SPECIALIST MEETING ON

PUMP PERFORMANCE
AND RELIABILITY

GRS Headquarters, Cologne, Germany
26th - 28th November, 1990

MEETING PROCEEDINGS
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COMMITTEE ON THE SAFETY OF
NUCLEAR INSTALLATIONS

The Committee on the Safety of Nuclear Installations (CSNI) of the OECD Nuclear Energy Agency (NEA), is an international committee made up of senior scientists and engineers. It was set up in 1973 to develop and coordinate the activities of the Nuclear Energy Agency concerning the technical aspects of the design, construction and operation of nuclear installations insofar as they affect the safety of such installations. The Committee's purpose is to foster international cooperation in nuclear safety among the OECD Member Countries.

The CSNI constitutes a forum for the exchange of technical information and for collaboration between organizations which can contribute, from their respective backgrounds in research, development, engineering or regulation, to these activities and to the definition of its programme of work. It also reviews the state of knowledge on selected topics of nuclear safety technology and safety assessment, including operating experience. It initiates and conducts programmes identified by these reviews and assessments in order to overcome discrepancies, develop improvements and reach international consensus on technical issues of common interest. It promotes the coordination of work in different Member Countries including the establishment of cooperative research projects and international standard problems, and assists in the feedback of the results to participating organizations. Full use is also made of traditional methods of cooperation, such as information exchanges, establishment of working groups, and organization of conferences and specialist meetings.

The greater part of the CSNI's current programme of work is concerned with safety technology of water reactors. The principal areas covered are operating experience and the human factor, reactor coolant system behaviour, various aspects of reactor component integrity, the phenomenology of radioactive releases in reactor accidents and their confinement, containment performance, risk assessment, and severe accidents. The Committee also studies the safety of the nuclear fuel cycle, conducts periodic surveys of reactor safety research programmes and operates an international mechanism for exchanging reports on safety related nuclear power plant incidents.

In implementing its programme, the CSNI establishes cooperative mechanisms with NEA's Committee on Nuclear Regulatory Activities (CNRA), responsible for the activities of the Agency concerning the regulation, licensing and inspection of nuclear installations with regards to safety. It also cooperates with NEA's Committee on Radiation Protection and Public Health and NEA's Radioactive Waste Management Committee on matters of common interest.
FOREWORD

(Prepared by the NEA Secretariat)

Since pumps play a major role in the operation of nuclear power plants, and since they constitute the hearts of process and safety-related systems, it is desirable from the economic and particularly from the safety standpoints to exchange views and experience, at the international level, on these vital components. For this reason, the Specialist Meeting on "Pump Performance and Reliability" was organized by the Committee on the Safety of Nuclear Installations of the OECD Nuclear Energy Agency.

This Meeting, which was held in Cologne, Germany, in November 1990, was hosted by Gesellschaft für Reaktorsicherheit (GRS) and addressed the following pump-related aspects:

a) requirements of sustained high reliability and availability;
b) design versus applications;
c) performance (operating experience), routine testing and maintenance;
d) effects of continued simultaneous exposure to high radiation fields and high temperature fluids;
e) materials;
f) initial qualification tests (e.g. seismic, environmental,...);
g) advances/improvements (materials used, design, manufacturing processes); and
h) regulatory requirements related to pump performance.

Forty-five participants from ten countries and two international organizations attended the Meeting. Pump manufacturers as well as plant designers, vendors, operators and regulators were represented. Twenty-two papers were presented in five sessions, which led to the final panel discussions on "Pumps in Nuclear Power Plants: Safety Aspects, Reliability and Developments".

These proceedings consist of a compilation of the papers presented at the meeting, the Chairmen's closing remarks and the panel discussions. Not included in these proceedings is the paper entitled "Incidents attributed to pump problems", by G. Ishack; this paper will be issued at a later date as an Addendum. It should finally be noted that some linguistic editing was effected, at the Secretariat, to the panel discussions' transcript to enable publication.
SESSION #1: "OPENING SESSION"

Chairman: K. Kotthoff

* Welcoming Address, K. Kotthoff, GRS, Germany

* Opening Remarks, J. Reig, CSN, Spain

* Invited Paper: "Pump damage due to low flow cavitation", Chi-Chiu Hsu, NRC, United States (presented by J. Rosenthal, NRC)

* Invited Paper: "Nuclear pump design from the past to the future", R. Martin, EDF-SPT, France

WELCOMING ADDRESS
Dr. K. Kotthoff, GRS, Germany

On behalf of the Director of GRS, who regrets not being able to be with you this morning, I am very pleased to welcome all participants in Cologne at GRS headquarters to our CSNI Specialist Meeting on Pump Performance and Reliability. This Specialist Meeting has been organized by the CSNI Principal Working Group No. 1 dealing with "Operating Experiences and Human Factors". I am very glad that we have with us today Mr. Reig from Spain, the Chairman of the Principal Working Group No. 1, who will welcome you on behalf of the Nuclear Energy Agency of the OECD. Furthermore I would like to welcome the members of the Principal Working Group No. 1 and last but not least Mr. Ishack from the Nuclear Energy Agency of the OECD who carried a large burden in organizing this meeting.

Our meeting today is one of a series of CSNI Specialist Meetings promoted by the Principal Working Group No. 1 under the auspices of the CSNI. CSNI Specialist Meetings are intended to provide a forum to specialists of the OECD member countries for the in depth discussions of specific relevant technical topics.

By tradition, Specialist Meetings are hosted by member countries of the NEA. I would like to emphasize that it is an honour for the FRG and especially the GRS to host the CSNI Specialist Meeting on Pump Performance and Reliability.

The meeting has been organized by a Programme Committee which included nominated members of the CSNI Principal Working Group No. 1 from Belgium, FRG, Japan, Spain and the United States. The Programme Committee met in September in Paris to agree on the programme and the practical arrangements for the meeting, and to select the papers.

I hope that this meeting provides an in-depth exchange of knowledge and experience with respect to pump performance and reliability in nuclear power plants. I hope that all participants will benefit from the papers presented and from the discussions. Have a very pleasant stay in Cologne, and hopefully you will have some time, good weather permitting, to have a look around in this interesting city with a history of more than 2000 years.
OPENING REMARKS

Javier Reig
PWG1 Chairman

I am very pleased to be here and to add my words of welcome to you, at the beginning of the CSNI Specialist Meeting on Pump Performance and Reliability which the OECD Nuclear Energy Agency is sponsoring.

Pumps are not a safety concern but an availability problem. This statement was commonly accepted more than 10 years ago and still may be accepted from time to time. Then, why is Principal Working Group No. 1 (PWG1), which deals with safety concerns derived from operating experience, and is composed mainly of regulatory authorities organizing a meeting on Pump performance?

The reasons could be traced back even before the Group was created. Already in TMI the operation of main coolant pumps played a significant role in the event evolution. Later on numerous problems related to pump performance have been reported. They appeared in different systems: main coolant and recirculation pumps, auxiliary and main feedwater pumps, ECCS pumps, RHR pumps, etc. They happened in all countries, across all reactor designs (PWR, BWR, CANDU) and are not peculiar to specific pump designs.

In recognition of this common problem, the PWG1 representatives of France and Germany proposed back in 1986 to hold a specific discussion on this topic within PWG1. Finally the group approved that a specialist meeting on this subject be convened; GRS, Germany graciously offered to host such a meeting.

The problems encountered varied from mechanical ones to those related to operation and design, and most of them have a clear impact on safety.

Most of these deficiencies came to light through events which occurred during the operation of nuclear power plants, or through routine testing and maintenance results. For this and many other reasons it is essential that we systematically collect and exchange all reports which describe these incidents, and that we determine the root causes of those malfunctions. You will be aware that the Nuclear Energy Agency has operated an Incident Reporting System since 1980, and we are now increasingly devoting resources to collectively analysing these events. We are certain that today no incident with safety significance, in the some 300 power reactors which operate in the OECD area, escapes our system; this provides also a sound basis for your investigations.

This meeting will deal with pump performance in many different aspects: pump design, pump testing and maintenance, pump operational characteristics, specific applications, research activities, operating experience and future designs.

I hope that your exchanges during the coming week will broaden your knowledge and increase your ability to improve pump performance and reliability which- as I mentioned earlier-are so important for nuclear safety.

Additionally you will have a chance to visit an experienced pump manufacturer and a nuclear power plant. This, I expect, will be a magnificent complement to your discussions during the meeting.
Safe nuclear power can only be assured by responsible nuclear operators who are supervised and controlled by responsible governments. Both of these parties rely on the competence and quality of the work performed by designers and manufacturers. It is vital, for the success of our industry, that all parties co-operate to attain this goal. I am therefore very happy that we have specialists here from all these fields, which will allow us to broaden our discussions by adding different perspectives.

This is no easy task, as we are aware, given the complexity of nuclear power plants and their safety systems, and the diversity of designs and operating practices, both among Member countries and in many cases also within countries.

In closing, Ladies and Gentlemen, I would not want to leave the floor without extending my thanks to the German government for having invited us to hold our discussions here in Cologne and to GRS and the other organizations and individuals who have contributed to the preparation of our meeting.

This is not the first time that CSNI has sponsored a meeting in this beautiful country and I know from past experience that we can expect a perfect organisation and gracious hospitality.

There now only remains for me to wish you a successful and informative meeting, and a pleasant stay in the lovely city of Cologne.
PUMP DAMAGE DUE TO LOW FLOW CAVITATION

Prepared by: Chi-Chiu Hsu

Presented by: Jack E. Rosenthal

U.S. Nuclear Regulatory Commission
Washington, D.C. 20555

ABSTRACT

In reviewing operational experience at U.S. nuclear power plants, cavitation damage of centrifugal pumps due to operation at flows lower than the design flow was identified. An evaluation of the problem found that the damage was due to pump impeller suction recirculation which occurs at these low flows. Such damage induces slow deterioration of pump internals, which, during the early stages of cavitation, do no affect pump operation. Plant routine surveillance tests for pumps may not be capable of detecting these early degradations. Hence, this type of damage is not easily detectable. There is the potential that recirculation cavitation on the pump impeller could go undetected until total failure of the pump occurs. These cavitation indication on pump internals cannot be observed without disassembling the pump.
I. INTRODUCTION

This study was initiated based on a report of an event at Susquehanna 1 involving severe erosion damage of pump internals. The erosion of pump internals was caused by recirculation cavitation. Recirculation within centrifugal pumps is flow reversal at the inlet or discharge tips of the impeller when pumps are running in off-design regimes. Flow reversal causes vortex action near the impeller vanes, inducing pressure surge and pulsation which cause rapid deterioration by cavitation of impeller metal in the inlet or outlet region even when adequate net positive suction head (NPSH) is provided (Ref. 1).

Recirculation has been one of the most persistent and puzzling problems encountered in the operation of centrifugal pumps in recent years. Although serious failures had not been reported previously, direct evidence of low flow induced failure of pumps in nuclear plants and experience gained in both the laboratory and the field during the past decade, had shown that hydraulic instabilities and imbalance can occur in a pump running significantly below the design flow. Furthermore, it has been proven through analysis and tests that the effects of recirculation can be very damaging not only to the pump's operation, but also to the life of the impeller and casing.

II. DISCUSSION

Licensee Event Report (LER) 86-021-00, dated May 24, 1986, for Susquehanna 1 describes an event involving pump damage due to erosion caused by recirculation cavitation. The pump damage was discovered in the emergency service water (ESW) and residual heat removal service water (RHRSW) systems. The event began on May 22, 1986, while the plant was operating at full power when an overcurrent alarm for the "C" ESW pump was received in the control room. Investigation revealed the pump motor to be running at low amperage, with the pump discharge check valve closed. The pump was declared inoperable and the plant entered a limiting condition for operation (LCO). Subsequent disassembly of the pump revealed that the bottom portion of the pump suction bell had separated from the pump body and had fallen into the pump pit. Inspection of the damaged parts revealed that the suction bell had been penetrated around its entire circumference by cavitation.

On May 24th, an inspection of the "A" pump by a diver revealed similar but less severe damage to the pump suction bell. The "A" pump was also declared inoperable.

Since the condition of the remaining "B" and "D" pumps was not known, they were also declared inoperable and a controlled shutdown of both units was begun. Although the "A", "B" and "D" pumps were declared inoperable, they were functional and continued to perform at or near their design capability during and following the unit shutdowns.

A subsequent inspection of the "B" and "D" pumps revealed cavitation damage similar to the "A" and "C" pumps. The suction bell wall was eroded considerably but had not been penetrated. The impellers showed signs of erosion on the high pressure side of their vanes at
the suction end. However, the vanes had not been penetrated and had retained their original shape. The "B" and "D" pumps were declared operable on May 28th. Due to a lack of spare parts, temporary repairs were made to the "C" pump and the "A" ESW loop was restored to a functional status on the same day.

The ESW pumps normally operate at about 60 percent or less of their design flow of approximately 6000 gpm per one pump. When the loop supplying cooling water to the diesels comes from two operating pumps, each pump delivers approximately 3500-3900 gpm. The other loop that does not serve the diesels (usually the B loop) is normally run with only one pump at approximately 1000-1500 gpm. Based on the inspection, the licensee concluded that impeller suction recirculation cavitation was the major contributing factor to the ESW pump failure. The type of cavitation occurred when the pumps ran at flow significantly lower than its design flow -- flow less than 60 percent.

Following receipt of the required spare parts, the "A" and "C" ESW pumps were repaired, retested and declared operable on June 6, 1986. The "B" and "D" ESW pumps were also repaired and retested on June 10th. Repair of all four ESW pumps was accomplished by the replacement of all suction bells and impellers. The replacement impeller is NiAl-Bronze which has a higher resistance to cavitation damage than the original impellers of carbon steel. The replacement suction bells are made of the original carbon steel material. Stainless steel liners were installed on the suction bells of the "A" and "C" pumps. Same liners will also be installed in the "B" and "D" pumps.

Due to their similar design, the Residual Heat Removal Service Water (RHRSW) pumps were also inspected. These pumps are two stage vertical circulator pumps, Byron Jackson type 28KXL. Cavitation damage was found on the impeller liners on the Unit 1 pumps "A" and "B", and the liners were replaced. Cavitation damage to the Unit 2 pump "A" was minimal and impeller liner replacement was not warranted at that time. The Unit 2 pump "B" was not inspected until the next refueling outage.

Although degraded, the RHRSW pumps were capable of performing their design functions. The "A" and "B" pumps of Unit 1 have been able to pump 9000 gpm per pump to their respective heat exchangers (design flow) during subsequent tests. Subsequent inspections found that the impeller liner degrades before any significant damage can be seen on the impeller and the liner damage did not seem to cause a noticeable drop-off in pump performance as evidenced by the flow data. There were no indications that the cavitation was attributed to flow vortexing or inadequate NPSH. The cavitation was a result of flow recirculation which was caused by operating the pumps at low flow rates. These RHRSW pumps had operated at least 50 percent of design flow most of the time. The licensee indicated that the cavitation damage can be avoided by operating the pump above 50 percent of design flow; specifically 75-100 percent of design flow is desirable. The RHRSW system design and method of operation would be reviewed to determine what changes could be possible to avoid recirculation cavitation.
The damage to the ESW and RHRSW pumps was determined to have been caused by impeller suction recirculation cavitation. The cavitation is caused by operating the pumps at flows which are significantly below the design flow. The cavitation erodes the suction bell wall. The impeller was also eroded but at a slower rate. Once the suction bell wall is penetrated, erosion of both the suction bell wall and the impeller is accelerated as water is drawn through the suction bell penetrations.

This event suggests a common cause failure mode for the pumps with low flow operation modes. The cited damage, which caused the pumps to be inoperable, included eroded impellers and suction bells.

Operation at lower flow creates mismatches of flow angles within the pump and causes water to recirculate back towards the suction. The recirculating currents cause local pressure zones which are below the vapor pressure of the water. This causes vapor bubbles to form which collapse when a high pressure zone is reached, eroding the local material. The erosion begins on the high pressure side of each impeller range at its suction end. Prolonged operation of a pump at low flow can result in cavitation damage on impellers.

When running a pump at a low flow, flow recirculation can occur at the discharge regions as well. This is called discharge recirculation. Discharge recirculation also creates surges and local deterioration by cavitation at the impeller tips. Recirculation in the suction and discharge regions does not necessarily occur at the same flow rate. The recirculation cavitation is not related to inadequate pump NPSH. Since inadequate NPSH would also cause pump cavitation, the similarity between patterns of cavitation damage from recirculation and from inadequate NPSH may often lead to an erroneous conclusion as to the cause of the damage. However, the mechanisms that cause the damage are entirely different. The cavitation damage from recirculation proceeds from the high pressure side of the inlet edge of the vane through the metal towards the low pressure side, while the damage from inadequate NPSH starts in the opposite direction, from the low pressure side of the vane and proceeding through the metal toward the high pressure side.

Recirculation characteristics are dependent on the design of the impeller. It is inherent in the dynamics of the pressure field that every impeller design must begin to recirculate at some point of low flow. Recirculation becomes progressively pronounced as a pump is operated further from the design flow. The percentage of design flow rate at which recirculation will begin is dependent on many factors. The most critical factors which influence low flow pump performance and minimum continuous flow are: power intensity (pump size), suction specific speed, and specific speed. Although pump manufacturers (Ref. 3 and 4) have recently developed guidelines for establishing low flow limits on pump operation, studies on low flow aspects are still continuing. Most of the guidelines which have been published (Ref. 5) stipulating recommended minimum flow for pumps, present minimum flow rates as a function of suction specific speed; the lower the design suction specific speed of a pump, the lower the flow rate the pump can operate without recirculation problems.
Low flow operations are generally required for the standby systems when performing in-service surveillance testing of pumps through the miniflow bypass line, and for systems with a wide range of flows when operating pumps in low flow modes. These low flow operations are general design configurations for emergency core cooling system testing in nuclear power plants. In most plants, the pump bypass lines were sized only on the basis of limiting the temperature rise of the pump when operated in the testing or minimum flow mode. Typically, this temperature rise based minimum flow is in the order of 10 percent of the best efficiency point (BEP) flow. In response to the concern by pump manufacturers that testing pumps at low flow on the order of 10 percent of BEP flow, may lead to premature failure of pump components as a result of higher vibration during low flow testing, EPRI (Ref. 2) conducted a study on surveillance testing of standby pumps in operating nuclear power plants in 1985 to determine whether test-related failures were caused by some aspects of the tests. Although this study neither provides conclusive evidence against nor vindicates the use of low flow testing practices, it does support the expectation that low flow test operation will lead to degradation and premature failure of pump internals and concludes that prolonged operation of pumps at very low flow (in the range of 10 percent BEP flow) can cause high vibration from hydraulic instability from flow recirculation in the pumps.

Lately, it is increasingly recognized by pump designers and manufacturers that factors other than temperature rise, such as energy level, suction specific speed, developed head per pump stage, and impeller design details, should play a role in establishing the reference value for flow rates during in-service tests. Several pump manufacturers are now recommending that standby pumps be tested at a flow no less than 25 percent of BEP flow(s).

Five additional events involving either pump failure or potential for pump degradation from low flow operations were found in a search of operational experience data base files. These five events occurred at Vermont Yankee, H.B. Robinson 2, Turkey Point 3, a foreign reactor and at Haddam Neck. The pump damage at Vermont Yankee was attributable to insufficient miniflow recirculation line capacity. The potential for pump degradation identified at H.B. Robinson and Turkey Point were also associated with the inadequacy in the original design of miniflow recirculation line and that of the last two plants was caused by prolonged operation of pumps in low flow modes.

The effects of recirculation resulting from low flow operation manifest themselves not only in material degradation — cavitation, but also in the form of pressure pulsations and vibrations. Hydraulic pressure pulsation and pump vibration are also significant contributors to deterioration of pump components because of the high amplitude dynamic forces that they produce. The pump damage at Vermont Yankee was a result of cavitation, while those at the foreign reactor and at Haddam Neck were attributed to pressure pulsation and pump vibration. H.B. Robinson and Turkey Point identified the existence of the potential for pump degradation and failure due to the insufficient flow rate designed for the recirculation lines.

Low flow testing can be a possible source of impeller cavitation damage. The resulting hydraulic recirculation present inside the pump at low flow can create cavitation damage.
Vermont Yankee was notified by the pump vendor, Bingham/Willamette, on November 13, 1986 that the minimum flow rates for the RHR pumps should be made higher than previously indicated to Vermont Yankee. Similar notifications for increase of mini-flow rates have also been sent to four other plants: Cooper, Pilgrim, Browns Ferry and Peach Bottom.

The pump vendor indicated that the minimum flow requirements established for the RHR pumps at Vermont Yankee since plant startup may not be adequate for all pump operating modes. Specifically, the pump vendor recommended that the value for continuous minimum flow for the pumps should be 2700 gpm, or about 38 percent of the pump design flow of 7200 gpm, and the value for intermittent operation (less than 2 hours of operation within a 24-hour period) should be 2075 gpm. The RHR pump has a flow orifice in the recirculation line designed to limit flow to about 350 gpm. This orifice sizing was based solely on the pump flow required to limit the temperature rise of pump when operated in the minimum flow mode. The pump vendor apparently has since determined that additional factors must be considered in determining the minimum flow requirements, including pump inlet and outlet circulation flow patterns that will occur at lower flow modes. Recirculation flow patterns can occur and result in component damage event if there is sufficient available NPSH. In a similar letter dated November 21, 1986, the vendor also recommended a continuous minimum flow of 1500 gpm (versus about 350 gpm) for the Bingham 12 x 16 x 4 1/2 CVDS core spray pumps. A minimum flow of 1350 gpm was recommended for intermittent operation.

As a result of this advice, Vermont Yankee initiated a review which determined that this matter does not pose a substantial safety hazard for the plant because of plant specific application of the pumps. The length of time that the RHR and core spray pumps would be required to operate in the minimum flow mode over the 40-year design life would be far less than the maximum times at minimum flow recommended by the pump vendor. Bingham/Willamette defines "intermittent operation" as less than 2 hours of operation in a 24-hour period over the 40-year design life. This translates to a value of up to a total 29,200 hours of operation. Vermont Yankee has no significant accumulated time in the minimum flow operating mode to date (other than successful pre-operational testing). Monthly surveillance testing does not utilize the minimum flow path for more than 15 to 30 seconds per month and therefore is considered to be negligible with respect to the 2075 hours allowable. In the event of a small break LOCA, RHR and Core Spray would be required to operate in the minimum flow mode for a maximum of 4 or 5 hours. Vermont Yankee's evaluation estimated a total of 5 to 10 hours of operation in the minimum flow mode for the life of the plant. As can be seen, these operating durations are far below the 29,200 hours of operation that Bingham/Willamette considers to be in the "intermittent" operating range, and that sufficient operating times available in the minimum flow mode for either the RHR or the Core Spray pumps would not be attained for recirculation cavitation failures to develop.

However, this matter could create a substantial safety hazard at another nuclear facility, depending on the application of this manufacturer's pumps and the length of time that a pump would be required to operate in the minimum flow mode. The licensee determined the matter
was potentially reportable under 10 CFR 21 and notified the NRC of the potential design
defect on March 20, 1987. As an added precaution, the licensee would incorporate a caution
statement into the appropriate procedure to alert the operators of the need to minimize time in
the minimum flow mode.

NRC Information Notice 86-39, "Failure of RHR Pump Motors and Pump Internals,"
discussed the failure of a RHR pump at Peach Bottom Unit 3 due to impeller wear-ring
failure as a result of intergranular stress corrosion cracking (IGSCC). After receiving this
notification, the Vermont Yankee licensee performed an inspection of pump internals on the
RHR pumps during the period from April to May 1987, since they are the same model and of
the same manufacturer (Bingham-Willamette single-stage vertical mounted, model CIVIC,
centrifugal pumps). In addition to the findings of through wall impeller cracks on two of the
RHR pumps and wear-ring cracks on one RHR pump, the inspection coincidently found
evidence of erosion resulting from suction recirculation on all four of the RHR pumps. The
licensee also indicated that the wear-ring with cracks in the "A" RHR pump was a stationary
ring, cast from a material not susceptible to IGSCC cracking. However, the cause of
impeller and wear-ring cracks was not determined, pending the completion of the licensee's
destructive examination. Based on the recirculation flow erosion effect observed on the
suction side of several impellers, the licensee would re-evaluate the previous engineering
conclusions regarding the adequacy of the RHR minimum flow lines in light of the results of
the pump internal inspection.

The impeller and wear-ring cracks may also appear related to excessive flow turbulence as a
result of suction recirculation. The flow reversal at the impeller eye under the condition of
suction recirculation creates a vortex action which induces pressure surges and pulsations,
causing rapid deterioration by erosion of impeller metal. The pressure surge and pulsation
can produce dynamic forces which may rise high enough to cause vibration and add undue
stress on the wear-ring, resulting in the cracking failure.

The event at H.B. Robinson was reported in LER 87-026. On October 30, 1987, the
potential for degraded recirculation flow for the RHR pumps was identified by the licensee
during a review in response to the Westinghouse (NSSS Vendor) concerns of inadequate
miniflow design for the RHR pumps. The NSSS designer had identified two concerns
recently involving the potential for dead heading of one of two RHR pumps in systems that
have a common miniflow recirculation line serving both pumps, and, the potential for
insufficient miniflow recirculation line capacity event for single pump operation. Based on
the licensee evaluation, the potential for insufficient miniflow recirculation for the RHR pump
is due to inadequacies in the original design based on today's criteria. The miniflow
recirculation line was designed on the assumption that the two pumps have equal flow/head
curves and that each would achieve a flow of about 250 gpm while both were operated
simultaneously. Recently, however, the pump vendor recommended a minimum flow of
5000 gpm for each pump to prevent excessive vibration and pump binding caused by heat up
of the recirculated fluid.
A similar case at Turkey Point 3 was described in LER 88-030. On October 27, 1987, the licensee discovered that the existing minimum recirculation design configuration was potentially inadequate. The miniflow recirculation line, which was shared by the two RHR pumps, was insufficient and had potential for dead heading one of the two pumps in the RHR system. The licensee’s corrective action was to install independent minimum recirculation lines for each pump. The modified recirculation system would allow operation of both pumps for at least 30 minutes without affecting pump operability.

In response to the concerns of inadequate miniflow design for RHR pumps, NRR issued an Information Notice on November 17, 1987 (NRC IN 87-059, Potential RHR Pump Loss) which indicates that the problem may be generic to all water-cooled reactor designs, regardless of the pump application or the NSSS manufacturer. This is based on the belief that miniflow lines have traditionally been designed for only 5 percent to 15 percent of pump design flow, while some pump manufacturers are advising that their pumps should have minimum flow capacities of 25 percent to over 50 percent of best efficiency flow for extended operation.

Another event occurred at a foreign reactor. During a periodic inspection, maintenance personnel heard a loud noise from an RHR pump. The pump was immediately stopped. Upon disassembly, the shaft was found broken, and slight contact marks were found on the wear rings. The markings on the broken area of the shaft indicate that failure was due to a low cycle fatigue fracture. Subsequent evaluations concluded that the pressure distribution around the impeller becomes uneven during low flow operation increasing the radial thrust, and the bending stress becomes approximately three times that at rated flow. The pumps had run approximately 5963 accumulated operating hours in low flow operation when the failure occurred. In addition to the main shaft replacement, the licensee modified its long term operation management of low flow operation. Previously, both modes of operation of the RHR pump, the clean-up and the cooling modes, were performed by one pump for each purpose. Modified operation is to operate one pump for both purposes to minimize accumulated low flow operations of one pump.

The event reported in LER 88-003 at Haddam Neck occurred on February 4, 1988. With the plant in mode 6 and the reactor core off loaded, the electric driven fire pump was declared inoperable due to a high amperage condition noted after a manual start during a routine surveillance. The normal indication of 200 amps increase to 340-360 amps. The indication increased to 1000 amps during the following manual restart. The cause of the inoperability was physical damage to the stuffing box brass bushing located in the upper shaft area of the electric driven fire pump. This caused the bushing to shear, resulting in a locked rotor condition. Based on the licensee’s evaluation and their discussion with the manufacturer, it was concluded that prolonged low flow operation of the pump may have caused the problem. Operation of the pump at or near shut off head had occurred during the Containment Integrated Leak Rate Test (CILRT) in which the fire pump operates at low flow mode only to provide cooling water to the air compressors.
These events illustrate that recirculation cavitations were caused by operating pumps at flows significantly below the design flow rates, either at low flow modes or at miniflow testing through a bypass line. The damage was a result of slow deterioration accumulated over a long period of time during which the pumps were still functional and remained operable at early degradation. Cavitation indication on the pumps' internals can only be observed by disassembly of the pumps. The routine surveillance tests of pumps provided in the plant inservice test programs may not be capable of detecting early impeller degradation. In addition, since the pumps operated in the specified operating range, the plants were not aware of the problem until the occurrences of pump failure. There is the potential that recirculation cavitation on a pump impeller could go undetected until total failure of the pump occurs. Such a failure could prevent the associated system from performing its safety function.

In the ongoing NRC/RES effort on aging and service wear of the auxiliary feedwater (AFW) pumps for PWR nuclear plants, testing of AFW pumps at flow less than 25 percent of BEP has been identified as a source of hydraulic instability and unbalance which will accelerate component wear and lead to premature failure of pump internals. The RES study found that the miniflow line for AFW pumps was typically established only to prevent pump overheating and thus is normally 10-15 percent of BEP flow. Although it is suggested that the surveillance testing of AFW pumps be completed at increased flow, in most cases, it is not practical to test the pump at higher flows without either modifying the existing miniflow circuit to increase its flow capacity or to perform testing while the plant is shutdown (In response to these concerns, one of the contractors of the RES study has developed improved auxiliary feedwater pump testing guidelines). The preliminary findings and conclusions of the RES studies correlate well with the operating experience evaluated in this report.

III. FINDINGS AND CONCLUSIONS

Based on the preceding discussion and related follow-up activities conducted for the study, the following findings and conclusions are provided:

1. Operation of centrifugal pumps at low flow conditions for extended periods of time can cause cavitation damage in spite of available NPSH. A centrifugal pump is designed for best performance at a specific combination of capacity, head, and speed, that is, the best efficiency point (BEP). At the design or BEP flow rate, the fluid motion is compatible with the physical contours of the hydraulic passage and is therefore well-behaved. Once deviating from design flow, the operation starts to create mismatches of flow angles within the passage and diverts part of flow to recirculate within the pump at certain low flow rates. The circulating currents cause local pressure zones which are below the vapor pressure of the water. This causes vapor bubbles to form which collapse when a high pressure zone is reached, leading to the erosion of the local material. Such flow recirculations can occur at the impeller eye and exit, as well as outside the impeller shroud and hub.
2. Low flow operations are generally required for the standby systems when performing inservice surveillance tests of pumps by restricting discharge flow through the mini-flow bypass line, and for systems designed for wide range of flows when operating the pumps in the low flow regime. Many of the emergency core cooling systems in most operating plants are designed to operate with wide range of flows and use a mini-flow bypass line for inservice testing of pumps during the standby mode.

3. Recirculation cavitation has caused damage to the pumps of the ESW and RHRSW systems at Susquehanna 1 and the pumps of the RHR system at Vermont Yankee. The recirculation cavitations were caused by operating pumps at flows significantly less than their design flow rates, and the damages were the result of prolonged operations of these pumps at the low flows. The cavitation damage of the ESW pumps at the Susquehanna plant was very severe. The impeller vanes were eroded through the wall, and suction bells were penetrated around most of the circumference. One of the suction bells had separated from the pump body and fell into the pump pit. The damages to the RHRSW pumps were less severe. The ESW pumps normally operate at about 50 percent or less of the design flow and had run approximately 18,000 hours when the failure occurred. The 18,000 hours of operation is only a small fraction of the design life. The RHRSW pumps had operated at less than 50 percent of design flow for most of the time and had run approximately 9,000 hours. The recirculation cavitation of the RHR pumps at Vermont Yankee could be associated with low flow operation during the monthly surveillance tests. The RHR pump mini-flow bypass line is a single line sized to bypass about 5 percent of the design flow. The accumulated time in operation at this low flow is not known.

4. The effects of recirculation manifest themselves not only in material degradation--cavitation, but also in the form of pressure pulsations and vibrations. Hydraulic pressure pulsations and pump vibrations are also significant contributors to deterioration of pump components because of the high amplitude dynamic forces that they produce. Excessive forces on the impeller and pump vibration have caused damage to the RHR pump at a foreign reactor and the fire pump at Haddam Neck. The licensees for H.B. Robinson 2 and Turkey Point 3 have identified the existence of the potential for pump failure due to insufficient flow rate designed for the recirculation lines of their RHR pumps.

5. It is inherent in the dynamics of the pressure field that every impeller design must recirculate at some point of flow -- it cannot be avoided. The flow rate at which recirculation occurs is dependent of the design of the impeller. Although pump manufacturers have recently developed guidelines for establishing low flow limits on pump operation, the pump bypass lines, in most operating plants, were sized solely on the basis of limiting the temperature rise of the pump when operated in the testing mode or minimum flow mode. Generally, the flows are in the order of 10 percent of design flow.
6. In response to a concern by pump manufacturers that testing pumps at low flows on the order of 10 percent BEP flow, may lead to premature failure of pump components, EPRI conducted a study in 1985 on surveillance testing of standby pumps in operating nuclear power plants. The result of this study does support the expectation that low flow test operation will lead to degradation and premature failure of pump internals and concludes that prolonged operation of pumps at very low flow (in the range of 10 percent to BEP flow) can cause high vibration which is a hydraulic instability.

7. Several pump manufacturers have recently recommended the standby pump be tested at flow no less than 25 percent of BEP flow(s). One pump vendor, Bingham/Willamette, has informed Vermont Yankee of inadequacy in the minimum flows designed for the RHR pumps and the core spray pumps. The minimum flow requirements were established for the pumps at plant startup. The value for minimum flows for the pumps, according to the vendor, should be increased to about 38 percent of the pump design flows. Similar notifications for increase of minimum flow rate for the RHR pumps also have been sent to four other plants: Cooper, Pilgrim, Browns Ferry and Peach Bottom.

8. In the ongoing RES program on aging and service wear of the auxiliary feedwater (AFW) pumps for PWR plants, surveillance testing of AFW pumps at flows below 25 percent of BEP has been identified as a source of hydraulic instability and unbalance which will accelerate component wear and lead to premature failure of pumps.

9. It appears that degradation caused by recirculation cavitation due to low-flow operation will require a great length of time to cause catastrophic failure of a pump. Hence these types of damage are not easily detectable. In addition, the degradation will also be difficult to detect in the basis of measurements taken during inservice surveillance testing. This follows from the fact that, in most plants, the bypass flow test provides neither the proper operating range of low nor sufficient running time to comprehensively trend and predict degrading condition. Recirculation cavitation damage of the pump impeller could go undetected until total failure of the pump occurs.

10. Although the mechanism that causes cavitation damage from recirculation is entirely different from that of inadequate NPSH, the similarity between patterns of the cavitation damage from both may often lead to an erroneous conclusion as to the cause of the damage. This may be one of the reasons that the concern of recirculation cavitation has not been widely recognized.
IV. SUGGESTIONS

1. The effects of pump recirculation can cause not only operational problems, such as vibration, but also lead to damage and loss of life of the impeller and casing. Major degradation of pump impellers and casings have occurred to centrifugal pumps running continuously at low flows. Many of these problems can be avoided by specifying and designing pumps for lower suction specific speeds and limiting the range of operation to capacities above the point of recirculation. For example, low flow operation of centrifugal pumps could be avoided in a system in which multiple pumps operate continuously at low flows by reducing the number of pumps running and thus increasing the flow through each pump, or by realigning the system to increase flow through the pumps. It is suggested that pumps specified for operation in a wide range of flows should be checked to determine whether any point in the operating range fall in the recirculation zone of the pumps.

2. The bypass flow rates for the pumps with bypass line should be reconsidered and acceptable values established to avoid operating pump in the recirculation zone. If the condition of running a pump in a recirculating zone cannot be avoided, a procedural control should be established to limit the length of operation in the bypass mode such that premature failure of the pump can be prevented.

3. For pumps having the potential for recirculation cavitation, (i.e., pumps not meeting items 1 and 2 above) appropriate inspection intervals should be established to facilitate early detection of recirculation damage of the pumps.

V. REFERENCES


NUCLEAR PUMP DESIGN
FROM THE PAST TO THE FUTURE

Roger MARTIN - Electricité de France
Service Etudes et Projets Thermiques et Nucléaires
VILLEURBANNE - FRANCE

SUMMARY

Pumps play an important part in the operation of nuclear power plants. Often they are the main piece of rotating machinery in the process. For this reason, they are extremely important, particularly for safety systems. We should also bear in mind that in a nuclear power plant:

- there are many pumps
- their sizes, duty conditions, performance and costs differ enormously,
- the cost of insufficient reliability or availability can be very high, especially for safety systems or systems which do not include back-up pumps.

This means that it is absolutely necessary to design pumps with high levels of reliability.

In the following paper, we will demonstrate the importance of specifications in order to obtain a healthy design of auxiliary nuclear pumps leading to high reliability. A subject like this is very ambitious and could take up hundreds of pages. This is not possible here. Accordingly, this paper will be an attempt to highlight some of the sensitive points concerning nuclear pumps. Then an overview, probably excessively swift, will demonstrate how it is always preferable to select "tolerant" or "forgiving" technologies. Finally, we will refer to the fact that over-sizing the equipment may be a detrimental process.

1. SURVEY OF PUMPS INSTALLED IN A PWR POWER PLANT

To get the inventory of the pumps installed in a power plant, computerized equipment lists are often used. With these, it is simply a matter of selecting the criterion "Pump" and asking for a print-out.

Indeed, the term "pump" applies to very different equipment:

- vacuum pumps,
- main pumps,
- auxiliary pumps,
- test pumps,
- lubrication pumps,
- proportioning pumps,
- exhaust pumps, etc.

In this manner, when the files are sorted, a highly diversified list of equipment is generally produced. Nevertheless, in spite of these difficulties, we will attempt to analyze below the number and kinds of pumps existing in a power plant.
1.1 HOW MANY PUMPS ARE INSTALLED IN A POWER PLANT?

Table 1 below gives a count of the pumps installed in a typical power plant with two 1300 MWe type plant units.

<table>
<thead>
<tr>
<th>Power Range</th>
<th>Number of pumps in each class</th>
</tr>
</thead>
<tbody>
<tr>
<td>less than 1 kW</td>
<td>320</td>
</tr>
<tr>
<td>1 to 3 kW</td>
<td>120</td>
</tr>
<tr>
<td>3 to 10 kW</td>
<td>92</td>
</tr>
<tr>
<td>10 to 30 kW</td>
<td>122</td>
</tr>
<tr>
<td>30 to 100 kW</td>
<td>58</td>
</tr>
<tr>
<td>100 to 300 kW</td>
<td>25</td>
</tr>
<tr>
<td>300 to 1000 kW</td>
<td>23</td>
</tr>
<tr>
<td>1000 to 3000 kW</td>
<td>14</td>
</tr>
<tr>
<td>more than 3000 kW</td>
<td>16</td>
</tr>
<tr>
<td>TOTAL</td>
<td>790</td>
</tr>
</tbody>
</table>

Although this count is probably incomplete, it already shows the great number of pumps encountered in a power plant. Some interesting statistical elements can be obtained from this information.

1.2 DIVERSITY OF PUMP TYPES:

Generally, in a nuclear power plant, there are only a few identical pumps.

There may be, for instance:
- 3 or 4 reactor coolant pumps,
- 2 feedwater pumps,
- 3 condensate extraction pumps

Even among waste transfer pumps, there are no series of pumps strictly identical to one another.

Each pump has to be adapted to the specifications imposed by the process.
If we classify pumps by type we obtain table 2 below:

<table>
<thead>
<tr>
<th>Localization</th>
<th>Total number of pumps</th>
<th>Number of types</th>
<th>Average no. of pumps of the same type</th>
</tr>
</thead>
<tbody>
<tr>
<td>- Site</td>
<td>138</td>
<td>76</td>
<td>1.8</td>
</tr>
<tr>
<td>- Pair of units</td>
<td>7</td>
<td>4</td>
<td>1.7</td>
</tr>
<tr>
<td>- Each Unit</td>
<td>323 x 2 = 646</td>
<td>123</td>
<td>5.3</td>
</tr>
<tr>
<td></td>
<td>790</td>
<td>203</td>
<td>3.9</td>
</tr>
</tbody>
</table>

### 1.3 DIVERSITY OF OPERATING MODES

Among the installed pumps, some operate continuously, others are on stand-by or in reserve. Therefore their operating modes differ enormously.

Pumps can be classified as shown in table 3 below:

<table>
<thead>
<tr>
<th>Operating mode</th>
<th>D typical (h)</th>
<th>H typical (h)</th>
<th>Examples of pumps concerned</th>
</tr>
</thead>
<tbody>
<tr>
<td>- on stand-by</td>
<td>1 to 10</td>
<td>&lt;150</td>
<td>safeguard pumps</td>
</tr>
<tr>
<td>- Intermittent</td>
<td>10 to 30</td>
<td>150&lt;H&lt;1500</td>
<td>start-up pumps (auxiliary feedwater pumps, RHR pump...)</td>
</tr>
<tr>
<td>- Alternating (1/2 or 2/3)</td>
<td>30 to 100</td>
<td>1500&lt;H&lt;5000</td>
<td>Condensate pumps</td>
</tr>
<tr>
<td>- Continuous</td>
<td>&gt;100</td>
<td>H&gt;5000</td>
<td>RCP, feedwater, Condenser-cooling water pumps</td>
</tr>
</tbody>
</table>

D: average operating time at each start-up (in hours).

H: accumulated operating time over one year.

NOTE: The "alternating" mode corresponds to cases where pumps are installed in reserve (system 3 x 50% or 2 x 100%). It is considered that regular changeover of pumps on duty is being carried out and that the pumps therefore have similar utilization times.

### 1.4 PUMP INVESTMENT COSTS

By including the driving machine (electric motor or turbine) in the cost of the pumps the following approximate comparisons can be obtained:

- primary circuit pumps ........................................... 57.0%
- auxiliary nuclear pumps ........................................ 3.0%
- other nuclear pumps ............................................. 6.5%
- conventional island main pumps ............................... 31.0%
- other pumps in conventional island .......................... 2.5%

TOTAL ............................................... 100%
Compared to the total investment cost of the power plant, the pumps represent approximately 5% of the cost. The reactor coolant pumps alone represent more than half of that cost.

Thus, a nuclear power plant includes a large variety of pumps that are of low relative costs.

2. **PUMP OPERATING EXPERIENCE**

Operating feedback is very useful in evaluating pump standards. In the following section, we will very briefly analyze the availability and reliability aspects of these pumps.

2.1 **UNAVAILABILITY OF PLANT UNITS DUE TO PUMPS**

Without going into fine detail on the operational statistics of all the French PWR nuclear power plants, let us attempt to estimate how important pumps are regarding plant unit availability.

Plant units can only become unavailable through pump failure:

- if the system is indispensable for safety or availability of the plant unit,
- if there is no installed back-up pump.

In practice, only main pumps (reactor coolant pumps and main feedwater pumps) can cause such unavailability.

The table below summarizes fortuitous unavailabilities and extended shut down due to primary pumps.

<table>
<thead>
<tr>
<th>Unit</th>
<th>Unavailability type</th>
<th>1986</th>
<th>1987</th>
<th>1988</th>
<th>1989</th>
</tr>
</thead>
<tbody>
<tr>
<td>900</td>
<td>- fortuitous</td>
<td>0.28</td>
<td>0.3</td>
<td>0.2</td>
<td>0.1</td>
</tr>
<tr>
<td></td>
<td>- extended outage</td>
<td>n-d</td>
<td>n-d</td>
<td>0.3</td>
<td>0.2</td>
</tr>
<tr>
<td>1300</td>
<td>- fortuitous</td>
<td>0</td>
<td>0.4</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>- extended outage</td>
<td>n-d</td>
<td>n-d</td>
<td>0.1</td>
<td>0</td>
</tr>
</tbody>
</table>

The following table summarizes fortuitous unavailabilities due to main feedwater pumps.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>900</td>
<td>0.04</td>
<td>0.16</td>
<td>0.1</td>
<td>0.1</td>
</tr>
<tr>
<td>1300</td>
<td>0.43</td>
<td>0.20</td>
<td>0.3</td>
<td>0.2</td>
</tr>
</tbody>
</table>
NOTE: 0.1% unavailability means that on average for each year a plant unit can not produce for 8760 h x 0.1% = 8.76 h. This corresponds to a production loss of 8.76 x 900 = 7884 MWh for a 900 MW plant unit, i.e. around 1200 kFF.

Overall, it will be observed that pumps cause relatively little unavailability.

2.2 PUMP RELIABILITY

To carry out probabilistic safety studies on the 1300 and 900 MW plant units, it was necessary to estimate the reliability of the various power plant components and therefore of the pumps. The studies were carried out using incident files.

The data bases used mainly concerned the 900 MW plant units during the early part of their life, i.e. in the period from 1978 to 1983 or approximately 80 reactor years. Table 6 below recapitulates the reliability data concerning the pumps:

<table>
<thead>
<tr>
<th>System</th>
<th>Pump Designation</th>
<th>LAMBDA (1/h)</th>
<th>Gamma (1/d)</th>
<th>Tau (h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AFW</td>
<td>Auxiliary feed water system</td>
<td>3.2E-4</td>
<td>2.5E-4</td>
<td>17</td>
</tr>
<tr>
<td>FWP</td>
<td>Main feed water system</td>
<td>8.6E-6</td>
<td>3.0E-4</td>
<td>42</td>
</tr>
<tr>
<td>CWS</td>
<td>Condenser-cooling-water</td>
<td>4.0E-6</td>
<td>4.0E-4</td>
<td>14</td>
</tr>
<tr>
<td>CSS</td>
<td>Containment spray system</td>
<td>2.5E-5</td>
<td>8.0E-4</td>
<td>23</td>
</tr>
<tr>
<td>RCP</td>
<td>Reactor coolant system</td>
<td>2.9E-6</td>
<td>3.0E-4</td>
<td>40</td>
</tr>
<tr>
<td>LHSI</td>
<td>Safety Inj. system, LP</td>
<td>2.5E-5</td>
<td>8.0E-4</td>
<td>23</td>
</tr>
<tr>
<td>HHSI</td>
<td>Safety Inj. system, MP</td>
<td>5.3E-5</td>
<td>1.1E-3</td>
<td>25</td>
</tr>
<tr>
<td>RHR</td>
<td>Residual heat removal system</td>
<td>3.5E-5</td>
<td>2.8E-4</td>
<td>48</td>
</tr>
<tr>
<td>CCS</td>
<td>Component cooling system</td>
<td>1.3E-5</td>
<td>3.5E-5</td>
<td>30</td>
</tr>
<tr>
<td>SEC</td>
<td>Essential service water system</td>
<td>5.8E-6</td>
<td>2.4E-5</td>
<td>23</td>
</tr>
</tbody>
</table>

With: Lambda: Failure rate during operation (1/ hour)

Gamma: Failure rate on prompting (1/ start-up)

Tau: Average repair time (in hours).

It should be observed that in general pump reliability is high (Lambda <5.10^-5/h) except for the AFW pumps. This point is worthy of explanation.

If we analyze the incidents affecting Auxiliary feedwater (AFW) pumps, we remark that a particularly high number (75%) are due to the same type of pump. Accordingly, in the same AFW pump inventory, for the 900 MW plant unit analyzed, there is one type of machine which is particularly fragile (the replacement of these pumps is under consideration).
Thus the following observations can be made:
- generally the pumps are reliable.
- conversely, the pumps may be relatively unreliable if they are badly designed. Accordingly, we come to the conclusion that machines with insufficient reliability may also lead to significant plant unit unavailabilities.

3. FEEDBACK AND DESIGN

We have considered a few particularities about the pumps used in nuclear power plants:
- high numbers,
- high diversity of types and operating modes,
- low relative costs.

Operating feedback demonstrates that even if their behaviour is generally good, pumps can lead to insufficient system reliability and/or availability levels.

We have not yet brought up the maintenance aspects, in particular the more and more strongly affirmed needs of predictive, simplified, economical and especially fault-free maintenance. These needs should be taken into account as of the design stage of nuclear power pump units.

In the following chapters, we will demonstrate how these aspects must and can be taken into consideration when pumps are selected or designed. So as not to overgeneralize, we will refer mainly to examples of auxiliary nuclear power pumps. The highly specific case of RCP pumps will not be brought up.

4. A FUNDAMENTAL STEP: DELIMITING THE OPERATING FIELD

It might appear trivial to point out that delimiting the operating conditions is an essential and preliminary stage to that of design.

In practice, many examples of incidents due to operation under unscheduled design conditions are encountered. The most typical cases correspond to operations:
- at overflow or underflow during longer periods than expected,
- with unexpected ambient temperature levels,
- with unexpected intake conditions or conditions which can only be tolerated for a short period of time,
- consecutive to maintenance operations not taken into account at the design stage (e.g. excessive lubrication of ball bearings, or lubrication at unsuitable frequencies).

4.1 FLOW RANGE

A few years ago, a pump was often specified for a single operating point and that meant for a single flow rate. It was very rare that the pump was actually used constantly at that flow rate. This situation was particularly tricky for feed water pumps during plant unit start-up or when the plant was on load follow-up.

In the case of safety injection pumps, the normal flow rate cannot be defined: it is evident that only a useful flow rate discharged to the primary circuit can be predicted, falling between:
- 0: when the primary pressure is greater than the closed valve head supplied by the pump, and
- \( Q_{\text{max}} \): the maximum flow rate limited by the flow restrictors when the primary pressure is very low.

In practice, we have to provide for a recirculating circuit for the minimum flow of \( Q_{\text{min}} \) as sketched out in Fig. 1.

Fig. 2 is a schematic representation of the operation of a safety injection pump. It is clear that when the primary pressure is greater than \( P_A \) no flow is discharged toward the primary circuits. Conversely, when the primary pressure becomes \( P_M < P_A \), the pump flows toward the primary circuit. The representative point \( M \) moves from \( A \) to \( B \) when the primary pressure drops.

### 4.1.1 CHOICE OF MINIMUM FLOW RATE VALUE

This choice is extremely important for pump reliability. Indeed, at a low flow rate, the pump sustains:

- considerable internal recirculation in the impellers. This circulation increases in proportion to how wide open the eye is, i.e. in proportion to how high the \( N_{ss} \) (1) of the impeller is.
- extreme overheating.
- high dynamic internal forces. Generally, it is found that these forces are 3 to 10 times greater than those which exist for the most efficient flow rate \( Q_{\text{BEP}} \).
- considerable axial and radial forces.

Figure 3 is an example of the minimum recommended flow rates for large pumps. This figure also demonstrates the influence of the form of the impellers due to parameter \( N_{sq} \), (2)

Our experience confirms this graph as indicated in the few examples below:

<table>
<thead>
<tr>
<th>Designation of the pump</th>
<th>Main characteristics</th>
<th>( Q_{\text{min}} ) in %( Q_{\text{BEP}} )</th>
<th>( Q_{\text{max}} ) in %( Q_{\text{BEP}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main feed water pumps</td>
<td>( N_{ss} ) 30, ( N_{sq} ) 150</td>
<td>30</td>
<td>115</td>
</tr>
<tr>
<td>High head safety injection (HHSEI)</td>
<td>( N_{ss} ) 24, ( N_{sq} ) 160</td>
<td>18</td>
<td>160</td>
</tr>
<tr>
<td>Auxiliary Feedwater (AFW)</td>
<td>( N_{ss} ) 17, ( N_{sq} ) 170</td>
<td>12</td>
<td>110</td>
</tr>
<tr>
<td>Low head safety injection (LHSI)</td>
<td>( N_{ss} ) 40, ( N_{sq} ) 180</td>
<td>15</td>
<td>150</td>
</tr>
<tr>
<td>Containment spray system (CSS)</td>
<td>( N_{ss} ) 45, ( N_{sq} ) 240</td>
<td>60</td>
<td>125</td>
</tr>
</tbody>
</table>

(1) \( N_{ss} \) : specific suction velocity : \( N^*Q^{0.5}/NPSH_T^{0.75} \)

(2) \( N_{sq} \) : specific velocity = \( N^*Q^{0.5}/H^{0.75} \)
The AFW, HHSI and LHSI pumps are designed to operate with a low minimum flow rate. Satisfactory behavior can be obtained through a carefully studied design and in-depth testing. It should be pointed out that for AFW pumps in a 1300 MW plant, the minimum flow rate is less than 0.10 QBE. The pumps are highly solicited at this rate and we have observed that there are considerable pressure pulsations at the intake. Fortunately, these pumps have a strong shaft and greatly oversized bearings. Accordingly, these machines can operate with a relatively low flow.

4.1.2 CHOICE OF MAXIMUM FLOW

At maximum flow, the main risk is failure due to cavitation:

- in the diffusers,
- in the eye of the first stage impeller.

Generally these risks are well known and taken into account at the design level. In practice, this point brings us back to the problem of cavitation criteria.

4.2 DIMENSIONING WITH RESPECT TO CAVITATION

Pump dimensioning with respect to cavitation is a major problem. We will look at this from the point of view of a pump user and of an architect/engineer.

4.2.1 WHAT DO WE ASK OF A PUMP?

A pump user requires that his pump, whatever the operating rate, complies with two types of specifications:

- on a short term basis (i.e. effective immediately):
  - no ruin (rotor seizure, breakage of diffuser fittings, ruin of thrust bearing or radial bearings, etc)
  - no unacceptable noise.
- on a long term basis:
  - a reasonable component life span. In practice, this basically means that erosion by cavitation must be limited.

As a complement, the A/E will impose that, to comply with all the specified service conditions:

- the pump does not undergo any unacceptable loss of characteristics,
- the technology is kept as simple as possible,
- costs of investment, installation and operation remain reasonable. In particular it is generally desirable to limit the “depression” of the pump i.e. the change in level between the tank on the intake and the level of the first impeller of that pump.

Once again, from the standpoint of cavitation, it is evident that it is very important to define operating conditions and in particular the available NPSH. Also observe that a trade-off often has to be found between the 2 following extreme solutions:

- low "depression" + complex pump, costly, etc.,
- high "depression" + simple and sturdy pump.
4.2.2 DEFINITION OF THE AVAILABLE NPSH (NPSH<sub>av</sub>)

The NPSH<sub>av</sub> on the pump intake depends more particularly upon:

- the level of the fluid in the intake tank,
- the flow (in reality, head losses in the intake piping),
- the partial incondensible pressures which can prevail in the tank,
- the GAGE effect which appears during transient operation (de-pressurization of the tank).

In addition, the time factor is very important for estimating the erosion caused by cavitation and therefore the life of the equipment. It may therefore be worth specifying the predictable operating times for each configuration.

Figure 4 is an example of presentation of NPSH<sub>av</sub>.

When pumps have to accept high transients (for instance main feed pumps), it becomes necessary to clearly specify the operating conditions during transient states (principally the rate of de-pressurization).

Remark: when models exist, it is worth carrying out a complete simulation of the transient states. The feed pumps, the feed water tank, the regulations etc., must be modeled.

Simulation in this manner is a way of checking that the entire system is dimensioned coherently.

4.2.3 OPTIMUM INSTALLATION CONDITIONS

At the power plant draft project stage, it is often necessary to find a trade-off between the "depression" and the complexity of the pump. In all cases, at the pump inlet, the flow must be as healthy as possible, i.e., with the minimum amount of free rotation and of flow distortion.

For instance, the intake on main feed pumps is particularly well optimized to accept the de-pressurization of the feed water tank up to 50 mbar/s.

Figure 5 schematizes a few of the set-ups used for smooth running of the feed water pumps. These dispositions have been applied to the 900 CP2, 1300 and N4 plant units.

4.2.4 CAVITATION CRITERIA

As they are expressed above (4.2.1), pump user needs are difficult to quantify: at the present time it is very difficult to predict the life of an impeller pump which sustains erosion by cavitation. Therefore it is difficult for the pump manufacturer to propose a guarantee (particularly if he wants a long pump life). It is even more difficult to define the tests and criteria for pump acceptance.

Accordingly, the pump user and/or architect/engineer has so far had to use, unfortunately, known cavitation criteria which are not very good.

Because cavitation is a very vast subject, we will not go into all the developments here but refer the reader to the abundant existing literature. We will simply present the main guidelines we habitually use. In practice, they are based on a few simple ideas:
a - The criteria must be complied with throughout the operating range and not only for the normal configuration. Operation at partial flow, from the cavitation standpoint, can be more erosive and therefore more dangerous than the full rate.

b - It is essential to take into account the material and its resistance to cavitation under the conditions of use. Unfortunately, there is no simple indicator for the time being. In practice, attempts are made to determine, through experience, a velocity $U_{1\text{ lim}}$ for each material.

This velocity is estimated by checking that most of the impellers do not erode when they are operating with $U_1 < U_{1\text{ lim}}$ and with a low ratio \text{NPSH}_{av}/\text{NPSH} 3%.

This is a highly empirical notion.

It is only to be used as a guide. Let us give a few typical values:

<table>
<thead>
<tr>
<th>MATERIAL</th>
<th>Velocity $U_{1\text{ lim}}$ (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copper-aluminum</td>
<td>20</td>
</tr>
<tr>
<td>Austenitic steel</td>
<td>30</td>
</tr>
<tr>
<td>Martensitic steel</td>
<td>35</td>
</tr>
</tbody>
</table>

Note: It is assumed that the medium (the conveyed fluid) has no corrosive effect.

c - For pumps operating at high speeds, we have to estimate the cavitation intensity. At the present time, the display of cavitation figures is the most effective way of estimating cavitation intensity. Unfortunately this is a long and costly method because a model has to be built.

