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Survey of Feed Pump Outages

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ABSTRACT

The emergence of power plant availability as a critical parameter for the electric utilities has motivated this investigation on the reliability and failure mechanisms of feed water pumps. A comprehensive industry-wide survey was made which provides ample data to identify the causes of availability loss resulting from failures of boiler feed, nuclear feed and feed water booster pumps. The specific design, operation and maintenance deficiencies and system related problems responsible for most outages involving feed pump failures are summarized. It is concluded that these pumps need more of the high-technology engineering that is currently being applied to many other types of rotating machinery. Also, it is concluded that less emphasis on maximizing pump efficiency at full load and more emphasis on operating performance over the entire load range is needed. Specific recommendations are made herein which, if implemented, will substantially improve feed pump reliability, both in present and future applications.

EPRI PERSPECTIVE

PROJECT DESCRIPTION

This final report, Survey of Feed Pump Outages is one of several surveys being conducted by the Fossil Fuel Power Plants Department to more clearly specify the major generic equipment and/or operating problem areas responsible for utility power plant outages. This survey includes input from 138 utilities throughout the United States, covering 240 generating units with an average size over 600MW, utilizing 1204 large feed pumps.

PROJECT OBJECTIVES

The main objective of this 26 month investigation was to determine the underlying causes of feed water pump failures. Other objectives were to identify design changes to reduce failure frequency, to determine the required instrumentation and shop-witnessed tests for detection of failure related problems before equipment is shipped to the plant site, and to identify those problem areas where current technology developments need to be expanded and utilized.

CONCLUSIONS AND RECOMMENDATIONS

Data analysis resulting from this survey clearly demonstrates the primary feed water system/pump/component problem areas contributing to generating unit unavailability. Recommendations are made to maximize the availability of existing feed pumps and areas are indicated where future research efforts should be directed and existing high-level technologies should be applied to substantially improve feed pump reliability. It is also recommended that users place less emphasis on the initial costs of feed pumps and more emphasis on reliability-design features to encourage manufacturers to accelerate their improvement efforts in critical technology areas. It is felt that implementation of the recommendations included in this report by users and manufacturers will substantially improve the availability of feed pumps.

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Section 1

INTRODUCTION AND SUMMARY

Reliable feed pump operation is essential to steam power plant availability. Nuclear steam generators achieve partial redundancy by sharing full-load duty between two and occasionally three pumps. Some large fossil fired plants operate with single (100% capacity) feed pumps. That plant availability is a direct function of feed pump reliability is quite clear. EPRI, in conjunction with the EEI Prime Movers Committee, has identified feed pumps as a primary area where improved reliability is needed and has sponsored this investigation by Energy Research and Consultants Corporation (ERCO) to provide data on the nature of feed pump failures.

The primary thrust of this document is to report the major causes of feed water pump failures. However, to support the findings summarized herein, technical sections and appendicies relevant to the major failure mechanisms are included.

The main objective of this investigation is to determine the underlying causes of feed water pump failures. Other objectives are to identify design changes to reduce failure frequency, to determine the required instrumentation and shop-witnessed tests for detection of failure related problems before equipment is shipped to the plant site, and to identify those problem areas where current technology developments need to be expanded and utilized. In meeting these objectives, recommendations are developed as a basis for the specification and design of these pumps with the ultimate objective of improving power plant availability.

The strategy taken in this investigation was first, to assemble a large data base on pump failures and related operating problems, second, to organize and analyze this data so that the major failure mechanisms would be readily identified, and third, to correlate these results for the purpose of isolating the design features and/or operating parameters primarily responsible for each of the failure categories.

The data base was assembled from an industry-wide survey, which was conducted as part of this investigation and supplemented with ERCO's recorded experiences on numerous power plant pump failures. The survey was made in the form of a questionnaire which was submitted to 138 electric utilities throughout the U.S. Of these, 96 responded, covering 240 generating units with an average size over 600 MW. The total number of feed pumps in this sample was 1204 of which one-third experienced at least one failure, and in several cases multiple failures were cited for the same pump.

Analysis of reported failures frequently involved direct communication with plant operators and utility engineers. Many unresolved failure cases reported were sufficiently troublesome for the particular utilities to warrant on site inspection of the pumps and comprehensive analyses. In most of these cases, this involvement provided key inputs for resolving the failure causes. This resulted in resolving similar failure problems in other generating stations and also in intercepting potential failure problems in plants presently being designed.

The design of centrifugal pumps has long been considered an art rather than a science. This is because of the complex fluid dynamical phenomena that take place inside centrifugal pumps. It is only recently that it has been feasible to predict and explain various pump hydraulic phenomena using advanced computerized methods of analysis. These methods are still under continuing development and have not yet had an impact at the design level. However, a reliable pump involves more than just hydraulic design. Feed water pumps are high-technology machines which need application of the best available technology in other areas as well. These areas include rotor dynamics, bearings, seals, structural integrity, materials and controls. All of these fields have been greatly advanced in the last two decades as reflected in other high-technology rotating machines, such as turbine-generator sets and jet engines. However, present pump designs frequently do not exhibit application of advancements in the various critical technologies.

The economic benefits of increased plant availability make it essential that users encourage the manufacturers to accelerate their development efforts in the critical technology areas so that the needed improvements in pump reliability are accomplished. The users must therefore place less emphasis on the initial cost of pumps and more emphasis on demonstrated reliability.

Section 2

CONCLUSIONS AND RECOMMENDATIONS

Industry-wide data on feed pump failures clearly demonstrates there is considerable margin for improvement in feed pump reliability. The present high priority given to plant availability therefore makes an upgrading of feed pump reliability highly attractive. It is recommended that specifications place more emphasis on pump design features which effect reliability and that accelerated efforts be made to incorporate the best available technology into feed pump designs. Major areas affecting feed pump reliability are summarized below.

EFFICIENCY AND HYDRAULIC INSTABILITY - Too much emphasis has been placed on maximizing pump efficiency at full load, resulting in unfavorable hydraulic performance (hydraulic instabilities) at part load operation in many applications. Severe hydraulic instability causes high vibration and pressure pulsation levels, resulting in frequent pump component failures (i.e., seal, bearing, shaft, impeller and axial thrust balancing device). Requiring stable head-capacity curves and setting appropriate upper limits on allowable vibration and pressure pulsation levels will help to minimize this problem.

COMPREHENSIVE SHOP-WITNESS TESTS - The costs associated with lost unit availability resulting from feed pump failures make it economically attractive to bear the cost of thorough shop-witnessed tests of new pumps before they are shipped to the plant site. Specifications should require these tests at full speed (for constant-speed applications) over the entire operating flow range. Variable speed applications should require tests over the entire operating speed and flow ranges. All performance parameters reflecting pump reliability (as described in text) should be carefully measured using appropriate instrumentation.

FIELD INSTRUMENTATION - Installation of permanent instrumentation on feed pumps can improve plant availability in three basic ways: First, to intercept potential failures before they occur, thus avoiding costly down time, second, to determine if unscheduled maintenance is needed, and third, to determine if scheduled maintenance can be postponed.

Feed pumps should have at least the following items monitored by indication (I), continuous recording (CR) and alarms (A): Radial vibration at both journal bearings (CR and A), axial position (I and A) and axial vibration of the shaft (A on nuclear only), balance line leak-off flow if disk type (CR and A), lube oil pressure and temperature (CR and A), casing temperature at four opposite points during start-up (I), and pressure difference across the suction strainers (I).

SEALS - The most frequently failed components are seals, mainly the floating-ring and mechanical types. There are very few failures with labyrinth seals, which demonstrates their long recognized high durability. To optimize reliability and avoid continuous maintenance and failure problems, labyrinth seals are recommended, especially for large pumps.

AXIAL BALANCING DEVICES - Failure of disk type balancing devices is frequent while failure of the drum type devices is rare. However, when a balancing disk fails, the damage is usually localized whereas when a balancing drum fails the complete rotor can be destroyed. Balancing drum failures are the results of other pump deficiencies, most commonly, high rotor vibration levels. Balancing disk failures are usually caused by insufficient load carrying capacity coupled with hydraulic instability. Balancing disk geometry must be carefully designed and adequately sized, keeping in mind that as the pump wear-ring clearances open up through normal wear, the thrust load increases considerably.

JOURNAL BEARINGS - It is rare that a pump outage is directly attributed to a journal bearing failure. However, replacement of damaged journal bearings is a frequent occurance and is usually associated with other operating difficulties, primarily hydraulic instability and the accompanying large dynamic forces and high vibration levels. It is felt that journal bearings are indirectly responsible for pump outages by not adequately limiting the vibration levels which result from hydraulic instability. Improved bearing designs with optimized damping properties would improve overall pump reliability by providing more dissipation of the potentially destructive energy in the hydraulic forces, especially at part load operation. Use of tilting-pad bearings is a step in that direction.

CAVITATION DAMAGE AND PUMP NPSH - Pump outage resulting from cavitation damage to internal components is frequent. Many of these outages occur because present standards for determining the pump NPSH requirements are usually misinterpreted or misapplied, frequently on the optimistic side. Larger margins than currently used on available NPSH are required to protect against cavitation damage to pump internal hydraulic components. It is recommended that the presently used standards be revised to insure that the NPSH available at the pump suction is ample to meet the requirements of the feed pumps.

PUMP MINIMUM (RECIRC.) FLOW - Operation of large feed pumps at flows below 25% of pump best efficiency design point may result in severe hydraulic instability, frequent failure of pump components and excessive vibration of pump or feed water piping. If the manufacturer's recommended minimum flow is excessive, for example 45%, this frequently indicates previous low-flow operating difficulties with the design and therefore the failure history of the design should be reviewed to see if a redesign of the hydraulic components is required.

HEAD PER STAGE - Field experience suggests that pump reliability suffers when delivered head per stage exceeds 2200 feet, especially if the available NPSH is marginal. It is recommended either that head per stage be kept below 2200 feet or that a heavy duty impeller construction be required with ample available system NPSH. For single-stage double-suction nuclear feed pumps and boiler feed pump double-suction first stages, staggering of the left and right side impeller vanes is essential to minimize hydraulic forces at the vane passing frequency.

SHAFT BREAKAGE - Failure of feed pump shafts is not infrequent and is usually by fatigue, particularly in cyclic units. Shaft fatigue is the result of dynamic hydraulic forces at part load operation combined with improper design and manufacturing procedures. These failures can be avoided by eliminating undercuts and sharp corners at keyways in shaft designs. Also, proper heat treatment of shafts will help alleviate this problem.

FEED WATER CONTROL - Tripping of generating units as a result of feed water control problems is a frequent occurance in power generating plants. Investigations have shown that an improper combination of pump and flow control valve, producing unstable hydraulic conditions, is a major cause. Solutions to this problem can be aided by investigating the operating history in other power stations applying the same pump-valve combination. The probability is high that the same problem has already been experienced and solved in other plants. To avoid this difficulty, it is recommended that more interfacing between the feed pump design and the feed water control system be required. Specifically, when the design analysis of the feed water control system is performed, measured flow characteristics of the feed pump and control valve should be used rather than the manufacturer's predicted characteristics, which are frequently more stable than the actual characteristics at low flows.

TECHNOLOGY DEVELOPMENT AND FUTURE RESEARCH - The majority of feed pump component failures and feed water system control problems which reduce plant availability result from unfavorable hydraulic performance. More fundamental research is needed to better understand the complex fluid dynamical phenomena which take place when pump hydraulic instability occurs. This is a necessary first step to develop improved hydraulic design procedures. A companion research area which needs more attention is the interaction of hydraulic instability with feed water system control. Interaction of pump and piping hydraulics with system control is active rather than passive, particularly at low flows. Reliable analysis of feed water control systems therefore requires a more sophisticated total systems approach, including the analytical description of the pump, control valve and piping hydraulic properties.

The large number of pump component failures and operating difficulties caused by large dynamic rotor forces and vibrations justifies development and utilization of high-level technology in the area of rotor-bearing dynamics. The development of advanced rotor systems which introduce large amounts of damping into the system would improve reliability by better controlling the large dynamic rotor forces, thus producing a considerable attenuation of vibration levels.

Section 3

PUMP CONFIGURATIONS AND SUMMARY OF FAILURES

Centrifugal pumps used in utility applications can be categorized as horizontal or vertical, single or multi-stage, and low or high head applications. A pump stage can be diffuser or volute type, while the pump casing may be cast or forged. The pump drive can be a constant speed electric motor coupled directly, through a gear box, or through a variable speed hydraulic coupling, or the pump can be driven by a turbine. Figure 1 shows typical large pump services for a fossil fueled station. Of the pump types shown, only boiler feed, nuclear feed and feed water booster pumps were included throughout the survey.

3.1 GENERAL DESCRIPTION OF PUMPS

Boiler feed pumps are always horizontal, multi-stage, with forged outer barrel and with welded-on suction and discharge nozzles. The nozzles can be upward or downward oriented depending on the plant feed water piping design. The impellers can be in line as shown in Figures 2 and 3, or as shown in Figure 4 opposed to keep hydraulic forces produced by the impellers in balance. Of the 1,044 boiler feed pumps surveyed, approximately 76% are of the in-line design. The choice of diffuser or volute lies with the manufacturer's design philosophy, and are equally favored. The decision on electric motor or turbine drive is made by the horsepower (hp) limitation on electric motors, which today is approximately at 14,000 hp.

Nuclear feed pumps are often called reactor feed pumps in a BWR, and steam generator feed pumps in a PWR system. It is usually a single-stage double-suction design similar to a conventional feed water booster pump. A nuclear feed pump with forged casing and diffuser is shown in Figure 5. Figure 6 shows a volute configuration. The casing can be cast or forged, and it may have a diffuser, volute, or a combination of the two. A volute is always a double volute to minimize the static radial force on the impeller caused by off-design flow conditions. A small percentage of the presently operating nuclear feed pumps are multi-stage units as shown in Table III.

Feed water booster pumps are often called boiler feed booster or condensate booster pumps. They are, with almost no exceptions, of the horizontal single-stage double-suction type with cast casing and with a single or double volute outer casing. If a power generating station applies boosters, the first stage of the boiler feed pump is a single suction normal stage, and the condensate pump is a low head application (see Figure 1). In the absence of boosters, the first stage of the feed pump is a suction stage, which can be single-stage single-suction, single-stage double-suction, or a twin-suction geometry. Figure 2 shows a single-suction design and both Figures 3 and 4 show double-suction designs which are also called double-entry or inlet first-stage impeller designs. Double-suction and twin-suction configurations are usual, while single-suction first stage designs are rare if the feed water system does not apply a booster pump. Without booster pumps high head condensate pumps are needed. This increases the pressure levels in the L. P. heaters considerably. Advantages and disadvantages of feed water systems with or without boosters is disputed among designers and users. If NPSH is adequate and if the pumps are properly selected, both concepts give equally reliable service.

3.2 FEED PUMP FRAME SIZES AND SPECIFIC SPEED

Classification of the information obtained from the survey questionnaires first required a rational classification of pump sizes to allow a reasonable condensation of the results. For all three pump types investigated a system was adopted which first reduced the many designs to five basic frame sizes based on a reference pump flow (Q_{REF}) produced at a reference speed of 3,570 RPM. This reference flow is obtained from the actual design flow (Q) and speed (N) by the equation*: $Q_{REF} = 3570 * Q/N$. Similarly, the design head (H) developed per stage is reduced to a reference head (H_{REF}) using the equation*: $H_{REF} = H * (3570/N)^2$. The power absorbed per stage is adjusted to the reference condition by the equation*: $HP_{REF} = HP * (3570/N)^3$.

Comparison of all vendors' pump sizes led to a frame size designation of A to E shown as follows:

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* From basic similarity laws for centrifugal pumps (see References 5, 13 to 15).

FRAME SIZE	REFERENCE PUMP FLOW (GPM)
A	UP TO - 2,200
B	2,000 - 4,400
C	4,000 - 9,000
D	8,000 - 16,000
E	15,000 - UP

BOILER AND NUCLEAR FEED PUMP FRAME SIZES

The five size ranges shown above cover all boiler feed, nuclear feed and feed water booster pump sizes investigated. Figure 19 shows graphically the capacities of these frame sizes for 500 MW and larger applications at the reference speed of 3,570 RPM. The A frame pump sizes (not shown in Figure 19) are used in these power stations as start-up pumps only. The incidence of failures as a function of frame size is shown for boiler feed, nuclear feed and feed water booster pumps in Table IV.

All customary feed pumps have a "low" specific speed (1,000 to 2,000) and therefore the impeller passages are essentially radial. Specific speed is an important parameter for a centrifugal pump stage (not for the whole multi-stage pump, it applies to one stage only) and is defined as follows:

$$N_s = N * Q^{1/2} / H^{3/4}$$

Suction specific speed is equally important and usually designated as S_s. It has the same equation form as the specific speed, but head (H) is replaced by the NPSH required for that pump. It is discussed in more detail in Section 4.2.

3.3 DISCUSSION OF FAILURE MODES

The survey data shows that certain boiler feed pump components consistently fail more frequently than others. This finding made it imperative to determine if a particular component failure was its own cause or the result of other factors. To give an example, a large number of failures involved simultaneous failures of floating ring type seals and axial groove journal bearings. Many also showed simultaneous balance disk failures. The question to explain is: Is the bearing, seal, or the balance disk responsible for the failure, or is another component, or perhaps lagging technology, or operating procedure the one that initiates the problem? Perhaps the fact that the first critical speed of these multi-stage boiler feed pumps is below 50% of the maximum operating speed makes the shaft overly receptive to any subsynchronous hydraulic excitation force. Those that contributed the most to power station outages as well as the general technical background of each is given in Section 4 of this report. References are given in the Appendix if one requires more detail. Although the title of Reference 5 refers to nuclear primary coolant pumps used in the primary loop of a PWR, it contains detailed information and calculation methods for bearings and seals applied in any rotating equipment.

The survey information obtained from participating utilities was organized from various points of view, such as pump types, frame sizes, vendors, applications, components and technologies. Table V indicates the ten major failure causes, each of these discussed in Section 4 in more detail. Related technologies are also discussed in Section 4. The relatively low failure rate attributed to unstable Head-Capacity characteristics as shown in Table V is misleading as this phenomenon may not be obvious in many cases. Probably many were reported as "vibration" which is one of the associated symptoms of pump hydraulic instability.

Only a limited number of truly cyclic plants contributed to the failure records, since there are only a small number of large fossil-fired generating stations today that shut down regularly every day. Nevertheless, with the large number of nuclear units coming on line presently and in the near future, the necessity to build more and larger cyclic units is unavoidable since nuclear units are more economical for base load performance. Problems with pumps, as well as with other equipment, are not only more severe with cyclic units but are also different in nature. There is insufficient data presently available to assess these problems. However, it is likely that all difficulties associated with part load operation, shut-down and cold start-up are more troublesome.

3.4 BOILER FEED PUMP FAILURES

Of the 138 electric utilities contacted, 96 responded with information on boiler feed, nuclear feed and feed water booster pumps from 240 power generating stations with an average size somewhat over 600 MW as shown in Tables I and XIV. Information on other pump types from 40 generating units were also reported, however they are not part of the discussion of this report. They are presented in Table I for completeness only. Of the responding utilities 28 also reported estimated costs associated with the reported outages from 69 power generating units. Discussion on cost of outages is given in Section 3.6 of this report.

Table I summarizes the survey work on all pump types. It shows that of the 1,044 boiler feed pumps surveyed, 362 pumps were exposed to at least one failure. This is a very high number and means that one out of every three feed pumps in service suffers some damage. The total number of boiler feed pump failures reported from large operating fossil units is 763, i.e. the average number of failures of those damaged is 2.1 per pump.

The "A"-Frame size boiler feed pumps are also applied as auxiliary nuclear feed pumps for PWR service, and as special purpose nuclear pumps. The survey work was not extended to that portion of the A-Frame feed pumps, hence are excluded from the statistical figures. Boiler feed pumps used in the petrochemical industry, as well as foreign utility feed pump applications are also excluded. Statistics on component failures are presented in Tables VII to XIII.

3.5 NUCLEAR FEED AND F. W. BOOSTER PUMP FAILURES

The information from questionnaires relevant to nuclear feed pumps are summarized in Tables I, III, IV and V, while for feed water boosters information is given in Tables I, IV and V. Statistics on component failures are given in Tables VII to XIII. The order of failure causes are somewhat similar for these two pump types, but are markedly different than the multi-stage boiler feed pumps, except for first stage cavitation, bearings and seals. Balance disks are not used in single-stage double-suction designs since the two sides of the impeller, acting in opposition, tend to cancel thrust loads, leaving a residual thrust load sufficiently small to be carried by a conventional thrust bearing. All boiler feed pumps operate above the first critical speed of the rotor over their entire load range. Nuclear feed pumps and booster pumps are usually designed to operate below the first critical speed.

Table I also summarizes nuclear feed pump failures. Presently 58 large (over 430 MW) nuclear units are in operation in the U.S., specifically 38 PWR and 20 BWR. Of the 160 pumps surveyed, 61 had at least one failure. The total number of failures reported from 30 operating nuclear units is 133, hence an average of 2.2 failures per failed pump. This ratio is very similar to that of boiler feed pumps. Most of the problems occurred with 22 high speed motor driven nuclear feed pumps.

Several more units are undergoing hot functional testing, showing that the number of operating large nuclear units is continuously growing. The total number of nuclear feed pumps in continuous nuclear plant operation is 135 as shown in Table III and two additional in fossil units as boosters. The number of motor and turbine driven pumps is about the same. Most motor driven high speed pumps are in the smaller units built in the early period of nuclear power development. All nuclear units above 900 MW apply turbine drive, except North Anna, which has motor driven feed pumps. If a third pump is added as standby, such as at Zion 1 and 2, it may be motor driven. An additional 25 nuclear feed pumps, all large turbine driven were surveyed while undergoing factory or hot functional testing, totaling 160.

Only three PWR units have 3 feed pumps, while 14 out of 20 presently operating BWR units apply 3 feed pumps. The others operate with 2 half capacity pumps. One BWR (Fitz Patrick) and two 800 MW fossil units (Tradinghouse Creek 2 and De Cordova) apply nuclear feed pumps as their feed water boosters. This puts the total number of operating nuclear feed pumps to 137.

The number of feed water booster pumps in service is not comprehensively tabulated because large numbers are used as booster pumps, petrochemical process and water supply pumps, only a small portion of which were reported. However, there is a large resemblance between boosters and current high speed nuclear feed pumps with similar failure symptoms. Independently from this project the petrochemical industry reported major problems with these pumps, verifying our findings and thus justifying their inclusion in our study with nuclear feed pumps. Minimum flow operational problems, shaft low-frequency axial vibration at low flows, impeller eye cavitation and bearing and seal failures are frequent with booster pumps. The total number of booster pumps surveyed was 123 in 40 power station units, reporting 155 failures as shown in Table I.

An additional 254 primary coolant, condensate, circulating and service water pump failures were reported from 40 power station units, involving 223 pumps. These pump types are outside of the scope of this investigation and are listed in Table I for completeness only.

3.6 FEED PUMP OUTAGE COST

Available EEI data (Table XV) confirms that boiler feed pumps (and their drives to a lesser degree) are one of the most costly sources of unit outage, with a partial forced outage five times as frequent as full-forced outages. This reflects the practice of providing multiple feed pumps for each unit, although large units with single 100% feed pumps reported comparatively good availability.

Twenty-eight utilities supplied feed-pump-caused outage cost data from 69 power generating units which represents 28.7% of the survey sample. Other pump types are excluded from our outage cost analysis. These costs are influenced by factors such as stand-by pump capacity, number of pumps and available spare parts. Pump outage times tend to fit into two categories, (1) with local availability of spare parts a repair may be made in 1-2 days, but occasionally (2) major damage requires factory repair that takes several weeks to complete. Average pump outage times were about 3 days, and the average size of the 69 power generating units represented in this cost survey was 617 MW. The average size of all reported units on feed pumps was also somewhat above 600 MW. Loss of 50% capacity on a base load unit of this size would incur a loss of approximately $50,000 per day. The failure rate in the sample examined was a little over 900 pumps in three years, or 300 per year. The annual cost of reported pump failures may therefore be computed as 300 (35,000 + 150,000) = 55,500,000 dollars, which is certainly a conservative figure. Several cases were experienced where individual generating units have suffered loss of generation equivalent to or more than 4 million dollars in a year. These cases, being isolated special occurances were not entered into our statistical study of outage cost, making our study conservative.

3.7 ACCEPTABLE LIFE EXPECTANCY OF PUMPS

The subject of how long a pump, or its various components should last, comes up frequently with utilities and especially between users and manufacturers. Searching for failures causes during this survey work also furnished information on reliable pumps. This information is summarized below and is useful for purposes of comparison with life data from pumps experiencing failures. A maximum overall life time of 35 years was selected. This number is only a symbol indicating no need for repair or repurchase during the life-time of the power station whether it is 35 years or more.

		LIFE EXPECTANCY (YEARS)
Pump overall life time		35
External components (foundation, piping)		35
Hydraulic components:	Diff./volute, case	35
Impellers	Not less than	10
" with exchangeable wear-rings		35
Wear-rings (stationary)	Min.	3
Rotor components:	Shaft	35
Balance device	Not less than	6
Seals: Labyrinth		10
" : Other (floating, etc.)		6
Journal bearings	Not less than	10
Thrust bearings	Not less than	10
Coupling: Fluid	Not less than	15
" Dry	Not less than	35

The above figures do not apply to cyclic units. However, if in a base load unit the above components do not perform as listed above, the pump design or perhaps the operating mode should be examined in order to determine the cause of the deficiency.

Section 4

HYDRAULIC DESIGN AND RELATED FAILURE MODES

The basic hydraulic designs of boiler feed, nuclear feed and feed water booster pumps are similar since the specific speed ranges are approximately the same. Specific speed (previously defined in Section 3.2) is a very important parameter and closely defines the impeller passage shape as shown in Figure 18. It is the speed in RPM at which one pump stage with suitable diameter, would need to run in order to deliver one GPM fluid and produce one Ft head. Specific speed (N_s) is correctly determined by using the head produced per stage at the best efficiency point (BEP). It is incorrect if the head per stage and flow are taken at other flow conditions than at the BEP. According to the standard U.S. definition of specific speed, the pump revolution should be in RPM, the flow in GPM and the head in FT.

The actual velocity distributions existing within a centrifugal pump are extremely complex, particularly at off-BEP operation. However, for practical purposes, a simplified two-dimensional analysis serves to illustrate the basic hydraulic concepts and, in fact, has served as the basis of hydraulic design for virtually all centrifugal pumps ever built. A typical centrifugal stage cross section with a volute is shown in Figure 7 along with standard nomenclature. While booster pumps employ volutes exclusively, the majority of feed pumps are designed with diffusers.

The design methods used today are still graphical techniques that depend on the use of many experimentally determined correction factors. Many of these factors are strongly interdependent. As an example, Figure 33 shows interdependence between impeller blade discharge angle and impeller width, and their influence on the required impeller diameter for a fixed inlet flow pre-rotation. Discussion of these parameters is beyond the scope of this document and the reader is referred to References (5, 8, 13 and 14) for specific details. Only geometries that are responsible for high failure rates in utility pumps are given detail discussion in this report. For example, gap "A" in Figure 13 controls the severity of pressure pulsations behind the impeller which give rise to high dynamic forces (both radial and axial) with distinct frequencies. Similarly, gap "B" controls the strength of hydraulic shock-created amplitudes at blade passing frequencies as shown in Figure 11, and discussed in Section 4.4.

