

REALISTIC DESIGN SPECIFICATIONS INCREASE PUMP RELIABILITY

by

Thomas R. Morton

Manager of Field Engineering

and

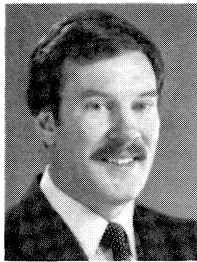
Gary L. Olin

Chief Engineer

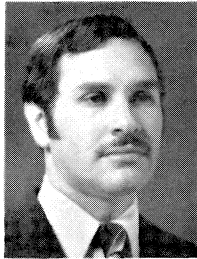
Bingham-Willamette Company

Portland, Oregon

Thomas R. Morton was educated at Stow College, Glasgow, Scotland where he received his national certificate in Mechanical Engineering. After coming to the United States in 1962, he was employed as a senior design engineer with Borg-Warner Corporation. He joined Bingham-Willamette at the Vancouver plant. He returned to Portland in 1980 to assume his present position. Among the assignments he is responsible for is the company's Reliability and Maintainability Program and the resolution of field technical problems.



Gary L. Olin, chief engineer, Bingham-Willamette Company, graduated with a degree in Mechanical Engineering from Oregon Polytechnic Institute in 1965. After graduation he rejoined the Bingham Pump Division where he had been employed during summers while attending college. Design engineering responsibilities over the years included: group leader, section leader and assistant chief engineer for horizontal specialty pumps. In January of 1990, he was promoted to his current position as chief engineer. He is a member of the American Society of Mechanical Engineers.



case studies. Field experience can provide a basis for establishing realistic design requirements toward obtaining optimum pump performance and system reliability. In addition, this paper will review the corrective action required to minimize these problems through optimum design specification.

INTRODUCTION

Technical specifications are the means by which the architect/engineer or user transmits his requirements to the manufacturer of sophisticated mechanical equipment — in this instance, the pump manufacturer. Unless these specifications are complete and define accurately all design criteria and operating modes, the pump designer can misinterpret, or remain ignorant through lack of detail, those aspects of the pump system that affect pump reliability and performance. In the cases presented, failure to properly define the pump operating modes caused major pump outages resulting in costly modifications to the pump and pumping system.

The basis of the original pump selections will be investigated and the design philosophy and operating prediction discussed. It will be evident through field engineering evaluation and in-depth factory testing that the original equipment was inadequate for the job due primarily to the lack of definitive operating conditions in the specifications. Assessment of the pump application was made by the equipment designer and the user after the initial outages which led to a more complete understanding of the equipment requirement. Through this mutual interchange of information, an equitable solution was reached which satisfied the application without undue compromise to the pump design.

A guide to the requirements of technical specifications will be outlined later in this paper with special emphasis on design parameters and testing requirements. It is hoped that this section will provide a basis for architect/engineers and users to communicate their requirements to pump manufacturers.

It is not the intention of the authors to imply that all technical specifications are inadequate; many good specifications exist for centrifugal pumps. It is more the intention to demonstrate through actual field cases where specification improvement and operation detail can provide the designers with sufficient insight to provide equipment with maximum reliability for the service.

CASE I

Review

Specifications were furnished to the pump manufacturer in the proposal stages of this contract which indicated that the

ABSTRACT

Many design specifications are incomplete relative to pumping system operation and can inadvertently place unrealistic restrictions on the pump designer. Meaningful cooperation between pump suppliers, architect/engineers (A.E.), and users can result in specifications being written to meet the customer's requirement and also provide design parameters to the manufacturer which will allow production of a machine with optimum reliability and operating life.

Pressure pulsation levels and vibration amplitudes are usually lowest when centrifugal pumps operate at capacities near the best efficiency point. On the other hand, pulsation and vibration are generally higher as the pump operates at "off peak" capacities. These conditions are due to hydraulic instability resulting from secondary flows and separation, with resultant stall in the hydraulic passages.

This paper, which will study hydraulic instability and the phenomenon of low flow recirculation, is based on actual field

pumps were essential to plant operation, including accident conditions, and would be of a design which would permit continuous operation under all conditions of operation from zero flow to design flow without any undue strain or damage to any part of the pumping units. Pump characteristic curves showing the pump head (ft.), efficiency (%), NPSH (ft.) and power (BHP) plotted against capacity (GPM) were to be provided and the curves extended to indicate pump characteristics out to 130% of design capacity.

Each pump would be designed for the following conditions:

Service: Service Water Pumps

Temperature (°F): 32-95

Capacity (GPM): 12,500

Net Developed Head (ft.): 295

Maximum Rise to Shut Off Head from the Design Point: 30%

Minimum NPSH (Net Positive Suction Head) Available (ft.): 36

Armed with these "facts", the application engineer in conjunction with the hydraulic engineer selected a pump with an enlarged eye impeller capable of meeting the design point of 12,500 GPM at 295 total differential head with a NPSH requirement of 31 ft.

What was not readily identifiable to the pump manufacturer was that conditions outlined were related to the "safety" design point and during "normal" operation the pumps would operate at much lower flows.

The pump selected was a 16x18x20 single stage, double suction, horizontally-split, double volute, double bearing pump (Figure 1) operating at 1760 RPM. With the exception of the impeller "eye" modification, this pump was of a well proven design, with a seven vane impeller. Similar units were operating extensively in many varied applications with a maximum of reliability.