In practice, we have 3 types of situations as schematized in figure 6. A threshold velocity $U_{1*}$ appears: it also corresponds to an experimental value linked to the material. As an example, $U_{1*}$ is around 60 m/s for martensitic steel.

It will be observed that the major questions concern the intermediate areas (area 2 in figure 6). We strongly hope that all the research carried out over the last few years will allow better prediction of erosion by cavitation and therefore better forecasting of the impeller life-span.

### 4.2.5 CAVITATION AT PARTIAL FLOW

In this paragraph, we will dwell on the risks linked with partial flow cavitation and to do so, refer back to a problem which occurred in 1980-81 on the 900 CP2 plant feedwater pumps. Cavitation was displayed on a model in 1977-78. It showed that cavitation figures appeared at a partial load. At the time they were considered to be acceptable.

It was believed that feed water pumps would rarely operate at a partial load.

Unfortunately, experience has demonstrated that 2,000 hours at partial load is sufficient to erode the impellers considerably. Therefore, we had to design new impellers offering higher performance at a partial load, i.e. capable of producing short bubbles.
5 THE NOTION OF "TOLERANT PUMPS"

a. - The specific aspects of nuclear plant pumps have been underscored in chapter 3. In chapter 4, we discussed ranges which are often considerable within which pumps have to operate without causing trouble. Accordingly, pumps have to be able to accept variations in the influencing parameters.

b. - Finally, a pump is a set of mechanical components (bearings, shaft, seals, couplings, etc.). Each component may deteriorate: a rotor can become deformed and its residual unbalance can increase, a coupling can lose its lubricant, etc. Obviously, reliable components must be chosen. Component reliability is never perfect and the consequences of incidents which may occur have to be provided for.

Nevertheless, experience shows that for a similar cost, one technique may be "more tolerant" than another when an incident occurs. Let us consider an example.

In the event of accidental cavitation, a vertical barrel type pump featuring several bearings lubricated by the pumped fluid, has a high probability of sustaining damage. Conversely, a pump with several external bearings (lubricated with grease or oil) will generally accept this type of incident more easily.

When selecting pumps, the choice of architecture and/or "tolerant" components becomes a rule to be complied with.

Feedback from experience is extremely important: it is a way of detecting problems and therefore of establishing priorities. The example of figure 7 shows which components are most often the subject of incidents. This statistic was established on the basis of incidents which affected pumps in PWR 900 units between 1977 and 1987 or so.

For instance, it will be seen that shaft seals are components which are most affected by failure. This observation justifies all the efforts EDF has made to select, qualify and monitor the mechanical seals used.

6 EXAMPLES OF PREFERENTIAL CHOICES

6.1 EFFICIENCY

It is a natural trend for a pump supplier to propose pumps with good overall efficiency. To obtain good overall efficiency, in particular with single stage pumps, the pump manufacturer minimizes internal clearances and thus the leakages from impeller labyrinths and balancing pistons. This improves the volumetric efficiency.

This type of practice is very dangerous for power plant pumps and it is preferable to sacrifice 1 or 2 percent efficiency rather than risk the seizing up of the rotor causing a pump or a plant unit to become unavailable.

Furthermore, we will see (sections 6.6 and 7.2) how important it is to study the performance of a pump with much greater tolerances so as to be able to cope with the considerable off-centering and deflections of the shaft.
6.2 **HORIZONTAL OR VERTICAL SHAFT PUMPS?**

When an installation allows a choice between a vertical disposition and a horizontal arrangement, the latter is far to be preferred for several reasons:

- bearings are loaded by the weight of the rotor,
- lubrication is easier,
- the supporting arrangements are naturally more rigid.

6.3 **SINGLE-STAGE PUMP: BETWEEN TWO BEARINGS OR OVERHANGING ASSEMBLY?**

The standardized pump (ISO 2858) with end intake and overhung impeller is very widely used, in particular for small units (power < 160 kW).

For large pumps which have to deal with considerable solicitation (for instance high cavitation), the mounting of the impeller between two bearings is naturally more rigid and offers greater tolerance. Unfortunately, a pump mounted between two bearings is slightly more costly than an overhung pump.

6.4 **BEARINGS LUBRICATED BY PUMPED FLUID**

In vertical barrel type pumps, the internal bearings are lubricated by the pumped fluid. This fluid may contain particles. Generally, it is necessary to feed the pump bearings with a filtered fluid.

This arrangement is a delicate matter. Obviously it is preferable to use a design with oil or grease lubricated external bearings.

6.5 **LUBRICATION BY OIL OR BY GREASE?**

Lubrication of motor and pump bearings by grease is very widespread. Operating experience on nuclear power plants however indicates that this type of lubrication has some drawbacks.

Figure 8 shows the conventional evolution of the temperature of a bearing cage on starting, and then on the injection of additional grease into the bearing. In diagram a, this injection takes place when the machine is running in the steady state. We observe:

- a temperature stabilization time R (this time can be up to 2 to 3 hours) when the pump is started,
- a heating "peak" following the addition of grease. This heating peak can be characterized by its amplitude T and its duration d.

Diagram b shows what happens when grease is added too frequently.

Diagram c symbolizes the case of grease addition carried out before the machine is put into rotation.

Cases b and c can be avoided by adapting the greasing intervals. However, this is a constraint that we are making every attempt to lessen.

Case a is unavoidable. However, the arrangement of the bearing should be optimized so that amplitude T and its duration d are as small as possible. We have encountered values T > 50 K and d > 5 h on 600 kW asynchronous motors running at 1500 rpm.
To recapitulate, lubrication by grease, in spite of its apparent simplicity, sometimes poses problems and involves some operational drawbacks.

Oil bath lubrication is to be preferred as soon as bearing performance approaches the utilization limits of grease lubrication. Accordingly we have completed our specifications with this in mind.

6.6 DYNAMIC BEHAVIOR

Pump dynamic behavior (in particular rotor behavior) is an extremely broad subject. In addition, for several years now, much progress has been made in the field of modeling and experimentation. In this paragraph, we will simply indicate how our approach has evolved, one aspect being the attempt to obtain more "tolerant" pumps.

In the specifications of pump buyers, reference is still often made to the notion of critical speed. In such cases, it is imposed that the first critical speed be greater than 110 % (ISO 5199), 115 or 120 % (API 610 6th issue) or even 125% of the maximum operating speed.

In practice, pump users are unaware of the critical speeds of their machines. All they want is for their pumps to operate in a stable manner whatever the operating conditions and in particular:

- for all the predictable flow rates,
- whether the pump is new or worn,
- for all the speeds of rotation under consideration.

What is more, examination of the many pumps in service in our power plants would indicate that many of them are running very well even though their critical speed is much closer to their speed of rotation. The corresponding resonance is damped to such an extent that the rotor response to dynamic forces (unbalance, hydraulic forces, etc) is insignificant.

This means that we have to abandon criteria based upon the positions of the critical speeds and choose criteria based upon stability.

Today, our specifications have not been established once and for all. We still have to answer a few major questions:

- Which parameter should be retained? (damping ratios for the least damped mode, amplification factor of the first mode, response to an unbalance?)
- What limit threshold should be imposed?
- Can we confine investigations to mathematical modeling and in this case, how can the model be validated?

7 OVER-SIZING CAN BE DETERIMENTAL

The above paragraphs might suggest that we want substantially over-sized pumps with high margins and which might even be extremely sophisticated. This is not our approach. First, because even if the relative cost of the pumps in a nuclear power plant is low, cost is nevertheless an important factor. Second because excesses can be detrimental. We propose two simple and significant examples which demonstrate the dangers of over-dimensioning.
7.1 BALL BEARING LIFE-SPAN

Generally our specifications want the ball-bearings of our pumps and motors to be dimensioned for a life of more than 100,000 hours. The extra cost of over-sizing bearings is generally low and it therefore goes unnoticed.

Unfortunately, there are drawbacks in this practice. They can result in serious incidents on important pieces of equipment.

An over-dimensioned bearing is in essence an under-loaded bearing. The forces transmitted from the inner cage to the active ball, then to the outer cage are therefore small. The lubricant, particularly if this is grease, has a braking effect on the balls. In some cases the balls slide on the cages instead of turning. This obviously causes abnormal wear and failure of ball-bearings at an early stage. The real life of a ball-bearing is, in reality, far less than the expected 100,000 h.

Finally, it should be noted that machines are serviced relatively frequently i.e., after operating periods which are far less than 100,000 h (typically every 15,000 to 30,000 h). But on each servicing, it is almost always necessary to replace the bearings. Therefore, ball-bearings with a life-span of little more than 30,000 h could be considered acceptable.

7.2 ROTOR DIMENSIONING

Around the middle of the 70's, one of the major concerns was the problem of the dynamic behavior of multi-stage pump rotors. The notion of critical speed was the "word" and the main discussions at the time concerned the taking into account of Lomakin effects. As a short cut, specifiers wondered whether it might be sufficient to use the calculated critical speed "in the water" (thus with Lomakin's effect), or that calculated "in the air" (i.e. without Lomakin's effect).

The AFW pumps of the 1300MW units (of figure 9) were designed according to the specification: maximum speed < first critical speed in the air. To comply with this constraint, the manufacturer opted for a large diameter shaft, drilled to reduce its mass. This led to a very rugged pump from the mechanical standpoint. Indeed, it operated accidentally with very high cavitation without any consequences on the pump.

Unfortunately, the impeller eye is far too large and hydraulic operation is far from perfect (rotating stalls, surges, etc.), causing erosion by the cavitation of the first impeller. At the current time, this erosion is estimated to be quite tolerable and we intend to replace the impeller every 6 to 8 years according to the programmed servicing of each pump.

8 CONCLUSION

This paper refers to many technological aspects which are sometimes considered secondary. But pumps used in nuclear power plants must have excellent reliability. No detail should be considered insignificant.

The great number and wide diversity of pumps used will not enable any generalized specifications to be established. Analysis of real pump user needs is always essential. This is carried out principally by feedback from experience and by active cooperation between the user, the architect engineer and the manufacturer.
- Figure 1 -
Schematic safety injection system

- Figure 2 -
Typical HHSI pump working characteristics
Both nuclear and fossil power generating units are included.

The inner line is for a properly-designed impeller stage. The design margin is a function of the designer's ability and experience.

Impellers with eye larger than normal, or with high values for suction specific speed (Nss), will have a more limited operating range at off-design flows that may require very large recirculation flow rates.

Source: FBR LCS 1512 Report
How to characterize the NPSHav?

A typical graph

Legend:
- Q min.: Minimum continuous flow-rate
- Q max.: Maximum continuous flow-rate

Description of each pump operating range (for example)

<table>
<thead>
<tr>
<th>Range</th>
<th>Operating conditions</th>
<th>Anticipated operation time</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Usual operating conditions (continuous)</td>
<td>80 000 h</td>
</tr>
<tr>
<td>B</td>
<td>Exceptional operating conditions</td>
<td>1 000 h</td>
</tr>
<tr>
<td>C</td>
<td>Exceptional operating conditions</td>
<td>2 000 h</td>
</tr>
<tr>
<td>D</td>
<td>Transient conditions</td>
<td>100 h</td>
</tr>
</tbody>
</table>
Figure 5

Typical feedwater pump arrangement

COMMENTS:

1 - Connection suction pipe is designed in order to avoid flow whirl
2 - Geometric head h optimised (typically h \approx 20 m). (1)
3 - Pipe diameter optimised. (1)
4 - Suction pipe length optimised. (1)
5 - All pipe elbows are in the pump symmetrical plan.
6 - Suction pipe has a continuous down slope.
7 - Suction pipe has a long vertical length.
8 - On-off conduit valve is installed at a low level.
9 - Each feedwater pump has its own suction pipe.
10 - No permanent strainer is installed forward the booster.

A permanent strainer is provided between booster and main feedwater pump.

(1) Nota: Optimize feedwater pump suction consists to find the best compromise between - GAGE effects.
- Friction head losses.
which occur during transient operations.
Figure 6: Typical Classification of Cavitation Criteria

<table>
<thead>
<tr>
<th>Class #1</th>
<th>Class #2</th>
<th>Class #3</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Effects of cavitation bubbles</strong></td>
<td>- No erosion</td>
<td>- Low erosion rate</td>
</tr>
<tr>
<td><strong>Major consequences of cavitation</strong></td>
<td>- Loss of pump characteristics (H (Q) for example)</td>
<td>- Reduced lifetime of hydraulic parts of the pump (impeller)</td>
</tr>
<tr>
<td><strong>Cavitation criteria</strong></td>
<td>- $\text{NPSH}<em>{av}/\text{NPSH}</em>{3%} &gt; k$ with $k &gt; 1$ (typically 1.5 to 2)</td>
<td>- ?</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- Very short lifetime of impellers $\rightarrow$ Cavitation erosion not admissible</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- No bubbles</td>
</tr>
<tr>
<td></td>
<td></td>
<td>- $\text{NPSH}<em>{av} &gt; \text{NPSH}</em>{l}$</td>
</tr>
</tbody>
</table>

**Scale of Cavitation Intensity**

- $U_{1\text{lim}}$
- $U_{1}^*$

- Examples of concerned typical pumps (PWR pumps)
  - Condenser circulating
  - RHR
  - LHSI
  - Containment spray
  - Component cooling water
  - Booster feedwater
  - ....

- Auxiliary feedwater
  - High head safety injection

- Main feedwater
- Figure 7 -

Pumps Incidents Statistics

French PWR Power Plants
(1977 to 1987)

Location of Damaged Parts

- Pump casing (17.0%)
- Bearings (23.1%)
- Rotor (19.6%)
- Shaft seals (18.8%)
- Gears (2.0%)
- Couplings (4.6%)
- Miscellaneous (14.9%)
- Figure 8 -

Typical behaviour of a grease lubricated ball bearing

LEGEND
T : Bearing temperature
t : Time
S : Start-up
GI : Grease injection (topping-up)

a/ REGULAR OPERATION

b/ ABNORMAL OPERATION:
2 grease injections in a short time


c/ ABNORMAL OPERATION
Grease injection before ball bearing start-up
Mr. Roger Martin is head of the Pump Section at EDF SEPTEN (Electricité de France - Service Etudes et Projets Thermiques et Nucleaires/Design Department for thermal and Nuclear projects). He deals with research and development, technology, specifications and qualification of thermal and nuclear powerplant pumps.

Roger Martin is a graduate Engineer from "Ecole Nationale Supérieure des Arts et Métiers".

Before joining SEPTEN in 1982, he was head of Technical Department in a natural gas liquefaction plant in Algeria for 3 years.

Earlier he was involved in managing studies for conventional islands of several thermal and nuclear power plants within EDF's Design Department.
The Use of Pumps in Nuclear versus Non-Nuclear Applications

by

F. J. Bernsteiner

J. Schill

KSB AG Frankenthal

Presented to

CSNI - Specialist Meeting
on Pump Performance and
Reliability

Cologne, Nov. 1990
0. Abstract

Nuclear pump design was originally derived from the design of conventional pumps, but after extensive research and development called for by the increasing demands in the fields of hydraulics (Q, H, pump characteristic), reliability of operation and availability, it advanced into new territory. Using the examples of residual heat removal, boiler feed and main coolant pumps, this paper will show the process of this development.

Before reaching today's level of technology, tremendous progress had to made in the design and manufacturing methods; the development of new materials as well as a sizable quality assurance expenditure complement the spectrum of innovations and further developments, to the benefit of both fields of application, nuclear and non-nuclear.

Numerous problems remain to be resolved in the foreseeable future: for example pumping systems requiring still less maintenance and simplified still further or new pump series for inherent safe reactor systems, to mention just a few.
1. Introduction

In the past decades, power station engineering has been characterised by

- increasing unit ratings of fossil-fired power stations to between 600 and 800 MW in Europe.
- optimisation in respect of improving fuel utilisation by the development of new plant configurations
- the development of commercially operated nuclear power plants accompanied by a steep increase in unit sizes to 1300 - 1500 MW.

A superficial comparison of system duty data diagrams of fossil-fired and nuclear power plants, (see Fig. 1), shows that power station pumps have many common features, starting from the cooling water pumps, through feed and condensate pumps and ending with circulating pumps producing the forced-flow of the steam boiler or used for the coolant circulation through the primary loop.

The reason for developments in nuclear power station engineering deviating from those in conventional power station engineering were, and are, the legitimate user specifications, which demand that it should be possible to safely shut down the reactor under any circumstances and at any time, i.e., for example, to dissipate the residual-heat. The factors that had to be accounted for during the course of development were, among others, earthquakes, gas cloud explosions and crashing airplanes. According to specification requirements, the integrity of some of the safety systems had to be intact after an incident of such magnitude. In some case, such as for example in residual heat removal systems, our pumps have to be capable of starting up and operating safely during the incident. For none of the conventional power stations are specifications this stringent!

The list of typical power station pumps (Fig. 2) show that the principal difference between the two types of power station lies in the scope of auxiliary and safety systems in primary and secondary loops. For simplicity's sake we based our comparison on a standardized PWR.

From today's point of view it can be said that nuclear pumps originally evolved from the experienced and standardized designs; numerous requirements then led to modifications and further developments of pumps requiring extensive basic research in the fields of hydraulics, stress analysis and manufacturing technologies. The insight gained through this research could be cashed in on in form of so-called "overflow effects" in the non-nuclear business (Fig. 3).

This paper seeks to highlight the relations and differences in design, quality assurance, material selection and maintenance experience between the two fields of power engineering using various examples.
2. Design and Design Methods

During the late '60s and early '70s, interest focussed in particular on developing the hydraulics capable of producing the increased volume rates of flow resulting from increased unit ratings; significant problems in connection with partial and overload operation, cavitation, steadiness and steepness of the pump characteristic curve, radial and axial forces and the vibration behaviour had to be studied and mastered (Fig. 4).

The demand for extensive prototype and acceptance testing under realistic operating and incident conditions has led to the construction an erection of largest test loop for testing of main coolant pumps worldwide (Fig. 5).

The proof of safety to be furnished regarding the integrity of pressure shell already during the early stages of the development years led to complex and demanding problems calling for a solution. Initially, proof was furnished by working out formulas manually on just a few pages. This requirement was followed by the demand for stress analyses at thermal transients, which in turn were followed by the complex, three-dimensional analyses accounting for pipe fractures, earthquakes, airplane crashes, etc.. Research efforts led to the development of strain gauge measurements with epoxy resin models (Fig. 6) and computer codes according to the principles of the shell theory.

It took some time before it became possible to enhance the capacity of 2-D and 3-D finite element programs and related computers to the level necessary for solving highly complex problems at a reasonable expenditure (Figs. 7 and 8), standard practice today, also for non-nuclear applications.

Taking residual heat removal, feed and main coolant pumps as an example, we will now come to some design details.

2.1 Residual Heat Removal Pumps Type RHR

The single-stage, double-volute pumps used in the demanding field of process engineering served as the point of departure from which the first residual heat removal pumps were developed (Fig. 9.1). From the point of view of hydraulics, they certainly were the best possible solution. The efficiency of this type of pump is excellent, the radial forces acting on the shaft are minimal, and that over the entire operating range from partial load to overload, and the stability and strength of the pump casing is fortified by an additional volute plate. The drawbacks of such a casing are, however, that they are difficult to test and if a flaw is found, they can only be repaired at an immense expenditure, if at all!

The next step development took was the use of a single volute at the expense of high radial forces outside the point of the hy-
draulic optimum (Fig. 9.2). It hardly came as a surprise that this almost immediately resulted in several broken shafts, thicker shafts and heavy bearing pedestals in combination with impeller/casing wear ring materials that were supposedly capable of being operated with mixed friction without seizing. Casings of this type can in fact be tested, albeit with some difficulty. If re-machining is required, this does not pose any problems, because all surfaces are within reach. One fact to be remembered in this context is, however, that there were no binding calculation methods for volute casings during the '70s. Only finite element calculations were generally recognized in those years and they were not only prohibitively expensive, but in spite of their cost not entirely free from difficulties of interpretation. Heated discussions took place about the secondary stresses at the apex of the volute and at the cutwater tip. In the end this type of casing proved to be no satisfactory solution, either!

The logical conclusion was to separate pressure shell and hydraulics. This resulted in voluminous, spherical casings (Fig. 9.3), which were easy to manufacture, easy to test and could be calculated with the formulas found in relevant technical literature. Hydraulically speaking, these pumps were new territory to the engineers involved. Combinations of this type had already been developed for main coolant pumps, but these were fitted with mixed flow impellers for high specific speeds and could, therefore, not be compared to the radial hydraulics of residual heat removal pumps. The characteristic curves to be expected were an unknown factor and the hydraulic radial forces couldn’t even be approximated, because literature didn’t provide any clues on the subject.

An extensive development program was started, which resulted in light-weight annular casings and a well-founded insight into static and dynamic radial forces. Following these results, additional specification requirements led to the present level of technology incorporated in the forged RHR-pumps.

Modern residual heat removal pumps with cast or forged annular casings (Figs. 9.4, 9.5, 10) can today be operated safely across the entire operating spectrum ranging from minimum flow to far into the overload range thanks to the research and development efforts performed in the past. The attainable efficiency figures are only slightly worse than those of a comparable volute casing pump.

2.2 Feed Pumps

There can be no doubt that feed pumps of conventional design are the dominating pump type in units rating up to 800 MW. Several nuclear power stations in Germany, Argentina, Sweden and Switzerland have been equipped with competitively priced, tested ring-section type pumps with cast end casings. Barrel-type
pumps were not in demand, not for boiling water reactors, either!

It was not possible to subject the highly complex end casings to non-destructive testing completely. Some areas could only be observed with the aid of mirrors. This applies both to volumetric testing and to surface crack testing. The overview (Fig. 11) shows that these pumps fully coincided with the trend in feed pumps for conventional power stations. The only difference was to be found in the hydraulics! The relatively low pressures led to small numbers of stages, between 2 and 4, while the larger flow rates resulted in radial hydraulics and the highest specific speeds possible. Pumps with these technical features are installed in the Obrigheim, Stade and other NPSs and have to date been in operation for over 100,000 hours without any complaint.

A new hydraulic design had to be found as units were expanding to sizes in excess of 1,000 MW. The pump volume rates of flow went up from 1,800 t/h to 3,750 t/h. Basically, it would have been possible to increase the number of pumps from 3 to 6, but this is no alternative, as this solution is uneconomical. A further option would have been to drastically lower the speed in order to return to feasible hydraulics.

Using a 1,300-MW PWR as an example, we will demonstrate which size single-suction ring-section type pumps this would have resulted in.

Duty point:

\[
\begin{align*}
\text{Flow rate} & = 4,100 \text{ m}^3/\text{h} \\
\text{Head} & = 880 \text{ m}
\end{align*}
\]

These data could have been achieved with one of the three design variants below:

<table>
<thead>
<tr>
<th>Variant</th>
<th>Speed</th>
<th>Stage number</th>
<th>( n_q )</th>
<th>Impeller ( \varphi )</th>
<th>Booster</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2950</td>
<td>2</td>
<td>33</td>
<td>650</td>
<td>yes</td>
<td>single-suction</td>
</tr>
<tr>
<td>2</td>
<td>1450</td>
<td>3</td>
<td>22</td>
<td>1000</td>
<td>yes</td>
<td>single-suction</td>
</tr>
<tr>
<td>3</td>
<td>1450</td>
<td>3</td>
<td>22</td>
<td>1000</td>
<td>no</td>
<td>double-entry suction impeller</td>
</tr>
</tbody>
</table>

One feature all of these pumps have in common is their heavy weight and their consequently poor cold starting ability. Pump variant 1 can be expected to have characteristic curves that are
hardly suited for parallel operation and the accuracy it requires.

In the face of this situation, the double-suction, single-stage design with a low speed booster pump and a high speed full-load pump virtually forced itself upon those involved in the search for a solution to the problem. The booster pumps employed in conventional power stations and the large single-stage, double-suction pumps from the water supply sector were then taken as a basis. Both types of pump are characterized by a lucid and simple design. Also the high pressure produced by one stage of the main pump and the high velocities associated with that had already been experienced. The fact that was initially overlooked was that one would be entering new territory with regard to additional pump characteristic requirements and frequent part-load operation by throttling keeping the pumps at constant speed. The result of this oversight was cavitation, which destroyed the impellers after only a short period of operation (300 h). The problem was solved by taking a larger booster pump with a head of approx. 140 m and impellers of waste-wax castings with armoured-plated and profiled flow-in edges for the main pump. The pump casing selected was a cast double-volute which initially led to fractured shafts at some manufacturers’. Because of the large casting tolerances it sometimes happened that the impeller did not run in the hydraulic centre of the double-volute, and that this eccentricity in turned gave rise to extremely high radial forces, as extensive and costly test series showed. All the same, these cast casings are still state-of-the-art on the world market (Fig. 12). They are by no means test-friendly and could not be calculated exactly, either. Although the latter fact is possible today, albeit at some expenditure.

Departing from above marginal conditions, the hydraulic data obtained for booster and main pump were as follows:

<table>
<thead>
<tr>
<th></th>
<th>Booster pump</th>
<th>Main pump</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow rate</td>
<td>4,100 m³/h</td>
<td></td>
</tr>
<tr>
<td>Total head</td>
<td>880 m</td>
<td></td>
</tr>
<tr>
<td>Head [m]</td>
<td>140</td>
<td>740</td>
</tr>
<tr>
<td>Speed [min⁻¹]</td>
<td>1490</td>
<td>5190</td>
</tr>
<tr>
<td>Impeller-Φ [mm]</td>
<td>700</td>
<td>480</td>
</tr>
<tr>
<td>n_q for each impeller shroud</td>
<td>28</td>
<td>28</td>
</tr>
<tr>
<td>NPSH [m]</td>
<td>8</td>
<td>140</td>
</tr>
<tr>
<td>Efficiency [%]</td>
<td>87</td>
<td>86</td>
</tr>
</tbody>
</table>
This solution is a markedly economical one, because it entails a total efficiency of over 86%. A solution incorporating a ring-section type pump would have 83% at the most.

The design concept based on a larger booster pump allows the main pump to run in the duty point "bubble-free" (incipient cavitation), while both profiling and armour-plating allow it to run in the partial load range for longer periods of time.

The close of the '70s saw a sudden change in the safety philosophy and there was a growing demand for pumps with a forged pressure shell (Fig. 13), and this applied to both booster and main pump. This requirement led to a division in a computable, 100% testable pressure shell and a cast hydraulics section characterised by optimised hydraulic geometries for minimised outer diameters.

KSB supplied a total of 75 double-suction reactor feed pumps with the necessary booster pumps. Of this number, 28 pumps have a forged casing.

Standard conventional power station engineering knows of no other component that compares to these large, double-suction pumps with a power requirement of up to 12,000 kW. A full-load pump for a 700 MW coal- or oil-fired boiler has following duty data:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow rate</td>
<td>725 kg/sec</td>
</tr>
<tr>
<td>Total head</td>
<td>325 bar</td>
</tr>
<tr>
<td>Speed</td>
<td>5275 min⁻¹</td>
</tr>
<tr>
<td>Power requirement</td>
<td>28,000 kW</td>
</tr>
</tbody>
</table>

As a rule, these pumps (Fig. 14) have 3 or 4 stages, yet the development work for nuclear pumps has triggered an enormous wave of innovations, among them:

- Profiled, armour-plated suction impellers

- Safe design of booster and main pumps with a view to avoiding cavitation erosion

- Observation of cavitation bubble behaviour as a criterion for acceptance testing

- Production of waste-wax castings using the waste-wax process handling unit weights of up to 50 kg

- Armour-plating of in-flow edges with stellite

- Manufacturing by spark erosion of suction and discharge sides of the blades at the entry into the impellers.

- Spark erosion or milling of diffusers.
2.3 Main Coolant Pumps

It is not difficult to see that circulating pumps for conventional furnaces originate from the same source. This type of heating plant, operating with a system pressure of up to 110 bar and temperatures of up to 325 °C, have not undergone any changes and are still equipped with horizontal, shaft-sealed volute-casing pumps. At higher pressures, i.e. up to 320 bar and operating temperatures of max. 420 °C, the proven pump type best suited to fulfill the requirements is a glandless vertical design initially of the double volute type, later with a spherical casing and a water-filled motor, the so-called wet motors (Figs. 15.1 and 15.2). As the motor has to withstand the full system pressure, its housing has to be dimensioned accordingly. The motor windings are surrounded by the water and are not protected by a can. Because of this arrangement, there is no need for a shaft seal. The hot pump is separated from the motor by a heat barrier. The heat loss of the motor is dissipated through a cooling circuit. In the Federal Republic, these conventional circulating pumps are subject to the "Technical Rules for Steam Boilers", TRD, which means that the scope of testing as well as the testing criteria themselves are laid down in detail. The constructional details, including a verification of the strength calculation, the quality control plan and the examinations required by the QCP are subject to the approval of an independent inspection agency.

At first glance, the main coolant pumps of innovative nuclear reactors feature the same construction properties. The pumps employed in pressure-water reactors will serve as the basis for our reflections in the following. As the time frame available for this discourse is limited, we will not include the pumps installed in heavy-water and sodium-breeding reactors, or the plug-in pumps for boiling-water reactors.

Previously, smaller size BWR, such as Oskarshamn II and Barsebeck I and II, were still fitted with external pumps with wet rotor motors (Fig. 15.3). Apart from their voluminous, spherical casings, these pumps do not differ greatly from conventional circulating pumps. This design concept, incorporating water-filled electrical motors, could not be used for pressure-water reactors, because the required motor ratings were far beyond the range of experience. As the horizontal volute casing pumps with shaft seal could only provide the basic principles from which to fill the requirements for conventional units (Fig. 15.4) -- the reason being that these pumps were designed for similar pressures and temperatures, but had a much too small power rating -- designers had to enter new territory. The volute casing was soon dropped, because the ever growing unit sizes called for ever larger volume rates of flow, which only mixed flow impellers could cope with. Attention therefore diverted to the spherical casing and all the advantages it has to offer: simple to calculate, to manufacture and to test, able to transform considerable nozzle forces and moments, and relatively insensitive to thermal shock. Also in this case, increasing requirements caused the forged design to gradually take the place of the cast version (Fig. 15.6 and 15.7).
The pump in Fig. 16 represents the present level of technology. The features worth noting in this standard pump for the so-called "Konvoi Units" are:

- its forged, one-piece spherical casing;
- its impellers and diffusers cut from forged blanks;
- the fact that the driving torque is no longer transferred from the shaft to the impellers through keys, but through Hirth-type serrations with a necked-down bolt, which accounts for the remarkably low-vibration operation of this pump;
- oil-lubricated axial and radial bearings with integrated oil supply plant on top, and a water-lubricated hydrodynamic carbon bearing in close proximity to the impeller guiding the rotor;
- with regard to the shaft seal, the pump is fitted with hydrostatic and, to an increasing extent during recent years also with hydrodynamic seals, whose advantages are their long service life and emergency-operating properties allowing each of its three stages to keep operating at full system pressure for several thousand hours after a failure of either or both of the other two stages. Several plants are presently being retrofitted with hydrodynamic seals in place of the old hydrostatic seals.

3. Material Selection and Requirements

In non-nuclear applications, material selection is normally based on economic aspects and on experience, e.g. the level of corrosion resistance of gray cast iron up to defined levels of water chemistry, or the fatigue limit proved by parts in comparable applications. There are some requirements for pressure shells of boiler circulation and boiler feed pumps.

For nuclear pumps, material selection is determined by authorities and customers. Extensive basic examinations are necessary to qualify new material for nuclear applications. Supplies to different markets complicate the application of uniform materials, because these have to meet the requirements of different sets of standards (ASME, VdTÜV, etc.). The testing criteria often also vary from inspection agency to inspection agency, undoing any ideas of combining blanks to reasonably priced batches. Fig. 17 shows a selection of material requirements in comparison.

4. Manufacturing and Quality Assurance

From what was said before, you could conclude that the widely adopted use of forged components for nuclear pumps represents today’s level of technology and this is so, to some extent, and present developments are definitely going in that direction.

The quality of cast components has, however, also been improved considerably in recent years. If we consider their price, weight and the base materials used for their production it is clear that much can be said for the use of cast parts. Thanks to the
availability of highly sophisticated testing methods, such as X-ray examinations, we have detailed information at our disposal and no longer need to speculate about the more complex geometries we are dealing with.

As the metal removal rates have also seen a remarkable improvement, the use of forged blanks is becoming less of a question of cost. Not just the parts of the pressure shell with its simple geometry, but also the complex hydraulic sections are produced from 100%-tested blanks. Hydraulic designs based on ruled surfaces are fed directly into modern 5-axis-milling machines; the impellers and diffusers produced this way are characterized by accurate contours, the absence of joint flaws and hydraulically smooth surfaces (Fig. 18).

The same is true for the production of pumps for fossil-fired power plants. The pump manufacturer is additionally faced with questions of optimising the stock of components, economic batch-sizes and minimising delivery times.

Supplying good quality products is something all pump manufacturers strive for, but quality has its price. However, there is no room for compromise in the production of pumps for nuclear applications, in particular in areas where the safety of operation of these pumps is of paramount importance! All areas of nuclear operation are today governed by an extensive scope of national and international quality assurance requirements (Fig. 19). Combined with general specifications of nuclear power system supplies, these requirements are reflected by the QA-systems of the manufacturers.

Many of these requirements also find their way into the QA-systems for non-nuclear fabrication, the foundation of these systems being formed by the classification of the relevant products in different categories according to their application, and within these categories in parts that are important from a safety viewpoint, parts that are important for the functioning of the plant, and small parts. All quality assurance activities are based on this classification. At leading manufacturers, these general specifications are normally complemented by internal QA-activities, which are proven improvements resulting from long-term experience.

Fig. 20 shows that the scope of quality assurance conducted in the design and fabrication of nuclear and non-nuclear pumps can be very different. A vast amount of documents subject to approval have to be compiled for nuclear applications. All stages of fabrication have to be monitored, hold points for customers and inspectors planned, and subcontractors qualified. The workshop staff is obligated to document every single nonconformance and to have these examined through formal, specified channels instead of taking impromptu decisions in keeping with the production plan. To sum it up it can be said that the pump manufacturer has to strictly adhere to a Quality Assurance Manual and that this manual has to be approved both by the customer and the inspection agency. A further factor needing to be considered are the regular audits that have to be scheduled. We do not want to argue the need for all of these activities here, but the few lines above explain the long lead times in this field!
5. Summary and Prospects

Our long-term experience with the development and production of pumps for nuclear and non-nuclear applications shows that the two fields are presently in a state of positive coexistence through individual problem-solving and subsequent know-how transfer.

The design of some pumps has branched off in several very different sub-designs as a result of the contrasting requirements that had to be met. However, the underlying aim accompanying all activities is to use the positive experience made with components, assembly groups and systems for the benefit of other applications. The use of simple shapes suitable for checking, the trend towards forgings and the application of internationally documented materials have become increasingly popular in the conventional business as well.

Design methods, hydraulic designs and tests, fabrication and vibration analyses as well as production processes are utilized in both fields of application and developed further. The nuclear business sets the standards of quality and quality assurance to the benefit of the non-nuclear business.

A glance into the future shows it holds many a large problem in store for the pump manufacturers. These problems are partly the result of technological progress in general, and partly necessitated by new requirements to be fulfilled by future reactor systems. The objectives of future research and development are, among others:

Low-maintenance pumps - low dose rate;

- modern sealing systems made from low-wear materials having a long service life (e.g. hydrodynamic seal systems)
- systems for extensive in-service-inspections (e.g. crack examinations of main coolant pump shafts)

Simplified pumping systems characterized by a further reduction of their susceptibility to failure;

- increased use of medium-lubricated bearing arrangements doing away with the need for auxiliary supply and cooling systems

Intelligent pumping systems allowing early-warning fault indication and preventive maintenance;

- the systems available today for the monitoring of vibrations, temperatures, and pressures will serve as the basis for the development of expert systems that can be integrated in the supply systems of the power station.

New pump series for inherent safe reactor systems;

- raising of the hydraulic data is expected to lead to the development of glandless pumps, either designed as canned motor pumps rated at up to 2,000 kW, or as wet motor pumps with a power input of over 5,000 kW.
Pumps in auxiliary systems

Fig. 1
## Major Pumps in Power Plants

<table>
<thead>
<tr>
<th></th>
<th>Fossil Fired Power Plant</th>
<th>Nuclear Power Plant</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coolant circulation</td>
<td>boiler circulation pumps</td>
<td>main coolant pumps</td>
</tr>
<tr>
<td>Auxiliary and</td>
<td>none</td>
<td>high pressure safety injection pumps</td>
</tr>
<tr>
<td>safety systems</td>
<td></td>
<td>high pressure charging pumps</td>
</tr>
<tr>
<td></td>
<td></td>
<td>residual heat removing pumps</td>
</tr>
<tr>
<td></td>
<td></td>
<td>emergency component cooling pumps</td>
</tr>
<tr>
<td></td>
<td></td>
<td>fuel pool cooling pumps</td>
</tr>
<tr>
<td>Feedwater</td>
<td>boiler feed pumps</td>
<td>boiler feed pumps</td>
</tr>
<tr>
<td></td>
<td>(multistage,</td>
<td>(single stage-high speed or multistage)</td>
</tr>
<tr>
<td></td>
<td>sectional/barrel type)</td>
<td>booster pumps</td>
</tr>
<tr>
<td></td>
<td>booster pumps</td>
<td>booster pumps</td>
</tr>
<tr>
<td>Auxiliary systems</td>
<td>(none)</td>
<td>emergency feedw. pumps</td>
</tr>
<tr>
<td></td>
<td></td>
<td>startup and shutdown pumps</td>
</tr>
<tr>
<td>Condensate</td>
<td>condensate pumps</td>
<td>condensate pumps</td>
</tr>
<tr>
<td>Condensor cooling system</td>
<td>cooling (circulating)</td>
<td>cooling (circulating)</td>
</tr>
<tr>
<td></td>
<td>water pumps</td>
<td>water pumps</td>
</tr>
</tbody>
</table>

Fig. 2
Contribution and Interdependebility of both nuclear and non-nuclear to design improvements and innovation

**non-nuclear**

Experienced and Standardized Designs

**nuclear**

Modifications

- examination requirements
- safety aspects and in-service-inspections
- easy and quick replacement of parts after being in service
- fluid chemistry, contamination
- seals / leak tightness
- increased Q-H requirements
- steep pump characteristic curves

New Developments

Overflow benefits

- design and calculation methods
- material development
- manufacturing technique
- quality assurance

*Fig. 3*
Geschäftsbereich Pumpen Energietechnik
Engineered Pumps Division

Strömungslabor des Forschungszentrums im Werk Frankenthal

Hydraulic test centre within the Rand D facilities at Frankenthal plant

Fig. 4
Vollastprüffeld
Hauptkühlmittelpumpen
für Kernkraftwerke

Full Load Test Facilities
Main Coolant Pumps for
Nuclear Power Stations

Fig. 5
Strain gauge measurements with epoxy resin models
Finite Element Models

MCP (PWR Konvoi-type)

Impeller

Boiler Feed Pump Barrel

Double Volute Casing with Nozzles

Fig. 7
Finite Element Calculations

Temperature Distribution

Stress Concentrations of Shrinked Bearing Sleeve

Stresses Caused by Nozzle Loads

Fig. 8
Evolution of Residual Heat Removal Pumps

Fig. 9
Baureihe: RHR
Horizontale, einstufige
Ringgehäusepumpe

Type: RHR
Horizontal, single-stage
annular casing pump

Pumpe für Hilfs- und
Sicherheitssysteme

Pump for auxiliary
and safety systems

Fig. 10
Baureihe: RHD
Topfgehäusepumpe
Type: RHD
Barrel pump

Speisewasserpumpe
Boiler feed pump

Fig. 12
Baureihe: RHD
Topfgehäusepumpe

Type: RHD
Barrel pump

Speisewasserpumpe       Boiler feed pump

Fig. 13
Baureihe: CHT
Hochdruck-Mantelgehäusepumpe

Type: CHT
High pressure pump barrel design

Hochdruck-Kesselspeisepumpe

High pressure boiler feed pump
EVOLUTION OF MAIN COOLANT PUMPS
GLANDLESS DESIGN - WET MOTOR

Fig. 15.1
Fig. 15.2
Fig. 15.3

DESIGN WITH SHAFT SEALS

Fig. 15.4

Fig. 15.5
Fig. 15.6
Fig. 15.7

Fig. 15
General material requirements

<table>
<thead>
<tr>
<th>Requirements</th>
<th>fossil fired application</th>
<th>nuclear application</th>
</tr>
</thead>
<tbody>
<tr>
<td>material selection</td>
<td>relatively free general specifications for BCPs and BFPs</td>
<td>specifically prescribed by authorities and plant owners</td>
</tr>
<tr>
<td>ductility</td>
<td>required for pressure retaining parts</td>
<td>min. ductility and critical fracture toughness (KIC) specified</td>
</tr>
<tr>
<td>corrosion</td>
<td>only important for special cases (e.g. water chemistry)</td>
<td>full corrosion resistance required due danger of system contamination</td>
</tr>
<tr>
<td>corrosion fatigue</td>
<td>few justifications required</td>
<td>justification of life time expectance required</td>
</tr>
<tr>
<td>strength</td>
<td></td>
<td></td>
</tr>
<tr>
<td>degree of cleanliness</td>
<td>not necessarily specified</td>
<td>rigorous cleanliness and surface quality specified because of NDE requirements and for detection of surface flaws and their interpretation, free from heavy metals and halogen</td>
</tr>
<tr>
<td>heat treatment</td>
<td>depends on mechanical properties to be achieved</td>
<td>to enhance mechanical and technological properties, to eliminate residual stresses</td>
</tr>
</tbody>
</table>

Fig. 17
Manufacturing of MCP-Impeller
Quality assurance systems

non-nuclear

System requirements

National / international requirements

ASME Sec. I
TRD
(for BCPs)

nuclear

System requirements

National / international requirements

ASME / ANSI / NQA
DIN ISO 4001 - 4003
KTA 1401
10 CFR 50 App B
BS 5750
NS 5801 - 5803
CAN CSA Z 299.1 - 4

+ General Specifications of system suppliers:

KWU AVS D 100/50
KWU QSP 4a

Manufacturers QA-System
(standardized fabrication)

Audits

Audits by

Customer, ASME,
Regulatory organisations or their agents

Manufacturers Audit Program (optional)

Manufacturers Audit Program
Quality assurance emphasis in design and manufacturing

**non-nuclear**

- design
  - concentrated on development of new generic and modular products
    (BCPs acc. to codes)

- procurement and manufacturing documents incl. calculations

- control and release of instructions procedures, drawings
  - bill of material
  - welding qualification procedures and plans
  - material and weld metal testing, examination and inspection plans
  - procurement of subcontracted items

- manufacturing process:
  - examination, tests inspection
  - deviations and repairs

- documentation:
  - retention after delivery

**nuclear**

- specified by customer and authorities e.g.
  - design input basis
  - design requirements
  - regulatory requirements
  - codes, standards identified and documented and their selection reviewed and approved by customer, RPI and third party agencies

- control of documents generated by manufacturer and his suppliers

- standardized and activated by starting the actual part-list

- by manufacturers engineering and QA-departments

- extensive surveillance activities by customers and authorities only with approval of customer/authorities

- docu of all records for a period of 40 years

Fig. 20
SESSION #2: "OPERATING EXPERIENCE OVERVIEW"

Chairman: M. Maris

* "Pump performance and reliability follow-up by the French safety authorities",
  J.-P. Clausner, IPSN, France,
  X. de la Ronciere, IPSN, France,
  E. Scott de Martinville, IPSN, France,
  P. Courbiere, SCSIN, France
  (presented by J.-P. Clausner)

* "Trend of incidents and failures of pumps in Japanese nuclear power plants",
  S. Nakamura, MITI, Japan,
  M. Harima & M. Hada, NUPEC, Japan
  (presented by M. Hada)

* "Reliability of steam turbine-driven standby pumps used for safety-related applications in U. S. light water commercial power generating plants",
  J. R. Boardman, NRC, United States
  (presented by J. Rosenthal, NRC)

* "Operating feedback on pumps in French PWR nuclear power plants",
  R. Larue, EDF - SEPTEN, France

* "Incidents attributed to pump problems",
  G. Ishack, OECD Nuclear Energy Agency
PUMP PERFORMANCE AND RELIABILITY FOLLOW-UP

BY THE FRENCH SAFETY AUTHORITIES

Jean-Pierre CLAUSNER, Xavier de la RONCIERE, E. SCOTT de MARTINVILLE
Institut de Protection et de Sûreté Nucléaire
Commissariat à l’Energie Atomique
Fontenay aux Roses, France

Pierre COURBIERE
Service Central de Sûreté des Installations Nucléaires
Ministère de l’Industrie et de l’Aménagement du Territoire
SUMMARY

Since pumps play a major role in the nuclear power plant operation, the Safety Authorities have to pay special attention to the pump performance and reliability, in particular to the pumps used in the safeguard systems that are operated only during periodical tests. The Nuclear Protection and Safety Institute (IPSN) is the technical support of the Safety Authority: the Central Service for the Safety of Nuclear Installations (SCSIN). For this reason, the IPSN assesses the safety-related pump performance and reliability and expresses its view on findings observed during the startup tests period and plant operation.

The means used by IPSN to ensure that pump performance and reliability are in accordance with technical specifications are mainly based on:

- startup test program analysis,
- surveillance test programs, maintenance programs and requalification test programs analysis,
- significant incident reports evaluation.

Regarding, particularly, the safety-related pump follow-up during plant operation, the goals of the safety analysis are:

- to assess the effectiveness of the surveillance programs,
- to detect the failure modes and track the degradation trends, by using measurable parameters including functional indicators, and monitoring degradation factors.

This follow-up is especially important for the safety-related pumps which are not used during normal operation, but are only actuated under abnormal situations such as emergency core cooling signals and, consequently, are of a primordial importance for the plant safety. However, these pumps being, during most of the plant life, in a standby mode, it is important to assess whether maintenance, surveillance and monitoring practices are elaborated enough. In particular, wearing out of components, identified during the qualification test programs as being the most sensitive to the long term accidental running conditions, has to be closely monitored.

Maintenance doctrine applied to the pumps determines for a large part their reliability and availability. Maintenance program application and requalification test results are analyzed and conclusions on the program validity are drawn. By analyzing event files and incident reports, the DAS tracks which design, operation, maintenance, and condition-monitoring factors require investigation and improvement.

This paper will present, through actual examples, the methodology of the performance and reliability safety-related pumps evaluation applied by the French Safety Authorities and the lessons drawn from this evaluation.
1 INTRODUCTION

The Institute for Nuclear Safety and Protection (IPSN) is the technical support of the Central Service for the Safety of Nuclear Installations (SCSIN), the French Safety Authority. The IPSN assesses the control and maintenance programs for safety-related equipment, such as engineered safety feature system pumps and analyses the program of periodical tests and their results.

As concerns PWR units, the present French situation is as follows:

- construction of units in standardized series, derived from the previous ones by controlled development,
- operation of all the units by a single operating organization, also prime contractor for their construction, Electricité de France (EdF),

which creates favorable but demanding experience feedback conditions.

The maintenance doctrine defined by EdF, considers two aspects:

. surveillance of the functional capabilities,
. surveillance of the material status.

The functional capabilities are checked each 2 months through the periodic tests, and after material inspections which involve disassemblies. The periodic testing is also used to have an insight in the material state, but maintenance activities with different programs and frequencies are performed during the refuelling periods. A summary description of the maintenance activities performed on the safeguard pumps is presented in table 1. Application of these maintenance programs is complemented in each plant by examinations and actions which are considered to be necessary. A feedback towards the central organization allows to generalize useful actions and to make studies on all the machines.

Reviewing the event files shows that the pumps are the active components which require the most servicing. Therefore, the events, significant incidents, maintenance activities and actions taken, are analyzed by IPSN in order to ensure that functional requirements as well as reliability of these pumps are met. The pumps follow-up during plant operation requires a special attention, and the goals of the safety analysis are:

- follow-up of the actions engaged by the utility after detection of a deviation or an abnormal condition,
- assessing the effectiveness of the test and control programs,
- highlighting the weaknesses of the maintenance programs by comparison with the event files and the incidents reports,
- tracking the degradation trends by using measurable parameters including functional indicators.
Two types of equipment should be considered, according to the frequency of their use:

- for the equipment with a frequent use, like the turbine-driven pumps of the steam generator auxiliary feedwater system, the experimental data base is large enough, so that it is possible to observe results obtained during operation and adjust their maintenance program and improve their reliability on a rather short time period with a well-known methodology. For these components, the standardization of the French nuclear program is favorable, since each anomaly is used to draw lesson for all components.

- for the equipment which is used only during periodical tests, like the safety injection or the containment spray pumps, the reliability is more difficult to evaluate in particular because during periodical tests, pumps are not used as during accidental conditions. Flow rate, temperatures, duration could be different and the results obtained during tests should be extrapolated to demonstrate the availability for accident mitigation.

This paper intend to show how the Safety Authorities deal with these two points. In order to clarify the technical issues, the equipment which is used in normal operation will be dealt with separately from the standby safety systems. For this standby equipment, the qualification, the maintenance and the qualification following intervention will be examined successively. On each item, the Safety Authorities preoccupations will be illustrated by significant examples that will allow to back up the final discussion and justify the conclusions.

2 PUMPS USED IN NORMAL OPERATION

The Safety Authorities analyze periodically the incidents files and deliver an opinion on the maintenance quality based on the operation results. The first example to be dealt with in that prospect is the reactor coolant pumps maintenance for which a summary will be presented. A second example will be given on the auxiliary feedwater system in order to confirm the methodology.

2.1 Reactor coolant pumps

For these pumps, which are used continuously during operation of the plant, the French Safety Authorities monitor carefully to the operating feedback in order to be sure that EdF is drawing lessons of each incident. Examples of improvements brought to the 900 MWe maintenance program, following three items that were highlighted by the operating feedback are described hereafter:

- seals and studs of the pumps casing,
- shaft seals
- pumps bearings.
The defects on the seals and studs of the pumps casing resulted in a large number of small leaks inducing a risk of stud corrosion.

The main actions taken consist in:
- improving the technical specifications for the seals,
- increasing the clamping strength of the studs,
- and periodically visiting the joint face.

The standard maintenance program now includes the joint face visit at any disassembly, or at least every two years. A magnetic particle testing is also foreseen in case of stud dismounting, which is replaced by an ultra sonic control at each complete visit if the studs are not removed.

As for the shaft seal number one, the main incidents were due to impurities in the fluid, too low pressure difference across the seal, bad seal profile and difficulties in slidding of the movable ring. The results of the operating experience for the seals are an increase of the visit frequency and a more systematic change. In order to correct the bad seal profile risk, the frequency of the control was increased to each three refuelling outages for visual inspection with more restrictive criteria on the seal mirror quality. For the problem of the movable ring, an increase of the visit frequency was decided to control its erosion.

For the number two and three seals, the incidents types were similar and the surveillance is performed with the same kind of criteria as for the seal number one.

Among the incidents that affected the pump bearings, only one resulted in the shutdown of the unit, when a sudden increase of the seal number one leak was measured. The other anomalies were detected during periodic visit; in some cases, the observation of small cuttings on the seals led the utility in pushing the visit program further. The follow-up of the pumps vibrations is considered as the main anomaly detection system. The spectral analysis of the vibrations is now performed on all the pumps because it was shown to bring a precursory warning for different anomalies. This system is still under progress.

2.2 Auxiliary feedwater system (AFWS)

The "AFWS" system provides the steam generators with an auxiliary feeding, which is used to extract the residual heat generated in the core down to the point where the specific residual heat removal system can take over. The AFWS system is also designed for removing the residual heat in accidental conditions: therefore, it is a safeguard for the nuclear plant.

The French 900 MWe unit auxiliary feedwater system includes two motor-operated pumps and one turbine-driven pump.

The turbine-driven pump is of a centrifugal type with an horizontal shaft and eight compression stages. The impulse turbine is a single stage turbine, fed by steam extracted on each steam line, upstream the isolating valves. These lines are permanently warmed up to the automatic starting valves which are closed in normal operation. The condensed water is evacuated through drain valves.
Every two months, the turbine-driven pump is tested on its low-flow circulation line, in order to assess its functioning and its reliability to start under demand. Each train (A or B) is tested separately.

Most of the incidents involved turbine-driven pump trips at nominal power, in particular during periodic testing and were related to the turbine inlet equipment. These incidents resulted in pump unavailability of variable duration without actual consequence, the motor-driven pumps being available. However, it would have been different in case of loss of off-site power supply.

Therefore, it is worth considering two types of incidents according to the duration of the unavailability:
- inadvertent trip, resulting in a short duration of the auxiliary turbine-driven pump unavailability, that could be repaired with a local action on a protection command system,
- mechanical coupling ruptures, that gave way to longer duration unavailabilities.

This last category include the most significant incidents. A common cause of these ruptures is the instability of the steam flow at the control valve when the turbine-driven pump starts. This phenomenon induces vibrations of the valve gate which were neither taken into account at the design stage, nor noticed during the commissioning stage of the first operated units. The periodic test being performed by only one valve opening, the problem could have been worsened in the real conditions. The analysis of these incidents highlighted several weaknesses in the design:
- bad evacuation of the condensates coming from the feed line,
- poor guiding of the valve gates,
- brittleness of the valve command,
- brittleness of the protection plugging mounting gear.

The generic difficulty for the control of these turbine-driven pumps comes from the fact that they are supposed to start either with low pressure in the steam generators or with high pressure. A particular attention was brought to the turbine speed control.

The first results obtained in 1986 show that the experimented reliability of this equipment was ten times smaller than the one considered initially in PSA studies; its value was consequently decreased which showed a critical importance of that piece of equipment.

The main modifications consist in an hydraulic speed control system, an improved design of the throttling valve, a new mounting gear for the over speed mechanical protection, and a water plug evacuation system.

Recommendations by Safety Authorities were aimed at pursuing the studies on the condensate evacuation, improving the experimental feedback on the event files and the preventive maintenance programs.

The lessons drawn from these events show that the follow-up of the material submitted to continuous operation allows to improve substantially their reliability. The difficulty is greater for the material which is in standby during normal operation.
3 PUMPS IN STANDBY EQUIPMENT

The pumps used in standby conditions include those of the containment spray system, of the intermediate pressure injection system, low pressure injection system, and of the auxiliary feedwater system used in emergency conditions.

The activities which allow to follow-up this equipment are factory tests for qualification in accidental conditions or tests on the nuclear plant, maintenance, and qualification after intervention. These activities will be successively dealt with in the following paragraphs, starting with the qualification.

3.1 Qualification in factory

Safety-related pumps are designed to fulfil their safety function in accidental conditions. This design must take into account safety rules and requirements, operating parameters in post-accident situation such as pressure, temperature, flow rate, activity, etc..., but also factors such as:

- duration of the safety mission,
- life time of components,
- instrumentation allowing to monitor operating parameters,
- means of control allowing to predict any degradation process.

This implies that during the design process all factors liable to affect the components have been identified and the necessary means, especially as regards monitoring, have been defined in order to allow the follow-up of the equipment status and assess that this status is compatible with its expected mission: ensure the long duration operating in post-accident conditions.

The qualification program for the safety-related equipment important for safety takes into account several types of loading: earthquake, accidental ambient temperature, particular loadings due to operating.

In a general manner, it includes considerations coming from analyse, analogies, with the component or the complete equipment, and operating experience. The qualification against earthquake is obtained through scientific analysis but some tests were performed on vibrating tables in order to confirm the modelling used.

As examples, containment spray and safety injection pumps are qualified through an endurance test performed with water at different temperatures up to 120°C, including thermal shocks.

3.2 Qualification on site during startup tests

For each engineered safety feature system, IPSN verifies that the test program takes into account:

- functions to be ensured for each equipment,
all configurations of the system, normal operating conditions but also incidental situations, such as operation with the system partly unavailable (within the frame work of the authorized technical specifications), functional requirements (minimum or maximum flow rate, criteria, allowances, etc...) for each configuration.

For the engineered safety feature systems, most of the requirements come from accidental situations which cannot be simulated even during commissioning. It is therefore necessary to transpose the experiments to these conditions.

Two connected incidents will be described in order to show that qualification may be a complicated matter.

**Endurance tests**

During the first months of 1985, on unit 2 of the St-Alban nuclear plant (1300 MWe), an endurance test of 2000 hours was performed on one pump of the containment spray system, in order to test this pump in “real” conditions.

During this test, the containment spray pump was running through the test flow line (700 M³/h) to the reactor water storage tank. When the temperature of the reactor water storage tank began to rise, vibrations on the pump increased and operators shutdown the pump. When they restarted the pump, vibrations had disappeared, but after a few hours the same phenomenon was observed.

Investigations showed that the thermal dilatation obtained by this change could not be accommodated by the coupling system between the motor and the pump, when the pump was running. Indeed the couple value was too high to allow the normal slippage that should occur along the longitudinal groove of the coupling shafts if the friction coefficient were correct. This resulted in a compressive axial force in the shaft that lifted the rotor part of the motor that was only sustained by a ball bearing. In these conditions the degradation of the motor bearings might be much quicker.

After a loss of coolant accident, the suction of the containment spray system pumps and safety injection system should be shifted from a cold reservoir to the containment sump water in order to recirculate the water. During this transient, the phenomena could occur, so it was decided to modify the bearing system in order to implement a double action bearing.

A new test was performed in order to check that the modification would not bring drawbacks. The lesson which was drawn by the safety authorities from these events is that a maximum number of tests has to be reformed on the safety systems in conditions similar to those that can be expected in accidental conditions.

