



pumpworld

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Cover

Since the creation of the human race, man has walked upon the earth's surface, plowed its fields, and partaken of its food. Unknown to him were the vast mineral treasures which lay beneath the ocean floor. One day he discovered these treasures, and he began to dig. Our cover illustrates a floating deep sea mining plant now operational in the Red Sea. As our mining article beginning on page 15 suggests, the key to mankind's existence may well lie beneath the sea.

Are you short on npsh? Nine ways to improve unfavorable suction conditions.

By Igor J. Karassik

The centrifugal pump, like any other machine, will perform properly if it is treated properly. The most important, most exacting of our obligations is to provide the pump with adequate net positive suction head, npsh. That is, the *available* npsh must equal or exceed the *required* npsh. If this obligation is not met, the pump will malfunction in a variety of ways. It will begin to cavitate, make unseemly noises, and, in extreme cases, inflict permanent damage on itself. All this is well known to pump users, and yet, some continue to mistreat their pumps. At worst, users fail to tell the pump manufacturer the real terms under which they plan to employ the pumps. At best, they ask for pumps designed for the minimum possible npsha. This article examines a number of approaches for solving the dilemma of seemingly inadequate npsha.

Net positive suction head available (npsha) is the energy in excess of the vapor pressure of the liquid at the pumping temperature. Net positive suction head required (npshr) (a function of the design of the specific pump model) is the minimum required margin between the suction head and the vapor pressure at a given capacity.

Npsha is expressed by the formula

$$H_{sa} = Z + (P_s - P_{vp}) - (h_{fs} + h_i)$$

in which: H_{sa} = npsha; Z = static head; P_s = pressure above the liquid level; P_{vp} = vapor pressure of the liquid; h_{fs} = friction loss from a to b; h_i = entrance loss at a.

In this formula, all heads and pressures must be expressed in feet of liquid at the

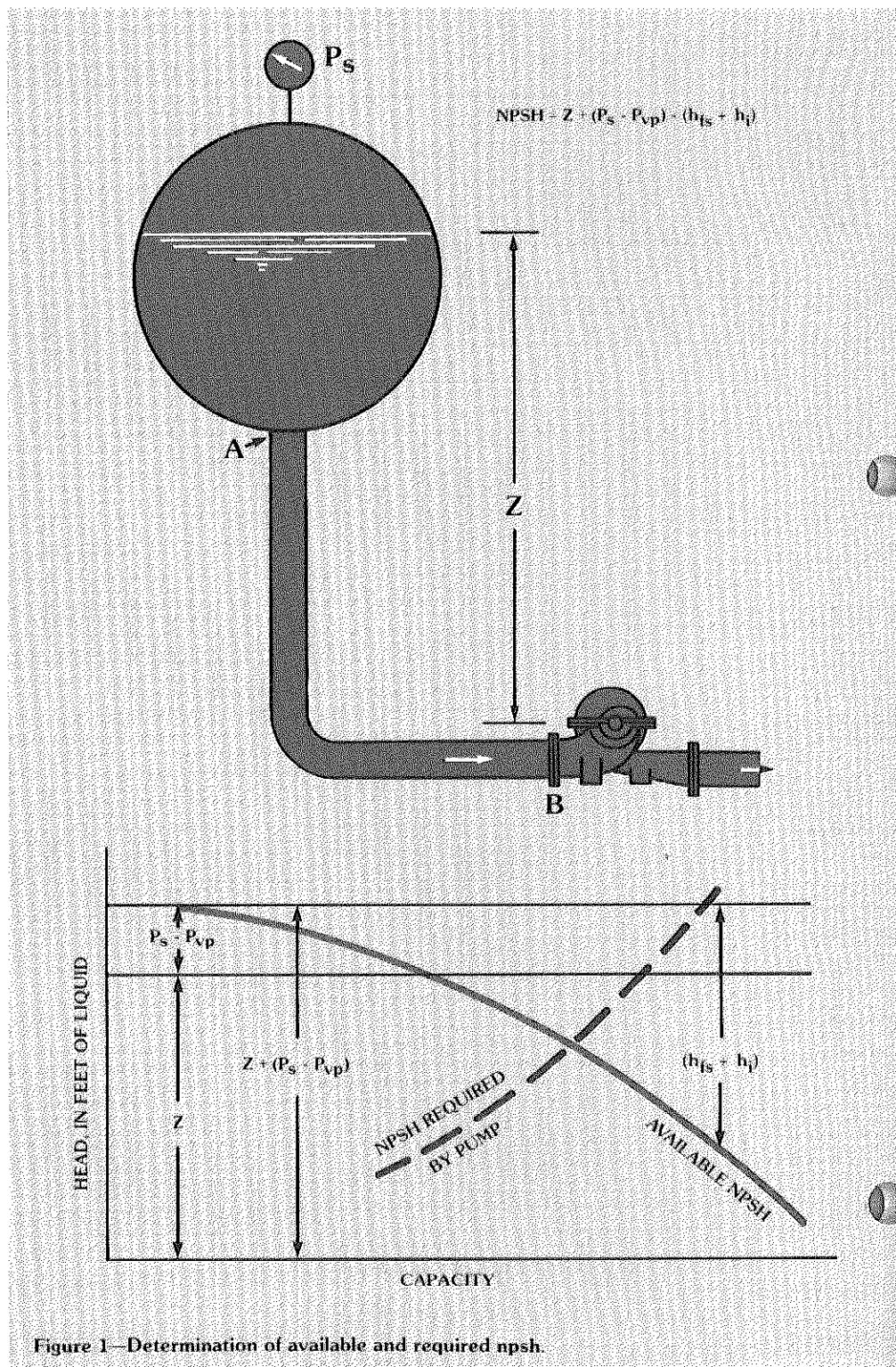


Figure 1—Determination of available and required npsh.

Igor Karassik is chief consulting engineer for Worthington Pump Inc.

pumping temperature with the proper algebraic sign. While P_s and P_{vp} can either be gage or absolute values, they must both be measured under the same conditions.