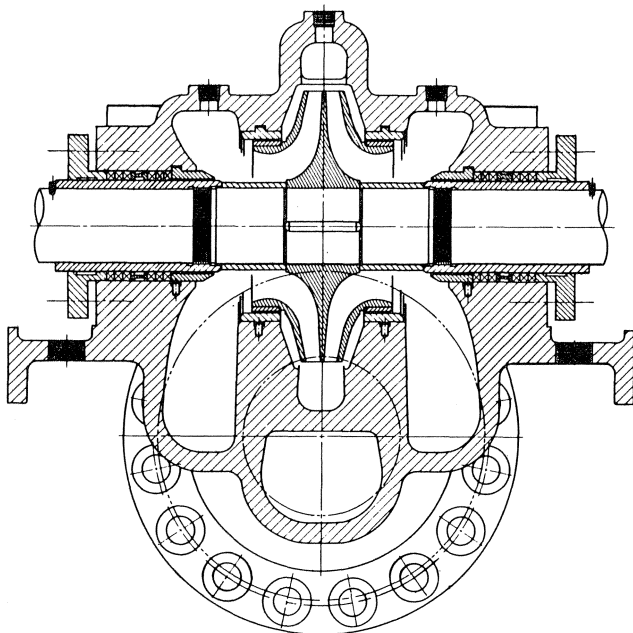


Figure 1. Single Stage, Double-Suction, Horizontally-Split Pump.

A minimum flow of 500 GPM was quoted by the pump manufacturer to the A.E. This value was predicated on the heat rise through the pump; however, this was misinterpreted by the A.E. to mean the minimum continuous operating flow.

On completion of the manufacture of the pump and in accordance with the specification and contractual requirements, the pump was "certified test" run. A "certified test" in the pump manufacturer's vocabulary usually means that the test is specified by the A.E./customer, is of a routine nature, and is not witnessed.

The pump was set up and hydraulically tested (Figure 2) with approximately five points of data taken. Since this order was for multiple pumps, all pumps were run in this manner and a log of the pump data and performance curve recorded. A certified NPSH test was also performed on the pumps and the same procedure was followed as for the certified performance test, except the NPSH was checked at the operating point only (i.e., 12,500 GPM).

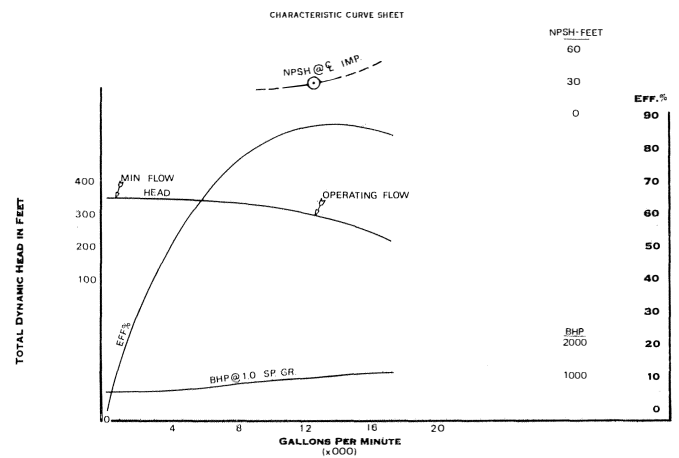


Figure 2. Horizontally-Split Pump Characteristic Curve.

The testing proved successful and was in line with the contractual requirements. The test data sheets and test performance curves were submitted to the A.E. for approval. The tests and data were subsequently approved and the pumps were released for shipment.

Field Evaluation

A field problem developed when on disassembly of one of the pumps, cavitation effects were observed on the impeller. At that time it was also observed that there was a build-up of silt above the pump suction. It was speculated that the silt build-up caused a suction restriction causing the pump to have insufficient NPSHA. To verify this, the silt build-up was removed and a second pump run; however, no change was noted in the noise level within the pump. This second pump was disassembled and the impellers displayed similar signs of cavitation as the first unit.

At this juncture the pump manufacturer was called in to discuss the problem with the A.E. and the user. From these discussions it was apparent there had been a major communication gap between the pump manufacturer and the user, which had not been covered explicitly in the technical specifications. The user fully expected the pumps to operate satisfactorily over a range of 0-100% flow; in fact the pumps had been operated extensively at a continuous minimum flow of 500

GPM, which represented approximately 3.5% of the best efficiency point (BEP) (Figure 2).

The pump manufacturer indicated that the specifications required the pump be designed to operate at a duty point of 12,500 GPM at 295 ft. head with an NPSH requirement of 31 ft. In order to meet this low NPSH requirement the impeller eye had been enlarged from 160 sq. in. to 180 sq. in. Enlarging the eye area had made the impeller more susceptible to "recirculation" effects at low flows. Part of the liquid flows back into the eye of the impeller at high rotational velocity and back into the main flow, causing recirculation or vortexing (Figure 3). This recirculation within the eye of the impeller and separation, with resultant stall effects, causes pulsations and cavitation type damage to the impeller material. Stalling occurs when the difference between the flow angle and the passage blade angles increases above a critical value, the stalled area washes out and reforms rapidly.