4.1 PUMP HYDRAULIC INSTABILITY AND EFFICIENCY

4.1.1 INSTABILITY - A properly designed pump stage operating at constant speed should produce a steadily rising head as the flow decreases, at least down to recirculation flow. However, if the head-flow curve has a "kink" in it, is flat in the middle flow portion, or is an "S" shape as shown in Figure 8, it is an unstable head-capacity curve which is symptomatic of hydraulic instability. The lowest permissible limit of flow (minimum, or recirculation flow) is determined by the vibration characteristics of the pump. If the head-flow curve is unstable, parallel operation, or even single pump operation in the unstable region is difficult, if not impossible. In the example given (Figure 8), the head-flow curve is unstable below 68% flow, resulting in various pump or feed water system malfunctions. The problem originates with the pump impeller hydraulic design, i.e., the number of impeller blades may be too high, blade exit angle may be too high, or the impeller/diffuser geometry may be mismatched. For reliable operation, the pump minimum flow should be higher than the onset of hydraulic instability, which is caused by various flow disturbances, secondary flows or stall in the pump hydraulic channels. Flow phenomena which are responsible for a large number of failures are shown in Figures 9 and 10. The obvious symptoms of hydraulic instability may be pump and/or piping vibration, control valve malfunctioning, pipe support and foundation vibration related problems, or pump internal component destruction. The selection of an oversize pump, while apparently conservative at full flow, will reduce stability margins at the lower delivery rates.

Figure 17 shows minimum flow standards for various pump types and applications. Due to keen competition in the pump market, all the major manufacturers' prices and pump efficiencies fall within narrow ranges. In many instances, this has prompted manufacturer application/sales engineers to offer an optimistically low recirculation flow, sometimes as low as 10% of rated capacity. The motivation is to reduce the size and cost of the minimum flow line and its control valve. This seems attractive initially, especially to the A/E. However, considering the recurring pump damage and availability loss that frequently results from severe hydraulic instability at too low a recirculation flow, the proper approach is quite clear. The design margin band shown in Figure 17 represents a regime in which the experience and assumptions of the pump designer come into play. Safe minimum flow for large nuclear feed or boiler feed pumps is usually not less than 25% of full design flow, whereas quotes as high as 45 or 50% indicate that the pump hydraulic components need design improvements as shown in Figures 23 and 25.

If a pump is already purchased but not yet delivered, a well planned shop test with test points at closely-spaced flow increments will detect this discrepancy. If the head is higher (or lower) at the midcapacities than at neighboring points, retesting of this flow region is essential to verify repeatability because of the oscillatory nature of pump delivery in the unstable region. If the project is in the proposal stage, properly written and executed specifications may take care of this problem by requiring the proper shop-witness tests as part of the purchase agreement.

Figure 27 shows the behavior of the balancing disk leak-off flow in the case of a hydraulically unstable turbine-driven feed pump impeller design. When instability sets in (always at some part load condition which is the characteristic of that particular hydraulic design) the hydrodynamic forces disrupt the equilibrium of the rotor also in the axial direction. The balance disk, which relies on a small clearance to properly function, is a sensitive device. It reacts to all disturbances of the rotor, including dynamic unbalance, misalignment, blade passing frequencies and also sub-synchronous rotor motion. A balance drum (piston), on the other hand, is a "soft" device in the axial direction and therefore does not offer any clues during trouble-shooting, regardless of the degree of hydraulic instability within the pump.

Unstable head-curve-produced outage is shown in Table V as a relatively low number (92). However, because in many cases its presence is not recognized, other associated component failures are mistakenly blamed as the cause of the problem. If carefully evaluated, this failure category would probably show a much higher number. Of the 151 cases reported, 92 were boiler feed pumps.

4.1.2 <u>EFFICIENCY</u> - The problem of low-flow induced hydraulic instability is worsened when, as is frequently the case, heavy emphasis is placed on obtaining the maximum possible pump efficiency at design flow (BEP). Overemphasis on full-flow efficiency prompts the manufacturers to bias their hydraulic design parameters to the full-flow condition, resulting in a degradation of hydraulic performance (hydraulic instability) at low-flow conditions. The present importance of unit availability therefore means that less emphasis on full-flow performance and more emphasis on reliability and low-flow performance is needed.

The pursuit of high feed pump efficiency is a controversial subject because it is difficult to measure in the field. Also, shop-witness test measurements for efficiency can easily be inflated by using various interpretation methods and extrapolation formulas permitted by the present guide lines of the Hydraulic Institute Standards, such as extrapolation to higher speeds, temperatures and sizes. During the past decade the subject of efficiency has been emphasized, especially by the A/E-s. In many cases the emphasis has been put on high efficiency at pump design capacity (BEP), while reliability of the equipment at part load operation was not emphasized. As a result, many pumps with high full-flow efficiencies have failed at part-load flows. Also, conservative reliability-minded manufacturers have often lost bids because they were lower on "apparent" efficiency at full load, however, had lower failure rates at reduced capacities. This may have been a justified course at one time when ample "spinning reserve" was available, but not at present with the heavy emphasis on unit availability and reliability. It is strongly recommended not to consider bids that offer unusually high efficiency at BEP without proof of validity of that performance figure or the method of testing, but more important, not without proof of hydraulic stability at part load capacities.

Figure 23 shows a case where prior to redesign by ERCO of certain pump hydraulic components, severe instability hindered operation below half the pump design flow. After introducing corrected components, pump efficiency improved at part load flows, and even did not suffer at full design capacity. Figure 18 is a fair guide to what pump efficiencies may be achieved as a function of pump size, flow and pump stage specific speed. Figure 12 shows an example of how an improper selection of impeller discharge geometry can reduce the useable operating range of a pump.

4.2 CAVITATION, NPSH AND SUCTION SPECIFIC SPEED

Failures caused by cavitation erosion damage are among the highest outage producing problems, because cavitation frequently causes severe pump internal damage, requiring new internal components or lengthy factory repair. Cavitation is caused by the production of very low local pressures adjacent to flow boundaries, such that vapor filled pockets form and then collapse violently as they are transported into higher pressure regions. Damage is most likely to occur in the inlet of the stage, but may be carried through the impeller causing erosion of the impeller exit or the diffuser/volute inlet. Cavitation can be caused by:

1. Inadequate NPSH of the feed water system (i.e. not enough pressure at the pump suction).
2. Flow recirculation at the impeller eye while operating in the off-design flow regimes.
3. Incorrect hydraulic design of the first stage impeller (incorrect blade inlet angle).
4. Localized high velocities caused by sharp corners and other flow disturbances such as misplaced inlet guide vanes.
5. Vortex formation due to obstacles in the flow path, sharp elbows in the suction piping, incorrect pump inlet geometry and blunt inlet guide vanes.
6. High frequency machine vibration can displace water particles perpendicular to a solid surface creating vapor pockets, hence creating cavitation (least important type with centrifugal pumps).

Operation of pumps at off-design conditions for extended periods of time can cause cavitation damage independent of available NPSH, due to high incidence angle caused stall and secondary flows like eye recirculation as shown in Figures 9 and 10. In a multi-stage boiler feed pump, impeller cavitation damage usually occurs in the first stage, but it can occur at other locations where flow conditions satisfy the above requirements. It is important to distinguish between first-stage impeller cavitation and pump internal cavitation, in that the latter is not related to pump NPSH. Feed pumps producing high head/stage are more receptive to cavitation damage because of the higher energy-input densities to the fluid. Velocities and dynamic forces are high enough to accelerate cavitation and fatigue damages of pump internal components. Further discussion is presented on this subject in Section 4.3 of this report.

The definition of specific speed (N_s) was given before and it determines the general shape of the impeller as shown in Figure 18. Suction specific speed has a similar form, $S_s = N \times Q^{1/2}/(\mathrm{NPSH})^{3/4}$, and it expresses the suction capability of a pump stage. The NPSH required for an impeller, not the system available NPSH, is substituted into the above formula. As with specific speed (N_s), the suction specific speed (S_s) is an applicable basis for comparison only at the design point. Calculated values of both N_s and S_s at off-design conditions are not as useful as indicators of hydraulic characteristics. For the calculation of S_s, the flow entering the impeller eye (Q) is used. Hence, for double suction impellers, only half the total flow should be used to be consistent with single-stage impeller eye properties. To determine N_s, the total flow at the impeller discharge is used. For feed water applications, such as feed, boosters, condensate and heater-drain pumps, a properly designed normal stage (i.e., other than first stage) has an S_s value of of about 8000. A suction impeller (i.e., first stage) has higher values, typically 11000, possibly as high as 13000. However, the higher the S_s value, the lower the stage efficiency, and more important, the stronger the tendency for hydraulic instabilities at low flow operation. That is, the usable operating range of the pump becomes narrower as S_s gets higher. Also, for high S_s applications, a larger than normal impeller eye must be employed. This produces more sensitivity to flow recirculation at the impeller eye, resulting in accelerated cavitation damage potential at flows other than the best-efficiency operating point.

It cannot be over emphasized that the available NPSH at the pump inlet must be substantially higher than the so called required NPSH (see Figure 32). The required NPSH is normally based on a 1 to 3% head reduction at constant flow, experimentally determined by the manufacturer. Most manufacturers use the less conservative 3% value to establish the required NPSH. At a 3% head reduction, the impeller is in a fully cavitating state. Operation in that state for extended periods of time results in accelerated cavitation erosion damage, regardless of the impeller material. The 3% drop in head criteria should be used only as an approximate guide to establish the NPSH vs flow performance curve. It is more important to establish for normal operation the required NPSH margin, over the minimum required NPSH, to effectively prevent cavitation damage. To accomplish this, instrumentation and testing must be refined to clearly indicate a smaller drop in performance, at least as small as a 1% head reduction. However, the safest way to avoid cavitation damage is to use a 0% head reduction to determine required NPSH where no cavitation exists (onset of cavitation).

To calculate NPSH with 0% head reduction, the following experimentally proven formula is suggested (see Figure 32):

Tested with 3% head reduction .... = $NPSH_3$
Tested with 1% head reduction .... = $NPSH_1 = 1.3 \times NPSH_3$
Onset of cavitation (0% head break-down) . = $NPSH_0 = 1.5 \times NPSH_3$

The feed water system and the feed pump will be free of cavitation damage if the system available NPSH is well above the $NPSH_0$ value as shown in Figure 32 (correct system available NPSH).

If cavitation damage occurs at design capacity due to insufficient NPSH, usually cavitation damage can be seen on either or both sides of the impeller blade inlet portion. The damage starts at the leading edge of the vane and may cover a large area. Another type of damage can be observed on the exposed side of the vane located in the corner where the blade joins the impeller hub. This type of damage indicates a mismatch between approach flow and impeller inlet angles which can be caused by extended operation of the pump in the low flow regime, even if NPSH is adequate to prevent cavitation. If severe impeller erosion appears somewhat downstream from the vane inlet edge at the periphery of the impeller eye, the damage may be caused by inlet flow recirculation (see Figure 9). The impeller is then operated in the off-design regime or the impeller eye is too large, causing development of flow recirculation at the impeller eye. If the damage starts from the vane inlet and is on the non-exposed side of the vane then the pump is undersized for the application, i.e., operated at substantially larger than best efficiency flow (BEP).

4.2.1 FIRST STAGE CAVITATION - First stage cavitation can be avoided initially by providing sufficient NPSH in the system, by proper hydraulic design and testing of the first stage impeller before the pump is put into service, by proper selection of pump minimum-flow requirements and by assuring that the pump will not operate in the low flow or the run-out regime for extended time periods. If cavitation occurs during operation, a newly designed impeller may eliminate further erosion damage, but in some cases expensive system changes are required. These include modifications to or installation of booster pumps. Modifications of the feed water piping is often tried, but rarely successful in resolving the problem.

4.2.2 INTERNAL CAVITATION - Internal cavitation includes pump inter-stage cavitation, discharge nozzle and discharge area cavitation erosion problems. Interstage erosion is usually caused by inadequate sealing between interstage components, giving rise to unusually high localized fluid velocities, i.e., "wire drawing" erosion. Discharge nozzle or discharge head damage can be caused by uneven flow distribution resulting in high localized velocities, or vortex formation, possibly combined with inappropriate component material. This is usually a "localized" problem associated with a particular manufacturer's product or a particular application of that product. Critical examination of the flow path, flow velocities and materials of design is required to eliminate these problems in the field.

Of the 271 cases in which cavitation damage was reported 192 suffered damage in the first stage. The majority of these pumps are operating without a booster pump. As expected, booster pumps and condensate pumps were found to suffer extensively from cavitation damage. A total number of 64 booster pumps and 46 of other pump types, mostly condensate pumps, were reported with first stage cavitation damage as shown in Tables V and X.

4.3 LIMITATION ON HEAD PER STAGE

The head produced by an impeller and the horsepower input are other important parameters. A limited number of presently operating boiler feed pumps produce more than 2200 FT head per impeller. The total number of pumps sold in the U.S. with head above this limit is 54. Of these, 34 are in operation, with 27 reporting various failures or operating difficulties. Two of the 34 installations reported trouble free operation, while three abstained from commenting. The major difficulties reported were continuous vibration problems (24 cases) and first stage cavitation damage (18 cases). Four of these 18 employed boosters ahead of the feed pumps. The total number of first stage cavitation cases reported is 17.6% overall, while the percentage among high head feed pumps is 56%.

4.4 PUMP INTERNAL CLEARANCES

The critical clearances in a high speed feed pump are those between rotating and stationary parts where high pressure differentials exist (wear-rings of impellers, balancing device cylindrical surfaces, and seal surfaces), and the gap between impeller periphery and diffuser vanes or volute tongues. All these clearances effect pump efficiency as well as pump reliability. The closer the clearances, the higher the efficiency, but in most components, the lower the reliability, i.e., seizure and internal breakage is more probable.

The commonly used close clearance internal dimensions, which have evolved over years of experience, are nearly the same for all manufacturers and are suitable for reliable operation. If unreasonably high pump efficiencies are specified or demanded, the manufacturer is inclined to reduce these internal clearances below the commonly used values. Such a reduction of these clearances improves hydraulic efficiency because of the resulting reduction in inter-stage leakage, balancing device leak-off flow and seal flow. However, this efficiency improvement exists only during factory acceptance test and for a short period of time in the field. The clearance surfaces wear-in to approximately the commonly used values and the artificially produced higher efficiency vanishes. However, in the process the reliability of the pump is jeopardized by an increased potential for rotor seizure and rubbing induced subsynchronous rotor vibration, either of which can result in destruction of the rotor and unexpected outage.

Reduction of normal radial gap (gap "B" shown in Figures 9 and 13) between impeller and diffuser/volute improves efficiency to some degree. As an impeller vane passes by a stationary blade (diffuser tip, or volute tongue) a hydraulic shock occurs that can be observed in the liquid, on the structure, or can be noticed on rotor vibration measured at the bearings or any part of the shaft. The influence of the radial gap on pressure pulsation at blade passing frequency (see Figure 28) and rotor deflection caused radial forces are shown in Figure 11. Numerical values are not given on the vertical scales, since they are also functions of other design parameters. The radial gap is given as a percentage of the impeller diameter. If the gap is too small, say 1%, the phenomenon can be self destructing, since the rotor exciting forces increase exponentially as shown in Figure 11. Rubbing at wear surfaces may also introduce sub-synchronous vibration amplitudes that can rapidly destroy the rotor. If the impeller and stationary components are structurally marginal, the result can be disintegration of these elements. If these structures are strong, the result may be complete destruction of the whole rotating element. If such failure occurs, the radial gap is to be examined and if found too small, it is to be opened up to normal dimensions. Generally accepted dimensions are:

Diffuser type, Minimum gap 3%

Volute type, Minimum gap 6%

Figure 12 shows that the radial gap also influences stability of the pump head curve at part load operation, as well as efficiency at design point. If the gap is too large, the useful operating range of the pump may be effected in that the minimum flow may have to be increased. It should be emphasized that while gap "B" controls the strength of hydraulic shocks created amplitudes at vane passing frequencies, gap "A" controls the severity of pressure pulsation behing the impeller hub and shroud giving rise to high dynamic forces with distinct frequencies. The head stability at reduced flows actually is controlled by both gaps, and in case of a failure or malfunctioning of a pump, both gaps should be examined. The impeller to diffuser/volute channel width as shown in Figure 13 is another important clearance. In general for a diffuser pump the accepted criteria is that the ratio of b_3/b_2 is not to be less than 1.15 and not more than 1.3. Volute type pumps need a somewhat larger dimension. Thorough details are given on this subject in References (5, 11, 13 and 14).

Many of the 184 impeller breakage caused failures reported during the survey were investigated. Usual symptoms were increasing pump vibration levels, or rapid wear of the cylindrical surfaces finally resulting in impeller casting damage. Increasing the impeller to diffuser/volute radial gap, increasing the wear surface clearances (if rubbing present), and/or increasing the minimum flow offered immediate relief to the problem. Other contributing causes were found: Overly flexible shaft, unstable head curve, cavitation on the non-exposed side of the impeller vanes, defective castings and unstable bearings. Each of these individually or combined may have been responsible for the above mentioned failures. Reduced impeller radial gap, combined with poor casting quality and lower than normal (25% of BEP) recirculation flow were identified as the most frequent causes of impeller and diffuser/volute tip breakages. Identification of this problem usually is straight forward, since it is accompanied by high vibration amplitudes at vane-passing frequency as shown in Figure 28. If rubbing is also present, a vibration component below operating speed may also be present, which normally represents the first lateral critical speed of the rotor assembly.

4.5 PUMP COMPONENT DESIGN

A review of pump components subject to frequent failures or malfunctioning is given in this section. Impeller, diffuser and volute design details, although they are extremely important, are omitted here since they are discussed in adequate detail in References (5, 13 and 14).

4.5.1 AXIAL BALANCING DEVICE - Axial forces in a boiler feed pump are in the minority of cases held in equilibrium by opposing equal number of impellers with a thrust bearing to assist the pump during start up and to take up the residual unbalance force caused by casting tolerances or minor dimensional differences (249 pumps in operation; 22% of the total). If the impellers all face the same way, either a balance drum or a balance disk is used to take the high axial thrust as shown in Figure 14. When the impellers are opposed, although the forces are in equilibrium, that design still utilizes a balance drum for safety, since at part load operation large unpredictable hydrodynamic forces are produced that could damage the thrust bearing.

A balance drum is basically a rotating piston which has the characteristics of an ineffective water-lubricated radial bearing, so it influences the dynamic behavior of the rotor (Figure 14-a). The balance disk (Figures 14-b and 14-c), on the other hand, is basically a water-lubricated hydrostatic thrust bearing. The small gap "e" controls the pressure and consequently the thrust in cavity "A". If the gap "e" becomes too small or closes entirely, the faces will touch and destruction of the mating parts results. Introduction of a small taper between faces is an effective design improvement successfully recommended by ERCO in numerous plants (see Reference 3 for more detail). It is the relative taper angle "alpha", and not the orientation of the faces, that is important. Figure 15 shows the parallel and the tapered face designs. Also shown is a force balance diagram, clearly indicating the superior behavior of the tapered disk design. The disk force counter balances the forces produced by the impellers, transmitted to the disk through the shaft. This force can easily have a magnitude over 100,000 lbs and is responsible for internal damage in many pumps. Figure 15 shows the force T_o, which is the force when the disk is closed. It is vital that the forces produced by the impellers can never be higher than T_o, otherwise failure results. Many failures are due to the fact that the disk design load capacity is marginal when the pump is new. As the wear-ring surfaces wear with normal use, the hydraulic forces on the impellers grow, resulting eventually in a higher force than the marginal thrust carrying capability of the disk. Figure 15 shows the basic principal difference between parallel and tapered face designs, the tapered design having only advantages over the parallel. Note that when the balancing disk is approaching closed position, the tapered face design is able to take much higher thrust loads than the parallel face design, hence it is much more reliable, particularly under large transient loads that accompany severe hydraulic instability. It can be seen in Figure 27 that a good disk design is especially important during unstable pump or system operation. It also shows that pressure pulsation and flow measurements in the balance disk leak-off flow line can be used to detect pump hydraulic behavior. It is recommended to monitor L.O. flows at all times.

Complete disintegration of the rotating balance disk reported from one large generating station called our attention to a typical failure mode. Customary material for that component is 416 SS type, where the material specification clearly states not to heat-treat above 42 Rockwell C hardness, since the possibility of surface cracking is high. When the failed component was tested for hardness, it was found to be over 50 Rockwell C. Other unused disks were tested and found to be over 42 Rockwell C hardness. Surface cracks were found severe enough to assure future failures. We found this discrepancy in at least three manufacturers' products, hence it is not a localized problem. It is a good practice to test for hardness everytime a new disk is put in service in order to intercept this type of failure. The best practice, however, is to heat-treat the stationary part to higher hardness since that part is not subject to centrifugal forces as is the rotating disk.

Of the designs surveyed, 533 feed pumps were equipped with balance disks, and 310 failures were reported. In 511 pumps equipped with balance drum there were 27 reported failures. Table VIII gives detailed failure data categorized by pump sizes. Though these figures show that the drum design is less sensitive to rotor behavior, the balance disk is capable of providing equally good service if correctly designed and adequately sized.

4.5.2 SHAFT SEALS - Failure of shaft seals is the highest among the reported failures as indicated in Table V. Figure 16 shows the three seal designs used for boiler and nuclear feed water pump services with single injection. Double injection is used for high temperature applications to avoid flashing in the seal area. Injection water can be regulated by temperature or pressure control. Low speed booster pumps may employ packing, however high speed applications (3600 RPM or above) are exempt from that seal application. Table VII shows the failures by seal types, clearly indicating that the majority of failures occurred with seal designs other than labyrinth types. Labyrinth seals are the least demanding "work horses" of utility pumps.

Of the multi-stage pumps surveyed, over 300 were equipped with Labyrinth seals, and reported only 32 failures. The majority of these failures were pump seizures caused by dirt in the feed water. Most seizures occurred during turning gear operation or during coast down at low speeds where the rotor momentum was not enough to grind up the dirt particles. Galling, then rotor seizure resulted. In a limited number of cases rubbing due to overly close clearances or high shaft flexibility introduced partial frequency shaft whip giving the appearance of oil-whip. In some cases the bearings were changed to tilting-pad types with various results. In four occasions the half frequency whirl recurred in spite of the analytic predictions of rotor-dynamics theory of rotor-bearing stability, that "tilting-pad bearings by definition are stable". Only changes in seal geometry and radial clearances eliminated the harmful subsynchronous vibration.

The number of pumps with floating-ring type or with mechanical seals were over 730, and 748 failures were reported which is the highest failure rate among all pump component failures. When in good condition, the mechanical seals have lower leakage rate than the labyrinth type, but obviously more prone to wear and failure. After some wear, the seal leakage increases rapidly, especially with the floating-ring type. This then requires excessive maintenance which results in lower pump, hence unit, availability, or if not repaired, rapidly decreasing pump efficiency.

Considering the high cost of unit down time to replace failed seals and the high repair cost with the mechanical or floating-ring seal types, it becomes obvious that the best seal type for feed pumps is the labyrinth type. Several utilities reported successful conversions to labyrinth seals after a long history of failures with the other seal types.

The reason for the demand of other seal types is because of the higher apparent leakage rate of the labyrinth type. The leakage of a properly designed labyrinth seal is not higher than for a floating-ring type. Mechanical face seals, in spite of their high failure rate, are favored many times because they do not require external water injection and when performing well have the lowest leakage rate among all seal types. The trade off between apparent higher efficiency and pump reliability should be fully recognized.

4.5.3 JOURNAL BEARINGS AND ROTOR - Practically all boiler feed pumps apply "flexible" shafts, which means that the pump operating speed is above the rotor first lateral critical speed. In addition to this, because of the relatively light rotor weight, the journal bearings are lightly loaded, which makes them prone to instabilities such as oil-whip (subsynchronous vibration component). Single stage nuclear feed pumps in general operate below the first critical speed, but the rotor weight is even lighter than boiler feed pump rotors. The difficulty with light bearing loads is that the speed at which rotor dynamic instability starts is lower than normally expected. This speed is called the threshold speed, above which the bearing fluid film looses its ability to damp out rotor excitation forces at frequencies below approximately half the rotational frequency. Self-excited subsynchronous rotor whirl instability (oil-whip) then occurs. This loss of low frequency bearing damping is particularly harmful in feed pumps (even at speeds below the threshold speed) because of the large low-frequency hydraulic forces which are produced by hydraulic instabilities, especially at part load operation. Subsynchronous or low frequency excitations also originate in the seals and wear-ring surfaces, induced both by fluid dynamical phenomena and by rubbing. Feed pump reliability would therefore be improved considerably by the development of advanced bearing and rotor configurations which introduce large amount of low-frequency damping into the system. The use of tilting-pad journal bearings is a good first step in this direction. A summary of commonly used bearing configurations is given below:

PLAIN JOURNAL BEARING - The simplicity and high static load-carrying capacity of plain sleeve or axial groove sleeve bearings has resulted in their wide acceptance, even though the static bearing loads are generally small as previously described. Although it is the least expensive bearing type, it does not offer resistance to sub-synchronous excitations, and as a result, vibration related failures with these bearings are much more frequent than with other bearing types presently used. Although the cause of such failures is the bearing, the bearing itself is not necessarily damaged or failed during the accident. This has created the misleading belief that this bearing type is well suited for the application.

PRESSURE-PAD BEARINGS offer a somewhat greater stability margin than sleeve bearings due to the hydrostatic pad in the upper half of the bearing. This bearing type is often referred to as "pressure-dam" bearing. It is adequate in many feed pump applications except when hydraulic forces become extreme or heavy internal rubbing occurs causing strong sub-synchronous rotor vibration.

TILTING-PAD BEARINGS, or often called pivoted-shoe bearings, generally have whirl-free characteristics, which makes them the best choice for use in high speed, lightly loaded machines, such as nuclear or boiler feed pumps. Although these bearings are the best available type, utilities reported feed pump failures with sub-synchronous vibration components present in spite the application of properly designed tilting-pad bearings. In these cases further improvement of the bearings alone will probably not furnish the "cure". Improvement of the hydraulic components is presently the only way these pump problems can be corrected. In at least one of the above mentioned cases it was clearly demonstrated that the problem was caused by applying smaller than the commonly used pump internal radial clearances resulting in strong hydraulic excitation forces giving rise to rotor sub-synchronous vibration. The development of advanced rotor-bearing configurations, optimized for vibration damping, is a promising approach for future applications.

Of the pumps surveyed 693 used axial groove bearings and 250 cases of bearing failures were experienced in these pumps. In the other categories 8 bearing failures were reported from 214 pumps with pressure-pad type and 3 bearing failures from 137 pumps with tilting pad type journal bearings. Pressure pad and especially the tilting-pad type bearing offers greater support in low loading conditions, but are more expensive at first cost.

4.5.4 THRUST BEARINGS are of the Kingsburry pivoted type in all feed pump and booster pump applications, and with almost no bearing-design caused machine failures. When thrust bearing failure occurs they are invariably associated with other primary problems (e.g., axial balancing device failure). The thrust bearing in a feed pump plays a role only during start-up and coast-down. During normal operation thrust bearings are essentially unloaded, the primary thrust load being carried by the axial balancing device. Thrust bearing failures usually trace down to faulty assembly of the rotor, failure of the axial balancing device, or other Q. A. problems. A minor number of failures were reported due to failure of the lub-oil system. Two failures were reported where the thrust collar was not properly seated on the shaft, resulting in fatigue failure of the shafts, after exactly the same number of operating hours in both cases.