On another reactor, some years later (August 1989), a periodic test containment spray pump exhibited a high level of vibrations. The test was repeated with a better instrumentation that showed a continuous increase of the vibrations amplitude throughout the test which was stopped after four hours. When restarted five minutes later, the vibrations amplitude had turned back to a normal level. It was shown there that the slidding of the coupling groove could not take place during the test, because of the degradation of the coupling device.
After this incident, the requalification tests were discussed again and improved by creating a qualification test at the manufacture level in order to test the sliding resistance of the coupling with the full power transmission. The spurious friction effect that appeared in these incidents is still under study.

The experience has shown that some specific problems have been revealed only during prolonged on-site operation. This is true, for example, for the motor jacking of vertical shaft pumps, and for the pump lubrication and cooling conditions. It is worthwhile repeating that the lubrication and coupling problems of certain pumps were identified owing to their continuous operation (charging pumps of the 900 MWe units). Lastly, it should be noted that for certain items of equipment, the demonstration of validity of operation was made by means of lengthy on-site tests.

It appeared that qualification of a machine model, such as a pump set, had to be completed by an endurance test on the first-off unit, to supplement in-shop tests intended to check performance parameters calculated at the series design stage and by special loop tests applied to the ability to withstand accident conditions. Verification of adaptation to actual operating conditions requires a test which specialists estimate should last at least 2000 hours (continuous, if possible). Environmental interaction is required (environment, effects of actual suction and discharge systems, auxiliary cooling systems, etc...) along with prolonged operation to reveal anomalies overlooked during tests. This is particularly important for emergency pumps which operate only during periodic testing. The French Safety Authorities requested on-site endurance tests to be carried out as part of first-off tests and asked Electricité de France to examine this possibility for the N4 standardized series.

**Incidental conditions**

The auxiliary feedwater system for the French 1300 MWe includes two redundant trains, each of them feeding two steam generators. Each train has two identical pumps, one powered by the steam of the corresponding steam generator, the second is powered by an electric motor which is backed up in case of emergency.

The procedure used to start the pumps during periodical tests involves two steps:
- first, put on the electrical pump of the lubrication circuit which has to establish a sufficient pressure,
- then start the AFWS pump (motor or turbine-driven pump) which will drive the pump for lubrication.

The electrical lubrication pump stops after a time delay.

The emergency procedure to start the AFWS involves the electrical lubrication and the main pump at the same time. Furthermore, in case of offsite power loss, the main turbine-driven pump starts alone, and the lubrication relies on the driven pump only. In order to qualify these emergency starts, the commissioning tests for the AFWS include a start-up without the electrical lubrication pump.

In 1987, during a unit hot functional testing, both motor-driven pump of the AFWS were put on without the electrical lubrication. The oil pressure was normally established in 15 seconds, decreased afterwards, and the bearing temperature increased, which was stopped by a voluntary action in the control room: the pumps were found blocked with a degradation due to temperature. The mechanical
blockage was associated with bearings degradations and small cuttings were found in the lubrication oil.

This insufficient lubrication could have resulted in the loss of both emergency feedwater lines in case of accident.

The analysis of this incident has shown a too low temperature resistance of the bearing pads, and problems in oil circuit design. It is worthwhile mentioning that this anomaly was detected after several hours of operation, because the same test on the new equipment was satisfactory.

Following this incident, the utility defined the following improvements:
- a new type of bearing pads was tested and exhibited a satisfactory behavior in case of starting without initial lubrication,
- the design of the lubrication circuit was improved and qualified during the commissioning of a new reactor.

4 REQUALIFICATION AFTER A MAINTENANCE OPERATION

One of the main sources of problems during maintenance is due to foreign material penetrating the circuits. The Safety Authorities have to look very carefully at the requalification program after each operation.

In August 89, during the 10-year shutdown at Fessenheim unit 1, after modifications on the safety injection system, a provisory filter was installed to protect the pumps in order to conduct a specific test. During the rinsing operation, a argon chamber used during the maintenance was forgot in the system and then displaced up to the provisory filter and plugged the pipe. During the requalification test, the pressure difference led to the filter deterioration and the plug came in the suction nozzle of the pump. The pump impeller was blocked and the circuit breaker tripped.

This incident was followed by an exhaustive inspection of all the safeguard systems to look for other possible foreign parts. A similar plug was found in the containment spray system, which could have resulted in closing one of the spraying lines.

Actions were therefore engaged by the utility in order to check the quality of the maintenance work performed and to improve the cleaning control procedure on the circuits. The Safety Authorities estimate that these actions are satisfactory, but they draw the attention of the utility on the 10-year shutdown during which a large number of similar operations should be performed.

The Safety Authorities are now very much concerned about the requalification of the material which appears to be an essential item in the overall safety evaluation. It is thought that important progresses have to be made in that area which is now under discussion.
5 CONCLUSION

For the Safety Organizations, the issue is to be ascertain that the engineered safety feature system pumps have been designed, tested and maintained in order to guarantee their safety function. Generally speaking, it appears that if defined surveillance and maintenance programs allow a satisfactory monitoring of the equipment status evolution versus time, there is still some room for improvement.

The question is the assessment of the damages found during periodic tests or during maintenance for their potential impact on the safety function: would the equipment be able to operate following a demand?

The safety organizations consider that the extrapolation of maintenance action on the reliability is the main issue.

The experience acquired during startup tests provides valuable information regarding design and qualification validation and some issues have been shown up during endurance tests on the site. EDF was requested to consider such a test performance for the future plants.

During unit operation, the only scheduled test programs are periodic tests and requalification tests after refuelling. Operational experience acquired on French units has shown that despite the care taken in the periodic test and maintenance program definitions, some problems come to light by chance, thereby confirming the need for a lengthy test period. Therefore, the IPSN considers that during the startup tests of future units, it will be needed to make the tests as extensive as necessary to guarantee the safety of installations and increase operating experience, particularly in the field of engineered safety feature systems.

Some events occurred, following modifications and maintenance, showing the need for EDF to review the intervention and requalification procedure organization.

To conclude, analysis of incidents related to emergency pump and analysis of maintenance programs applied on these pumps show some areas liable to improvements:

- carrying out first-off and endurance tests on site for engineered safety feature equipments which do not operate continuously in normal operating conditions,
- studying the risks of operating conditions liable to lead to pump vibrations and wearing out,
- performing the necessary arrangements regarding filtration methods to reduce the risks associated with foreign materials,
- considering the lubrication circuit linked to the pump reliability as well as the maintenance program adequacy,
- performing arrangement regarding maintenance on redundant equipments particularly regarding schedule,
- carrying out comprehensive requalification tests,
- pursuing improvements in reliability data collection related to periodic test and maintenance results,
- using PSAs results for assessment of program adequacy,
- studying alternative methods to assess the long term reliability for the material which is operated only during the periodic testing.

Maintenance and surveillance test programs are found generally efficient and ensure satisfactory performance and reliability of the pumps. However, some generic defects which could not be found during these activities and, therefore, liable to jeopardize the safety function, were revealed by performing additional actions such as qualifications tests representative of accidental conditions, endurance test performed on site and first-off tests. The results obtained through such actions, increased the confidence given to the safety related pumps in performing their safety function and brought a significant contribution in maintaining the level of safety of the units.
<table>
<thead>
<tr>
<th>TYPE AND FREQUENCY OF THE MAINTENANCE PROGRAM</th>
<th>CONTENT OF THE MAINTENANCE PROGRAM</th>
<th>COMPLEMENTARY MAINTENANCE ON SPECIFIC MATERIAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>TYPE 1: EVERY 3 MONTHS DURING PERIODIC TESTING</td>
<td>VIBRATIONS MONITORING BEARINGS TEMPERATURE SURVEILLANCE</td>
<td>FOR HHSI AND IHSI: OIL LEVELS AND LEAKAGES DETECTION FOR HHSI ONLY: PRESSURE/FLOW RATE OF LUBRICATION PUMP</td>
</tr>
<tr>
<td>TYPE 2: EACH REFueling</td>
<td>SHOCKS DETECTION ON BALL BEARINGS GREASING OF BEARINGS AND COUPLINGS OIL ANALYSIS</td>
<td>GREASING OF IHSI BEARING FOR HHSI ONLY: COASTDOWN TIME GEAR AND COUPLING STATES</td>
</tr>
<tr>
<td>TYPE 3: 1/3 REFueling FOR IHSI 1/4 REFueling FOR HHSI 1/8 REFueling FOR LHSI AND SPRAY</td>
<td>VISUAL EXAMINATION OF: MULTIPLYING GEAR MECHANICAL COUPLINGS ALIGNMENT CONTROL AFTER REASSEMBLY</td>
<td>FOR HHSI ONLY: SUPPORT PADS, OIL FILTERS DRIVEN PUMP OIL BEARINGS STATE</td>
</tr>
<tr>
<td>TYPE 4: 1/8 REFueling FOR IHSI AND HHSI 1/16 REFueling FOR LHSI AND SPRAY</td>
<td>COMPLETE DISASSEMBLY ALL SEALS NEW HEAT EXCHANGERS TIGHTNESS</td>
<td>FOR LHSI AND SPRAY ONLY: ALL WEARING RINGS NEW ALL BEARINGS NEW</td>
</tr>
</tbody>
</table>
Specialist Meeting on
Pump Performance and Reliability

Trend of Incidents and Failures of Pumps in Japanese
Nuclear Power Plants

S. Nakamura
Nuclear Power Operating Administration Office
Agency of Natural Resources and Energy, MITI

M. Harima and M. Hada
Nuclear Power Safety Information Research Center
NUPEC

Abstract

As of end of 1989, 36 light water reactor type nuclear power plants are being operated in Japan. The number of major events reported to MITI was 355 cases respectively, in which events relating to pump were 53 cases respectively.

This paper deals with the statistical evaluation of the events relating to pumps.

"Pump system" is defined to be consisted of four blocks, ie :
- Pump proper (mechanical part)
- Driving motor
- Power source equipment, and
- Controlling equipment
- Statistical assessment of events were performed specifically on:
  - trend of events (classified by system blocks and by the cause of events)
  - Operating years (aging related)
  - Effect on plant operation, and
  - Countermeasures taken
- Number of reports of the events on pumps is continuously decreasing for past years. This decreasing is considered to be attributable to the following :
  - Improvement in maintenance (preventive maintenance and self-imposed inspections)
  - Feedback of operating experiences into design and operation
  - Complete execution of countermeasures (including feedback to other utilities)
  - Connection and evaluation of overseas events information and its reflection to the domestic events
1. INTRODUCTION

Japanese utilities are obligated to report incidents and failures occurred at their commercial nuclear power plants (NPPs) to the Ministry of International Trade and Industry (MITI) in accordance with the laws (Electricity Utilities Industry Law and Law for the Regulation of Nuclear Sources Material, Nuclear Fuel Material and Reactors) and the administrative guideline of MITI.

There are 36 NPPs being operated in Japan at the end of 1989 and their operating experience counts about 370 reactor years. In this period, the number of reporting reached 355 cases. With the increase of operating data, requirement for the systematic compilation and the efficient utilization of the information reported became stronger.

The Nuclear Power Safety Information Research Center (NUSIRC) has been established to respond such requirement in 1984. The main objective of this organization is to operate a data bank of the incident and failure report (reported to MITI), to administrate and to assess the data mainly for government use.

NUSIRC is performing a series of evaluation on the incidents and failures to grasp the status of safety management of NPPs in Japan, to support regulatory activities of MITI on NPPs and to utilize effectively the information on incidents and failures. This paper describes the "Trend of Incident and Failures of Pumps in Japanese Nuclear Power Plants", which is one of the such assessment performed by NUSIRC.
2. DESCRIPTION OF INCIDENTS AND FAILURES

The number of commercial LWRs under operation amounted to 36 units at the end of 1989, and the number of reports for the past 20 years was amounted to 355 cases. Among them, incidents and failures related to pumps was 53 cases. The following will describe the outline of the incidents and failures.

2.1 History of Past Incidents/ Failures

Fig.2.1 shows the number of LWRs in operation, the number of reports, and the reporting rate (no. of reports/no. of units) on a calendar year basis.

- Concerning the number of units in operation -

Since the first commercial LWR started its operation in 1970, the number of units in operation has increased steadily, amounting to 36 at the end of 1989. The number will increase further according to the current construction plans of the electric power companies.

- Concerning overall incidents and failures -

The average reporting rate in the past 20 years from 1970 through 1989 was 0.96 cases/reactor year (355/370). In 1971, when three nuclear power plants were in their first operation, there occurred a number of initial failures, showing the very high rate 4.0 cases/unit (12/3). Since then the rate has been decreased steadily to the level of about 0.6 cases/unit in 1980 and after. Recently (for three years) the rate is 0.55 cases/reactor-year.

Although another small peak of reporting rate existed in 1981, this may be attributable to the fact that the interpretation of the term "failure, etc." in the LRNR was clarified in this year so as to include events and failures during the adjusting operation, commissioning and shutdown.
2.2 Description of Incidents and Failures in Pump Facilities

Among the incidents and failures reported to MITI, 53 cases were related to pumps. The following are the outline of the incidents and failures of pump facilities.

Fig. 2.2 (b) shows the trend of the number of reports related to the pump facilities, and the reporting rate (no. of reports/no. of units) during the 20 years on a calendar year basis.

The average reporting rate in the past 20 years was 0.15 cases/reactor-year (53/355). As described in the above, in 1971, high rate (0.63 cases/unit) was recorded. Since then the rate has been decreased steadily to the level of about 0.1 in 1981. In 1985, it showed a very low value of 0.03. In 1988 and 1989 the rates were 0.17 cases/no. of units (4/34), and 0.19 (7/36) respectively.

As mentioned above, the number of reports on the incidents and failures of the LWRs in Japan is steadily decreasing, showing the increased safety and reliability of plant operation. These reduction of incidents and failures can be attributed not only to guidance from the regulatory side, but also to the positive efforts made by the utilities to prevent the recurrence of incidents and failures.
3. ANALYSIS OF INCIDENT/FAILURE IN PUMP FACILITIES

This chapter concerns the further assessment of the incidents and failures relating to the pump facilities. First, the pump facilities are broken down into the following four categories from their functions: pump proper, control device, power supply and motor. An analysis is made mainly about the relation between the respective devices and the incidents/failures, cause analysis and influence on plant operation.

3.1 Classification of Facilities and Component Functions

Incidents and failures relating to the pump facility are classified according to above four categories.

Pump proper is regarded as a facility which directly form a part of the plant system, and other categories such as control equipment, power supply and motor are regarded as a device which belongs to the pump system.

- Classification of the above categories on a functional basis -

(1) Pump proper consists of the main shaft, bearing, shaft seal and casing. Piping is also included in the category of the pump proper.
(2) Control equipment for controlling pump function consists of pump control device, and instrumentation piping for the detector is included in the control equipment category.
(3) Power supply for driving the pump includes the breaker of the main power supply to the connector box in front of the driving portion. Therefore, the MG set also belongs to the power supply.
(4) Motor consists of drive device, cooling system for the drive portion, cables and terminals from connector box to the driving portion.
Fig. 3.1 (a) shows the classification which the 53 events are first classified on the facility basis and further on a system basis.

Among the 53 cases of incidents and failures,

34 cases took place in the pumps of reactor cooling system.
2 cases emergency core cooling system
4 cases reactor auxiliary system
9 cases condensate and feedwater systems

and

4 cases other systems than above

(1) Incidents and failures related to the reactor cooling system pumps amounts to 34 cases, thus according for 64.2% (34/53). They are classified as follows on a functional basis:

13 cases pump proper
2 cases control equipment
9 cases power supply
10 cases motor

(2) Incidents and failures related to the emergency core cooling system amounts to 2 cases, thus accounting for 3.8% (2/53).

One case took place in HPCI system, and the other core spray system.

(3) Incidents and failures related to the reactor auxiliary system amounts to 4 cases, thus accounting for 7.5% (4/53).

3 cases took place in RHR system, and the other one in reactor coolant cleanup system.
(4) Events and failures related to the condensate / feedwater system amounts to 9 cases, thus accounting for 17 % (9/53).

4 cases took place in FW system, 2 cases in condensate system and the other 3 in reactor recirculation system. On functional basis, one belongs to pump proper and the remaining 8 to control equipment.

(5) Events and failures related to other systems amounts to 4 cases, thus accounting for 7.5 % (9/53).

2 cases took place in instrumentation control system (CRD hydraulic), one cases in steam/turbine system ((turbine control) and the other on is radwaste system (liquid).

Fig. 3.1 (b) shows classifications of 53 cases events and failures on the functional basis, and further the component basis.

These incidents and failures are classified as follows:

<table>
<thead>
<tr>
<th>Cases</th>
<th>Component</th>
</tr>
</thead>
<tbody>
<tr>
<td>22</td>
<td>pump proper</td>
</tr>
<tr>
<td>10</td>
<td>control equipment</td>
</tr>
<tr>
<td>9</td>
<td>power supply</td>
</tr>
<tr>
<td>10</td>
<td>motor</td>
</tr>
</tbody>
</table>

and

2 cases others

These incidents and failures are further classified on a component or part basis as follows:

(1) The incidents and failures in pump proper are classified as follows:

<table>
<thead>
<tr>
<th>Cases</th>
<th>Part</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>main shaft</td>
</tr>
<tr>
<td>4</td>
<td>bearing</td>
</tr>
<tr>
<td>9</td>
<td>shaft sealing (6 shaft seal, 3 seal water piping)</td>
</tr>
<tr>
<td>3</td>
<td>others (1 flange, 1 impeller, 1 piping)</td>
</tr>
</tbody>
</table>
(2) The incidents and failures of control equipment are classified as follows:

- 7 cases detector
- 1 case control circuit
- 2 cases control power

(3) The incidents and failures of power supply are classified as follows:

- 8 cases variable frequency power supply (including 3 cases for control oil system and system and 5 cases for generator control (2 brushes))
- 1 case constant voltage adjusting power supply

(4) The incidents and failures in motor are classified as follows:

- 5 cases power supply cable and terminals
- 5 cases motor cooling system (2 cooler oil tank, 2 cooling pipe)

(5) Other 2 cases are misoperation which has no affect to the facility.

The incidents and failures classified on a basis of system facility and function are described above. Table 3.1(a), (b), (c) and (d) shows cases, the number of reports, corrective measures and evaluation of the measures for every component of these functional categories.
3.2 Cause Analysis

The initiating causes of the incidents/failures in the pump facilities are classified on a process basis such as design, manufacturing, installation, and maintenance.

Further, the maintenance of pump facilities are divided roughly into the following two: maintenance work and maintenance control. And, further the maintenance work and the maintenance control are further divided into inspection and restoration, as well as work control (such as incorporation of foreign material), work under plant operation, and planning (such as inspection plan, replacement plan).

Fig. 3.2 (a), (b) shows the results of the causes analysis of 53 cases of incidents/failures in the pump facilities. Causes of these incidents/failures are classified on a process basis as follows:

6 cases  design deficiency
9 cases  manufacturing deficiency
11 cases  installation deficiency
12 cases  maintenance deficiency
15 cases  maintenance management deficiency
(1) Design deficiency (6 cases)

includes 4 cases of deficient structure (3 cases in which pump vibrations propagated to instrumentation piping causing malfunction of pressure sensor, and 1 case in which vibrations caused fatigue failure of instrumentation piping, leading to water leakage), and 2 cases of deficient material selection for possible corrosion (1 case in cooling water pipe for air conditioner - water leakage affected the pump operation and 1 case in seal water piping for RW system - pump shaft seal).

(2) Manufacturing deficiency (9 cases)

includes 6 cases of fabrication/assembling deficiency cases of wrong fabrication dimension, 2 cases of wrong assembling dimension, 1 case of fabrication mar and 1 case of insufficient tightening, and 3 cases of welding deficiency (2 cases of insufficient penetration in welds at the hydrostatic bearing and bearing ring, and 1 case of deficient silver brazing at connection terminal of current transformer for designing the generator).

(3) Installation deficiency (11 cases)

includes 5 cases of assembling installment work deficiency at site, 2 cases of welding deficiency and 4 cases of work control deficiency.

(4) Maintenance deficiency (12 cases)

includes 3 cases of deficient overhaul inspection and reassembling in periodical inspection, 7 cases of insufficient tightening, and 2 cases of poor adjustment.

(5) Maintenance management deficiency (15 cases)

includes 5 cases of poor work control (2 cases in which tools necessary for overhaul inspection are left in the system, 3 cases of possibly poor water quality control in disassembling of shaft seal), 4 cases of wrong work under plant operation, 4 cases of poor maintenance
inspection planning, and 2 cases of misoperation.

Tables 4(a), (b), (c) and (d) show the situation of corrective measures taken.

Fig 3.2(c) shows the comparison of the trend of incident/failures for 1970’s and 1890’s. The former show the total number of reports while later is given for individual calendar years.

Total number of incidents/failures reports relating to pump facility was 26 cases for 1970’s in which installation deficiency was dominant for 9 cases.

After 1987, number of reports becomes some what higher total 13 cases in which 5 cases of maintenance management deficiency is the largest fraction among design deficiency (1 case), manufacturing deficiency (3 cases), installation deficiency (2 cases) and maintenance deficiency (2 cases).

From the standpoint of aging, 2 cases of maintenance management deficiency and 1 case of design deficiency are considered. Also, deficiencies during manufacturing or installation caused 4 cases of incidents/failures by combining with aging factor such as vibration or oxidation etc.

Fig 3.2(d) shows the comparison for the same periods classified by causes.

Installation deficiency in 1970’s (9 cases) decreased in 1980’s (2 cases). Incidents/failures relating to the construction of the plant (design, manufacturing and installation) were the most significant in 1970’s while incidents/failures for plant operation were the major part in 1980’s.

It will be noted that maintenance or maintenance management is the important factor for the plants with longer operating period.
3.3 Effect of Incidents /Failures on Plant Operation

Classification on the effect of the incidents/failures on the plant operation were as follows:

Table 3.3 shows the effect of the incidents/failures relating to pump facility.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Automatic shutdown</td>
<td>16 cases (30.2%)</td>
</tr>
<tr>
<td>Manual (forced) shutdown</td>
<td>25 cases (47.2%)</td>
</tr>
<tr>
<td>Power reduction</td>
<td>3 cases (5.7%)</td>
</tr>
<tr>
<td>No effect</td>
<td>9 cases (17.0%)</td>
</tr>
</tbody>
</table>

Note:

The effect of overall incidents/failures (355 cases) on plant operation is classified as follows:

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Automatic shutdown</td>
<td>35%</td>
</tr>
<tr>
<td>Manual (forced) shutdown</td>
<td>30%</td>
</tr>
<tr>
<td>Power reduction</td>
<td>1%</td>
</tr>
<tr>
<td>No effect</td>
<td>34%</td>
</tr>
</tbody>
</table>

Figs. 3.3(a) and (b) show the effect of incidents/failures of pump facilities on plant operation and the relations incidents/failures of pump facilities with functional categories and causes.
(1) Automatic shutdown

On a functional category basis.

8 cases of control equipment, 4 cases of power supply, 1 case of pump proper, 1 case of motor and 2 cases of others. Control equipment account for larger percentage (50%).

On a cause basis.

3 cases of design deficiency (pump vibrations propagated to piping, causing malfunction of sensor), 1 case of manufacturing deficiency (insufficient tightening at shop), 1 case of installation deficiency (insufficient tightening), 4 cases of poor maintenance (2 cases of insufficient tightening, 2 cases of poor adjustment), 7 cases of maintenance management deficiency (2 cases of poor replacement plan, 3 cases of wrong work under plant operation, 2 cases of misoperation).

(2) Forced manual shutdown

On a functional category basis.

13 cases of pump proper, 2 cases of control equipment, 2 cases of power supply, and 7 cases of motor.

On a system basis.

22 cases of reactor cooling system, and 2 cases of condensate/feedwater systems.
On a cause basis.

1 case of design deficiency (poor support design leading to fatigue failure of pipe due to vibration). 4 cases of manufacturing deficiency (2 cases of poor fabrication/ assembling, 2 cases of deficient welding). 4 cases of poor maintenance (insufficient tightening in periodical inspection). 8 cases of maintenance management deficiency (4 cases of poor work control, 2 cases of poor replacement plan).

On a situation basis.

6 cases of steam leakage in containment leading to an increase of floor drain. 6 cases of the increase in shaft seal water flow. 7 cases of vibration of bearing. 3 cases of high coolant temperature. 1 case of low system pressure. 2 cases of power reduction, and 1 case found by the patrol during re-startup.

The above results suggest the importance of the daily work made by the operators, and of the maintenance of sensors indicating precursor signs.

(3) Power reduction

On a functional category basis.

All 3 cases are due to power supply.

On a cause basis.

1 case of design deficiency (wrong material selection), and 2 cases of maintenance management deficiency (1 case of introduction of foreign material or metal pieces in control panel, resulting in short circuit, and 1 case of wrong work under plant operation).
Fig 3.3(c) shows the relation between plant operational situation on the initiation of incidents/failures of pump facilities and detection means of these incidents/failures.

The forty three cases of the incidents/failures which took place during the normal plant operation were detected by the following measures:

30 cases by alarming of instruments installed in the central control room:
12 cases by abnormal indication of instrumentation (that events were secured by manual shutdown)
1 case by patrol.

The 10 cases of the incidents/failures identified during periodical inspection were as follows:

2 cases during overhaul in the in-service inspection
4 cases during adjustment/performance test following the overhaul
3 cases during plant re-startup
1 case during commissioning after construction
3.4 Effect of Operating years on Incidents/Failures

The operational experience of LWRs in Japan has exceeded twenty years. From the standpoint of the extension of service life of the existing LWRs, the effect of operating length years on incidents/failures was surveyed.

Figs. 3.4 shows the progress of the reporting rates of incidents/failures of total and pump facility for every elapsed years of plant operation.

For the pump facility,

(1) The reporting rate of pumps is decreasing with increasing operating years, with similar trend of the total incidents/failures.

(2) During the periods of short operating years (0 to 1 year, or 1 to 2 years after commissioning), the reporting rate (no. of cases/ no. of units) shows a rather high value. While the average reporting rate over the past 20 years is 0.13, the reporting results in 0 to 1 year and 1 to 2 years were 0.36 and 0.18 respectively.

(3) When the operation year exceeds 10 years, the reporting rate becomes 0.09, which is somewhat lower than the overall average reporting rate (0.13) over the past 20 years.

This may suggest that not only the utility companies successfully performed or performing preventive maintenance of the component, but also the extension of service life of the pump facilities will not pose any particular challenges.
The incidents and failures which took place within one year after the start of commercial operation shows following trend:

(1) Trend of incidents/failures of pump facilities during the first year of operation

The reporting relating to pump facilities during the first year of operation were 15 cases for the past 20 years. These incidents/failures are classified as follows:

On a functional category basis.
5 cases Pump proper
3 cases Control equipment
2 cases Power supply (including wrong work in plant operation)
4 cases Motor
1 case Other

On a cause basis.
1 case Design
2 cases Manufacturing
9 cases Installation
2 cases Maintenance management

(1) 1 case of design deficiency is deficient support arrangement, which propagated pump vibrations to the sensor, resulting malfunction.

(2) 2 cases of manufacturing deficiency include one case for wrong fabrication dimension, and the other for incorrect clearance.

(3) 9 cases of installation deficiency are classified into 4 cases of poor construction management (3 cases of seal water leaks in the system due to the introduction of foreign material to shaft seal, and 1 case of the leftover of metal piece in the control box), and 5 cases of wrong installation (2 cases of insufficient tightening of electric components and 3 cases of insufficient tightening of fitting to mechanical parts).

(4) 2 cases of maintenance management deficiency are 1 case of operation in plant operation, and 1 case of wrong work.
Among the total 15 cases of incidents/failures, 13 cases took place before 1980, and only 2 cases took place thereafter.

In general, incidents and failures within one year after commissioning are said to take place depending on the comprehensive technical capability for plant construction. Therefore, it suggests that the capability of the manufacturers has been enhanced in these years, as well as the results of efforts for recurrence prevention of incidents/failures by the utility companies.

3.5 Effect of Operating Months Following Periodical Inspection on Incidents/Failures

Recently, the extension of operation cycle is being studied from the standpoint of the enhancement of capacity factor of NPPs.

Figs. 3.5 shows the trend of reporting/operation cycle of the total incidents/failures those relating to pump facilities for every month elapsed from the completion date of periodical inspection (first connection to grid).

- Total events/failures -

(1) Reporting rate within one month following the completion of periodical inspection is 0.184/cycle, higher than the mean value of 0.06 for an operation cycle.

(2) The reporting rates for one, two and three months after startup decrease rapidly, showing the minimum of 0.02/operating cycle during the 2-3 months. Then, it goes up slightly, and during 3 to 10 months levels off to 0.08/cycle.

(3) The reporting rates after 10 months are 0.03/cycle, smaller than the average value of 0.06 for an operation cycle.
- Incidents/Failures of pump facilities -

(1) The **maximum** peak of the reporting rate lies in 5 to 6 months after the re-startup, showing a difference from the trend of the overall incidents/failures.

(2) Reporting rate within one month following the completion of periodical inspection is 0.020/cycle, slightly higher than the average value of 0.013 for an operation cycle. The reporting rates for two, three and four months after startup decrease rapidly, showing the **minimum** of 0.07/operating cycle during the 3 to 4 months.

(3) Then, it goes up slightly, and during 5 to 6 months reaches to the **maximum** of 0.028/cycle.

(4) The reporting rates after 6 months are levelling off to 0.010/cycle, lower than the average value of 0.06 for an operation cycle.

(5) The reporting rates after 10 months are 0.008/cycle, lower than the average value for an operation cycle.

The results of above evaluation shows relatively low level of incidents /failures relating to the pump facility especially one month after the completion of periodical inspection.

This may suggest that the overhaul and performance confirmation test are effectively performed in the periodical inspection.
4. Conclusion

Number of reports of events on pumps is continuously decreasing for past years. This decreasing is considered to be attributable to the following:

- Improvement in maintenance (preventive maintenance and self-imposed inspection)
- Feedback of operating experiences into design and operation
- Complete execution of countermeasures (including feedback to other utilities)
- Connection and evaluation of overseas events information and its reflection to the domestic events
Fig 2.1 Trend of Incidents/Failures (in Calendar Year)

Fig 2.2 Trend of Incidents/Failures relating to Pump Facilities (in Calendar Year)
Fig 3.1(a) Incidents/Failures relating to Pump Facilities Classified by System/Facilities

Fig 3.1(b) Incidents/Failures relating to Pump Facility Classified by Components and Functions
<table>
<thead>
<tr>
<th>Location of Incidents/Failures</th>
<th>Outline of Cause</th>
<th>No. of Occurrence</th>
<th>Actions Taken</th>
<th>Countermeasure and Evaluation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main shaft</td>
<td>Initiating cause was machining damage which propagates to fatigue fracture due to bending stress of the shaft.</td>
<td>2 (1-CRD system and 1-FILL/HP Injection system)</td>
<td>- Integrity confirmation of materials by UT and MT-replacement of main shaft and mechanical seal.</td>
<td>Case of CRD system pump - Thorough materials inspection, monitoring improvement for vibration and temperature. - Temperature detector was newly provided for bearing part.</td>
</tr>
<tr>
<td></td>
<td>Initiating cause was fretting crack which propagates to fatigue fracture due to bending stress of the shaft.</td>
<td>2 (RHR pumps)</td>
<td>- Integrity confirmation of materials by UT - replacement of main shaft and impeller.</td>
<td>- Reinforcement of inspection (once in 2 years). - Improvement of operation scheme (pump was used for undesirable long period low flow operation, which was omitted in the revised operating manual).</td>
</tr>
<tr>
<td></td>
<td>Heated plastic twisted failure due to insufficient lubrication.</td>
<td>1 (CRD pump)</td>
<td>- Integrity confirmation of materials by UT - replacement of main shaft and mechanical seal. - Adjustment of feed line piping for lubricant.</td>
<td>- Management not to change the position of lubricant feed piping. - Temperature detector were newly provided for bearing part.</td>
</tr>
<tr>
<td></td>
<td>Damage was caused by foreign particle introduced into the system.</td>
<td>1 (RCP)</td>
<td>- Replacement of main shaft and impeller</td>
<td>- Strengthen monitoring of motor noise.</td>
</tr>
<tr>
<td>Bearing Hydrostatic bearing</td>
<td>Insufficient welding was the initial cause - fatigue fracture was resulted due to repeated stress by pressure pulsation</td>
<td>2 (PLR)</td>
<td>- Integrity confirmation of welded part by PT - replacement of hydrostatic bearing.</td>
<td>- Thorough inspection after welding of bearing plate. - Welding structure bearing will be replaced by single body structure.</td>
</tr>
<tr>
<td>Location of Incidents/Failures</td>
<td>Outline of Cause</td>
<td>No. of Occurrence</td>
<td>Action Taken</td>
<td>Countermeasure and Evaluation</td>
</tr>
<tr>
<td>--------------------------------</td>
<td>----------------------------------------------------------------------------------</td>
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<td>---------------------------------------------------</td>
<td>-------------------------------------------------------------------</td>
</tr>
<tr>
<td>Ordinary bearing</td>
<td>Wearing due to insufficient lubrication. (Condensate system)</td>
<td>1 (Condensate system)</td>
<td>Replacement of coupling part</td>
<td>Re-evaluation of frequency of changing lubricant oil.</td>
</tr>
<tr>
<td></td>
<td>Wearing due to drop off of the bearing pin (incomplete attachment).</td>
<td>1 (RCP)</td>
<td>Replacement of bearing assembly.</td>
<td>Shortening inspection period.</td>
</tr>
<tr>
<td>Shaft seal</td>
<td>Small foreign particle were flown into the seal surface and caused scratches which</td>
<td>6 (3 - PLR)</td>
<td>Replacement of shaft seal assembly.</td>
<td>Inspection improvement.</td>
</tr>
<tr>
<td></td>
<td>extend to wear resulting increase of seal water leakage.</td>
<td>(1 - CUW)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>(2 - RCP)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Piping</td>
<td>Cooling water leakage due to insufficient tightening.</td>
<td>2 (1 - PLR)</td>
<td>Control of torque for tightening the flange of the piping.</td>
<td>In case of PLR:</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(1 - RCP)</td>
<td></td>
<td>Reinforcement of monitoring for</td>
</tr>
<tr>
<td></td>
<td>Seal water leakage due to corrosion of the piping (wrong selection of material)</td>
<td>1 (RW)</td>
<td>Replacement of piping.</td>
<td>- Seal water temperature</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>- Control breed-off flow</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>- Seal purge flow</td>
</tr>
<tr>
<td>Casing flange</td>
<td>Reactor water leakage from selective crackings due to welding deficiency.</td>
<td>2 (PLR)</td>
<td>Repair welding (over-laying after removing the deficient part).</td>
<td>Thorough execution of torque control for tightening of piping.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>- Execution of hot inspection at the plant start-up.</td>
</tr>
<tr>
<td>Impeller</td>
<td>Vibration fatigue fracture due to stress concentration to improper shape of key way of the impeller.</td>
<td>1 (RHR)</td>
<td></td>
<td>Improved shape of key way (large corner radius).</td>
</tr>
<tr>
<td>Location of Incidents/Failures</td>
<td>Outline of Cause</td>
<td>No. of Occurrence</td>
<td>Action Taken</td>
<td>Countermeasure and Evaluation</td>
</tr>
<tr>
<td>--------------------------------</td>
<td>---------------------------------------------------------------------------------</td>
<td>-------------------</td>
<td>--------------------------------------------------------</td>
<td>--------------------------------------------------------------------</td>
</tr>
<tr>
<td>Control facility Detector</td>
<td>Deficient support of instrumentation piping caused propagation of delivery pressure pulsation to the instrumentation piping causing malfunction of the detector.</td>
<td>3 (1 - CS) (1 - Control oil system) (1 - FDW)</td>
<td>- Separate from common support. Independent support was newly provided.</td>
<td>- Modification of supporting method.</td>
</tr>
<tr>
<td>Control circuit</td>
<td>Multipurpose of detector due to submerged signal cable.</td>
<td>2 (Circulating water system)</td>
<td>- Anti-water measure was provided.</td>
<td>- Countermeasure for flooding.</td>
</tr>
<tr>
<td></td>
<td>Deficient inspection plan caused detector deterioration without inspection.</td>
<td>1 (Circulating water system)</td>
<td>- Replacement.</td>
<td>- Re-evaluation of inspection frequency considering deterioration of the detector.</td>
</tr>
<tr>
<td></td>
<td>Signal cable was short-circuited by foreign material and malfunctioned.</td>
<td>1 (PLR)</td>
<td>- Repair of failed part.</td>
<td>- Strengthening of work management.</td>
</tr>
<tr>
<td></td>
<td>Insufficient tightening of screw in the control circuit.</td>
<td>1 (Condensate system)</td>
<td>- Repair of failed part.</td>
<td>- Strengthening of work management.</td>
</tr>
<tr>
<td>Location of Incidents/Failures</td>
<td>Outline of Cause</td>
<td>No. of Occurrence</td>
<td>Action Taken</td>
<td>Countermeasure and Evaluation</td>
</tr>
<tr>
<td>-------------------------------</td>
<td>------------------</td>
<td>------------------</td>
<td>--------------</td>
<td>------------------------------</td>
</tr>
<tr>
<td>Power source</td>
<td>Cable terminals of hydraulic pressure detector was short circuited by foreign material and malfunctioned due to deficient adjustment.</td>
<td>2 (PLR)</td>
<td>Repair of failed part.</td>
<td></td>
</tr>
<tr>
<td>Control oil system</td>
<td>Fatigue failure of lubrication oil pump gear coupling due to wear by insufficient lubrication.</td>
<td>1 (PLR)</td>
<td>Replacement of main shaft bearing.</td>
<td>Re-evaluation of oil pressure setting value.</td>
</tr>
<tr>
<td>Control circuit</td>
<td>Deficient card due to temporary performance change of the parts and in sufficient tightening.</td>
<td>2 (1 PLR) (1 RCP)</td>
<td>Replacement of failed parts, &amp; card.</td>
<td>Re-evaluation of inspection period.</td>
</tr>
<tr>
<td>Generator</td>
<td>Insufficient contact due to adhering of foreign material to the contactor of field magnet breaker.</td>
<td>1 (PLR)</td>
<td>Repair of failed part.</td>
<td>Strengthening work management.</td>
</tr>
</tbody>
</table>
| Control panel                 | Wearing of rotation control brush. | 2 (PLR) | Replacement of brush. | Re-evaluation of replacement period.  
- Replacement practice - to replace brush when the length of brush becomes shorter than specified value. |
<p>|                               | Leaked water from air conditioner flooded inside of the panel causing loss of control functions. | 1 (PLR) | Repair of failed part. | Shielding was newly provided between electric component and piping. |</p>
<table>
<thead>
<tr>
<th>Location of Incidents/Failures</th>
<th>Outline of Cause</th>
<th>No. of Occurrence</th>
<th>Action Taken</th>
<th>Countermeasure and Evaluation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motor</td>
<td>Insufficient soldering of power cable of the motor, insufficient tightening of terminal within terminal box and short circuit due to miswork.</td>
<td>5 (4 - PLR) (1 - RCP)</td>
<td>Repair of failed part.</td>
<td>Improvement of solder.</td>
</tr>
<tr>
<td>Power cable and terminal</td>
<td>Insufficient tightening of the piping connecting part of the cooler and high-cycle fatigue due to deficient welding (initiating cause) and resonance.</td>
<td>3 (PLR)</td>
<td>Repair of failed part.</td>
<td>Addition of anti-vibration support. Torque control for tightening.</td>
</tr>
<tr>
<td>Piping</td>
<td>Leakage of cooling oil due to insufficient tightening of packing of oil tank.</td>
<td>2 (1 - PLR) (1 - RCP)</td>
<td>Replacement of packing.</td>
<td>Torque control for tightening of packing.</td>
</tr>
</tbody>
</table>
Fig 3.2(a) Incidents/Failures relating to Pump Facility classified by Cause

Fig 3.2(b) Cause-related to Systems of Incidents /Failures relating Pump Facility
Fig 3.2(c) Trend of Causes of Incidents/Failures relating to Pump Facility

Fig 3.2(d) Comparison of Causes of Incidents/Failures relating to Pump Facility (1970's vs 1980's)
### Table 3.3 Effect to Plant Operation

<table>
<thead>
<tr>
<th>Effect to Plant Operation</th>
<th>Total Incidents/Failures (%)</th>
<th>Incidents/Failures relating to Pump Facility</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>No of Reports</td>
</tr>
<tr>
<td></td>
<td></td>
<td>(%)</td>
</tr>
<tr>
<td>Automatic Shutdown</td>
<td>35</td>
<td>16</td>
</tr>
<tr>
<td>Manual Shutdown</td>
<td>30</td>
<td>25</td>
</tr>
<tr>
<td>Power Reduction</td>
<td>1</td>
<td>3</td>
</tr>
<tr>
<td>No Effect</td>
<td>34</td>
<td>9</td>
</tr>
<tr>
<td>Total</td>
<td>100</td>
<td>53</td>
</tr>
</tbody>
</table>

![Diagram](image)

**Fig 3.3(a) Effect to plant Operation by Incidents /Failures of pump Facility**
Fig 3.3(b) System of Pump Facility affected to plant Operation

Fig 3.3(c) Operating Status of Plants when Incidents/Failures of Pump Facility occurred
Fig 3.4 Trend of Incidents/Failures for Plant Operating years

Fig 3.5 Trend of Incidents/Failures for Operating Months after Periodical Inspection
RELIABILITY OF STEAM TURBINE-DRIVEN STANDBY PUMPS
USED FOR SAFETY-RELATED APPLICATIONS IN
U.S. LIGHT WATER REACTOR COMMERCIAL
POWER GENERATING PLANTS

Prepared by: John R. Boardman

Presented by: Jack E. Rosenthal

U.S. Nuclear Regulatory Commission
Washington, D.C. 20555

ABSTRACT

In the USA, pump failure experience is collected by two primary means: 1) Licensee Event Reports, and 2) the Nuclear Plant Reliability Data System failure reports. Certain safety-related turbine-driven standby pumps were identified by these data systems as experiencing a significant ongoing repetitive failures of their turbine drivers, resulting in low reliability of the pump units. The root causes of identified failures were determined, and actions to preclude these repetitive failures were identified.
1.0 INTRODUCTION AND SCOPE

This study of safety-related standby turbines was initiated because of the continuing and repetitive operational failures of these turbine assemblies (turbines with their associated governors, valves, valve operators, overspeed trip mechanisms, circuit breakers, and fuses) used as drivers for safety-related standby pumps installed in LRW commercial power generating plants. In pressurized water reactor (PWR) plants, the turbine-driven pumps are used as an independent and redundant means of removing reactor core heat by providing Auxiliary Feed Water (AFW) to the steam generators in the event of the operational failure of the Main Feed Water (MFW) system. In Boiling Water Reactor (BWR) plants, these turbine-driven pumps are used in the High Pressure Cooling Injection (HPCI) and Reactor Core Isolation Cooling (RCIC) systems to provide automatically initiated redundant and independent sources of reactor grade water to assure reactor core heat removal under specified off-normal and accident conditions of plant operation when the MFW system is not available. The US regulatory requirements for these standby pump turbines are identified in Appendix A.

In the US, pump failure experience is collected by two primary means: 1) Licensee Event Reports (LERs), and 2) the Nuclear Plant Reliability Data System (NPRDS) failure reports. NPRDS failure reports relating to these turbine assemblies for the years 1985 through 1989 were analyzed, as were LER abstracts from 1969 through 1989, and all identified generic communications and studies issued by the US Nuclear Regulatory Commission (NRC), or by US industry. Appropriate personnel were contacted at selected plants to gain a better insight into specific failures that had been reported. The turbine and governor manufacturers were contacted to gain a better understanding of the components involved. The recurring problems identified were analyzed for additional root causes which may not have been addressed in generic communications.

2.0 SIGNIFICANT STANDBY PUMP-TURBINE CONFIGURATION DEFICIENCIES

Most safety-related standby turbine-driven pumps in US plants use steam turbines as their drivers. All such turbines use governors manufactured by the Woodward Governor Company. Figure 1 shows the schematic diagram of a typical Woodward mechanical/hydraulic governor used with standby turbines. Figures 2 and 3 show the control diagrams for HPCI and RCIC turbines used with BWR plants, which were provided by Terry-Turbodyne, the design agent for the components as noted.

The design of HPCI and RCIC turbines and their governor systems was accomplished by the Nuclear Steam System Supplier (NSSS). The design of AFW turbines and their governor systems was performed by the Architect-Engineer (AE) for each US plant, and appears to be essentially plant specific. The scope of design included designing the turbine and
governor for standby service in a cold shut-down condition. In addition, in order to meet their safety-related requirement supplying water to remove heat and maintain reactor core integrity, these turbine pump-drivers were designed to reach their required speeds of from 3000 to 6000 revolutions per minute in less than 30 seconds from the cold shut-down condition. This is called a "cold quick start" and is required of all safety-related standby turbines.

Some AFW turbines use the trip and throttle valve supplied by Terry as the steam stop valve. This design requires the trip and throttle valve to be closed at turbine startup, and at startup to rapidly open coordinated with the position of the governor valve to prevent an overspeed trip. Other AFW turbines have separate steam stop valves as shown in the RCIC control diagram, figure 3. This design permits the trip and throttle valve to be open at startup as designed by the turbine manufacturer. This configuration requires the opening speed of the turbine steam stop valve to be coordinated with governor valve position to prevent a turbine overspeed trip during a cold quick start.

Many turbines do not have pressurized lubricating oil systems to provide lubrication at startup, others have shaft-driven lube oil pumps which do not provide oil pressure until the turbine is operating. Only HPCI turbines have an auxiliary motor-driven lube oil pump to provide prelubrication at turbine startup. Safety-related standby turbines utilize turbine lubricating oil as the hydraulic operating fluid for their governors or valve actuators.

Many variations and modifications of Woodward governors are used with these turbines. Basic types include both mechanical/hydraulic governors and electric governors with an electric/mechanical/hydraulic actuator. Governor modifications appear to be plant, and pump/turbine, specific. Governors having the same model numbers may not be interchangeable per se because of optional parts and subassemblies which can significantly affect governor operation. Governor nameplates do not necessarily identify modifications made after a governor leaves the factory, nor will the factory know of such modifications when supplying replacement parts.

The physical location of AFW turbines varies widely between plants, resulting in different ambient temperature and humidity ranges with their environmental effects on turbine deterioration, maintenance, and operational performance.

Normal commercial applications for these turbines and their speed control governors involve continuous operation. Normal turbine startup is slow and preceded by proper warmup of the steam lines and turbine to minimize transients and wear. By contrast, these standby turbine-driven pumps in nuclear power plant applications are normally in a cold shutdown condition in areas where humidity is typically high, and temperatures can vary from high to almost 0 degrees Celsius depending on the specific plant. Leaking steam inlet valves can worsen the situation by contaminating the turbine lubricating oil with water. This oil is normally used as the hydraulic fluid for the governor and actuator for which a primary failure cause is oil
contaminated with water. These standby conditions can lead to accelerated deterioration of the turbine and governor that may not be identifiable until pump-turbine startup.

3.0 SAFETY-RELATED TURBINE-DRIVEN STANDBY PUMP OPERATING EXPERIENCE

Operating experience has shown that failure detection for these standby pumps is very sensitive to the way in which surveillance testing is carried out. The closer the testing mimics the cold quick start profile of an actual demand, the more likely it is to disclose situations where the dynamic response of the turbine-governor-valve combination is out of calibration, resulting in an overshoot and trip on overspeed. The current surveillance testing requirements have historically allowed a range of approaches, some of which do not fully test that dynamic response.

Certain plants have tested their safety-related standby turbines using cold quick starts. Other plants have used hot quick starts, which do not challenge the governor hydraulic system and turbine lube oil systems. Certain plants have used hot slow start surveillance tests. These tests preclude the turbine experiencing either failures during cold quick start transients, or accelerated bearing wear. Surveillance tests to determine pump operability that are initiated by hot starts do not reflect the safety-related operational requirements of standby turbines, however. In general, AFW turbines normally experience more cold quick starts than either HPCI or RCIC turbines do. Depending upon specific plant requirements, the number of cold quick starts has varied by a factor as great as 18, from one test start per month to one per eighteen month fuel cycle. This variance in test methodologies can result in a proportional variance in failure rates between plants.

Over time, more and more plants have begun using cold quick starts for routine surveillance testing as a result of industry and regulatory feedback. The changing mixture of hot-start and cold-start testing, as well as plant specific changes in maintenance resulting from operational feedback, makes it difficult to accurately quantify turbine reliability, or to trend failure frequency from NPRDS failure reports and LERs. Thus tables of failure frequencies or trend charts can be misleading. However, the continued receipt of reports of the same repetitive failures observed for a number of years, along with the prominence given this component in industry improvement efforts, are strong indications that problems continue to arise too frequently and that additional effort may be warranted. Standby turbine and governor failure trend data from this study are shown in figures 5 through 7.

For this study, 443 NPRDS failures of safety-related turbines and their associated components, such as governors, valves, valve operators, overspeed trip mechanisms, circuit breakers, and fuses were identified covering the period of January 1985 through December 1989. Also, 538 LER abstracts were reviewed for the period of 1969 through 1989. Selected completed LERs were retrieved and reviewed. The analysis of these failures confirmed the continuing validity of earlier studies by the US NRC and by US industry that
the most significant factors in failures of turbine-driven standby pumps have been the failures of the turbine drivers and their controls, especially during quick start transients. For example, a historical overview of failures through 1986 for the auxiliary feedwater system, US NRC NUREG/CR-5404, found that the standby turbine was a principal contributor to AFW system failures, and the turbine governor and controls were the principal contributor to the AFW turbine failures. The present study has extended the data review to include more recent failures, and has assembled an integrated and comprehensive compilation of the failure mechanisms or root causes of the reported failures, as well as remedies developed by the industry which should prove effective if implemented. Sections 3.1 through 3.2 discuss in detail causes of turbine failures, while section 4.0 discusses reliability growth that has been generated through the operational feedback improvement process.

3.1 STANDBY TURBINE FAILURES DIRECTLY RELATED TO COLD QUICK STARTS

Direct turbine failures during cold quick start transients appear to primarily result from failures of governor control. Root cause determinations during this study, as well other NRC studies, industry studies, and generic correspondence, are discussed in the following subsections. In addition, there are a significant number of cases of turbines trip during initial startup which failure reports stated could not be duplicated during subsequent startups, or for which the cause was left as unknown. The design review performed for this study indicates that such failures might be linked to the time required to build up operating pressure in a turbines' governor and actuator hydraulic system. Except for HPCI turbines which have an electric auxiliary oil pump (AOP), the hydraulic system pressure for governor valve speed control of the turbine drivers normally is provided by a shaft driven pump. The turbine must be rotating before mechanical/hydraulic governors, or hydraulic valve actuators for electric governors, begin to reach operating pressure. This time delay can affect the governor's control of turbine speed. Pump reliability during cold quick starts appears to be affected by such factors as ambient temperatures below 100 degrees Fahrenheit (38 degrees Celsius) and relatively viscous turbine lubricating oil above 150 Saybolt Seconds Universal (SSU), as well as oil contaminants.

3.1.1 MAINTENANCE RELATED FAILURES AFFECTING STANDBY TURBINE COLD QUICK STARTS

A significant number of standby turbine governor failures appeared to be maintenance related. These failures are capable of being eliminated with the effective implementation of plantspecific preventive maintenance (PM) programs. The failure mechanisms identified from failure reports during this study are listed below.
Operational failures of turbine and governor sub-components and piece-parts presently identified by the turbine manufacturer as covered by periodic preventive maintenance, for which this maintenance was not, or may not have been, performed completely with specified periodicities prior to the reported failure.

These failures range from the loss of turbine lube oil caused by a clogged filter, to an apparent failure to calibrate an electric governor ramp generator circuit causing an overspeed trip. A primary factor in plant specific failure to accomplish the turbine manufacturer's identified periodic preventive maintenance appeared to be the use by certain plants of early versions of the turbine manual which did not contain all preventive maintenance dictated by turbine operational experience. Some plants still use their original manufacturer's manuals. Certain manuals were published as early as 1969 and do not contain the presently identified preventive maintenance actions and periodicities.

Failures caused by the mispositioning of valves in turbine lubricating oil systems. Since these valves are part of a component, the pump-turbine assembly, typically they may not have check lists to verify their position. Additionally, they may have been positioned initially during plant startup in a partially open position to balance flow in the lube oil system. This required position may have been documented only on a test procedure which is now in archival storage.

Governor failures caused by the failure of instrument air used for remote turbine speed setting with mechanical/hydraulic governors, and by the simultaneous mispositioning of the governor high speed stop. The turbine manufacturer stated that the original design concept for these turbines did not include the use of the connection of instrument air to these turbine governors, though it is an optional feature of certain governors to permit remote setting of the turbine speed.

Apparent failures of sub-components and piece-parts as a result of aging phenomena. A recent example was the identified repetitive failures of voltage dropping resistors in electric governors. Certain electrical component piece-parts such as voltage dropping resistors, and aluminum electrolytic capacitors, as well as mechanical/hydraulic component piece-parts such as elastomeric seals, will exceed their design life span during plant life. Determination of required replacement intervals is the responsibility of the individual plant because of plant specific conditions affecting component aging. Only one plant was identified during this study that accomplishes extensive periodic replacement of component parts to prevent age-related governor system failures. This plant replaces their electric governors and actuators every eight years because of age related failures.
3.1.2 TURBINE COLD QUICK START FAILURES RELATED TO DESIGN AND QUALITY CONTROL

Several turbine governor failures were caused by the removal of required, but uncontrolled and unidentified, modifications to the governors. The past frequency of this failure mechanism per se does not raise a concern, as it appears to be low. However, this failure mechanism by its nature can affect many plants, and can lead to common mode failure for plants which have redundant turbine-driven standby pumps.

These failures were caused by plant-specific governor modifications which were not documented, or indicated on the governors' identification plates. The modifications were removed during governor refurbishment at the manufacturer's plant when the governor was restored to its "as-built" configuration based on the governors' identification plates and factory records. Plant design control and quality assurance organizations did not identify the removal of the modification when the governors were returned to the plant after refurbishment. Failures caused by the removal of the modification occurred when the plant changed its standby turbine-pump operability verification surveillance test methodology from hot start to cold quick start.

Significant operational failures have resulted from the apparent lack of coordination of the opening of the turbine steam inlet stop valve, or the turbine trip and throttle valve, with the required response time and movement of the governor valve to prevent turbine overspeed trip during turbine startup.

For RCIC and AFW turbines, the turbine manufacturer supplies the turbine trip and throttle and governor valves. Another design agent provides the turbine inlet steam stop valve, or uses the trip and throttle valve as the steam stop valve. One study identified that as many as 20 percent of turbine failures may be attributed to the improper opening speed of the trip and throttle valve, causing turbine overspeed before the governor valve can control steam flow. Originally, the trip and throttle valve was designed to remain open at all times unless it is tripped closed. Apparently other design agents have changed valve timing coordination between the stop valve, the trip and throttle valve, and the governor valve resulting in marginal coordination that at times not prevent turbine overspeed trips during quick cold starts.

3.2 TURBINE FAILURES INDIRECTLY RELATED TO COLD QUICK STARTS

Indirect failures are those that result from accelerated wear caused by standby turbine cold quick starts. An example appears to be the higher reported number of AFW pump turbine bearing failures. NPRDS failure reports show that AFW turbines experience more failures from bearing wear than do HPCI or RCIC turbines. The following identifiable factors indicate that the higher failure rate for AFW turbine bearings is to be expected:
Many AFW turbines are lubricated only by oil slinger rings, while RCIC turbines have a shaft-driven lube oil pump, and HPCI turbines have both a shaft-driven pump, and an electric auxiliary oil pump (AOP) that is started immediately after the turbine stop valve switch is placed in the "open" position, before the turbine rolls. AFW turbine slinger rings provide less oil to bearings during the initial turbine roll, increasing wear.

Because of differences in the methodology of performance of the periodic tests to determine standby pump-turbine operability, AFW turbines normally experience more cold quick starts than either HPCI or RCIC turbines do. Depending upon specific plant requirements, the number of cold quick starts has varied by a factor as great as 18, from one test per month to once per eighteen month fuel cycle. Bearing wear would be expected to vary approximately by the same factor.

HPCI and RCIC pumps are normally located in interior compartments, while certain turbine-driven AFW pumps are located out-of-doors. Such AFW turbines and pumps may see temperatures as low as 2 or 3 degrees Celsius. The fluidity of certain allowable grades of turbine lubricating oil could slow the rate of oil pressure buildup for turbines having shaft driven oil pumps, and reduce the lubrication ability of slinger rings.

4.0 STANDBY TURBINE/PUMP PERFORMANCE FEEDBACK BASED ON THE ANALYSIS OF OPERATIONAL FAILURE DATA

Since 1978, the US NRC using operational experience has issued 17 notices, circulars, and studies dealing with various problems affecting the reliability of safety-related standby turbines. Pertinent US Nuclear Regulatory Commission documents are listed in Appendix B. This study identified a similar number of documents issued by US industry organizations since 1980, which covered the history of operational failures of the subject pump-turbines. Certain industry documents identified specific methodologies to enhance the reliability of safety-related standby turbines.