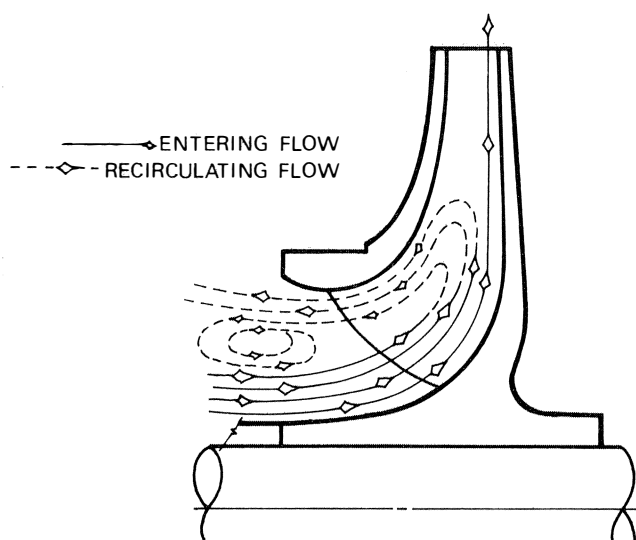


Figure 3. Impeller Eye Recirculation.

It was pointed out to the user a more realistic continuous minimum flow would be in the order of 4000 GPM or 30% of the pump best efficiency point. The user indicated the system design was such that to operate at a minimum flow of 4000 GPM presented a problem that only a major system modification to the piping would correct. It was, therefore, suggested the units be field tested to verify the hydraulic performance of the pumps and minimum flow as they were shipped from the manufacturer and to determine noise and vibration levels.

The pump was tested in the field; however, the minimum flow achievable was on the order of 4300 GPM due to plant requirements. The pump was extremely noisy, and high vibration was observed at the lower flow capacities (4300 GPM); however, noise and vibration decreased as the pump flow was increased and at flows around 12,500/13,000 GPM the cavitation noise decreased to almost zero and vibration was minimal.

Because of temperature control valve modulating, flow conditions were not steady or accurately controllable. However, the data that was accumulated substantiated that the pump was performing along its characteristic curve and corroborated the factory test. In the post test discussion between the user/A.E. and the pump manufacturer it was generally agreed that the conditions observed were due to "recirculation" and

separation caused by low flows, rather than NPSH problems at pump "runout" conditions.

It was suggested to the user/A.E. that the pump be returned to the pump manufacturer's facility for testing and further evaluation, and, to expedite correction of the low flow phenomenon, to fit the existing impellers with "anti-stall rings." The stall rings (Figure 4), it was theorized, would decrease recirculation at low flows, but would also tend to increase NPSH. Comparison tests would be made between the "stall rings" design and the original impeller design and to their compatibility with the existing system.

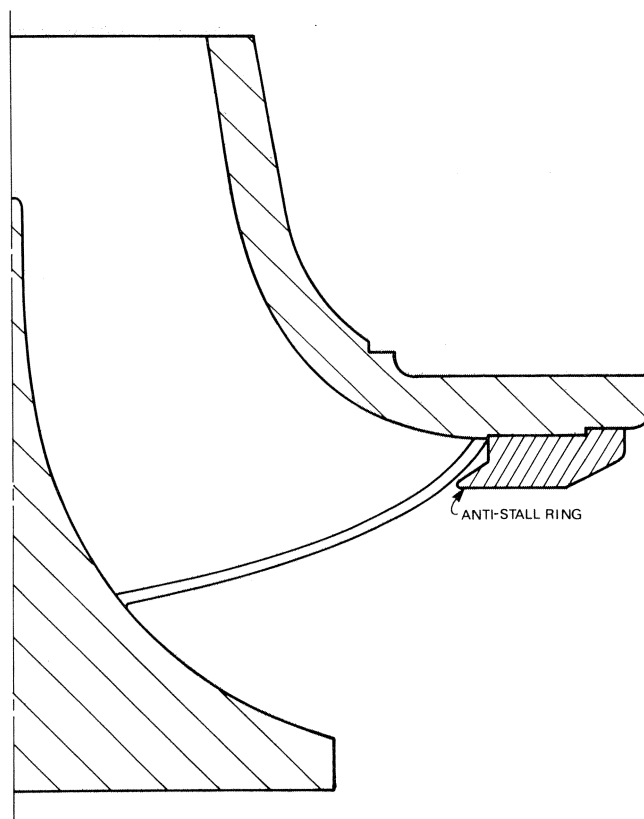


Figure 4. Schematic Showing Anti-Stall Ring.

Factory Test

The pump was received back at the pump manufacturer's facility and was set-up in a closed loop which included a suppression tank for NPSH testing. The general test loop arrangement, testing procedures and instrumentation were in accordance with the standards of the Hydraulic Institute and ASME Power Test Code PTC 8.2.

The original impeller design was tested to verify the basic performance. A six point performance test was run with noise and vibration data taken at 4000, 7000, 10,000, 12,500 and 13,200 GPM.

The noise data was taken at "all-pass," A-weighted readings and all octave bands using a General Radio octave band analyzer. The location was approximately four feet from the pump suction nozzle. Vibration data was taken on the bearing housings at all-pass, rotational, and other dominant frequencies. Additionally, shaft vibration readings using a "Vee" notch stick were taken at all-pass and at rotational frequency.

After completion of the original impeller design tests, the impeller was modified to incorporate an anti-stall ring (Figure 4). The hydraulic performance was duplicated and an NPSH test run to verify requirements with the anti-stall ring.

Comparison between the two tests indicated NPSHR (required) on the stall ring design at the design point of 12,500 GPM was 32 feet versus 31 feet on the unmodified design. The hydraulic performance comparison indicated appreciably lower noise and vibration levels at the lower flows (noise and vibration figures); however, at flows less than 7000 GPM vibration levels exceeded 0.002 in. peak-to-peak.

After the presentation of the data to the A. E. and the user, it was resolved that the pump system was such that the pumps would be required to operate significantly below the 7000 GPM level and the stall ring was not an acceptable solution. It was furthermore resolved that the pump manufacturer would undertake to review the existing pumping system for the purpose of establishing the true operating range of the pumps and designing a new impeller to meet these requirements.