Of the 78 total number of thrust bearing failures reported, 58 occurred with boiler feed pumps, and almost all were explained as a consequence of an axial thrust balancing device failure.

4.5.5 IMPELLER BREAKAGE usually results in major pump damages and causes are difficult to determine. It is a frequent result of other problems such as vibration, hydraulic instability and cavitation damage. However, damage may also result from design deficiencies such as stress risers, inadequate strength or faulty casting quality. The survey data indicates that the majority of the 184 impeller breakages reported were caused by unusual close clearances (1% or less) between the impeller and diffuser/volute tongue. Of the 169 boiler feed pump impeller failures, 114 were diffuser type pumps and 55 were volute type. Of the total number of boiler feed pumps surveyed, 672 were diffuser type, 372 were volute type. Hence, the failure rate is approximately the same (10%) for both volute and diffuser type boiler feed pumps.

4.5.6 SHAFT BREAKAGE - The existence of stress risers or material flaws can contribute to shaft breakage due to fatigue, but failure of other pump components will produce exceptionally high stress conditions. Of the 121 shaft damages reported, 77 occurred with feed pumps, several of them in 800 MW and larger units. In almost all cases reported, the shaft failure was diagnosed as a secondary failure mode caused either by another component disintegration just prior to the shaft breakage or as a result of high vibration amplitudes at impeller vane-passing frequency for an extended period of time. The possibility of shaft breakage due to fatigue is particularly high in cyclic units. Several failures were reported due to overloading the shaft by the high hydraulic forces acting on the axial balancing device. If the retaining mechanism of the disk/drum or the fitting of it is not proper on the shaft, very high cyclic forces result, accelerating fatigue failure. There were some cases where improper heat-treatment of the shaft was argued to be the cause of the failures.

4.5.7 IMPELLER WEAR-RINGS - Excessive wear of the impeller wear-rings is in most cases the result of excessive shaft flexibility and operation at conditions where large shaft vibrations are encountered. It can also follow journal bearing wear. Overly close wear-ring clearances in case of a new pump or new components will also lead to rapid wear of the rings. In any case, wear of these rings will make it impossible for pump efficiency to be maintained at the initial level. Therefore, new pumps should have sufficiently large clearances to insure against rubbing or seizure. As with balancing disks, a proper choice of component materials is important for providing good accomodation of occasional rubs, thus avoiding consequential pump failures and premature replacement of wear rings.

Section 5

OPERATION

With many utilities, reliability of boiler feed pumps is beginning to gain priority over such objectives as pump efficiency at design point only and initial cost. Better testing procedures at the manufacturers' shops, adequate instrumentation for witness testing, and proper vibration monitoring during normal operation are getting more attention than a few years ago. New standards are evolving in areas such as vibration and pressure pulsation safe levels, safe operating ranges and realistic efficiencies. Standards for new concepts such as shaft axial vibration of single stage nuclear reactor feed pumps are essential to determine safe operating ranges. Until recently, such standards either did not exist, were not known, or were not adequately emphasized.

5.1 VIBRATION AND VIBRATION STANDARDS

Generally accepted radial vibration amplitude levels are shown in Figure 20 as measured on the shaft relative to the bearing cap. The major disadvantage of most available standards is that they refer to synchronous frequencies only, while the majority of failures are the results of other than just synchronous unbalance. Figure 20, compiled by ERCO from field experience, shows vibration displacement at various frequencies, but shows also the equivalent velocities and accelerations. Hence the chart applies to readings with proximiters (displacement), velocity pickups, as well as with accelerometers. Amplitudes of vibration at the various major frequencies is considerably more informative than the synchronous component or the broad band values alone. Figure 21 shows various frequency ranges and some explanations for their causes and cures.

Axial vibration levels for single stage double suction pumps, such as the majority of nuclear feed and booster pumps, is an excellent indication of overall pump design quality. It can be used to detect hydraulic instability of the impeller or volute/diffuser, or to set safe minimum operating flow (recirculation flow). There are presently no established standards among pump manufacturers on danger threshold levels for axial vibrations or safe minimum operating flows. Figure 23 shows a field example to detect onset of hydraulic instability and what proper design correction can achieve. Figure 24 shows another actual case where ERCO used axial shaft vibration of a nuclear feed pump to establish the safe minimum flow.

Of the total number of pumps surveyed, 373 vibration problems were reported by responding utilities. Pump vibration problem is a loosely defined phenomenon, however, most obvious to detect. It is frequently a symptom of a bearing failure, hydraulic instability, internal rubbing of close clearance components or an indication of the beginning of a seal failure. However, it may also be an indication of an inherent deficiency in the design of the pump, foundation, piping, control valve or a combination of these. Multi-stage boiler feed pumps having relatively long shafts are almost always designed to operate above the first critical speed of the rotor, sometimes even above the second critical speed. This occasionally poses a problem when variable speed or cyclic operation produces frequent operation through or at the critical speed(s). In many cases, power plants specified (when ordered) for base load duty are now in the cyclic load category and may exceed their design fatigue life. New pumps for such duty should be more carefully specified. Vibration is seen to account for most component failures in both nuclear feed pumps and other single stage booster pump designs. Proper instrumentation to detect vibration levels and frequencies is therefore extremely important during both factory witness testing and field operation. Recommendations for testing and applied instrumentation is discussed in Section 5.4 below.

5.2 PRESSURE PULSATION - There are presently no pressure pulsation standards available among manufacturers. Often a 3% maximum of a pump-stage produced head is loosely quoted, as measured on the discharge side of the pump. This is satisfactory in most cases. Pumps analyzed during the survey revealed that vendors almost always quote pressure pulsation amplitudes at blade-passing frequencies (number of impeller blades "Z" times rotational speed "N"). Part (c) of Figure 26 shows hardly any amplitudes at vane-passing frequency, but the same data properly analyzed (Part (b) of Figure 26, and Figure 28) shows significant amplitudes at other than vane-passing frequencies. There was sufficient data collected on nuclear feed pumps having pressure pulsation difficulties to warrant setting standards on safe pressure pulsation levels as measured in the pump discharge nozzle or piping. Figure 22 is an attempt to explain frequencies and magnitudes of hydraulically induced dynamic forces within a high speed feed pump.

5.3 SAFE OPERATING RANGES - The subject of "Minimum Flow", often referred to as "Recirc Flow" is a continuously disputed subject when failure arises due to lower than necessary minimum flow, or because the necessary safe minimum flow of that pump design is too high. Figure 17 shows minimum flow standards for various pump types and applications. Due to keen competition in the pump market, vendor prices and pump efficiencies all fall within narrow ranges. In many instances, this has prompted manufacture application (sales) engineers to offer optimistically low recirculation flow or unrealistically high NPSH capability of the first stage impeller. For example, instead of guaranteeing 25% recirculation flow, a vendor may offer 10%. Then the minimum flow line and its control valve can be reduced in size considerably, and he can offer a saving in price. This seems attractive initially, especially to the A/E, but not when you consider the extensive pump and system damage that can result from low recirculation flow. The design margin band shown in Figure 17 represents a regime in which the experience and assumptions of the pump designer come into play. Safe minimum flow for large nuclear feed, or boiler feed pumps is not less than 25% of design flow, but may be as high as 45 or 50%, which is not acceptable, and the pump hydraulic components are to be improved as shown in the example given in Figures 24 and 25 in Appendix II. The right side of the design band shown in Figure 17 may represent the influence of incorrectly selected suction specific speed of the impeller. If the S_s is too high, the useful range of operation becomes narrower, also, achievable efficiencies are lower. A suction impeller (an impeller with high S_s, hence higher than normal NPSH capabilities) is always less efficient than a normal impeller design and has a narrower useful operating range.

5.4 TESTING AND INSTRUMENTATION - Many useful observations can be made during factory witness testing. One of the most important phenomenon to look for during acceptance test is the Head-Capacity curve of the pump. Figure 8 shows an example of a troublesome pump head curve shape. If the head curve is droopy, or flat toward decreasing flows, or if it has a "kink" in it, the pump is hydraulically unstable in that flow range, and should not be accepted, namely pump, piping, control valve control system, sometimes steam generator level control malfunctioning may result. Parallel operation of the feed pumps is difficult in this flow range, and may even result in occasional tripping of the entire unit if the pumps are operated in this range. This problem is usually caused by the pump impeller, or impeller and diffuser/volute hydraulic design. It is recommended to analyze peak-to-peak pressure pulsation ranges not just at the blade passing frequency, but in a wide range, from low frequencies to at least up to and including twice blade passing frequency. Proper observation of the rotor is also important. Figure 20 shows a typical case where improper measurements actually did more harm than good. Installing one proximity probe not only did not detect the problem, but actually was misleading and in the case shown in Figure 29 allowed the rotor to drive itself into destruction. It is recommended to use two probes always at each bearing to monitor shaft and bearing behavior properly. The two probes should be 90° apart. If the pump has a balancing device, the disk leak-off flow should be measured and analyzed in the complete operating range of the pump as shown in Figure 27. It shows a case where a boiler feed pump failure was intercepted but more important, the results of simple instrumentation allowed us to detect degradation of the feed pumps. For a reliable feed pump, test the following minimum number of locations should be considered: Two non-contact type probes at each bearing (90° apart), one for shaft axial movement especially for single stage double suction type pumps, at least one pressure transducer in the discharge nozzle or piping, one accelerometer on the pump casing (most sensitive location is to be determined during testing), and balancing line leak-off flow.

EPRI REPORT NO. RP 641

Appendix

I: QUESTIONNAIRES OF THE SURVEY

ENERGY RESEARCH & CONSULTANTS CORP.
900 OVERTON AVENUE - MORRISVILLE, PA

ENERGY RESEARCH & CONSULTANTS CORP.

900 OVERTON AVENUE - MORRISVILLE, PA. 19067

TELEPHONE: (215) 295-2850

EPRI PROJECT RP 641 COORDINATED WITH EEI

October 20, 1975

TO: ALL operating and future large Nuclear and Fossil power generating stations (500 MW and larger)**.

SUBJECT: Improve Power Plant Availability:"LARGE PUMP PROBLEMS".

Large pump operating problems are a major source of plant outage. Recognizing this, EPRI has initiated a project with the objective to "IMPROVE PLANT AVAILABILITY" by optimum utilization of existing pump-failure information and technology. The cooperation of utilities in supplying information requested on the enclosed questionnaires will enhance the success of this project.

Drawing from our previous experience and a thorough evaluation of the completed questionnaires a comprehensive guide will be prepared suitable as a ready reference to utilities on pump operating problems. We believe this guide will have a beneficial impact in the following areas:

- o Solutions to existing problems
- o Avoidance of duplicate failures
- o Procurement of future equipment
- o Identifications of present technology deficiencies

Our present scope includes centrifugal pumps for the following services:

- o Boiler Feed
- o Nuclear Feed
- o Condensate Booster

Please distribute questionnaires to the appropriate plant or project engineers of all your operating stations and plants not yet started. New plants may already have tested their pumps and discovered an inherent problem. This is vital information to other utility members in the process of purchasing the same problem you have already discovered or perhaps solved.

Where possible please provide us with failure evaluation reports. Such documentation will fill information gaps not accounted for by the questionnaires. We also welcome suggestions on how to increase plant availability through improved pump operations.

Please return all questionnaires, data, comments, etc. to the above address. For personal communication on any details please contact the undersigned. Your cooperation is appreciated.

Sincerely,

Elemer Makay

Elemer Makay, Technical Director
EPRI Project RP 641

Copies to:
List (All US Utilities)
Mr. D. Q. Hoover, EPRI-Palo Alto
Mr. H. A. Moshell, EEI-Tampa Elec.

** In case of a single 100% feed pump, 250 MW, or smaller if technology related to pump difficulties reflect on larger sizes or reveal GENERIC type problems.

ENERGY RESEARCH & CONSULTANTS CORP.

900 OVERTON AVENUE - MORRISVILLE, PA. 19067

TELEPHONE: (215) 295-2850

EPRI PROJECT RP 641 COORDINATED WITH EEI

QUESTIONNAIRE
(fill out one for each failure case)

TO: All operating and future large Nuclear and Fossil power generating stations (500 MWE/Unit and larger)**

FROM: Project Director of EPRI Project RP 641

SUBJECT: Improve Plant Availability:"LARGE PUMP PROBLEMS".

1. Does your Co. have Boiler Feed, Nuclear Feed and Cond. Booster pumps installed on 500 MWE** and larger Units, which have given operating, etc. troubles?

YES:_______ NO:______

If the answer is YES, please fill out also the more detailed (table type) questionnaire in addition to the questions below (for type and performance see item 6)

Name Pw. Co.:______________________ A.E.:__________

Name Pw. Sta.:______________________ Size:__________ MWE

2. Estimate Outage (If downtime caused by pump) in KW,Kwh,$,%Avg/Year,etc.:______

3. Do you classify this case as:

a) Pump Caused problem: YES:_____ or NO:____; System related:YES:___or NO:____
b) Generic to pump service?___________ If so,which services?_______________
c) Generic to manufacturers' design, fabrication, Q.A.,etc.?_______________
Name(s) of manufacturer(s)_______________________
d) Generic to a design (or sales) philosophy followed by manufacturers as a whole:_________ Explain:_______________________
e) Trouble unique to installation?_________ Describe:_______________
f) Other? Describe:_______________________
g) Cause not yet determined:_______________________
h) Case presently is disputed with: Vendor:____, A.E.:_____, Both:__________
i) Station learned how to live with problem:_______________________

** In case of a single 100% feed pump, 250 MW and larger. Or smaller sizes, if technology related to pump difficulties reflect on larger sizes, or reveal GENERIC type problems.

4. On the scale of 0 to 10 (0=poor, 10=excellent) please express the degree of satisfaction you have for manufacturer's efforts to correct the trouble with your pump(s):______________________________

5. Do you welcome ONE organization to make a detailed survey, then establish a dialogue with pump manufacturer about these troubles?

   YES_____ NO_____ , EPRI_____ , EEI_____ ,Other(describe)__________
   If answer is:YES-EPRI,will be handled under EPRI Project RP 641 coordinated with EEI, If:YES-EEI, this information will be transmitted to EEI Prime Movers Committee Chairman for further action.

6. Describe each pump with which you had trouble:

   Service______________ Pump delivered (~ year):__________
   In Operation since("):__________
   Capacity______________GPM Head______________ Ft
   Driver______________HP Speed(or Range)______________RPM
   Manufacturer______________ Model No.______________ No.of Stages:____

7. Nature of troubles (use next page Table form detailed questionnaire for details as well as space below if special remark) describe symptoms briefly:______

   ____
   ____
   ____

8. What changes were made in the pump or F.W. system (please also describe those that did not help):______________________________

   ____
   ____
   ____

PLEASE RETURN COMPLETED QUESTIONNAIRES TO:

Dr. Elemer Makay, Technical Director
EPRI Project RP 641
Energy Research & Consultants Corp.
900 Overton Ave.
Morrisville, Pa. 19067

ENERGY RESEARCH & CONSULTANTS CORP.

IMPROVE PLANT AVAILABILITY OF LARGE NUCLEAR AND FOSSIL STATIONS

EPRI Project No.: RP 641

Coordinated with EEI

							CAUSES OF OVERALL PROBLEMS																		TYPE OF VIBRATION (IF VIB.)																
VENDOR	NO.OF FAILURES OF SAME PUMP OR SAME TYPE	NO.OF PUMPS IN THE STA.	PUMP TOTAL LOSS	ROTOR REPLACED	PARTS REPLACED	NONE "	UNDETERMINED	HEAD-CAPACITY	UNSTABLE HEAD	AVAILABILITY	SHAFT BROKE	SEAL OR S.SYSTEM	BALANCE DEVICE	DYNAMIC UNBALANCE	WATER HAMMER	CAVITATION	IMPELLER BLADE BROKE	" SIDEPLATE "	CASTING QUALITY/CORE SHIFT	MATERIAL	Q. C.	INADEQUATE PIPE/P. SUPPORT	HIGH MAINT. COST	" SPARE PART "	UNDETERMINED	PUMP	DRIVER	FLUID COUPLING	DISCONNECT "	SUCTION PIPE	DISCHARGE PIPE	FOUNDATION	CONTROL VALVE	CONT. SYSTEM	CRITICAL SPEED	JOURNAL BEARING	THRUST BEARING	RADIAL	AXIAL	PRESSURE PULSATION	BASIC DESIGN
Al-Ch.																																									
B-J																																									
Bingham																																									
B & W Ltd.																																									
De Laval																																									
I-R																																									
Pacific																																									
Westinghouse																																									
Worthington																																									
Other:																																									

DATE: . . . .

APPROX. DATE OF FAILURE: . . . .

A. E.:

PUMP TYPE	CHECK
NUCLEAR FEED	
BOILER FEED	
BOOSTER	
OTHER:	

POWER STATION:

SIZE. . . . . . MWE

TYPE:

FOSSIL	
PWR	
BWR	
OTHER:	

NAME POWER CO.: . . . . NAME POWER STA.: . . . .

ADDRESS: . . . . .STREET; . . . . CITY; . . . . STATE; . . . . .ZIP

NAME TO WHOOM FURTHER CORRESP.
TO BE SENT ON ABOVE SUBJECT:. . . . . TITLE . . . . TEL.NO.:. . . . .

PLEASE RETURN INFO TO: Dr. Elemer Makay, ENERGY RESEARCH & CONSULTANTS CORP.,900 Overton Ave.,Morrisville, Pa. 19067
Telephone No.: (215)-295-2850

Appendix

II: TABULATION OF FAILURES BY MECHANISMS,

COMPONENTS AND FRAME SIZES

PUMP TYPE NO.	PUMP TYPE	FAILURES REPORTED FROM TOTAL NO. OF STATIONS	UNITS	NO. OF PUMPS IN THOSE STATIONS	NO. OF FAILURES IN THOSE STATIONS	TOTAL NO. OF PUMPS SURVEYED	NO. OF FAILURES/ FAILED PUMPS	% FAILURE RATE
1	BOILER FEED	150	203	362	763	1044	2.1	34.7
2	NUCLEAR FEED	20	30	61	133	160	2.2	38.1
3	F. W. BOOSTER	28	40	123	155	123	*	*
	SUB TOTAL:	178	240	546	1051	1327	*	*
4	PRIMARY COOLANT	24	29	85	*	106	*	*
5	CONDENSATE	25	31	89	*	89	*	*
6	CIRC & SERVICE W	10	12	49	*	49	*	*
	SUB TOTAL:	(30)	(40)	223	*	244	*	*
	TOTAL:	**178	**240	769	(1305)	1571	*	*

TABLE I: FAILURE RATE OF UTILITY PUMPS USED IN LARGE CENTRAL POWER GENERATING STATIONS

* NOT APPLICABLE, THOSE PUMP TYPES WERE NOT THOROUGHLY SURVEYED.

** PUMP TYPES 4, 5, AND 6 ARE EXCLUDED.

( ) APPROXIMATE NUMBERS.

NO.	BFP VENDOR	NUMBER OF PUMPS IN OPERATION					
		A	B	C	D	E	TOTAL
1	AL-CH	44	16	3	-	-	63
2	B - J	96	131	20	2	-	249
3	DE LAVAL	36	111	23	2	-	172
4	I - R	32	82	37	20	3	174
5	PACIFIC	211	91	22	2	-	326
6	WORTHINGTON	48	40	32	-	-	120
6	TOTAL:	467	471	137	26	3	1,104

TABLE II: BOILER FEED PUMPS IN OPERATION. MADE BETWEEN 1963 AND 1977 (FOREIGN SHIPMENTS EXCLUDED).

- NONE, OR NOT SURVEYED.

VENDOR	NO. OF UNITS		NO. OF PUMPS IN OPERATION						NO. OF PUMPS WITH NO. OF STAGES:			
			MOTOR DRIVE SPEED		TURB. DRIVE	T O T A L						
	PWR	BWR	LOW	HIGH	HIGH	LOW	HIGH	M+T	1	2	3	4
BINGHAM	7	1	11	2	4	11	6	17	12	5	–	–
B – J	14	9	19	12	22	19	34	53	28	17	5	3
DE LAVAL	7	2	2	2	16	2	18	20	18	2	–	–
I – R	3	1	3	0	6	3	6	9	6	–	3	–
PACIFIC	1	5	2	6	9	2	15	17	11	6	–	–
WORTHINGT	6	2	3	8	8	3	16	19	19	–	–	–
TOTAL:	38	20	40	30	65	40	95	135	95	30	8	3

TABLE III: TOTAL NUMBER OF NUCLEAR FEED PUMPS IN PRESENTLY OPERATING* LARGE ** BWR AND PWR UNITS, CLASSIFIED BY: VENDORS, DRIVE, SPEED AND NUMBER OF STAGES.

* UNITS UNDERGOING HOT FUNCTIONAL NOT INCLUDED (AS OF 10-10-77)

** UNITS SMALLER THAN 430 MW ARE EXCLUDED

FRAME SIZE		BFP	RFP	BOOS.	OTHERS	TOTAL
A	SURVEYED	406	*	*	*	406
	FAILURES	78	*	*	*	78
B	SURVEYED	472	2	23	*	497
	FAILURES	198	2	23	*	223
C	SURVEYED	137	72	71	*	283
	FAILURES	75	30	92	*	197
D	SURVEYED	26	78	15	*	120
	FAILURES	11	25	26	*	62
E	SURVEYED	3	8	14	*	25
	FAILURES	**	4	14	*	18
NO. OF PUMPS SURVEYED		1,044	160	123	244	1,571
NO. OF PUMPS FAILED		362	61	123	244	790
% FAILURE RATE		34.7	37.2	*	*	*
TOTAL NO. OF FAILURES		763	133	155	254	1,305

TABLE IV: PUMP FAILURES BY TYPES AND BY FRAME SIZES.

* NOT SURVEYED. Many utilities reported failures only with 500 MW and larger units, hence a large portion of A-Frame size pumps was not surveyed.

** NOT REPORTED BY THE USERS.

NO.	PUMP FAILURES: COMPONENT, SYMPTOM OR TECHNOLOGY	FEED PUMP	BOOSTER PUMP	OTHER PUMP TYPES	TOTAL NO.OF FAILURES
1	SEALS	602	178	198	978
2	VIBRATION: PUMP, PIPING, FOUNDATION	228	85	60	373
3	AXIAL BALANCING DEVICE	337	--	--	337
4	JOURNAL BEARING	209	52	37	298
5	CAVITATION	161	64	46	271
6	IMPELLER BREAKAGE	169	8	7	184
7	WEAR-RING: RAPID WEAR	155	3	-	158
8	UNSTABLE HEAD CURVE:	92 *	59 *	10 *	161 *
9	SHAFT BROKEN/DAMAGED	77	6	51	134
10	THRUST BEARING	58	11	9	78

TABLE V: TEN MAJOR OUTAGE PRODUCING FAILURE CAUSES.

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

EEI-LIST
SUMMARY OF ANSWERS
TO QUESTIONNAIRE
ON
"LARGE PUMP TROUBLES"
(SEP.1975)

1.	COMPANIES RESPONDING		60 TOTAL
			10 NO
			50 YES
2.	TOTAL NUMBER PUMPS WITH TROUBLES		101 TOTAL
			56 GENERIC
			45 NON-GENERIC
3.	DEGREE OF SATISFACTION WITH MFG. EFFORT		0-10 Range
			5.7 Avg.
4.	RECOMMEND DIALOGUE WITH MFG.		42 YES
			59 NO
5.	CLASSES OF PUMPS	Reactor Coolant (PCP) Reported:	8
		Reactor Feed	4
		Condensate	20
		Boiler feed	34
		Boiler Circ.	12
		Miscellaneous	23
		TOTAL:	101
6.	NUMBER OF PUMPS BY MFG.	I	6
		II	1
		III	20
		IV	31
		V	8
		VI	16
		VII	4
		VIII	5
		IX	10
		TOTAL:	101
7.	NATURE OF TROUBLES		
	A.	Capacity	2
	B.	Reliability and Availability	16
	C.	Seals and sealing arrangement	30
	D.	Cavitation	11
	E.	Materials	19
	F.	Vibration or Pulsation	17
	G.	High maintenance cost	8
	H.	Other	29

TABLE VI: EEI STATISTICS RECEIVED FROM THE EEI-PRIME MOVERS COMMITTEE IN SEPTEMBER 1975.

PUMP FRAME SIZE	NO.OF FAILURES REPORTED WITH SEAL TYPE			TOTAL NO.OF FAILURES	NO.OF PUMPS SURVEYED
	MECH.	FLOATING RING	LABYR.		
A	34	63	0	97	406
B	34	197	17	248	497
C	240	151	11	402	280
D	2	21	4	27	119
E	4	2	0	6	25
TOTAL:	314	434	32	780	1327

TABLE VII : SEAL FAILURES TABULATED BY PUMP FRAME SIZES, AND SEAL TYPES (BFP*,NFP* AND BOOSTERS).

* For abreviations see List of Symbols at the end of this Report.

PUMP FRAME SIZE	BOILER FEED PUMPS NO.OF FAILURES	NO.OF PUMPS SURVEYED
A	68	406
B	192	472
C	62	137
D	15	26
E	0	3
TOTAL	337	1044

TABLE VIII: AXIAL BALANCING DEVICE (DISK/DRUM) FAILURES BY FEED PUMP FRAME SIZES.

TOTAL NO.OF IMPELLER BREAKAGES REPORTED	184
Of the 184 pumps, 169 are feed pumps.	
Impeller breakage at EXIT	112
Impeller breakage at INLET	72
Volute Type (All pump types)	63
Out of 372 BFP	55
Diffuser Type (All pump types)	121
Out of 672 BFP	114

TABLE IX: IMPELLER BREAKAGES REPORTED FROM UTILITIES

TOTAL NO.OF CAVITATION DAMAGES REPORTED	271
Total No.of FIRST STAGE CAVITATION	192
Boiler Feed Pumps	89
Nuclear Feed Pumps	20
F.W.Booster Pumps	47
Other Pump Types	36
TOTAL NO. OF INTERSTAGE CAVITATION	79
Boiler Feed Pumps	46
Nuclear Feed Pumps	6
F.W.Booster Pumps	17
Condensate Pumps	10

TABLE X: CAVITATION DAMAGES REPORTED FROM UTILITIES

PUMP FRAME SIZE	NO. OF FAILURES REPORTED WITH			TOTAL
	BFP	NFP	BOOSTER	
A	25	-	-	25
B	7	2	-	9
C	35	11	34	80
D	22	4	13	39
E	0	3	0	3
TOTAL	89	20	47	156
OTHER PUMP TYPES:				36
TOTAL (ALL PUMP TYPES):				192

TABLE XI: NUMBER OF FAILURES CAUSED BY FIRST STAGE CAVITATION.