The most extensive and focused source of operational data to enhance the reliability of standby turbines identified during this study were four General Electric Company Nuclear Service Information Letters (SILs) issues since 1980. These SILs contain design and operational changes that appear to significantly enhance the reliability of HPCI and RCIC standby turbine-driven pumps. These SILs were not mandatory, and their implementation in the field varies. These SILs addressed operational problems affecting HPCI and RCIC turbine reliability as follows:
GE SIL 336, dated 11 July 1980, emphasized the need to perform cold quick start tests of these standby turbines to duplicate actual operational demands for the turbines to start, since most failures occur during the cold quick start transient. The SIL provided guidance on the sequencing of the HPCI turbine stop valve and methodology for the performance of periodic surveillance tests for pump operability, including test instrumentation to enhance the assurance of pump and turbine operability.

GE SIL 352, dated 18 February 1981, identified operational problems resulting from the improper adjustment of the HPCI turbine steam stop valve's steam balance chamber which resulted in overspeed trips during the cold quick start transient. The SIL included a detailed procedure for the proper adjustment of this balance chamber.

GE SIL 377, dated June 1982, identified operational problems with RCIC turbine speed control during the cold quick start transient. It provided a design modification for a turbine steam by-pass line to significantly reduce this transient and overspeed trips.

GE SIL 480, dated 3 February 1989, identified a design change to the provide oil pressure from the HPCI turbine's electric auxiliary lubricating oil pump to the governor valve's model EGR hydraulic actuator prior to the turbine rolling to assure instant governor response during a cold quick start. This SIL also changed the type EG electric governor's turbine idle speed voltage setting. Figure 4 shows typical operational parameters during HPCI turbine cold quick start before and after the accomplishment of these changes.

GE SIL 337, Revision A, dated 8 December 1989, updated the original revision, incorporating additional operational data. Enhanced procedures were included for the assurance of proper turbine operation and test.

Other design features that addressed by GE include such potential turbine failure mechanisms as 1) the maximum allowable particle size when using turbine lube oil as a hydraulic fluid for mechanical/hydraulic governors and actuators, and 2) fluctuations in the reference voltage of electric governors.

The US NRC is reviewing a new MAINTENANCE GUIDANCE DOCUMENT, NP-6909, titled TERRY TURBINE CONTROLS, which was issued in late October 1990 by the Electric Power Research Institute (EPRI). This document provides manufacturers’ data on the preventive maintenance of standby turbines and their governors that generally has not been available at specific plants to support the maintenance of these turbines and governors.
5.0 SUMMARY AND CONCLUSIONS

While failures of safety-related standby turbine-driven pumps continue to occur, analysis of existing operational experience indicates that the majority of failures have been repetitive in nature. Elimination of the causes of these failures, with the resultant improvement in the reliability of standby turbines, appears to be achievable by enhanced industry-wide implementation for these turbines of presently invoked requirements, such as more rigorous design control and dedication of commercial grade items used in safety-related applications, and by the accomplishment of actions specifically relating to these standby turbines based on failure report root cause determinations such as:

- Implementation of the design and test changes contained in the GE SILs for all applicable HPCI and RCIC standby turbines. Some of the design and test enhancements in SILs address design and test weaknesses in AFW turbine.

- Implementation of the latest applicable Terry-Turbodyne identified maintenance for the standby turbines, and the maintenance guidance contained in EPRI NP-6909 for the turbine governors. Applicable plant specific design features relating to these turbines and their governors need to be identified, maintained, and incorporated in the appropriate plants procedures.

- Replacement of component piece-parts which affect plant safety before the end of their design life to prevent failures.

- Verification that the opening of turbine steam inlet stop valves is coordinated with the opening of their associated turbine trip and throttle valve and governor valve based on the criteria of the turbine manufacturer, who supplied the trip and throttle, and governor valves.

- For plants which use the AFW turbine trip and throttle valve as the turbine steam inlet stop valve, assurance of the proper coordination of its opening with governor valve response during cold fast starts.

- Verification that the extremes of all variables, such as internal governor and model EGR actuator hydraulic fluid temperature and viscosity at the initiation of startup, and of variations in electric governor reference voltages resulting from maximum and minimum station battery voltage, are included in the design time responses for standby turbine cold quick start transients, as well as continuous turbine operation.
APPENDIX A

REGULATORY REQUIREMENTS RELATED TO STANDBY TURBINE-DRIVEN PUMP DESIGN AND PERFORMANCE

The present regulatory requirements for these standby pumps applicable to plants being licensed are contained in sections of the U. S. Nuclear Regulatory Commission Standard Review Plan for the Review of Safety Analysis Reports for Nuclear Power Plants (NUREG-0800) as described below. Since these requirements have evolved based on operational experience, certain earlier plants may have somewhat different design criteria.

- **Section 10.4.9, AUXILIARY FEEDWATER SYSTEM (PWR)** - Requires that when a turbine-driven AFW system pump is installed, it must be of sufficient capacity to remove reactor decay heat during normal operating conditions, and following any accident or transient. Pump capacity must assure cooling the plant to the temperature at which the decay heat removal system is designed to remove the reactor core's residual heat. Pump-turbine reliability must support an overall system unreliability of E-4 to E-5 per demand based on an analysis using methods and data presented in US NRC NUREG-0611 and NUREG-0635. Individual plant technical specifications define surveillance tests to assure continued reliability of the pumps during plant operation.

- **Section 5.4.6, REACTOR CORE ISOLATION COOLING SYSTEM (BWR)** - Requires a steam-driven turbine-pump unit designed to have an extremely high probability of performing its safety function in the event of anticipated operational occurrences. These operational occurrences include reactor coolant system makeup for protection against small breaks in the reactor coolant boundary, removal of fission product decay heat and other heat from the reactor core to preclude fuel damage or reactor coolant pressure boundary overpressurization. The RCIC system functions in conjunction with the High Pressure Coolant Injection (HPCI) or High Pressure Core Spray (HPCS) systems, with the safety/relief valves, and with the Residual Heat Removal (RHR) system in response to certain off-normal and faulted plant conditions and to accidents. Plant technical specifications define surveillance tests to assure the continued reliability of these standby turbine-driven pumps during plant operation.

- **Section 6.3, EMERGENCY CORE COOLING SYSTEM** - Contains the design criteria for the HPCI system that is installed in certain BWR plants, including HPCI standby turbine-driven pumps. The design criteria are performance-based in terms of preventing reactor core damage, and do not address specific requirements for the
turbine-driven pumps. Later design BWR plants use a High Pressure Core Spray (HPCS) system with an electric pump powered by a standby diesel powered generator, in lieu of a HPCI system. Plant technical specifications include surveillance tests intended to assure the continuing operability of HPCI turbine-driven standby pumps during plant operation.
APPENDIX B

The following US Nuclear Regulatory Commission Studies, Information Notices (IN) and Circular (IEC) were reviewed during this study.

**CASE STUDY REPORT AEOD/C602.**  "OPERATIONAL EXPERIENCE INVOLVING TURBINE OVERSPEED TRIPS"

**NUREG/CR-5404, Volume 1:**  AUXILIARY FEEDWATER SYSTEM AGING STUDY

**INFORMATION NOTICE IN 88-09:**  REDUCED RELIABILITY OF STEAM-DRIVEN AUXILIARY FEEDWATER PUMPS CAUSED BY INSTABILITY OF WOODWARD PG-PL TYPE GOVERNORS

This IN covers a series of interrelated problems at Calvert Cliffs involving speed control of their AFW pumps. The causal factors in these problems were identified as:

- Use of replacement governors with incorrect buffer springs. The correct buffer springs are required to dampen out turbine speed oscillations. The installed springs were of less than the degree of stiffness that was originally specified by Woodward for this installation and resulted in unacceptable oscillations.

- Degraded and improperly adjusted governor linkage, resulting in excessive tolerances.

- Binding of the governor valve stem-disc assembly, resulting in improper control of turbine speed control.

- Damaged and misaligned turbine overspeed trip linkage and mechanisms, which resulted in an over sensitivity to tripping when disturbed by vibration, jarring, or water hammer in adjacent piping.

- A failed governor.

- Excessive condensate in the turbine steam supply line when the turbine was started. The condensate resulted in damage to the governor valve and the governor linkage, and in loss of turbine speed control as condensate impinged upon the governor valve and the turbine wheel.
INFORMATION NOTICE IN 81-24: AUXILIARY FEED PUMP TURBINE BEARING FAILURES

This IN identifies damage to Terry Turbine type GS-2 auxiliary feed pump turbine bearings as a result of the failure to maintain turbine lube oil levels within the required operating band. The involved turbines used oil pickup rings only. Type GS-2, and GS-1, turbines have been built using a combination of force feed lubrication and oil pickup rings. Maintenance of proper oil level is required for both oil system designs.

INFORMATION NOTICE IN 81-36: REPLACEMENT DIAPHRAGMS FOR ROBERTSHAW VALVE (MODEL NO. VC-210)

This notice identified a failure of a neoprene diaphragm in the ROBERTSHAW diaphragm control valve installed in the mechanical-hydraulic overspeed complex of a Terry Turbine used with a HPCI pump. Because of the lube oil used in such standby pumps, a non-standard, fabric reinforced diaphragm is required for this application. Replacement valves will have the incorrect diaphragm. The correct diaphragm must be ordered separately and installed by the end-user.

INFORMATION NOTICE IN 84-66: UNDETECTED UNAVAILABILITY OF THE TURBINE-DRIVEN AUXILIARY FEEDWATER PUMP

This IN identified five events at operating reactors during 1982-1983 where standby turbine driven AFW pumps were inoperable because the steam supplies were isolated. The isolations were caused by a failure to return AFW pump turbines to operability after the performance of surveillance or maintenance, or to verify operability after work had been performed in the area of the turbine control systems.

INFORMATION NOTICE IN 86-14: PWR AUXILIARY FEEDWATER PUMP TURBINE CONTROL PROBLEMS

This IN dealt with four overspeed trip events during AFW pump turbine startup, or restart. Two of the trips resulted from residual oil pressure in the governor control oil system prior to turbine restart. Two trips resulted from condensate in the turbine steam supply lines at startup causing loss of speed control.
INFORMATION NOTICE 86-14, SUPPLEMENT 1: OVERSPEED TRIPS OF AFW, HPCI, AND RCIC TURBINES

This IN dealt with four generic standby turbine speed control problems which result in overspeed trip. This IN was based on AEOD study AEOD/C602, "Operational Experience Involving Turbine Overspeed Trips." The four speed control problems were:

- Slow response of the governor during quick startup (including binding of the governor valve stem-disc assembly).
- Entrapped oil in the governor speed setting cylinder.
- Incorrect governor setting.
- Water Induction into the turbine (condensate in the steam supply line at startup).

INFORMATION NOTICE 87-53: AUXILIARY FEEDWATER PUMP TRIPS RESULTING FROM LOW SUCTION PRESSURE

This IN deals with AFW pump trips resulting from low AFW pump suction pressure caused by the simultaneous starting of two AFW pumps.

INFORMATION NOTICE IN 88-67: PWR AUXILIARY FEEDWATER PUMP TURBINE OVERSPEED TRIP FAILURE

This IN deals with a failure of a standby turbine for an AFW pump to trip on overspeed. The overspeed trip failure was caused by an excessively worn polyurethane tappet ball in the turbine overspeed trip system linkage.

INFORMATION NOTICE IN 89-14: INADEQUATE DEDICATION PROCESS FOR COMMERCIAL GRADE COMPONENTS WHICH COULD LEAD TO COMMON MODE FAILURE OF A SAFETY SYSTEM

This IN deals with the inadequate 10 CFR 21 dedication by a licensee of packing for an AFW pump.
INFORMATION NOTICE IN 90-45: OVERSPEED OF THE TURBINE-DRIVEN AUXILIARY FEEDWATER PUMPS AND OVERPRESSURIZATION OF THE ASSOCIATED PIPING SYSTEMS

This IN deals with the combined failures of the turbine governor and overspeed trip mechanism, with resulting turbine overspeed and overpressurization of the AFW system.

INFORMATION NOTICE IN 90-51: FAILURES OF VOLTAGE DROPPING RESISTORS IN THE POWER SUPPLY CIRCUITRY OF ELECTRIC GOVERNOR SYSTEMS

This IN deals with the affect on governor operability of the failure or degradation of voltage dropping resistors used in the power supplies of electric governors used for the speed control of standby turbines and emergency diesel generators.

NRC CIRCULAR 78-02: PROPER LUBRICATING OIL FOR TERRY TURBINES

This circular deals with the use of lubricating oils containing vapor phase inhibitors in Terry turbines used for pump-drivers for AFW/EFW, RCIC and HPCI standby pumps to prevent excessive rusting of the turbine interiors.
HPCI System

**FIGURE 3**
Before Modification

- GOVERNOR VALVE POSITION
  - Open

- Volts
  - EGM OUTPUT

- Volts
  - RGSC* OUTPUT

- Open
  - STOP VALVE POSITION
  - Closed

- RPM
  - SPEED

**FIGURE 4**
After Modification

- GOVERNOR VALVE POSITION
  - Open

- Volts
  - EGM OUTPUT

- Volts
  - RGSC* OUTPUT

- Open
  - STOP VALVE POSITION
  - Closed

- RPM
  - SPEED

* Ramp Generator Signal Converter

**FIGURE 4**
RCIC PUMP TURBINE/VALVES/OPS/CBKR
Composite Failure Rate Trend

Failures/Million Component Calendar Hrs.

Failure Discovery Date (Month/Year)

Trend
AFW PUMP TURBINE/VALVES/OPS/CBK
Composite Failure Rate Trend

Failures/Million Component Calendar Hrs.

Trend

Failure Discovery Date (Month/Year)
OPERATING FEEDBACK ON PUMPS IN FRENCH PWR NUCLEAR POWER PLANTS

R. LARUE
EDF - NUCLEAR AND FOSSIL GENERATION
MAINTENANCE DEPARTMENT

*****

SPECIALIST MEETING ON
PUMP PERFORMANCE AND RELIABILITY

COLOGNE - GERMANY

26th - 28th NOVEMBER 1990

OECO - NUCLEAR ENERGY AGENCY

09/11/1990
FEEDBACK ON PUMPS IN FRENCH
PWR NUCLEAR POWER PLANTS

INTRODUCTION

In order to improve the safety and reliability of all PWR nuclear power plants in France, EDF has taken a number of successive measures since start-up of the first units.

Certain of these measures are concerned with operation, and others with maintenance. In this article, the author reviews the general organization setup for the pumping units, and the special measures adopted by the Rotating Machines Section of the Maintenance Department, to achieve this objective.

This article does not take account of measures applied by the Engineering and Construction Division and the power plant builders, occurring ahead of the operational phase, but which naturally contribute also to the final results.

I. FEEDBACK PARTNERS

A well designed, manufactured and installed pumping unit, which is operated correctly and adequately maintained, would leave no trace in feedback records.

Nevertheless, as in any other industrial company, we unfortunately encounter a number of incidents on pumping units each year.

These incidents, which are minor from the point of view of safety (so far all have been classified at a level not exceeding 1, on a 6-level scale), must nevertheless be taken properly into account in order:

- to maintain or improve the safety and reliability of these components,
- to reduce non-availability and maintenance costs.

The measures adopted for effective implementation of this feedback must also take account of scale effects. It should be remembered that the French PWR facility operated by EDF comprises some 50 power plants, with more than 200 pumping units per plant, corresponding to a total exceeding 10,000 pumping units.
Incidents encountered on these components are of various origins, ranging from design errors to defective reassembly, and including defective manufacture, operating and assembly errors, and inadequate operating, inspection and maintenance procedures.

The partners of the Machinery Section of the Nuclear and Fossil Generation Division Maintenance Department are numerous:

- manufacturers, including their design departments, production facilities and subcontractors,

- the Design Department for Thermal and Nuclear Projects and the regional authorities of the Engineering and Construction Division, which monitor the design, manufacture, installation and commissioning of the pumping units,

- the operators and their various departments (operation, maintenance and technical control),

- the Operations and Nuclear Safety Departments, which establish doctrines at national level, and analyze incidents in their specific fields.

Alongside these direct partners, we must also include the Research and Development Group, the Chemical and Metalurgical Laboratories and Technical Support Unit of the Nuclear and Fossil Generation Division, and the French Safety Authorities, each of which can also serve as an interlocutor, in its specific area, for the engineers of the Machinery Section.

Generally, the origin of an incident or deficiency (whether operational or relating to maintenance) has a number of causes. This aspect requires the development of both functional and organizational synergy, in order to solve the problems encountered.

II. RESOURCES APPLIED TO IMPROVE THE FEEDBACK PROCESS: STRENGTHS AND WEAKNESSES

2.1 Preventive maintenance basic programmes and doctrines

Various resources have been applied to meet the objectives indicated above. Firstly, we should mention the drafting of a Maintenance Doctrine, and a set of Basic Preventive Maintenance Programmes. These documents, prepared by the Maintenance Department, have a number of different purposes.
- Improved safety and reliability of the installations: this aspect should in fact be considered first. The actual content of the maintenance programmes, and the maintenance operations which lead to an increase in the safety and reliability of the components are built on the solid base of doctrine.

The expression "hard core" is frequently used in our company to refer to doctrine, and this clearly demonstrates that this is the base, or the foundation of maintenance actions.

In the same way as for plant design and control actions, it was necessary to define a set of true notions on the basis of which EDF could orient or direct maintenance actions, for the purpose of improving the safety and reliability of the equipment.

Obviously this doctrine can evolve in time, not in order to challenge those notions said to be "true" (although this can happen!), but to complete the existing notions with others, not included in the initial draft for a number of reasons: insufficient knowledge of degradation phenomena, changes in control techniques, etc.

- Identical concept for the maintenance of a component: this concept must be independent from the geographical location of the unit and, all things being equal, independent from operation of the component, already made uniform by prior actions (e.g. common set of operating procedures). Indeed, it is easy to accept that a component, the design, manufacture and operation of which are identical to those of another unit, is covered by the same maintenance doctrine whether it is physically located to the north or south of the Loire!

- Theoretical integration of "global facility effect": identification of a non conformance on a component during inspection in a unit, allows rapid reporting of the non-conformance to the Maintenance Department, which then analyzes the anomaly and initiates corresponding feedback to the other power plants. This is the same method as used in the aviation industry (AIRBUS INDUSTRIE or BOEING), and in the relations between licensor and licensee (e.g. WESTINGHOUSE and JEUMONT SCHNEIDER), and is also the method used by our Safety Authorities, via their site inspection teams and Regional Industry and Research Divisions.

- Planning of maintenance operations: apart from organizational aspects, these doctrines and programmes are used for initial definition of workloads in terms of budget, human resources, dedicated tooling, etc., and consequently to achieve improved rationalization of
maintenance resources, from both the financial and logistic points of view.

Other aspects can emerge from these resources: safety of maintenance personnel, personnel training, simplification of procedures, etc. We shall not consider these aspects in the present article, as they in fact stem from those already mentioned.

These positive aspects are counterbalanced by a certain number of drawbacks (advantages by drawbacks, and vice versa!).

The first drawback stems from the premise of operational identity, valid in theory, but clearly influenced by the human factor in practice. Beyond the singleness of design, procedures and equipment, we have the men who operate the plant and carry out maintenance tasks, all of them essential factors for safety.

The example described in Chapter 3, concerning reactor coolant pumps, demonstrates a number of aspects of this apparent weakness.

Apparent but nevertheless real: each of us, whatever position we hold in the organization, is first and foremost an active participant, required to take daily decisions concerning changes or observed situations which are not fully covered by procedures, standing instructions or rules.

Thus in the area of maintenance, and in our opinion more so than in the area of plant control, decision-making is subject to discussion, for example as to whether a fault state detected on a machine can be left as is, in the certainty that it will not have an adverse effect on safety. Sometimes on the basis of calculated, although not fully validated non-criticality, and frequently according to the "intuition" of the engineer, based on comparison with other more or less similar cases encountered, we agree to reintegrate components showing some defects, at the more or less systematic price of enhanced future inspection.

This practice is not particular to our industry, and can be found all around us. For example no aircraft, after several thousand flying hours, is entirely free from a number of faults left uncorrected. Nevertheless, we do not consider that we are prejudicing the safety factor in this way.

The decisions which are taken, although subjective and not totally dictated by established physical laws, are accepted by all. However, these decisions must have a
limit, and the entire problem lies in the definition of this limit.

Unless we use mathematical models to achieve full definition of a degradation phenomenon, we are obliged to admit an uncertainty factor, based on the reasoning and intuition of the engineer, consequently affecting the safety factor although to an undeniably low degree.

The second aspect relates to the fact that one machine is not operated in exactly the same way as another. Operated at the limits authorized by established procedures, one machine will ultimately perform, in all probability, differently from another machine which has been operated in the median area of its tolerance range. We can thus assert, without however being able to provide sure, irrefutable proof, that component faults observed on one machine, and not present on another, are the result of the actual operating method, all other things being equal. The trivial example of the motor car is, nevertheless, an example accepted by all. The difficulty of this aspect lies in the identification of actual operating conditions, for the purpose of explaining the degradation observed. How can we establish the true temperature, flow-rate and other values actually experienced by the defective component, after 70,000 or 80,000 hours in service? Very often the values recorded in the control room, on the weekly chart recorder output, require several months (or even years) research work on the part of the operator. Must we nevertheless guarantee the traceability of all parameters, using faster and more certain methods? The answer to this question depends on both the following considerations:

- cost related to safety,
  - cost related to the type of maintenance adopted.

The third aspect which we shall consider relates to operating data feedback. To be totally effective, in addition to the advantages resulting from the effect of standardization, this requires establishment of a continuous stream of data between operator and central departments.

This same stream should also provide feedback of analytical results from the central departments to the operator.

Obviously this vector must be accompanied by others between the central departments and the industrial architect (EDF Engineering and Construction Division), and the plant builders and manufacturers.
Furthermore, as for any vector, in addition to its directions, the "intensity" factor must be both complete and reliable.

In objective terms, what is the situation in 1990?

- Firstly, this stream exists, although it can take a number of different itineraries of variable rapidity. Certain streams do not even reach their destination! We must therefore reformalize the stages, objectives and responsibilities involved.

- The quality of the information ("intensity" factor) received by the central departments, and consequently passed on, after analysis, to all sites, is dependent on efficient "first level" analysis.

In our opinion, this objective is not achieved to a sufficient degree. Here again, objectives must be clearly defined, and more appropriate resources provided.

These two essential points are currently the subject of reflection by the main authorities involved at EDF, and we believe that it will be possible in the future to improve the level of feedback quality, already recognized as good, and consequently improve plant safety and availability, as a result of the options identified by this reflection.

III. EXAMPLE: REACTOR COOLANT PUMPS

Although not intrinsically considered as safety equipment, the reactor coolant pumps nevertheless represent a component of the primary circuit, and consequently contribute, if only in containment system terms, to the integrity of the circuit, and finally to plant safety.

From the point of view of maintenance, the Basic Preventive Maintenance Programme (BPMP) stipulates the following, in particular:

- inspection of n° 2 and n° 3 seals every 2 years,
- inspection of n° 1 seals every 3 years,
- inspection of the pump bearing block every 6 years.

This programme was established in 1984, and was based on the Operating and Maintenance Guide published by the manufacturer, JEUMONT SCHNEIDER INDUSTRIE, a WESTINGHOUSE licensee for type 93D pumps (used for the 900 MW PWR programme), and type 100 pumps (1,300 MW PWR programme).

The BPMP programme was also drafted according to the existing knowledge of EDF specialists. The only
references available on this subject at the time (1984) were as follows:

- similar equipment installed in conventional power plants (vertical pumps with one bearing block, fitted with mechanical seals, etc.). This equipment was somewhat remote from the design of the primary circuit pump, and its method of operation was also different,

- operating and maintenance results on certain US utilities. To acquire this data base, an EDF engineer, who was also a very high level technician, was sent to the USA to establish operational feedback in this particular area.

This maintenance programme can be considered, in its temporal context, as the most likely to meet the potential difficulties envisaged at the time. What in fact is the reality of the matter?

The answer is to be found in the feedback data since analyzed by the Maintenance Department of the Nuclear and Fossil Generation Division. The Maintenance Department indicates the following aspects:

3.1 Negative aspects

The specialists at the time had not "envisaged" the following forms of degradation:

- cracking of shafts due to thermal cycles induced by injection flow-rates, and primary circuit water flow-rates, on the thermal rings at the lower end of the shaft.

This aspect, although apparently perceived by the WESTINGHOUSE designers (Westinghouse jacketed the pump shafts after several thousand hours testing and in service), was not taken into account either by the licensor, the licensee (JEUMONT SCHNEIDER INDUSTRIE) or the EDF industrial architect (Design Department for Thermal and Nuclear Projects). So far we cannot certify that operational feedback concerning French pumps is effectively taken into account by the foreign production units using the same type of pumping unit.

In defence of the designer, our manufacturer and our industrial architect, it should be pointed out that it is currently extremely difficult to model the phenomenon comprehensively. We are faced here with a technological limitation on the mathematical representation of a physical phenomenon. This being so, from the safety point of view, we are able to detect critical cracking of the pump shafts by means of quasi-continuous monitoring of vibrations, and by the measures taken to start up the pump
units at a very low global vibration level (< 100 um). It should be remembered at this point, that global shaft displacement levels recorded in the case of reactor coolant pumps shaft ruptures, were in excess of 500 um, or five times greater than the currently recommended figure.

- Cracking of thermal barrier labyrinth seals: for reasons identical to those concerning the cracking of pump shafts (thermal shocks generated by the cold water/hot water interface), we also observe labyrinth seal cracking phenomena. In this case, given the static aspect of the part, and on the basis of propagation and criticality calculations, we propose leaving the defect uncorrected.

Here again, for reasons similar to those for shaft cracking, there is nothing which enables us to anticipate this type of degradation, and to the best of our knowledge, this phenomenon is neither described nor predicted by any of the participants concerned (designer, manufacturer or industrial architect).

- Impeller cavitation: discovered very recently on an impeller in service for more than 80,000 hours (August 1990), and as we have not so far completed all investigations required to explain this phenomenon, it is merely mentioned in support of our demonstration. Nevertheless, here again, we can claim with quasi-certainty that no precursor element came to the notice of the operator, which could have made it possible to overcome these difficulties, subject to appropriate maintenance action.

3.2 Positive aspects

After the negative aspects reviewed above, we must now mention the positive aspects induced by the Basic Preventive Maintenance Programmes.

- Improved operation of n° 1, n° 2 and n° 3 seals.

With the backing of the operational feedback system set up by the Nuclear and Fossil Generation Division on initial start-up of the power plants, EDF drafted a set of recommendations relating to seal maintenance and operating actions, back in 1986.

This document, prepared conjointly by the Maintenance Department and Operations Department, was extremely well received by all units. Four years after its publication we can observe its effect: no cases of random non-availability, intrinsically due to the n° 1, n° 2 and n° 3 seals, have been recorded. Furthermore, these analyses of data concerning the 900 MW PWR power plants, were immediately adopted for their 13,00 MW counterparts with,
at least in this area, a highly beneficial result. No cases of non-availability due to the seals have been encountered, any more than for the 900 MW power plants. This positive aspect, from both safety and availability points of view, must nevertheless be counterbalanced by the question as to whether our preventive maintenance policy is not "too rich" and too conservative. This question is currently being examined, and we believe that we shall be able to alleviate constraints to some extent, in terms of maintenance and consequently of cost, in order to obtain the sought-after optimum, while maintaining the same degree of reliability in terms of safety.

- Improved efficiency of casing seals: feedback for this component quickly indicated the essential efforts required. This work (modification of tightening torque and manufacturing improvements) enabled us to clear this problem in less than 3 years, for the total of 102 type 93D pumps, and to have had no further resurgence of this problem so far. These improvements led to simplification of the basic maintenance programme, and obviously reduced any impairment of safety by this component to a quasi-zero probability factor.

- Improvement of motorized pump unit vibration performance: up to 1983-84, only alarm (250 um) and shut-down (380 um) values were specified for operation of this equipment. Very frequently the units were operated between these two values. Without representing an intrinsic safety problem, it is obvious that such performance is prejudicial to the effective life time of pump components (seals, bearing blocks, balls, etc.) in the medium or long term. Actions aimed at improved personnel awareness of the problem, associated with the drafting of balancing procedures, and the development of frequency status monitoring procedures, made it possible to reduce the vibratory state of the majority of pumping units, after inspection, to a value below 100 um. Apart from this extremely important improvement with respect to pumping unit behaviour, we were then able to achieve a comfortable margin with respect to shut-down level (380 um), thus giving sufficient time to analyze any possible drift before reaching this level.

Furthermore, this margin enables us to state that any shaft cracking phenomenon which might develop outside current hypotheses, would necessarily be detected by the substantial increase in vibration level with respect to initial level. All shaft ruptures experienced so far have shown levels exceeding 500 um, with a gradient exceeding 100 um/day as from the current level.

The list of negative and positive aspects described above makes no claim to be exhaustive. It is merely
intended to serve as an example, in support of the preceding demonstration.

IV. CONCLUSIONS

As with any procedure in the area of human organization, that set up by EDF for pumping units possesses the "drawbacks of its advantages" and vice versa. Furthermore, a high degree of balance is always difficult to achieve. The cost/profit ratio of any organization of this kind is never really quantifiable. Nevertheless, reliability levels with respect to safety are highly commendable, and are improving each day, each year and each decade. This aspect would justify maintaining the existing organization on its own. In our opinion it is not sufficient to achieve this, and it is essential to develop an increasing degree of ingenuity, to ensure that we do not "go to sleep on the job" in the medium and long term.

These resources, which depend as much on organization, function, discipline, etc., must be reinvented, and implemented in our daily activities.
SESSION #2
Chairman's Closing Remarks
(Mr. M. Maris, Vinçotte, Belgium)

1. "Pump performance and reliability follow-up by the French safety authorities", presented by Mr. J.P. Clausner

A description is given of the methodology of the evaluation of the performance and reliability of safety related pumps, as applied by the French safety authorities. The lessons learned from this evaluation are also presented.

The approach is characterized by two main lines:

- surveillance of the functional capabilities and
- surveillance of the equipment status through testing and maintenance.

The authors make a distinction between pumps used in normal operation on one hand and pumps in standby on the other. Examples of both categories are given:

- primary pumps (seals, studs), auxiliary feedwater system pumps (inadvertent trips, coupling problems), containment spray pumps (thermal dilatation impairing the coupling), auxiliary feedwater system pumps in incidental condition testing (black-out, inducing lubrication problems and bearing damage).

Comprehensive requalification testing after a maintenance operation (e.g. safety injection system) is stressed.

The authors conclude that, although maintenance and surveillance test programmes are in general efficient, additional actions such as qualification tests representative of accidental conditions, endurance tests performed on site, may contribute to an increased confidence in the reliability of pumps.

2. "Trend of incidents and failures of pumps in Japanese nuclear power plants", presented by Mr. M. Bada.

The paper presents the statistical evaluation of the events related to pumps in Japanese light water reactors for the period 1970-1989. A total of 53 cases of incidents and failures have been reported. The analysis of these cases, categorized in four functional blocks (mechanical part, motor, power source, controls), addresses the following aspects:

- classification of the events on a functional basis and on a component basis;
- cause analysis, identifying deficiencies in design, manufacturing, installation, maintenance, maintenance management;
- effect on plant operation;
- effect of the number of years of operation on incidents or failures;
- effect of the number of operating months following periodic inspection.
The authors conclude that the number of event reports on pumps is continuously decreasing for the past years, and consider that this is due to improvement in maintenance, domestic experience feedback, complete execution of counter measures and evaluation of the applicability of overseas events.

3. "Reliability of steam-driven standby pumps used for safety-related applications in U.S. light water reactor commercial power generating plants", presented by Mr. J. Rosenthal

Referring to the U.S. data on pump failures, the author indicates that no clear composite failure rate trend can be identified. The paper therefore focuses on the continuing problems with standby turbine-driven pumps in U.S. boiling water and pressurized water nuclear power plants. The question "Can the hardware meet the design requirements?" was illustrated by examples, more particularly by the example of the cold quick start requirement. In the latter case, problems with the lubrication and control oil system have been reported resulting from the periodic tests which are carried out in the same design conditions.

The paper presents, next to useful references, the measures to be taken that, once implemented, should improve the reliability of the turbine-driven pumps, which present, for the time being, a reliability which is of an order of magnitude inferior to their motor-driven counterparts.

4. "Operating feedback on pumps in French PWR nuclear power plants". presented by Mr. R. Larue

The various organizations taking part in the feedback process are presented: manufacturers, SEPTEN (design department for thermal and nuclear projects), the operators, the operations and nuclear safety departments. The goals of the return of experience are

- to maintain or improve the safety and reliability of the components;
- to reduce the non-availability and the maintenance costs.

The author mentions the establishment of a preventive maintenance policy and the implementation of a maintenance programme, aiming at improving the feedback process. He then discusses positive aspects and weaknesses of this approach.

Positive points are the identical concept for maintenance for a component, "global facility effect" (quick feedback of a non-conformance to all plants concerned) and maintenance planning inducing rationalization of resources.

As drawbacks, the author mentions the human aspect (devalitating the premise of operational identity), different operation conditions of identical machines and the deficiencies in the operational data feedback (insufficient rapidity, quality of the information).

The example of reactor coolant pumps has been used to illustrate some of the advantages and weaknesses of the present operational feedback system discussed before.

The author finally recommends reliability levels with respect to safety and underlines permanent alertness.
During the discussion period, much attention was given to the vibration levels and setpoints for primary pumps. Next to these vibration levels, for which no general absolute values can be adapted, displacement monitoring and observation of double speed frequency for early detection of shaft cracking were mentioned.

5. "Incidents attributed to pump problems", presented by G. Ishack

The author made an overview of the IRS reports between January 1980 and June 1990. From these 1500 reports 191 reports related to pump problems were further analyzed. Most of the reports came from NEA members. The reports were grouped according to three reactor types (PWR, BWR, PHWR) and further separated into events caused by faults in design, operating procedures and inspection strategy.

Finally the events in primary pumps were highlighted, as well as the causes and the corrective actions.

For PWR seal failures, shaft problems (cracks, leaks, impeller separation), bolt corrosions, were found to be recurring problems.

A much smaller number of reports concerned BWR and PHWR.

As a conclusion, the corrective actions common to all reports were highlighted: adequate vibration monitoring, response to high vibration levels, and QA in manufacturing.

For PWR, the following corrective actions were identified: adequate seal and shaft inspection, seal reliability, seal cooling, installation, bolt corrosion resistance, fatigue resistant materials.
SESSION #3: "PUMP DESIGN"

Chairman: W. Minners

* "Turbulent flow analysis of inlet and exit flows of internal pumps installed in an internal pump plant vessel", T. Okamura, T. Takagi, Y. Yoshimoto & H. Utsuno, Hitachi Ltd., Japan (presented by T. Okamura)

* "NPSH requirements of large PWR boiler feed pumps at part load pumping conditions", F. Schubert, Siemens, Germany

* "Need for rotor dynamics considerations in design of vertical pumps for reliable operation", A. N. Kumar, AECL, Canada

* "Construction and qualification experience on the RRA and ISMP pumps of the French nuclear reactors", C. Mech, GEC ALSTHOM, France

* "Cavitation characteristics study of a nuclear reactor internal pump", H. Komita, I. Ohshima, S. Fukuda & K. Suga, Toshiba Corporation, Japan (presented by H. Komita)
Turbulent Flow Analysis of Inlet and Exit Flows of Internal Pumps Installed in an Internal Pump Plant Vessel

Tomoyoshi OKAMURA, Takeo TAKAGI
Mechanical Engineering Research Laboratory, Hitachi, Ltd. Japan
and
Yuichiro YOSHIMOTO, Hideaki UTSUNO
Hitachi Works, Hitachi, Ltd. Japan

For Presentation at
"Specialist Meeting on Pump Performance and Reliability",
Committee on the Safety of Nuclear Installations,
Nuclear Energy Agency of OECD,
GRS Headquarters, Cologne, Germany,
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Tomoyoshi OKAMURA, Takeo TAKAGI
Mechanical Engineering Research Laboratory, Hitachi, Ltd. Japan
and
Yulchiro YOSHIMOTO, Hideaki UTSUNO
Hitachi Works, Hitachi, Ltd. Japan

Abstract

The inlet and exit flows of internal pumps installed in an internal pump plant are examined analytically and experimentally to confirm pump reliability. The finite element method using a two-equation model (k-ε model) for turbulence is applied to analyze the actual turbulent flows. Analysis is conducted for two sector-shaped vessels, that is, one with an internal pump and the other with three internal pumps. To verify the computational results experiments using a half-scaled model and 1/5-scaled model are carried out. It is confirmed that the turbulent flow analysis is useful to estimate the flows around the internal pumps installed in the reactor vessel. It is clear that the velocity near the shroud wall is higher than that near the vessel wall at the pump exit. The effects of nonuniform velocity distribution at a pump exit on hydraulic loss, and radial thrust on an impeller are quantified.

1. Introduction

An internal pump system is applied to an internal pump type of a reactor plant. Since internal pumps are installed at the bottom of the pressure vessel as shown in Fig. 1, extremely high pump reliability is required. From the hydraulic design viewpoint of pumps, the following parameters are important to evaluate pump reliability: hydraulic forces acting on the impeller, a pressure drop at the pump exit and flow patterns downstream of the pump exit. Therefore, flow distribution near the inlet and pump exits needs to be clarified. However, it is very difficult to experimentally obtain velocity distribution around the internal pumps. On the other hand, recent development of supercomputers has made it possible to simulate turbulent flow in an actual flow region of industrial equipment. [1] In this study, the turbulent flow analysis at inlets and exits of internal pumps was performed. Experiments using a half-scaled and one fifth-scaled hydraulic model were also carried out to verify the analytical results. In this paper, these results are described and some comparisons of computational results with experimental results are also described.
2. Flow Analysis

2.1 Analytical Method

A computer program code called HISTREAM [2] was applied for three-dimensional turbulent flow analysis. This code was developed in the Mechanical Engineering Research Laboratory of Hitachi, Ltd. A finite element method based on the SMAC method is used to treat a flow field having arbitrarily shaped geometry. The k-ԑ two-equation model is applied to the turbulence model. Computation was made by using the supercomputer S810 of Hitachi, Ltd..

2.2 Analytical Flow Region

Two kinds of flow regions were analyzed. One is the sector region of a pressure vessel with one internal pump, as shown in Fig. 2. The other is a sector region with three pumps, as shown in Fig. 3. Analysis was made separately for the upper-region of the pump deck (pump suction) and the lower-region of the pump deck (pump delivery) because of the computer's memory capacity. For an example, a finite element model is constructed for the lower part of the three pumps sector consisting of 15,357 nodes and 11,520 elements.

2.3 Analytical Conditions

The analytical conditions are shown in Table 1. Analysis was mainly performed at the duty flow rate for the prototype flow region under the boundary conditions indicated in Fig. 4. Further computation was made as follows. In the sector with one pump, the effects of the residual rotational flow at the pump exit on the flow pattern and pressure distribution were examined. In the sector with three pumps, the effect of a tripped pump on the flow pattern were examined. The boundary conditions for computation are shown in Fig. 5. The model's flow was also compared with that of the prototype.

3. Results of Flow Calculations

3.1 Pump Inlet

The computational results of flow patterns near the pump inlet in the one pump sector are shown in Figs. 6 and 7. Figure 6 shows the three-dimensional velocity vectors on the horizontal section at the pump inlet height. Figure 7 indicates the two-dimensional velocity vectors within a vertical section including the center axes of the pressure vessel and pump. Comparison between the computational velocity and experimental distributions obtained in the 1/5- scaled model by using a Laser Doppler Velocimeter is shown in Fig. 8. The following results are obtained from these figures.
• The velocity distribution at the pump inlet is not so distorted that pump performance and reliability are influenced.
• Some flow disturbance is observed in the vicinity of the shroud and vessel walls at the pump inlet. This flow is caused by the upward stream between both walls and the cylindrical pump casing.
• The computational flow pattern is similar to that obtained experimentally.

3.2 Pump Exit

1) Flow Patterns

Figure 9 shows the velocity profiles obtained analytically at the pump exit region in the sector with three pumps. Large vortices and a severely skewed flow profile do not exist. The velocity inside the shroud is very low as compared with that near the pump exit. Upward-reverse flows appear between the pressure vessel wall and cylindrical pump casing. These flow patterns result from the passage blockage at the vessel bottom.

2) Velocity Distributions

The constant velocity contours of the pump exit region in a vertical cross section are shown in Fig. 10. Since the flow passage at the downstream pump exit is not axisymmetric, the flow at the pump exit is not uniform around the pump hub. The velocity near the vessel wall (left side of pump hub) is lower than that near the shroud wall (right side). As mentioned above, this flow profile is caused by downstream blockage at the vessel bottom. The main flow heads toward the opening between the shroud support legs, similar to a wall jet.

The normalized meridional velocity distribution at the pump exit is shown against the circumferential direction angle in Fig. 11. The velocity reaches its maximum mean velocity \( C_m \) of 115% at the circumferential angle \( \theta = 270^\circ \) i.e. the nearest passage to the shroud. It decreases to its minimum \( C_m \) of 86% at \( \theta = 90^\circ \), i.e. the nearest passage to the pressure vessel wall. This distorted velocity distribution increases the hydraulic losses in the pump. The pump diffuser is usually designed on the assumption that inlet velocity is uniformly distributed in the circumferential direction. The inlet blade angle of the diffuser is decided from the inlet velocity triangle which is derived from the design flow rate. So, if the inlet velocity decreases or increases, the inlet flow angle differs from the design blade angle. Thus, the hydraulic loss increases at the diffuser blades. Considering that the pump consists of many small partial pumps as shown in Fig. 13, hydraulic loss is expressed by the following equation:

\[
\Delta h = H_s - \frac{1}{2\pi} \int_0^{\pi} H(Q) \lambda(\theta) \, d\theta
\]

\[
= \frac{H_s}{2\pi} \int_0^{\pi} \left(1 - \frac{H(Q_s)}{H_s} \lambda(\theta)\right) \, d\theta
\]

(1)
where, \( \Delta h \) : hydraulic loss \([\text{m}]\),
\( H_e \) : total head at design point \([\text{m}]\),
\( H(Q) \) : total head at flow rate \( Q \) \([\text{m}]\),
\( \lambda(\theta) \) : velocity distribution coefficient (Fig. 10),
\( Q_e \) : design flow rate \([\text{m}^3/\text{s}]\).

From Eq. (1) it is evident that hydraulic loss in the pump increases with increasing of both the gradient of the pump characteristic curve \((H_e - H(\lambda Q))/(Q - \lambda Q)\) and the velocity distribution coefficient at the pump exit \( \lambda(\theta) \).

3) Pressure Distributions

The static pressure distribution at the pump exit on the horizontal cross-section in the sector with one pump is shown in Fig. 15. The static pressure near the vessel wall is higher than that near the shroud wall. Figure 12 shows the pressure distribution against the circumferential direction at the pump exit in the sector with three pumps. The maximum value reaches 120% of the mean pressure. This result is easily forecasted from the velocity distribution of Fig. 11. That is, the maximum pressure location has the lowest velocity and vice versa. The dotted line indicates the results obtained computationally. The solid line denotes the pressure obtained by experiments at the diffuser inlet, i.e., the impeller exit of a 1/2-scaled model. The experimental values were measured at the pressure taps installed on the hub wall between the diffuser blades. As a result, they fluctuate much like saw teeth. The coincidence of both results seems to be qualitatively sound. According to this pressure distribution, the radial thrust on the impeller is thought to act in the vessel-to-shroud direction. The approximate magnitude of the radial thrust is calculated by Eq. (2).

\[
F_r = \int_0^{\pi} P(\theta) b_2 R_2 d\theta.,
\]

where, \( F_r \) : radial thrust \([\text{N}]\),
\( P(\theta) \) : pressure at impeller exit \([\text{Pa}]\),
\( b_2 \) : axial length of impeller exit \([\text{m}]\),
\( R_2 \) : outer mean radius of impeller \([\text{m}]\).

Figure 15 also shows the effect of residual rotational flows at the pump exit. The rotational angle of 10° was assumed as the calculating condition. It was made clear that the effect of the rotational flow on the pressure distribution was not great.
4) Flow at One Pump Tripping

An internal pump plant system is designed so that even if one pump tripped, plant operation continues. So, it is significant from the viewpoint of pump reliability to examine the flow around the tripped pump. The internal pump is equipped with mechanical stoppers (anti-reverse unit) so that the tripped pump is unable to reverse when a counter flow occurs in the pump. Therefore, flows near the stopped pump are also analyzed. The calculated flow profile at the upstream pump is shown in Fig. 16(a). The reverse flow through the tripped pump reaches approximately the height of three times the pump inlet diameter from the pump deck. It is confirmed that the reverse flow has little effect on the inlet flow of adjacent ordinary pumps such as inducing prerotation or distorting velocity distribution. The analytical velocity profiles of the downstream pump are shown in Fig. 16(b). It is observed that extremely disturbed flow does not appear around the normal and tripped pumps.

5) Pressure Drop at Downstream

Evaluation of pressure drop in the vessel is necessary to decide the pump head specifications at the rated core flow. The pressure drop is affected by the geometry and arrangement of complex structures inside the vessel. Since quantitative estimation is quite difficult by applying the pressure loss data of hydraulic passage elements, simulation of the core flow is fairly useful. The pressure drop coefficient defined by Eq.(3) is shown in Fig. 14.

\[ \zeta = \frac{\Delta H_s}{1/2 g (Q/A)^3}, \]  

where, \( \Delta H_s \): pressure drop between pump inlet and exit (inside the shroud) [m], 
Q : pump flow rate [m³/s], 
A : hypothetical pump delivery area [m²], 
g : acceleration of gravity.

The pressure drop is evaluated between the suction passage wall and the shroud inside wall. The calculated and experimental results are compared in Fig. 14. The results coincide fairly well. Hence, it is confirmed that flow analysis is also effective to obtain pressure information around the internal pump installed in the pressure vessel.

6) Comparison between Model and Prototype

The flow profiles in a half-scaled model with ordinary water temperature and a prototype with 547° K (274°C) water are shown in Fig. 17. These flow patterns are quite similar to each other. So, the effect of the Reynolds number on the flow pattern seems to be small. However, optimization of parameters concerning the artificial viscosity needs so many calculations that the analytical parameters used for this computation may not be optimized. Therefore, the calculated flow may be more viscous than the actual flow. As such, more computational studies are necessary to obtain conclusions regarding the effect of the Reynolds number.
4. Conclusions

From analytical and experimental study of flows around the internal pumps, the following results were obtained:

1) It was confirmed that turbulent flow analysis is effective for the evaluation of flow around the internal pumps.
2) It was made clear that the velocity and pressure differences exist at the circumference of the pump exit. This is caused by the non-axisymmetric flow passage at the downstream of the pump. This distorted flow profile results in hydraulic loss in the diffuser, and radial thrust on the impeller.
3) The flow pattern between the model and prototype appears to be similar, but more analytical studies are necessary to optimize the viscosity parameters.

Acknowledgements

The authors wish to thank Dr. Ikegawa of the Mechanical Engineering Research Laboratory of Hitachi, Ltd. for his valuable advice concerning computation. They also wish to thank Mr. Sakamoto of Hitachi Works of Hitachi, Ltd. for his help with experiments and Mr. Hatakeyama of Hitachi Information Systems for his assistance with calculations.

5. References

Table 1  Calculating Conditions

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*) Re=CD/ν

where,  C : Pump Inlet velocity [m/s],
       P : Pump Inlet Diameter [m],
       ν : Kinematic viscosity [m²/s].
Fig. 1 Internal Pumps Installed in Reactor Vessel

(by courtesy of NUPEC Japan)
Fig. 2 Analytical Region (One Pump)

Fig. 3 Analytical Flow Region (Three Pumps)
Fig. 4 Boundary Conditions

Fig. 5 Boundary Conditions (One Pump Trip)
Fig. 6 Flow Pattern at Pump Inlet

Fig. 7 Flow Pattern near Pump Inlet

Fig. 8 Velocity Profiles near Pump Inlet
Fig. 9  Velocity Profiles at Pump Exit Region

Fig. 10  Constant Velocity Contours at Pump Exit Region
Fig. 11 Velocity Distribution at Pump Exit

Fig. 12 Pressure Distribution at Pump Exit
Fig. 13  Partial Pumps and $Q$-$H$ Performance

Fig. 14  Pressure Loss Coefficient (1/5-Scaled Model)
Fig. 15 Pressure Distributions at Pump Exit (One Pump)
Fig. 16  Flow Profiles at One Pump Tripping
Fig. 17  Flow Profiles of Pump Downstream in Model and Prototype
NPSH-REQUIREMENTS OF LARGE PWR-BOILER FEED PUMPS AT PART LOAD PUMPING CONDITIONS

Dr. F. Schubert

SIEMENS POWER GENERATION KWU
INTRODUCTION

"You can never have too much NPSH" - that's the title of a publication of James Healy. It is the best way to deal with cavitation problems, but often it is not possible to go it.

So cavitation, cavitation erosion and cavitation damage had been topics since pumps had been built, and they are topics up till now. And it can be expensive because of spare parts needed, disassembly and assembly and maybe even plant outage.

In 1978 Makay presented the results of an examination of pump outages, where cavitation is on the 5th place behind seal failures, vibration and bearing problems (table 1). 10 years later Gopalakrishnan presented the results of a similar examination (table 2). The order of succession had changed a little bit, but cavitation damage is still on place 5.

So I would like to present some of our experiences we have made with the main pumps of the main feed water pump units of some 1300 MW PWR power plants that are built by Siemens/KWU.

In table 3 the main data of a 1300 MW PWR power plant feed water pump are compared with a 770 MW coal fired power plant (CFP). As I said before we will only have a closer look on the main pumps.

The head of the CFP-pump is about 5 times as high as the head of the PWR-pump, so the CFP-pump has a 5 stage impeller instead of 1 stage of the PWR-pump. The mass flow - or better for comparison related to cavitation - the volume flow is 3 times greater at the PWR-plant, so we have 2 pumps on parallel service. The CFP has one 100% pump for normal operation, thus this pump is a very large one. It is turbine driven with variable speed, and the speed for 100% rated discharge at normal operation is 4880 rpm. The PWR-pump is motor driven with 5296 rpm constant speed.

In figure 1 we can see a schematic view of the PWR main feed water pump unit, with the one stage double suction booster pump, the constant speed electrical motor, with the gear box and the one stage double suction high speed main pump.
CAVITATION

Before we will have a closer look on this main pump let me say some words about cavitation.

As you all know, cavitation occurs if the value of NPSH-available becomes smaller than the required value NPSH-incipient. This is the point when the first bubbles appear at the blade inlet of impeller.

MEASURING METHODS

The very common measuring method is the determination of a certain head drop (mostly 3% head drop) compared with the cavitation free operation.

The visualization method - i.e. the direct observation with use of stroboscopic light - is an other but very expensive method. A special test rig with original size is necessary.

The acoustic method is the 3rd one. A lot of experience is necessary for interpretation of the measuring results. No special test rig is required.

CRITERIA OF ASSESSMENT

At 3% cavitation caused head drop a considerable amount of cavitation occurs. The disadvantage is that mostly only the NPSH - value at 3% head drop is known and not the complete head drop versus NPSH characteristic. As an example the head drop characteristics of two different impellers are compared in figure 2a. The NPSH value for 3% head drop is the same at both characteristics. Imagine the two impellers have 5m NPSH-available, respectively. So with impeller #2 no problems may occur, while impeller #1 at least keeps the latent possibility of damage.

If an impeller is special designed to deal with a certain amount of bubbles to produce good NPSH-3% values (a simple method is to enlarge the inlet area of impeller), it can get cavitation erosion problems instead of this (as we just saw at impeller #1). On the other hand this impeller - with a high suction specific speed - might
have problems with recirculation flow or unsteady flow patterns at part load conditions.

The visualization method and the acoustic method can detect bubbles before any effect on head occurs. In our little example with the two head drop characteristics the points where incipient cavitation is detected may be as shown in figure 2b. So it is possible that cavitation occurs long before the head is affected. If an impeller is designed for bubble free operation it might be better at part load conditions than an impeller designed only for low NPSH-3% head drop values, instead of the higher suction specific speed. So it is necessary to deal with the details if problems are expected, t.ex. with high energie impellers.

CAVITATION DAMAGE

The existence of mild cavitation seems not to be sufficient for cavitation damage, as we can learn from the practice. Many pumps operate with a small amount of cavitation without cavitation problems.

A strong influence on cavitation erosion comes from the inlet velocities, water temperature, operation mode and the material of impeller blades. Erosion factors are published by different authors, t.ex. Doolin or Galich.

These erosion factors can only be used for estimation of cavitation erosion if cavitation already occurs. And cavitation erosion can only occur if firstly there are cavitation bubbles and secondly the resistance of the material against cavitation erosion is smaller than the attack of it.

NPSH-required VALUE AT PART LOAD

A normal NPSH versus flow characteristic at 3% head drop as it can be found in many catalogues, ordinary has a smooth shape with decreasing NPSH-required values at decreasing flow. No problems seem to exist at part load if NPSH-available is big enough at rated discharge.

But if you choose a stricter criterion the things may change. In figure 3 I've chosen an impeller with extreme behaviour for better realization. It is an impeller of a
submersible pump.

The testings were carried out at Pfleiderer Institute for Fluid Flow Machines of Technical University of Braunschweig.

It can be seen that the part load peak is the higher, the stricter the criterion is. If "bubble free" would be demanded the peak would be higher again.

In the literature indications can be found that the part load peak establishes at onset of recirculation or even short before.

But here are big individual differences between different pumps depending on hydraulic design, geometries of impeller inlet and inlet chamber and kind of diffusor (volute, vaned...). Even things like the in- leading of internal leakage flow are important.

At the present time no theoretical procedure to calculate the NPSH peak at part load in any way, is known.

So it seems to be the best (and only) way to test the pump by visualization (or acoustic) method, if the precise knowledge of the complete cavitation behaviour of that pump is necessary.

OPERATING EXPERIENCES

Now I'd like to give a short overview of our operating experiences with PWR feed pumps referring to cavitation, within the last 15-16 years. Only the main pump is of interest in view of cavitation problems, because the booster pump has none.

Figure 4a is a section drawing of an elder main pump type. It has an double suction one stage impeller and a double volute type diffusor. The casing is of casting type. We will have a closer look on the main data later.

By the way, I've chosen the pumps of one manufacturer only because we have the most experiences with part load service with these pumps.

When these pumps in 11/1974 were startet up for initial operation in the plant, it could be detected after a
short time of service that cavitation caused erosion had occurred, although design was correct in the light of the experiences of that time.

It was decided to build a bubble test rig (BTR) of original size at the workshop of the pump manufacturer and to test the impellers.

In figure 5a the characteristics of NPSH-incipient versus flow are shown schematically as well as the head of the booster pump. The reference point is the point of original design of the booster pump because NPSH-available and the head of the booster pump are nearly the same.

The curve "a" is a result of the first bubble testings. Curve "a" has it's minimum NPSH-incipient value at 100% load but the impellers were not bubble free even at the design point. At the complete range of operation cavitation occurred.

Now a design review was started and new impeller geometries were developed in several steps with the aim to reduce bubble length (or the amount of bubbles at the first steps). An important step was to profile the blade inlet area and to manufacture the profile shape of all blades identically. The value of NPSH at 3% head drop became insignificant.

In 03/1978 the impellers were bubble free at 100% load, at part load conditions a bubble length of 30mm was measured (curve "b").

The NPSH-incipient curve "b" could be reduced at lower flow by modifications of the inlet casing geometrie - the short diffusor type inlet geometry was developed. The head of the booster pump was raised as much as possible.

The results of these modifications in 07/1978: The bubble length at part load conditions was only 10-15mm.

This was a good result. With the so modified pumps a long service life could be reached. Figure 6 shows pictures of an impeller after more than 65,000 hours of full load operation. The shining areas on the pictures are only areas with a little increased surface roughness.
Some times later the plant started a longer period of part load service. The result at the impellers due to cavitation: considerable cavitation erosion.

Finally the impellers were replaced.

In the meantime the development and the design of impellers with a shorter bubble length was carried on by the manufacturer.

A new system design of powerplants with more electrical power output caused a development stage at the feed water pumps. The main data of the new and the old feed pump design are compared in table 2.

The flow was increased by 27%, the speed was increased by 4.3%. The head of the booster pump could be increased by 50% by means of a new designed impeller.

As we saw some minutes before a higher relative velocity at impeller entrance and a lower water temperature cause an increase in cavitation erosion (if cavitation occurs, of course). The numbers at the bottom of table 2 are calculated values of this increase. The reference plant is the first one in table 2 (former pump design).

So a new pump design was necessary.

Figure 4b shows a section of the new designed pump. It is a forged-casing-type pump with new designed impeller and inlet chamber - the short diffusor type inlet canals - and the pump has an vaned diffusor additional to the double volute.

The NPSH-incipient versus flow characteristic of the new pump is shown in figure 5b schematically.

In 03/1982 these impellers had only bubbles at a limited range of flow. Within this range the bubbles were 5-10mm long. At a service smaller than 60% rated flow or greater 80% these impellers were bubblefree.

By the way - the "Feedwater Pump Technical Specification Guidelines" from Electric Power Research Institut give an definition of cavitation inception. In figure 4.1-5 of these guidelines cavitation inception is defined with a bubble length of 5mm. Our definition of cavitation inception is "no bubbles". So an NPSH-incipient - line according to our definition is the boarder line between the first noticeable bubbles above and the area of
absolute bubblefree operation under (and on !) this line. This is not an exact definition for laboratory measurements but it is on the safe side in the practice.

During the start-up commissioning of one unit the pumps were operated at an 1000 hour test at minimum flow conditions. At an inspection after this test no cavitation could be indicated.