The relationship between these individual components of $npsha$ is illustrated in Figure 1 with a curve of $npshr$ superimposed. The pump cannot operate beyond the intersection of $npsha$ and $npshr$.

When a system offers insufficient $npsha$ for an optimum pump selection, there are several ways to deal with the problem. Each method presents its own special advantages—and disadvantages. Depending on the specific situation, we can:

Raise the liquid level.

This is the simplest solution—when it can be done. Referring to our $npsha$ equation, increasing the value of Z increases the $npsha$, foot for foot. Of course, there are times when this is not practical. The liquid level may be fixed, as in the case of a river, pond, or lake; the amount by which the level would have to be raised may be completely impractical; or the cost of raising a tank, a fractionating tower, or a deaerating heater may be excessive.

Nevertheless, this approach should be considered. Sometimes just a few extra feet of suction head may permit the selection of a much less expensive or much more efficient pump. The resultant savings in first cost, energy, or maintenance may far outweigh the additional costs incurred by raising the source of the liquid.

Lower the pump.

Just as in the case of raising the liquid level, the cost of lowering the pump may be a wise investment, since it may permit the selection of a higher speed, less costly, and more efficient pump. If it's not practical to lower the floor level at which a horizontal pump is installed, the pump user can try a vertical pump operating in a sump. If an open sump is impractical—as is the case with condensate pumps or pumps handling volatile, inflammable, or toxic liquids

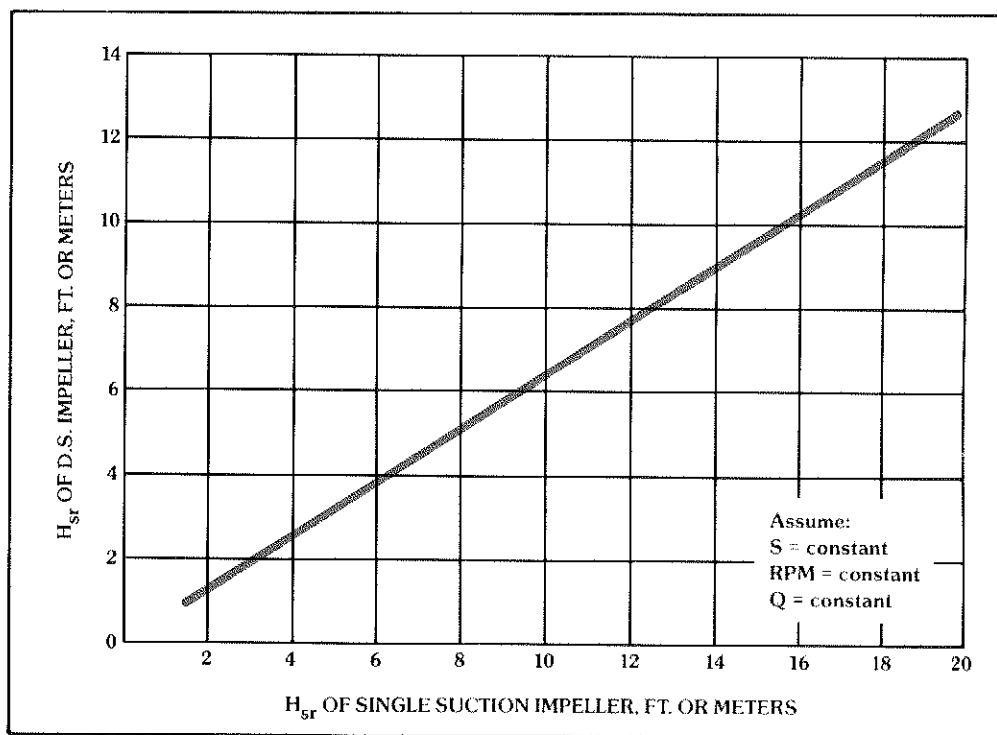


Figure 2—Effect of using a double-suction impeller on required $npsh$.

— the solution may be a can pump. This is essentially a vertical turbine pump mounted in its own closed sump: a tank or “can” which is sunk into the floor. The pump is simply made long enough to reach the level where sufficient $npsh$ is available for the first-stage impeller. The diameter and length of the can must allow proper flow between the pump and the tank, and for the turn and flow into the bellmouth of the pump.

The disadvantage of vertical pumps lies in the fact that the pump bearings must usually be lubricated by the liquid pumped. While some modern bearing designs and materials are quite satisfactory, generally they do not last as long as oil- or grease-lubricated external bearings.

Use slower speeds.

In weighing the relative adequacy of a particular pump design from the point of view of required $npsh$ values, it is most practical to make comparisons of the suction specific speed that it represents. Suction specific speed (S) is defined as:

$$S = \frac{n \sqrt{Q}}{(H_{sr})^{3/4}}$$

where n = pump speed in rpm; Q = flow in gpm; H_{sr} = $npshr$. For single-suction impellers, Q is the total flow. For double-suction impellers, Q must be taken as half the total flow.

The $npsh$ curves recommended by the Hydraulic Institute are based on S values from 7480 to 10,690 with most of the curves falling below 8500. These values are relatively conservative and may be raised somewhat, but values of 8500 to 9000 should not be exceeded, particularly if the pump is required to operate over a fairly broad range of capacities.