Assessment of Pump Application

The existing pumping system was reviewed and it was confirmed that the pumps were not compatible with the system. No minimum flow recirculation loop had been installed as the user fully expected the pumps to operate over the full range of zero to runout flow conditions.

It was identified by the user that the minimum flow that had been established by field testing indicated that the system would use about 2000 GPM. To accommodate this, the piping would be required to be modified. The pump manufacturer indicated that 2000 GPM only represented 16% of the design flow and, as had been addressed in the factory testing, vibration levels at this capacity would be high and severely affect the operation and reliability of the pump.

A compromise was struck with the user accepting the pump manufacturer's recommendation of a minimum continuous flow of 4000 GPM without short-term damage to the pump. It was pointed out to the user that at these low flows catastrophic damage would not occur to the pump, but there would be accelerated wear until the pump could be returned to the design point.

The user indicated the criticality of the pump service and stressed the importance of safe, reliable operation, and agreed to allow the pump manufacturer to proceed with a new design impeller and reserve any modification to the system until the completion of the pump manufacturer's development program.

Redesign Recommendations

The background information from the field and factory testing indicated improvement must be achieved in the cavitation characteristics of the new impeller at the lower flows. Since a centrifugal pump can be designed for optimum performance at only one capacity at a given head and speed, attention must be given to the design to minimize undesirable effects at other capacities. Even with the optimum design, the characteristic noise and pulsations generated within the pump will be highest at low percentage flow capacities and reduce to a minimum at the best efficiency point. The intensity of these pulsations will also increase as the flow is increased beyond the best efficiency point capacity.

To achieve an "optimum compromise," the suction inlet of the impeller was modified resulting in a decrease of 8 sq. in. in the net through area of the impeller, a decrease in the relative area between the vanes of approximately 6.8 sq. in. and a reduction in the suction inlet angles.

Pressure pulsations are defined as low and high frequency response of fluid particles to complex, unsteady non-linear forces. High amplitude pressure pulsation levels may degrade the pump performance and cause accelerated damage to the pump and system components. It is important to maintain the pulsation generating mechanisms within the pump to reasonable proportions for pump unit reliability and performance.

To minimize these pulsations without affecting the discharge performance of the pump, the vanes on the new impeller were staggered 18° from side-to-side using seven vanes on each side. The vanes on the original impeller design were not staggered. A reduction of the pulsation level is obtained by the use of staggered vanes due to the reduced flow capacity passing the volute cutwaters at the vane passing time and the greater area available in the non-passing vane side of the impeller for pulse dissipation.

The impeller sidewall clearance and the volute cutwater clearance to the impeller diameter ratio were also checked for conformance with good design practice. If the cutwater clearance to impeller diameter ratio is too small, excessive pressure pulsations will be generated. If this clearance is too large, the pump performance will be impaired.

For the subject pump, the volute lip diameter as originally manufactured was 21¾ in. and the new design impeller was 19½ in.; thus, the diametrical clearance was 2¼ in., giving a 11.5% clearance ratio. Figure 5 illustrates the influence of the clearance ratio on the pump noise (and thus pressure pulsation) as a function of flow. As can be seen, the clearance ratio has a very strong influence. Also important to notice is the influence of the pump flow with respect to noise (and vibration). The further removed from the best efficiency point flow — the greater the increase. The best efficiency point of the subject pump was 13,800 GPM; therefore, the minimum flow suggested of 4000 GPM represented 30% of BEP.

The sidewall clearance (the clearance between the shroud of the impeller near the OD and the volute wall) is important since excessively tight clearance will also produce excessive pressure pulsations. The clearance for this pump was 9/16 in. on each side, which was considered adequate to preclude excessive pulsations. If this sidewall clearance is too large, circulation effects may impair the pump performance.

Although the new impeller proposal was considered optimum for the range of operation required for this installation and the internal clearances of the pump were reasonable to provide reliable operation for the pump internals and low

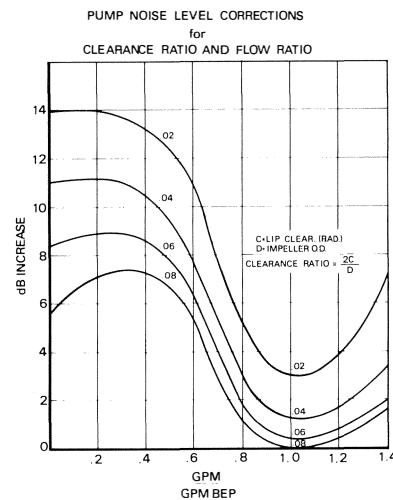


Figure 5. Pump Noise Level Corrections.

pulsation levels, only a full hydraulic test program would "prove out" the design and cavitation characteristics.

Development Testing

The test procedures and acceptance criteria were discussed with the A.E./user. It was agreed that vibration in itself was not conclusive of pump performance and sound levels and pressure pulsation levels should also be recorded to supplement the analysis. The basis of acceptance of performance was the requirement by the user that the pumps operate out of the destructive recirculation range at 4000 GPM. The pump manufacturer indicated that although the hydraulic analysis had indicated acceptable design criteria at these levels, a number could not be qualified without actual pump testing.