At the beginning of 1989 - this new type of pumps is on service in power plants since 1985 without cavitation problems - some of the plants are on part load service at about 60-80% rated electrical output, in part up to 45% of the operating time.

Figure 7 shows the results of this kind of service conditions. In this diagram the average depth of cavitation erosion - the average of all blade inlet areas of an impeller - is drawn versus total time of service. This diagram is up to date, so the total time of service for each plant has the value of it's revision '90, respectively. The reference value is the cavitation erosion at one blade (the worst one), where the exchange of the impeller was recommended up to now.

In this diagram only power plants with the new type of pumps are represented.

In table 2 we could see that the feed water temperature has an influence on cavitation erosion, too. This we will have to take into consideration when we compare the characteristics.

* Power plant #1 with low feed water temperature and 45% part load service of the total operation time has the deepest cavitation erosion.

* Power plant #4 has an erosion rate like the plants #2 and #5 instead of it's relative high part load portion of 30% of operation time. The plants #2 and #5 have about 10% part load portion, but they have a low feed water temperature obvious to plant #4 having a high feed water temperature.

Instead of the fact that the line of power plant #4 in figure 7 can be compared very good with the line of plant #1 by means of calculation, the erosion factors should be used carefully. Inaccuracies can occur at measuring the erosion depth, by informations about the operation mode (which pump, which kind of operation, how long in service...) and by system induced influences.
In this particular case it has to be considered that cavitation can only occur in a certain range of flow rate, so cavitation erosion is limited to this range, too.

Let's come back to our history: Nobody had expected that this little bubble which only occurred between 60 and 80% rated flow would be able to cause problems. So the results of inspection at beginning of 1989 led to the new experience, that a small bubble even if hard materials are used would cause erosion if the operation time is in the amount of thousands of hours with these operation conditions.

In 07/1989 model impellers with a completely new designed inlet area were tested. The blades had a special profile shape to avoid bubbles within the complete range of operation.

Remember: if all main parameters are given, t. ex. speed/inlet velocities, water temperature, NPSH-available, cavitation caused erosion can only be avoided by

- a material with much more cavitation erosion resistance (that is not possible because stellited surface is already used)

- avoiding bubbles at the complete operation range.

So it was decided to replace the impellers of the main pumps in the 3 power plants with low feed water temperature.

At beginning of 1990 the original impellers were tested in the BTR. Figure 5b shows that these impellers are bubble free within the complete range of operation.

The impellers were replaced during the overall maintenance inspections '90.
CONCLUSIONS

Incipient cavitation and cavitation erosion cannot be calculated theoretically up till now, especially if part load service is required. So it often is necessary at pumps with high energetic impellers or with special features related to NPSH, to deal with the cavitation details.

At extreme demands on service (full operation range without limits in operation time) and at pumps with high relative velocity values it is the best (and only) way to design impellers without bubbles at the complete range of operation.
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3. Mansell, C.J. "Impeller cavitation damage on a pump operating below it's rated discharge" I MechE conf. publi. C167/74, p. 185 - 191

4. Gopalakrishnan, S., "A computer program for the diagnosis of feed pump problems" EPRI C5-5857 06'88, p. 4 - 25...

5. Güllich, J., et.al. "Quantitative prediction of cavitation erosion in centrifugal Pumps" 13th. IAHR symposium, Montreal '86

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11. Cooper, P., Antunes, F., "Cavitation damage in boiler feed pumps" EPRI CS-3158, p. 2 - 24 ...

12. Stoffel, B., Hergt, P., "On problems of the specific suction number as evaluation criterion for operational safety and reliability of a centrifugal pump" VDMA Pump Congress, Karlsruhe '88, Paper B8
<table>
<thead>
<tr>
<th>NO.</th>
<th>PUMP FAILURES: COMPONENT, SYMPTOM OR TECHNOLOGY</th>
<th>FEED PUMP</th>
<th>BOOSTER PUMP</th>
<th>OTHER PUMP TYPES</th>
<th>TOTAL NO. OF FAILURES</th>
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<tr>
<td>1</td>
<td>SEALS</td>
<td>602</td>
<td>178</td>
<td>198</td>
<td>978</td>
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<td>228</td>
<td>85</td>
<td>60</td>
<td>373</td>
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<td>3</td>
<td>AXIAL BALANCING DEVICE</td>
<td>337</td>
<td>--</td>
<td>--</td>
<td>337</td>
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<tr>
<td>4</td>
<td>JOURNAL BEARING</td>
<td>209</td>
<td>52</td>
<td>37</td>
<td>298</td>
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<td>5</td>
<td>CAVITATION</td>
<td>161</td>
<td>64</td>
<td>46</td>
<td>271</td>
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<td>IMPELLER BREAKAGE</td>
<td>169</td>
<td>8</td>
<td>7</td>
<td>184</td>
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<td>7</td>
<td>WEAR-RING: RAPID WEAR</td>
<td>155</td>
<td>3</td>
<td>--</td>
<td>158</td>
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<td>8</td>
<td>UNSTABLE HEAD CURVE:</td>
<td>92 *</td>
<td>59 *</td>
<td>10 *</td>
<td>161 *</td>
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<tr>
<td>9</td>
<td>SHAFT BROKEN/DAMAGED</td>
<td>77</td>
<td>6</td>
<td>51</td>
<td>134</td>
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<td>10</td>
<td>THRUST BEARING</td>
<td>58</td>
<td>11</td>
<td>9</td>
<td>78</td>
</tr>
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</table>

TABLE V: TEN MAJOR OUTAGE PRODUCING FAILURE CAUSES.

Survey of Feed Pump Outages

FP-754
Research Project 641
Final Report. April 1978
Work Completed, December 1977
Makay, E.

* UNSTABLE HEAD CURVE SHOWS AS A RELATIVELY LOW NUMBER, NAMELY IN MOST CASES IT IS NOT RECOGNIZED UNTIL THOROUGHS EXAMINATION OF THE PUMP RECORDS, WHICH IN MANY CASES ARE NOT AVAILABLE.
### Table 1

**FAILURE MODES**

<table>
<thead>
<tr>
<th>FAILURE MODE</th>
<th>WEIGHTING FACTOR</th>
<th>ESTIMATED COST IN 1987 (Million of Dollars)</th>
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</thead>
<tbody>
<tr>
<td>1. Vibration</td>
<td>0.093</td>
<td>53.8</td>
</tr>
<tr>
<td>2. Impeller Breakage or Cracking</td>
<td>0.085</td>
<td>49.2</td>
</tr>
<tr>
<td>3. Shaft Seal Failure</td>
<td>0.084</td>
<td>48.6</td>
</tr>
<tr>
<td>4. Rapid Wear of Wear Rings</td>
<td>0.081</td>
<td>46.8</td>
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<tr>
<td>5. Cavitation Damage</td>
<td>0.077</td>
<td>44.5</td>
</tr>
<tr>
<td>6. Axial Balancing Device Failure</td>
<td>0.077</td>
<td>44.5</td>
</tr>
<tr>
<td>7. Broken or Damaged Shaft</td>
<td>0.075</td>
<td>43.3</td>
</tr>
<tr>
<td>8. Journal Bearing Failure</td>
<td>0.072</td>
<td>41.6</td>
</tr>
<tr>
<td>9. Seizures of Wear Rings, etc.</td>
<td>0.072</td>
<td>41.6</td>
</tr>
<tr>
<td>10. Thrust Bearing Failure</td>
<td>0.071</td>
<td>41.0</td>
</tr>
<tr>
<td>11. Unstable Head Curve</td>
<td>0.068</td>
<td>39.3</td>
</tr>
<tr>
<td>12. Auxiliary System Reliability</td>
<td>0.062</td>
<td>35.8</td>
</tr>
<tr>
<td>13. Hot Misalignment</td>
<td>0.043</td>
<td>26.0</td>
</tr>
<tr>
<td>14. Gear-Type Couplings</td>
<td>0.038</td>
<td>22.0</td>
</tr>
<tr>
<td><strong>TOTAL</strong></td>
<td></td>
<td><strong>578</strong></td>
</tr>
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### TABLE 2

4. Gopalakrishnan, S., "A computer program for the diagnosis of feed pump problems"  
EPRI CS-5857 06'88, p. 4 - 25...
<table>
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<tr>
<th></th>
<th>770MW coal fired plant</th>
<th>PWR 1300 MW</th>
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<tr>
<td></td>
<td>system</td>
<td>main pump</td>
</tr>
<tr>
<td>pumps for normal service</td>
<td>1x100%</td>
<td></td>
</tr>
<tr>
<td>main pump head [bar]</td>
<td>317</td>
<td>317</td>
</tr>
<tr>
<td>mass flow [kg/s]</td>
<td>678</td>
<td>678</td>
</tr>
<tr>
<td>volume flow [m³/s]</td>
<td>0.775</td>
<td>0.775</td>
</tr>
<tr>
<td>speed [rpm]</td>
<td>4880</td>
<td>variable</td>
</tr>
<tr>
<td>number of stages</td>
<td></td>
<td>5</td>
</tr>
</tbody>
</table>

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TABLE 3
main data of boiler feed pumps (main pumps):
comparison 770MW coal fired plant and 1300 MW PWR plant
<table>
<thead>
<tr>
<th>Plant (date of commissioning)</th>
<th>Former main pump design</th>
<th>Advanced main pump design</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1975..</td>
<td>1985</td>
</tr>
<tr>
<td>Main pump head [m]</td>
<td>642</td>
<td>645</td>
</tr>
<tr>
<td>Flow [m³/s]</td>
<td>0.92</td>
<td>1.165</td>
</tr>
<tr>
<td>Speed [rpm]</td>
<td>5079</td>
<td>5218</td>
</tr>
<tr>
<td>Specific speed (metric units)</td>
<td>27</td>
<td>31</td>
</tr>
<tr>
<td>Water temperature [°C]</td>
<td>180</td>
<td>180</td>
</tr>
<tr>
<td>Density [kg/m³]</td>
<td>884</td>
<td>884</td>
</tr>
<tr>
<td>Booster pump head [m]</td>
<td>141</td>
<td>212</td>
</tr>
<tr>
<td>Impeller material</td>
<td>1.4008</td>
<td>1.4313</td>
</tr>
<tr>
<td>Hard facing finishing</td>
<td>stellite eroded</td>
<td>stellite eroded</td>
</tr>
<tr>
<td>Impeller entrance velocities at 100% load:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Meridional [m/s]</td>
<td>12.1</td>
<td>14.8</td>
</tr>
<tr>
<td>Circumferential [m/s]</td>
<td>70.7</td>
<td>76.5</td>
</tr>
<tr>
<td>Relative [m/s]</td>
<td>71.7</td>
<td>77.9</td>
</tr>
<tr>
<td>Erosion factors related to:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Relative velocity (t.ex. Güllich/Pace)</td>
<td>1</td>
<td>1.6</td>
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<tr>
<td>Water temperature (Doolin)</td>
<td>1</td>
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</table>

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Table 4
Boiler feed pumps of PWR plants
data of main pumps
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FIGURE 2A HEAD DROP/NPSH - CHARACTERISTICS
point of cavitation detection

curve 2
optical method acoustical method

curve 1
optical method acoustical method

HEAD

100%

97%

0 2 4 8 10 NPSH [m]

point of operation

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FIGURE 2B HEAD DROP/NPSH - CHARACTERISTICS
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FIGURE 3  NPSH\text{required}  OF  SUBMERSIBLE  PUMP  IMPELLER

experimental testings carried out at
PFLEIDERER INSTITUTE FOR FLUID FLOW MACHINES,
TECHNICAL UNIVERSITY OF BRAUNSCHWEIG
FIGURE 4B
BOILER FEED PUMP, MAIN PUMP, FORGED CASING

type KSB Frankenthal
SIEMENS

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FIGURE 5A
DEVELOPMENT OF NPSH/FLOW CHARACTERISTICS
former main pump design
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FIGURE 5B
DEVELOPMENT OF NPSH/FLOW CHARACTERISTICS
advanced main pump design
Figure 6  Main Feed Water Pump - Main Pump Runner

time of operation:  65 673 h
service:  full load service
 cavitation:  no damage, only surface roughness
 (light areas at blade inlet)
SIEMENS

power generation KWU

FIGURE 7  EROSION DEPTH (AVERAGE VALUES) AT DIFFERENT PLANTS
NEED FOR ROTOR DYNAMICS CONSIDERATIONS
IN DESIGN OF VERTICAL PUMPS FOR RELIABLE OPERATION

A.N. Kumar
AECL-CANDU, Mississauga, Ontario
Canada, L5K 1B2

Presented at the Specialists Meeting on
“Pump Performance and Reliability”
Cologne, Germany
1990 November 26-28
NEED FOR ROTOR DYNAMICS CONSIDERATIONS IN DESIGN OF VERTICAL PUMPS FOR RELIABLE OPERATION

Ashok N. Kumar*

ABSTRACT

Modern system demands require some centrifugal pumps to operate at one or more "off-design" flow points in addition to the rated point. In such applications the pump hydraulics, its bearing design and rotor dynamics have to be optimized for a reliable pump design. The significance of these factors is often underestimated by the Pump Supplier.

This paper presents a case history of a vertical two-stage centrifugal pump design for a CANDU** nuclear reactor. These pumps experienced a series of bearing failures during the performance tests at the Pump Supplier's plant. The stiffness of all the water-lubricated pump bearings was inadequate to accommodate the hydraulic forces and vibrations at low flow modes resulting in unacceptable bearing wear. This was the main cause of the failures. Diagnostic solution involved the pump bearings to be redesigned along with improved pump operating points. This resulted in a dramatic reduction in the bearing wear rate by a factor of nearly 50.

The need for optimized rotor dynamic for such pumps, operating at "off-design" flows, is clearly demonstrated. Critical design consideration for design and reliable operation is presented in a simplified matrix form.

* Process and Safety Engineering Department. AECL-CANDU, Sheridan Park Research Community, Mississauga, Ontario, L5K 1B2, Canada.

** Acronym for CANadian Deuterium Uranium Reactors
1. INTRODUCTION

Generally, a centrifugal pump is designed for operation at a single point, close to the Best Efficiency Point (BEP) of the pump. However, there are some applications where a centrifugal pump is required to operate for several hours at one or more "off-design" flow points in addition to the rated point of the pump. In such applications the pump hydraulics and its rotor dynamics have to be optimized for a sound and reliable pump design. This factor is often underestimated or overlooked by the Pump Supplier.

This paper presents an interesting case history of a series of bearing failures which occurred on two-stage vertical centrifugal pumps of turbine type for a CANDU** Nuclear Reactor. These pumps were required to operate at three points in addition to the rated design point. All the failures occurred during the pump performance tests at the Pump Manufacturer's plant prior to shipment to the Reactor Site.

The pumps failed to meet one of the acceptance criteria defined in the AECL Pump Technical Specification for successful completion of the pump performance test. Considerable refinement to the pump bearing design and its rotor dynamics was necessary by the Pump Supplier before the pumps were acceptable.

2. RECOMMENDED PUMP DESIGN CONSIDERATIONS

Typically the hydraulic and other unbalance forces on any pump rotor bearing assembly are minimal close to the best efficiency point (BEP) of the pump. If the pump is required to operate at the points well removed from its BEP, then the hydraulic unbalance forces and the resultant vibrations acting on the pump rotor assembly increase. For reliable operation of any vertical centrifugal pump operating at one or more "off-design" flow points, the following design objectives are recommended.

2.1 Pump bearings should be designed to accommodate the highest vibrations and unbalance forces corresponding to the "worst" off-design flow point (farthest from the BEP of the pump).

2.2 Pump rotor assembly including all couplings should be balanced to as high an achievable balancing limit as practical.

2.3 Rotor dynamics of the pump motor rotor assembly should be optimized with respect to critical speeds, unbalance forces and misalignments to give an acceptable design.

2.4 Detailed match marked assembly procedure, especially for a multi-shaft pump design, should be established and followed. This procedure should enable repeatable assembly results.

2.5 The Pump Technical Specifications should be adequately detailed to include relevant tests with monitored measurements which could enable detection of unacceptable pump design.
These pump design considerations may be grouped under two broad categories:

a. Hydraulic design
b. Mechanical design

When the pump is required to operate at the "off-design" flow, two additional parameters, which are connected with mechanical design, need special consideration. These are:

c. Rotor dynamic optimization
d. Tribological design consideration

These four design parameters are schematically summarized in a simplified square matrix form in Figure 1.

When the pump is designed for operation at a single point, most pump manufacturers have a standard hydraulic and mechanical design nearest to the desired operating point. This design would have a proven performance. Hence such a design would require little or no modifications. The inner square in Figure 1 (shown shaded) represents such a design.

On the other hand, if the pump is also required to operate at one or more "off-design" flows, in addition to the rated point, then the pump manufacturer instinctively selects the nearest pump from the standard range which meets the rated point requirements. All the outer square considerations (Figure 1) apply in this case, with parameters (c) and (d) assuming a major design role. In some cases a completely new pump bearing design may be essential to accommodate the forces resulting from "off-design" flow operation. Sound design practice requires an optimization of the pump rotor dynamics to cater to the worst operating conditions.

General approach of most pump manufacturers has been to avoid the special considerations of the outer square in Figure 1. This results in either a marginal or unacceptable pump design. The bearing design and the rotor dynamic considerations are generally low in most pump manufacturers' design priorities. Nevertheless these are the critical design considerations for all "off-design" flow operation. Current trend towards multi-disciplinary design approach demands that all pump manufacturers pay suitable attention to such critical parameters. The case history presented in this paper fully illustrates the importance and need for such considerations in pump design, where necessary.

3. DESIGN DESCRIPTION OF THE PUMP MOTOR SETS

3.1 General

This case history pertains to a set of vertical, two-stage, multi-shafted centrifugal pumps of turbine type. The pumps were designed for Emergency Core cooling system of a CANDU Nuclear Reactor. There are four pumps per reactor, two operating and two on "standby". Two pumps either operate in parallel or operate one at a time depending upon the flow requirement for each operating mode. The contract was for procurement of eight pumps for two reactors.
FIGURE 1  PUMP DESIGN MATRIX

1. CRITICAL SPEEDS
2. MODE SHAPE vs. BEARING LOCATION
3. UNBALANCED RESPONSE ANALYSIS
4. MISALIGNMENTS

ROTOR DYNAMICS ANALYSIS

ROTOR DYNAMICS

MECHANICAL DESIGN

PUMP DESIGN

TRIBOLOGICAL DESIGN

HYDRAULIC DESIGN

1. MATERIAL SELECTION
2. SURFACE COATING
3. MECHANICAL SEALS/STUFFING BOX AND DESIGN
4. COUPLING DESIGN
5. BEARING DESIGN

1. NPSH AND SPECIFIC SPEED
2. HYDRAULIC FORCES
3. IMPPELLER DESIGN (HEAD/FLOW REQUIREMENTS)
4. NUMBER OF STAGES

1. OVERALL DESIGN OF ALL PUMP COMPONENTS
2. TOLERANCES; FITS
3. ALIGNMENT/BALANCING PROCEDURE
4. ASSEMBLY - MATCH MARKED
5. VARIOUS ANALYSES AND MIT PLANS
Each pump was designed to operate at the rated capacity of 600 litres/second (8000 IGPM) and a head of about 76 metres (250 feet). In addition to the rated flow, each pump was also required to be designed for operating at approximately 7%, 20% and 60% of the rated flow. The pump supplied has an efficiency of 80% at the rated point which is also its BEP.

The pumps were designed and manufactured to the requirements of ASME Code Section III Class 2 and to the quality assurance requirements of Canadian Standards Association (CSA) Code Z299.2. All the detailed design requirements were outlined in the AECL Technical Specification for the pump.

The pump rotor consists of two shafts connected together by a line shaft coupling. The total length of the pump rotor bearing assembly is about 5 metres (16.5 feet). The pump rotor bearing assembly is rigidly coupled and driven at 1200 rpm by a 600 kW (800 hp) electric motor.

The pump is also provided with flow stabilizers. The motor rotor assembly is about 2.3 metres (7.5 feet).

A schematic representation of the pump rotor bearing assembly is as shown in Figure 2.

3.2 Details of the Pump Bearing Design

When the pump is required to operate at one or more "off-design" flow points, as in this case, the bearing design of the pump is the most important parameter. Hence some bearing details are presented below for the pump design initially provided by the Supplier.

The pump rotor assembly is radially supported by four (4) water lubricated plain journal bearings, as shown in Figure 2. Each bearing assembly comprises a shaft sleeve of 440 A material provided with a ceramic coating (chromium oxide) and is mounted on the pump shaft. The surface hardness of this coating was in the range of (62-68) HRC. The bearing bushing is also of 440 A material which was hardened to 47-53 HRC after machining the three axial grooves at 120° orientation. The ceramic coating thickness after final machining was in the range of 0.30 mm (0.012"). The differential hardness between the sleeve and the bearing bushing is provided to minimize the possibility of wear and galling.

The nominal bearing diameter is 101 mm (4 inches) with a length to diameter (L/D) ratio of 2.3. Each bearing is lubricated with pressurized water through a single 12.5 mm (1/2") diameter orifice positioned at the top end of the bearing. Each bearing is provided with three axial grooves suitably dimensioned to eliminate any grit or sand particles getting entrapped in the bearings. Water supply to each bearing is in the range of 45 litres/second (10 IGPM) at an estimated supply pressure of 275 KPa (40 psi). The bearings are provided with a diametral clearance in the range of 0.228 to 0.305 mm (0.009" to 0.012").
3.3 **Details of Motor Bearings**

The motor is provided with rolling element bearings at the top and bottom end. The bearings accommodate the radial and thrust loads and are grease lubricated. They are standard bearings and are designed for a B-10 life of at least 100,000 hours for the operating conditions of the pump motor set.

3.4 **Monitoring Instrumentation**

The pump motor set is provided with a limited instrumentation as follows:

- two non-contacting proximity probes (at 90° orientation) to monitor the pump shaft run out at the line shaft coupling;
- RTDs on the motor bearing housing;
- velocity transducers on the motor frame to measure bearing vibrations in the axial and the transverse directions.

4. **PUMP OPERATING MODES**

The Emergency Core Cooling Pumps circulate water through the reactor core during a Loss of Coolant Accident (LOCA). These pumps are safety related and hence a high degree of reliability is essential. The pumps are required to operate at the rated flow and three “off-design” flow modes, as dictated by:

a. the size of the LOCA; and
b. the mode of operation.

Some modes demand a single pump while other modes require two pumps operating in parallel to make up the total flow requirements. The pumps are also required to be tested for short durations every month. Various flow modes of the pump are presented in Table 1 as a percentage of rated flow with approximate operating duration. However, all the “off-design” flow modes had to be revised from the initial contractual values* to enable the pumps to operate at more favourable points. The contractual and revised flows are shown separately in the Table. These improved flow requirements, together with extensive pump design modifications, resulted in achieving an acceptable pump design. The system design modifications corresponding to revised flows required changes to the piping layout at the reactor site.

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* These were the values to which the pumps were to be supplied meeting all the requirements of the AECL Pump Technical Specification.
TABLE 1 Operating Flow Modes of the Pumps

<table>
<thead>
<tr>
<th>Flow Mode</th>
<th>Percentage of Rated Flow*</th>
<th>Operating Duration</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Contractual Flow/Pump (1)</td>
<td>Revised Flow per Pump (2)</td>
</tr>
<tr>
<td>1. Pump Testing (Single Pump)</td>
<td>25</td>
<td>40</td>
</tr>
<tr>
<td>2. Recirculation 2-Pump Operation</td>
<td>5</td>
<td>16</td>
</tr>
<tr>
<td>3. Small LOCA 2-Pump Operation</td>
<td>28</td>
<td>40</td>
</tr>
<tr>
<td>3.1 1-Pump Operation</td>
<td>35</td>
<td>65</td>
</tr>
<tr>
<td>4. Large LOCA 2-Pump Operation</td>
<td>=105</td>
<td>=105</td>
</tr>
</tbody>
</table>

* Rated Flow = 600 litres/second (8000 IGPM)

5. ACCEPTANCE CRITERIA FOR THE PUMPS

As a part of Standard AECL design practise, all nuclear pumps for a CANDU reactor are subjected to a performance test after their manufacture and assembly at the Pump Supplier’s plant. The pumps pertaining to this case history were also subjected to a 24-hour endurance test. The main purpose of these tests was to establish the acceptability of all the hydraulic and mechanical characteristics of the pump motor set. Any deficiencies in the pump operation observed during the test can thereby be rectified.

Only the first pump motor set after assembly was tested for 150 hours to establish the general characteristics for this type of pump. The 24-hour endurance test applied to all the pumps. During the endurance tests, each pump was operated at the rated flow for sixteen hours and for the remaining eight hours the pump was tested at all the three “off-design” flow modes. It was necessary for AECL to specify a suitable acceptance criterion which would quantify the detrimental effects of operating at the “off-design” flow modes, if any.
As stated earlier, when a centrifugal pump operates at “off-design” flows, not only the hydraulic forces on the pump rotor bearing assembly increase but also the vibration levels at the pump bearings. The turbine type of pump design made it difficult to monitor the vibration levels at the pump bearings by standard vibration transducers, although the motor bearing vibrations were recorded. Hence an alternate parameter was established to monitor the acceptability of the pump design. The Pump Supplier was required to show that the total surface wear at any of the pump bearings or shaft sleeves did not exceed 0.05 mm (0.002") during the 24-hour test period. This was the major acceptance criterion included in the AECL Technical Specification for the pumps. It was felt by AECL Design Engineering that the bearing wear levels not only reflected the effects of imbalance forces and vibrations but also the effects of misalignments in the pump rotor assembly. Hence the choice of wear levels was an excellent acceptance criterion. Without this criterion, the deficiencies in the pump design would have gone unnoticed.

The surface wear was established by measuring the diameters of the bearing and the shaft sleeve before and after each test. An accurate micrometer (accuracy: 2.5 microns or 0.00010") was used to record the surface wear.

One other criterion which was effectively used in monitoring the overall pump motor assembly was the “rundown” time of the motor (time taken for the motor speed to drop to zero after switching it off from its rated speed). A repeatable predetermined rundown time of the motor was essential.

Other standard criteria included acceptable pump shaft run out, bearing temperatures, pump and motor characteristic curves.

6. SUMMARY OF THE PUMP FAILURES

Main highlights of the endurance tests on these pumps are summarized below:

6.1 There were seven separate pump test failures;

6.2 All the failures were due to unacceptable wear levels (greater than 0.05 mm) at one or more pump bearing locations over a 24-hour test period;

6.3 The measured wear in all the failures varied between 0.10 mm (0.004") and 0.30 mm (0.012"). Mostly the bearing bushings and some sleeve were involved in the failures. There was no consistent pattern in either the orientation of the wear zone on the bearing surface or the failures being confined to specific bearing locations. This lack of consistent pattern made it very difficult to isolate the cause of the failures.

6.4 The characteristic curves plotted for all the tested pumps were acceptable.

6.5 Despite these failures, four of the eight pumps had acceptable wear levels.
There was continuous discussions after each test between the supplier and AECL Design Engineering. Each test failure caused concern to both the parties.

A rotor dynamic and unbalance response analysis by AECL after the first two failures showed that very large deflections were to be expected at the pump bearing locations. Hence AECL consistently maintained the pump bearing design in general and the bearing stiffness in particular was inadequate to accommodate the operating forces and vibrations.

The inability to isolate the cause of these failures caused some concern regarding the reliability of the pumps. After seven pump test failures, it was decided to review the overall pump design individually by the Pump Supplier and AECL, utilizing the services of dedicated specialists.

7. CONCLUSIONS FROM PUMP DESIGN REVIEW AND FAILURES

Detailed design review by AECL Design Engineering resulted in conclusions as summarized below:

7.1 Hydraulic forces at the "off-design" flows, especially at the lowest flow mode, were considered to be grossly under-estimated in the pump design. It was concluded that increased flow at this mode and/or shorter operation duration would greatly reduce the bearing wear.

7.2 The stiffness of all the pump bearings were considered to be inadequate to support the hydraulic forces and the resulting high vibrations as evident from the bearing wear.

7.3 The bearing design itself was considered to be deficient and needed significant design modifications to critical bearing parameter such as L/D ratio, shape of the lubricating grooves, quantity of lubricant and increased number of orifices for injecting water into the bearings.

7.4 Detailed metallurgical investigation revealed that the quality of ceramic coating was poor. High porosity (up to 20%) and poor surface finish (2 micron CLA) were evident. Also some doubts existed regarding the adhesion of the coating. Hence these parameters needed to be improved in quality.

7.5 The tolerances and assembly procedures used by the Pump Supplier were considered to be inadequate to produce a repeatable well aligned pump rotor assembly. This was reflected by inconsistent wear pattern and its location. Improved match marked pump rotor assembly was considered essential to improve the reliability.

7.6 Improved balancing of the line shaft coupling and tighter assembly tolerance were considered to reduce the unbalance forces and misalignment.
8. CORRECTIVE ACTIONS

Based on the conclusions in the preceding section, the design changes listed in this section were implemented.

8.1 Bearing Design

The bearing bushing at each pump bearing location was completely redesigned with the following modifications:

a. Three axial lubricating grooves oriented 120° apart were replaced by a single spiral groove over the 101 mm (4") diameter of the bearing. The spiral groove was in the same direction as that of pump rotation. This greatly assisted the hydrodynamic action of the bearing.

b. The L/D ratio of the bearing was reduced from 2.3 to 1.0. This assisted in improving the stiffness and stability of the bearing.

c. The orifice for injecting the pressurized water into the bearing was increased in size from 12.5 mm (1/2") to 19 mm (3/4"). The number of orifices was increased from one to three.

8.2 Bearing Stiffness

As already discussed, the bearing stiffness is the single most important factor for pumps operating at "off-design" flows. The quantity of water supplied to each pump bearing was increased by an order of magnitude to 8.25 litres/second (110 IGPM). Increased water supply coupled with improved configuration of the bearing grooves was considered to have improved the bearing stiffness by a factor of four to an estimated value of 9425 kg/m (76,000 lbs/inch).

8.3 Ceramic Coating

The quality of ceramic coatings was greatly improved to give a porosity of about 8%. The surface finish of the coating after final machining was about 0.4 microns. The coating itself showed a more uniform quality. These results were established by examining sample coupons which were coated at the same time as the sleeve.

8.4 Match Marked Assembly

The diametral clearance at the line shaft coupling was reduced. The whole assembly procedure was by match marking. This reduced the misalignments and provided a repeatable assembly.

8.5 Revised Flow Modes

The lowest operating mode flow of the pump was increased by a factor of about four to 93.75 litres/second (1250 IGPM). The vibration level at this flow mode (before flow revision) is estimated to be in the range of 10 to 15 times those at the rated flow. These changes to the flow rates resulted in modifications to the piping design at the reactor site.
9. RESULTS OF CORRECTIVE ACTIONS

The pumps with modification as specified in Section 8 above were tested. One pump was tested for a 70-hour duration at the revised flow modes (1, 2, and 3 in Table 1). This included a 10-hour test duration in the recirculation flow mode with the following results:

9.1 The overall wear at the end of the 70-hour test was below 0.002 mm (0.0008”). This was less than half the allowable wear in 24 hours. Thus an improvement by a factor of 15 to 40 in the pump bearing wear rate was achieved, in comparison to that given in Section 6.3.

9.2 All the four pumps tested with modified pump components and revised flow modes easily met the AECL bearing wear acceptance criterion.

9.3 Improved alignment, balancing and match marked assembly resulted in repeatability of the test data.

10. CONCLUDING REMARKS

10.1 It is essential that the Pump Suppliers clearly define the limitation of their pump design at the bidding stage. The most detrimental “off-design” flow mode, in particular, must receive a detailed evaluation by the Pump Supplier. If the Purchaser’s operating points are unrealistic the Pump Supplier should not hesitate to renegotiate them.

10.2 AECL experience on these pumps showed that the vibration levels could increase considerably when the pumps operate at flows below 30% of the best efficiency point of the pump. The vibration levels at flows less than 10% of the BEP flow could be at least an order of magnitude higher than those at the BEP flow.

10.3 It is necessary for the Pump Supplier to realize the importance of bearing design for applications involving low “off-design” flows. The pump bearing design is the key to an acceptable pump design in such applications. Hence all Pump Suppliers should have an “in-house” program for a detailed pump bearing design.

10.4 This case history clearly established that the initial design and the design approach used by the Pump Supplier was unacceptable. A multi-disciplinary design approach which considers as a minimum all the outer square requirements (Figure 1) is necessary. A pump design merely based on the Supplier’s engineering judgement, in our experience, is considered unacceptable. Current reliability requirements taboo such an approach.
CONSTRUCTION AND QUALIFICATION EXPERIENCE ON THE RRA AND ISMP PUMPS OF THE FRENCH NUCLEAR REACTORS

C. MECH (Technical Manager - GEC ALSTHOM La Courneuve)

GEC ALSTHOM has built the RRA pumps of the 900 MW and 1300 MW reactors and the ISMP pumps of the 1300 MW reactors.

This paper describes the experience gained in the construction and the qualification of those equipments.

1 - Description of the RRA and ISMP pumps

The following chart gives the main characteristics of the different models.

<table>
<thead>
<tr>
<th>TYPE</th>
<th>RRA 900 MW</th>
<th>RRA 1300 MW</th>
<th>ISMP P 4</th>
<th>ISMP P'4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow (nominal) (m³/h)</td>
<td>1250</td>
<td>910</td>
<td>245</td>
<td>245</td>
</tr>
<tr>
<td>Head (mcl)</td>
<td>75</td>
<td>77</td>
<td>1025</td>
<td>1025</td>
</tr>
<tr>
<td>Inlet Pressure (bars)</td>
<td>28</td>
<td>41.5</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td>15° - 180°</td>
<td>15° - 190°</td>
<td>10° - 120°</td>
<td>0° - 120°</td>
</tr>
<tr>
<td>Rotating Speed (rpm)</td>
<td>1495</td>
<td>1485</td>
<td>1485 - 4277</td>
<td>1188 - 4277</td>
</tr>
<tr>
<td>Power (kW)</td>
<td>355</td>
<td>560</td>
<td>1300</td>
<td>1275</td>
</tr>
</tbody>
</table>

Fig. 1 gives the drawing of the RRA 1300 MW showing the single stage overhung impeller with a diffusor and the volute.

The shaft is supported on a front roller bearing and two rear ball bearings. Sealing is insured by a mechanical seal.

The Pumps is coupled to the electric motor through a tooth coupling and supported on a fabricated support.
The impeller diameter is 540 mm for the RRA 900 MW and 610 mm for the RRA 1300 MW.

The ISMP (Medium pressure safety injection) is shown Fig. 2 and 3.

The booster pump is a single stage horizontal pump, similar to the RRA design for the P 4 reactors and vertical (Fig. 2) for the P 4 reactors.

The main pump is a multistage barrel type equipped with mechanical seals and oil lubricated bearings.

The complete train is shown Fig. 4 and includes:
- an electric motor,
- the gear box driving the
- main pump,
- a gear box driving the
- booster pump.

11 - General options of construction

The pumps are built according to the general specifications of FRAMATOME and EDF Standard and quality insurance procedures defined in our AQ manual.

All the external casing in contact with the process fluid are in stainless steel 316.

The internal components are in 13 or 17 % chromium steel.

The hydraulic performances have been adjusted to comply with the requirement of the nominal point but also with the limited head at the minimum flow and at the maximum flow.

Provisions have also been taken to avoid any risk of cavitation in the operating range.

A very limited rate of leakage has been achieved by the use of mechanical seals.

To complete an extensive programme of calculation some complementary developments have been conducted during the construction programme in order to improve the reliability of the pumps.
III - Qualification of components - Partial tests

1°) Ball and roller bearings of RRA pumps

Some scratches have been observed on the ball bearings of the first pumps and pitting on the roller bearings and on the sleeves.

Extensive testing has been conducted on a special test rig to explain and correct those defects.

The following improvements have been defined:
- Preloading of the ball bearings
- Reinforcement of the shaft and mounting of the bearings directly on the shaft without sleeves
- Bearings material in stabilized steel
- Improvement of the rubber seals between bearing compartment and shaft seals
- Selection of grease compatible with the radiation level and the operating conditions

2°) Qualification of bearings in accidental conditions

The bearing system of the RRA pumps has been qualified in accidental conditions corresponding to the steam ingress in the RRA room.

The system has been submitted to a pressure of 4 bars and a temperature of 160° at rest and then after decreasing of pressure and temperature, we have demonstrated its ability to restart at a temperature of 100°C without any damage.

3°) Shaft sealing system

Mechanical seals with an hydrodynamic effect have been selected.

The rotating collar is in tungsten carbide and the static counter part in graphite.

The seal includes an auxiliary pumping system forcing the cooling water into an external heat exchanger.

For the ISMP pumps the specification includes the possibility to operate with impurities in the circuit and this imposes very severe conditions for the shaft seals.

A system of giracyclones has been developed to provide clean water to the seals when impurities are handled in the pump.
The system has been qualified by several separate experiments and also on the complete set of pumps during the complete test carried out in the INDRET test loop as described in the paper presented by EDF.

4°) Some other tests and modifications have been performed on the other components.

That is:

- Tests on the piping reactions on the RRA pumps due to the pressure and thermal effects.

Due to the importance of these reactions we have had to reinforce the base plate of the RRA 1300 MW, improve the fixation on the base plate of the RRA 900 MW and modify the couplings.

- For the ISMP booster pump using a vertical shaft a special water lubricated bearing with an helicoidal groove has been developed in order to avoid the half speed whirl instability encountered on unloaded bearings.

IV - Qualification of the complete pumping system

In order to achieve the best quality a test at full load is performed on each motor and pump.

A special test rig (Fig. 5) has been constructed in the RATEAU plant with a double loop for the testing of the RRA and the ISMP.

The testing facility includes:

- the two hydraulic loops in stainless steel with flow and pressure measurements,

- the electric installation with an available power of 1 500 kW,

- a control room with all the facilities to establish the hydraulic characteristics (flow, head efficiency, cavitation) and the mechanical behaviour of each set of pump,

- the water treatment,

- the power dissipating device.

V - Present status

More than 100 RRA pumps and 40 ISMP have been delivered and most of them are now in operation.
Some troubles have been encountered on the RRA pumps in the early years of operations and solutions have been developed and applied to obtain the required reliability.

For the new 1500 MW reactors new machines have been developed taking into account the experience gained on the previous machines.

For the ISMP pumps the main development has been on the use of an inducer on the booster pump allowing a significant reduction of the length of this vertical pump and an economy on the corresponding civil works of the Plant.

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**Fig. 1**
GROUPE MOTO-POMPE ISMP1300

Fig. 4
CAVITATION CHARACTERISTICS STUDY
OF A NUCLEAR REACTOR INTERNAL PUMP

Hideo Komita and Iwao Ohshima
Nuclear Engineering Laboratory
Toshiba Corporation
Yokohama, Japan

Sinichi Fukuda and Kazuo Suga
Isogo Engineering Center
Toshiba Corporation
Yokohama, Japan

ABSTRACT

Cavitation tests of a reactor internal pump were performed at several water
temperature up to the reactor operating temperature. Required Net Positive
Suction Head (NPSH) decreased with increasing water temperature. An empirical
equation for a decrease of the required NPSH was obtained. Cavitation at high
temperature did not affect pump vibration. Changes in flow rate at constant
pump speed influenced pump vibration during cavitation at low temperature.

1. Introduction

A jet pump has widely been used as a recirculation pump for Boiling Water
Reactor(BWR). In the last 10 years, reactor internal pump(RIP) which are in-
stalled in a reactor pressure vessel(RPV) instead of jet pump has begun to
use in Europe. In Japan, the RIP will be used in near future. In order to
develop the RIP which are more reliable and are suitable for Japanese par-
ticular conditions such as earthquake, many studies have been performed to
evaluate seismic(1), two phase flow, bearing wear and cavitation(2)
etc.(3)(4).

As the RIP is a important component to cool reactor core, it has to continue
supplying the reactor core with coolant without undue vibration during
cavitation. It is importance to accurately understand the cavitation charac-
teristics of the RIP.
The RIP is a semi-axial pump which used at high temperature. Many studies on pump cavitation have been reported, but most have considered centrifugal pumps at low temperature. Literature related to the cavitation characteristics of the semi-axial pump at high temperature such as RIP is only reference [5] to the best of our knowledge, but quantitative evaluation for cavitation characteristics was not discussed in that article. The purpose of this paper is to quantitatively discuss RIP cavitation characteristics with several water temperature. The cavitation characteristics that were measured and discussed here are: effect of water temperature on required NPSH and pump vibration and effect of flow rate at constant pump speed on pump vibration.

2. Test equipment and procedure

2.1 Test facility

A test facility consists of the test vessel, a flow measuring loop and an auxiliary system to control temperature and pressure. These are shown in Fig.1. Figure 2 shows a schematic view of the test vessel and flow measuring loop. The test vessel is 6.5m in height and 2.5m in diameter. Since it is well known that configuration of test vessel significantly influences pump characteristics, reactor internals such as shroud wall, RPV wall, pump deck and control rod guide tubes were installed in the test vessel. The flow measuring loop is equipped with flow control valves and venturi flow-meters.

2.2 Test instrumentations

The pressure, temperature and vibration instrumentations are illustrated schematically in Fig.3. Flow rate through the pump was measured with a venturi flow-meter. The pump head was measured by the differential pressure between the suction side and the discharge side of the pump. Vessel pressure and temperature were measured at the suction side of the pump. Strain-Gauge pressure transducer were used for pressure measurement. A calibrated platinum resistance thermometer was used to measure temperature. Impeller vibration was measured using an eddy current type displacement meter suitable for use at high water temperatures (300°C) and pressure(80 bar). Vibration in the motor casing was measured with accelerometer. Vibration velocity which is given by integration of acceleration was evaluated. Data were automatically recorded and analyzed by a computer data acquisition system.
2.3 Test procedures
Pressure of 10 Kg/cm² more than saturation pressure was applied to the test vessel, and then pump speed and flow rate were stabilized. While these test conditions were held constant, pressure was reduced very slowly. But while data was recorded, pressure was held constant. Pressure were intermittently reduced until cavitation was finally established. The onset of cavitation was detected by monitoring changes in pump head and subcooling. Water temperature was kept constant during each test.

3. Test results and discussion

3.1 Pump hydraulic characteristics and cavitation characteristics
Flow rate to pump head (Q-H) characteristics and cavitation characteristics were discussed here. The RIP used for this test has the Q-H characteristics shown in Fig.4. Optimum flow rate was 2.3 m³/sec at the rated speed. This pump is normally operated at a flow a few percent lower than the optimum flow rate, and cavitation tests were performed around normal operating conditions. Figure 5 shows the cavitation characteristics at low temperature. To make comparison with other data, required NPSH is defined as the point where the pump head is reduced by 3 percent from pump head under the non-cavitation conditions. The required NPSH of this pump was 28m. Figure 6 shows comparison of the test results with an formula obtained by Stepanoff and also with Hydraulic Institute standards. The formula is well used to predict cavitation characteristics of a pump. Test results well agreed with the formula, but test results was a bit less than that. This is due to the improvement of impeller vanes which reduced the required NPSH.

3.2 Effect of water temperature on required NPSH and vibration
Figure 7 shows relationship between pump head changes and NPSH with parameter of water temperature. The required NPSH decreased with increasing water temperature. This is most likely due to the large difference in water to vapor density ratio at different temperatures. At 50°C this ratio is 13000, while at 279°C it is only 20. This means that if an equal amount of liquid were turned to vapor at 50°C and 279°C, the vapor at 50°C would occupy a much larger volume. Empirical equation related to effect of water temperature on required NPSH for centrifugal pump has already been reported by Stepanoff(6) or Spraker(7). Although RIP is a semi-axial pump, comparison of test results with their equation was done in Fig.8. The vertical axis(ΔNPSH) rep-
resents the difference in required NPSH value between low temperature (50°C) and another temperature. Our test data gave larger values of ΔNPSH than the previously reported equations. This is mostly due to the difference of pump type. A theoretical analysis of this has not been accomplished yet.

In order to obtain an empirical equation for the R1P, Spraker’s equation which agrees qualitatively with our test results, was modified to agree with the test results. The equation is:

\[ \Delta \text{NPSH} = 1 / (B \times (1/R-1)) \]

where B is thermal cavitation factor
R is an empirical coefficient
\[ R = 0.515 - 0.0633 \times \Delta \text{NPSH} \]

Since R is an empirical coefficient for a centrifugal pump, it was modified for the case of a semi-axial pump. It can be expressed as follows:

\[ R = 0.933 - 0.034 \times \Delta \text{NPSH} \]

Figure 9 shows effect of water temperature on pump vibration during cavitation. At both 50°C and 150°C, the level of pump vibration increased near the point of the required NPSH. When NPSH was reduced, vibration decreased to the value measured under non-cavitation conditions. The peak value of impeller vibration was smaller than gap between impeller and diffuser, and the peak value of motor casing vibration was lower than allowable value (7mm/sec).

On the other hand, at both 200°C and at 279°C the vibration level remained constant, and the above changes did not occur. This indicates that the water to vapor density ratio has an effect on pump vibration as it does on required NPSH.

3.3 Effect of flow rate on pump vibration

These tests were performed at several flow rates with a constant pump speed at low temperature. Figure 10 shows the changes in pump vibration. Below the optimum flow rate, as described above, pump vibration increased near the required NPSH, and it fell to the non-cavitation level with further reduction in NPSH. On the other hand, at optimum flow rate and more than it the level of vibration did not change even if NPSH decreased under required NPSH. This is most likely due to the difference in the location where cavitation vapor was generated. Figure 11 shows the photographs and sketch of cavitation which was taken in another test facility. At less than the optimum flow rate,
vapor was generated on the back side surface of the vane only. On the other hand at the optimum flow rate and above, vapor was generated on both sides, These difference was caused by static pressure profile along impeller vane. The static pressure profile changes with a change in attack angle. Figure 11 shows samples of the static pressure profile. At less than optimum flow rate, the static pressure on back side surface is very low. This shows that cavitation easily occurs on back side surface. This qualitatively agreed with the results of photograph. When cavitation vapor was generated only on back side surface, change in vibration was occurred. Increase in vibration level at less than the optimum flow rate are most likely a result of load fluctuations caused by vapor forming only on the back side surface.

4. Conclusions

Conclusions resulting from this study are:

1) The required NPSH decreased with increasing water temperature. An empirical equation related to the decrease in required NPSH for semi-axial pump was obtained.

2) The effect of cavitation on pump vibrations was found to be much more gradual at high than at low temperatures. At low temperature, the level of vibration was lower than allowable value.

3) The flow rate at constant pump speed was found to affect pump vibration during cavitation also. At less than the optimum flow rate, the level of vibrations peaked near the required NPSH. At above the optimum flow rate, vibration remained constant.

References


Fig. 1  Reactor Internal Pump Test Facility

Fig. 2  Schematic View of Test Vessel
Fig. 3 Instrumentation

Fig. 4 RIP Hydraulic Performance
Fig. 5  Cavitation Characteristics at 50°C

Fig. 6  Relations Between Specific Speed and Thoma's Cavitation Coefficient
Fig. 7 Effect of Water Temperature on Cavitation Characteristics

Fig. 8 Relations Between $\Delta$NPSH and Water Temperature
Fig. 9  Effect of Water Temperature on Pump Vibration

Fig. 10  Relations Between Pump Vibration and Flow Rate at Constant Pump Speed
Fig. 11  Cavitation around Impeller Vane

Fig. 12  Examples of Static Pressure Profile
SESSION # 3
Chairman's Closing Remarks
(Mr. W. Minners, NRC, United States)

Mr. Okamura's three dimensional analysis demonstrated the computational ability to evaluate complicated flow and pressure distributions and showed that a distorted flow profile exists at the exit of internal pumps which results in a hydraulic loss in the diffuser and radial thrust on the impeller.

Dr. Schubert stated that cavitation has been a significant cause of pump failures, but that pump design can reduce cavitation and special materials can control it.

Mr. Kumar noted that pump design is a multidisciplinary process that consider rotor dynamics, mechanical and hydraulic design and bearing design in an integrated manner. He also demonstrated that off-design operating points must be considered in the design.

Mr. Mech reviewed the construction and qualification of pumps in French reactors, and how this experience was used in the design of pumps for the new 1500 MW plants.

Mr. Komita described the cavitation tests of reactor internal pumps which showed that cavitation did not affect pump vibration at high temperature, but did at low temperature with changes in flow rate. Required NPSH decreased with increasing water temperature due to the decreasing ratio of water to steam density.
SESSION #4: "PUMP TESTING, MAINTENANCE AND OPERATING CHARACTERISTICS"

Chairman: J.-P. Clausner

* "CANDU heat transport pumps - a quality product", A. N. Kumar, AECL, Canada

* "Design and manufacture of primary sodium pumps for the prototype fast breeder reactor MONJU", Y. Yamagishi, PNC, Japan
  S. Yazawa, S. Nakadaira, Y. Hayashi & J. Kikushima, Hitachi Ltd. Japan
  (presented by S. Nakadaira)

* "Ultrasonic inservice inspection of PWR coolant pump bowl welds", Ph. Dombret, AIB - VINCOTTE, Belgium

* "Performance of the motors of primary pumps for nuclear reactors", E. Lejeune & J. L. Killian, Jeumont-Schneider, France
  (presented by J. L. Killian)

* "Routine measuring of vibrations as predictive maintenance", D. Meire, Doel Nuclear Power Plant, Belgium
CANDU HEAT TRANSPORT PUMPS –
A QUALITY PRODUCT

A.N. Kumar
AECL-CANDU, Mississauga, Ontario
Canada, L5K 1B2

Presented at the Specialists Meeting on
"Pump Performance and Reliability"
Cologne, Germany
1990 November 26-28
CANDU HEAT TRANSFER PUMPS – A PROVEN QUALITY PRODUCT
Ashok N. Kumar*

ABSTRACT
Consistently high capacity factors achieved by CANDU** reactors during the past few years has placed these reactors amongst the world leaders. A record of such high performance demands an excellent quality of all the major reactor equipment. The heat transport pump motor set represent an excellent example of one such quality equipment by virtue of its proven reliable performance of over 1000 years of pump operation in various CANDU reactors in Canada.

This paper outlines some of the salient design features which enabled the CANDU HT pumps in attaining such high levels of performance and reliability. Design and development background to various critical items in the pump motor set are also discussed.

* Process and Safety Engineering Department. AECL-CANDU Sheridan Park Research Community, Mississauga, Ontario, L5K 1B2, Canada.

** Acronym for CANadian Deuterium Uranium Reactors
1. INTRODUCTION

During the past few years excellent performance of CANDU Nuclear Power Stations has placed these reactors amongst the world leaders. At least four (4) of the top five (5) world ranked reactors have been CANDU reactors (Tables 1 and 2). These rating are based on a lifetime capacity factor.* Consistent achievement of such high performance record demands an excellent quality of all major reactor equipment. Heat Transport (HT) pump motor sets which circulate pressurized heavy water (D$_2$O) coolant through the reactor core represent a prime example of both quality and reliability. After over 1000 years of operation in various CANDU reactor units in Canada there has not been a forced reactor outage due to failure of a heat transport pump.

The reason for such excellent performance record of the HT pump motor sets may be attributed to an integrated design approach covering research and development, material selection, testing, operation and maintenance. The results of inspection of critical pump components at various reactor sites is fed back to the AECL pump design engineering which further refines an already sound design.

Some of the salient features, which enabled the CANDU HT pumps to attain such high standards, are summarized in this paper.

2. HEAT TRANSPORT PUMPS

2.1 General Background

The heat transport (HT) circulate pressurized heavy water (D$_2$O) coolant through the reactor core and the steam generators. The primary function of the HT pumps is to circulate the coolant in order to transfer the heat generated in the reactor core to the light water on the secondary side of the steam generators. The steam thus produced drives the turbine generator set and produces the electrical power. Thus the HT pumps form the "heart" of a CANDU nuclear reactor. Hence a great emphasis was placed on a sound and reliable design of the HT pumps right from the start on all CANDU reactors.

Initially because of lack of operating history, a degree of redundancy was built in the design by having spare HT pumps. Thus the first two CANDU reactor designs (Douglas Point and Pickering) each had spare pumps. However due to increased degree of confidence in the pump design generated by successful operating experience, the later CANDU reactors (Bruce, CANDU 6 and Darlington) eliminated the use of spare pumps. A degree of redundancy was built, instead, into the HT system which enables the reactor operation with one pump failed but with the power output derated to about 70% of the full power.

* Capacity Factor = \( \frac{\text{Actual electricity generated}}{\text{Perfect electricity generation}} \)
# The Top Ten

Lifetime World Power Reactor Performance to June 30, 1987*

from among 339 reactors over 150 MW

<table>
<thead>
<tr>
<th>Country</th>
<th>Ranking</th>
<th>Unit</th>
<th>Type</th>
<th>Capacity Factor %†</th>
</tr>
</thead>
<tbody>
<tr>
<td>🇨🇦 Canada</td>
<td>1.</td>
<td>Bruce 5</td>
<td>CANDU</td>
<td>88.7</td>
</tr>
<tr>
<td>🇨🇦 Canada</td>
<td>2.</td>
<td>Bruce 3</td>
<td>CANDU</td>
<td>86.6</td>
</tr>
<tr>
<td>🇨🇦 Canada</td>
<td>3.</td>
<td>Pt. Lepreau</td>
<td>CANDU</td>
<td>86.5</td>
</tr>
<tr>
<td>🇨🇦 Canada</td>
<td>4.</td>
<td>Bruce 4</td>
<td>CANDU</td>
<td>85.7</td>
</tr>
<tr>
<td>🇨🇦 Canada</td>
<td>5.</td>
<td>Bruce 7</td>
<td>CANDU</td>
<td>85.5</td>
</tr>
<tr>
<td>🇨🇦 Canada</td>
<td>6.</td>
<td>Pickering 7</td>
<td>CANDU</td>
<td>85.1</td>
</tr>
<tr>
<td>🇩🇪 FRG</td>
<td>7.</td>
<td>Philippsburg 2</td>
<td>PWR</td>
<td>84.0</td>
</tr>
<tr>
<td>🇧🇪 Belgium</td>
<td>8.</td>
<td>Doel 3</td>
<td>PWR</td>
<td>83.5</td>
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<td>🇨🇦 Canada</td>
<td>9.</td>
<td>Pickering 8</td>
<td>CANDU</td>
<td>83.2</td>
</tr>
<tr>
<td>🇩🇪 FRG</td>
<td>10.</td>
<td>Grohnde</td>
<td>PWR</td>
<td>82.7</td>
</tr>
</tbody>
</table>

*Source: Nuclear Engineering International
†Capacity Factor = \( \frac{\text{actual electricity generation}}{\text{perfect electricity generation}} \)

TABLE 1
## The Top Ten

Lifetime World Power Reactor Performance to March 31, 1990*  
*from among 339 reactors over 150 MW

<table>
<thead>
<tr>
<th>Country</th>
<th>Ranking</th>
<th>Unit</th>
<th>Type</th>
<th>Capacity Factor %†</th>
</tr>
</thead>
<tbody>
<tr>
<td>FRG</td>
<td>1.</td>
<td>Emsland</td>
<td>PWR</td>
<td>92.4</td>
</tr>
<tr>
<td>Canada</td>
<td>2.</td>
<td>Point Lepreau</td>
<td>CANDU</td>
<td>89.6</td>
</tr>
<tr>
<td>Canada</td>
<td>3.</td>
<td>Pickering 8</td>
<td>CANDU</td>
<td>88.3</td>
</tr>
<tr>
<td>Canada</td>
<td>4.</td>
<td>Bruce 5</td>
<td>CANDU</td>
<td>87.7</td>
</tr>
<tr>
<td>Canada</td>
<td>5.</td>
<td>Pickering 7</td>
<td>CANDU</td>
<td>87.5</td>
</tr>
<tr>
<td>FRG</td>
<td>6.</td>
<td>Grohnde</td>
<td>PWR</td>
<td>87.4</td>
</tr>
<tr>
<td>Hungary</td>
<td>7.</td>
<td>Paks 1</td>
<td>PWR</td>
<td>87.3</td>
</tr>
<tr>
<td>Finland</td>
<td>8.</td>
<td>Loviisa 2</td>
<td>PWR</td>
<td>86.6</td>
</tr>
<tr>
<td>Belgium</td>
<td>9.</td>
<td>Tihange 3</td>
<td>PWR</td>
<td>86.5</td>
</tr>
<tr>
<td>Hungary</td>
<td>10.</td>
<td>Paks 3</td>
<td>PWR</td>
<td>86.4</td>
</tr>
</tbody>
</table>

*Source: Nuclear Engineering International

1Capacity Factor = \( \frac{\text{actual electricity generation}}{\text{perfect electricity generation}} \)
A typical CANDU 6 HT pump and HT system configuration are shown in Figures 1 and 2 respectively. The coolant (Figure 2) passes through one set of steam generators, HT pumps and preheaters through the core in one direction in one half of the reactor core. In the other half the direction of flow is in the opposite direction. Thus a Figure "8" configuration is formed by the flow of the coolant.

2.2 Major Design Features

All HT pumps for a CANDU reactor are vertical single stage, single suction, double volute centrifugal pumps. The pumps either have a single or a double discharge arrangement. The pumps are designed and manufactured to the requirements of ASME Code, Section III, Class 1 and to the quality assurance requirements of the Canadian Standards Association (CSA) Code Z299.1. The pumps for various reactors are designed to operate at different heads and capacities as shown in Table 3. In all reactors the HT pumps have at least 200 to 300 metres of available NPSH. About 90% of the HT pumps on the existing CANDU reactors have been supplied by Byron Jackson Pump Company. KSB and Bingham Willamette Pump Companies supplied the rest.