TOTAL NO. OF BEARING FAILURES REPORTED		298
FEED PUMP BEARING FAILURES		261
FEED PUMP BEARING FAILURE RATE (261/1204)		21.7%
FAILURES ATTRIBUTED TO:*		
Unknown or undetermined	Over	10
Bearing design	Over	80
Incorrect type or application	Over	50
Lub system/System design	Over	10
Manufacturing (Q.A., Material)		3
BEARING FAILURES BY TYPES (BOILER FEED PUMPS)		
Axial groove Journal	Out of 693 Pumps	250=36.0%
Pressure Pad/Hydrostatic	Out of 214 Pumps	8= 3.7%
Tilting Pad Journal	Out of 137 Pumps	3= 2.2%

TABLE XII: JOURNAL BEARING FAILURES REPORTED BY UTILITIES

TOTAL NO. OF SHAFT FAILURES	134
BOILER FEED AND NUCLEAR FEED PUMPS	77
FEED WATER BOOSTER PUMPS	6
PRIMARY COOLANT PUMPS (PCP)	11
CONDENSATE PUMPS	21
OTHERS	19

TABLE XIII: SHAFT FAILURES REPORTED FROM UTILITIES.

* Symptoms of the reported failures were discussed with many of the Utilities. The above judgement was jointly concluded during the surveywork. In occasions the vendors were also consulted with, however, their oppinion may have been in contradiction with ours.

MANUFACTURER	TOTAL MW	NO. OF UNITS	AVG. UNIT MW	P U M P T Y P E
I	3,814 - -	10 - -	381 - -	BOILER FEED NUCLEAR FEED F.W. BOOSTER
II	- 2,210 -	- 3 -	- 737 -	BOILER FEED NUCLEAR FEED F.W. BOOSTER
III	14,350 4,650 9,225	23 7 12	624 664 769	BOILER FEED NUCLEAR FEED F.W. BOOSTER
IV	34,099 6,350 900	65 7 1	525 907 900	BOILER FEED NUCLEAR FEED F.W. BOOSTER
V	31,982 650 11,395	49 1 16	653 650 712	BOILER FEED NUCLEAR FEED F.W. BOOSTER
VI	15,446 1,700 -	32 3 -	483 567 -	BOILER FEED NUCLEAR FEED F.W. BOOSTER
VII	15,475 2,770 4,800	27 5 8	573 554 600	BOILER FEED NUCLEAR FEED F.W. BOOSTER
VIII	4,300	4	1050	NUCLEAR FEED
	144,240	240	601	TOTAL

TABLE XIV: TOTAL PLANT CAPACITIES IN "MW" WHERE FEED PUMP FAILURES CAUSED POWER PLANT OUTAGES.

ANALYSIS OF AUXILIARY COMPONENT OUTAGES, FOSSIL FIRED UNITS OVER 600 MW

	OUTAGE CAUSE OR PROBLEM AREA																			
	Condensers									Feedwater Heaters				Boiler Feed Pump					Fuel Handling	
	GENERAL	CLEANING	TUBE FAILURE	CW PUMP	COND PUMP	EXPANSION JOINT	MISCELLANEOUS	TOTAL LISTED	OUTAGE RATE	LEAKS	DIRTY	TOTAL LISTED	OUTAGE RATE	GENERAL	DRIVES	BF PUMP	TOTAL LISTED	OUTAGE RATE%	COAL CONVEYING	OUTAGE RATE
EEI FAILURE CAUSE CODE	800	801	802	803	804	809	899			901	902			905	918,919 920,921	922			906	
FFO (hr)	.92	.38	2.1	2.7	1.4	1.7	6.8	16	.2%	6.1	0	6.1	.1%	25	3.2	.95	29	.3	.77	--
INCIDENCE	.05	.02	.04	.05	.06	.01	.12	.35		.11	0	.11		.74	.19	.07	1.0		.02	
HRS/INC.	17	19	47	49	22	117	58			54	0			34	17	13			31	
EFPO (hr)	.45	1.0	4.9	3.8	2.9	.20	2.9	16	.2%	9.6	.22	9.8	.1%	45	6.2	3.2	54	.6	3.7	--
INCIDENCE	.08	.42	.92	.20	.56	.02	.48	2.7		.81	.06	.87		3.6	1.1	.53	5.2		.93	
HRS/INC.	5.3	2.5	5.3	19	5.2	10	6.0			12	3.7			13	5.6	6.1			4.0	
SCHED OUT (hr)	285	18	1.5	.2	1.4	1.9	0.3	308	3.5%	4.3	0	4.1	0	62	2.6	2.6	67	.7	1.7	
% TOTAL	100	95	18	3	25	50	3	91		22	0	20		47	22	38	44		27	
TO (hr)	286	19	8.5	6.7	5.7	3.8	10	340	3.8%	20	.22	20	.2%	132	12	6.8	151	1.7	6.2	.1%
INCIDENCE	.87	4.1	1.0	.33	.83	.05	.65	9.5		1.1	.06	1.17		5.6	1.4	.71	7.7		1.1	
HRS/INC	328	4.7	8.3	20	6.9	71	16			18	3.7			23	8.6	9.5			5.8	
ANNUAL REI COST* ($mm)								44				6.8					39		2.4	
REL PRIOR.								H				M					H			

KEY

FFO - Full Forced Outage
EFPO - Equivalent Forced Partial Outage
TO - Total Outage
Incidence - Average Number of Events/Unit Year

H - High Impact ($15 million pa)

M - Moderately High Impact ($5-$15 million pa)

* Relative Cost - Base is 100 units at mean outage cost
$4000 per hour forced outage per unit
$1000 per hour scheduled outage per unit

Data Source: Edison Electric Institute. Equipment Availability, Fossil Component Cause Code Summary Report 1964-1973

TABLE XV: BOILER FEED PUMP OUTAGE DATA FROM EEI SUMMARY REPORT 1964-1973. (REPORTED IN EPRI REPORT NO. FP-422, JUNE 1977 BY DR. DON ANSON).

Appendix

III: ADDITIONAL FIGURES.

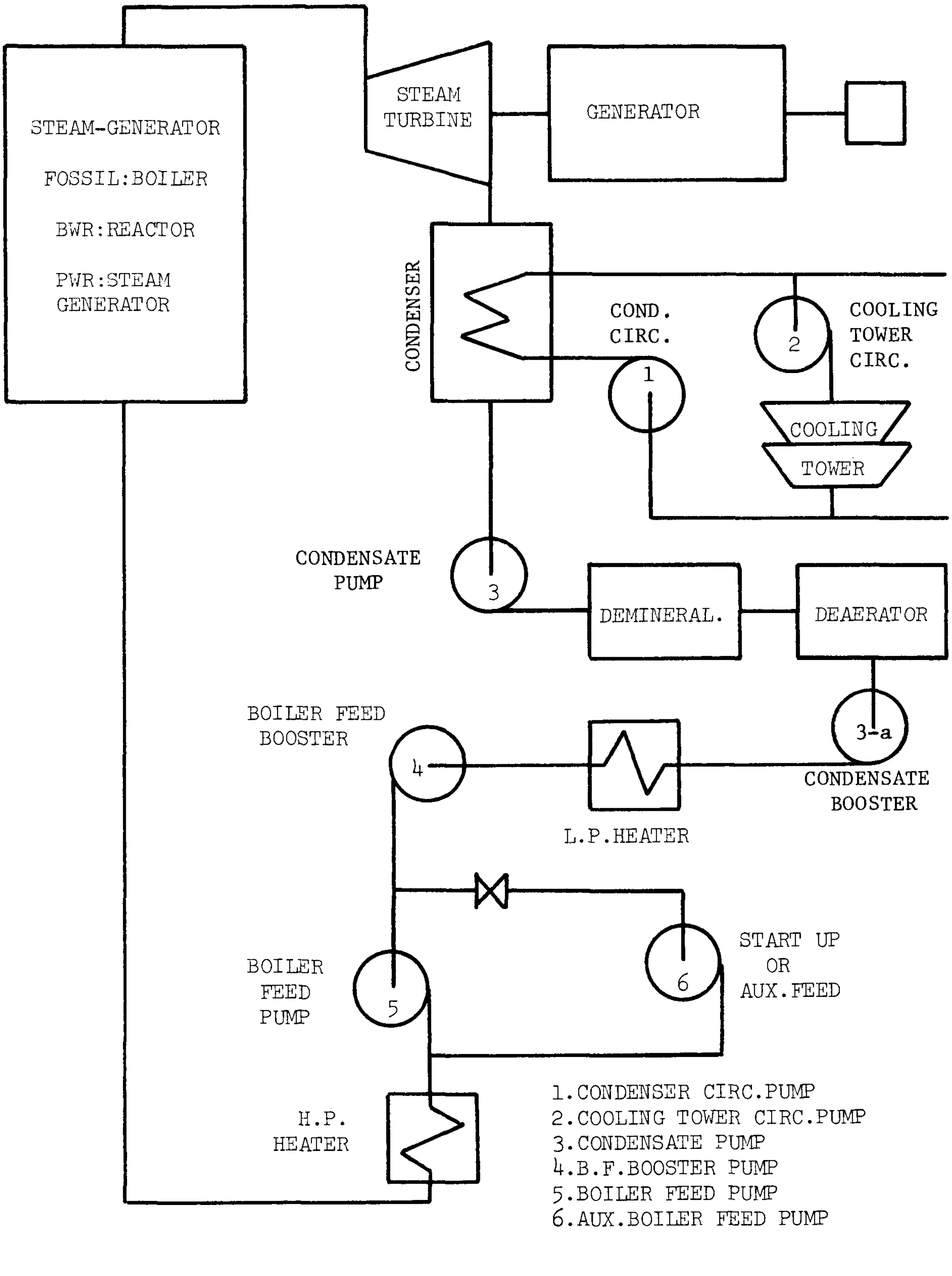


FIGURE 1: TYPICAL POWER STATION PUMP SERVICES.

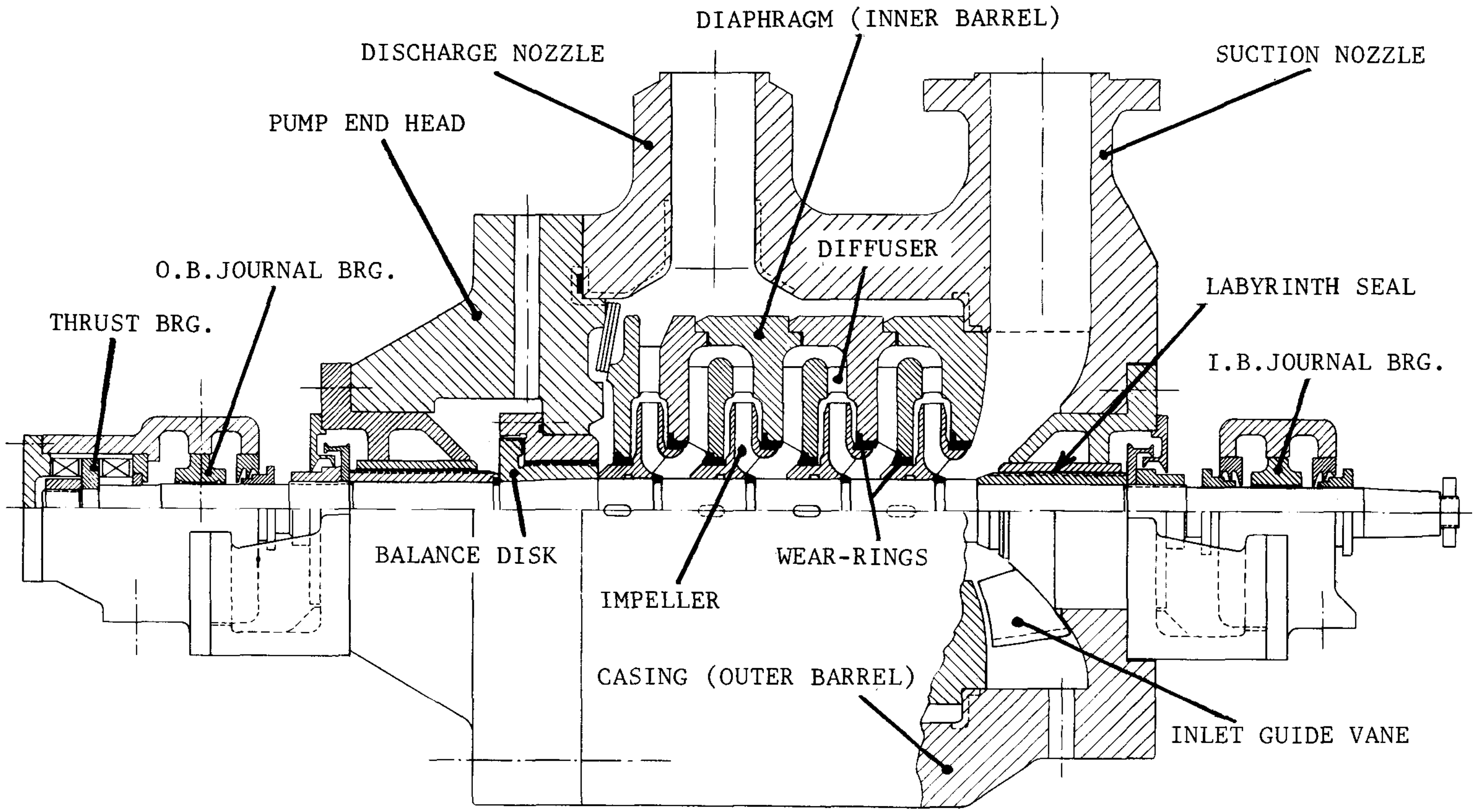


FIGURE 2: TYPICAL MULTISTAGE BOILER FEED PUMP. (IN-LINE IMPELLERS, SINGLE SUCTION FIRST STAGE, BALANCE DISK, LABYRINTH SEALS, DIFFUSER TYPE)

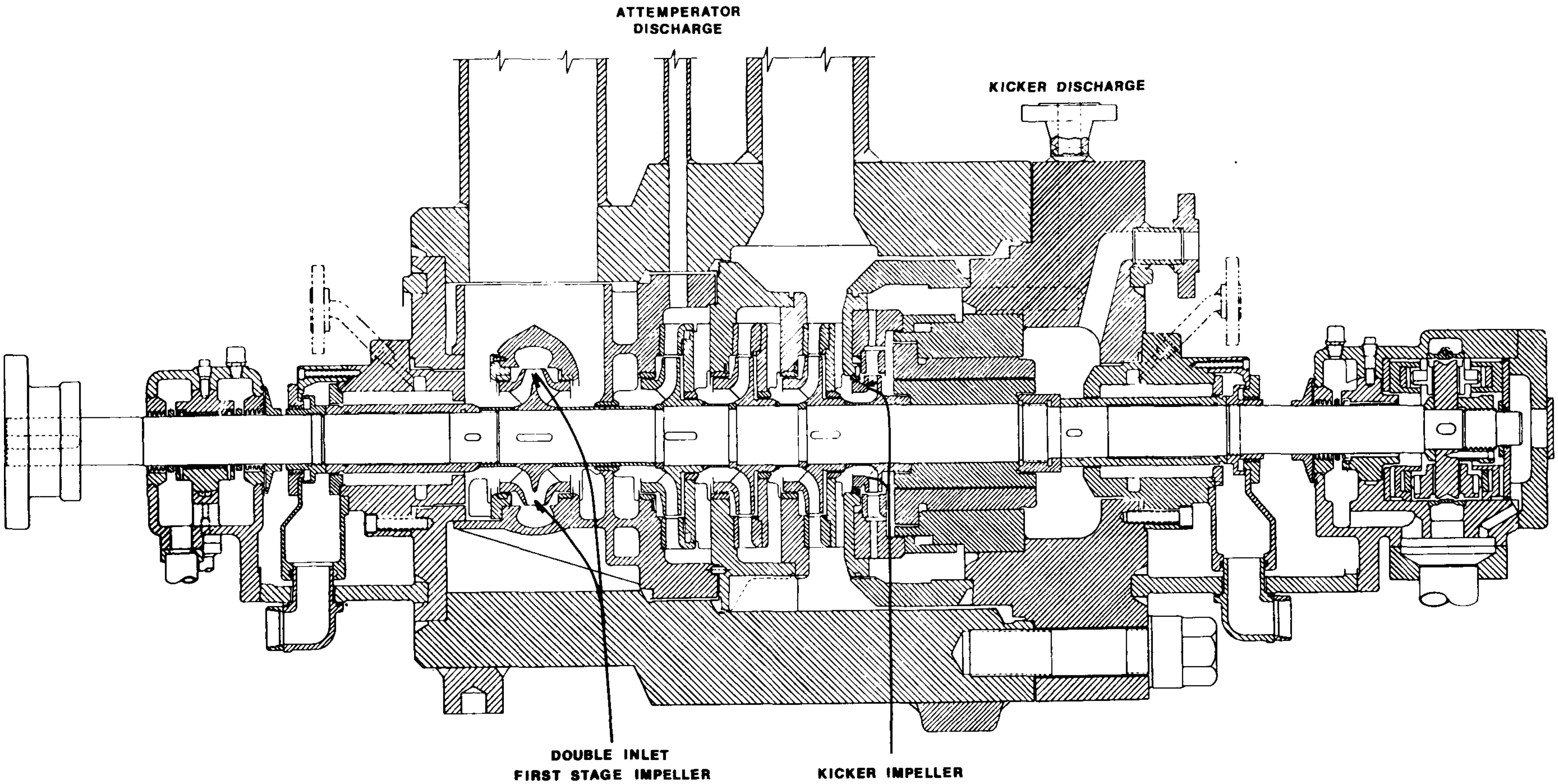


FIGURE 3: TYPICAL MULTISTAGE BOILER FEED PUMP. (IN-LINE IMPELLERS, DOUBLE SUCTION FIRST STAGE, BALANCE DRUM, SEAL TYPE NOT SHOWN-OPTIONAL, DIFFUSER TYPE)

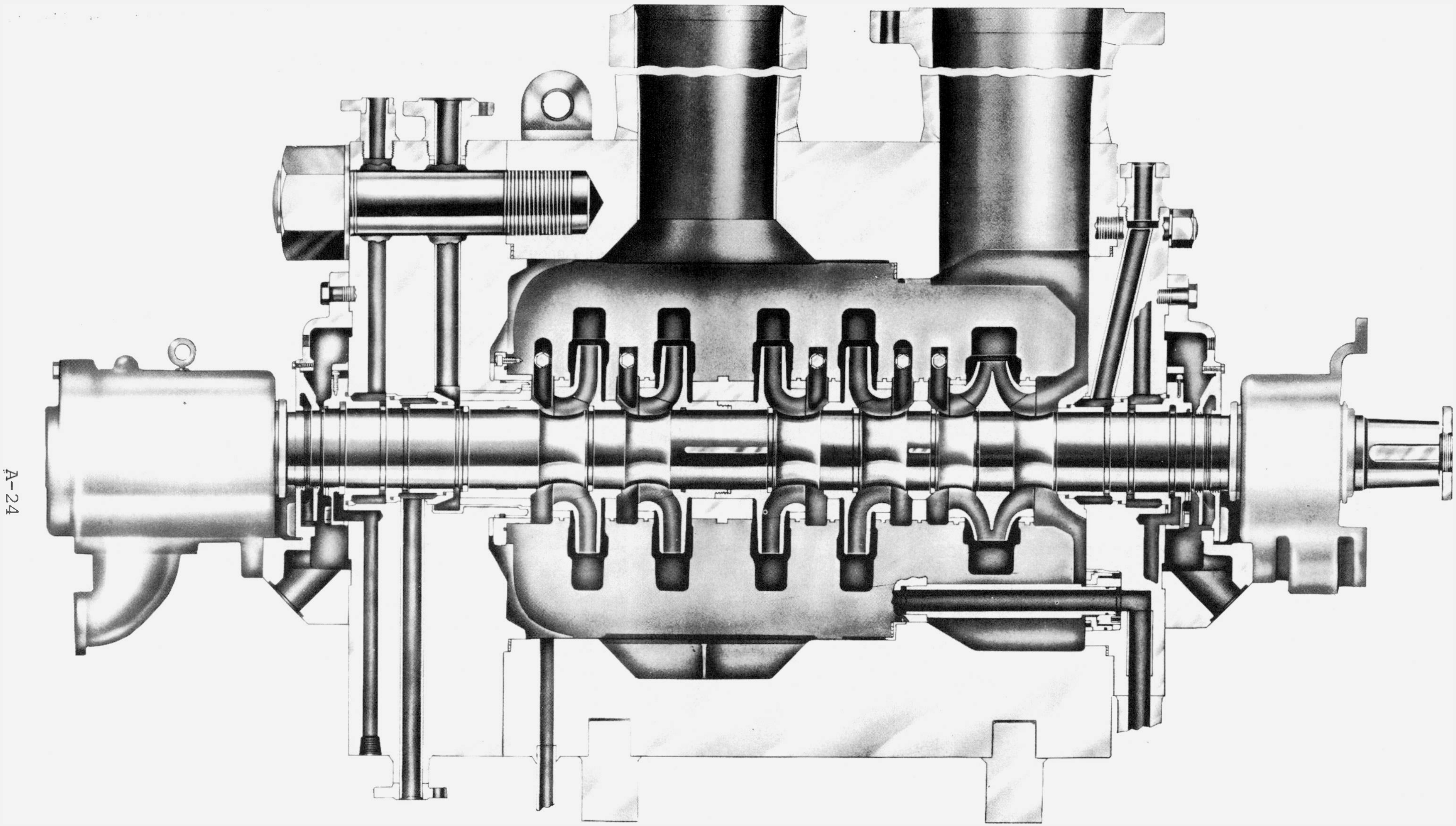


Figure 4. MULTISTAGE DOUBLE CASE BOILER FEED PUMP
(Forged steel outer casing, double volute inner casing, opposed impeller grouping for inherent axial balance, double suction first stage impeller, optional attemperator tap, and choice of shaft seals.)

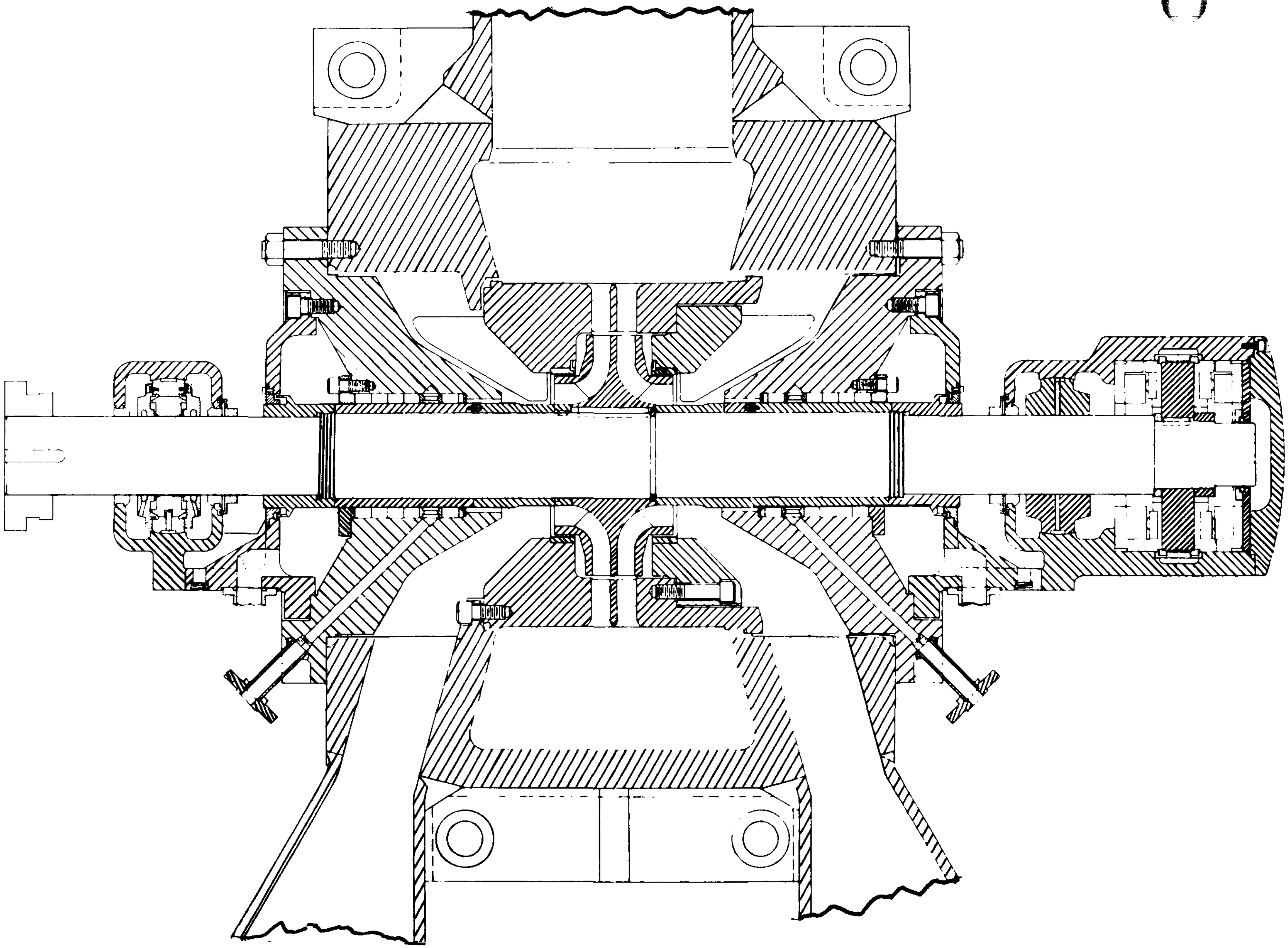


FIGURE 5: TYPICAL NUCLEAR FEED PUMP.(SINGLE STAGE DOUBLE SUCTION, FORGED CASE, SEALS OPTIONAL, DIFFUSER TYPE)

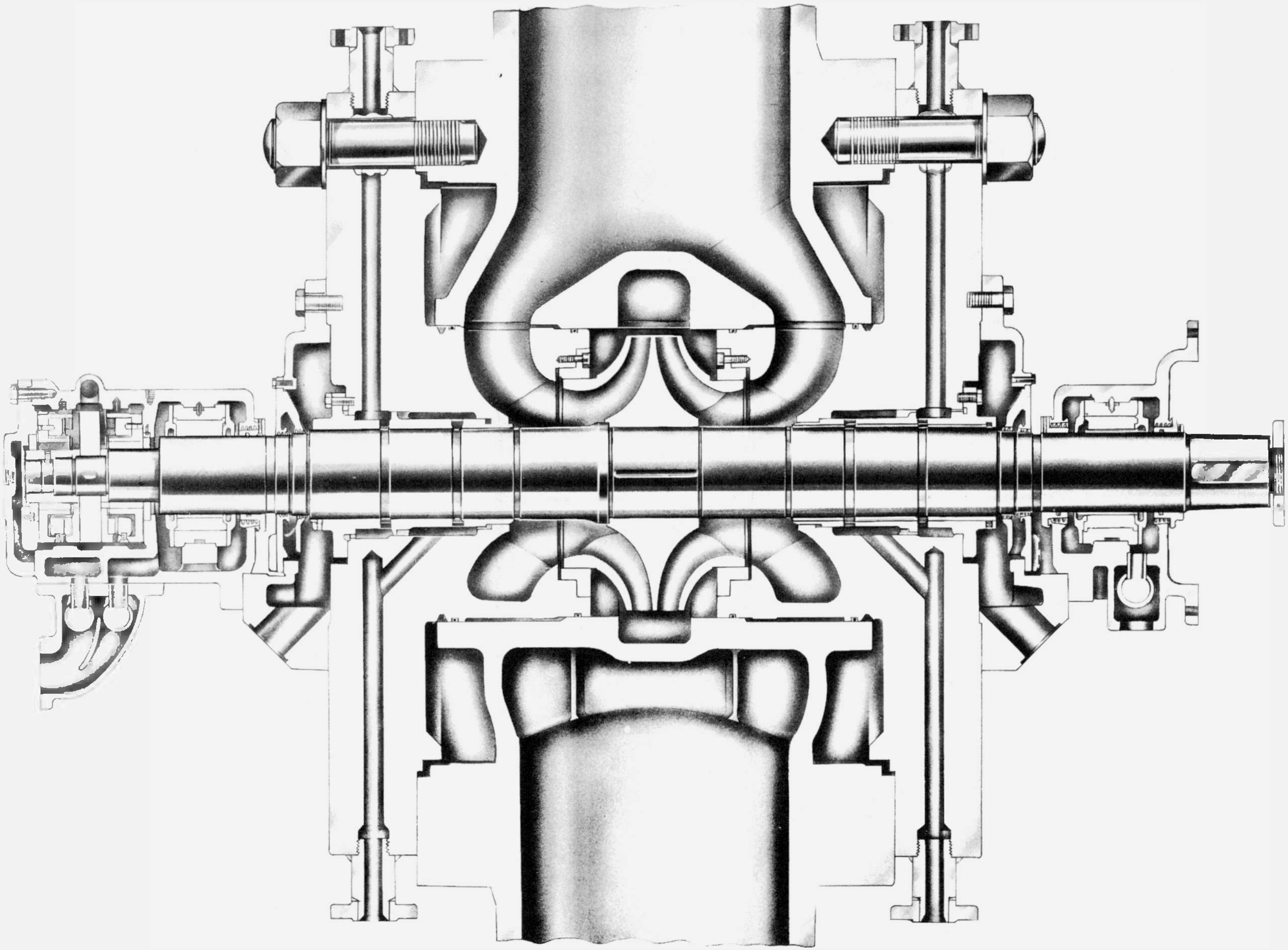


Figure 6. SINGLE STAGE REACTOR FEED / STEAM GENERATOR FEED PUMP
(Forged steel outer casing, double volute inner casing, double suction impeller, choice of shaft seals.)