Two separate witness performance and NPSH tests were planned for the pump. The first test would be with the original unmodified impeller; the second test with the newly designed impeller. Performance tests would consist of seven data points from shut-off to 135% rated (design) flow, including a minimum flow of 4000 GPM and rated flow of 12,500 GPM. Noise readings would be taken at each flow during the performance tests. Vibration readings would be also taken at each flow on the bearing housing and two proximity probes would record shaft readings.

The pump was instrumented to provide a maximum of performance characteristics. The two proximity probes to measure shaft radial vibration at the inboard and outboard bearings were mounted 45° off the vertical centerline. A proximity probe was mounted axially through an opening in the thrust bearing cover. This probe would give an indication of any axial vibration of the rotating element at low flows. Pressure pulsation measurements would be taken by installing pressure transducers in the pump discharge and suction nozzles. Again, as in previous tests, the noise data was taken at "all pass," "A" weighted and all octave bands using a General Radio octave band analyzer. The location was approximately four feet from the suction nozzle.

It was the pump manufacturer's intention on completion of the testing to assess and recommend permanent instrumentation to monitor vibrations in the field and, with the A.E./user, to establish test criteria, procedures and instrumentation for future surveillance testing based on the test results.

Real-time analyzer spectrums for the original impeller indicated that vibration levels were within acceptable levels and axial vibration decreased as the flow was increased (Figure 5). The predominant frequency for the axial vibrations occurred at approximately 2700 CPM. There was no significant axial vibration at frequencies less than the rotational speed. The radial vibration frequency was predominantly rotative (1800 CPM). Pressure pulsation measurements taken at the suction and discharge nozzle had predominant frequency components at vane passing, 12,320 CPM, twice vane passing and 8000 CPM.

Real-time analyzer spectrums for the new design impeller indicated the vibration and pulsation amplitudes were considerably less for the new impeller than for the original impeller. There were no significant low frequency axial vibrations and the amplitude was less than 0.001 in. peak-to-peak at rotative speed. The radial vibrations were essentially the same frequency characteristics as the original impeller. The pressure pulsation frequency characteristics showed primarily at vane passing and twice vane passing and, although the 8000 CPM component was still present, it was less pronounced.

The NPSH comparison for the new impeller versus the original impeller showed substantial improvement for the new impeller at low flows. While both the new and the original impeller were capable of performing at 31 feet NPSH at the

design flow of 12,500 GPM, only the new impeller would run satisfactorily at the minimum flow of 4000 GPM.

Summary

On completion of the development testing, the operating range of the new impeller pump was reviewed by the pump manufacturer and the A.E./user. The appreciable improvement in vibration and pressure pulsation and the capability to operate at reduced flows with low NPSHR indicated that maximum reliability for this given system could be obtained by installing the new design impellers.

The pumping system should be modified in the field by providing a bypass system to maintain a minimum flow of 4000 GPM. This bypass system would be complete with bypass valves controlled by actuators on the discharge leg of the pumps. In the event of an accident in this particular installation, the system flow could be reduced to 2000 GPM for an extended period of time of approximately three weeks.

This condition was acceptable to the pump manufacturer providing the user recognized that accelerated wear would occur at these low flows until the pump could be returned to its design point.

CASE II

Review

This case study is based on the history of a pump failure in a high energy pumping application under the following operating conditions:

Service: Steam Generator Feed Pumps

Operating Temperature (°F): 385

Capacity (GPM): 15,200

Net Differential Head (ft.): 1900

Continuously rising head characteristic as capacity decreases, for parallel operation.

NPSH Available (ft.): 286

NSPSH Available (ft.): 286

Tandem electric motor driven at 3560 RPM; 7400 BHP required at design conditions.

The pump selected to meet the above operating conditions was a type 16x18x24 style "CD" pump (Figure 6). This is a

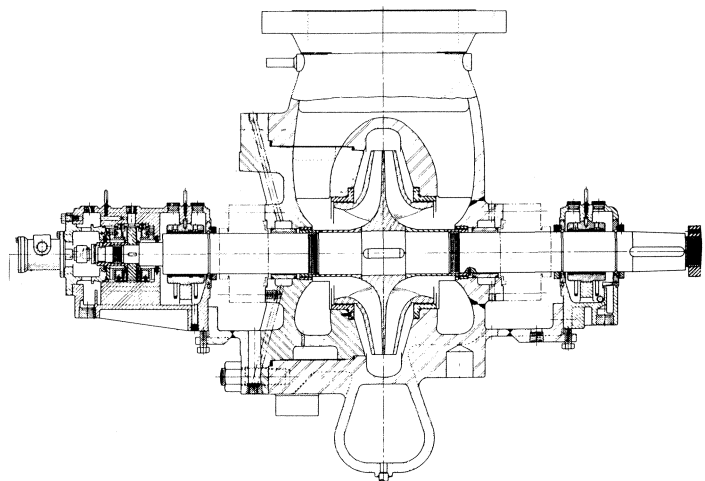


Figure 6. Single-Stage, Double-Suction, Radially-Split Pump.

single stage, double volute, radially-split, double suction casing design, which utilizes a 16 in. diameter discharge nozzle, 18 in. diameter suction nozzle and 24 in. nominal volute diameter. The double suction impeller was trimmed to 22 in. diameter and operates at 341 ft/sec peripheral velocity.

Although the specifications were generally detailed as to certain aspects of the pump mechanical design, they lacked specifics in the area of actual system operating conditions related to pump performance requirements. Service conditions were stated; however, again more details were stated as to physical location of the pump in the system rather than actual system operating conditions. A system performance curve graphically illustrated an operating range of 0 to 30,400 gallons per minute flow conditions (two pumps in parallel) plotted against system resistance (pressure psia).