Each pump is rigidly coupled to an electric motor, which is provided with an upper and a lower guide bearing. A combined up thrust and down thrust bearing assembly is located at the top end of the motor. All the motor bearings are oil lubricated tilting pad bearings. The motor is also provided with a pneumatically actuated brake to prevent its rotation in a non-powered state. A schematic representation of the pump motor rotor assembly is shown in Figure 3. Major design features of the HT pump motor sets are discussed in this section.

2.2.1 Pump Bearing Design

The pump has a single water lubricated bearing located just above the overhung pump impeller. The pump bearing is either of a hydrostatic or of a hydrodynamic type. In case of a hydrostatic bearings, the pump is supplied with pressurized heavy water (D2O) from the pump discharge by means of an auxiliary impeller. The bearing and the journal material selection along with their hardnesses are optimized to provide adequate resistance against wear and galling of the material. This characteristic is particularly desirable when the pumps are required to operate under two-phase flow conditions, where metal-to-metal contact is most likely to occur. The post-LOCA operational capability of HT pumps with hydrostatic bearing has already been established by extensive testing of these pumps (Reference 1). Bruce, Darlington and two of the CANDU 6 reactor stations have HT pumps provided with hydrostatic bearing. These pumps were supplied by Byron Jackson Pump Company from their Canadian plant.
1. UPPER OIL POT COVER
2. THRUST BEARING OIL POT
3. RUNNER
4. THRUST BEARING ASSEMBLY
5. DOWN THRUST BEARING
6. UP THRUST BEARING
7. THRUST BEARING COOLING COILS
8. BRAKE RING
9. MOTOR SHAFT
10. OIL LEVEL CONTROL
11. BEARING COOLING WATER PIPES
12. AIR COOLER WATER PIPES
13. SURGE CABINET
14. AIR SHIELD
15. AIR SHIELD
16. BLOWER RINGS
17. MOTOR FLYWHEEL
18. STATOR CORE
19. ROTOR ASSEMBLY
20. LOWER GUIDE BEARING
21. THRUST DISC
22. SPACER COUPLING
23. MOTOR STAND
24. PUMP SHAFT
25. VAPOUR CONTAINMENT SEAL
26. SECONDARY MECHANICAL SEAL
27. PRIMARY MECHANICAL SEAL
28. PUMP BEARING
29. PUMP CASE
30. CASE WEAR RING
31. PUMP DISCHARGE
32. SUCTION PIPE

FIGURE 1 CANDU 6 HEAT TRANSPORT PUMP
FIGURE 3  TYPICAL HEAT TRANSPORT PUMP MOTOR ROTOR SYSTEM
<table>
<thead>
<tr>
<th>Design Parameter</th>
<th>Decommissioned in 1986</th>
<th>Currently Operating NGS</th>
<th>Design Being Finalized</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>No. of HT Pumps</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Total</td>
<td>10</td>
<td>16</td>
<td>4</td>
</tr>
<tr>
<td>1.2 Operating</td>
<td>8</td>
<td>12</td>
<td>4</td>
</tr>
<tr>
<td>2. Rated Head &quot;H&quot;</td>
<td>160 m (525 ft)</td>
<td>146 m (480 ft)</td>
<td>213 m (700 ft)</td>
</tr>
<tr>
<td>3. Rated Flow &quot;Q&quot;</td>
<td>427.5 l/sec (5700 IGPM)</td>
<td>757.5 l/sec (10100 IGPM)</td>
<td>3270 l/sec (43600 IGPM)</td>
</tr>
<tr>
<td>4. Specific Speed &quot;N&quot;</td>
<td>1350</td>
<td>1920</td>
<td>3020</td>
</tr>
<tr>
<td>Motor HP</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>5.1 Operating</td>
<td>0.93 MW (1250 HP)</td>
<td>1.4 MW (1900 HP)</td>
<td>8.2 MW (11000 HP)</td>
</tr>
<tr>
<td>5.2 Rated (cold)</td>
<td>0.76 MW (1020 HP)</td>
<td>0.76 MW (1575 HP)</td>
<td>6.70 MW (9000 HP)</td>
</tr>
<tr>
<td>6. Distance between Discharge Pipes</td>
<td>Single Discharge</td>
<td>Single Discharge</td>
<td>Single Discharge</td>
</tr>
<tr>
<td>7. Pump Type/Supplier</td>
<td>12 x 12 x 24 DVSS BJ/Elect.</td>
<td>14 x 14 x 24 DVSS BJ/CGE</td>
<td>25 x 25 x 30 DVSS BJ/CGE</td>
</tr>
<tr>
<td>8. Pump Inlet Temperature</td>
<td>244°C (480°F)</td>
<td>244°C (480°F)</td>
<td>266°C (511°F)</td>
</tr>
<tr>
<td>9. Bearing Type</td>
<td>Hydrodynamic</td>
<td>Hydrodynamic</td>
<td>Hydrostatic</td>
</tr>
<tr>
<td>10. Motor Inertia</td>
<td>900 lb-ft²</td>
<td>1500 lb-ft²</td>
<td>300000 lb-ft²</td>
</tr>
<tr>
<td>11. Pump Support</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

* Ontario Hydro were the Responsible Design Engineers for Darlington HT Pumps
** In-service date of the first unit of a multi unit project
On the other hand, HT pumps with hydrodynamic bearings need a guaranteed minimum supply of filtered heavy water (D₂O) at controlled temperature for its successful operation. This bearing water supply is provided by a separate circuit which includes a filtering system, a heat exchanger and a jet pump. The HT pumps for Pickering and three of the CANDU 6 reactor stations are provided with the hydrodynamic bearing. CANDU HT pumps fitted with hydrodynamic bearings were supplied by Byron Jackson, KSB and Bingham Willamette Pump companies.

2.2.2 Mechanical Seals
The mechanical seals for the CANDU HT pumps vary in size from 100 mm’s to about 200 mm’s. General design approach has been to have either a two or a three stage seal assembly in series. Each stage is designed to withstand the full system pressure, although in practice it carries only half (for 2 stage) or one third (for 3 stage design) of the full system pressure. Thus a generous redundancy is built into the seal design. A pressure breakdown device in the gland seal circuit provides the staged pressure across each seal stage. The current seal life for HT pump using CAN-21 seals is 5 years. This has been achieved through extensive developmental effort, as discussed in Section 4.4

2.2.3 Pump Support Arrangement
There are two approaches used in designing the support arrangement for CANDU HT pump motor sets:

a. Pump floating to accommodate the thermal expansion while the header is anchored (Pickering and CANDU-6)
b. Pump is anchored while a floating header accommodates the thermal expansions (Bruce and Darlington)

Figures 4 and 5 show these arrangements.

2.2.4 Seismic Braces
All CANDU HT pump motor sets are provided with seismic braces to accommodate the lateral loads resulting from an earthquake or a postulated burst pipe event. Typically either three (3) (at 120° orientation) or four (4) (at 90° orientation) braces are welded to the motor mount. One restraint is placed against each of these braces (separated by an elastomeric pad) at one end while the other end of the restraint pad is embedded into the building concrete after further reinforcement.

2.2.5 Material of Construction
All materials used in the construction of the pump components are in accordance with either ASME or ASTM standards. All major finished pump components have tight controls in their composition with regards to cobalt concentration (which promotes radioactivity in the reactor components). Since the D₂O circulated through the reactor core is of controlled chemistry, the materials also have controlled concentrations of sulphur, phosphorus and halides.

1. The Can-2 seal design was developed by AECL and is used on Byron Jackson HT pumps for CANDU reactors (except Pickering).
FIGURE 5   TYPICAL CANDU 6 HEAT TRANSPORT PUMP SUPPORT ARRANGEMENT
Over 30 years ago, the AECL engineering team made a major decision to use carbon steel material for pump bowls and heat transport system piping and fittings. This was contrary to more conventional accepted practice of using stainless steel in American Nuclear Reactors. Main basis of this decision was that the carbon steel offers several important advantages over stainless steel. Carbon steel is cheaper, easier to decontaminate and is immune to intergranular stress corrosion cracking. The wisdom of this historic decision is very evident in all CANDU reactor components including the HT pumps. Stress corrosion cracking is non-existent in the major pressure boundary components of CANDU HT pumps. Where stainless steel components are used, their hardness is well controlled to a maximum value of HRC 40. All materials are subject to non-destructive examination.

The hardness of the pump shaft is controlled. The shaft notch radii, where there are changes in diameter, are generous, to avoid any chances of fatigue failures. Furthermore there are no components welded onto the pump shaft. By paying attention to such small details, CANDU HT pumps have grossly improved the quality of the overall design.

3. QUALITY AND RELIABILITY OF CANDU HT PUMPS

The performance history of CANDU HT pumps is in itself an adequate evidence of their reliability. Based on an average capacity factor of 80% for the Pickering nuclear reactors, since 1971, the non-availability of the reactor due to HT pumps was only 0.20%. This high availability of the HT pumps clearly establishes its reliability.

The reliability of HT pumps was achieved by sound fundamental design approach. Performance and reliability were always given preference even if it meant a slight sacrifice in the pump efficiency. It was recognized that a pump failure results in more than $0.30 \times 10^6$ per day in lost power alone. Hence although efficiency is critical from economic considerations, it is only second to its reliable performance.

Similarly a great emphasis was placed on the sound design and reliability of all the wearable items such as the mechanical seals, pump and motor bearings.

Defining the requirements fully and incorporating them through suitable control at various stages of manufacture assembly, testing, installation at site and during operation at the reactor site greatly contributed in achieving an outstanding quality of the pumps.

4. MAJOR CONTRIBUTORY FACTORS

There are several factors which have contributed towards achieving an excellent operating record of the CANDU HT pumps. Sound engineering principles backed by considerable feedback of operational data from various sites greatly assisted this process. The following are some of the major factors:
4.1 Quality of the AECL Technical Specifications

The design of any pump motor set is largely dictated by quality of its technical specifications. Thus a great emphasis is placed by AECL on producing a good quality of HT pump motor specifications. The design requirements of the pump motor set are defined clearly and completely, in sufficient details, in the AECL Technical Specifications. During the tendering period further clarification of technical requirements is provided to the Suppliers, if necessary. Hence the requirements are well understood and correctly interpreted by both the parties. This is the key to the reliable design of the CANDU HT pumps. All the design requirements are fully implemented by close monitoring and quality control procedures.

Some of the major requirements outlined in the AECL pump motor technical specifications permit a control on the following.

Material of construction, major internal pump components, balancing, misalignment, vibration levels, assembly tolerances, performance tests of the pump motor sets, welding, non-destructive examination, various analyses, seismic qualification, dimensional checks and inspection, special tools for maintenance, shipping and packaging.

4.2 Design and Documentation Control

All phases of manufacture are controlled by relevant procedures, inspection and test plans, detailed drawings and various analyses reports. Majority of these documents are subject to acceptance by AECL Design Engineering. The procedures of Supplier’s subcontractors are also subject to comments from AECL. All procedures have to be accepted by AECL at least 6 weeks before their use. Without acceptable procedures or documents, the manufacture cannot proceed.

The pump Supplier is required to provide a detailed Design Report which establishes the conformance of the pump design to the requirements of ASME Code and to the AECL Technical Specification. In addition the pump motor set has to be seismically and environmentally qualified. The pump Supplier is required to establish the capability of the pump to sustain the burst pipe loads. Post-LOCA operational requirements of the pump motor sets also have to be met.

All non conformities during the manufacturing processes and the subsequent corrective actions are subject to AECL acceptance. This is formalized through “Design Disposition Deviation” forms or “Concession Applications”.

Thus although the pumps are designed, manufactured and tested by the pump Supplier, AECL retains a complete control over the correct implementation of the specific design requirements.

4.3 Quality Control Procedures

The Supplier is required to produce a Q/A Manual to show that the manufacture, assembly and testing facilities in the shop conform to the Quality Assurance programs of CSA Z299.1 and ASME Code, Section II for Class 1 pumps. The Supplier’s plant is also audited by AECL to ensure the conformity to the required quality levels, before the award of the contract.
Throughout the manufacture of the pump components, the quality surveillance inspectors ensure that all the approved manufacture, inspection and test plans are followed. This ensures the desired high quality of the pumps.

4.4 Design Development of Major Pump Motor Components

Considerable advances have been made in improved life and reliability of critical pump motor components since the first experimental CANDU reactor NPD was started up about 25 years ago. The seal life at that time was only 200 hours. Since then the HT pump motor horse power has increased to 14,000 from 500 and the shaft diameter to over 200 mm. Two examples of design and development work on pump motor components are presented in this section.

4.4.1 Mechanical Seals

The only pump component requiring period maintenance is the mechanical seal package. A reactor shutdown forced by seal failure would cost about $0.3 \times 10^6 \text{ per day in terms of lost power alone. Hence a reliable and long seal-life is a major factor in minimizing operating costs. To ensure this, a seal development program was initiated in collaboration with the pump supplier. The existing seal designs were modified, the seal face and seal sleeve materials changed and a convergent wedge was built in the seal faces in the direction of seal leakage, using a computer code developed by AECL. This design was extensively tested both in the vertical seal tester at AECL Chalk River Nuclear Laboratories and in heat transport pumps at the manufacturer's plant before the design was accepted.}

Simultaneously another program was initiated at one CANDU station where all the replaced heat transport pump seals were inspected. Initially the seal packages were being conservatively replaced annually. A record of the "as assembled" seal packages, along with a complete list of assembly tolerances, was maintained at the site. These controlled assembly tolerances were further optimized to maximize the seal life based on the results of inspection of replaced seals. It was concluded that the seal packages that were being replaced even after two years service were capable of at least two to three years more of operation. This process of inspection feedback enabled the seal life to be increased to period of five years. This approach incorporates all the design, assembly and operational aspects in assessing the seal life and has resulted in significantly improved reliability. A schematic arrangement of the seal operating faces for Can-2 seals is shown in Figure 6.

4.4.2 Motor Bearing

The motors are provided with oil-lubricated tilting pad bearings. The design parameters of these bearings were also optimized to provide the necessary stiffness and damping to provide acceptable rotor dynamics of the pump motor set. Extensive developmental testing was carried out at the manufacturer's plant and in AECL laboratories to attain acceptable Bruce A motor bearing design characteristics. A high-pressure oil life-pump supplies pressurized oil to the bearings just before starting the heat transport pump motor. Thus bearing wear during the motor start-up is eliminated because of the existence of pressurized oil film separating the bearing surfaces.
FIGURE 6  SEAL OPERATING FACES
4.5 Condition Monitoring

The pump motor set is provided with extensive instrumentation (Figure 7) to continuously monitor the health of the pumps throughout their operation. Most of the monitored parameters are displayed in the Control Room. Every monitored parameter indicates some aspects of the “health” of the pump motor set. From the very outset, AECL set tight but realistic acceptance limits on certain monitored parameters which are crucial to reliable operation of the pump motor sets. These include:

- vibration levels at various bearing locations,
- shaft run-out at the pump coupling,
- motor bearing temperatures
- motor stator winding temperature
- seal cavity temperatures and inter stage seal pressures.

The acceptance limits on these parameters are specified in the AECL Technical Specifications. During operation of the pumps at the reactor site, the acceptance limits are classified into “Normal”, “Alarm” and “Shutdown” categories. The pump motor sets must operate within the “Normal” limits at all times. If “Alarm” level is reached, immediate corrective action is necessary. However if the “Shutdown” limit is reached the relevant pump motor set has to be shutdown as soon as practical or the pump trips if the parameter dictates an automatic trip.

One advantage of this monitoring is that the pumps are always operated under known and acceptable conditions. This assures the pump motor set of achieving its design life. In addition any operating problems can be diagnostically resolved from the monitored data. As can be seen from Figure 7, the extensive monitored parameters provide a complete health status of the heat transport pump motor sets. This unique condition monitoring has provided a tremendous boost to the reliability of the pump motor set.

4.6 Pump Performance Tests

The intent of performance tests on the CANDU HT pump motor sets is to verify that the sets as designed and assembled fully comply with the AECL specified requirements. The tests are extensive and simulate various operating conditions including start/stop cycles in addition to stop/depressurize, repressurize/start cycles. The pump characteristic curves provide the hydraulic data on the pumps. Various monitored parameters provide the data regarding the performance of the pump motor sets. The pump motor set is disassembled after the tests and is inspected to meet the specified acceptance criteria.

The accepted pump motor sets are installed at the reactor site to comply with various elevations and tolerances specified in the detailed installation procedure provided by AECL.

The pump motor set are subject to various commissioning tests after installation at the reactor site. This is to ensure that the pumps “as installed” at the reactor with various attached piping has an acceptable performance in the HT system loop.
FIGURE 7  HEAT TRANSPORT PUMP SIMPLIFIED INSTRUMENTATION
This extensive testing to verify an acceptable pump performance before putting the reactor in-service ensures a very high degree of reliability in the pump motor sets as installed at the reactor site. It is then left up to the operating staff to ensure the continued sound operation of the pumps by good maintenance practices.

4.7 **Operation and Maintenance Practices at the Reactor Site**

Detailed operation and Maintenance Manuals for both the pump and the motor are provided to the site personnel which ensures that the pump motor set is always maintained and operated in accordance with consistent and acceptable procedures. There is a wide range of special tools provided to assist the site maintenance staff. These special tools include:

- **Seal removal device**: enables seal replacement within 5 hours involving removal of only the pump half coupling.
- **Shaft Hold down device**: enables the replacement of the pump motor in a pressurized state.
- **Coupling Removal device**: assists in coupling removal and assembly by hydraulic tensioning.
- **Stud tensioner**: enables in assembling the case to cover bolts (75 mm diameter) by hand tightening and vice-versa.
- **Lifting Gear**: capable of lifting the coupling components, the hold-down device and the seal assembly and transporting them through the access opening in the motor mount.

Thus the pump motor set is assembled and operated within the specified tolerances with the right tools provided for each operation. This greatly assists in smooth and predictable operation of the pump motor set.

4.8 **Feedback of Operational Data**

For the successful operation of any sophisticated equipment, such as the CANDU HT pumps, it is essential to have a continuous feedback of information between the designer and the user. There is a “CANDU Owners Group” (COG) set up in Toronto, Canada, which provides an excellent feedback of all the data from various reactor sites to the AECL designers. This makes the design team to device improved concepts where a deficiency is observed. The deficiency could be related to either equipment operation, instrumentation, material or the system operating conditions.

The ‘COG’ also funds research and development programs, where necessary, which may be beneficial to the utilities in both short and long terms. The experience and lessons learnt from resolution of problems at one reactor station is feedback to other nuclear stations. Thus all generic problems can be resolved expeditiously and economically.
The operational feedback data from one reactor site has assisted in development of a sound seal design (CAN-2) for the HT pumps. The current reliable seal life of 5 years for the Can-2 seals was achieved by deriving the critical design parameters and incorporating them into the seal design and assembly. These specific features were obtained from the results of inspection of seals which operated in the reactor HT pumps for durations varying between 1 and 3 years.

Lessons learnt from operational experience also resulted in:

a. improved design of the pump support arrangement,
b. improved choice of material of construction for some pump components,
c. improved assembly procedures of the pump motor assembly and
d. improved instrumentation reliability.

5. CONCLUSIONS

Some major design features which enabled the development of a highly reliable performance of the CANDU HT pump motor sets have been presented in this paper. Ontario Hydro is the major user of the CANDU reactors (Table 3). Significant funding and research efforts by Ontario Hydro in promoting the technical excellence and development of HT pump motor components is acknowledged. Without this strong backing of Ontario Hydro, the current performance of the HT pump motor set may not have been possible.

Close cooperation between the Design Engineering team, the pump and motor suppliers and the Canadian Utilities, spearheaded by Ontario Hydro, was another key factor to the success of this product.

Lastly, the dedicated effort of the Design Engineering team as the driving force behind developing and implementing the design process as outlined in this paper was a major factor in establishing the CANDU HT pump motor set as a quality product.

Acknowledgement

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References

Design and Manufacture of Primary Sodium Pump for the Prototype Fast Breeder Reactor "MONJU"

Yoshiaki Yamagishi
Power Reactor and Nuclear Fuel Development Cooperation (PNC)

Setsuo Yazawa, Shiro Nakadaira, Yojiro Hayashi, Jun Kikushima
Tsuchiura Works, Hitachi, Ltd.

Abstract

The circulating pump for the primary heat transport system of the prototype fast breeder reactor "MONJU" circulates high-temperature (approx. 670° K (400°C)) liquid sodium. This pump must perform satisfactorily under all power plant operating conditions. Therefore, this pump is one of the most important components and must be highly reliable.

Hitachi has been carrying out development programs concerning mechanical sodium pumps for over 20 years on consignment from Power Reactor and Nuclear Fuel Development Corporation (PNC). This includes the construction and operation of the experimental fast breeder reactor "JOYO", and the execution of endurance tests on the mockup pump for "MONJU". The primary sodium pump of "MONJU" was designed based on the results of R & Ds, and it was manufactured and inspected under strict quality control standards which agree to law regulations and classifications of plant operation. Also, its design requirements have been confirmed. The pump has undergone in-water shop tests and is now under construction at the plant site.

1. Introduction

The primary heat transport system’s circulating pump (called primary pump hereafter), is used to circulate the primary coolant and to supply the reactor core with a discharge necessary for removing the core heat during the plant’s operation as well as during a low-temperature shutdown or accident.

The primary pump circulates radioactive liquid sodium and must function under various conditions, such as sodium temperature changes from approximately 470° K (200°C) to 670° K (400°C). The pump must also control a wide range of discharge from approximately 10% to 100%.

Therefore, Hitachi has promoted research and development related to the primary pump: developing basic technologies of dealing with radioactive high-temperature liquid sodium, developing technologies on each component of the pump, and executing endurance tests on the mockup pump.

All of this is to be based on domestic technology and in accordance with the fast breeder development plan of the national project led by Power Reactor and Nuclear Fuel Development Corporation (PNC).

The primary pump for the fast breeder reactor MONJU power plant (shortened to MONJU hereafter), whose design was based on the results of various research described above, has come to the present point of completion under a special manufacturing system of securing a high-quality reactor heat transport system pump.

In this document, an overview of the progress in research concerning the primary pump, the contents of the designing, manufacturing, and testing of MONJU’s primary pump will be stated.

2. Characteristics of the Primary Pump

The principal specifications of MONJU’s primary pump are shown in Table 1. The pump itself is shown in Fig.1. The basic structure of the primary pump includes the hydrodynamic components (impeller and diffuser), the shaft, the bearings and the shaft sealing elements.

The pump is also equipped with parts specifically for a fast breeder reactor plant in which the coolant is radioactive high-temperature liquid sodium.
Table 1 Principle specifications of WOJU's primary pump:

<table>
<thead>
<tr>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
</tr>
<tr>
<td>Mechanical Vertical</td>
</tr>
<tr>
<td>Centrifugal</td>
</tr>
<tr>
<td>Capacity</td>
</tr>
<tr>
<td>Approx. 100 m³/min</td>
</tr>
<tr>
<td>Pump head</td>
</tr>
<tr>
<td>Approx. 92 mNa</td>
</tr>
<tr>
<td>Rated speed</td>
</tr>
<tr>
<td>Approx. 840 rpm</td>
</tr>
<tr>
<td>Operating temperature</td>
</tr>
<tr>
<td>Approx. 670°C (400°C)</td>
</tr>
<tr>
<td>Major dimensions</td>
</tr>
<tr>
<td>Shell's outer diameter</td>
</tr>
<tr>
<td>Approx. 1.8 m</td>
</tr>
<tr>
<td>Overall height</td>
</tr>
<tr>
<td>Approx. 10.6 m</td>
</tr>
<tr>
<td>Main material</td>
</tr>
<tr>
<td>Austenite stainless steel JIS SUS304</td>
</tr>
<tr>
<td>Pumps per plant</td>
</tr>
<tr>
<td>3</td>
</tr>
</tbody>
</table>

(1) Double casing is used so that the internal structure of the primary pump can be pulled out as a whole to make overhaul possible.

(2) The primary pump has a long shaft structure in which the impeller and lower bearing are immersed in liquid sodium. This enables the necessary core heat removal flow to be secured even at the minimum free liquid level, providing for leaks in the primary heat transport system.

(3) The free liquid level is created in the pump, because it is difficult to directly seal the shaft of high-temperature liquid sodium. The upper part of the casing is covered with inactive argon gas (called the cover gas hereafter), which is sealed off. The overflow nozzle is placed at the middle section of the casing to restrict an increase of the liquid sodium level.

(4) The sealing device consists of two mechanical seals, one upper and one lower. The double seal structure contains oil between the seals, thus completely shutting out the cover gas and air. A bellows seal is used for the lower mechanical seal especially to improve reliability.

(5) In order for the upper face of the pump to be accessible, the thermal shield plate is installed on the cover gas layer of the upper part of the free liquid level, and the γ-ray shield plug is installed on top of this plate.

(6) The pump material is austenite stainless steel (JIS SUS304), which is highly compatible with sodium.

(7) The electric heater and the heat insulating material are installed on the outside of the casing, so that a temperature increase can be achieved before the sodium is charged.

Furthermore, in order to maintain a higher reliability in the pump function, the following points were considered in the design.

(1) The natural convection-prevention plates are installed on the ring-shaped gas space area of the double casing to prevent bending in the casing due to nonuniform temperature distribution in the circumferential direction.

(2) A hollow, large-diameter shaft is employed, so that the long shaft does not resonate during any operation situation required of this pump and that a high thermal shield effect may be obtained.

![Diagram](image.png)
3. Details of Primary Pump Development

One of the major problems in the development of the fast breeder reactor is to establish the technology of using high temperature liquid sodium as a coolant. This research has been promoted as a national project with PNC as the leading group.

The primary pump is also one of the major devices developed by Hitachi. As the only rotating device directly dealing with radioactive high temperature sodium, this pump has, through the scale-up process, gained high reliability and large capacitance. Hitachi, which was granted with the Consignment Research Expenses On Peaceful Uses of Atomic Energy in 1965 and 1966, completed the first domestic pump as an experimental model. With this as a starting point, as shown in Fig. 4, Hitachi has manufactured four types of pumps in the scale-up ratio, which are about 5 times in the flow ratio from the previous model. In parallel with manufacturing and operation research of these sodium pumps, Hitachi has conducted a wide range of research in the fields of fluid performance, material, heat, vibration, measuring, bearing, shaft seal and cleaning.

(3) A static pressure bearing with an enlarged bearing gap is employed as the lower bearing because it is used at a high temperature. This bearing is lubricated by liquid sodium whose pressure is raised by the pump.

Details of the hydrodynamic components and the lower bearing are shown in Fig.2. The upper bearing installed between the upper and lower mechanical seal is a ball bearing which uses sealing oil as the lubricant.

The flow control of the primary pump is carried out by changing the power frequency by means of the W-G (motor-generator) set and then by controlling the rotation speed of the main motor. Besides this main motor, the primary pump is equipped with a pony motor (output 22kW) used for core decay heat removal operation and a turning motor (output 1.5kW) used to measure the starting torque and to straighten the thermal deformation of the pump rotor. These motors are operated by means of the overrunning clutch on the upper part of the main motor. Fig.3 shows the driving mechanism of the primary pump.
The progress of the research and development is shown in Fig. 5.

Development of the natural convection prevention technology of the cover gas part is introduced here as a characteristic research example related to the pump's function.

At first, in the MONJU mockup pump test, a large temperature difference occurred in the circumferential direction of the casing of the cover gas layer. Furthermore, it was confirmed through experiments that this temperature difference lead to thermal deformation, which in turn caused an excessive load on the pump driving system.

Therefore, the cause of this occurrence was investigated, and the actual temperature difference occurring in the circumferential direction was quantitatively researched by means of size reduced visible model experiments and actual size model tests. As a result, it was understood that the temperature difference in the circumferential direction was caused by the natural convection of the cover gas which had occurred in the ring-shape space between the inner and outer casings of the cover gas. In order to prevent this convection, installing strip partition plates (convection prevention plates) on the relevant ring-shape space was found to be effective.

The pattern of the natural convection is shown in Fig. 6. The distribution of the temperature difference in the circumferential direction on the mockup pump is shown in Fig. 7. As a result of installing the above described convection prevention plates on the relevant ring-shape space, the maximum temperature difference of the outer casing was reduced from the initial 74° K to about 10° K. This made it possible to substantially reduce the thermal deformation quantity of the casing.

The influence of the natural convection at the ring-shape space on casing deformation has become prominent through scaling-up from the fast breeder reactor JOYO (shortened to JOYO hereafter) to MONJU. This phenomenon is a good example of showing that the actual liquid mockup test, which has simulated the temperature conditions, is important for high temperature
### Progress of research and development related to the sodium pump for fast breeder reactors

For the past 20 years, research and development have been conducted on the main body and each component of the sodium pump for fast breeder reactors. The pump has been scaled up by 5 times the previous model.

#### Pump Development
- **1m³/s Experimental Model (First Model in Japan)**
- **1m³/s Sodium Testing Model**
  - Manufacturing
  - Operation

**JOYO mockup**
- Manufacturing
- Operation

**JOYO actual pump**
- Manufacturing
- Operation

**MONJU mockup**
- Manufacturing
- Endurance test

**MONJU actual pump**
- Adjustment
design
- Manufacturing
- Preparation design
- Plan design
- Manufacturing
design

#### Fluid performance

#### Material research

- **Structural strength research**
- **Thermal study and research**
- **Vibration study and research**
- **Bearings and shaft seal**
- **Instrumentation technology**
- **Maintenance and cleaning**

- **Manufacturing technology**
- **Driving mechanism**

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**Devices and problems**

- Devices and problems come to the surface for the first time through scaling-up.

- Furthermore, a large volume of precious information has been obtained on measures for preventing sodium attachment around the shaft, on the thermal shield plate design method, and on improved reliability of static pressure bearing in sodium through high-temperature sodium tests using the full-scale model. The operation results of the pump showed that a high reliability was obtained through sodium in actual flow rate tests. Over 70,000 hours of operation experience in JOYO's primary pump and 23,700 hours of endurance tests in MONJU's mockup pump were obtained. The results of the above tests have been used in designing and manufacturing MONJU's primary pump.

**Ring-shape space natural convection pattern based on reduced visible model experiments**

Reduced visible model experiments confirmed that when the lower part is heated, an ascending flow occurs in one part of the space and a descending flow occurs in the location symmetrical to it by 180°.
4. Manufacturing the Primary Pump

As a nuclear reactor heat transport system, strict quality control regulations were applied to manufacturing MONJU’s primary pump. In manufacturing the primary pump, the classification of the plant’s operation was put into consideration, instead of simply satisfying law regulations. Also, efforts were made to secure high quality by using a special manufacturing system and manufacturing knowledge as described below.

(1) Educational lectures were held for all the managers and employees involved in manufacturing to help them increase their awareness of the importance of the pump.

(2) A separate area was dedicated to the primary pump. Specialized working garments and shoes were used to distinguish this pump from other products, thus creating a specialized working environment.

(3) When processing the parts, effort was made to protect parts in the manufacturing process. This included applying protection sheets on the material surfaces not relevant to the processing and using specialized boxes (colored differently for different materials) for transporting the materials.

(4) Computers were used for managing the history of the pump parts, thus strengthening quality control.

(5) Special jigs/tools were used in many cases to minimize the unevenness in manufacturing precision among the three primary pumps and to secure the centering precision.

(6) Careful attention was paid to the bending in the shaft, for example conducting vibration tests in the same high temperature environments as the conditions for actual use. When processing, the offset of the hollow part was strictly restricted to secure a high-precision balance. Also, the wall of the shaft was thickened at several locations to facilitate revision of the imbalance. The shaft processing situation is shown in Fig. 8. The high temperature vibration test situation is shown in Fig. 9.

---

**Fig. 7** DISTRIBUTION OF CIRCUMFERENTIAL TEMPERATURE DIFFERENCE ON THE MOCKUP PUMP

By installing the strip partition plates on the ring-shape space part, it was confirmed that the maximum temperature difference of 74° K in the circumferential direction was reduced to about 10° K.

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**Sodium loop temperature: 663° K (390°C)**

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**Note:**

- Before installing convection prevention plates
- After installing convection prevention plates
(7) Parts related to the hydraulic performance, such as the impeller and the diffuser, were controlled with a dimensional tolerance stricter than that specified by the Japan Industrial Standards (JIS). The impeller finishing situation is shown in Fig. 10.

(8) Colmonoi, which contains little cobalt, was used as a reinforcement of the bearing surface to reduce the quantity of radiation during repairs. Technology on cladding methods and defect detection methods was established before using colmonoi to assure its reliability as a bearing sliding material.

(9) The large-scale cleaning facilities dedicated to the pumps in the shop were installed. The pumps were cleaned after confirming with test pieces that no problems existed in the cleaning process.

Careful attention was paid to each manufacturing process from the material stage to welding, heat treatment, machining, assembly, testing and preparation for shipment. The internal assembly body of the primary pump manufactured under the control described above is shown in Fig. 11.

About two and a half years were required to manufacture the pump from the arrangement of materials to the water tests. In the meantime, witnessed inspections of material checking, welding inspections, non-destructive inspections, and pressure, dimension and performance tests were carried out at the shop by institutions, such as PNC, to make sure that the soundness and reliability were sufficiently secured before the product was completed. These records are considered to mean that the manufacturing system of the mechanical sodium pump for fast breeder reactors has been established and may be used with high confidence in designing and manufacturing the primary pump for the demonstration reactor.
5. Shop Tests

In the shop, pump performance tests driven by the main motor or pony motor were conducted with specialized closed loop test facilities using demineralized water at room temperature. Also, the function for controlling the pump rotating speed was confirmed by combining the main motor with the actual M-G set.

The test loop system is shown in Fig. 12. The pump testing situation is shown in Fig. 13. The pressure adjustment tank was installed to adjust the suction head for the NPSH (Net Positive Suction Head) characteristic test. A cooler was installed to keep the loop water from rising by the main motor and maintaining it at room temperature.
Fig. 12. SYSTEM CONFIGURATIONS OF PLANT PERFORMANCE
TEST LOOP:
The closed loop type was used to control the temperature and
good of the water. The main motor was operated in
combination with the M-G set.

Cooler
Pressure adjustment tank
Main pipe
Main motor
Pony motor

Refrigerating machine
Oil unit

Fig. 13. WHOLE VIEW OF SHOP PERFORMANCE TEST FACILITY:
The water test loop and auxiliary facilities are
installed on the floor level. The primary pump body
is placed in the pit below the floor.
The test results for Q-H characteristics in the 100%, 70%, 50% and 30% rated rotating speed and pony motor operations are shown in Fig. 14 and Fig. 15. In all three pumps, the results sufficiently satisfied multiple-point specifications as predicted in the design, without revising the impeller.

Characteristics between pumps showed satisfactory correspondence, proving the adequacy of quality control in manufacturing.

From designing to manufacturing and shop testing of the primary circulating pump, experience obtained over many years through JOYO at the PNC O-ARAI Engineering Center and in various sodium device test rooms have been used.

Concerning the maintenance after commencing commercial operation of the power plant, experience at the above-mentioned facilities, such as developing technologies in the remote control cleaning apparatus dealing with sodium, has been reflected from the design stage.

6. Conclusion

MONJU’s primary pump is a type not blessed with many manufacturing opportunities. The product was completed after research and development for over 20 years. This primary pump has the largest capacity for its kind in Japan, and as one of the major components of the fast breeder reactor, the completed product has become an object of strong interest among people engaged in the development of the fast breeder reactor. The manufacturing situation of this pump has currently been observed over 50 times by hundreds of people wanting to see the workmanship of the product. Presently, preparations are being made for in-sodium trial operations of this pump which is in the process of being installed on the site. It is considered that the manufacturing experience will be used in designing and manufacturing the next generation sodium pump for the demonstration reactor, in combination with current technology and the experience that will be obtained during on-site operation.
ULTRASONIC IN-SERVICE INSPECTION OF PRR COOLANT PUMP BOWL WELDS

CSNI Specialist Meeting on Pump Performance and Reliability
OECD NEA, Cologne (D), November 1990

Ph. DOMBRET
ULTRASONIC IN-SERVICE INSPECTION OF PWR COOLANT PUMP BOWL WELDS

Ph. Dombret
AIB-Vincotte, Brussels (B)

Abstract

The volumetric inspection of PWR main coolant pump bowl welds is difficult because of their heavy thickness and of the cast austenitic steel used for the pump bowl fabrication. An ultrasonic technique has been developed, which allows such examinations to be conducted from the outer surface of the pump. The method is based on local immersion focusing probes, which are operated automatically by a dedicated instrumentation. The paper reports on the capability of the method, and on the first field inspection.

INTRODUCTION

Most pressurized water reactor main coolant pump casings are made from cast austenitic stainless steel. Before steel makers became able to manufacture the entire pump bowl in a single weldless piece, electroslag welding was used to assemble the three, or later two, parts moulded separately.

Although radiography may not be considered as the optimal inspection technique for such heavy thickness components, it has been used [1] to comply with the Section XI of ASME code [2], which prescribes a volumetric examination of the weld(s) of one pump per unit every ten years. A practical further drawback of the method lies in the time-consuming prerequisites, such as draining of the pump and disassembling of rotor and water guide. In addition, radiography can be inadequate as to the geometric design of the pump bowl.

A study, conducted by EPRI to evaluate the technical need for non-destructive examination of those welds during the plant life [3], concluded that, for the considered pump types, the interval between inspections could be safely extended to 15 or 20 years. This led several utilities to seek - and obtain - from their Safety Authorities deferral of the 10-year inspection, although no long term solution was hereby provided.

The following describes an alternate ultrasonic technique, to be carried out from the outer surface only, hereby reducing significantly the inspection cost as well as the radiation doses incurred by personnel.
PWR MAIN COOLANT PUMPS

The inspection method described below refers to the most common two-weld or one-weld pump bowl geometry in 900 MWe three-loop PWR's of Westinghouse or Framatome design. Recent investigations have shown that adaptation to smaller (two-loop) reactor pumps would be relatively easy.

The development was primarily undertaken for the Tihange 1 nuclear unit (900 MWe Westinghouse PWR), taking benefit from the outcomes of a laboratory study carried out several years ago [4]. Fig. 1 illustrates the geometry of the relevant pump bowls: the casing elements made from cast austenitic steel (SA-351 CF8A) are assembled by electroslag welding. Both the inner and outer surfaces are manually ground flush, and the actual wall thickness ranges from 180 to 200 mm.

A steel sample cast with the pumps was made available for the examination method development; that flat block is 200 mm thick and contains no weld. Another specimen, with a portion of a casing lower weld and containing simulated fabrication defects, but made from a different material, was kindly lent by Electricité de France.

Fig. 1 - PWR Main Coolant Pump Bowl
MATERIAL CHARACTERIZATION

Extensive data were collected on the Tihange 1 pump and on the available specimens to evaluate the representativity of these. Metallurgical measurements (delta-ferrite content, chemical content, and, where practical, macrography) and acoustic measurements (peak amplitude and frequency content of backwall reflection) enabled to conclude that, resulting from a high ferrite percentage and a comparatively small grain, the energy scattering rate is much lower in the curved block than in the real pump bowl.

On the contrary, the pump material, in which the weld and the parent metal appear quite similar acoustically, is fairly well represented by the flat block, provided that the curvature difference is taken into account through corrections. These were quantified by calculations and by measurements carried out on a custom-made carbon steel sample. Consequently, most of the method development work, as well as the reference calibration of the equipment, were achieved on these two blocks.

Regarding the design of the examination technique, the acoustic experiments showed that correct propagation of sound pulses through the coarse structure of the weld and cast materials can be achieved only with low frequency (0.5 MHz or lower) compression waves.

Fig. 2 – TRL (left) and Focusing (right) Transducers
PULSE-ECHO EXAMINATION TECHNIQUE

The first layer, in the through-wall direction, can be conveniently examined by contact TRL transducers. Such probes were manufactured, based on 0.5 MHz crystals, refracting compression waves at 45° in steel (Fig. 2). A standard search unit of the same frequency was selected for the straight beam examination. As however the efficiency of the TRL technology is limited in depth, focused beam transducers were specifically developed for the 60-200 mm depth range (Fig. 2). The required focal characteristics led to 0.5 MHz piezoelectric elements with a diameter of 140 mm, backed by a heavy damping material. Each transducer is mounted on a local immersion chamber, providing for efficient coupling and minimizing leakage, and supports its own pulser-preamplifier triggered by the data acquisition instrument.

Several such straight beam and angle beam (30°) compression wave transducers were manufactured to cover the required depth range, with respect to the various surface curvatures to be considered. Acoustic beam dimensions have been measured in steel to be from 15 to 25 mm in the focal depth range.

Fig. 3 shows two angle probe sensitivity curves, which are the references actually used for the examination of the lower weld zone. The curves result from data collected on 9.5 mm diameter (4 mm for the TRL) holes drilled in the flat block, and corrected for the curvature difference. The diagram also displays the peak amplitudes obtained from a 4 mm deep milled notch and from a 20 mm deep slot machined by spark erosion. Fig. 4 illustrates the A-scan signal generated by the latter defect: the upper tip echo can clearly be distinguished from the corner reflection, allowing for accurate sizing.

Fig. 3 - Sensitivity Curves of Angle Beam Probes

Fig. 4 - A-Scan Display of Spark-Eroded Slot
PITCH-CATCH EXAMINATION

Near to the inner surface, it may be difficult, particularly in a material featuring high metallurgical noise and acoustic velocity variations, to discriminate between crack tip signals, corner reflections, and specular echoes from volumetric defects, whereas the relevance of those reflectors as to the integrity of the component is quite different. Therefore, a pitch-catch mode configuration, with two focusing probes, can be implemented to examine the far-surface region.

As the distance between emitter and receiver is set manually, wide areas cannot be examined conveniently, but nevertheless the pitch-catch setup can be very helpful locally, where pulse-echo data call for further flaw characterization.

The pitch-catch disposition is illustrated by Fig. 5.

Fig. 5 – Test of the Pitch-Catch Mode on the Pump Mockup
IMPLEMENTATION AND QUALIFICATION

The huge size and weight (about 14 kg) of the focusing probes request mechanized scanning. Two dedicated manipulators (one for each weld), confined within the available clearance of 150 mm around the pump body, were developed. The motions are driven by a control unit, that also digitizes and processes the ultrasonic data. The inspection results are displayed on a colour graphic terminal, as B, C and D-scan views.

According to the ASME Section XI requirements, angle beam scanning is conducted under four orthogonal directions; defect characterization in pitch-catch mode can also be performed along the same orientations.

The operation of the equipment was assessed on a polyester full-scale mockup (Fig. 5), in which windows are accommodated to permit the insertion of the available steel blocks. The installation eventually allows to calibrate dynamically the equipment in the laboratory, so that field operation can rely on simple sensitivity checks. A polyethylene sample was made for that purpose.

The mockup was also used to qualify the method and the supporting instrumentation: the actual accuracy and the variability of many parameters could be measured under fairly realistic conditions; in addition, conservative figures were postulated where no experimental data could be obtained. The combination of those individual contributions resulted in reflector location and sizing estimates.

Some results of the analysis are presented in Fig. 6, where the ideal conditions refer, by way of reference, to a carbon steel flat component. The two scanning step values considered (4 and 1 mm) are those specified for the regular scanning and for the characterization of indications. The diagrams do not consider the possible influence of the flaw morphology, nor the capability of the particular sizing technique.
FIELD EXPERIENCE

The first field implementation was achieved in Tihange 1, during a refueling shutdown. Penetrant testing was applied on the outer surface, to detect possible near-surface flaws, and the TRL probe inspection was carried out manually.

Focusing probe examination was performed, according to schedule, with no major trouble (Fig. 7). In practice, however, the pump outlet nozzle and the supporting feet prevent scanning of a part of the weld zone. The focusing transducer examination detected some indications, which turned out to be of no concern as to the integrity of the component.

The whole inspection, including the installation and the dismantling of the UT-equipment, was successfully completed in two weeks. Equipment driving requires two operators with an appropriate training.

Further applications of the method are currently envisaged on different coolant pumps.

Fig. 7 - Field Examination at Tihange 1

a) Scanning of the Pump Bowl Upper Weld
b) Remote Control System
SUMMARY

An ultrasonic technique, based on large focusing transducers, has been developed for the volumetric examination of PWR pump bowl welds.

An evaluation of the examination capability has been conducted with all available means, and resulted in the qualification of the method and of the associated automatic instrumentation. Field efficiency has been demonstrated by the Tihange 1 performance.

As the inspection is carried out from the external surface, opening the pump and dismantling internal parts are not required. Consequently, the inspection can be completed within the duration of a refueling outage.

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PERFORMANCES OF THE MOTORS OF PRIMARY PUMPS FOR NUCLEAR REACTORS

E. LEJEUNE - J. L. KILLIAN

JEUHOND - SCHNEIDER INDUSTRIES
Design, control of equipment in operating conditions and preventive maintenance are key points acting on reliability of a mechanical device. After a brief description showing the main solutions chosen for French primary pump motor sets, we will speak about some parameters taken into account in our manufacturing and maintenance programme.
In the PWR, water circulates between reactor core where the fuel is placed and the steam generators: this circuit is called primary circuit.

This water circulates through the primary pumps cools the fuel assemblies in the reactor vessel and transfers the calories to the exchangers which are going to produce the steam in the secondary circuit supplying the turbine.

This figure represents the operative conditions of this systems where the main elements of the primary circuit are shown:

- vessel, fuel, control rods, pressurizer, piping, primary pump and steam generator.

You can see that the whole primary circuit is closed in the tight reactor building and the primary pumps are located on the cold leg. They take suction from the steam generators to discharge water into the vessel.

Since the primary pumps carry this coolant they are very important for the power station in terms of good and safe operative conditions of the power station.

We want to insure the best service and it was the basic care for designing the primary motor pump sets because it is clear that any unexpected outage of the equipment means a lack of current production.

Primary pump driving motors play, as well as primary pumps itself, a key role in power plant operation and safety.

After a short description of the equipment and its evolution we shall treat some subjects which show that the factor of unavailability of a nuclear power station because of a pump driving motor failure is practically equal to zero.
We only display the 3 main types which equip the main part of our French nuclear power stations.

We can observe that the increase of the power of each unit requires the development of appropriate motor pump sets. After a description of the set, we will deal with the main parameters and more particularly with motor parameters which have to be carefully analyzed and met so that the equipment can be quite reliable and the unavailability ratio practically equal to zero.
<table>
<thead>
<tr>
<th>UNIT POWER (Mw)</th>
<th>900</th>
<th>1300</th>
<th>1450</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of loops</td>
<td>3</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Flowrate (m³/hour)</td>
<td>22700</td>
<td>22890</td>
<td>24500</td>
</tr>
<tr>
<td>Head (m)</td>
<td>78</td>
<td>99</td>
<td>106</td>
</tr>
<tr>
<td>Shaft power (Kw)</td>
<td>5000</td>
<td>5850</td>
<td>6600</td>
</tr>
<tr>
<td>- hot conditions</td>
<td>5000</td>
<td>5850</td>
<td>6600</td>
</tr>
<tr>
<td>- cold conditions</td>
<td>6600</td>
<td>7900</td>
<td>8900</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Type of the motor</th>
<th>Squirrel cage induction motor</th>
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</thead>
<tbody>
<tr>
<td>Rated power (Kw)</td>
<td>5300</td>
</tr>
<tr>
<td>Set inertia (kg m²)</td>
<td>3730</td>
</tr>
<tr>
<td>Motor weight (T)</td>
<td>43</td>
</tr>
<tr>
<td>Amperage (A)</td>
<td>500</td>
</tr>
<tr>
<td>- in hot conditions</td>
<td>500</td>
</tr>
<tr>
<td>- at start-up</td>
<td>3750</td>
</tr>
<tr>
<td>Efficiency (%)</td>
<td>91,8</td>
</tr>
<tr>
<td>Insulation class</td>
<td>B</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Number of operating machines</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>France</td>
<td>84</td>
<td>72</td>
</tr>
<tr>
<td>Export</td>
<td>12</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Number of machines with manufacturing in progress</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>France</td>
<td>8</td>
</tr>
<tr>
<td>Export</td>
<td>8</td>
</tr>
</tbody>
</table>

TABLE I

Main characteristics of primary motor pump sets
* Build to obtain heating characteristics of class B
This figure shows a whole motor pump set.

The water in the primary circuit flowing through the primary pumps has a temperature of about 280° and pressure about 155 bars.

Since the pressure is very high in this circuit, the outer wall of the pumps form a pressure vessel and meet the French regulation for nuclear pressure vessels.

The vertical axis set has an overall height of 8.3 m and a diameter of 2.7 m at the level of the pump casing; the total weight of the pump is about 90 tons: 48 tons for the pump element and 42 tons for the motor.

On the figure the following subassemblies are displayed:

- the hydraulics (casing, diffuser, impeller)
- the thermal barrier
- the pump bearing
- the sealing system

for the motor:

- the electric equipment (rotor - stator)
- a lower guide bearing
- an upper guide bearing and a double thrust bearing
- an anti reverse rotation device
- the cooling system

The electric motor which drives the pump is a squirrel cage induction motor with direct on line start and supplied with 6600 volts.

The function of this motor is therefore easier and safer on one hand and the supplying network is simpler on the other hand.

Whereas this motor is similar to common motors, its design is special because of the high inertia level required from the motor pump.

The rotor of the motor has the largest possible diameter in compatibility with the material resistance.

Motor cooling is ensured by ambient air circulation but water cooling systems are mounted on the air return system so as to avoid evacuating motor losses into reactor room.
This solution has been selected to limit the ventilation in the building and to use the air volume in the room to control the temperature in case of loss of cooling water.

The following elements compose the electrical part:

**STATOR**

The magnetic core of the stator is composed of a stacking of high silicon steel laminations, insulated on both faces.

The winding is composed of rectangular section copper conductors which are insulated with mica tape.

They lie in the magnetic core slots and the end turns are firmly braced to support rings so as to withstand the stresses during full-voltage start.

The stator is impregnated in a tank after preheating. In high vacuum atmosphere the stator is covered with epoxy resin, then a neutral gas under a pressure of several bars is introduced to improve the impregnation. Afterwards the rotor is drawn out of the tank to be placed into an oven for resin polymerization. This insulation process is called "Thermalastic-Global Epoxy".

**ROTOR**

The magnetic core is composed of high silicon steel laminations, insulated on both faces.

The cage is constituted of rectangular section copper alloy bars.

2°) **The mechanics assembly is composed of**

**Motor thrust and guide bearings**

In normal operative conditions the high pressure coming from the primary circuit generates a thrust upwards which is much higher than the hydraulic thrust of the pump impeller and the rotor weight. That is the reason why the set is equipped with a double thrust bearing.
This thrust bearing is a double acting of Kingsbury type which is designed to support loads of about 60 tons. Each single thrust bearing is composed of 8 babbitted steel shoes. The thrust is transmitted from the shaft to the pads through the thrust runner connected to the shaft.

The thrust bearing can be located at different points of the shaft line above the shaft sealing system. On the primary pumps of the French PWR, the thrust bearing is above the motor. The thrust bearing bracket can also be integral part of the motor frame. This sort of design combines a smaller mass with reduction of the shaft length.

The thrust bearing is oil lubricated. A viscosity pump located in the runner circulates the oil between thrust bearing oil pot and an oil to water heat exchanger installed on the motor side. With this system it is not necessary to install an auxiliary circulating system for oil cooling.

The thrust bearing shoes assembly is completely covered with oil:

- the motor rotation ensures the self feeding of the oil film between shoes and runner.

The system is equipped with two pivoted pad oil lubricated journal bearings.

The bearing losses are evacuated through oil to water heat exchangers.

Tests have shown that the motor pump set could operate up to 30 min with loss of cooling water without damages for bearings.

**Flywheel and anti-reverse rotation device**

The primary electropump is equipped with a flywheel supplying about 2/3 of the set inertia.

This flywheel weight is about 6.5 tons.

In the French PWR this flywheel is located at the top of the group above the thrust bearing. Through this arrangement it is easy to access to the flywheel for performing all ultrasonic tests and dye penetrant tests which can be required during the in-service inspections.

Since there are risks of missile effect which could lead to a fracture in the rotating material having a high inner energy and subsequent destruction risks in the nearest radioactive circuits, a strict regulation for flywheel design and calculation must be respected.
Moreover, the materials are carefully tested at acceptance so as to find all begins of possible crackings; the whole part is then strictly ultrasonic tested in the material mass. The impact toughness characteristics are also carefully controlled.

At its bottom part the flywheel is equipped with an anti-reverse rotation device.

EXTERNAL ELEMENTS

Oil lift system

An external oil lift system ensures the formation of an oil film between thrust bearing shoes and runner when the motor pump set is started.

This system is then tripped 1 or 2 minutes after starting up because the thrust bearing shoes are then self lubricated.

This external system delivers high pressure oil through flexible pipes to a groove machined on the surface of each shoe of both thrust bearings.

Coolers

External tube bundle heat exchangers through which cooling water circulates ensure oil cooling in the upper bearing and air cooling in the motor.

An internal coil in the lower oil pot ensures oil cooling in lower bearing.
Reliability has to be taken into account in three stages:

- Design
- Control of equipment in operating conditions
- Preventive maintenance.
DESIGN:

- MECHANICAL ASPECT
- ENVIRONMENTAL CONSIDERATIONS
- RELIABILITY IN OPERATING CONDITIONS
- MAINTENANCE CONSIDERATIONS
In design stage, we can consider:

Mechanical aspect

The motor pump set is one of the only moving elements of the pressure vessel.

In such sets the 17 tons rotating parts run with a speed of 1500 rpm; it is sure that vibration and shaft displacement is a design concern.

As other requirements determining the design of the different motor pump sets, we could mention:

Environmental considerations

The equipment must withstand, without jeopardizing safety, the consequences of earthquakes or of the reference accident. This has an impact on motor frame.

Reliability of the equipment in operating conditions

The difficult access to the reactor building, in which the motors are, increases the importance of the good operating conditions.

So it is necessary to find a good compromise between quickness and easiness of intervention in case of incident on one hand, reliability and safety of the equipment on the other hand.

Maintenance considerations

Maintenance must be performed on radioactive pieces which have to be decontaminated.

The intervention time should be as short as possible to limit losses due to an outage.

As a whole, design should be the result of a compromise between those various requirements but we think more operation reliability indirectly determines the other aspects of designing.
4 PARAMETERS:

- Quality test in design and manufacturing
- Permanent improvement
- Cooperation between
  - Utilities
  - Maintenance teams
  - Design departments
- Definition of a severe test program
So as to obtain maximum reliability we must aim at perfection in each design phase, that is to say, in designing phase itself as well as in procurement, manufacturing and test phases.

A quality check performed in each phase first of all determines successful results.

The second element of this success is to introduce in the equipments any improvement which may result from field experience.

We do not want to consider technical details but we can name as examples some improvements on motors which increased reliability, while meeting requirements of safety in the reactor containment, easy operating conditions, quick intervention during preventive maintenance operations.

- Continuous research of improvement in shaft line behavior.
  
  * changing the preload of upper and lower guide bearing pads
  
  * change of material in the various oil baffles in order to avoid problems induced in vibration monitoring when they rub against shaft.

- Designing of a specific system to reduce or control the oil leakages from upper and lower oil pot and avoid oil level alarms and fire danger due to them.

- The design of lower and upper bearings on the first motors of earliest type has been modified to improve their seismic resistance.

Generally speaking many little modifications result from a cooperation between utilities, maintenance teams and design departments.

This third aspect, that is to say, collaboration is a determining factor to increase material reliability.

A fourth component is the definition of a severe test program performed on the motors as soon as they leave the fabrication line.

Two series of tests are carried out:

- no-load tests
- on-load tests
The following procedure is applied:

- For each new motor type, tests are performed on a reference machine which is called "lead unit". Those tests consist of no-load and on-load tests including 1500 start-ups.

- For all serial motors the no-load motor tests are only carried out so as to compare their data with test results obtained during "lead unit" tests.

With this procedure we can validate all machines before shipping to site.
OPERATION FOLLOW UP :

. Monitoring from the control room

Examples:

- Temperatures
- Oil levels
- Vibration levels

. Strict operating instructions
The following completes the description of the motors because we deal with safety devices so as to monitor from the control room the most important parameters insuring good operation conditions.

Some of those parameters give alarm when they reach minimum or maximum values. Most of those devices are equipped with a two levelled signal and strict operating instructions are given to the first level to reduce all incidental consequences bringing loss of production or causing irretrievable damages on the equipment.

The different meters installed on the motors control oil level, temperatures, and vibration levels.

* temperature measurement of: - stator windings
  - bearing oil
  - bearing shoes and pads

There are two measurement devices so as either to switch to the second equipment when there is probe failure, or to perform contradictory measurements when an alarm is given.

* level measurement in upper and lower oil pot with alarms for high and low levels

* vibration level measurements on the flange of the lower bracket

* shaft displacement measurement at pump motor coupling.

Moreover, the oil injection system is equipped with a pressure gauge and a pressure switch which impede any startup of the motor as long as the pressure conditions in the double thrust bearing are not reached.

Preventive field maintenance actions on nuclear sites aim at reducing all incidental possibilities in operating conditions and avoiding all difficult interventions in the reactor building.

It is clear the cost for a preventive action is reduced in comparison with a loss due to outage resulting from an accident on an equipment in the pressure vessel.

Moreover those preventive actions have the following positive aspects:

* The inspection or replacement of the more sensitive elements which could badly damage the equipment, in case of failure of one of them, is so possible.
Such incidents could have heavy consequences on the equipment because this motor is a rather essential part and the 17 tons heavy rotor is rotating at 1500 rpm.