It is obvious from our suction specific speed equation that a lower speed pump means lower required $npsh$. The problem, however, is that a lower speed pump is more expensive than a higher speed pump designed for the same conditions of service. Most frequently, it is also less

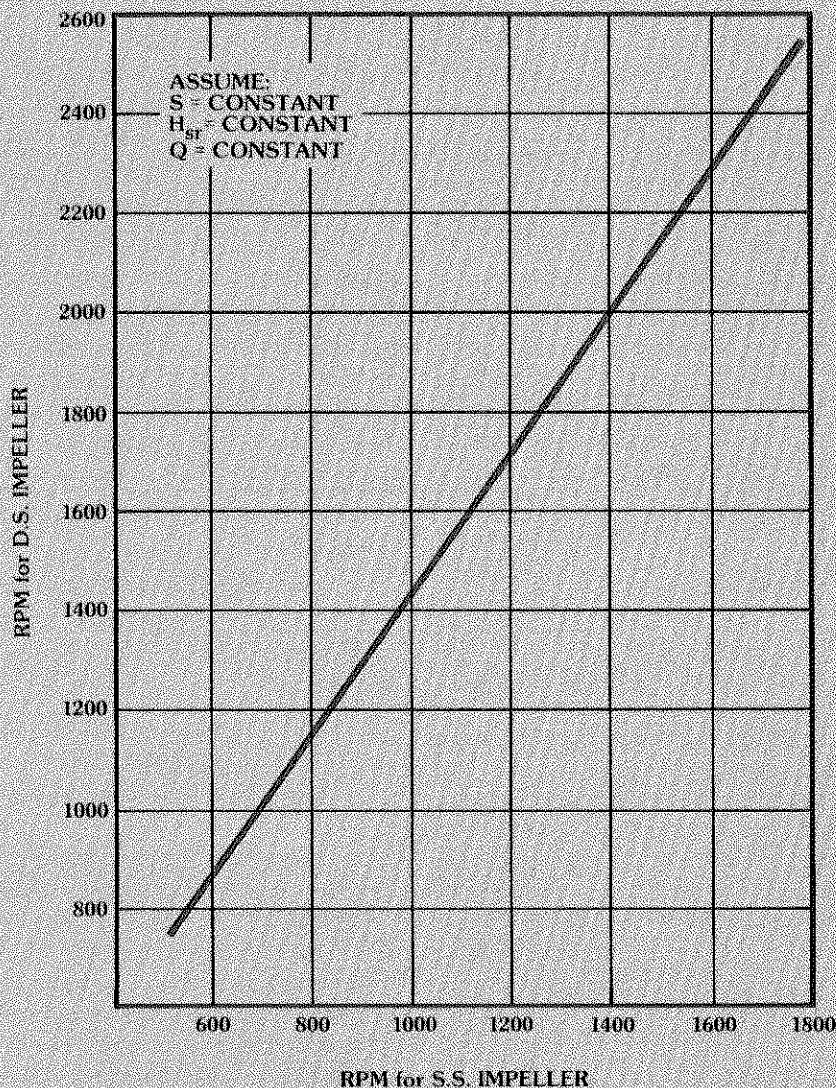


Figure 3— Effect of using a double-suction impeller on permissible speed.

efficient. Considering today's energy bills, lowering the pump speed seldom proves to be an economic proposition.

Use a double-suction impeller.

Whenever a double-suction impeller design is available for the desired conditions of service, it is likely to be the most desirable solution. A double-suction design allows a 27% reduction in nps_{hr} —

or a 41% higher speed. These benefits are based on the following considerations:

$$S = \frac{n_1 Q_1^{1/2}}{H_{sr1}^{3/4}} = \frac{n_2 Q_2^{1/2}}{H_{sr2}^{3/4}}$$

where subscript (1) refers to a single-suction impeller and subscript (2) refers to a double-suction impeller

$$Q_2 = \frac{Q_1}{2}$$

and for the same value of S, we can assume

$$(a) \quad n_2 = n_1$$

in which case $H_{sr2} = 0.63 H_{sr1}$ or

$$(b) \quad N_{sr2} = N_{sr1}$$

in which case $n_2 = 1.414 n_1$

Keeping the pump speed the same in both cases, as in Equation (a), we can reduce the nps_{hr} by 27% if we use a double-suction impeller (Figure 2). Alternately, with a given nps_{hr}, as shown in our equation, we can operate a double-suction pump at 41.4% higher speed (Figure 3). In both cases the value of S, the suction specific speed, remains the same.

Use a larger impeller eye area.

In this case, the idea is to decrease the nps_{hr} by reducing the entrance velocity into the impeller. Within limits, the practice is feasible; these lower velocities will probably not affect pump performance at or near the best efficiency point. When such pumps run at part capacity however, reduced velocities can lead to noisy operation, hydraulic surges, and premature wear: symptoms of internal recirculation at the impeller suction.

At lower capacities the flow at the outer eye diameter tends to reverse itself. Part of the liquid leaves the outer part of the impeller at a rotational velocity and then "folds back" into the main flow in the form of a vortex, as shown schematically in Figure 4. This vortexing causes surges and pulsations and leads to rapid deterioration of the impeller through cavitation damage — the very problem we wanted to avoid

The lower the nps_{hr} required by the impeller, the closer will be the inception of internal recirculation to the capacity at best efficiency, and the higher the operating flow at which surges and hydraulic pulsations will start (Figure 5). That is why this solution should be avoided if the pump is intended to operate over a wide range of flows below design conditions. On the other hand, it may be practical for applications like condenser circulating pumps, since they generally operate at or beyond their design capacity. This is particularly true when several circulating pumps