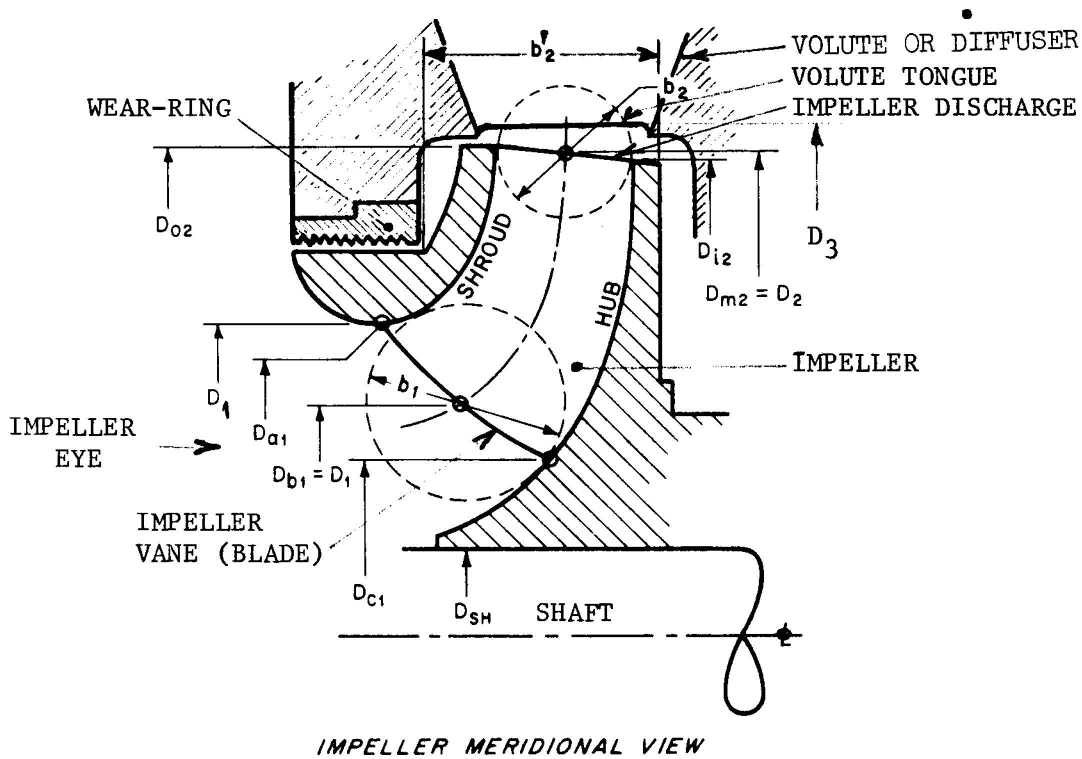


FIGURE 7: PUMP IMPELLER MERIDIONAL VIEW. (AN IMPELLER CAN BE MATED WITH A VOLUTE AS SHOWN ABOVE, OR WITH A DIFFUSER. BOTH DESIGNS ARE POPULAR)

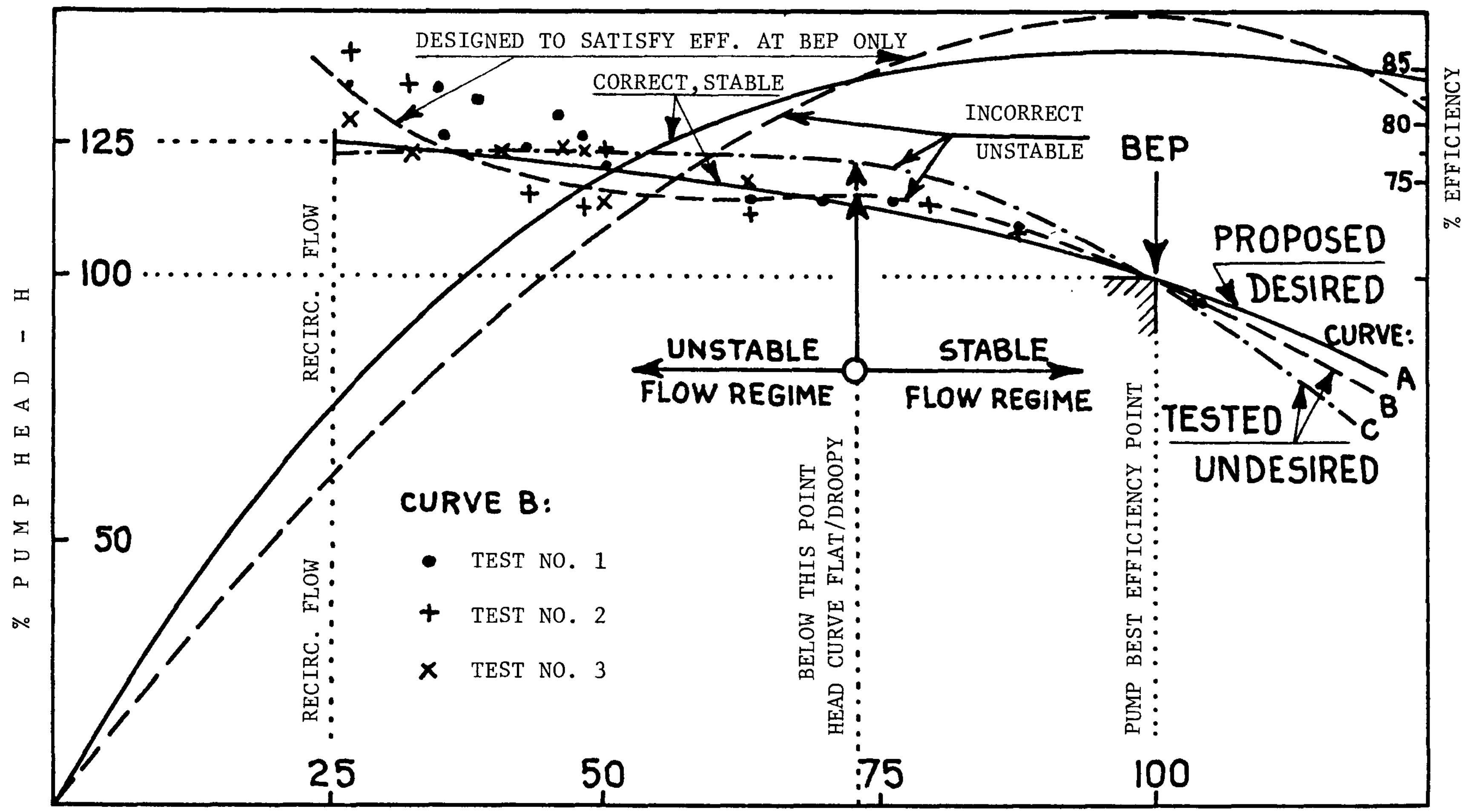


FIGURE 8: HEAD-CAPACITY CHARACTERISTICS OF MULTI-STAGE BOILER FEED PUMPS. CURVE "A" IS CORRECT AND DESIRED FOR STABLE SYSTEM OPERATION. CURVE "B" REPRESENTS A HYDRAULICALLY UNSTABLE IMPELLER-DIFFUSER DESIGN. PARALLEL AS WELL AS SINGLE PUMP OPERATION IS DIFFICULT IN THE UNSTABLE FLOW REGIME. CURVE "C" SHOWS A DESIGN WITH FLAT HEAD CURVE AT PART LOAD RESULTING IN MALFUNCTIONING OF THE CONTROL SYSTEM. SINGLE PUMP OPERATION IS POSSIBLE IN THE UNSTABLE REGIME. DESIGNS "B" AND "C" ARE NOT ACCEPTABLE FOR UTILITY APPLICATIONS.

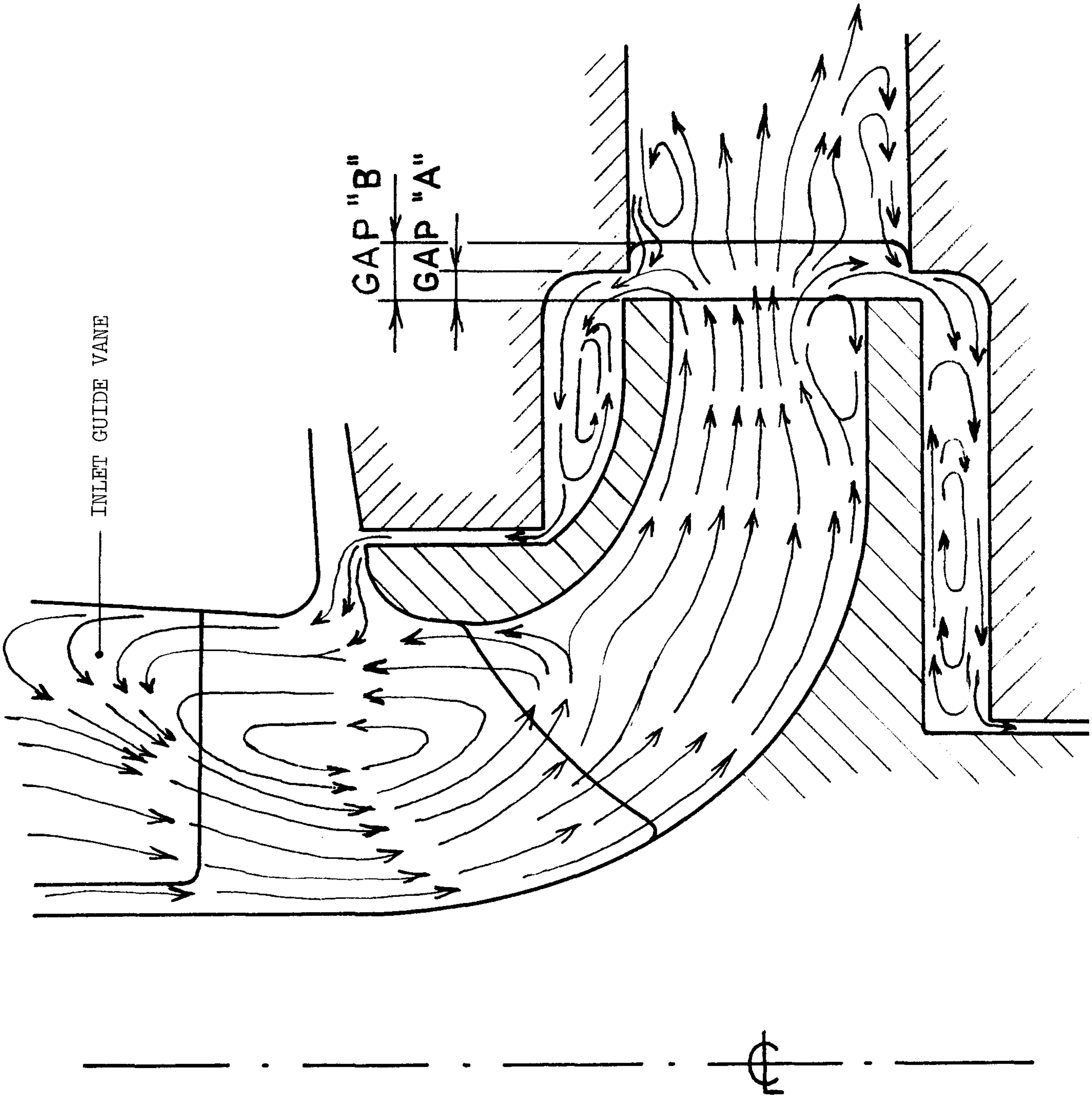
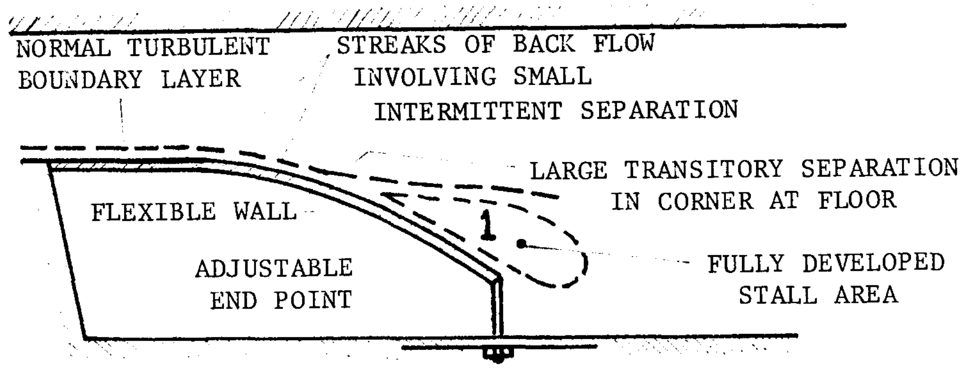
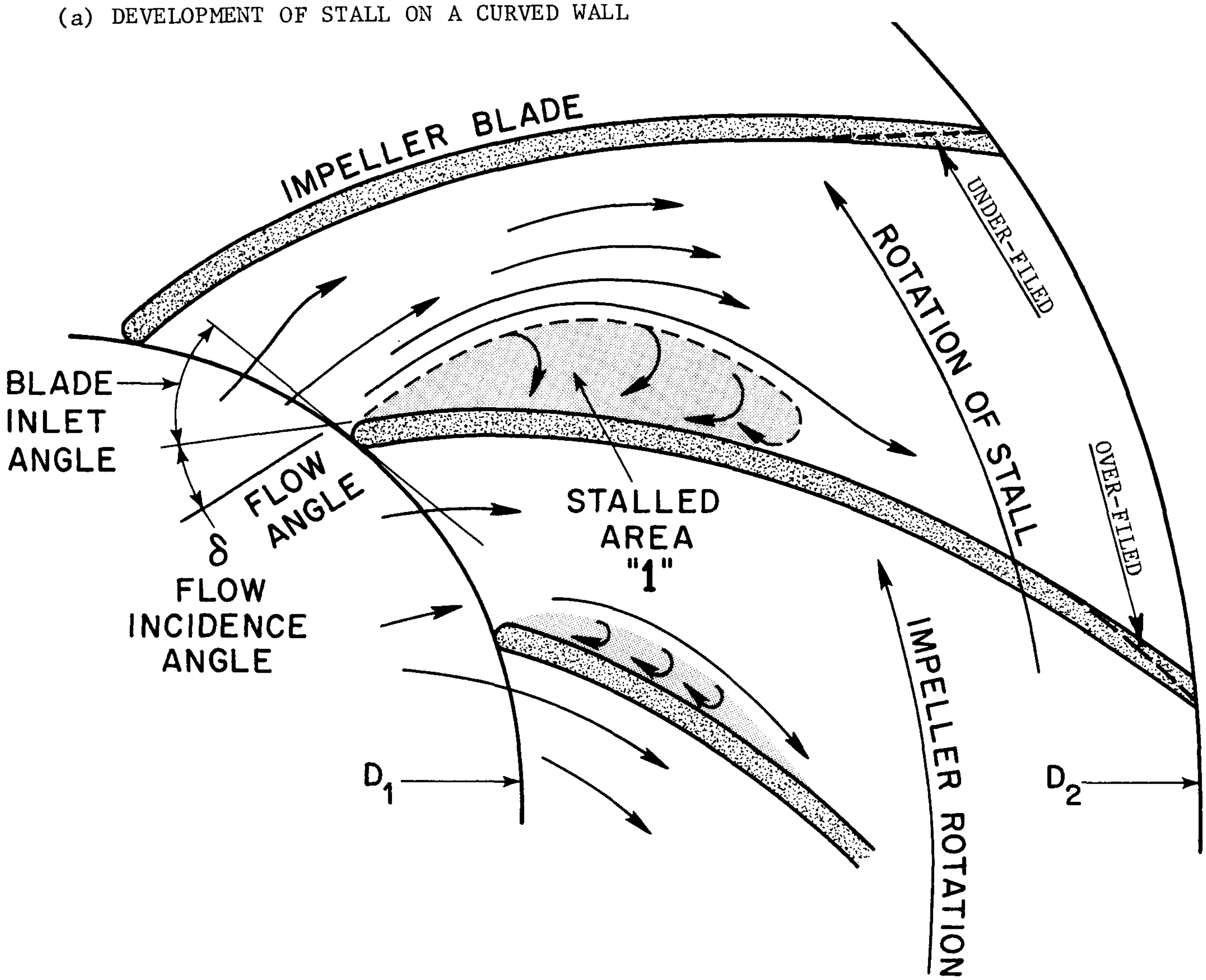


FIGURE 9: SECONDARY FLOW PATTERN IN AND AROUND A PUMP IMPELLER STAGE AT OFF-DESIGN FLOW OPERATION.



(a) DEVELOPMENT OF STALL ON A CURVED WALL



(b) FORMATION OF STALL IN AN IMPELLER EYE DUE TO FLOW INCIDENCE ANGLE
(VISUALIZED ON EXPERIMENTAL TEST RIG)

FIGURE 10: FORMATION OF STALL (a) ON A CURVED WALL, AND (b) IN AN IMPELLER EYE (ROTATING STALL)

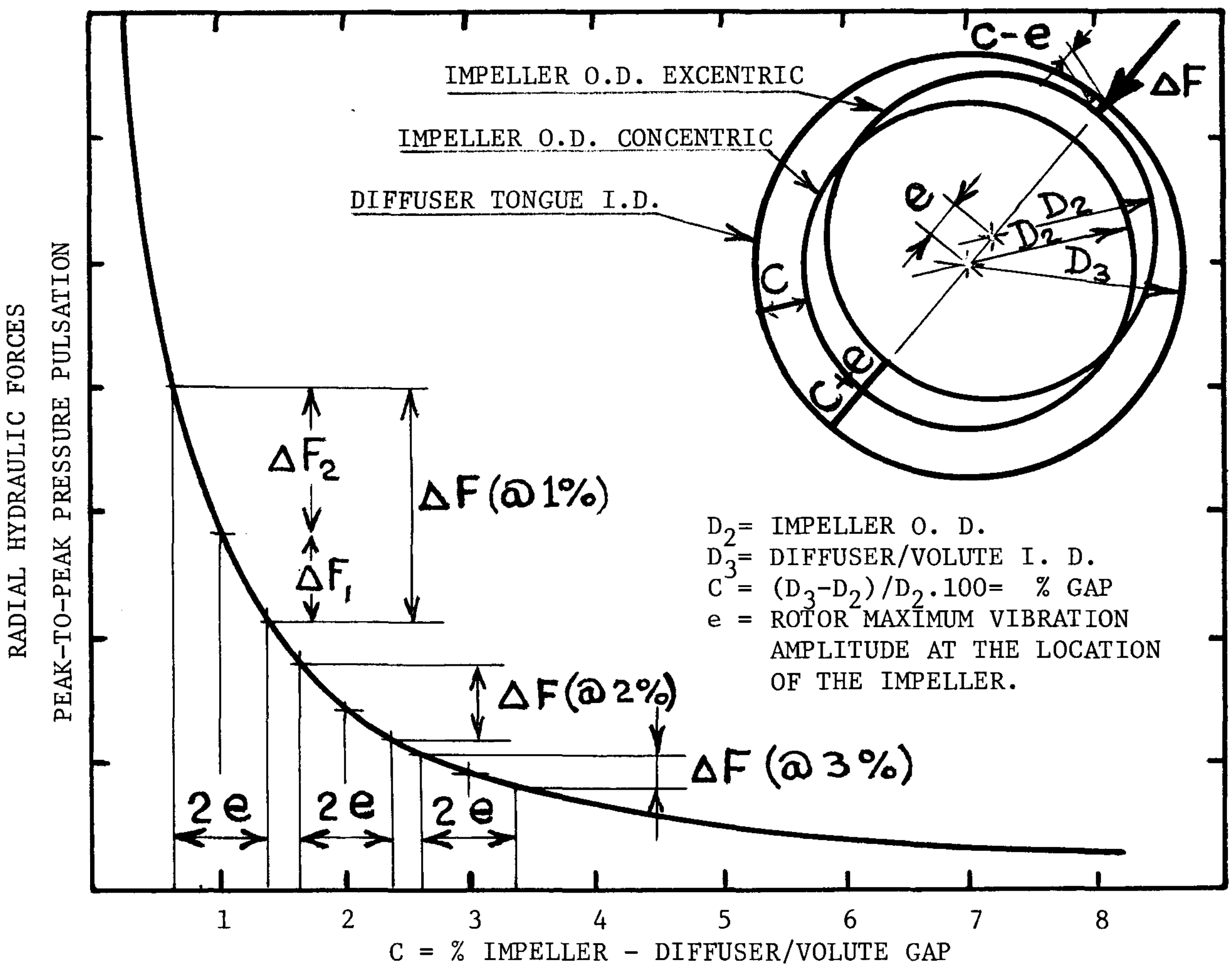


FIGURE 11: INFLUENCE OF PUMP IMPELLER TO DIFFUSER/VOLUTE RADIAL GAP ON PRESSURE PULSATION AT BLADE PASSING FREQUENCY AND ROTOR DEFLECTION CAUSED RADIAL FORCES.

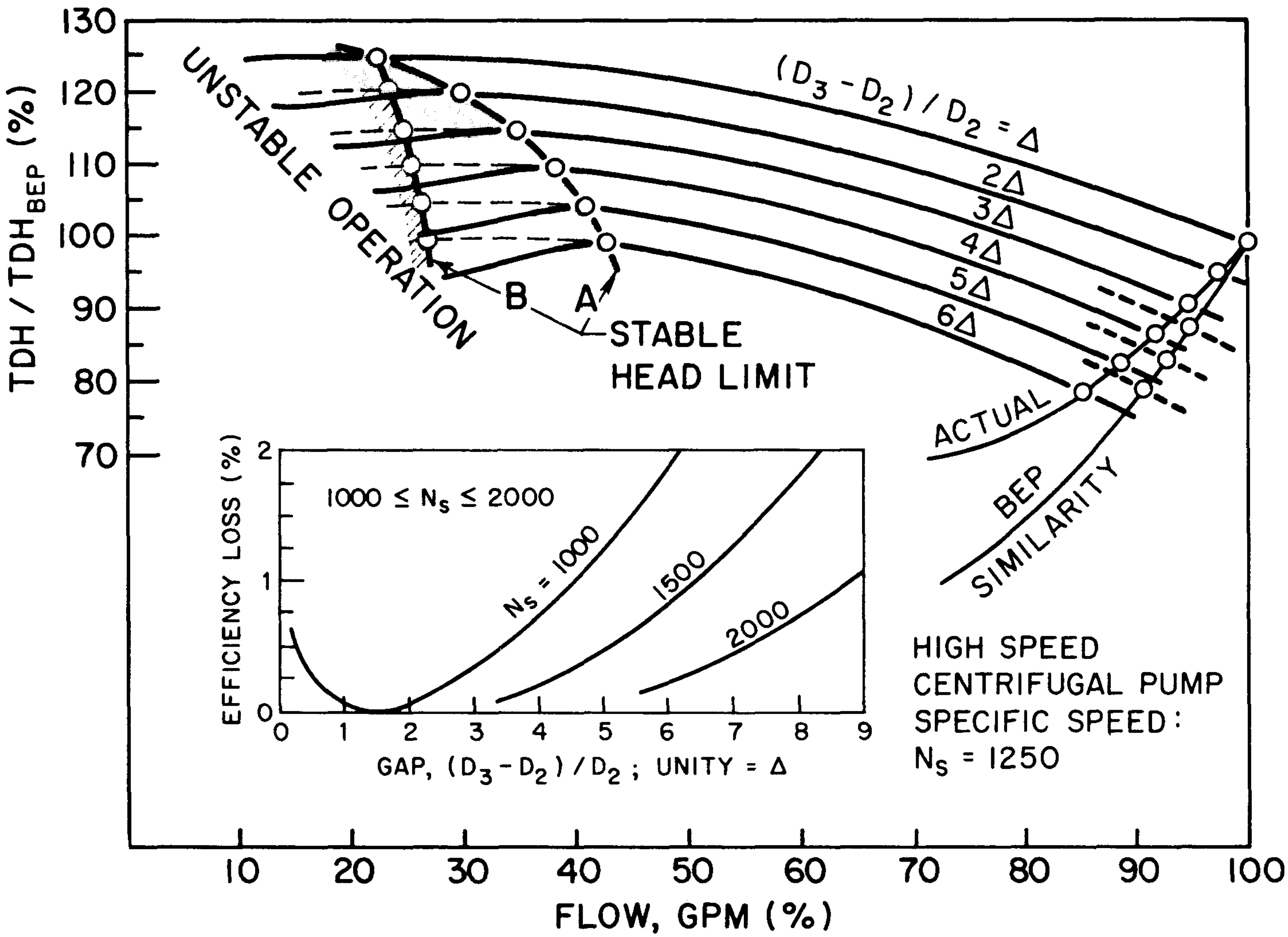
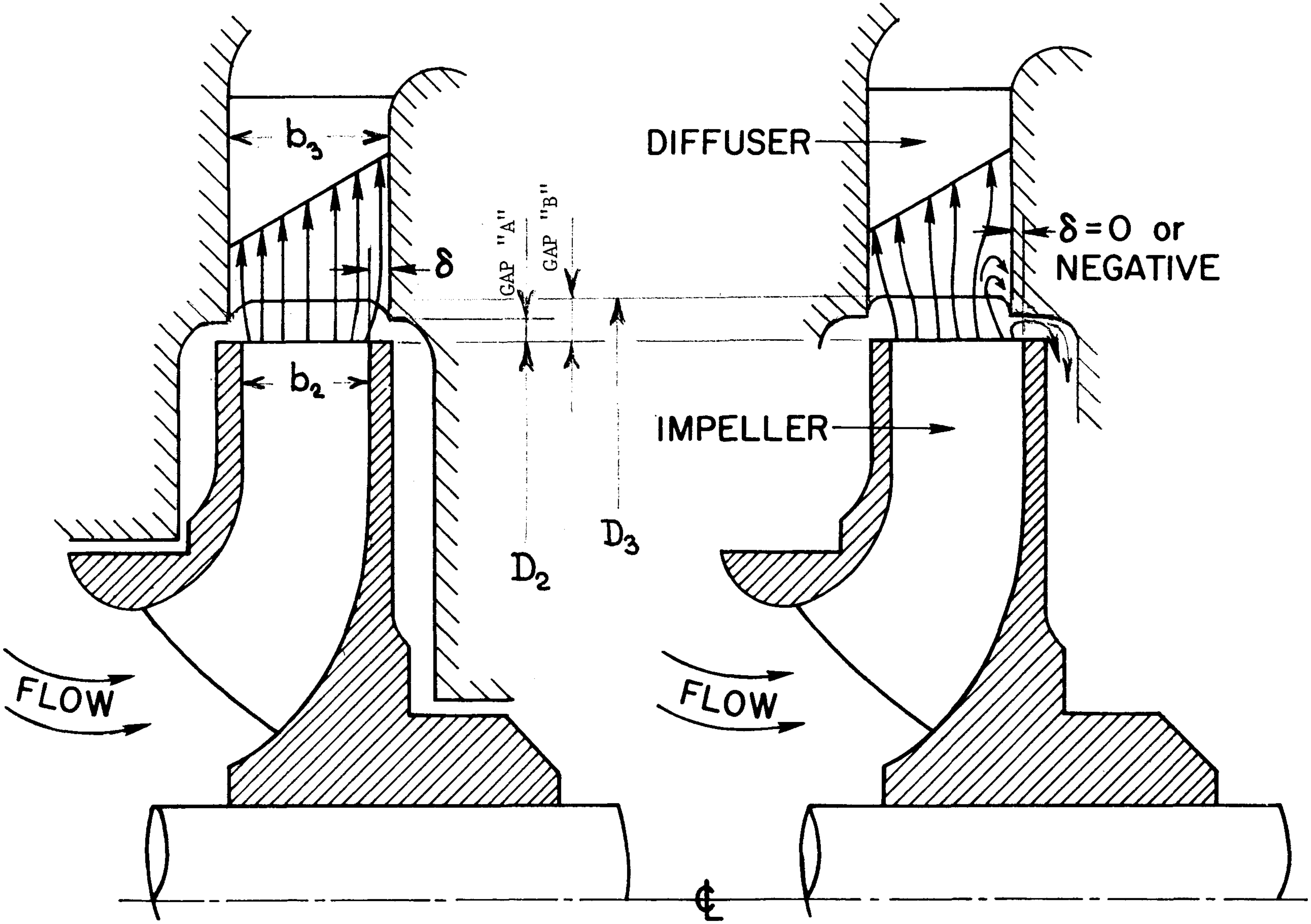


FIGURE 12: REDUCTION OF IMPELLER DIAMETER IN THE SAME CASING FOR A DIFFUSER PUMP STAGE. [IMPELLER DIAMETER IS REDUCED IN CASE (A), WHILE IN CASE (B) ONLY THE VANE GAP IS INCREASED. INFLUENCE ON EFFICIENCY IS ALSO SHOWN]



(a) CORRECT ALIGNMENT

(b) INCORRECT ALIGNMENT

FIGURE 13: IMPELLER - DIFFUSER RADIAL AND AXIAL GAPS, AND AXIAL ALIGNMENT OF HYDRAULIC CHANNELS.