* A systematic inspection at regular intervals of some elements has the advantage to detect every generic defect and to plan all corrective actions as we showed it here above to improve its reliability.

As an example, the preventive maintenance programs for motors are defined to be introduced as not affecting critical path into the general maintenance outage planning of the unit.

Moreover so as to have the best intervention conditions, an analysis of all previous possible incidents occurred after the last maintenance action is performed, this analysis is performed with all recordings made by the different probes we quoted here above (temperature, vibration).
## PREVENTIVE MAINTENANCE PROGRAM

<table>
<thead>
<tr>
<th>INTERVALS (Years)</th>
<th>DESCRIPTION</th>
</tr>
</thead>
</table>
| 1                 | - Checking of possible oil and water leakages  
                   | - Replacement of the oil and filters  
                   | - Checking of safety devices |
| 3                 | MOTOR UNCOUPLED FROM ITS PUMP  
                   | - Insulation check of the bearings  
                   | - Anti reverse rotation device inspection  
                   | - Ultrasonic testing of the flywheel  
                   | - Electrical measurements (insulation - polarization index) |
| 6                 | UNCOUPLED MOTOR  
                   | - Check of clearances at guide bearings and at thrust bearings  
                   | - cleaning of air and oil cooler tubes |
| 12                | MOTOR REMOVAL  
                   | - Return to factory for dismantling and full inspection of the motor |

**TABLE II**

DESCRIPTION OF THE SCHEDULED INTERVENTIONS
The list of the main checks which are periodically carried out on the motors during the inspections is displayed in the following table, with the intervals between inspections.

We see that those controls aim at the different aspects which were recalled several times in our survey:

- safety in the reactor building
- operation reliability

Every 12 years the motors are sent back to the manufacturing plant where they are completely dismantled for extensive inspection and component refurbishment.

Those interventions are performed in a specialized shop in nuclear maintenance services. The motors are reconditionned and equipped with all improved technical elements.

CONCLUSIONS

The feedback of informations about 170 operating motors results in the constitution of a data base which allows a good follow up of those motors and their sensitive elements.

The improvements, however slight they may have been, immediately brought to these components during the 12 year inspections or during manufacturing, together with strictly respected operation requirements, and a precise definition of a preventive maintenance program, allow us to consider that the motor is one of the most reliable element in the primary circuit.

The statistical results of last years fire evidence that the unavailability ratio of a power station due to a motor failure is practically equal to zero.
SPECIALIST MEETING ON
PUMP PERFORMANCE AND RELIABILITY
COLOGNE, NOVEMBER 1990

ROUTINE MEASURING OF VIBRATIONS
AS PREDICTIVE MAINTENANCE

DIRK MEIRE
MAINTENANCE ENGINEER
DOEL N.P.P.
BELGIUM
Routine measuring of vibrations as predicitif maintenance.

2. Diagnostic maintenance.
4. Conclusion.

Figures.

1. Data collector.
2. Automatic detection.
4. ISO-standard with levels.
5. Comparison ISO-levels and Doel data.
1. Monitoring at Doel 3 and 4.

Since 1983, pumps, ventilators and their drives have been vibration monitored. Originally the monitoring program was based on the ASME-code: safety-related pumps had to be monitored every three months. This program was extended as part of a program for predictive maintenance. Currently, 1000 pumps, ventilators and their drives (out of 2500 installed at Unit 3 and 4 of the Doel Nuclear Power Plant) are monitored. Vibrations are measured on all bearings in three directions (when possible).

Initially an 'analog' instrument (IRD, Phillips...) was used. This was a time-consuming way of measuring: displacement, velocity, and acceleration of every bearing in the three directions, a minimum of 15 minutes for 1 pump. This method gave the possibility to detect an abnormal situation, but gave little information for a diagnosis.

In 1987 the first digital data-collector was introduced (figure 1). The data-collector is a portable Fourier-analyser with a data-base for measurements: name and place of the machine, name and description of the measurement point, previous value, measurement settings (window...), alert- and fault-levels... It is possible to measure 20 to 40 machines in half a day.

The data-collector stores the measured values (and in some cases its spectrum) in a computer data-base. The measurement is fast and precise (autoranging...), 1 pump takes less than 5 minutes. The inspection frequency was augmented to once a month on most equipment that is functioning continuously. Actually around 10,000 measurements are stored monthly.

The automatic detection of suspect machines is based on fixed ISO-levels, and on automatic trend analysis (figure 2). The analysis and/or diagnosis is still 'man-made'. A helpful tool is that the calculated fault-frequencies of bearings can be shown on the measured spectrum (figure 3).

The result of the analysis can be an augmented inspection program (every month, every week and even every day) or an advice (bearing, alignment...) to the maintenance department.

2. Diagnostic maintenance.

3 typical problems with rotating equipment are due to the normal operation of the equipment:
- balancing
- alignment
- bearings.

Fortunately, these 3 problems have each a very particular vibration pattern. For a trained technician, these patterns are easily recognisable, which makes diagnosis easy.
Other vibration problems can be: low-frequency flow induced vibrations, vibrations from 'zero or mini-flow' operation, resonance, cavitation... Normally, these are start-up problems, and disappear after a few years of operation. When they (re-)appear, it can be difficult to recognise them.

At Doel, the vibration / diagnostic program has two major effects:

- Maintenance is more and more based on a diagnosis: only the minimum of work is done. For correcting a mis-alignment, the ventilator has not to be balanced, and bearings are not replaced any more. Vibrations on the blade-pass frequency is not a maintenance problem, but an operations problem.

- Diagnostic maintenance is gradually replacing the preventive maintenance program. This results also in a higher availability of the equipment. In some cases, the period for scheduled maintenance (replacement of bearings) was doubled.


The ISO-levels for 'Alert' and 'Fault' are based on the principle that large machines can bear more vibrations than small machines (figure 4). These fixed levels are needed for the automatic detection of suspect machines (figure 2). At Doel, the 'Fault' level was kept unchanged, but the 'Alert' level was slightly modified (in function of the frequency, plus or minus 10 percent). This was done in an early stage of the introduction of the data-collector, and was only a minor improvement in the automatic detection.

The use of the ISO-vibration levels for urgent (unplanned) or predictive maintenance can give some problems:

- when used on ventilators, 50% should be stopped
- ISO levels can be too low for certain small pumps
- the level is independent of frequencies
- ...

With the intention of optimizing the program for diagnostic maintenance at Doel, a new approach is being studied, based on the information of previous measurements and the levels of vibrations as they are acceptable for experienced maintenance engineers and formen.
As an example, the ISO-levels and the Doel-measurements on a specific group of pumps are compared.

Group 2 of ISO (medium pumps, > 15 KW and < 300 KW), has two vibration levels: 2.8 mm/sec (Alert) and 7.15 mm/sec (Fault). These levels are valid for measurements from 10 to 1000 Hz.

Out of all measurements at Doel 3 available in the computer data-base, information was selected following these criteria:
- Horizontal pumps with antifriction bearings, without gearboxes, electric drives, no belts...
- > 15 KW and < 510 KW.
- Age of equipment: between 6 and 10 years in operation (only Doel 3).
- Running time: from a few hundred hours (stand-by equipment) to 80,000 hours (production process).

The result is a database with 85 pumps and 11,530 measurements. One measurement consists of 7 values (the 'parameter' set). These are defined as:

<table>
<thead>
<tr>
<th>Name</th>
<th>From (order)</th>
<th>To (order)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall value</td>
<td>4 Hz</td>
<td>30</td>
</tr>
<tr>
<td>Subharmonic</td>
<td>0.10</td>
<td>0.80</td>
</tr>
<tr>
<td>1 X RPM</td>
<td>0.80</td>
<td>1.50</td>
</tr>
<tr>
<td>2 X RPM</td>
<td>1.50</td>
<td>2.50</td>
</tr>
<tr>
<td>3 X RPM</td>
<td>2.50</td>
<td>3.50</td>
</tr>
<tr>
<td>Higher orders</td>
<td>3.50</td>
<td>30</td>
</tr>
<tr>
<td>High Frequency Detection</td>
<td>&gt; 5 Khz</td>
<td></td>
</tr>
</tbody>
</table>

With these measured values it was possible to calculate mean values, comparison with ISO-levels, distributions...
Part of the results is shown in the next tables (and figure 5):

Mean value of vibration levels (in mm/sec):

<table>
<thead>
<tr>
<th>Group</th>
<th>ISO Fault</th>
<th>ISO/Doel Alert</th>
<th>Doel Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall value</td>
<td>7.15</td>
<td>3.10</td>
<td>1.52</td>
</tr>
<tr>
<td>Subharmonic</td>
<td>7.15</td>
<td>2.80</td>
<td>0.54</td>
</tr>
<tr>
<td>1 X RPM</td>
<td>7.15</td>
<td>3.10</td>
<td>0.47</td>
</tr>
<tr>
<td>2 X RPM</td>
<td>7.15</td>
<td>2.80</td>
<td>0.44</td>
</tr>
<tr>
<td>3 X RPM</td>
<td>7.15</td>
<td>2.80</td>
<td>0.27</td>
</tr>
<tr>
<td>Higher orders</td>
<td>7.15</td>
<td>2.25</td>
<td>0.94</td>
</tr>
</tbody>
</table>
Distribution in percent of the measurements:
Reference: ISO-Fault level: 7.15 mm/s.

<table>
<thead>
<tr>
<th>From</th>
<th>To</th>
<th>Level</th>
<th>Overall</th>
<th>1X RPM</th>
<th>Higher</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.45</td>
<td>very good</td>
<td>11.1</td>
<td>71.2</td>
<td>30.1</td>
</tr>
<tr>
<td>0.45</td>
<td>1.12</td>
<td>good</td>
<td>47.0</td>
<td>23.9</td>
<td>48.2</td>
</tr>
<tr>
<td>1.12</td>
<td>2.80</td>
<td>allowable</td>
<td>36.8</td>
<td>4.2</td>
<td>19.9</td>
</tr>
<tr>
<td>2.80</td>
<td>7.15</td>
<td>just tol.</td>
<td>4.9</td>
<td>0.6</td>
<td>1.7</td>
</tr>
<tr>
<td>7.15</td>
<td>....</td>
<td>not perm.</td>
<td>0.2</td>
<td>0.0</td>
<td>0.1</td>
</tr>
</tbody>
</table>

Some conclusions are:

- The Doel vibration levels are highest in the 'higher orders' region (blade pass frequency, bearings...). Other vibration levels are in general 10 times lower than the ISO Fault level.
- Based on the ISO-levels, only 0.2% of the 'overall level' measurements require 'immediate action' or 'unplanned maintenance'. This can lead to (Technical Specifications) unavailabilities.
Based on these figures this should happen only once every 6 years at Doel 3.
- Based on the ISO-levels, about 4% of the pumps should be looked after (extra monitoring or maintenance). This means at Doel 3 only 3 pumps each year, or a service interval of 25 years!

In reality, more maintenance is done, not only caused by vibration indications, but also replacement of seals, changing of oil, complete overhaul...

With regard to vibrations, the maintenance engineer will base his decision mostly on the trend analysis. Since ISO-levels are much higher than normally acceptable levels of vibration, maintenance will be performed at a lower level of vibration.

It is seldom that the ISO-levels are used as a reference, except with Technical Specification problems.

At Doel, the ISO 'Alert' levels will be replaced with the newly derived values, using the approach described in this chapter. Using these lower detection values, combined with the trend analysis, an inspection period of three months should be possible, instead of one month.
4. Conclusion.

Vibration monitoring is a powerful tool for improving the reliability of pumps: less frequent and more efficient maintenance can be based on the vibration diagnosis.

Where ISO has 4 groups of machines for vibration monitoring, it could be useful to determine the 'acceptable levels of vibrations' for a larger number of specific machines (large vertical pumps, electrical drives, ventilators with belts...).

The rapidly extending amount of information existing in electronic databases can be used to calculate these levels. Depending on the detail of the information available, it can be possible to determine a spectrum which is typical for each group of machines.
Figure 1: Data collector.
**Figure 2: Automatic detection.**

**"A" alarm: Time to alarm**

<table>
<thead>
<tr>
<th>SIGNAL</th>
<th>PARAMETER</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>&quot;D&quot;/FAULT</td>
<td>&quot;C&quot;/ALERT</td>
<td>&quot;B&quot;/WARNING</td>
</tr>
</tbody>
</table>

![Graph showing time to alarm with signal levels and time intervals]

**"B", alarm: Baseline ratio warning**

<table>
<thead>
<tr>
<th>SIGNAL</th>
<th>PARAMETER</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>&quot;D&quot;/FAULT</td>
<td>&quot;C&quot;/ALERT</td>
<td>BASELINE VALUE x INPUT RATIO E.G. 1.4 x BASELINE</td>
</tr>
</tbody>
</table>

![Graph showing baseline ratio with signal levels and time intervals]

**"B", alarm: Statistical deviation warning**

<table>
<thead>
<tr>
<th>SIGNAL</th>
<th>PARAMETER</th>
<th>VALUE</th>
</tr>
</thead>
<tbody>
<tr>
<td>&quot;D&quot;/FAULT</td>
<td>&quot;C&quot;/ALERT</td>
<td>MEAN VALUE</td>
</tr>
</tbody>
</table>

![Graph showing statistical deviation with signal levels and time intervals]
Figure 3: Spectrum with bearing fault.
**Figure 4**: ISO-standard with levels.

<table>
<thead>
<tr>
<th>RMS Velocity (mm/s)</th>
<th>Class I</th>
<th>Class II</th>
<th>Class III</th>
<th>Class IV</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.18</td>
<td>Good</td>
<td></td>
<td>Large machines operating at speeds above foundation natural frequency. (e.g. Turbo-machines)</td>
<td></td>
</tr>
<tr>
<td>0.28</td>
<td>Good</td>
<td>Medium machines 15-75 kW or up to 300 kW on special foundations.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.45</td>
<td>Good</td>
<td>Large machines with rigid and heavy foundations whose natural frequency exceeds machine speed.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.71</td>
<td>Allowable</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.12</td>
<td>Allowable</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.8</td>
<td>Allowable</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.8</td>
<td>Just tolerable</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.5</td>
<td>Just tolerable</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8 db</td>
<td>Just tolerable</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>11.2</td>
<td>Not permissible</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>Not permissible</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>28</td>
<td>Not permissible</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>45</td>
<td>Not permissible</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**VIBRATION CRITERION CHART (10 — 1000 Hz)**

VDI 2056  ISO 2372  BS 4675
Figure 5: Comparaison ISO-levels and Doel data.

Doel 3 Medium pumps.

% exceeding ISO level
--- ISO fault level
--- ISO alert level
--- Doel alert level
% exceeding DOEL level
--- Doel data (mean)

(Oct '90)
Session # 4  
Chairman's Closing Remarks  
(Mr. J.-P. Clausner, IPSN, France)  

During this session five papers were presented. An additional paper dealing with "Operational problems of the ECCS high pressure injection pumps of Paks Nuclear Power Station" [Hungary] should have been presented; unfortunately the author was not able to attend the meeting.  

The first presentation was made by Dr. A.N. Kumar, AECL, Canada. His paper entitled: "Candu heat transport pumps. A proven quality product", emphasized the high degree of quality reached by the Candu Heat Transport Pump which results in outstanding records of capacity factor achieved by the Candu reactors. This high reliability is attributed to an integrated design approach covering all aspects - research and development, material selection, testing, operation and maintenance. This result is also a significant contributing factor to the safety of the CANDU plants.  

The second paper entitled: "Design and Manufacturing of primary sodium pumps for the prototype fast breeder", is a joint paper written by Mr. Y. Yamagishi, PNC and Messrs. S. Yazawa, S. Nakadaira, Y. Hayashi and J. Kikushima, Hitachi, Ltd. Japan. In his presentation, Mr. Nakadaira made a comprehensive description of the circulating pump development project carried out over 20 years and which should be awarded with the initial criticality of the MONJU prototype reactor presently scheduled for October, 1992.  

The third paper has been presented by Mr. Ph. Dombret, AIB-Vinçotte, Belgium, and dealt with: "Ultrasonic in-service inspection of PWR coolant pump bowl welds".  

Mr. Dombret described the ultrasonic technique, carried out from the outer surface only, which was developed in Belgium and qualified at Tihange unit 1, during a refuelling shutdown. This methodology, using focusing probes, reduces significantly the inspection duration, as well as the radiation doses incurred by personnel.  

The fourth presentation made by Mr. J.L. Killian, Jeumont-Schneider Industrie, France, was a comprehensive overview of the "Performances of the Motors of Primary Pumps for Nuclear Reactors".  

Mr. Killian has showed that the reliability relies on a continuous process at each step of the equipment lifetime from the design up to the monitoring during operation and maintenance. In particular, the importance of a good preventive maintenance program was pointed out in order to achieve a high degree of reliability.  

Finally, the fifth presentation was made by Mr. D. Meire, Doel 3 & 4 NPP, Belgium and dealt with "Routine Measuring of Vibrations as Preventive Maintenance". Mr. Meire described a new approach being studied, based on the database from 85 pumps and more than 11,000 measurements, in order to improve the diagnosis and optimize the preventive maintenance programme. This paper pointed out that good vibration monitoring is a powerful tool for improving the reliability.
SESSION #5: "PUMP PERFORMANCE IN SPECIFIC APPLICATIONS"

Chairman: M. Hada

* "Advantages of wet motor pumps in nuclear power plants based on ABB Atom AB experience", L. Törnbloom & S. Nylen, ABB Atom AB, Sweden (presented by L. Törnbloom)

* "Charging pumps in French PWR nuclear power plants", A. Duhamel, FRAMATOME, France J. Dhote & J. Vauchel KSB - Pompes Guinard, France (presented by A. Duhamel & J. Vauchel)

* "Design modifications carried out on RCP at C. Trillo I", J. M. Burriel & G. Blasco, Trillo Nuclear Power Plant, Spain (presented by J. M. Burriel)

* "Reactor coolant pump seal failure considerations", J. E. Jackson, R. L. Baer, W. Minners & C. J. Heltemes, NRC, United States
Advantages of wet motor pumps in Nuclear Power Plants based on ABB Atom AB Experience

Presented at the CSN1 Specialist Meeting on "Pump performance and Reliability", GRS, Colgne, Germany, November 1990.

Written by Lars Törnblom and Stig Nylén, ABB Atom AB, S-72163 Västerås / Sweden. Telephone +46 (0) 21 107000.

Abstract

In BWRs, designed by ABB Atom, use is made of pumps with wet motors for Main Recirculation System and Reactor Water Clean-up System.

Development work on this type of pumps has been made in cooperation between ABB Atom and pump manufacturers. The general favourable operation of wet motor pumps covers two decades of ABB Atom experience. A description of this experience and associated development is the subject of the present paper.

Gu
INTRODUCTION OF ABB ATOM

ABB Atom is an independent supplier of nuclear reactor systems, and offers turnkey plants, nuclear islands and nuclear steam supply systems based on Boiling Water Reactors. The company has supplied eleven nuclear power plants, whereof nine are located in Sweden and two in Finland. ABB Atom is the only LWR supplier outside the USA and the USSR which has developed, designed and engineered its reactor system without having resorted to foreign licenses at any time.

Further ABB Atom is an established manufacturer of control rods and of complete fuel assemblies for both BWRs and PWRs.

WET MOTOR PUMPS

Pumps for the recirculation system and for the clean up system have for all nuclear plants of ABB Atom design been chosen to be of the wet motor type. The high radioactivity of the process water and the close connection to the reactor have been the reasons. The absence of a moving sealing surface was regarded to give possibilities to better reliability and less demand for service. Another important factor was that service work to a greater extent could be made in areas with less radiation.

Table 1 gives the main the data for the eleven plants ABB Atom has delivered to date. The table shows the number of wet motor pumps in each power station.

PUMP SUPPLIERS

It is an ABB Atom policy not to rely upon one supplier to important components in the plant.

Suppliers employed by ABB Atom for recirculation pumps are Hayward Tyler and KSB, and suppliers for pumps to the clean up systems are KSB and Pompes Guinard.

MAIN RECIRCULATION SYSTEM

EXTERNAL PUMPS

Design

General

The Main Recirculation System for the first generations of ABB Atom reactors consists of an external and one internal part - related to the reactor pressure vessel. The external part consisting of 4 or 6 loops in parallel. One of these loops is shown in figure 1. The pumpcase is welded into the pipework.

Each pump motor unit consists of a single stage centrifugal pump driven by an induction motor of the wet stator type. The motor is cooled by passing the water through an external heat exchanger. Glandless construction is employed and as such the interior of the motor is pressurised by the system media.
The motor stator is wound with special cable able to operate submerged and in a pressurised environment. The electrical supply passes into the pressure containment of the motor, via pressure sealed terminal glands.

The unit design also includes means (thermal neck or heat barrier) of restricting the flow of heat from pump to motor.

Unit design
The construction of the unit is shown on the sectional arrangement drawing in figure 2 and 3.

The unit is designed for vertical mounting with the pump case positioned above the motor. The pump and motor share a common shaft, which locates a rotor on the lower portion and an impeller at the upper end.

The impeller of the shrouded mixed flow type is fitted with a renewable wearing ring. The ring runs within the case wear ring.

The continuous purge water applied just above the motor keeps undesirable fluids and smaller solids from penetrating the motor.

The motor is of the squirrel cage induction type, wound with special water-tight covered cable in the stator. The conductors are made of copper.

Bearings
The hydraulic thrust of the pump is taken by the thrust bearing situated immediately below the cover end journal bearing. This thrust is of the tilting pad type and is water lubricated.

The weight of the rotating member and thrust imposed at start-up and shut-down periods are taken by the reverse thrust bearing, also of the tilting pad design.

The rotating assembly runs on two water lubricated journal bearings. Due to the design of the journal bearings, reverse rotation of the rotor is permissible.

Motor cooling circuit
The heat exchanger dissipates heat generated in the motor windings and also any heat passed from the pump body, but should this flow of cooling water cease; thermo syphon circulation takes place to remove conducted heat from the pump end of the motor.

The thrust disc at the base of the motor incorporates an auxiliary impeller which circulates the internal water of the motor through the bearings, and motor windings. The water passes out at the top end of the motor, through an external pressure tight heat exchanger and back to the bottom of the motor.

Terminal glands
The electrical supply to the motor is fed through special high pressure single lead cable inlets of the self sealing type.
Heat exchanger

The heat exchanger dissipates the heat generated in the motor windings. The unit is of the shell and tube type, having high pressure system water on the shell (primary) side and low pressure cooling water on the tube (secondary) side.

Brackets are provided on the side of the motor case to mount the heat exchanger, whilst interconnecting pipework links the unit to the motor at the top and bottom of the case.

Alarm systems

Probes are provided to relay temperature, vibration, power, and flow rate to the control room.

Development

At the time when the external pumps were ordered some of the specified demands were unusual to this type of pumps as they were dependent on the application. Four of them were

1. Variable speed. The pumps were used to regulate the power of the reactor by varying the flow.
2. Purge water. In order to give a low radiation level of the motors at service.
3. Good behaviour at low NPSH. The pumps were used to heat the reactor during pre-nuclear hot tests.
4. Improved design of pump house to facilitate non-destructive testing.

Experience

During the years some modifications have been imposed. (The accumulated duty time for all the external pumps is about $2.5 \times 10^6$ hours). The most important modification that was imposed to all pumps was due to thermal fatigue of pumpshafts. There has also been modifications of the pump house close to the shafts due to fatigue.

The mixing of the cool purge water with the hot process water is not stable due to movement of the shaft. In this case temperature difference and movement of the mixing zone was too big. The cure that reduced but not completely eliminated the fatigue was thermal sleeves and reduced purge flow.

There have been some minor problems:

- Leakage from flanges - specially at heating up or cooling down.
- Some motors have been rewound due to aging of the cables.
- In some instances broken screws, holding details in the trust bearing, have given problems when the rotor has moved axially when rotating.
Some disturbances have occurred in connection with handling and service. E.g. mechanical damage of winding for example when cleaning the stator.

An important factor to take into account is that after uprating the power of the reactors the pumps have been used at higher speeds. This of course implies higher duties on e.g. bearings and cables. Some of the above mentioned disturbances has been caused by these higher duties.

INTERNAL PUMPS

As an important step in the ABB Atom development of the reactor design the external loops and pumps of the main recirculation system were replaced by internal pumps.

Design

Unit design

The circulator is a single stage centrifugal pump driven by a wet stator induction motor arranged as shown in figure 4 with the pump impeller and diffuser inside the base of the reactor vessel and the motor housed in a suspended pod forming part of the pressure boundary.

Figures 5 and 6 show a longitudinal cross section of a pump and motor. The pump impeller is bolted to the upper end of the vertical pump shaft. Reactor coolant enters the impeller axially from above and is discharged downwards into the diffuser. The diffuser is held in position on top of the penetration nozzle, by a stretch tube which is retained by a nut locked inside the motor case. Coolant recirculation is restricted by piston rings contacting the sealing bush welded into the reactor vessel down comer, and by renewable wear rings on both the impeller and diffuser.

The impeller, with pump shaft attached, and the diffuser are removed through the top of the reactor vessel, by means of mechanical handling equipment. Similarly, the complete motor unit can be removed for maintenance with the aid of primary and secondary static seals. A collar on the pump shaft inside the reactor vessel forms a static seal when lowered to contact the upper surface of the stretch tube flange. This is backed up with an inflatable muff-type secondary seal immediately below the stretch tube nut.

The motor, complete with hollow rotor tube and bearings, is lifted to engage the suspended pump shaft. The torque is transmitted at the upper end of the rotor tube. A long coupling screw at the cover end completes the coaxial shaft system.
Cooling and lubrication is provided by low temperature coolant water circulating through the windings and bearings at full reactor pressure. This water is in connection with the reactor coolant water through the angular gap between pump shaft and strech tube. The coolant water is passed through an external shell and tube heat exchanger before being recirculated through the motor. A continuous cold purge is injected to the top of the motor case to keep radioactive debris out of the motor. Water-lubricated radial and thrust bearings support the rotor tube. The thrust disc also acts as an auxiliary impeller to circulate the motor coolant.

A speed sensor head, mounted in the thrust bearing housing, generate signals from passing slots in the perifery of the rotating thrust disc. A simple back-stopping device is also incorporated in the thrust disc and housing to prevent reverse rotation of the unt.

The pressure tight submersible winding cables pass through the main motor cover into the terminal box. An auxiliary cover provides access to the coupling screw to disengage the pump shaft. As there are no valves in the circulation system, control of the pump output is achieved by using a thyristor converter to vary the supply frequency to the motor.

Development

The basic design of the reactor internal pumps differs only slightly from the conventional design of wet motor pumps. The influence of the reactor pressure vessel has, however, entailed two changes:

1. The shaft is divided, giving the possibility to handle the impeller and pump shaft through the reactor vessel

2. The diffuser is of the vane type due to the limited space available.

To give a base for and to check the design, a large number of different tests have been made:

1. Component and model testing
2. Prototype tests
3. Delivery testing of production pumps
4. Tests in reactor during commissioning

The development work has of course also involved design and calculation. The calculation has covered various areas, such as analysis of hydraulics, stresses, temperature distribution, stability, dynamics and so on. These have covered components (e.g. pump shaft, bearings and reactor vessel nozzle), the circulation system and the whole plant including turbine, generator and switch yard. The work had been going on for over twelve years before the pumps were operating in a power plant.

Component and model testing

Model testing, component and procedure testing have been performed by ABB Atom, the two pump suppliers and to some extent by other companies or institutions.
The tests have for example covered the influence of different reactor pressure vessel configurations on pump performance, the hydraulic behaviour of the pump, working as a turbine with the rotor running free or with locked rotor, mechanical performance of a back rotation stop device.

Prototype tests

The two pump manufacturers employed by ABB Atom have each delivered two full-scale prototypes, which have been tested at full pressure and temperature in a special test rig at ABB Atom's laboratory. The first set of prototypes was tested for 8000 and 1400 hours respectively, and the second set 3200 and 2000 hours.

The test programme conducted at the test rig included:

a) Assembly and disassembly under simulated reactor conditions with a water column of 20 m. The tests were performed in a series which resulted in a gradual improvement of the handling equipment.

b) The prototypes were tested for their performance. Performance was based on the static pressure difference between the outlet and the inlet of the pump.

c) Extensive measurements of vibration have been undertaken.

d) Cavitation tests were performed at different temperatures, speeds and flows.

e) The temperature distribution in the pump nozzle at steady state condition and for transients has been measured.

f) Changes in alignment of the motor house because of vessel pressurization and temperature.

g) Tests on diffusion of particles have been performed to establish the minimum purge water flow.

h) Performance of the primary cooling circuit during both normal and abnormal conditions.

A number of other tests and checks have also been made, e.g. endurance, tightness, and electrical disturbance on in-core measurement, and on the main electrical net.
Experience

The reactor internal pumps have to date had a very good performance. The accumulated duty time is approximately $3 \cdot 10^6$ hours. As a total only a couple of days of production have been lost for all 6 plants with internal pumps.

For pumps from both suppliers there were during the first year problems with the backing stop device. After changing some smaller details they have functioned satisfactorily.

Some pumps have been rewound and/or got their terminals repaired.

At two occasions handling has caused seizing between mating metallic surfaces. That was cured by redesigning handling tools and changing a procedure.

Service intervals were from the beginning 4 years. Experience shows that they can be increased. There have been pumps running for 6-7 years between services.
REACTOR WATER CLEAN-UP SYSTEM

SYSTEM FUNCTION

Two systems in the ABB Atom nuclear power plants, Reactor Water Clean-up System and Shut Down Cooling System (Residual Heat Removal System) are served by common pumps in the high-pressure part. The main system functions are as follows:

- Cleaning the reactor water for maintaining an appropriate water chemistry.
- Cooling the reactor to cold shut-down conditions.

A simplified flow diagram is shown in figure 7.

The pumps are situated outside the reactor containment. Two pumps are installed in parallel, one is in operation for normal clean-up water flow and both pumps for maximum (forced) flow.

Design

General

The design of the RWCU pumps are in all essential parts similar to the external pumps in the main recirculation system. Table 4 gives main data for RWCU pumps.

Some common features for the RWCU pumps in all ABB Atom plants can be stated:

- Centrifugal type.
- Glandless.
- Wet motor.
- One or two pump stages depending on total head.
- Vertical shaft.
- Internal motor cooling circuit. A heat exchanger dissipates heat from the motor part to a secondary cooling system.
- Heat barrier for preventing heat to be transferred from the process water to the motor part. The heat barrier is provided with a jacket for cooling by the secondary cooling system.
- Purge water with controlled chemical properties and pressure exceeding the process water pressure is supplied in the hot neck part between the pump and the motor. The purge water makes a fluid barrier that prevents reactor water to enter the motor and contaminate the motor internals.
- Pump casing is provided with nozzles for welded connections to the process pipes. All maintenance can therefore be performed with the casing still in the system.

Cross sections of the two types of RWCU pumps are shown in figures 8 and 9.

A flow diagram for cooling water and purge water supply is given in figure 10.

Development

The original development work was mainly performed by the pump manufacturers during the design period.

Development, in which ABB Atom has been involved, is made in a later stage and implies:

- Modifications of the pump design as a result of operation experience.
- Development of equipment for handling and maintenance.

The modifications and equipment are described below.

Equipment for handling and maintenance

Equipment for handling and maintenance has been designed and manufactured by ABB Atom and consists of remotely controlled devices for the following purposes:

- Transport and lifting of the pump.
- Radiation shielding.
- Disassembly or replacement of internal parts in the pump such as impellers and wear rings.

Experience

Operation experience of wet motor pumps in the RWCU system as of today are very good in all ABB Atom plants.

However, operation experience in the older plants has lead to some modifications of principally three different kinds:

A. Electrical
B. Material
C. Mechanical

A. Electrical

The insulation of the stator windings has in some cases been punctured at handling or cleaning by flushing. At least in some cases the insulation have had defects from the beginning and not been caused by damage. Such faults are now eliminated by raised requirements on wire manufacturing and winding. Testing of the insulation by resistance measuring is performed in water at operating pressure of both the wire before winding and the complete
Leakage and insulation problems in the cable penetration (through the motor housing) have occurred in one pump type. The problems are solved by a revised design of a type of double packing box.

B. Material

Galvanic corrosion has occurred between stator and rotor plates of carbon steel and details of copper or brass. In some pumps new stators and rotors are installed with plates of stainless steel.

C. Mechanical

Wear of bearings was in the beginning a problem in some pumps for two main reasons:

- Corrosion particles in the motor cooling water created excessive wear in one pump type. The corrosion problems were solved by exchanging the carbon steel plates in the stator and rotor to stainless steel.

- The axial forces had in one of the pump types, under certain operational conditions, an upward direction for which the pump were not designed. The pumps were redesigned and equipped with double-acting axial bearings and the wear problem was eliminated.
SUMMARY

The wet motor pumps have in ABB Atom experience had a good performance. Specially the ones of later design have had a considerably better performance partly as a result of the costly and time consuming development work. Also the pumps with shaft seals have of course improved their reliability during the years, but it is, however, our opinion that it is easier to get a good performance and a low man dose at service when using wet motor pumps.

However, considering the accumulated experience of wet motor pumps in nuclear applications, it is the intention of ABB Atom to also use this type of pump motors in future plants.
# Table 1 - Nuclear Power Plants delivered by ABB Atom AB

<table>
<thead>
<tr>
<th></th>
<th>Oskarshamn 1</th>
<th>Oskarshamn 2</th>
<th>Oskarshamn 3</th>
<th>Ringhals 1</th>
<th>Barseback 1</th>
<th>Barseback 2</th>
<th>Forsmark</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal power, MWth A)</td>
<td>1365</td>
<td>1700</td>
<td>3000</td>
<td>2270</td>
<td>1700</td>
<td>1700</td>
<td>2700</td>
</tr>
<tr>
<td>Net electric power, MWe A)</td>
<td>440</td>
<td>570</td>
<td>1060</td>
<td>750</td>
<td>570</td>
<td>570</td>
<td>900</td>
</tr>
<tr>
<td>Thermal power (uprated)</td>
<td>-</td>
<td>1800</td>
<td>3300</td>
<td>2500</td>
<td>1800</td>
<td>1800</td>
<td>2220</td>
</tr>
<tr>
<td>Operating pressure, bars</td>
<td>70</td>
<td>70</td>
<td>70</td>
<td>70</td>
<td>70</td>
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<tr>
<td>Number of fuel assemblies</td>
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<td>444</td>
<td>700</td>
<td>648</td>
<td>444</td>
<td>444</td>
<td>676</td>
</tr>
<tr>
<td>Number of absorbers</td>
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<td>102</td>
<td>169</td>
<td>157</td>
<td>109</td>
<td>109</td>
<td>161</td>
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<td>Reactor vessel ID, mm</td>
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<td>5200</td>
<td>6400</td>
<td>5200</td>
<td>5200</td>
<td>5200</td>
<td>6400</td>
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</table>

**Wet motor pumps**

- in recirculation systems  
  - 4 ext.  
  - 4 ext.  
  - 8 int.  
  - 6 ext.  
  - 4 ext.  
  - 4 ext.  
  - 8 int. 

- in clean up systems  
  - 2  
  - 2  
  - 2  
  - 2  
  - 2  
  - 2  
  - 2

A) Initial
### Table 2 - Main data for external recirculation pumps

<table>
<thead>
<tr>
<th></th>
<th>Oskarshamn 1</th>
<th>Oskarshamn 2</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Ringhals 1</td>
<td>Barsebäck 1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Barsebäck 2</td>
</tr>
<tr>
<td>Head</td>
<td>45 m</td>
<td>44 m</td>
</tr>
<tr>
<td>Quantity</td>
<td>2.2 m³/s</td>
<td>2.3 m³/s</td>
</tr>
<tr>
<td>Design pressure</td>
<td>95 bars</td>
<td>95 bars</td>
</tr>
<tr>
<td>Design temperature</td>
<td>300°C</td>
<td>300°C</td>
</tr>
<tr>
<td>Voltage</td>
<td>3 kV</td>
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</tr>
<tr>
<td>Purge flow</td>
<td>69 g/s</td>
<td>97 g/s</td>
</tr>
</tbody>
</table>

### Table 3 - Main data for internal recirculation pumps

<table>
<thead>
<tr>
<th></th>
<th>Forsmark 1</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>TVO 1</td>
</tr>
<tr>
<td></td>
<td>Forsmark 2</td>
</tr>
<tr>
<td></td>
<td>TVO 2</td>
</tr>
<tr>
<td></td>
<td>Forsmark 3</td>
</tr>
<tr>
<td></td>
<td>Oskarshamn 3</td>
</tr>
<tr>
<td>Head</td>
<td>33.5 m</td>
</tr>
<tr>
<td>Quantity</td>
<td>1.72 m³/s</td>
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<tr>
<td>Design pressure</td>
<td>85</td>
</tr>
<tr>
<td>Design temperature</td>
<td>300°C</td>
</tr>
<tr>
<td>Voltage</td>
<td>890 V</td>
</tr>
<tr>
<td>Purge flow</td>
<td>7 g/s</td>
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# Table 4. Main data for RWCU-pumps

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<th>F1/F2</th>
<th>B1/B2/O2</th>
<th>R1</th>
<th>O1</th>
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<td>178</td>
<td>160</td>
<td>165</td>
<td>125</td>
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<td><strong>Quantity (m³/h)</strong></td>
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<td>400</td>
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<td>10</td>
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<td>500</td>
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Figure 1
External recirculation loop
Baureihe: PSR  
SWR interne Pumpe  
mit Naßläufermotor

Type: PSR  
BWR internal pump  
with wet motor

Hauptsächlich-  
Umwälzpumpe

Primary coolant  
circulation pump

Figure 6
Figure 7. Simplified flow diagram

312 Feedwater lines
321 Shut down cooling system
331 Reactor water clean up system
Figure 8. Cross section of RWCU pump
Figure 9. Cross section of RWCU
Figure 10.

Flow diagram for cooling water and purge water to RWCU pump
CNCI SPECIALIST MEETING ON PUMP PERFORMANCE AND RELIABILITY

Cologne, 26th - 28th November, 1990

CHARGING PUMPS IN FRENCH PWR NUCLEAR

A. DUHAMEL
FRAMATOME - TOUR FIAT
92084 PARIS-LA-DEFENSE

J. DHOTE - J. VAUCHEL
KSB - POMPES GUINARD
39, avenue du Pont de tasset
B.P. 435 - 74020 ANNECY CEDEX
SUMMARY

1. GENERALITY

2. IMPROVEMENTS ISSUED FROM PLANT OPERATION EXPERIENCE
   2.1. THRUST BEARING
   2.2. OIL CIRCUIT DEVELOPMENT
   2.3. COUPLINGS
   2.4. SPEED INCREASER

3. IMPROVEMENTS FOR SI FUNCTION
   3.1. STARTING UP WITHOUT AUXILIARY LUBRICATION PUMP STARTUP
   3.2. HYDROSTATIC BEARINGS ARRANGEMENT

4. IMPROVEMENTS FOR WORKERS PROTECTION REDUCTION OF NOISE LEVEL

5. CONCLUSION FUTURE TRENDS
1. CHARGING PUMP

2 FAMILIES

1) 3 LOOP PLANTS

FUNCTIONS:

- PROVIDE CHARGING FLOW TO RCS
- SUPPLY COOLING WATER TO SHAFT SEALS OF RCP
- HHSI

OPERATING CONDITIONS

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2) 4 LOOP PLANTS

FUNCTIONS

IDEM 3 LOOP PLANTS WITHOUT SI FUNCTION

OPERATING CONDITIONS

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2.1. THRUST BEARINGS

900 MW REACTOR PROGRAM
INITIALLY DESIGNED WITH:

- ONE PIECE CASING IN CAST IRON
- ONE PIECE BEARING, LEMON TYPE, ASSEMBLED AND TIGHTENED IN THE CASING
- PAD THRUST BEARING WITH DOUBLE EFFECT
- LABYRINTH SEALING
- LUBRICATION AT START-UP WITH OIL CHAMBER AT THRUST LEVEL

1300 MW REACTOR PROGRAM
EVOLUTION COMPARED WITH 900 MW REACTOR PROGRAM:

- POSITION OF BEARING MODIFIED - BETTER DISTRIBUTION OF LOADS
- USE OF THRUST BEARING WITH PADS - BETTER STABILITY
- SEALING WITH DEFLECTOR, COUPLING SIDE

CHINA REACTOR PROGRAM
EVOLUTION COMPARED WITH THE 1300 MW REACTOR PROGRAM:

- CAST IRON CASING IN TWO PIECES
- DOUBLE THRUST IN TWO PIECES
- THRUST BEARING WITH PADS IN TWO PIECES
- PADS REPLACED WITHOUT UNCOUPLING THE PUMP
- INCREASE OF THE OIL VOLUME
- OIL CHAMBER SEALING WITH FLOATING GASKETS
2.2. OIL CIRCUIT DEVELOPMENT

2.2.1. FESSENHEIM

INITIALLY DESIGNED WITH:

- STEEL PIPINGS
- VALVES, VANES, CONNECTIONS: STEEL - BRONZE CAST IRON
- SCREWED CONNECTIONS
- WATER/OIL EXCHANGER WITH STEEL TUBES
- PRESSOSTATS WITH METALLIC FANS

INCIDENTS:

- LEAKEAGES FOUND AT SCREWED CONNECTIONS
- CLOGGING UP AND CORROSION OF TUBES AT ROUGH WATER EXCHANGER
- DISTORTION OF PRESSOSTAT FANS - ALTERNATE STRESSES DUE TO PRESSURE PULSATIONS OF THE OIL CIRCUIT

REMEDIES:

- SCREWED CONNECTIONS REPLACED BY FLAT FLANGEs AND GASKETS WITH GRAPHITE BASES
- EXCHANGER STEEL TUBES REPLACED BY STAINLESS STEEL TUBES
- METALLIC PRESSOSTAT FAN REPLACED BY PERBUNAN
- FILTRATION OF THE ROUGH WATER IN THE EXCHANGER CIRCUIT
2.2.2. 900 MW REACTOR PROGRAM

INITIALLY DESIGNED WITH:

- PIPING CONNECTIONS WITH FLANGES AND FLAT GASKETS (PETROLEUM APPLICATION TYPE)
- OIL/AIR EXCHANGER ON OIL CIRCUIT
- PRESSOSTATS WITH PERBUNAN FANS

INCIDENTS:

- LEAKAGES FOUND AT FLANGES CONNECTIONS
- GEOMETRICAL DEFECT (PERPENDICULAR AND PARALLEL)

REMEDIES:

- MACHINING OF FLANGES
- GASKETS WITH GRAPHITE BASES
2.2.3. 1300 MW P4 AND P'4 REACTORS

INITIALLY DESIGNED LIKE THE 900 MW REACTOR PROGRAM NO SPECIFIC PROBLEM

2.2.4. N4 REACTOR

INITIALLY DESIGNED LIKE THE 1300 MW REACTOR PROGRAM WITH ADDITION OF:

- UNCOUPLING OF THE OIL CIRCUIT FROM THE GROUP (EXTERNAL STATION)

- CONNECTION TO THE INCREASER AND PUMP WITH HIGH PRESSURE FLEXIBLES
2.3. COUPLINGS

900 MW REACTOR PROGRAM

DESIGN
- GEAR COUPLING
- LUBRICATION WITH GREASE

INCIDENTS
- SEPARATION OF OIL GREASE AND SOAP
- OIL LEAKAGE
- BAD LUBRICATION
- UNBALANCE DUE TO SOAP WHICH GIVES VIBRATION
- DAMAGE OF GEAR

REMEDIES
- REINFORCEMENT OF SEALINGS
- REPLACEMENT OF THE GREASE

1300 MW P4 ET P'4 REACTOR PROGRAMS

DESIGN
- GEAR COUPLING
- OIL COUPLING

INCIDENTS
- OIL LEAKAGE

REMEDIES
- REINFORCEMENT OF SEALINGS

1300 MW N4 REACTOR PROGRAM

DESIGN
- METAL DISK COUPLING
- NO LUBRICATION

INCIDENTS
- NONE
2.4. SPEED INCREASER

INITIAL DESIGN
- "INDUSTRIAL" COMPONENT WITHOUT SPECIAL SPECIFICATION
- NO SPECIFIC CALCULATION
- GEAR MATERIALS: QUENCHED AND TEMPERED STEEL

INCIDENTS
- GEAR DAMAGE (PITTING) AFTER FEW RUNNING HOURS (< 1000 H)
- HIGH NOISE LEVEL ON DAMAGED GEAR

REMEDIES
- EQUIPMENT SPECIFICATION FOR SPEED INCREASER
→ APPLICATION OF AGMA STANDARD FOR CALCULATION
→ GEAR MATERIALS: CASE HARDENED STEEL
3. IMPROVEMENTS FOR SAFETY INJECTION FUNCTION

3.1. STARTING UP WITHOUT THE AUXILIARY START UP LUBRICATION PUMP

INCIDENTS:

- THRUST OIL TANK TOO SMALL
- LUBRICATION OIL INLET TIME TOO LONG

REMEDIES:

- MODIFICATION OF THE OIL CIRCUIT PIPING PROFILE TO INCREASE THE OIL TANK AND REDUCE THE LUBRICATION OIL INLET TIME.
- ELIMINATION OF THE DRAINING OF PIPING DURING A LONG STOP.

THE QUALIFICATION WAS PERFORMED IN OUR WORKSHOP AND THE ABOVE MODIFICATIONS APPLIED TO ALL CONTRACTS
3.2. HYDROSTATIC BEARING ARRANGEMENTS

INCIDENTS:

- MAXIMUM FLOW DURING SAFETY INJECTIONS = 160 m³/h
  → ΔP FEEDING OF THE BACK HYDROSTATIC BEARING TOO LOW → DAMAGE DUE TO SEIZING

- CIRCULATION OF SOLID PARTICLES POSSIBLE → DAMAGE DUE TO SEIZING OF HYDRAULIC GASKETS AND OF THE HYDROSTATIC BEARING

REMEDIES:

- INCREASE OF THE ΔP FEEDING OF THE BACK HYDROSTATIC BEARING (2 STAGES INSTEAD OF 1)
- ADDITION OF A LABYRINTH ON DIFFUSOR STATIONARY GASKETS

- PITTING OF THE BACK HYDROSTATIC BEARING FEEDING IN A CALM AREA AND PROTECTED BY LAMINATION

THE QUALIFICATION WAS PERFORMED ON THE E PEC LOOP AND ABOVE MODIFICATIONS APPLIED TO ALL CONTRACTS

FRAMATOME
4. NOISE LEVEL OF CHARGING PUMPS

IN FIRST PLANTS, HIGH NOISE LEVEL: 98 dB(A)

→ INVESTIGATION AND RESEARCH

→ IMPROVEMENTS IN NEW PLANTS:

  - SEPARATION OF FRAMES FOR PUMP AND SPEED INCREASER

  - LUBRICATION STATION SEPARATED

  - USE OF FLEXIBLE TUBING BETWEEN COMPONENTS OF LUBRICATING SYSTEM

  - USE OF CASTED MATERIAL FOR GEAR CASING

→ DIMINUTION OF NOISE LEVEL UP TO 87 dB(A)
   FOR THE "N4" CHARGING PUMP DURING WORKSHOP TEST
5. **CONCLUSION**

**FUTURE TRENDS**

- The demonstration of the charging pump's reliability leans on 170 manufactured pumps with 154 in service in France and in the world.

- In future plants, our objective:
  - Increase speed's suppression
  - Use of variable speed motor
  - Increasing of the rotating speed
  - Pump more compact
DESIGN MODIFICATIONS CARRIED OUT ON RCP AT C. TRILLO I

JUAN M. BURRIEL
GONZALO BLASCO

NOVIEMBRE 1990
The present paper expose different problems that happened in the RCP at C. Trillo I. The pumps were supplied by ANDRITZ-SIEMENS (KWU) and the solutions to the problems involved modifications of the original design.
REACTOR COOLANT PUMPS. FEATURES

VIBRATION MONITORING SYSTEM OF THE RCP

REMOVAL OF FLOW RECTIFIERS. REPLACEMENT OF CAST STEEL DIFFUSERS BY FORGED ONES

REPLACEMENT OF HP HOSES OF THE LIFTING OIL SYSTEM

SYSTEM TO COLLECT OIL DROPS AND PROJECTIONS

LOWER RADIAL BEARING PROBLEMS
REACTOR COOLANT PUMPS. FEATURES

- THE REACTOR COOLANT PUMPS ARE PROPELLER TYPE, VERTICAL SHAFT, SINGLE-STAGE, CENTRIFUGAL PUMPS.

- THE RCP ARE SIMILAR IN DESIGN AND MAIN CHARACTERISTICS TO THOSE OF OTHER INTERNATIONAL SUPPLIERS.

- THEY HAVE THE AUXILIARY SYSTEMS NECESSARY FOR THE OPERATION: OIL SUPPLY, SEAL WATER, LEAKAGE CONTROL, COOLING WATER AND NITROGEN SUPPLY.

- FROM THE DESIGN AND CONSTRUCTION POINT OF VIEW IT'S CONVENIENT TO POINT OUT THAT ITS CASING HAS NO WELDING, WHICH ASSURES ITS INTEGRITY, FACILITATES ISI, ETC.

- ON THE OTHER HAND THE INTERNALS DESIGN IS RATHER COMPLEX WHAT MAKE DIFFICULT THE DISMANTLING AND ASSEMBLY OF THE PUMP AND INCREASE THE DOSES TO WORKERS.

- THERE IS A STUDY TO IMPROVE THE PUMP DESIGN GROUPING PIECES AND MODIFYING THEIR CONSTRUCTION.
* MAIN CHARACTERISTICS:

HEAD 104.2 m WATER COLUMN

FLOW 25 685 m³/h = 7.135 m³/s

DENSITY 741.7 kg/m³

DESIGN TEMP. 3500°C

DESIGN PRES. 176 bar

MOTOR POWER 9870 kW COLD

7320 kW HOT

SPEED 1482 r.p.m.
VIBRATION MONITORING SYSTEM OF THE RCP

* At the end of 1987 a vibration monitoring system (IRD Mechanalysis Model 5915) was incorporated to the RCP.

- Each pump is provided with 5 sensors (3 in the upper bearing of the motor and 2 in the axial bearing of the pump).

- Other 2 sensors (of non-contact type) are placed in the zone of the seal system of the pump.

- The signals originated are centralized in a monitor placed in a room close to the control room. The overall vibration level and the alarm values are observed.

- The overall vibration values indicated in the monitor are written down by hand every week, and their trend is observed.

- Frequency analysis of the signals are performed monthly and their evolution is studied.

- The data collection and signals analysis periods are performed daily in case of abnormal condition of the pump.
BESIDES THIS SYSTEM, THERE ARE FOR EACH PUMP 2 SENSORS OF RELATIVE DISPLACEMENT (THEY BELONG TO THE VIBRATION MONITORING SYSTEM OF THE PRIMARY CIRCUIT).

- WITH THESE SENSORS THREE REGISTERS OF THE TEMPORAL SIGNALS ARE PERFORMED IN A MAGNETIC TAPE AT BOC, MOC AND EOC.

- THESE SIGNALS ARE ANALYZED AND POWER DENSITY SPECTRA ARE OBTAINED IN THE ZONE OF FREQUENCIES 0-50 Hz, COHERENCES AND PHASES. VARIATIONS ARE STUDIED WITH RELATION TO BEGINNING OF THE FIRST CYCLE.
REMOVAL OF FLOW RECTIFIERS
REPLACEMENT OF CAST STEEL DIFFUSERS BY FORGED ONES

* AFTER HOT FUNCTIONAL TEST 1 (540 HOURS OPERATION).

- CRACKS AND FISSURES APPEAR PRODUCED BY FATIGUE IN FLOW RECTIFIERS AND DIFFUSERS.

- A PLAN TO INVESTIGATE CAUSES WAS ESTABLISHED. IT WAS CONSIDERED FATIGUE DUE TO VIBRATION AS THE MAIN CAUSE.

- THERE WERE INTRODUCED MODIFICATIONS TO REDUCE STRESSES. LARGER TRANSITION RADII, SHOT PEENING.
* HOT FUNCTIONAL TEST 2 (680-780 HOURS OPERATION).

- AFTER THE SUBSEQUENT INSPECTION, DAMAGES SIMILAR TO THOSE PREVIOUSLY INDICATED WERE OBSERVED.

- IT WAS CONSIDERED THAT THE EXCITATION PRODUCED BY THE PREVIOUS RECTIFIER WAS IN RESONANCE IN THE CRITICAL ZONE OF THE FREQUENCY OF BLADE PASSING.

- THE RECTIFIER WAS REMOVED (REPLACED BY A LINER).

- CAST STEEL DIFFUSERS WERE REPLACED BY FORGED ONES THICKER AND WITH A LARGER TRANSITION RADIUS (X5CrNi134 MATERIAL).

- THE VIBRATION RESONANCE EFFECT WAS AVOIDED BY THE LARGER STIFFNESS AND MASS OF THE NEW DIFFUSER.

* NO PROBLEMS AFTER 1989 AND 1990 REVISIONS.
REPLACEMENT OF HP HOSES OF THE LIFTING OIL SYSTEM

- The HP hoses of the lifting oil system appear broken and with erosion due to friction. It was detected before criticality in May 1988.

- It was considered to be due to installation faults. KWU-ANDRITZ sent new instructions to make easy the right installation.

- Damages were still being produced. KWU studied the possibility to replace the HP hoses by 2-loop metallic tubes.

- A calculation program that justified it was performed and fatigue tests were carried out in a laboratory.

- The HP hoses were replaced by A.M. tubes during first refueling outage.

* KWU completed in July 1990 the fatigue tests of the new tubes, with satisfactory results.
SYSTEM TO COLLECT OIL DROPS AND PROJECTIONS

* INITIAL DESIGN.

- THE OIL SYSTEM AUXILIARIES (TANK, PUMPS, HEAT EXCHANGERS, ETC.) WERE PLACED IN AREAS INDEPENDENT TO THOSE OF THE RESPECTIVE PUMPS.

- THE PUMP DESIGN PREVENTS THE POSSIBILITY OF OIL DROPS OR PROJECTIONS OUTWARDS.

- A GATHERING TRAY FOR OIL DROPS THAT COULD LEAK FROM FLANGES OR VALVES WAS PLACED.
* REQUIREMENTS OF THE "CSN" (NUCLEAR SAFETY COUNCIL) TO GRANT THE OPERATION LICENSE.

- TO MINIMIZE THE RISK OF FIRE IN THE ZONE OF THE PRIMARY PUMPS DUE TO OIL DROPS OR PROJECTIONS, THE CSN REQUIRED IMPROVEMENTS TO BE IMPLEMENTED.

- THE TRAY SIZE WAS ENLARGED TO GATHER THE MAXIMUM AMOUNT OF OIL WITHIN THE PUMP AREAS. INTERFERENCES WITH OTHER ELEMENTS WERE SOLVED.

- A NEW PIPING SYSTEM WAS INSTALLED FROM THE GATHERING TRAYS TO A NEW TANK WITH LEVEL AND ALARM INSTRUMENTATION.
LOWER RADIAL BEARING

* INITIAL DESIGN:

- CARBON BEARING WITH ANTIMONY AS AGGLUTINANT.

- BEARING HOUSING OF X5CrNi134.

- BEARING SLEEVE OF X5CrNi134 LOWER PART STELLITED ATTACHED TO THE UPPER PART WITH SCREWS OF X22CrNi17.

* INSPECTION AFTER HFT 1 (AUGUST 1987).

- SURFACE SCRATCHES WERE DETECTED ON THE STELLITED ZONE OF THE BEARING SLEEVE (PUMPS 10 AND 30) LOCATED MAINLY IN THE LOWER PART. THEY WERE CONSIDERED ACCEPTABLE.
HFT 2 (APRIL 1988).

THE BEARINGS AND THE BEARING SLEEVES OF THE THREE PUMPS WERE INSPECTED, BEING DETECTED SCRATCHES AND PORES SIMILAR TO THOSE OF THE PREVIOUS INSPECTION.

FIRST REFUELING OUTAGE (OCTOBER 1989): ONLY PUMP 30 WAS DISMANTLED.

THE BEARING WAS REPLACED BY ANOTHER SIMILAR WITHOUT ANTIMONY TO AVOID RADIOLYSIS PROBLEMS OF THE PRIMARY COOLANT.

THE BEARING SLEEVE WAS INSPECTED BEING DETECTED A LARGE NUMBER OF SCRATCHES IN THE STELLITED ZONE THAT DO NOT AFFECT THE BASE MATERIAL.

THIS BEARING SLEEVE WAS REPLACED BY ANOTHER NEW ONE.
* REACTOR SHUT-DOWN OF FEBRUARY 1990: PUMP 30.

. TEMPERATURE RISE IN THE BEARING ZONE BEYOND THE ALARM SETPOINT OF 1200°C. THE PUMP WAS SHUT-DOWN.

. IT WAS CHECKED THAT IN PARALLEL WITH THE RISE IN TEMPERATURE THERE WAS AN INCREASE OF THE VIBRATIONS MEASURED IN THE SHAFT.

. IN THE INSPECTION OF THE INTERNALS IT WAS DETECTED: A CRACK OF NEARLY 300 mm LONG AND 2 mm WIDE IN THE BEARING SLEEVE LOWER PART.

. IMPORTANT DAMAGES OF EROSION AND MATERIAL LOOSENESS IN BEARING AND ASSOCIATED SUPPORT PARTS.

. ALL DAMAGED PARTS WERE REPLACED BY SPARE PARTS. THE BEARING SLEEVE LOWER AND UPPER PARTS WERE REPLACED BY A NEW ONE INTEGRAL PIECE OF X5CrNi134 WITHOUT STELLITED.