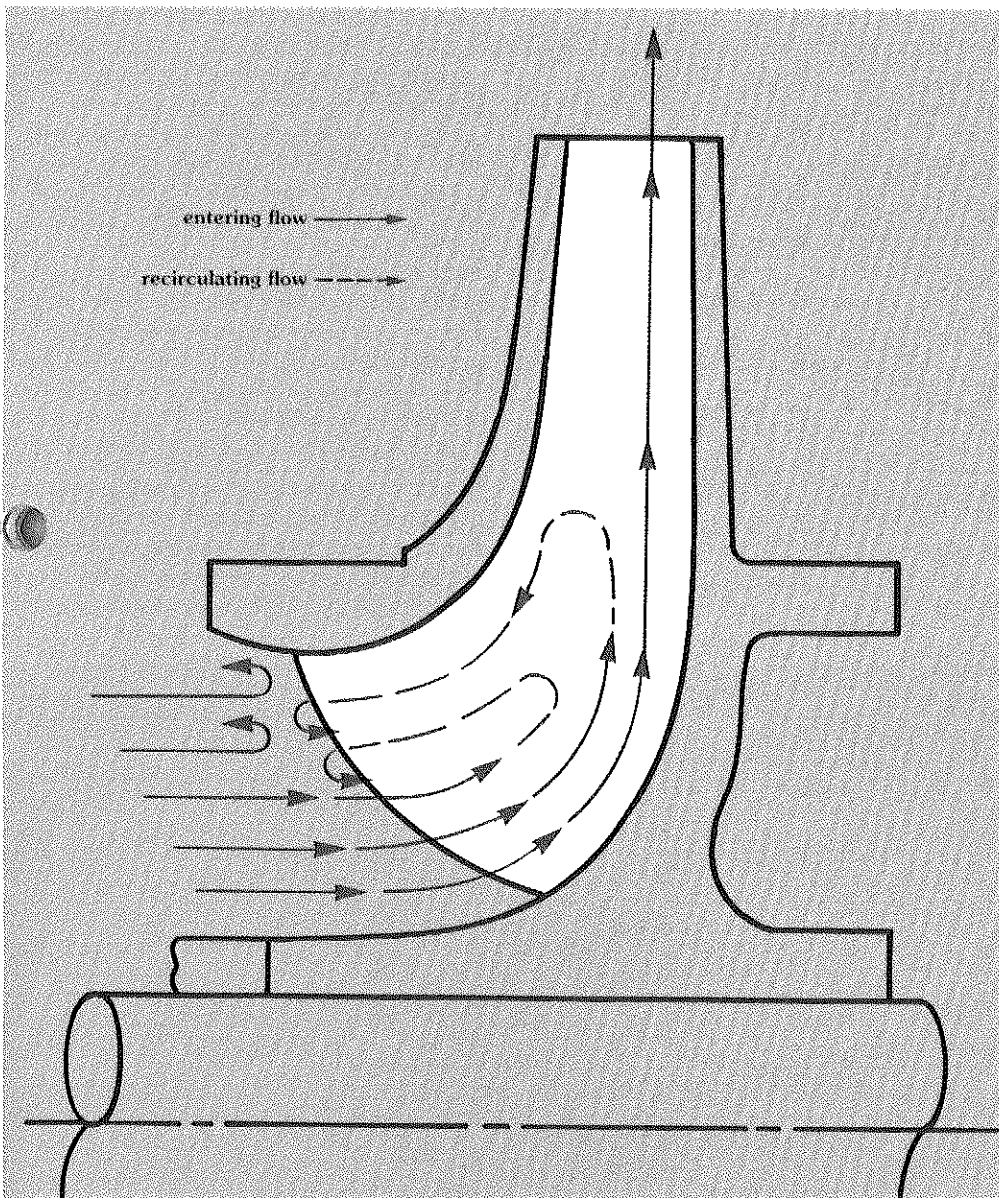


Figure 4—Suction through a single-suction impeller indicating schematically the recirculation of liquid at the inlet during operation at low capacities.

operate in parallel. Shutting down one of the pumps automatically causes the remaining pumps to run out further on their head-capacity curves, intersecting the system head curve at a lower total head value.

Use an oversized pump.

Npsh required by a pump decreases as capacity decreases. That is, npshr is lower

at part capacity. For that reason, a larger pump than would otherwise be applied to the service is occasionally selected (Figure 6). This practice is risky and can lead to undesirable results. Oddly enough, an oversized pump may operate at too low a flow as well as at excessive capacities.

If an oversized pump is permitted to operate unthrottled, the head-capacity curve will

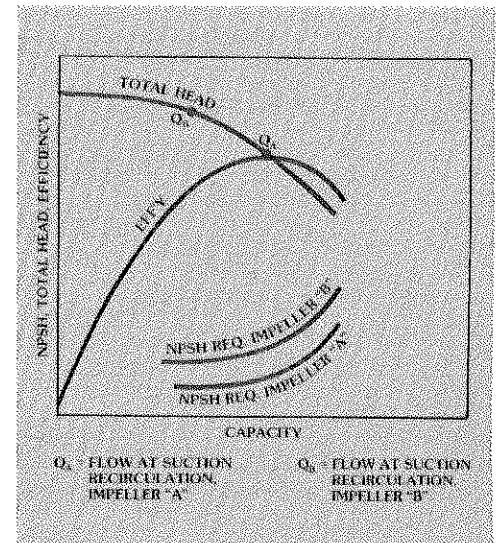


Figure 5—Location of suction recirculation

intersect the system head curve at a capacity much in excess of that desired. This raises the npshr, negating the intended result. In addition, low and medium specific speed pumps have power consumption curves which increase with capacity and will require more power. The pump can be throttled to reduce power use somewhat, but if the pump head and capacity had been selected to match the true system requirements, the pump would have operated closer to its best efficiency and used less energy.

If a pump is oversized and operates with variable flow requirements, it will always be operating at a lower percentage of its capacity at best efficiency. This will lead to exactly the same unfavorable results as using a larger impeller eye area with the same dangers of surges and hydraulic pulsations.

Use an inducer.

An inducer (Figure 7) is a low-head, axial-type impeller with few blades which is located in front of a conventional impeller. By design, it requires considerably less npsh than a conventional impeller, so it can be used to reduce the npsh requirements of a pump (Figure 8), or to let it operate at higher speeds with a given