The specification contained very generalized statements such as:

- "All pumps shall be free from tendency to vapor bind or unreliability from other causes."
- "Reliability of the feed pump shall be the highest obtainable for this type equipment."

The pump specification also stated: "During periods of low rates of flow means will be provided for by-passing a portion of the pump discharge when the power loss within the pump will cause excessive temperature rise which will lead to possible flashing or binding at the running clearances." The customer specified the pump manufacturer to provide a multiple, welding type orifice to accommodate minimum flow operation which was established by the manufacturer at 20% of design flow or 3040 GPM. Note that the pump specification did not state any approximation of running time at minimum flow operating conditions (Figure 7).

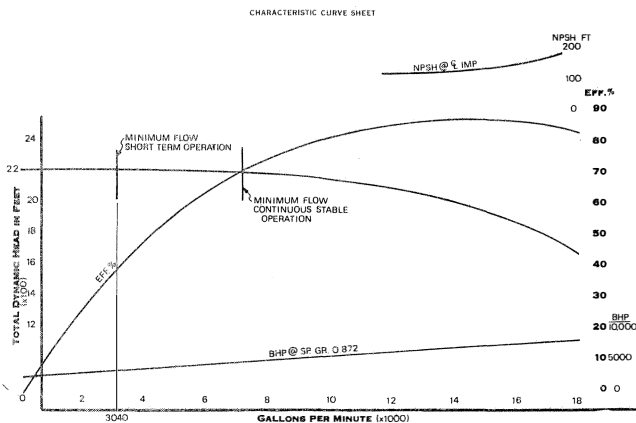


Figure 7. Radially-Split Pump Characteristic Curve.

Specification testing requirements called for the following in addition to routine shop development testing on each pump:

1. Full range performance testing at 3560 RPM and 385°F to establish complete performance characteristics in accordance with ASME Power Test Code for Centrifugal Pumps.
2. NPSH testing at 3560 RPM and 385°F to determine NPSH required at various capacities.
3. Temperature transient testing. Each pump to be subjected to a full speed temperature transient ramp of

385°F, to 90°F in a two minute period at a flow rate between 500 to 1500 GPM.

4. Post test inspection of all parts for condition and disposition after completion of all the testing above.

Pump Design

To meet the hydraulic design conditions, a small factor was applied to proven, existing hydraulics. This is a common practice in centrifugal pump design and presented no problems to the experienced hydraulic designer. Exact impeller trim diameter to meet field operating conditions was determined after the initial hydraulic test.

Mechanical design presented a somewhat greater challenge since it was recognized that the nature of the service and the energy level involved required significant attention. Potential loss of suction due to plant upset conditions required a "stiff shaft" design to limit rotor deflection, thereby minimizing contact of rotating versus stationary wear parts to avoid potential pump seizure. Shaft L/D ratio (where L = length between journal bearings, D = average shaft diameter between bearings) was 9.9, which is an excellent stiffness ratio. For this application a maximum ratio of 11.0 was targeted by the designer.

The wear part material must be selected to be of non-galling nature, while also being compatible with the pumpage, for the same reason. In this case, 17% chrome, 4% nickel precipitation hardened alloy stationary wear parts were selected to mate with integral impeller rotating surfaces of 13% chrome, 4% nickel material. The impeller was heat treated to achieve a 315 to 350 Brinell hardness number (BHN) while the stationary rings were heat treated to a 400-450 BHN range. These values not only allowed a minimum hardness differential of 50 BHN for galling resistance, but also kept the impeller in a ductile, yet high strength condition for maximum reliability.

A radially-split casing design was selected in deference to an axially-split casing due to high pressure, high temperature application requirements. Other design features include mechanical seals, equipped with pumping rings to circulate pumpage through a heat exchanger, then to the seal faces; babbitted style plain journal bearings using forced feed oil lubrication with oil ring backup; and a tilting pad type thrust bearing, sized for thrust loads anticipated plus 50% minimum additional capacity, also with force feed lubrication.

Factory Testing

All testing was completed in accordance with the manufacturers and customers specification requirements as previously outlined. All testing results were satisfactory, including the temperature transient shock test. Vibration levels as measured on the pump shaft adjacent to the journal bearings were all within acceptable specification values (0.002 in. maximum peak-to-peak amplitude). Post test inspection revealed no undue wear or signs of distress on any pump components.

Field Evaluation

Approximately ten months after being installed in the field, a series of three failures occurred within a short period of time, one which was of a catastrophic magnitude. Following is a description of the three respective failures; an explanation of the failures based on field data, manufacturer's analysis, and foundry metallurgical study.

Failure One

The first minor failure occurred in pump number one when the casing developed a small leak through its pressure

retaining boundary at the top exterior surface near the volute centerline. The crack like defect extended to the interior of the casing at the discharge side of the volute cutwater (Figure 8). Subsequently, the pump case was returned to the pump manufacturer's facility, repaired per approved procedures, machined as required to reestablish the proper fits and registrations, then returned to the job site and reinstalled.

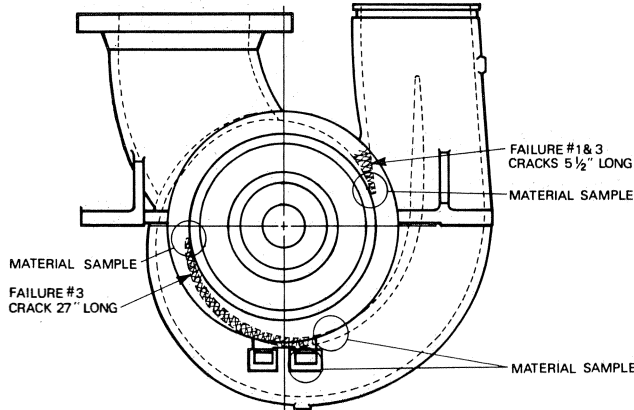


Figure 8. Pump Casing Metallurgical Sample Locations.