BINGHAM
BYRON JACKSON
HITACHI
PACIFIC

DE LAVAL
WEIR, LTD.
WORTHINGTON

INGERSOLL RAND
KSB
SULZER
ALLIS CHALMERS

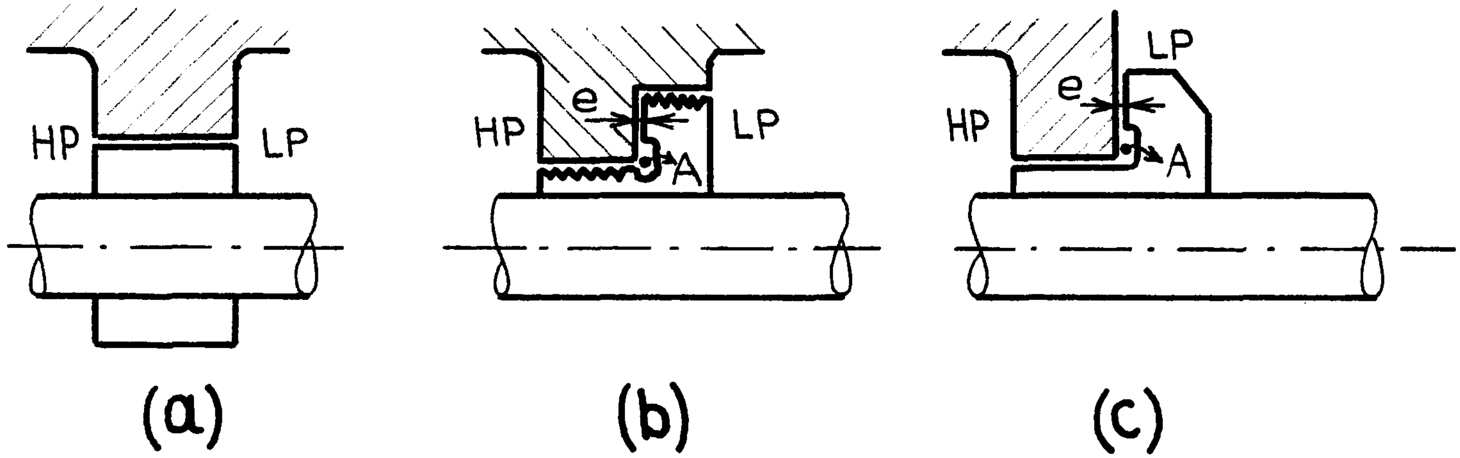
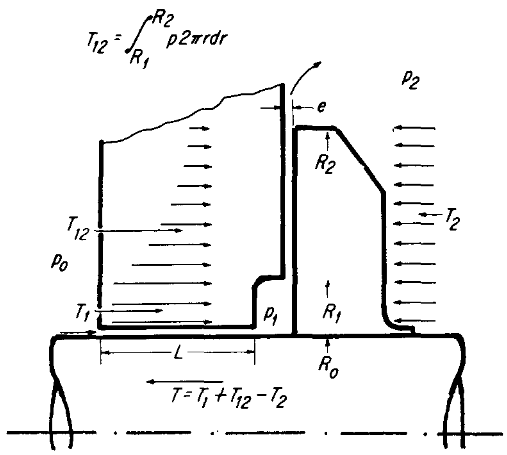
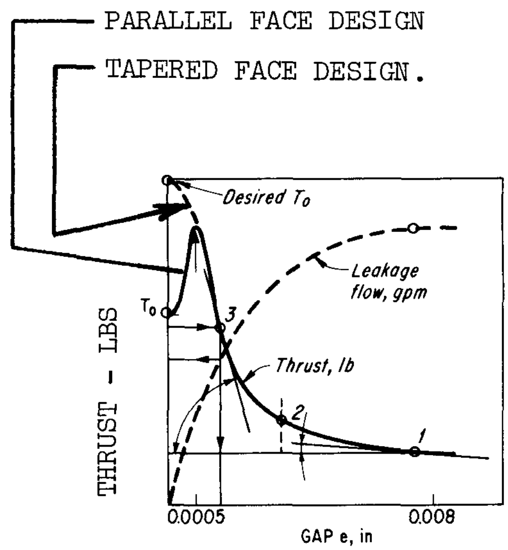


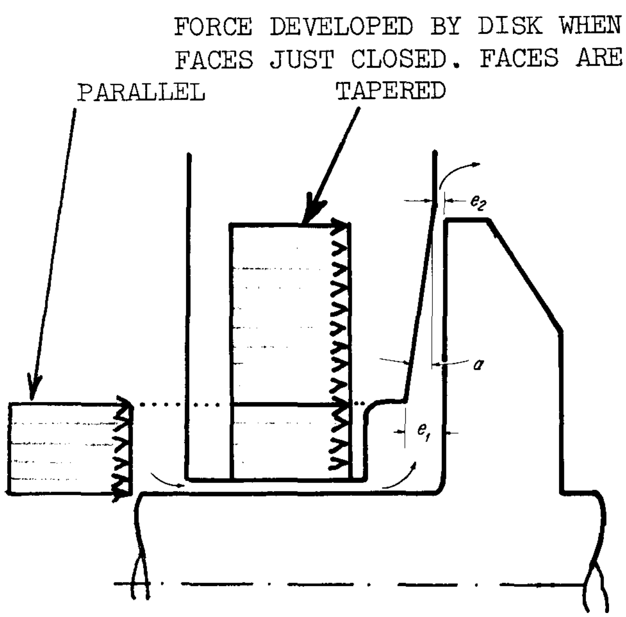
FIGURE 14: CUSTOMARY AXIAL BALANCING DEVICES FOR HIGH PRESSURE MULTI-STAGE BOILER FEED PUMPS: (a) BALANCE DRUM, (b) & (c) BALANCE DISK.



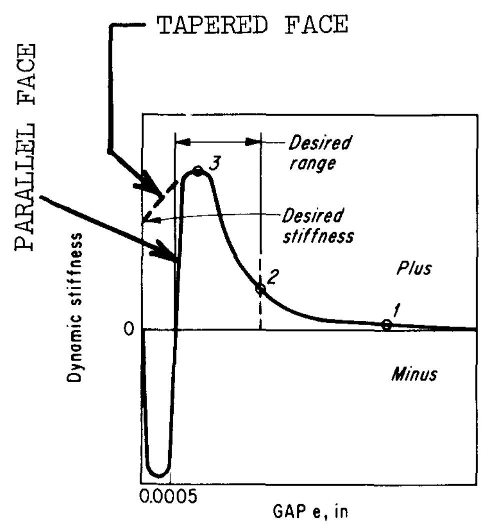
(a) BALANCE DISK WITH PARALLEL FACES. FORCE EQUILIBRIUM IS SHOWN WHEN DISK IN NORMAL OPEN POSITION.



(c) BALANCE DISK THRUST CARRYING CAPABILITY AND LEAK-OFF FLOW.

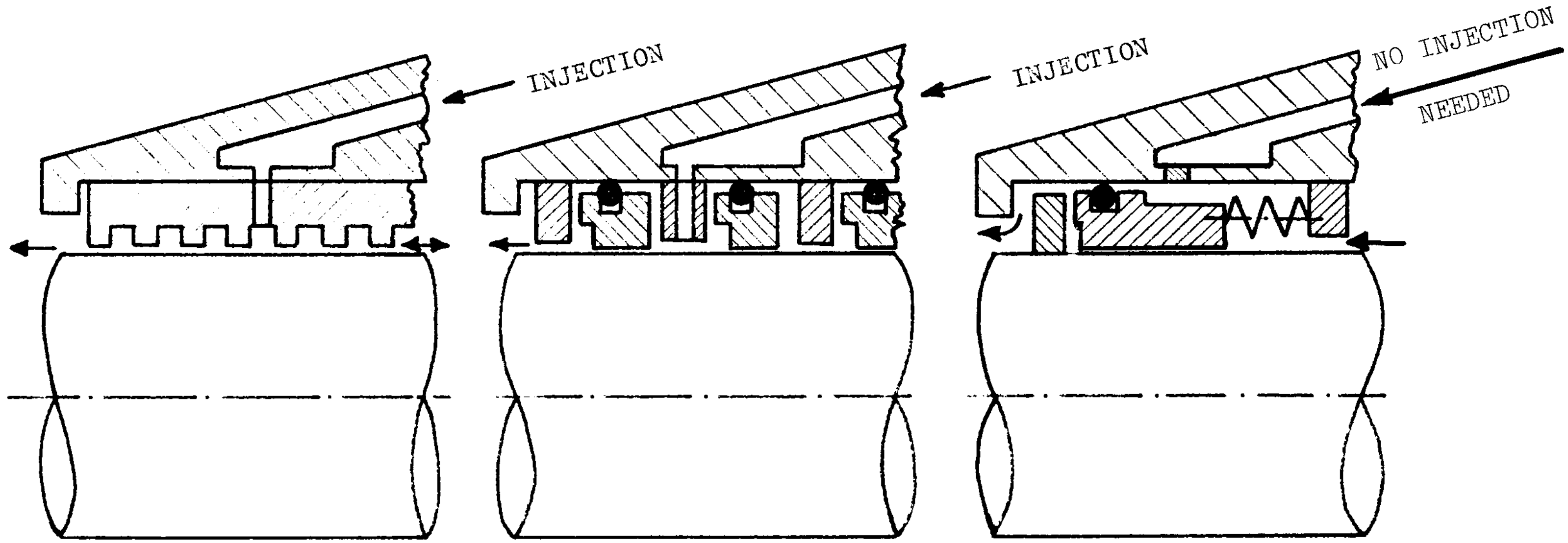


(b) TAPERED FACE BALANCE DISK



(d) DYNAMIC STIFFNESS OF BALANCE DISK.

FIGURE 15: PARALLEL AND TAPERED FACE BALANCE DISK DESIGNS.



(A) (B) (C)

FIGURE 16: SHAFT SEAL TYPES USED IN BOILER FEED, NUCLEAR FEED, AND FEED WATER BOOSTER PUMPS: (A) LABYRINTH, (B) FLOATING RING SEALS WITH SINGLE INJECTION, AND (C) MECHANICAL SEAL.

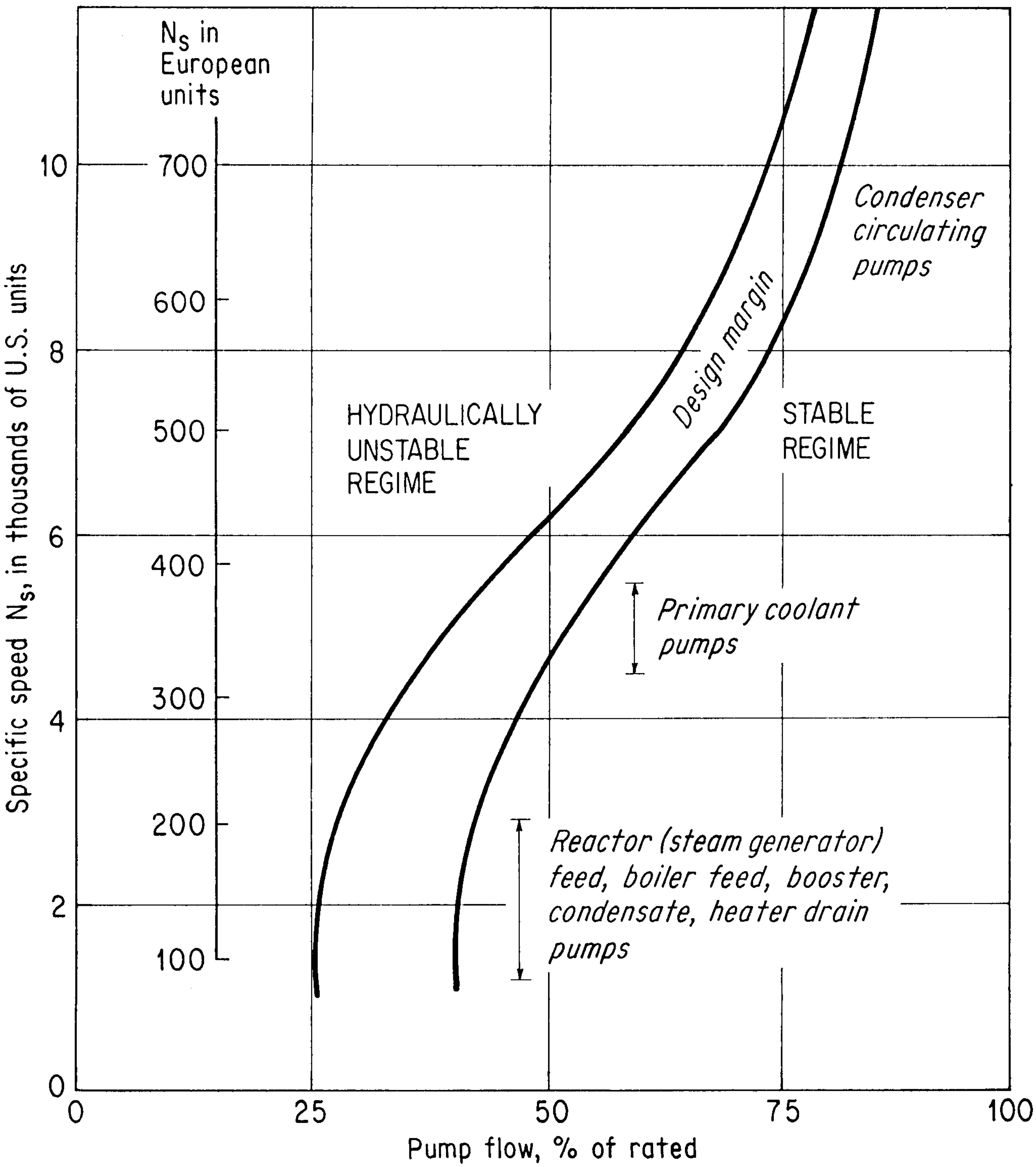


FIGURE 17: ANTICIPATED USEFUL OPERATING RANGES FOR PUMPS USED IN LARGE NUCLEAR AND FOSSIL POWER GENERATING UNITS.

(THE INNER LINE OF THE DESIGN MARGIN AREA IS PREFERRED, IF HYDRAULIC INSTABILITY OCCURS AT HIGHER FLOWS VARIOUS PUMP AND SYSTEM PROBLEMS CAN BE EXPECTED)

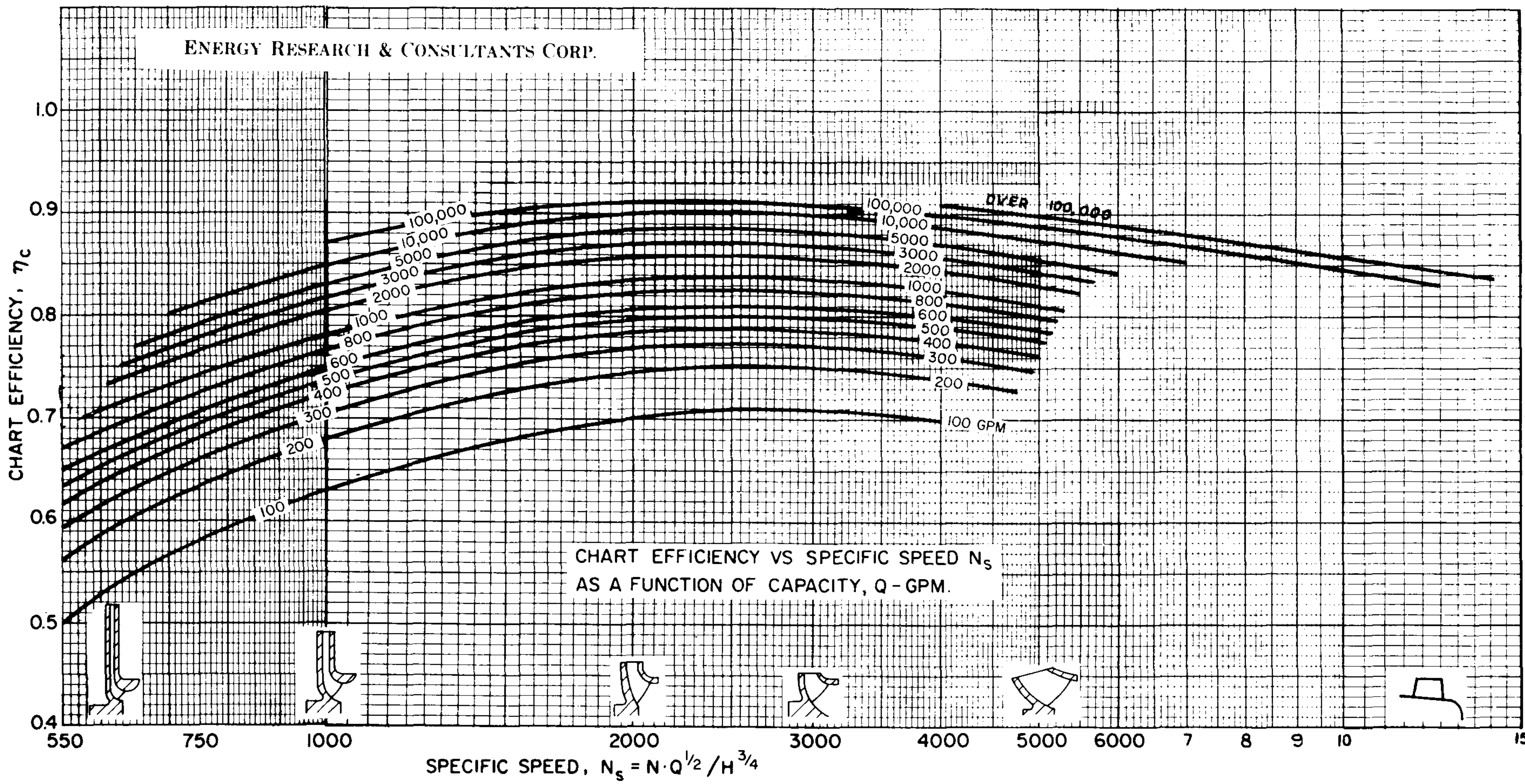


FIGURE 18: EXPECTED EFFICIENCIES OF UTILITY CENTRIFUGAL PUMPS (BASED ON EXPERIENCE) AS FUNCTION OF SPECIFIC SPEED AND PUMP DESIGN CAPACITY (GPM)

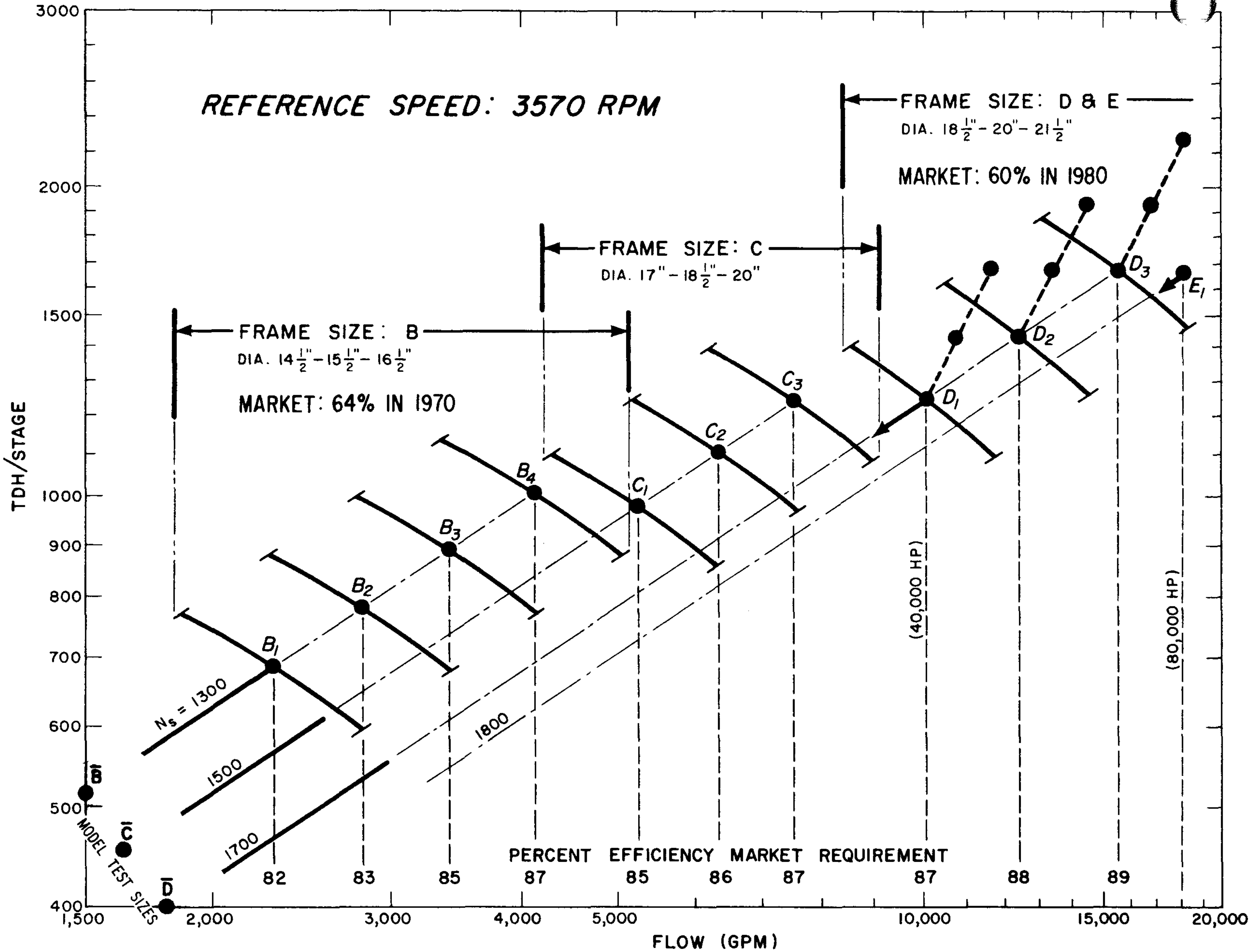


FIGURE 19: CUSTOMARY BOILER FEED PUMP RANGES. (REFERENCE SPEED IS 3570 RPM. A-FRAME NOT SHOWN, USED IN SMALLER THAN 500 MW UNITS AS MAIN FEED PUMPS, OR FOR LARGER UNITS AS A START-UP BOILER FEED PUMP)

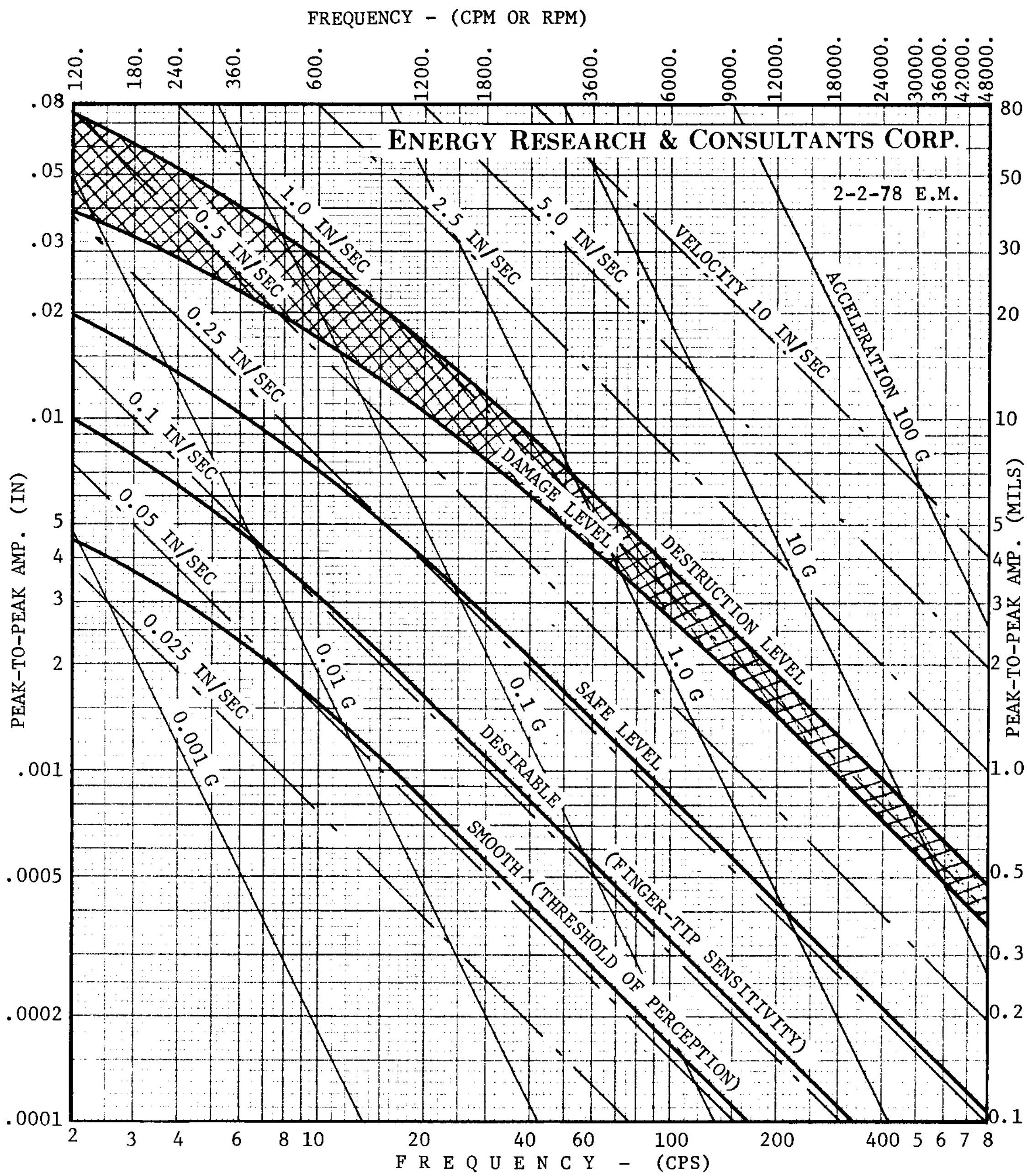


FIGURE 20: ALLOWABLE ROTOR VIBRATION LEVELS MEASURED RELATIVE TO THE BEARING CAP (VALUES SHOWN ARE FILTERED READINGS AT THAT PARTICULAR RPM OR FREQUENCY;EXPERIMENTAL STANDARDS).