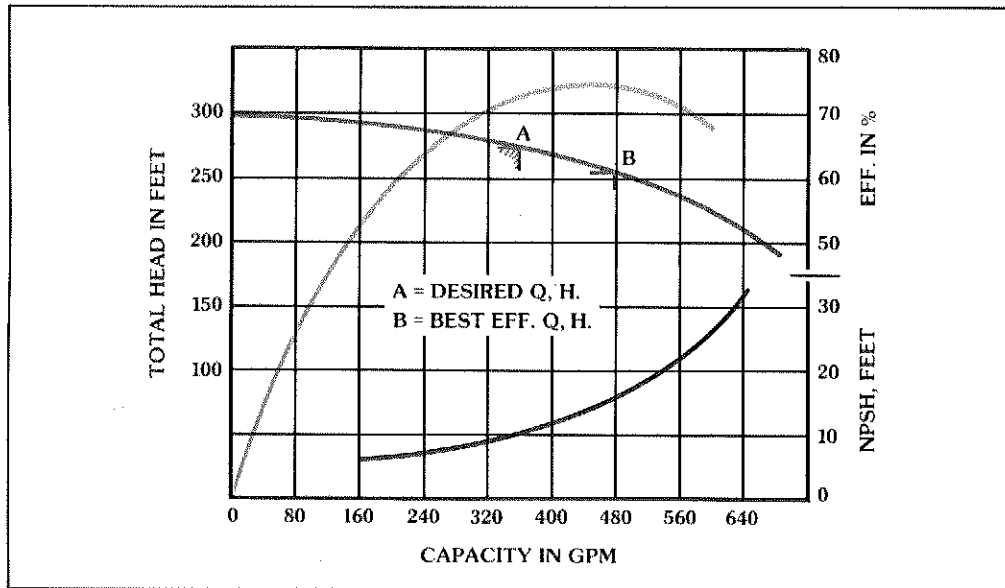


Figure 6—Effect of oversizing a pump.

nps_h. The inducer is a very adequate solution for many low nps_h situations must be applied with care, as the permissible operating range of pumps with inducers is generally narrower than with conventional impellers. (For more on inducers, see Pumpworld, Vol. 4/No. 1978, and Vol. 2/No. 1, 1976.)

Use a booster pump.

The principle of the booster pump is simple. A low-speed, low-head pump, generally of single-stage design, is installed ahead of the main pump to provide it with greater nps_h than can be made available strictly from static elevation differences.

Booster pumps are often used ahead of high-pressure multistage boiler feed pumps. Modern practice uses operating speeds well above 3600 rpm for the feed pumps, so nps_h values may vary

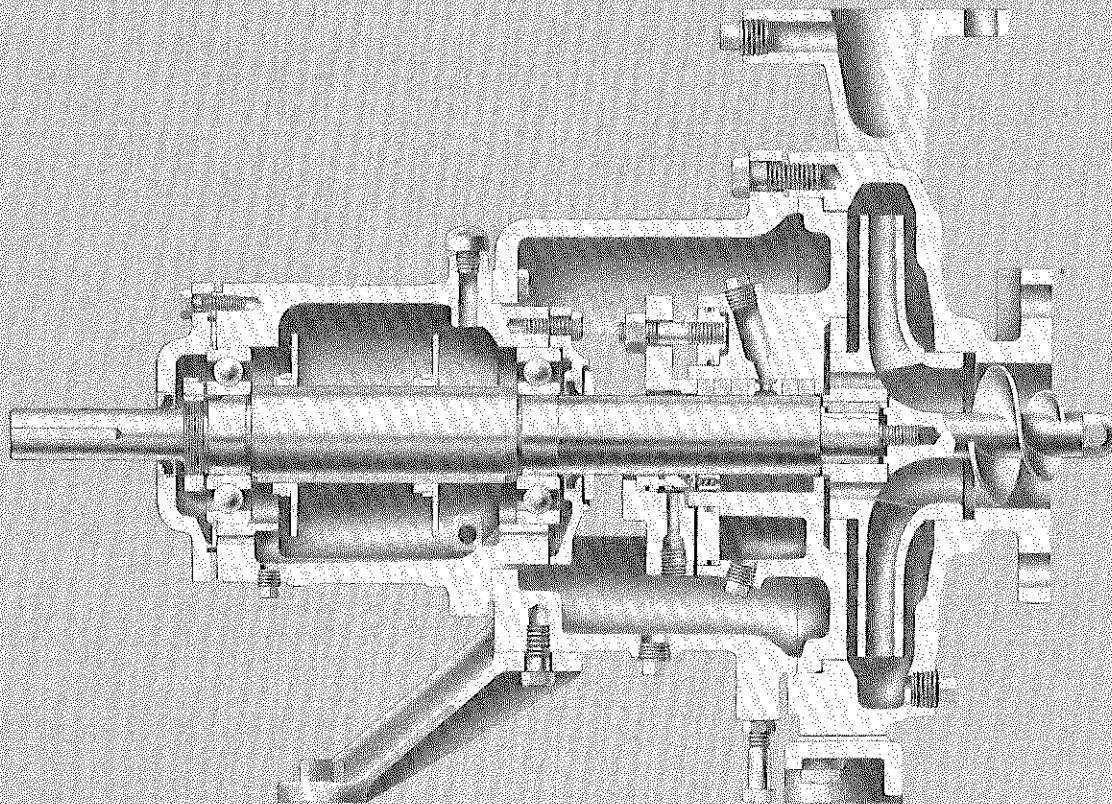


Figure 7—Centrifugal pump with inducer.