Failure Two

During subsequent operation a catastrophic mechanical failure occurred in pump number two. Although many parts were severely damaged, a thorough investigation led to the conclusion the thrust bearing failed first. Evidence of fretting and pitting on the thrust support parts indicated vibratory forces were of considerable magnitude and that the thrust bearing oil film could not support this loading.

The thrust bearing failure initiated a shaft whip due to unequal forces on the rotating thrust bearing collar. This allowed the outboard end impeller locking sleeve to contact and gall with the mating stationary throat bushing. The resultant heat at the locking sleeve destroyed its strength, causing the threads to strip, thus allowing the impeller to be moved off its center position towards the outboard end.

When the impeller was off center, the axial force against the shroud increased on the more widely spaced side due to lower vortex velocities. Extremely high axial forces were then created when the impeller contacted the volute sidewall near the impeller periphery causing a metal seal to form, creating still lower shroud pressure on that side. This additional thrust completed the destruction of the thrust bearing. The outboard casing wear ring, impeller, and outboard mechanical seal were also destroyed during these events.

Failure Three

Two days after failure two on pump number two, pump number one failed again. The mechanical seal assembly on the outboard end of the pump moved axially on the shaft and made contact with the bearing cover deflector disc. The force was great enough to move the deflector disc against the bearing cover with the resultant friction heat damage to these parts.

During subsequent repair, it was decided to further examine the pump internals. Crack type linear defects were found in both suction and discharge side volute splitters terminating at the cutwater ends. The crack in the discharge side splitter ran approximately 5 1/2 in. from the end (volute lip) of

the splitter, while the crack in the suction side splitter extended approximately 27 1/2 in. or 105° of arc from the end of the volute lip (Figure 8).

Failure Analysis and Recommendations

During the course of events described above many studies were made to determine the causes for the pump failures. One fact became apparent to the pump manufacturer at the outset of the problems: the pumps were being operated at a low flow rate continuously for long periods of time. This mode of operation was required to complete system functional testing. Being direct motor driven, the pumps were required to run at full speed at 25% design flow rate (4000 GPM) for hundreds of hours. It was later explained that the exact running time in this mode could not be accurately predicted due to many unknowns during plant and system shakedown. The pump specifications did not specify the plant start-up mode of pump operation.

It was concluded the pump failures were initiated by high magnitudes of vibration caused not only by the pulsation generating mechanisms of the pump, but also as a result of the dynamic interaction of the piping and motors. This was concluded based on data collected from vibration surveys conducted at the job site. Following failure three, permission to undertake a metallurgical study was requested by the pump manufacturer since the same casing had failed twice. To accomplish this, the casing would have to be cut into various samples (Figure 8), therefore rendering it useless for further service.

Since the original supplier of the castings was no longer in business, a reputable foundry was requested to conduct the analysis. The material furnished for the pump casing was 13% chrome, 4% nickel alloy. The pump specifications did not require chemical or mechanical test reports and no non-destructive examination was specified.

Conclusions of the metallurgical report indicated a high hardness value in one volute cutwater area attributed to improper welding techniques used in an original casting repair. However, it was clearly determined that both cracks were fatigue type failures caused by high stress levels across the volute splitter. These were a combination of primary, secondary, and peak stresses produced by operating pressure and other mechanical loads, including pump nozzle loads and effects of structural discontinuities. Piping interaction included the potential nonuniformity of flow into the pump suction nozzle, potential resonant amplification of the pump pulsations involving either or both the fluid or pipe wall, and potential pulsations generated within the piping or other connected equipment.

The pumps had been operating for a considerable period of time at a flow rate of 4000 GPM with the resultant detrimental effects. The pump manufacturer recommended installation of staggered vane impellers to reduce the pulsation levels developed by the vanes passing the cutwater at low flow operating conditions. Also recommended was increasing the minimum flow to 7000 GPM for extended full speed, continuous low flow operation.

Subsequently, both these recommendations were accepted and implemented. Over three years of trouble free service for both pumps has proved the merits of these recommendations.

OPTIMIZING PUMP SPECIFICATIONS

General Discussion

The specification writer must be specific in constructing a meaningful document, yet use good judgement as to subjects

dealt with. In other words, totally define all operating conditions, system design parameters, and specific preferences for the options required. Also, the writer must specify testing requirements and acceptance criteria to be satisfied to meet field operational needs.

The pump manufacturer should be relied upon to apply these specification requirements to design an efficient, reliable machine that will meet the user's needs for the specified service life. Thorough factory testing will be the ultimate proof that the specifications have been satisfied.

Reviewing various pump specifications, it is evident that more emphasis has been placed on high pump efficiency rather than ultimate reliability to meet actual field operating conditions. Today, reliability is more important than ever when considering the high cost of "down time" for applications related to power production, petro-chemical and many others. Generally, a trade-off in reliability for artificially high efficiency is not to the user's benefit and must be carefully evaluated in the proposal stage.