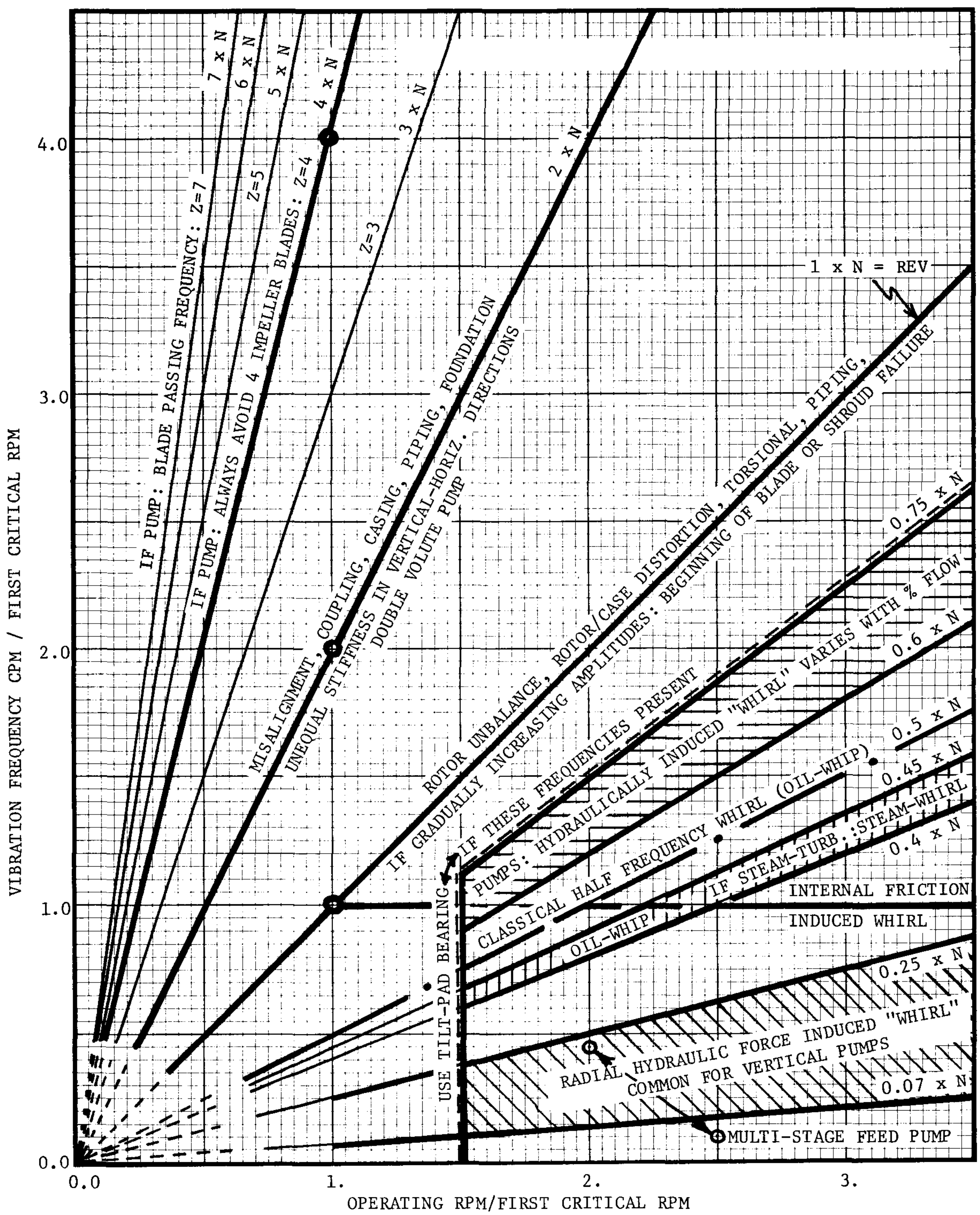


FIGURE 21: VIBRATION FREQUENCY - VS - SPEED. CAUSES AND CURES OF COMMON VIBRATION PROBLEMS OF CENTRIFUGAL PUMPS, STEAM & GAS TURBINES, FANS & BLOWERS, COMPRESSORS, AND OTHER CENTRIFUGAL EQUIPMENT.

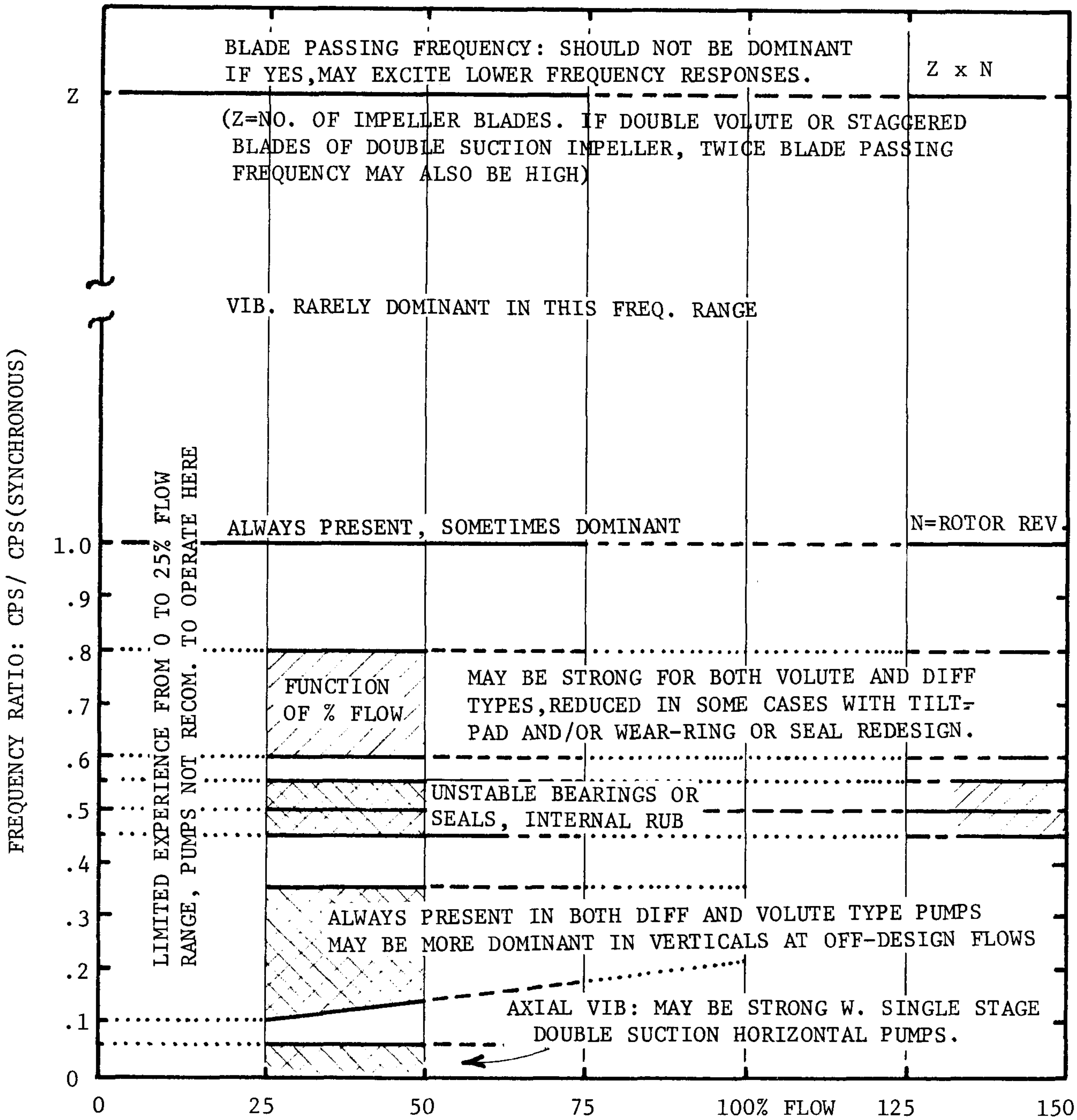


FIGURE 22: FREQUENCIES OF HYDRAULICALLY INDUCED DYNAMIC FORCES ACTING ON THE ROTOR OF A CENTRIFUGAL PUMP.

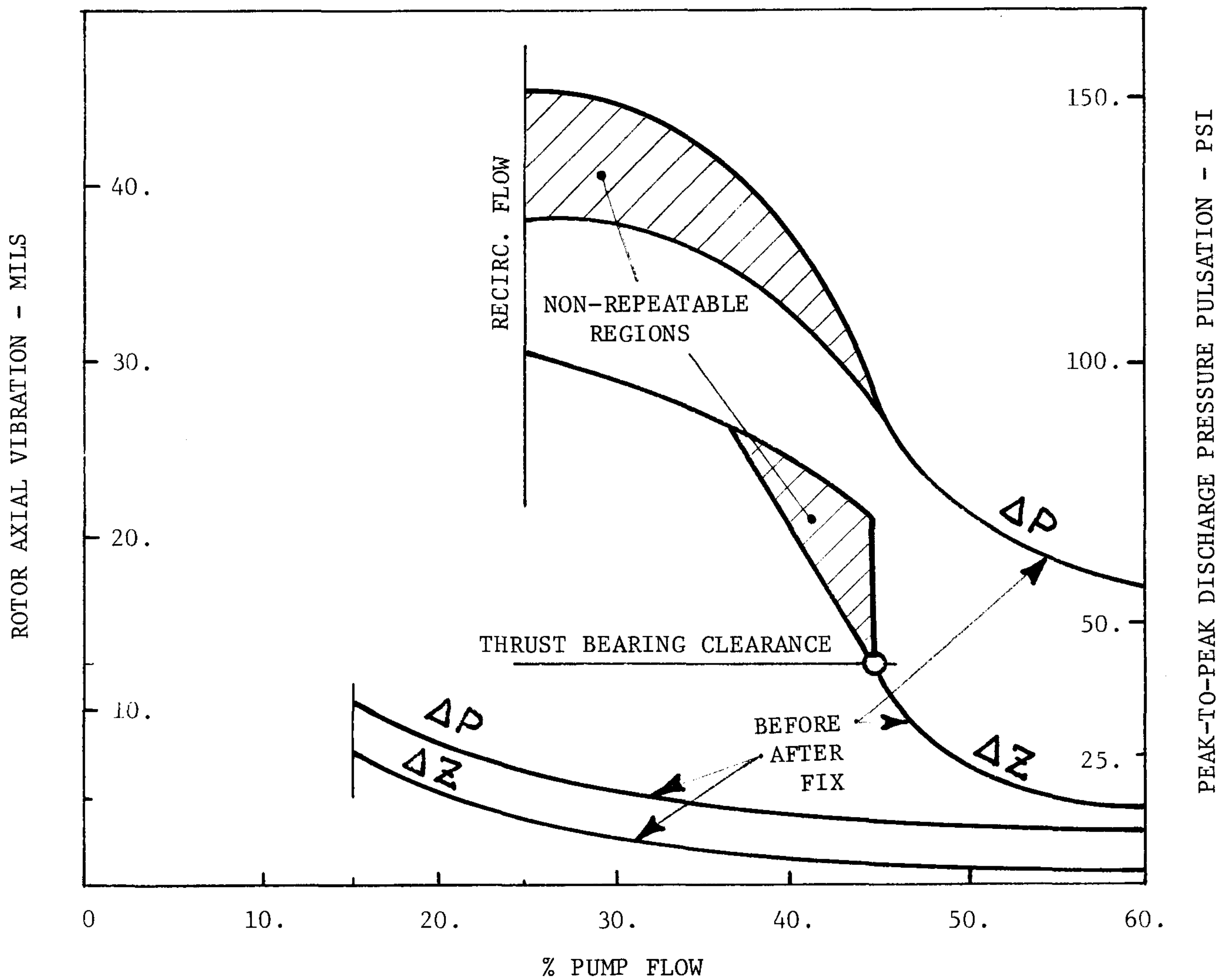
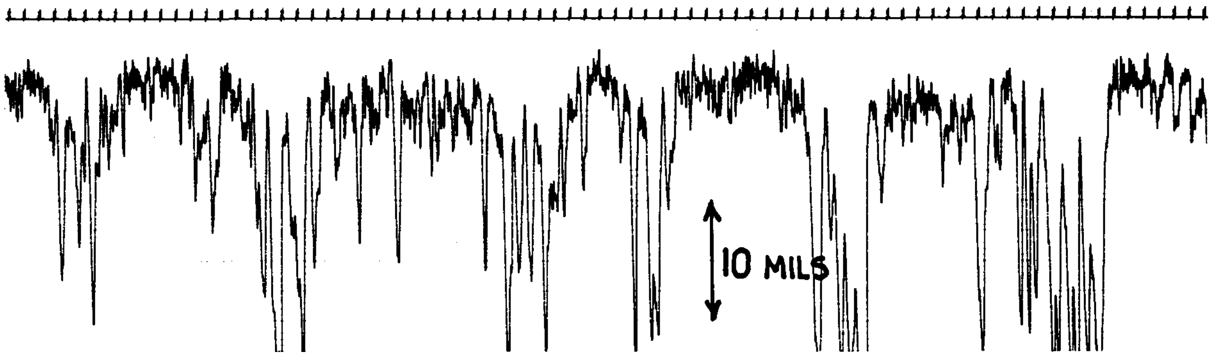
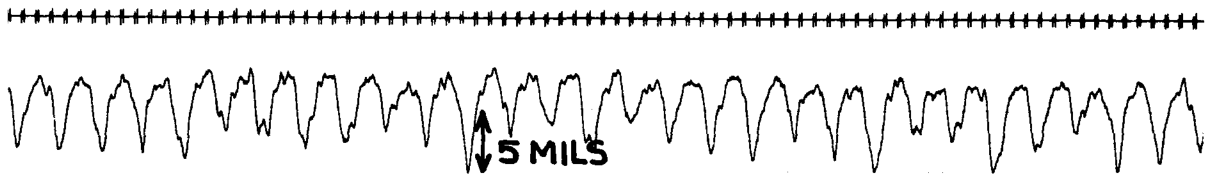


FIGURE 23: PEAK-TO-PEAK PRESSURE PULSATION MEASURED IN THE DISCHARGE NOZZLE, AND ROTOR AXIAL VIBRATION OF A HIGH SPEED SINGLE STAGE DOUBLE SUCTION NUCLEAR FEED PUMP WITH TWO DIFFERENT IMPELLER-TO-DIFFUSER GEOMETRY.



(a) MANUFACTURER RECOMMENDED RECIRC. FLOW



(b) RECIRC. FLOW INCREASED BY 10 %

(c) FINAL RECIRC FLOW, 20 % HIGHER THAN ORIGINAL

FIGURE 24: SHAFT AXIAL VIBRATION OF A HIGH SPEED, SINGLE STAGE, DOUBLE SUCTION NUCLEAR FEED PUMP AT MINIMUM FLOW OPERATION. (DIRECTLY APPLICABLE TO ANY SINGLE STAGE DOUBLE SUCTION CENTRIFUGAL PUMP, SUCH AS CONDENSATE BOOSTER, OR ANY CHEMICAL PROCESS PUMP)

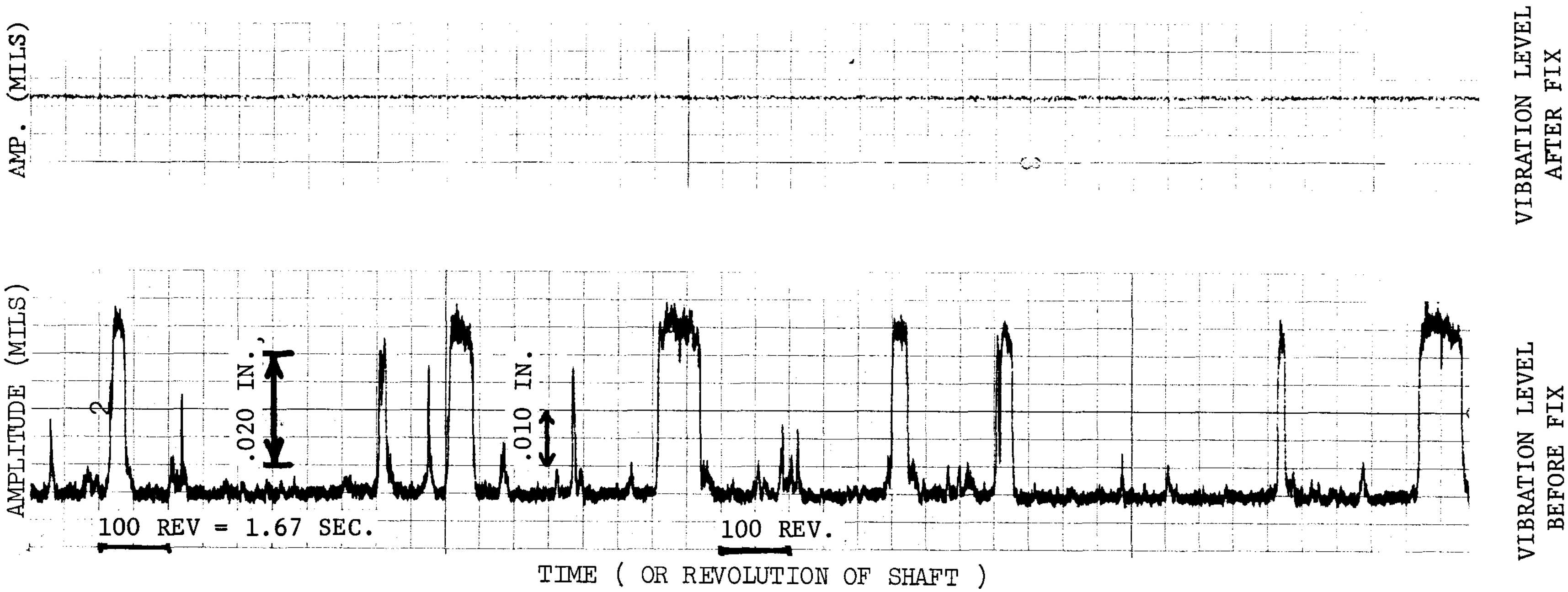
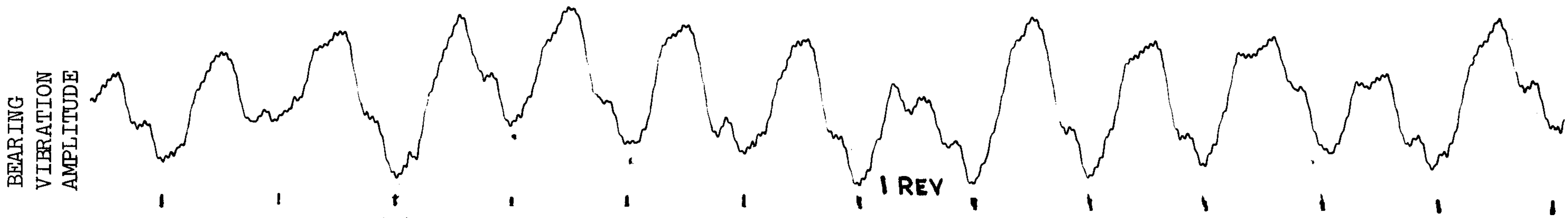
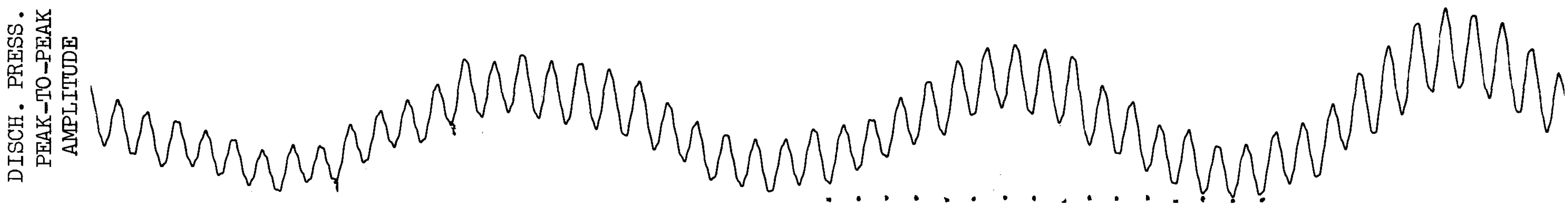


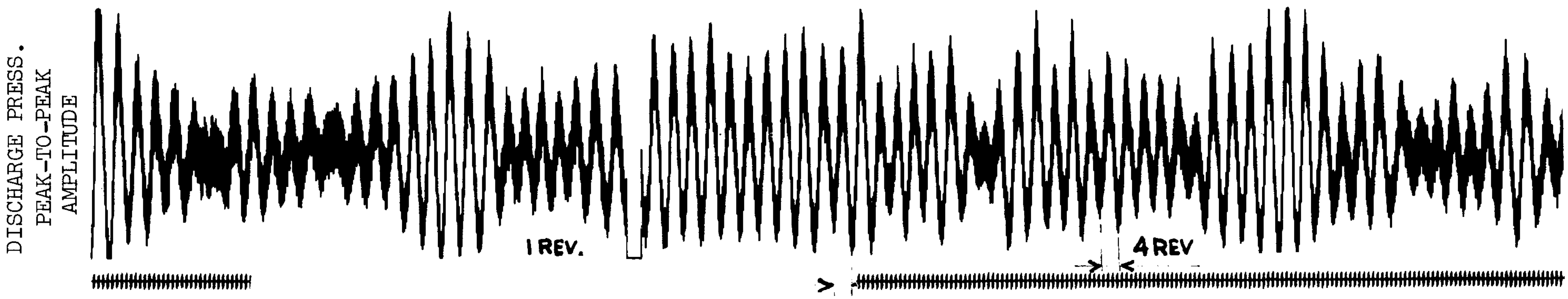
FIGURE 25: SINGLE STAGE DOUBLE SUCTION FEED PUMP SHAFT AXIAL MOTION CAUSED BY LOW FLOW HYDRAULIC INSTABILITY (AT ORIGINAL RECIRC FLOW, 25%)



(a) OUTBOARD BEARING RESPONSE. X-DIRECTION.



(b) PEAK-TO-PEAK DISCHARGE PRESSURE PULSATION AMPLITUDES - PSI.



(c) PEAK-TO-PEAK DISCHARGE PRESSURE PULSATION AMPLITUDES - PSI.

FIGURE 26: PRESSURE PULSATION AMPLITUDES MEASURED IN THE DISCHARGE NOZZLE OF A HIGH SPEED, SINGLE STAGE, DOUBLE SUCTION NUCLEAR FEED PUMP. (DATA PLAYED BACK WITH TWO DIFFERENT TAPE SPEEDS TO SHOW HIGH AND LOW FREQUENCIES. PUMP REV. AND OUTBOARD BEARING RESPONSE ARE ALSO SHOWN FOR COMPARISON. PUMP ON RECIRCULATION LINE)

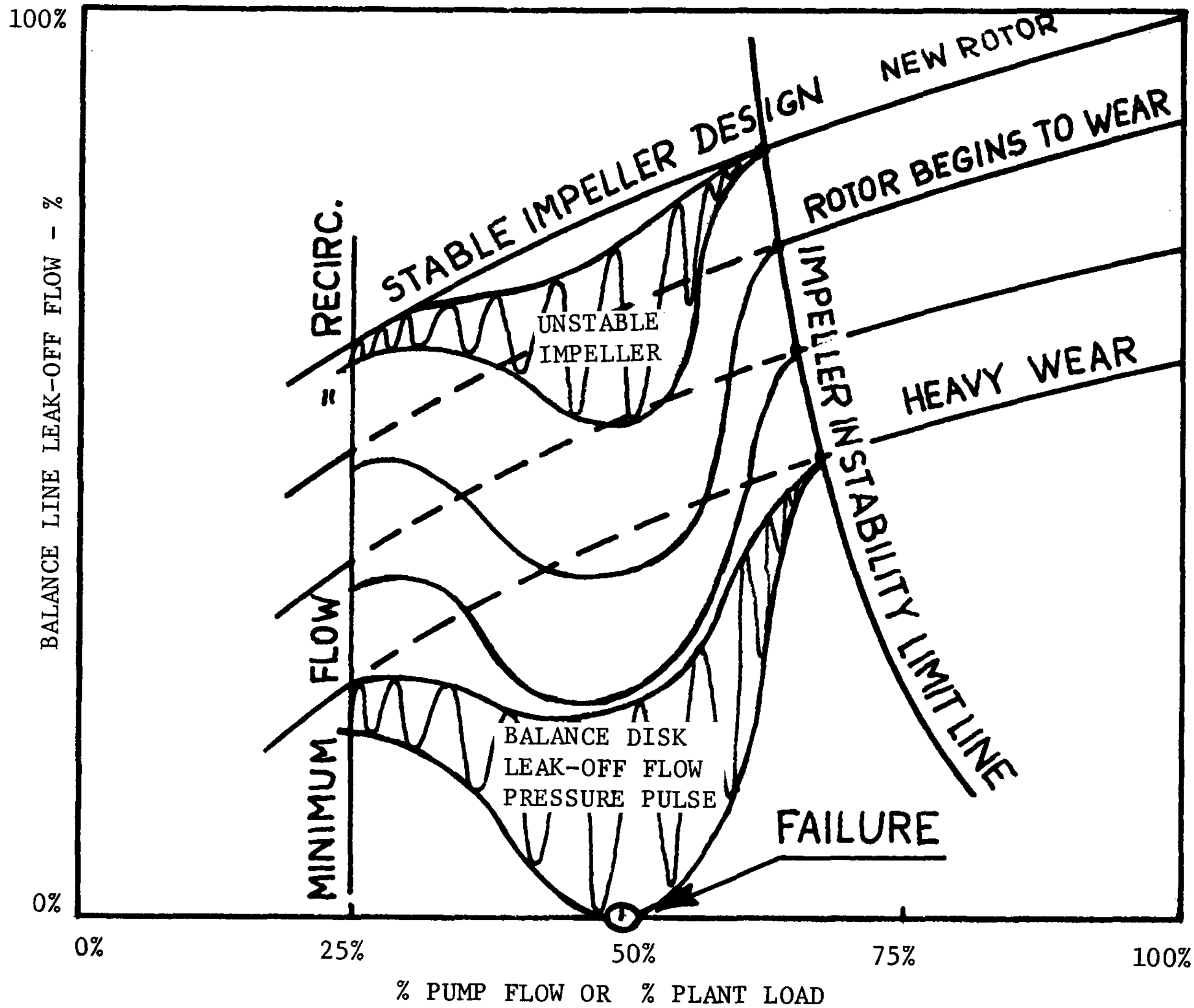


FIGURE 27: BALANCING DISK LEAK-OFF FLOW CHARACTERISTICS WITH STABLE AND UNSTABLE PUMP HYDRAULIC COMPONENTS.