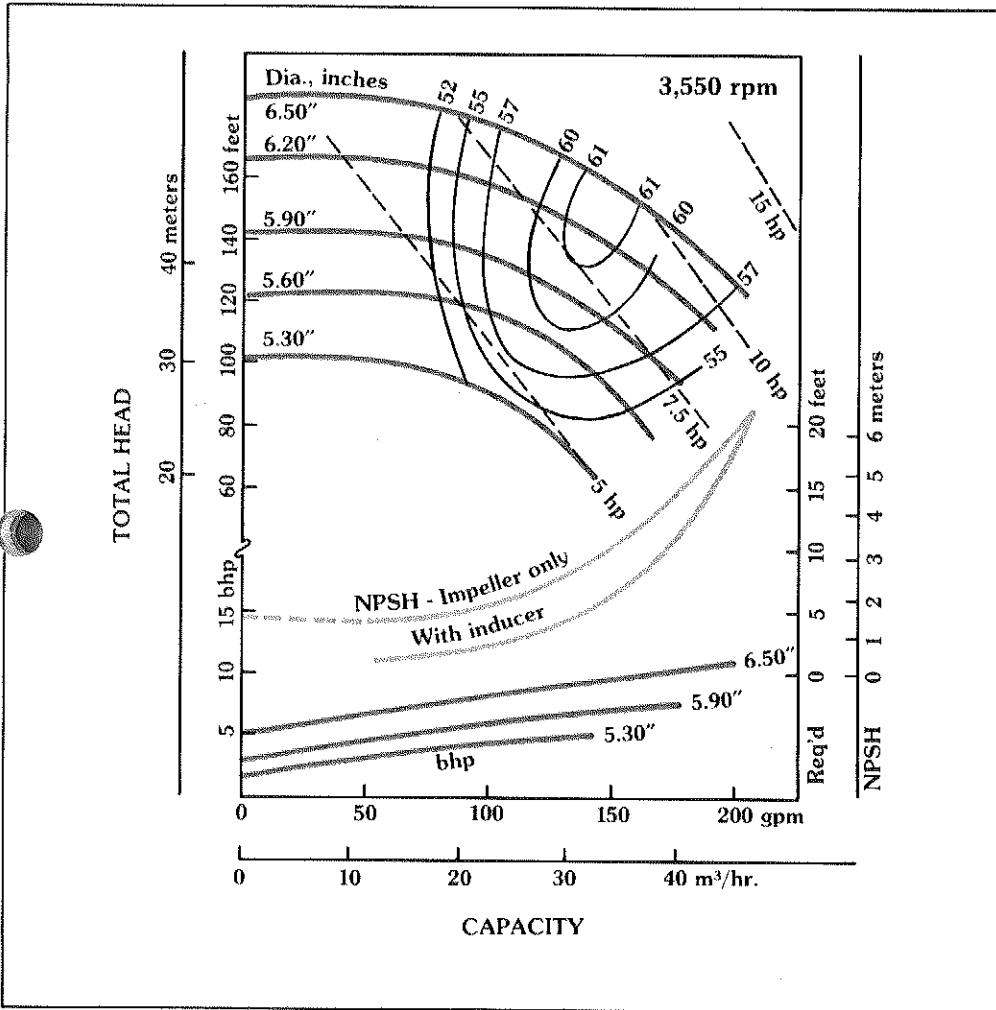


Figure 8—Performance of pump with and without inducer.

high as 150 to 250 feet. It is seldom practical to install the deaerating heaters from which the boiler feed pumps take their suction at an elevation high enough to meet such requirements. On the other hand, single-stage, double-suction booster pumps operating at the lower speed of 1750 rpm may require as little as 25 to 35 feet of npsH, neatly solving the problem.

Subcool the liquid.

Available npsH was defined earlier as $H_{Sr} = Z + (P_s - P_{vp}) - (h_{fs} + h_j)$ and by definition is the energy in excess of the vapor pressure of the liquid at the pumping temperature.

All the methods examined up to now have increased npsHa by increasing the static component Z or reduced npsHr in various ways. There is one other possibility: npsHa may be increased by decreasing the liquid's vapor pressure, which is done by reducing the pumping temperature.

The most obvious way is to inject colder liquid ahead of the pump suction as illustrated in Figure 9. Figure 10 shows the effect of subcooling (or temperature depression) on npsHa at various initial temperatures. Figure 11 shows the temperature depression resulting from cold water injection plotted against the differ-

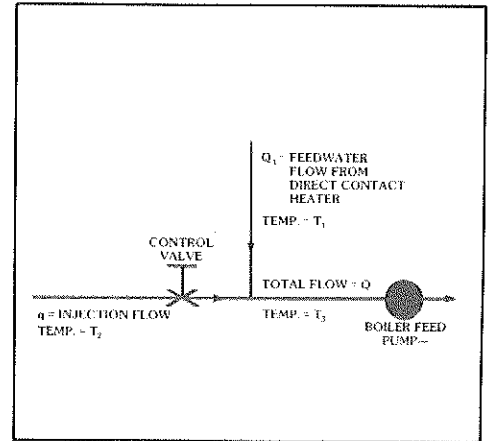


Figure 9—Subcooling by injection of colder water.

ence in temperature between the hot water and the injection stream and given for varying ratios of injection flows.

For instance, we want to provide 20 feet additional npsH to the pump in question, which handles 325°F water. The required temperature depression is 6°F. The injection water temperature is 227°F, giving us a 98°F difference between feed water and injection water temperatures. From Figure 10 we can see the injection flow must be 6.2% of total water flow.

A typical installation using this system might be a steam-electric power plant. The colder injection flow is taken from the upstream side of the feed water cycle, ahead of the deaerating heater. Of course, introducing nondeaerated condensate in the boiler feed pump suction is not recommended, especially in the case of high-pressure boilers where the possible effects of oxygen contamination may be severe. On the other hand, if deaeration in the condenser hotwell is reasonably good, the introduction of a little over 6% water which has bypassed the deaerating heater is occasionally feasible. (This solution is more frequently used as a protection against reduction of npsHa during the transient conditions which accompany sudden load rejections in a steam-electric power plant equipped with a deaerating heater.)