Although it is true that many pump specifications must be somewhat general in nature due to universal usage, it is extremely important to carefully define *actual* field operating conditions for every unit ordered. Unfortunately, it is common for a customer specification to only state hydraulic requirements for a specific design condition, rather than accurately and totally defining all pump operating modes that will be encompassed. Another common fault is to use "generalized statements," such as "pumps must be capable of satisfactory operation over the entire operating flow range from zero to runout without undue vibration." This statement alone means little to the pump designer. He must design the pump to meet a specific hydraulic operating condition; therefore, logic dictates the units will normally be operated at or near this point unless the specification elaborates otherwise. Obviously, from the preceding case studies, this practice can result in a costly, time consuming corrective action cycle, especially when "high energy" pumping equipment is involved. Many problems will be avoided through the diligent use of definitive, practical specifications.

There are options available to the designer to satisfy most operating conditions with reliable pumping equipment when all facts are known. As an example, if it is known that an off-peak low capacity condition is imminent for a length duration (as in Case II), a lower capacity impeller can be designed to improve pump performance and resultant reliability during this interim period.

Design Specification Content

The following listing reflects the author's views on minimum technical specification content requirements to insure procurement of reliable centrifugal pumps. It should be noted that these are *minimum* requirements and may not include some specific requirements for certain applications or other customer requirements based on previous experience, interchangeability standards, etc.

I. General Requirements

- A. Codes and Standards
- B. Definitions
- C. Application Description including All System Design Operating Parameters.
 1. Minimum Flow Operation and Control System
- D. Pump Hydraulic Condition Design Requirements

II. Mechanical Design Requirements

- A. Shaft Sealing System
- B. Bearing Requirements
- C. Auxiliary Equipment
 1. Instrumentation
 2. Auxiliary System Requirements
 - a) Testing Requirements
 3. Coupling Requirements
 4. Prime Mover Specifications
 - a) Over Speed Requirements (When Applicable)
 5. Support System (Baseplate) Requirements

III. Materials of Construction

- A. Major Component Material Identification
- B. Material Certification and Documentation Requirements
- C. Nondestructive Examination Requirements

IV. Testing Requirements

- A. Hydrostatic Testing
- B. Hydraulic Performance Testing
- C. NPSH Testing
- D. Transient Testing (When Applicable)
 1. Temperature
 2. NPSH

V. Technical Data Requirements

- A. Engineering Drawings
 - Certified Outline Drawings
 - Piping and Instrumentation Drawings
- B. Instruction Manual
 1. Sub-Vendor Information & Drawings
 2. Technical Job Specific Details
 3. Materials of Construction

Elaborating further on the above specification outline, the following design parameters are most commonly neglected and require elaboration:

Application Description Including All System Operating Parameters

The importance of this information to the pump designer cannot be overstated. The two case studies included in this paper are only typical examples of what can happen when all field operating conditions are not clearly stated for the pump designer's use. Low flow or "off peak" operating conditions must be identified, including reasonable operating time estimates.

Testing Requirements

To obtain pumping equipment of optimum reliability for the intended application, the requirements for mechanical integrity testing should be carefully considered. Testing requirements should be established to determine if the pump will function, as designed, under conditions as close as practicable to those encountered in the field. It is necessary to test pumps under full load, full speed conditions to insure mechanical integrity. This is especially important when high energy pumps are involved.

For critical/high energy applications, "real time" plots of vibration data should be taken during factory testing for later comparison with field data as needed. The entire operating range from minimum flow to 120% of design flow should be covered to insure acceptable field operation at flow rates required. Also, when variable speed units are required, the entire speed range should be scanned for the same reason. If the pump in question must operate at a minimum flow condition for long periods of time, a mechanical run on the test stand for a specified duration should be considered.

The sixth edition of API Standard 610 dated January 1981, "Centrifugal Pumps for General Refinery Services," now has an optional provision for testing at *minimum continuous stable flow*. This paragraph is intended to limit allowable pump vibration to specification levels, at minimum continuous flow, for application the purchaser considers this operating mode critical to the ultimate reliability of the unit(s) in question.

Other forms of testing, such as temperature and NPSH transient testing, can be equally important to insure field reliability. If the pump will be subjected to transient operating modes in the field, then similar factory test conditions should be strongly considered. Auxiliary systems may also be affected by pump transients. As an example, consider a high speed boiler feed pump with a packingless stuffing box design being fed by a pressure controlled injection system. If the pump is subjected to an NPSH transient to simulate a loss of suction condition in the field, at operating temperature (say 300°F), will the pump survive during a loss of injection flow? This can only be absolutely determined by factory testing if the manufacturer's previous experience does not satisfy this question.

CONCLUSION

Outlined in this paper are some of the problems surrounding incomplete or unrealistic design specifications and their effect on pumping equipment. These case studies are indicative of many instances where the pump manufacturer supplies equipment that does not meet the needs of the ultimate user. In many cases a solution is never achieved because the service is not essential and the operators put the unreliability of the equipment down to "poor quality" or "bad design." In the case instances cited, the problems showed themselves early in the functional testing of the plants because of the "essential" nature of the services.

Communication must be improved between the A.E., pump manufacturers, and the user. The A.E./user is not always cognizant of the limitations of a pump design for a given application and may inadvertently place unrealistic restrictions or conversely no restriction at all on the equipment manufacturer. This paper has attempted to outline the parameters necessary in technical specifications to guide and assist the pump designer.

It is believed through discussion with A.E.'s and industry representatives, pump manufacturers can assist and improve technical specifications and can offer guidelines, such as those offered in this paper, which will ensure that the equipment purchased in the future will be designed to obtain optimum pump performance and system reliability. Only through these combined efforts will the common goal of increased component availability be achieved.