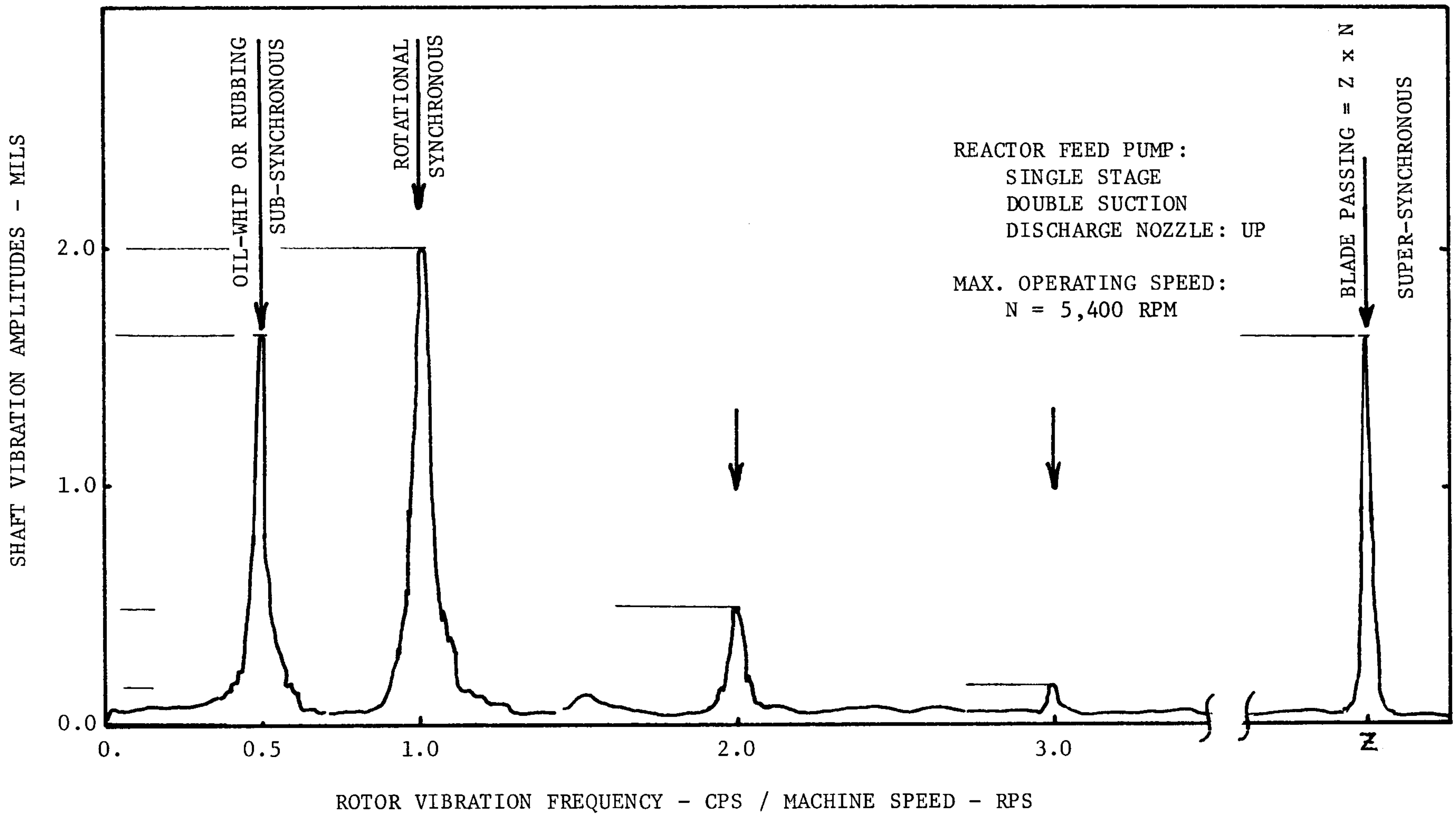
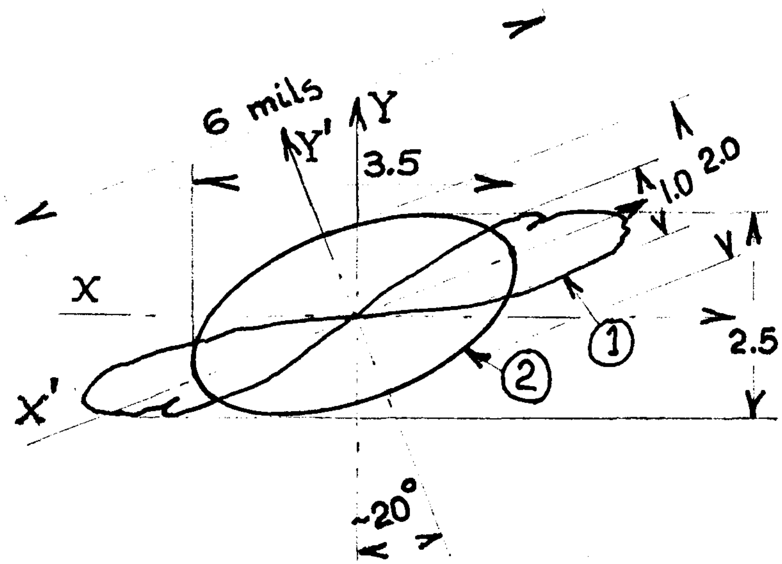
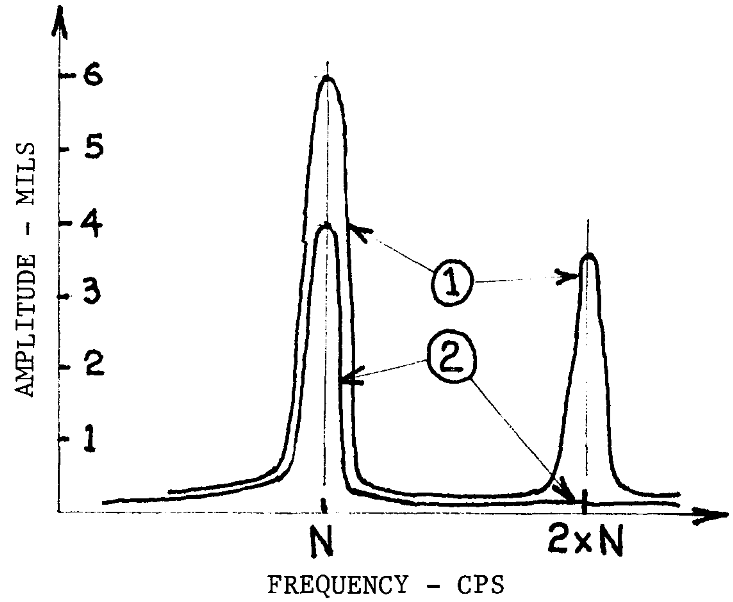


FIGURE 28: HYDRAULICALLY INDUCED INSTABILITY CAN DESTABILIZE THE JOURNAL BEARINGS RESULTING IN HALF-FREQUENCY WHIRL CALLED CLASSICAL "OIL-WHIP".



OSCILLOSCOPE OUTPUT WITH TWO PROBES



FREQUENCY ANALYZER OUTPUT: X'

FIGURE 29: ORBIT SHAPES MEASURED AT THE BOILER FEED PUMP TURBINE DRIVE INBOARD BEARING (CASE 1: BEFORE HOT ALIGNMENT, CASE 2: AFTER HOT ALIGNMENT AND WITH PROPERLY REDUCED BEARING CLEARANCE. NOTICE THAT ONE VERTICAL PROBE WOULD NOT SHOW ANY CHANGE IN VIBRATION AMPLITUDE, ONE PROBE 20° OFF VERTICAL WOULD SHOW THE OPPOSITE EFFECT. TWO PROBES PERPENDICULAR TO EACH OTHER, OSCILLOSCOPE AND FREQUENCY ANALYZER GIVE CLEAR PICTURE)

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VENDOR:____________

PUMP TYPE:__________

N= RPM
Q= GPM
H= FT
HP= BHP

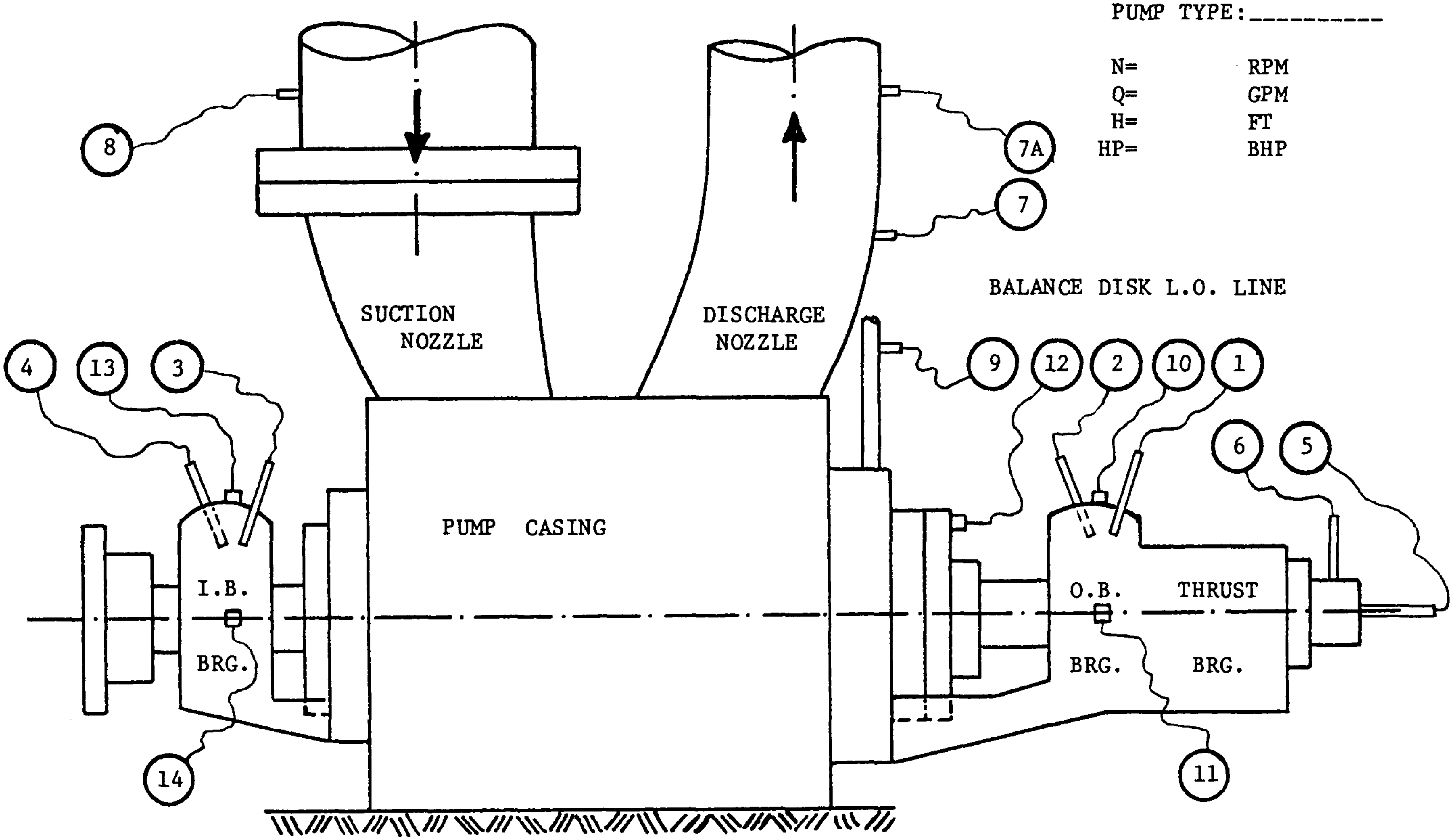

FIGURE 30: RECOMMENDED FEED PUMP INSTRUMENTATION FOR TROUBLE-SHOOTING OR WITNESS TESTING. THE NUMBER OF CHANNELS SHOWN IS 15: 6 PROXIMITY PROBES, NOS. 1 TO 6, 4 PRESSURE TRANSDUCERS, NOS. 7, 7-A, 8 AND 9, AND 5 ACCELEROMETERS, NOS. 10 TO 14. FOR STANDARD MEASUREMENTS THE NUMBER OF CHANNELS IS 14.

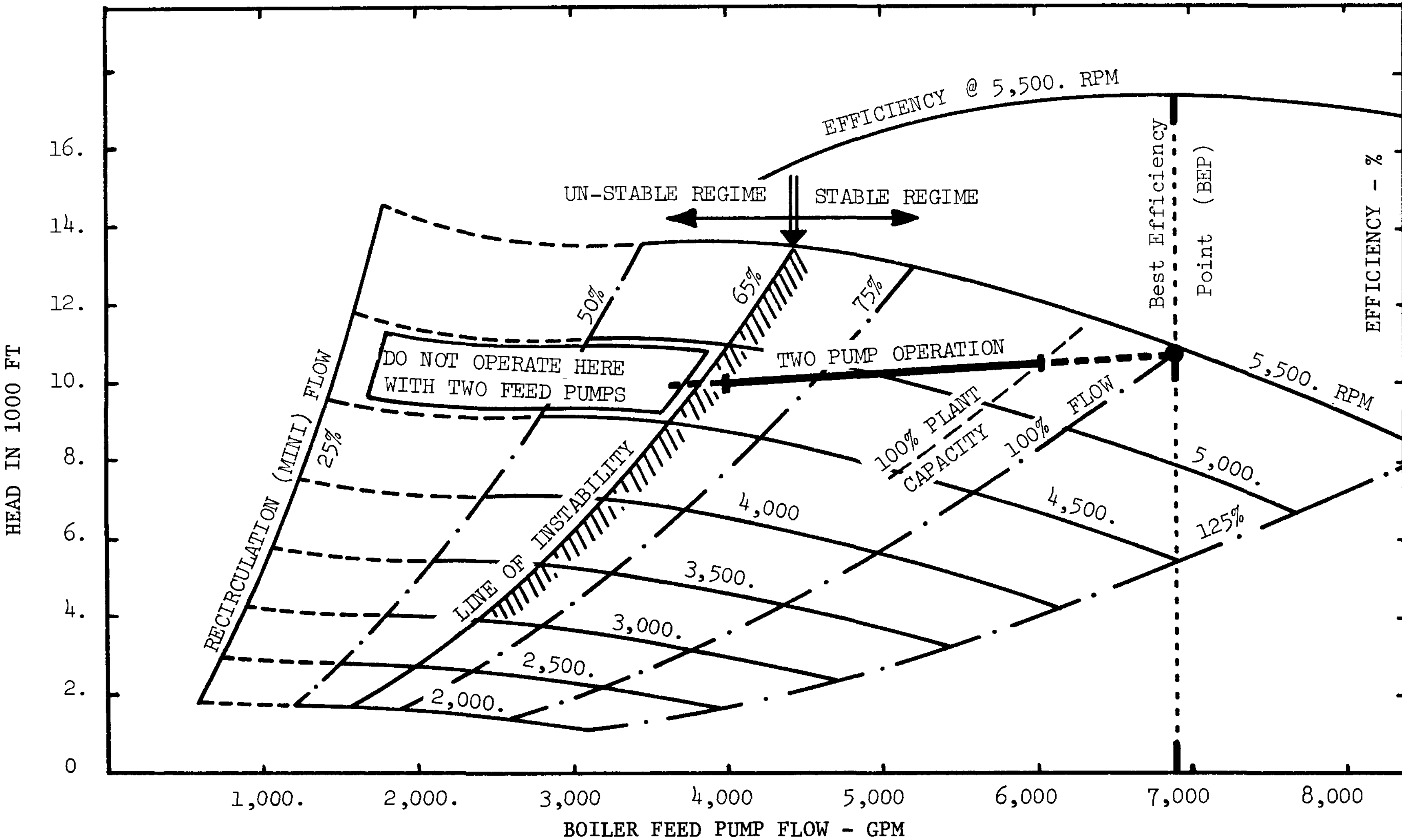

FIGURE 31: OPERATING RANGE RESTRICTION ON A HYDRAULICALLY UNSTABLE LARGE BOILER FEED PUMP.

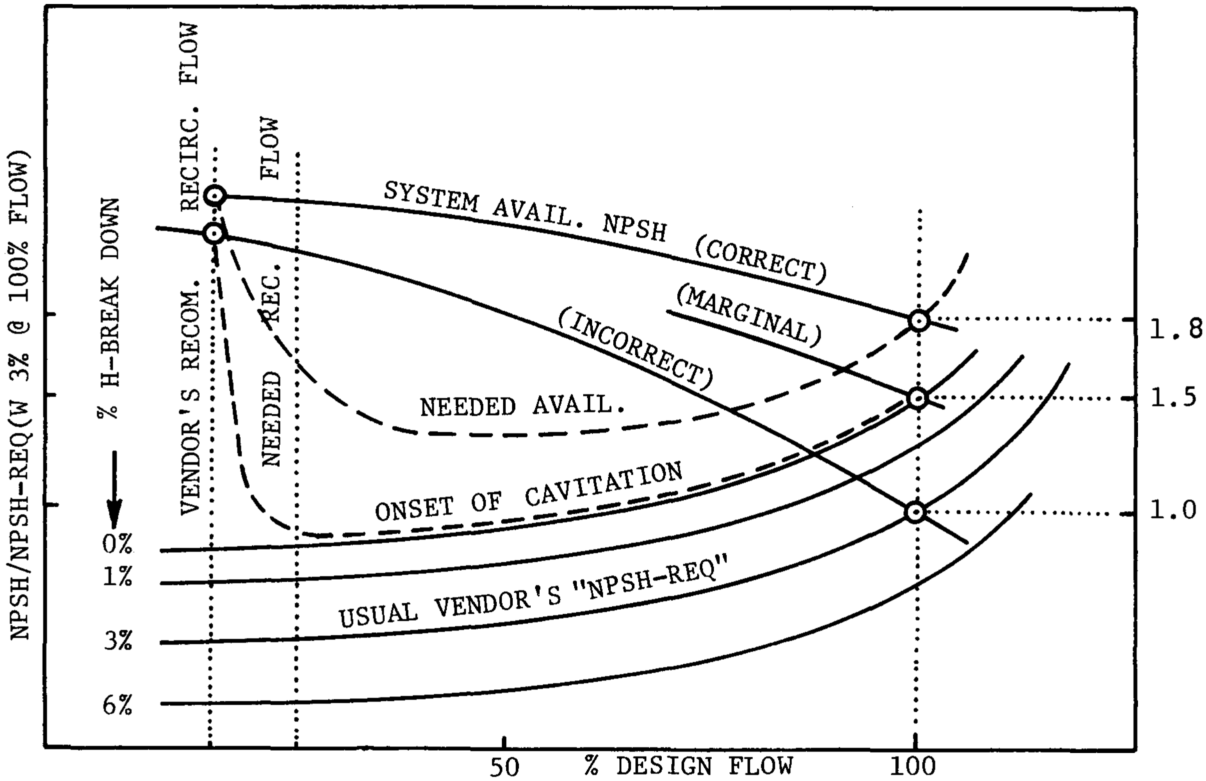

FIGURE 32: PUMP NPSH AVAILABLE AND REQUIRED AT A GIVEN SPEED.

(SYSTEM AVAILABLE NPSH SHOULD ALWAYS BE HIGHER THAN REQUIRED AT THE ONSET OF CAVITATION TO PROTECT THE BOILER FEED PUMP FROM CAVITATION DAMAGE)

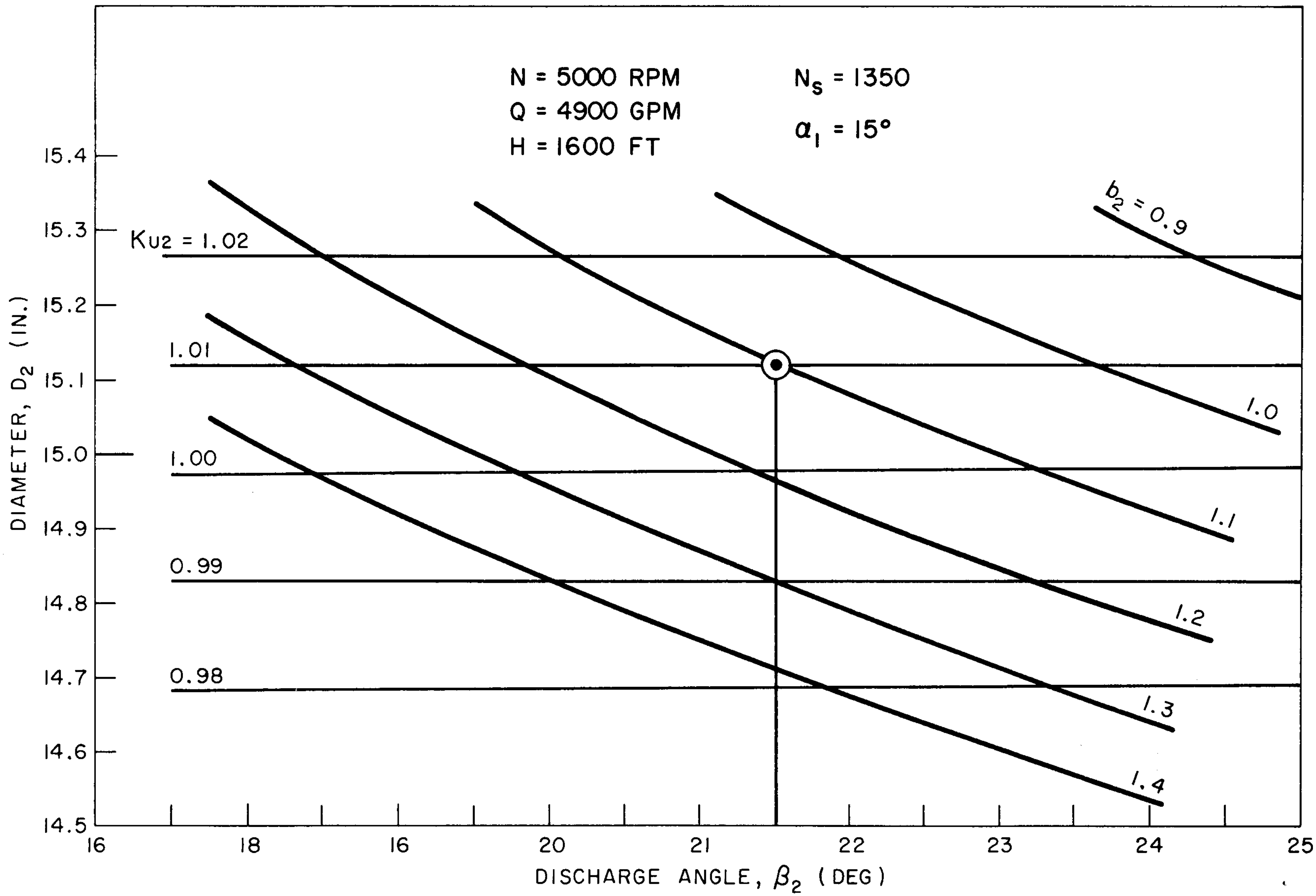

FIGURE 33: INFLUENCE OF IMPELLER BLADE DISCHARGE ANGLE AND EXIT WIDTH ON IMPELLER DIAMETER FOR A "B"-FRAME SIZE MULTI STAGE BOILER FEED PUMP.

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V: LIST OF FIGURES

VI: LIST OF SYMBOLS

VII: LIST OF REFERENCES

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LIST OF SYMBOLS

A/E	Architect Engineer
A_2	Impeller Exit area
A_3	Diffuser Throat Area
BEP	Best Efficiency Point
BFP	Boiler Feed Pump
BWR	Boiling Water Reactor
C_1	Impeller Inlet Velocity (Absolute), ft/sec
C_{1m}	Inlet Meridional Velocity Component, ft/sec
C_{1u}	Inlet Tangential Velocity Component, ft/sec
C_2	Impeller Exit Velocity (Absolute), ft/sec
C_{2m}	Exit Meridional Velocity Component, ft/sec
C_{2u}	Exit Tangentional Velocity Component, ft/sec
C'_{2u}	Actual Exit Tangentional Velocity Component, ft/sec
CPM	Cycle Per Minutes
CPS	Cycle Per Seconds
D_1	Impeller Inlet (Eye) Diameter, in
D_2	Impeller Exit Diameter, in
D_{SH}	Shaft Diameter at Impeller, in
FW	Feed Water
H	Head, ft
H_0	Shut-Off Head (at zero pump flow), ft
H_1	Head at Design Flow, ft
Hz	Hertz (CPS)
N	Machine Speed, rpm

LIST OF SYMBOLS (CONT.)

N_s	Stage Specific Speed ($=N.Q^{1/2}H^{-3/4}$)
NFP	Nuclear Feed Pump
NPSH	Net Positive Suction Head, ft
p	Pressure, psi
PCP	Primary Coolant (or Circulating) Pump
PWR	Pressurised Water Reactor
Q	Pump Flow, gpm
RFP	Reactor Feed Pump
RTA	Real Time Analyser
S_s	Suction Specific Speed ($=N.Q^{1/2}/NPSH^{3/4}$)
TDH	Total Dynamic Head, ft
U_1	Peripherial Velocity at D_1, ft/sec
U_2	Peripherial Velocity at D_2, ft/sec
W_1	Relative Flow Velocity at D_1, ft/sec
W_2	Relative Flow Velocity at D_2, ft/sec
Y	Area Ratio ($=A_2/A_3$)
Z	Number of Impeller Vanes
Z'	Number of Diffuser Vanes

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