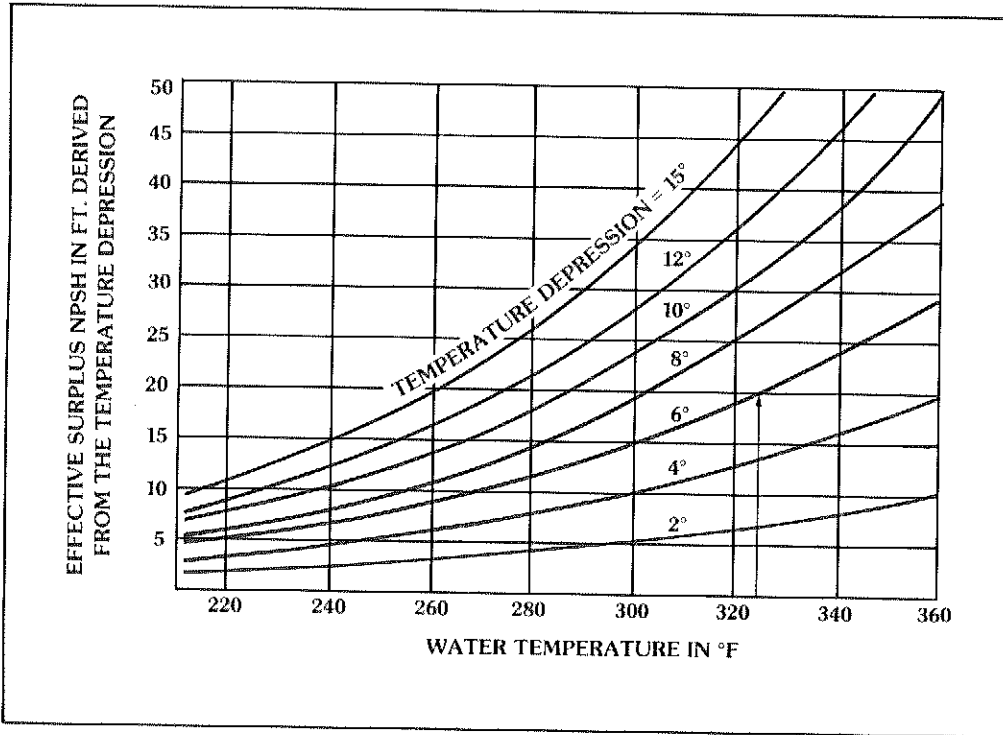


Figure 10—Effect of subcooling on available npsH at various initial water temperatures.

A more highly recommended practice to eliminate all possibility of oxygen contamination by installing a heat exchanger in the suction piping to subcool the deaerated feed water. Friction loss through the heat exchanger would have to be added to the desired increase in npsH and the amount of subcooling determined from Figure 10. To avoid wasting heat, the cooling medium can be the condensate itself on the way to the deaerating heater as shown in Figure 12.

Which way to skin the cat?

The old adage tells us there is more than one way to skin a cat. Likewise, the pump user has many ways to resolve whatever problems may be placed in his way by unfavorable suction conditions. Each solution has disadvantages as well as advantages, and the user must carefully evaluate the solution in relation to his own specific conditions of service.

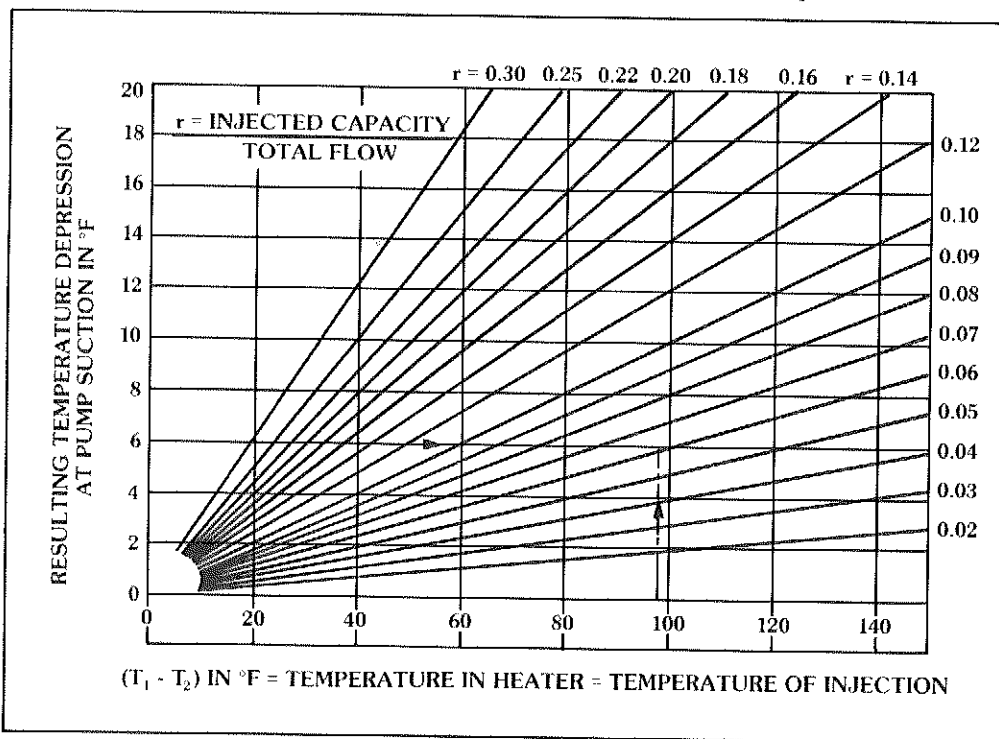


Figure 11—Required amount of cold water injection for a given temperature depression.