. A TEST PROGRAM WITH THE DAMAGED PIECES WAS PREPARED TO DETECT THE CAUSES OF THIS PROBLEM, INITIALLY CONSIDERED DUE TO A SEAL GASKET FAILURE.
* PUMP 10 SHUT-DOWN ON SEPTEMBER 1990.

  TEMPERATURE RISE IN THE BEARING ZONE BEYOND THE ALARM SETPOINT. THE PUMP 10 WAS SHUT-DOWN AND 2-LOOP OPERATION PROCEEDS (NEARLY 50% FULL POWER) UNTIL BEGINNING OF REACTOR SHUT-DOWN WHICH WAS SPEEDED UP TO THE 24TH SEPTEMBER.

  IT WAS ESTIMATED THAT THE PROBLEM COULD BE SIMILAR TO THE ONE HAPPENED IN THE PUMP 30 ON FEBRUARY 1990.

  IT WAS DECIDED TO DISMANTLE AND INSPECT THE INTERNALS OF THE THREE PRIMARY PUMPS DURING THE 2ND REFUELING OUTAGE.
* INSPECTION RESULTS AND CONCLUSIONS.

. AT THE INSPECTION OF THE INTERNALS IT WAS DETECTED A CRACK AND DAMAGES VERY SIMILAR TO A.M. IN RELATION WITH PUMP 30.

. THE CRACK AND SCRATCHES OF THE STELLED BEARING SLEEVES WERE PRODUCED BY VIBRATION FATIGUE. ALSO SOME GASKETS FAILURES COULD CONTRIBUTE, DUE TO TEMPERATURE RISE, TO THE DAMAGE.

. THE BIGGER LOADS ON THE BEARING (ABOUT 30% MORE) ARE ALSO A SIGNIFICANT DIFFERENCE WITH OTHER SIMILAR PUMPS.

* TO AVOID PROBLEMS IN THE FUTURE:

. THE BEARING SLEEVES WILL BE MANUFACTURED WITHOUT STELLED.

. THE BEARING SLEEVE MUST SUPPORT THE THERMAL LOADS PRODUCED BY A GASKET FAILURE.

. THE ACTUAL ASBESTOS-RUBBER GASKETS SHOULD BE CHANGED BY GRAPHITE.

. THE SEAL WATER FLOW WILL BE INCREASED AS MUCH AS POSSIBLE.
critical machinery.
Modificaciones en el rectificador

Ausgangszustand
Estado original

modifiziert
modificado
Skizze
Austrittskante Leitschaufel

Z Croquis
Borde de salida de la paleta directriz

Difusor fundido
Modificación (1.4313)

Guss-Leitapparat
Modifikation (1.4313)

Schnitt A-A
M1: 2

Detalle
Detail Z
M1: 2

Anlage 2: Modifikation des Leitapparates (Guss)
ANEXO 2: Modificación del difusor (fundido)
Aenderungen am Gleichrichter

Modificaciones en el rectificador

Ausgangszustand
Estado original

modifiziert
modificado

Einlaufring
camisa

Anlage 1: Ausführungsformen Gleichrichter / Einlaufring
ANEXO 1: Versiones rectificador / camisa
Skizze
Austrittskante Leitschaufel

Z borde de salida de la paleta directriz

Geschmiedete Ausführung
Ejecución forjada

ANEXO 6: Forma del borde de salida de la paleta directriz del difusor forjado

Anlage 6: Ausbildung der Austrittskante der Leitradschaufel des geschmiedeten Leitapparates
Tuerca racor aplicada antes
Überwurfmutter vor Schweißen
daufgelösten de sbldar

Presión de prueba 250 bar
Abpreßdruck 250 bar

Ver plan de soldadura SP 1
Der Schweifplan SP 1

Anexo 4 al informe de trabajo Siemens
U9 122/88101

Anlage 4 zum Siemens-Arbeitsbericht U9 122/88101

1:1 Tubería de aceite de alivio

2HR100575
AO301: OIL SUPPLY SYSTEM ROOM OF RCP-10; PLANT EL. +1.250
AO311: PRIMARY SYSTEM FLOOR; PLANT EL. +0.000
A0407: RCP-10 ROOM; PLANT EL. FROM +0.000 TO +18.300
A0423: RCP-20 ROOM; PLANT EL. FROM +0.000 TO +27.360
A0428: OIL SUPPLY SYSTEM ROOM OF RCP-20; PLANT EL.+3650
A0431: OIL SUPPLY SYSTEM ROOM OF RCP-30; PLANT EL.+3650
A0442: RCP-30 ROOM; PLANT EL. FROM +0.000 TO +18.300
BAND. ACEITE DE BOMBAS PRINCIPALES

NOTAS

1. Se dispondrá de 12 apoyos seguido detalle de anclaje repartidos de forma homogénea.

2. El anclaje de la bandeja se hará seguido detalle; se dispondrá de una placa guía de medidas a definir en obra con taladro rascado en la dirección 210°-90° y sentido 210°-89°. Para permitir el desplazamiento de la bandeja; esto es aplicable a todos los soportes de la bandeja.

3. La bandeja exterior tendrá una pendiente del 2%.

4. El sector de bandeja comprendido entre 5° a 15° sera de 11400 mm.

5. En bomba N° 3 la bandeja sera de 450 mm. de ancho.

6. El material de las bandejas es ACR110X. 204-L
"Reactor Coolant Pump Seal Failure Considerations"
J. E. Jackson, R. L. Baer, W. Minners and C. J. Heltemes
U. S. Nuclear Regulatory Commission

Introduction
Reactor coolant pump (RCP) seal failures of a magnitude which could result in a loss-of-coolant accident (LOCA) have been the subject of Generic Issue 23 (GI-23). The concerns have been divided between (1) normal operation failure rates which appear to exceed the previous assumptions for small break LOCA's by an order of magnitude and (2) the potentially large RCP seal leak rates which could result from loss of seal cooling during events such as station blackout. A number of research studies supporting resolution of GI-23 have added to a better understanding of the potential RCP seal failure modes and the magnitude of the resulting leak rates.

Tests to evaluated the effects of loss of seal cooling on RCP seals under station blackout conditions were performed by Atomic Energy of Canada Limited (AECL) under NRC sponsorship. The results of these tests and analyses are contained in NUREG/CR 4077 and 4821. This program included 1) axial seal extrusion tests to determine the extrusion resistance of polymer seal materials, 2) frictional force tests to measure the drag forces for extruded polymer seals, and 3) tests and calculations to identify the parameters which effect seal face stability under conditions of two-phase flow.

Extrusion Tests
AECL performed a series of tests to assess the extrusion resistance of some RCP polymer seals under station blackout conditions. These tests were primarily aimed at assessing Westinghouse polymer seal performance under loss of cooling conditions.
Polymer seals are used as both static seals to provide leak tight joints between certain mating parts of the RCP seal assembly and as secondary seals to prevent leakage between RCP seal components that undergo limited relative motion during operation. In order to conduct seal assessment tests, it was necessary to identify certain seal design features such as the sealing gap, differential pressure, and materials employed for both static and secondary seals under loss of cooling conditions.

Typical elastomer polymer seal materials used in various RCP seal assemblies are as follows.

Westinghouse RCP seals contain static elastomer O-rings of Parker E515-80 ethylene propylene and filled (proprietary material) Teflon "Channel" seals of Tetrafluor Inc. Tetralon 720, design TF388. The channel seals are energized with O-rings made from E515-80. Byron Jackson secondary seals utilize U-cups formulated of nitrile elastomer; the BJ static seals are typically nitrile O-rings. Bingham-International secondary seals employ Parker ethylene propylene O-ring compound E740-75, size 2-349, backed with Parker "Parbak" anti-extrusion rings of ethylene propylene compound E652-90. The Bingham-International static seals are E740-75 O-rings.

Results of the AECL high temperature extrusion tests on E515-80 O-rings, E740-75 O-rings, and Tetralon 720 channel seals energized with E740-75 O-rings are shown in Figures 1 and 2. Tests under worst-case station blackout conditions indicated that a blowout failure was probable with Parker ethylene propylene compound E515-80 O-rings currently used in the Westinghouse RCP seal. Tests on Tetralon 720 channel seals (backed with E740-75 O-rings) revealed severe extrusion of Tetralon 720 in many tests. At small gaps this extruded material prevented subsequent sealing by the O-ring and a blowout failure occurred.
At large gaps, the Tetralon extrusion was sufficient to allow proper O-ring sealing, and the results corresponded to those for O-rings alone. The expected gaps for those areas where channel seals are used in the Westinghouse RCP seal assembly are large enough that a blowout failure is not expected, if the channel seal is energized with a proven high temperature O-ring, such as E740-75.

The tests have shown that Parker ethylene propylene compound E740-75 O-rings exhibited superior resistance to high temperature extrusion. Westinghouse is currently developing a replacement O-ring material with superior high temperature extrusion resistance for use in their RCP seals.

A comparison of potential seal gaps in Bingham-International seal assemblies with the results of tests documented in NUREG/CR-4077 indicated that one static O-ring in each stage of the Bingham-International seal assembly would experience gap and pressure conditions which could result in a potentially significant probability of extrusion failure if subjected to full system pressure during a station blackout. This O-ring is located between the OD of the stationary ring and its carrier, where the gap could exceed 0.02 inch depending on tolerances in fabrication and assembly and upon relative thermal expansion. The nitrile compounds used in Byron Jackson static and secondary seals have very good extrusion resistance and are not expected to fail, by extrusion, in station blackout conditions. This is a result of the nitrile U-cups becoming extremely hard after exposure to high temperature. However, this hardening is so severe that the material effectively ceases behaving as a seal. This could lead to seal failure under certain perturbed conditions although no leakage occurred in any of the AECL tests during static high temperature exposure. In addition, this hardening could significantly increase the friction in the BJ secondary seals if
they are required to slide over worn or roughened surfaces.

As a result of Phase I testing and analysis (NUREG/CR-4077), AECL suggested the following possible design changes to improve RCP seal performance during a station blackout event.

- Elimination of staging flow - to reduce the maximum temperature seen by the seals on Byron Jackson and Bingham-International seals.

- Thermal syphoning circuit for stationary cooling - to reduce the maximum temperature encountered by the seals.

- High temperature resistance elastomers - to make best use of available materials on Westinghouse seals.

- Addition of a back-up "safety" seal to each cartridge - to provide a seal designed for the job.

- Increase in seal closing force (balance ratio) under station blackout conditions - to prevent "popping open."

- Increase in flow restrictions, particularly towards the exit from the seal cartridge - to reduce "worst-case" leakage rate by insuring choked flow.

- Design the seal to promote divergent face deflection due to flashing - to reduce the susceptibility of the seal "popping open."

- Steam-driven cooling system - to provide a means of cooling that is independent of the station blackout event.
Friction Tests
In addition to the evaluation of static O-rings, the subject research included tests to determine the friction forces developed between degraded secondary seals and the movable hydraulic seal rings. Three secondary seal types were investigated—channel seals energized by O-rings (used in Westinghouse Electric Corporation seals), U-cups (used in Byron Jackson Seals), and O-rings with backup rings (used in Bingham International seals). The cross sections for the seals are shown in Figures 3. The seals were chosen for testing on the basis of being typical of the polymer seals between the movable seal ring and its supporting structure.

Operation of a secondary seal depends on the pressure differential across the secondary seal to tightly press it against the two surfaces that define the gap to be sealed. For the Westinghouse channel seal, the pressure acts on both the channel seal and energizing O-ring but the actual seal occurs at the contact region between the metal surfaces and the channel seal. The Byron Jackson U-cup flexes when loaded with the pressure and seals against the metal surfaces. The Bingham International seal is formed by contact of the O-ring and backup ring to the metal surfaces. The backup ring, made of harder polymer material, serves to reduce the effective size of the gap that is sealed by the O-ring and prevents extrusion of the softer O-ring into the gap.

All secondary seals tested at half scale (4 1/2-in. shaft diameter). The extrusion resistance of a polymer seal is considered to be independent of changes in hoop (shaft) diameter for thin-section, flexible seals such as those tested, since extrusion of such seals is a localized phenomenon, and moments and forces due to shear and twisting cannot be transferred very far around the circumference of the seal. Similarly, the axial drag force per unit circumference is considered to be independent
of seal hoop diameter for seals of the large hoop diameter to cross-sectional diameter ratios in this study. Results of other AECL tests on O-rings, channel seals and U-cups of different sizes have substantiated these premises.

Tests were conducted on Westinghouse channel seals with a diametral clearance of 0.009 inches, 550°F temperature and a test pressure of 1800 psig, the calculated "worst-case" conditions for a station blackout event. The highest axial drag recorded for a pair of 4 1/2 inch diameter seals was 210 lbs. or 7.4 lb/in. of circumference per seal.

Similar tests on Byron Jackson type U-cups resulted in the material becoming embrittled to a glass-hard state, with permanent set to the shape of the holder and shaft. Leakage did not occur, but fracture of the U-cup occurred on stroking (0.12 in. of movement) during one test, resulting in high leakage. The highest axial force measured for a pair of 4 1/2 in. diameter U-cups was 625 lbs. or 22 lb/in. of circumference per seal. This force is partially compensated for with the approximate 200 lb of closing force exerted by the axial loading springs in the Byron Jackson design. Because the seals become so hard, it was judged that in an actual seal assembly the friction would be much higher due to molding of the seal into typical surface irregularities (pitting and wear) on the shaft. In this event there is also the possibility of pieces breaking from the sealing lip when the shaft moves axially.

Tests of the Bingham International seals (E652-90 Parbak backup rings with E740-75 O-rings) showed that although the backup rings underwent various degrees of extrusion, leakage did not occur in any of the tests. The maximum drag force measured for two sets of seals of 4 1/2 in. diameter was 660 lb, or 23.3 lb/in. of circumference per seal. Similar to the case of the Byron Jackson design, axial loading springs would provide a closing force of
about 350 lb for a full-size 9 1/2 in. seal and would partially compensate for the increased drag force.

The friction force combines with the hydraulic and spring forces to determine the overall balance of forces on the movable seal ring. The magnitudes of the spring and hydraulic forces are obviously very dependent on the specific geometries of the assemblies. The friction loads determined for Westinghouse shaft seals are less than 1% of the nominal hydraulic closing force in the first stage. Therefore, the overall balance of forces in the first stage is basically unaffected by friction from the channel seal, even in its degraded condition. Friction forces in the second stage make up a larger portion of the total closing force, and the increased friction can be significant with respect to operating margins.

The nominal hydraulic closing forces in both the Byron Jackson and Bingham International shaft seals are less than in the Westinghouse units. Given the higher friction loads produced by degraded secondary seals, friction has more effect on these shaft seals than for Westinghouse. The friction forces are about 9% of the hydraulic closing forces for both Byron Jackson and Bingham International shaft seals. This is a relatively small part of the total load on the movable seal ring and the influence on the behavior of the primary seal is small for many station blackout conditions. There are, however, some conditions in which minor changes in the closing forces on the movable seal ring can cause a relatively large change in the margins to unstable primary seal behavior.

Seal Stability
Analytic and experimental investigations of the stability of primary seals under station blackout conditions were performed by AECL (NUREG/CR-4077 and NUREG/CR-4821). The designed gap in a primary seal stage is maintained by a balance of the hydraulic,
spring, and friction forces acting on the seal rings. The increased, off-design temperatures (−500°F) experienced during a station blackout can cause degradation of polymer seal components and also affect the pressure distribution of the flow through the gap between the seal rings. Increased pressure loading between the seal rings has the potential of causing the primary seal to open to a large gap (bistable seal), or to pop completely open (unstable seal), resulting in increased leakage.

Since RCP end face seals are hydraulically balanced, the closing load due to the axial loading springs (when included in the design) and the pressure on the back of the seal rings is balanced by the hydraulic opening load due to the pressure between the seal faces. For small variations in seal face separation, the closing load is essentially independent of seal face separation. The seal faces are normally separated by a very thin lubricating film of water. If the seal face gap is convergent, then the movable seal ring will move towards or away from the other seal ring until it reaches an equilibrium position. For a small seal face separation, pressure is high across most of the seal face and drops sharply at the inside; for a large seal face separation, the pressure profile across the seal face approaches a linear drop from the outside to the inside. Thus, the opening force, which is equal to the integral of the pressure across the seal face, decreases as seal face separation increases. The seal rings move apart until the opening and closing forces are equal. This is the stable operating position for the seal.

When the temperature of the water entering the seal gap is greater than the saturation temperature at the downstream pressure, some water will flash to steam in the gap or at the exit. Flashing causes the pressure profile to become less linear because the pressure drop becomes concentrated in the two-phase region near the exit. As the fluid flashes, the pressure
gradient increases because the fluid velocity increases due to its greatly increased specific volume. This is aggravated by the further expansion of the vapor as the pressure decreases through the gap. Also, when the flow chokes at the exit of the gap, there is a sharp drop in pressure. Concentration of pressure drop near the exit increases the opening force, so the seal rings move apart until the opening force is reduced to match the closing force. Unfortunately, when flashing becomes very significant, the opening force does not continue to drop as the rings move apart, but instead it reaches a minimum and then increases again for wider gaps. If the closing force is less than the minimum opening force, then the seal is unstable and will open to the limits of axial travel permitted by the seal configuration.

The AECL analysis indicated that the pressure distribution could change to a degree sufficient to cause bistable or unstable operation by the increased fluid temperatures and subsequent flashing associated with a station blackout. The primary parameters influencing seal stability were determined to be:

- Inlet fluid conditions.
- Back pressure on the discharge side of the seal rings.
- The radial taper in the gap.

Additional factors affecting the seal-assembly leakage rate are seal ring surface roughness and exit area. The exit area is a significant parameter in two-phase flow since choking is likely to occur near the exit. Therefore, the leakage rate for fully opened seals can be minimized by designing the exit areas as small as possible.

Stability regimes, based on these primary parameters, are presented in NUREG/CR-4821 for a single seal, representative of one stage of a typical RCP seal-assembly. In general, unstable
behavior can be expected for inlet fluid conditions near saturation if the seal rings are exposed to low back pressure, unless the seal faces are divergent and are not worn or warped enough to cause significant leakage.

It was not possible to predict seal leakage during unstable operation due to geometrical complexities and differences between individual designs. However, it was generally concluded that high leakage flow will develop during station blackout unless the seal-face convergence is low (~10 μin.) or the back pressure is high. Limited analysis suggests that leakage can be limited to less than 10 gpm if back pressure is greater than 50% of saturation pressure at inlet temperature, convergence is not more than 100 μin., and the balance ratio is greater than 0.63.

Even though stability of the seal is a necessary condition for limiting leakage, it is not sufficient to assure acceptable leak rates. A calculation at 2250 psig with a convergence of 300 μin. resulted in a leakage rate in excess of 200 gpm, even though the seal was operating in a stable manner. The leakage rate is a strong function of the primary stability parameters and the closing balance ratio, but the functional form of this dependence has not been determined.

Based on two-phase flow calculations and test results, AECL arrives at the following conclusions on the effects of two-phase flow resulting from station blackout conditions (NUREG/CR-4821):

1. All RCP seals are at risk of popping open under two-phase flow conditions. The principal cause of instability is a result of the hydraulic opening force between the seal faces exceeding the net closing force on the movable seal ring. Thus, the potential for popping open is increased for seals with low balance ratios, especially if secondary seals, degraded by
friction, further reduce the closing balance ratio. (Bingham-International and Westinghouse seals have significantly lower balance ratios than do Byron Jackson seals).

2. RCP seals will remain stable during station blackout provided that:

a. Inlet pressure is sufficiently above saturation,
b. Back pressure is sufficiently high, and
c. Seal face convergence is sufficiently low, or
d. The seal faces are divergent and are not roughened or warped enough to cause significant leakage.

3. Seals with divergent faces are not immune from instability and will pop open if a maximum stable leakage is exceeded.

4. Fast transients in fluid temperature could cause a seal to pop open before beneficial thermal distortions take place because the fluid mechanics time constant is so much less than the time required for thermal distortions to take place.

5. The inner and outer diameters of the seal face are of secondary importance in determining its stability.

RCP Seal Tests
The results from some full scale tests are available for public disclosure with restrictions on certain details. These details are considered proprietary by the sponsoring organizations.

A full scale test was conducted at a coal fired 250 MW generating station at Montereau, France in May 1985. The tested hardware was a 7-in. Westinghouse (W) seal cartridge, typical of the seals
8-in. seals used in U.S. RCPs. The most significant differences between the tested 7-in. and 8-in. designs used in U.S. RCPs include polymer seal materials, seal ring thicknesses and mounting or support configurations, flow restriction downstream of the gap between seal rings, and the balance ratio of the second stage seal.

The most important differences in design are those of component geometry and more critically, balance ratio of the second state (No. 2) W seal. Saturated inlet conditions, low back pressure, and low balance ratio make the No. 2 seal highly susceptible to popping open under two-phase flow conditions. The 8-inch No. 2 seal has a significantly lower balance ratio than the 7-inch seal, increasing its potential for instability. Once maximum stable leakage is exceeded via scratches on the seal face, accumulated wear, general degradation and/or a momentary leakage spike during initiation of station blackout, the high divergence of the No. 2 seal gap may be insufficient to prevent it from popping open. This would release the upstream backpressure and impact heavily on the stability of the No. 1 seal. Thus the stability of the W No. 2 seal under two-phase flow conditions remains a major concern for the overall integrity of the seal package.

The 8-inch seal has a much thicker No. 1 seal faceplate which would have a much slower thermal response during a station blackout. This lower susceptibility to distortion caused by pressure and temperature would appear to be beneficial under station blackout conditions. However, under normal operating transients, a slow response to heat generated by momentary rubbing contact can lead to premature degradation and possibly failure. Therefore, in actual use, an 8 inch seal could have a higher degree of degradation at the initiation of a station blackout and therefore be more susceptible to failure than the 7-inch seal.
Other differences between the 7-inch and 8-inch designs are in the location of the hydrostatic clamping load with respect to the faceplate centroid and the filler used in the teflon secondary channel seals. The clamping load application above the centroid on the 8-inch seal would adversely affect distortion and the resultant leakage. Also the 7-inch seals tested have carbon-filled teflon secondary seals, while the 8-inch seals have organic-filled teflon, that was shown to extrude at the stated test temperature.

The E515-80 O-ring material used in Westinghouse seals presents the possibility of partial extrusion at high temperatures as indicated by the tests conducted by AECL. A different material with the same label was used in the full scale test and no extrusion occurred, although there was evidence of permanent set on all O-rings present in the test carridge. However, there are significant differences between the elastomers tested in the 7-in. seal and those typically installed in Westinghouse U.S. RCPs.

The test procedure was the result of necessary compromises to meet the objectives of the participants and to stay within facility capabilities. Actual predicted station blackout sequences were not accurately duplicated. The general sequence consisted of establishing near normal seal operating conditions of 2200 psig and 100°F for about one hour, increasing temperature to an intermediate level and holding for an hour, and then further increasing temperature to the maximum obtainable from the coal fired facility (near 540°F). Pressure was then decreased to about 1300 psig and held; temperature was reduced; pressure was reduced to about 600 psig and held for 15 hours; then the test rig was depressurized and allowed to cool down.

The following observations were made during the test and the subsequent disassembly of the seal:
When hot water was introduced to the seals, initial thermal distortion caused a brief leakage spike of about 61 gpm through the first stage seal. Otherwise, leakage remained controlled with no indication of inherent instability. There was indications that the second stage seal did open briefly to allow some leakage, but then reclosed.

There was little indication of extrusion of any of the O-ring seals or channel seals; however, all O-rings exhibited some degree of permanent set.

This full scale test provides valuable information to be considered in the overall effort to resolve the question of RCP seal performance during a station blackout. However, this information must be used with an understanding of the differences between the 7-in. seals that were tested and typical U.S. 8-in. seal assemblies.

Tests on a 4 1/2 inch Bingham International seal assembly similar to the 9-inch seal assembly installed in RCPs at the San Onofre Nuclear Generating Station were conducted on an operating boiler recirculation pump under station blackout conditions in November 1985 at Long Beach, California. The objective was a demonstration of seal performance for the first 30 minutes after loss of all seal cooling. Test results were later compared to AECL analytical predictions. Two tests were performed in which the seal inlet temperature was ramped up from normal operating temperature (120°F) to 485°F and 500°F in twenty and fifteen minutes, respectively. The seals operated at temperatures at or above 485°F for longer than 1-1/2 hours.

Examinations after the test did not reveal any damage to the rotating/stationary seal ring or secondary polymer seal. Comparisons of the measured performance of the third stage seal
to AECL calculations were subsequently made at selected points on the heatup ramp. No instability was observed throughout the test and the maximum leakage was 2 gpm. Under some conditions the seal assembly exhibited bistable behavior. Bistable behavior occurs due to limited fluid flashing and results in the seals opening to an new stable position where the separation gap is less than the fully open position associated with instability, but larger than the gap for normal stable operation.

The analysis predicted earlier instabilities and higher leakage than the tests showed. For two third stage tests where bistable or unstable behavior was predicted, stability was observed in one and bistability in the other. For a case where unstable behavior was predicted, bistability was observed.

These results led AECL to conclude that the analytical results are conservative for the conditions of these tests, but that the existence of a bistable regime is confirmed.

The pump seal cartridge from the Byron Jackson Pump at St. Lucie 2 was subjected to a simulated station blackout test, that is, no shaft rotation and plant operating temperature and pressure. The four seal stages were mounted in a cartridge which was connected to the Byron Jackson test loop in order to simulate actual plant conditions. The test was run for 50 hours without loss of seal function. The seal controlled leakage remained within normal limits (approximately 1.0 gpm) and vapor seal leakage did not exceed 0.25 gpm. Vapor seal leakage is that which leaks past the fourth seal and potentially into the containment. Controlled leakage exits the seal cartridge between the third seal and fourth (vapor) seal and is piped off to the volume control tank. Seal pressure breakdowns remained within acceptable limits and the vapor seal temperature reached approximately 400°F. Post test inspection showed a cracked vapor seal rotating ring and permanent compression of the "O" rings and hardening of the "U"
cups. After the test, the seal cartridge was rebuilt to operational standards by replacing the broken rotating ring and all elastomers.

The St. Lucie seal cartridge test did not fully simulate a station blackout event due to the following conditions. The seal inlet temperature and pressure remained constant throughout the test, therefore the inlet subcooling remained at a maximum, a condition which favors seal stability. Since, only the seal cartridge was tested and not a full pump, shaft thermal movement was not simulated. Also, the seal cartridge was new and therefore did not simulate normal seal wear which could be present.

**Probabilistic Risk Assessment**

Using the information on failure mechanisms of seal components an event tree model was constructed for the Westinghouse seal which systematically displays the degradation phenomena as nodal questions (branches), and provides the long-term leakage rates at the end-points of a given path in the event tree. Figure 4 illustrates the hypothetical success and failure paths in the event tree. O-ring extrusion (failure) is considered to be time-dependent, and therefore all event-tree paths containing this failure mechanism will have associated leakage rates which are time-dependent.

The significance of the flow leakage rates is evaluated by considering the probability and time at which core uncovering is postulated to occur. The time between event initiation and the onset of core uncovering is used as an indicator of the time frame available for recovery. A time-dependent probabilistic model was constructed based on experimental and theoretical investigations along with substantial engineering judgment to obtain the probability of core uncovering as a function of time. The model makes use of the event-tree based time dependent leakage rates
and the failure rate probabilities of seal components.

Failure probability distributions are obtained for the two failure mechanisms: seal face failure (combined popping open and seal ring binding) and O-ring extrusion. Only O-ring failure is considered to be time dependent.

The probability of exceeding the leakage rate that leads to the core becoming uncovered is calculated for each path for each hour. These probabilities are then summed over each hour, obtaining the cumulative probability of core uncovery as a function of time. Combining this distribution with the frequency of a station blackout event and the probability of failing to restore power in a given time yields the probability of core uncovery. The results are shown in Figures 5, 6 and 7 and Table 1.

No fully qualified O-ring material has been identified, the best currently available is an improved O-ring material. It is likely that the improved O-ring material would survive when subjected to the station blackout conditions predicted by Westinghouse (2250 psia, 550°F upstream of the first stage and an intermediate pressure and temperature between the first and second stages). However, failure of one of the face seals would result in the full system conditions (2250 psia, 550°F with no back pressure) being applied across the elastomer seals of the other stage. Also, at these full system conditions, the extrusion gaps would not be the same as the ones reported in WCAP-10541 for the predicted conditions. For the improved material to be conclusively shown to be "qualified" for these extreme conditions, test results would have to be provided for these O-rings and channel seals at the appropriate gaps.
Figure 1. Pressure vs. gap for short-term (typically 2- to 4-hour) blowout by extrusion of un lubricated static O-rings and channel seals in water at 520°F (NUREG/CR-4077)
Figure 2. Pressure vs gap for short-term (typically 2- to 4-hour) blowout by extrusion of unlubricated static O-rings and channel seals in water at 550°F (NUREG/CR-4077)
Figure 3. Typical secondary seals in RCP shaft seal assemblies.
<table>
<thead>
<tr>
<th>Loss of Seal Cooling</th>
<th>Does No. 1 Bind?</th>
<th>Does No. 2 Bind?</th>
<th>Does O-Rings Fail?</th>
<th>Does No. 2 O-Rings Bind?</th>
<th>Does No. 3 O-Rings Fail?</th>
<th>Resultant Leak Rate</th>
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Fig. 4. RCP seal leakage following loss of all seal cooling.
Figure 5. Probability of core uncover for the best estimate, upper bound, and lower bound models with unqualified O-rings. For interest, the effect on the lower bound model of reducing B3 from its assumed value of 1.0 to 0.54 is also plotted. All curves assume plant cooldown, except the one labelled "Best Estimate (Without Cooldown)."
Figure 6. Probability of core uncover for the best estimate, upper bound, and lower bound models with qualified O-rings.
Figure 7. Probability of core recovery for the best estimate, upper bound, and lower bound models with improved O-rings.
TABLE 1- CORE UNCOVERY FREQUENCY ESTIMATES FOR VARIOUS COMBINATIONS OF SEAL FAILURE ASSUMPTIONS AND BLACKOUT MODELS

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Cooldown</th>
<th>Qual. Sls</th>
<th>Seal Fail Mdl.</th>
<th>Blkout Mdl.</th>
<th>NRC Cluster</th>
<th>Core Damage Frequency (/Yr)</th>
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(1) This is the plant blackout model from NUREG-1032; 1 designates NRC cluster 1 (lowest plant blackout frequency) and 2 is NRC cluster 2 (average frequency of station blackout), and 5 is NRC cluster 5 (highest frequency of station blackout).
SESSION #5  
Chairman's Closing Remarks  
(Mr. M. Hada, NUPEC, Japan)

1. BWR designs tend to favour the use of internal pumps because of various advantages as mentioned in the presentation of Mr. Törnblom. He has described development experiences on external and internal pumps for main recirculation as well as conventional clean up and shutdown cooling system pumps. (As my colleagues presented their papers yesterday, Japan is also in the state of building ABWR's in the near future. Pumps for these plants are also internal pumps thus the presentation was very interesting).

2. Charging pump development has been explained by Messrs. Duhamel and Vauchel. Problems and countermeasures have been discussed and were interesting. Problems relating to relatively low flow and high head pumps would be common and also applicable to the pumps of similar purposes.

3. Pump design modification based on its operating experience was reported by Mr. Burriel of Trillo I plant. It was rather peculiar that the pumps of the same plant had such frequent troubles in a rather short period of time. It should also be noted that the countermeasures taken by the Trillo plant people were comprehensive and excellent in improving operating performance of its main coolant pumps.

4. Report of test to evaluate the effect of loss of cooling on RCP seals under station blackout conditions was made by Mr. Winners together with risk assessment on core uncovering and core damage probability. Test was made for actual seal materials used in RCP of existing designs to give the base of assessment. Results also suggest the possible design change to improve RCP seal performance. This kind of activity is recommendable for possible plant performance improvement.
PANEL DISCUSSION: "PUMPS IN NUCLEAR POWER PLANTS: SAFETY ASPECTS, RELIABILITY AND DEVELOPMENTS"

Chairman: K. Kotthoff

Panelists:
- M. Zwingelstein, EDF - SPT, France
- M.-C. Dupuis, IPSN, France
- W. Minners, NRC, United States
- F. J. Bernsteiner, KSB, Germany
- V. Sola, Trillo Nuclear Power Plant, Spain

Panel Subtopics:

* Measures and strategies for ensuring and increasing pump reliability;

* Impact of pump performance on safety; and

* Future developments in pump design.
Chairman's Introduction:

We are now coming to the last part of our specialists' meeting, the panel discussion, and I am very glad to have such competent panelists here. I will first introduce the panelists, each of whom will follow by making a short statement to stimulate discussion; the floor will then be opened for comments and general discussions.

You all already know Mr. Bernsteiner and Mr. Minners from previous presentations, and so I will not re-introduce them.

It is really a great pleasure for me to have Mrs. Dupuis with us because she has, among other things, been the chairman for more than five years of our Principal Working Group No. 1, which promoted this meeting. Both Mrs. Dupuis's competence and charming chairmanship, have contributed significantly to the success of our Group. Mrs. Dupuis is working for the Institute of Protection and Nuclear Safety in France, IPSN, and she started working there when the French PWR programme was launched. She worked first in the Safety Analysis of PWRs, and for the past four years has been deputy head of the Safety Analysis Department.

Next I would like to introduce Mr. Zwingelstein. He is from Electricité de France, EDF, and is deputy manager of the maintenance department of the Nuclear and Fossil Generation Division. He has an engineering degree from the Ecole Nationale Superieur d'Electrotechnique et de Hydraulique Controleque de Toulouse and a Ph. D. in automatic control from the Bordeaux University. He started work in 1966 as a group leader of the French Atomique Energy Commission, and from 1981 to 1990 he worked for Electricité de France in the Research and Development Division.

Finally, Mr. Sola is plant manager of the Trillo Nuclear Power Plant in Spain. He has an engineering degree, and prior to his current position he was Operations Superintendent from 1977 till 1984 at the Zorita Nuclear Power Plant.

Now I invite the panelists to make their statements and, as usual, ladies first.

Please, Mrs. Dupuis.

Mrs. Dupuis:

Because of my current involvement, I would like to share with you some impressions from the point of view of nuclear safety.

I felt during the meeting that reliability of pumps appears to be a high priority topic, and that pump performance was a primary consideration for plant availability.

But safety seemed to hover in the background. It rarely surfaced in the foreground. So in order to advance safety in the foreground and in order to stimulate discussion, I will highlight some important safety concerns associated with pumps. First, pumps contribute to the defence-in-depth of plants. The first level of defence-in-depth is the prevention of accidents of primary pumps which in turn contribute to the integrity of primary circuits.
Some pumps, not safety-related such as feedwater pumps, contribute to the prevention of transients that could lead to incidents or accidents. Also, pumps contribute to the mitigation of accidents, hence the second line of defence-in-depth. Pumps included in engineered safety features are usually the best example.

We have seen in some of the presentations that some pumps are dedicated to specific safety functions. In some cases, equipment serves both safety functions as well as some normal operation in the plant. That was my first point, namely, putting safety into a new perspective.

The second point pertains to the stringent design requirements for pumps in nuclear plants. Environmental conditions prompt safety bodies to require that pumps be seismically qualified in addition to loading conditions due to accidents, etc. Regulators require, that the licensees demonstrate the functional capacity of pumps in every condition. Furthermore, operating experience provides a means of assessing the performance of pumps.

Some safety-related pumps, though, perform only a "stand-by" function. So the question is, how to ensure that those pumps will be able to run under accident conditions and during a very long time.

As a third point, safety has to be considered in the context of the entire plant, and pumps therefore are not independent from other components in systems. Pump malfunctions could damage plant components in the case of missiles, for example, that could result from flywheel breakup or loose parts.

Conversely, pumps could be adversely affected during plant operation due to their dependence on systems such as electrical supplies cooling water, and cooling air. They are also not independent from man intervention, especially during maintenance, surveillance and testing.

Another matter to stimulate the discussion is that operating experience appears to highlight the vital importance of pumps, and their contribution to reportable incidents and the attending corrective actions.

I would like now to summarize this introduction by one question:

Do we really expend sufficient efforts to ensure that the reliability of pumps is proportional to their high safety significance?

Dr. Zwingelstein:

Since I work in the maintenance department I will try to comment on the measures and strategy for ensuring and increasing pump reliability. I will subsequently try to establish an equation to evaluate equipment reliability.

Safe and economic operation of nuclear power plants requires a highly developed functional reliability. This high level of the functional reliability demands an equally high level of reliability of components, and control systems. This is true of course for pumps. An important step in developing a valid maintenance plan is understanding the relationship between the reliability of a piece of equipment and its maintenance needs. There are several reasons other than maintenance performance which can result in variations in equipment
reliability. Therefore, reliability of a new component varies between manufacturers and depends on the model design and on the process of fabrication for any manufacturer. We have also variations due to the improper application, improper maintenance or equipment abuse which can strongly influence equipment reliability. Even if maintenance is ultimately responsible for the equipment field reliability, it is useful to identify other factors in order to have a more complete picture of the root causes of failures and a better definition of the options available. So we will try to introduce the so called "intrinsic reliability" concept.

I will define the intrinsic reliability concept under non-operating conditions and given no preventive maintenance at all. An equipment exhibits a so-called "natural value" of its reliability based only on the quality of the design and of the manufacturing process. Intrinsic reliability of an equipment is the value in principle other than its natural reliability. It is worth while to note that an optimal preventive maintenance program, where each action is efficient in countering one type of failure, allows to improve somewhat the intrinsic values. For such a program more preventive maintenance leads to improved reliability. For less optimized or poorly engineered PM-programs, more preventive maintenance can lead to a lower reliability. A specified field reliability range can only be achieved by using equipment with very high intrinsic reliability or by using equipment with intrinsic reliability improved by a required volume of optimized PM.

So traditionally, in the energy industry, high intrinsic reliability has been associated with over-design, achieved by a careful definition of operational stresses placed on the equipment, consideration of every significant degradation mechanism or failure mode that is likely to take place, and selection of design features and manufacturing processes which eliminate these factors or at least minimize them. Qualification testing and early real life experience are also included in this design activity. Providing data is useful to improve the design until its reliability level is achieved.

As a result, equipment with high reliability only needs maintenance to counteract the degradation mechanisms. To summarize the foregoing, I contend that actual field reliability is equal to the intrinsic field reliability plus or minus changes due to maintenance.

Thus the goal of the maintenance plan for each equipment is to reach and maintain a generally implicit target for actual reliabilities. Because of its importance, this equation will take into account concerns of the manufacturer, designer, utility and of course the safety body. Consequently determining the actual reliabilities through failures is not the same practice. Rather, it is necessary to predict it using engineering reviews and analyses of a level of detail generated by qualification and operational testing, inspection, monitoring and diagnostics. When these reviews and data, and of course the actual history of failures, indicate that reliability is too low, and that deterioration is occurring, it is time to manipulate; that means improve or restore the component's reliability, which in turn means changing the preventive maintenance program in order to improve its intrinsic reliability. The optimization of the maintenance program demands a great deal of cooperation and technical information exchange between the utility maintenance staff and the manufacturer at the time of purchase as well as during operation and through any design modifications. In addition, the maintenance organisation collects,
analyzes, and integrates data relative to equipment field reliability, often available through independent organisations.

In conclusion, I pose several questions for the floor; first of all the manufacturer and the designer: How can you assess intrinsic reliability? Secondly for utilities; how are they able to manage and to measure the actual field reliability? Finally, for the safety authorities I would like to know what would be a reasonable and achievable intrinsic reliability of safety-related systems.

Mr. Minners:

Mr. Jack Rosenthal of the NRC, who reviews operating experience in more detail than I do, generally agrees with me in that pumps in American plants are not major contributors to the residual risk of operating these plants; possible exceptions are the turbine-driven pumps and, as my talk demonstrated, possibly the reliability of seals may be a contributor. This could therefore be analogous to experience with other pieces of equipment such as motor-operated valves which are considered to be very prominent plant components whose reliability is inadequate; for this reason we have extensive programmes to try and improve their reliability. However, we have no similar programmes to improve pump reliability. This does not mean though that pumps are unimportant. The safety importance is high and the only reason that they do not show up in the residual risk is because they have good reliability, as has been discussed here several times. The American experience has shown that adequate maintenance is important, in addition to high manufacturing quality, to ensure reliable pump performance.

Mr. Sola:

My point of view is that of an operator and is more related to higher plant availability through the increased reliability of the reactor coolant pumps. In Trillo we have several reliability-related problems with these pumps and my colleague Mr. Burriel explained our technical problems detected during the commissioning phase of the plant, and that we are solving step by step. My first concern is about this type of problem which is long and difficult to solve. We requested the main supplier KVU and the designer of the pump, Andritz, to review the design including a testing program with the model. This program has two characteristics; one is compatible with the commissioning phase and with the operation of the plant and the other one was done in cooperation with the main supplier of the plant and the designer of the pump. The second concern is about testing. During the commissioning phase, an extensive program of testing was undertaken with the objective of identifying all the problems associated with the reactor coolant pumps. This commissioning program included complete disassembling of the pumps and inspection of all internal parts. The third concern was about surveillance during operation. During the past three years we have had three main facets related to surveillance. The first is measurement of operational parameters, e.g. temperature, flow, and pressure of different auxiliary systems, and periodic analyses of these parameters. The second one is predictive maintenance based on surveillance of vibration and periodic analyses of the data of the vibration system. The third one is the preventive maintenance which involves changing of the parts having a limited life like gaskets or O-rings, and inspection of special parts using non-destructive techniques like ultrasonic, eddy current tests, etc. In
preventive maintenance we have special considerations about radiation protection since about 30% of the total doses during a refuelling outage is due to the inspection of pumps. The fourth facet is about external experience supervision. In this area we have a program of analyzing the external experience with three main objectives: the first one is to give more data in support of design change decisions, the second one is to permit analyses of whether a certain problem is a generic one; and the third is to prevent the occurrence of problems. Finally, the fifth facet is about documentation of the quality assurance program. Our experience shows that we need to improve the quality of internal parts of the pumps, as well as the quality of manufacturing. We thus have now a program that covers specification of materials, specification of the manufacturing process and a program of review and testing for future inspections.

Mr. Bernsteiner:

Let’s have a look on future developments in pump design from the viewpoint of a pump manufacturer. Mrs. Dupuis pointed out the concern of safety aspects, and the need to feedback operating experience. Mr. Zwingelstein asked for intrinsic reliability. Mr. Minners said, that pumps are not a major concern in nuclear power plants. Mr. Sola has had some very bad experience with a pump design that was used only in a few nuclear power plants. So what can we do in future designs? There are several ways to address this question. Way number one is to do nothing, or better still, to effect only those changes that are deemed strictly necessary. Hence, you remain within the field of experience, you will not incur any additional safety problems, you need not have any additional expenses, you can reduce your research and development costs, and you can spend your time coping with the stringent and often unfortunately divergent requirements of your customers in countries over the world. An alternative approach is to do something; i.e. to use developments made in your own company or other companies to improve your pumps in mini steps, as shown in several presentations, or to devise ways for coping with increased safety requirements, low dose rates and reduced maintenance. Examples include sealing systems with long service life, back-up seals for station blackout, independent cooling systems, and new impeller-to-shaft connections to cut down assembly time of hot parts, which in turn means lower dose rates. A third alternative is to be innovative. The innovative push of the seventies and early eighties is gone. Are we now ready to re-start innovative activities, and to put a lot of money in prototypes? Are we ready to struggle with the so-called children diseases and are we ready to incur additional risks? There are many design features to be introduced in the nuclear business, but what are the requirements to be introduced tomorrow? A few examples are: Would you like to have improved gap designs to raise the efficiency of multi-stage pumps? Should we offer new medium-lubricated maintenance free bearing systems, so we can save complicated oil supply systems? Can you imagine having a charging pump without any gearbox, without oil supply systems, and without seals? Will you integrate pump monitoring systems with logic event tree analysis in your reactor system? Finally, what are the prospects of new reactor designs, and will there be a market beyond 1995 or the year 2000?

Dr. Kotthoff:

I would like to thank all panelists and open the floor for comments or discussions.
Mr. Rosenthal:

To the pump manufacturer. We anticipate reliable high powered models with a variable speed, and a variable frequency. That has been alluded to in several talks, and has the potential for simple, fine designs etc. Where do you see this going?

Mr. Bernsteiner:

These variable speed drives will help to simplify pump design. Therefore, we are very interested to have these variable speed drives and we also have undertaken some design studies to use these, for instance, in the charging pumps; the result is thus quite a minipump, I would say, compared with the size of the piston pumps used in the United States.

Mr. Killian:

One other difficulty for pump manufactureres to introduce in new technologies is, as Mr. Bernsteiner underlined, the need to design the pump with all requirements asked for. So you have to know what the future requirements are because you cannot be sure of the technology, improvements, and design; an example is the use of variable speed drives.

Mr. Bernsteiner:

I agree with that.

Mr. Kumar:

I have a few questions on the safety issue. Mrs. Dupuis mentioned fire retardant lubricants and fire retardant capabilities in the pump itself. The use of fire retardant lubricants is one way of preventing the fires from starting. Also Mr. Bernsteiner mentioned the seal behaviour under accident conditions, having a standardized back-up seal. I think that is the future way to go, i.e. controlling the accidents by passive measures rather than by active measures.

My last point pertains to condition monitoring both in design and maintenance. Several participants have indicated that active condition monitoring is essential for monitoring individual pump performance, to enable taking corrective actions whenever necessary. Could any panel member comment on this?

Mr. Bernsteiner:

I have one comment concerning condition monitoring systems. Up to now we know that these are only monitoring systems; hence we get some data either on screen or on paper, and this data is examined with a delay varying from one day to one second. Currently, the next step is being developed in addition to that monitoring system. The next step involves the combination of all this information in some form of logic trees to find the root cause of a given problem within the pump or within the system. These logic trees depend always on the pump system; they can be very complicated, and to set them up much experience and data is required. This means that this cannot be done for the first design of main coolant pumps. Once data has accumulated, such a system
can be introduced into the market. The customer can benefit from such a system since it will enable him to detect at a very early stage the possible cause(s) of a failure. This approach can also be used to predict maintenance activities.

Dr. Zwingelstein:

EdF is adopting a similar approach. We were in the process of designing a monitoring system for primary pumps and then we started developing an expert system in order to help plant personnel perform analyses for primary pumps.

Dr. Kotthoff:

I agree that monitoring systems are very helpful and that there should be an increased use of them. However I would like to say a word of caution because I feel that monitoring systems would be most helpful for continuously operating components, but not for stand-by components, in which case the benefits would be much less. The second point is that monitoring systems are very efficient for those expected failure modes for which the monitoring systems were designed for; but any other failure modes may not be detected by the monitoring system. One example is the pump shaft cracks we had. We did have a monitoring system, but we did not expect a pump shaft crack. The first pump shaft cracks were therefore not identified at an early stage by the monitoring system. Following improvements to the monitoring system, one severe crack was detected. I am therefore raising the question of how to cover the area that is not covered by the monitoring systems.

Dr. Zwingelstein:

From EDF's standpoint, I agree that monitoring equipment is necessary but not sufficient; hence it must be supplemented with inspections to monitor non-expected degradation.

Mr. Kumar:

Generally, there are no code requirements for pump internals. We therefore need to have data control on the pump internal requirements; that is definitely a need for the future. Pump shaft cracking is a difficult example, and we have had cracks in the stainless steel components, where the hardness was not controlled. 440 stainless steel is a very typical example, where you have stress corrosion cracking. Failures have also occurred in other components such as screws on the impellers, the journals sometimes, and the shafts. But they are all related to hardness. By controlling the hardness, you can avoid these failures. So in several cases the method of measurement of hardness itself is unreliable, since it involves just one single point measurement which doesn't give how it varies on a big component from one side to the other. I therefore believe that manufacturers should have better control on hardness measurement; hence, instead of giving a one-point measurement, an average of eight points should be made and the spreads should be within specified limits.

Mr. Bernsteiner

Of course, we check hardness, if required by the specifications. But we have also several additional checks which is particularly important for the nuclear components. We are therefore used to doing more than simply what the specifications stipulate.
Mr. Clausner:

From the safety authority's point of view, we feel that this is not a matter of concern since we believe that there is a sufficient amount of operating experience information to enable measurement of the reliability of this equipment. As far as reactor coolant pumps are concerned, the questions seem to be more with regard to the deterioration of stand-by equipment; we think that this is where our safety concern lies. Some examples on the French plants have been found where generic problems were identified during normal surveillance or during preventive maintenance. This is one reason why our problem seems to be how to extrapolate the results of the periodic maintenance program to the reliability of these components. This is also why we consider it necessary to get as much information as possible by other means such as endurance tests, as already mentioned, or maybe commissioning tests.

Mrs. Dupuis:

I would like to know the point of view of utilities and manufacturers about some design options; for example this morning a paper was presented addressing two very different designs. For the 900-MW-plants the same pumps ensure, under normal operating conditions, the chemical and volume control system functions as well as the safety injection function. For the 1300 MW-plants, EdF proposed to use dedicated pumps for each of those functions. I would therefore like to know the point of view of EdF regarding the advantages and drawbacks of both options.

Concerning the question raised by Mr. Clausner, it is important to determine the future requirements pertaining to pumps devoted to safety functions such as safety injection, where their operation must be ensured under accident conditions.

Mr. Kumar:

To address that question, I would like to point out that we have in our specifications quite detailed requirements from the safety point of view. For events such as seismic loading, we brought out floor response spectra, and the pumps have to be qualified for it. But for loss of coolant accidents, the integrity of the pump has to be retained, and the pump should be able to operate at the environmental qualification levels, namely, radiation and 100% humidity. Those are some examples apart from a whole list of safety features incorporated in the design itself.

Dr. Kotthoff:

I would also like to add that pumps are designed in accordance with the specifications of the plant designer, who takes into consideration flow rates for normal conditions as well as abnormal conditions such as design basis accidents. But these are, as we say, conservative covering events. However, an event may occur which exhibits flow rates and conditions that are quite different from those considered by the designer. My question is therefore whether we should design the pumps for all events that can possibly happen.
Mr. Bernsteiner:

Sometimes a pump is designed for different kinds of operations; and sometimes we are quite unhappy with the stringent requirements, imposed on some of our pumps, which are often difficult to meet. From a pump manufacturer's viewpoint, it's often better to have a pump for each requirement, since QH-data differ sometimes so extremely that it's very difficult to meet. We must also check prototypes for a very long time, and even then we cannot be sure whether our tests do represent the actual conditions which will be imposed in the power plant.

Mr. Hada:

I just want to refer to the presentation by Mr. Minners. In one of our PWR-plants one pump which is designed for RHR-purposes was used for cleanup; design of that pump was therefore based on the requirements stipulated by the utility, which were in turn based solely on the RHR operation which requires higher flow and lower head. The designer did not satisfy the requirements for the clean-up operation, where a higher thrust force on the impeller is encountered, which causes higher bending stresses on the shaft. This resulted in fatigue which eventually sheared the pump shaft. This is a very simple example, but I fully agree with your comments that the actual user or the utility should identify the purpose of the pump exactly; but one pump should not be switched easily to operate in modes or for purposes for which it was not originally designed.

Mr. Verry:

Coming back to the problem of a safety injection and charging function, I quite agree with Mr. Rosenthal. In general it's difficult to design the pump to satisfy the two functions. In the 900 MW-French reactors, pump operation was at the limit of which it seemed reasonable; but for the 1300 MW, it was not reasonable. I think that if I come back to the tolerance concept, there is a tolerance machine concept that was presented by Mr. Martin the first day. I think that if we had two very different design points for the same machine, we are at the limit of tolerance for both. It would therefore be a pity because you don't have enough margins on both. Conversely, if you have two separate machines, you can have much more tolerant machines, with margins that could allow you to have slight deviations from the operating conditions stipulated in the design. You would thus minimize the risk of significant divergence between specifications and actual pump utilization.

Mr. Gonzales:

Coming from a utility, I would say that I suffered from pumps for twenty years. My opinion, which contrasts with that of some of the other participants, is applicable to Westinghouse, KWU, and other designs, and is the following. There are two kinds of generic problems; the first one is the inadequate design or the incompatibility of the pumps with the system; this problem arises during the life of the plants because no changes are made to the pumps, following any maintenance or modifications to the system. The second one is the maintenance staff of the utility, who are sometimes not qualified to perform the right maintenance on a given pump design; equally important is the qualification of the manufacturing personnel who supervise pump maintenance. These two constitute actual problems that utilities suffer from.
Mr. Bernsteiner:

Thank you for your question. It should not only be answered by pump manufacturers but also system designers. From my point of view, we know that many system designs are such that the pump, once installed, will not run in the particular duty point for which it was originally designed. That depends of course on the different designers and on the various countries. But there really are safety margins which can cause minor problems, but with which systems should normally be able to cope.

Concerning the second point regarding maintenance personnel in the utilities and their qualifications, and also the vendors' personnel qualifications, I can only cite from our experience in German Nuclear Plants. The utility itself generally would not have the specialists to maintain the pumps in the nuclear island; that is the NSSS and boiler feed pumps, condensate pumps, cooling water pumps; these pumps are maintained by the vendor's staff. That means that in our case, there are about 200 persons doing that work in the 20 German power plants, and consequently responsible for any failures that may subsequently occur.

Mr. Sola:

Mr. Bernsteiner, I have two questions concerning the hardness tests you mentioned during the manufacturing process. One of them pertains to fatigue, and the second one to the residual loadings arising from the manufacturing process. Do manufacturers, do you think, consider these two problems during manufacturing?

Mr. Bernsteiner:

Past experience showed that these problems cannot be ignored. Our raw material requirements, related to hardness and other properties, are very stringent, normally even more stringent than imposed by the specifications. Fatigue and other problems have been investigated to a very high extent, particularly during the last three to five years, and are therefore taken into consideration during the manufacturing process.

Dr. Kotthoff:

This has been quite an interesting panel discussion, and in the absence of any further questions or comments, I would just like to summarize it. I think, the presentations and the contributions during the discussions showed that due to the special situation in the nuclear industry, the requirements imposed on pumps as well as on other pieces of equipment in nuclear power plants are much more demanding than in most other industries. It is hard for pumps to meet these requirements. Sometimes pumps have to be further developed in accordance with substantial evaluations up to complete new designs. Due to this situation there have been problems, and there are still problems with pumps and we have to be aware that problems may continue to exist in future, with pumps. To overcome these problems, significant measures have been taken; these include in-depth experimental and theoretical investigations both to overcome problems observed and improve the design of new equipment, increasing efforts during design and construction to assure high quality and to meet the requirements, increasing efforts in the qualification of pumps including full scale test
facilities in the workshops and increasing the systematic use of the feedback of operating experience. All these methods, I think, are effective and we had some presentations giving an overview of the overall reliability of pumps and these papers demonstrated an increasing reliability of the pumps with time. Nevertheless, I think, there have to be further efforts to solve the problems we still have and to assure that no significant problems will occur in future. I hope that these efforts will be successful. This ends my summary and it remains for me to thank all the speakers for the interesting presentations, to thank the session chairmen for their professional chairmanship and to thank the participants for the fruitful and interesting discussions. Last but not least, I would like to thank Mrs. Laue and Mrs. Kurth for their work in the background to guarantee that no major incidents occurred during this meeting. I wish all of you who cannot participate in our technical meeting, enjoy Cologne and have a good flight back home. Thank you very much, and that closes the meeting.
## ANNEX #1

### List of Participants

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<td>Organizations</td>
<td>G. Ishack (Scientific Secretary)</td>
<td>OECD/NEA</td>
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## ANNEX #2

### List of Papers and Authors

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<td>Ultrasonic In-service inspection of PWR coolant pump bowl welds.</td>
<td>Ph. Dombret</td>
<td>Groupe AIB-Vincotte</td>
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<td>Routine measuring of vibrations as predictive maintenance.</td>
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<td>Germany</td>
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<td>A.N. Kumar</td>
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<td>Canada</td>
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<td>Atomic Energy of Canada Ltd.</td>
<td>Canada</td>
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<td>Reliability of steam turbine-driven standby pumps used for safety-related applications in U.S. light water commercial power generating plants.</td>
<td>J.R. Boardman</td>
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<td>Advantages of wet motor pumps in Nuclear Power Plants based on ABB Atom AB experience.</td>
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<td>Trend of incidents and failures of pumps in Japanese Nuclear Power Plants.</td>
<td>S. Nakamura</td>
<td>MITI )</td>
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<td>M. Hada</td>
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<td>M. Harima</td>
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<td>11</td>
<td>Design and manufacturing of primary sodium pumps for the prototype fast breeder reactor &quot;MONJU&quot;.</td>
<td>Y. Yamagishi, S. Yazawa, S. Nakadaira, Y. Hayashi, J. Kikushima</td>
<td>PNC, Hitachi Ltd.</td>
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<td>12</td>
<td>Construction and qualification experience on the RRA &amp; ISMP pumps of the French nuclear reactors.</td>
<td>C. Mech</td>
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<td>Performances of primary pump motors for nuclear reactors.</td>
<td>E. Lejeune, J.L. Killian</td>
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<td>Turbulant flow analysis of inlet &amp; exit flows of internal pumps installed in internal pump plant vessel.</td>
<td>T. Okamura, T. Takagi, Y. Yoshomoto, H. Utsuno</td>
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<td>Charging pumps in French PWR nuclear power plants.</td>
<td>A. Duhamel, J. Dhote, J. Vauchel</td>
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<td>Incidents attributed to pump problems</td>
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<td>R. Larue</td>
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<td>Nuclear pump design from the past to the future</td>
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