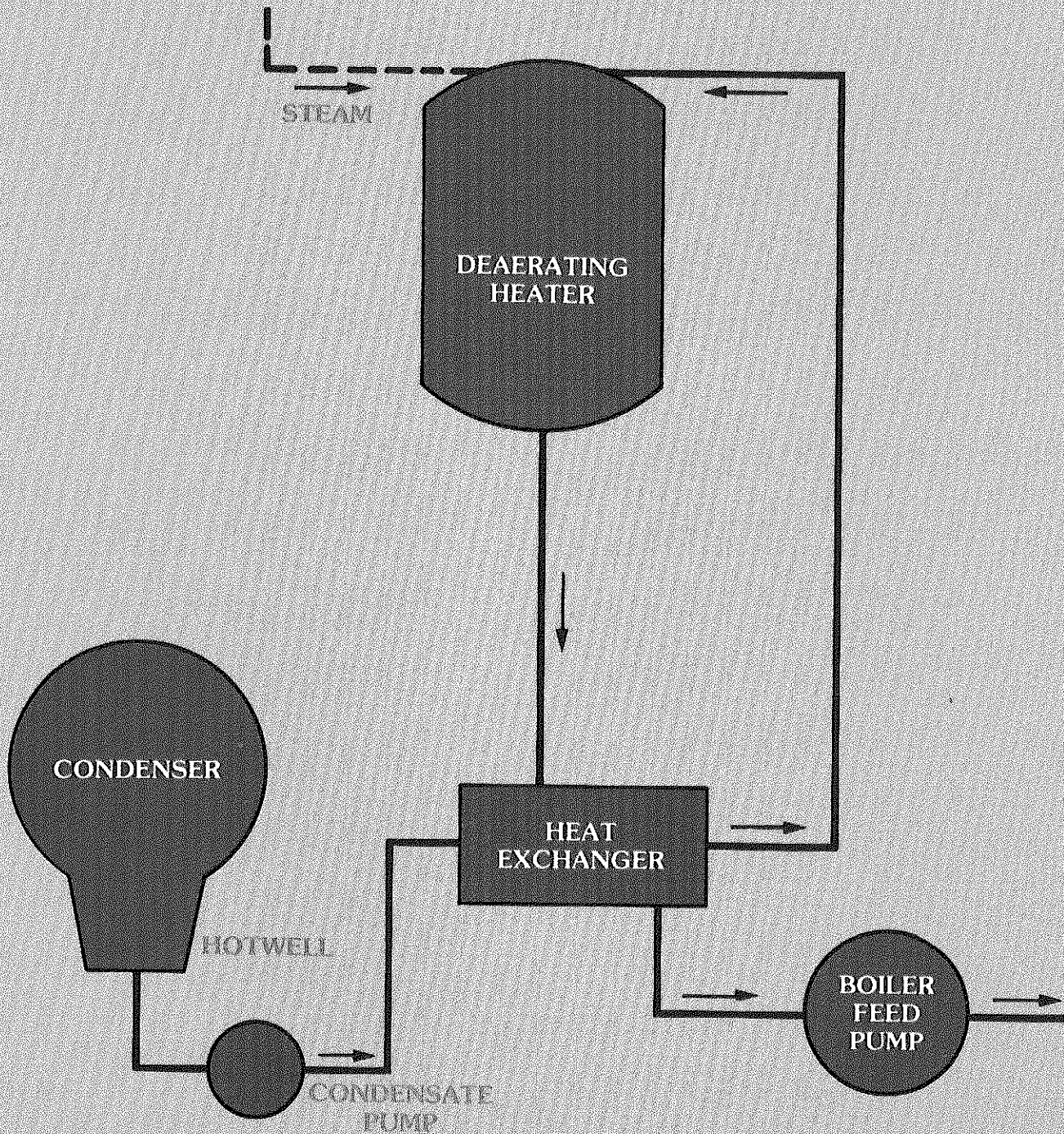
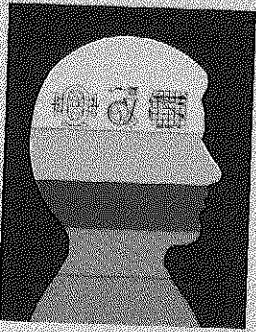


Figure 12—Subcooling system which uses condensate as the cooling medium.



Pump Refresher: The World of Pumps

Part 7

Pumps

Kinetic

Displacement

Centrifugal

Regenerative
Turbine

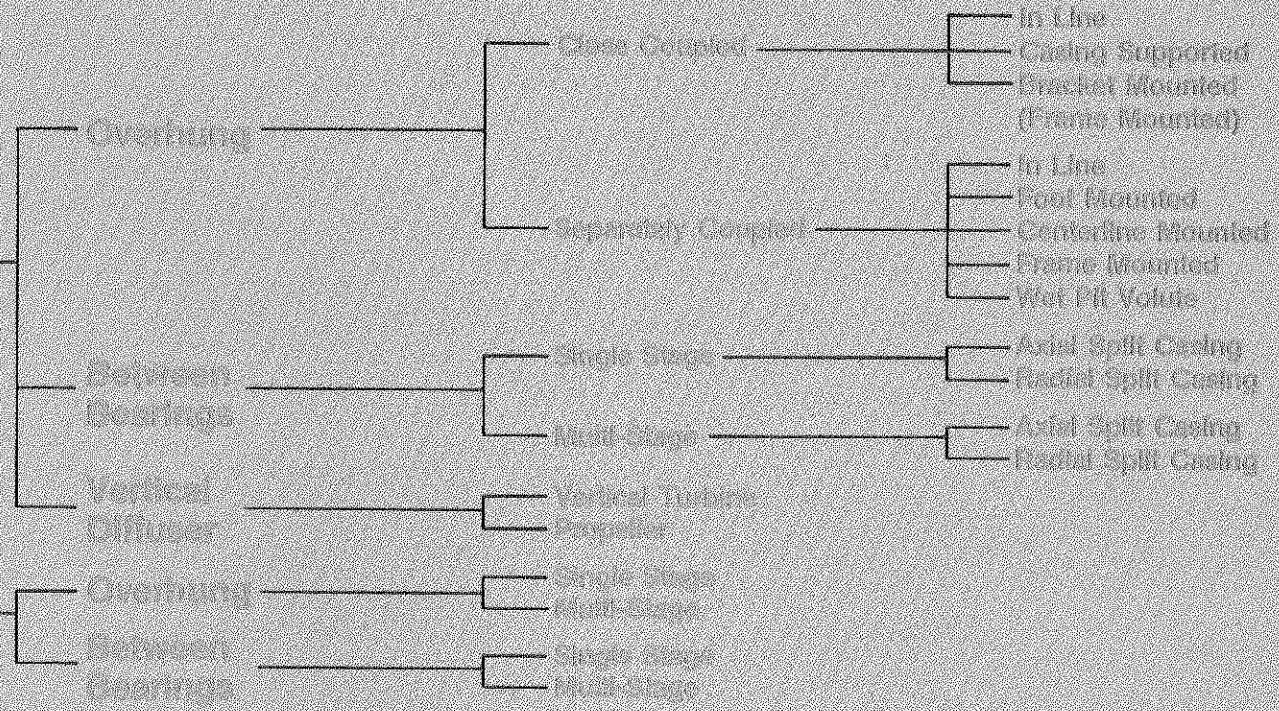
Special
Effect

Reciprocating

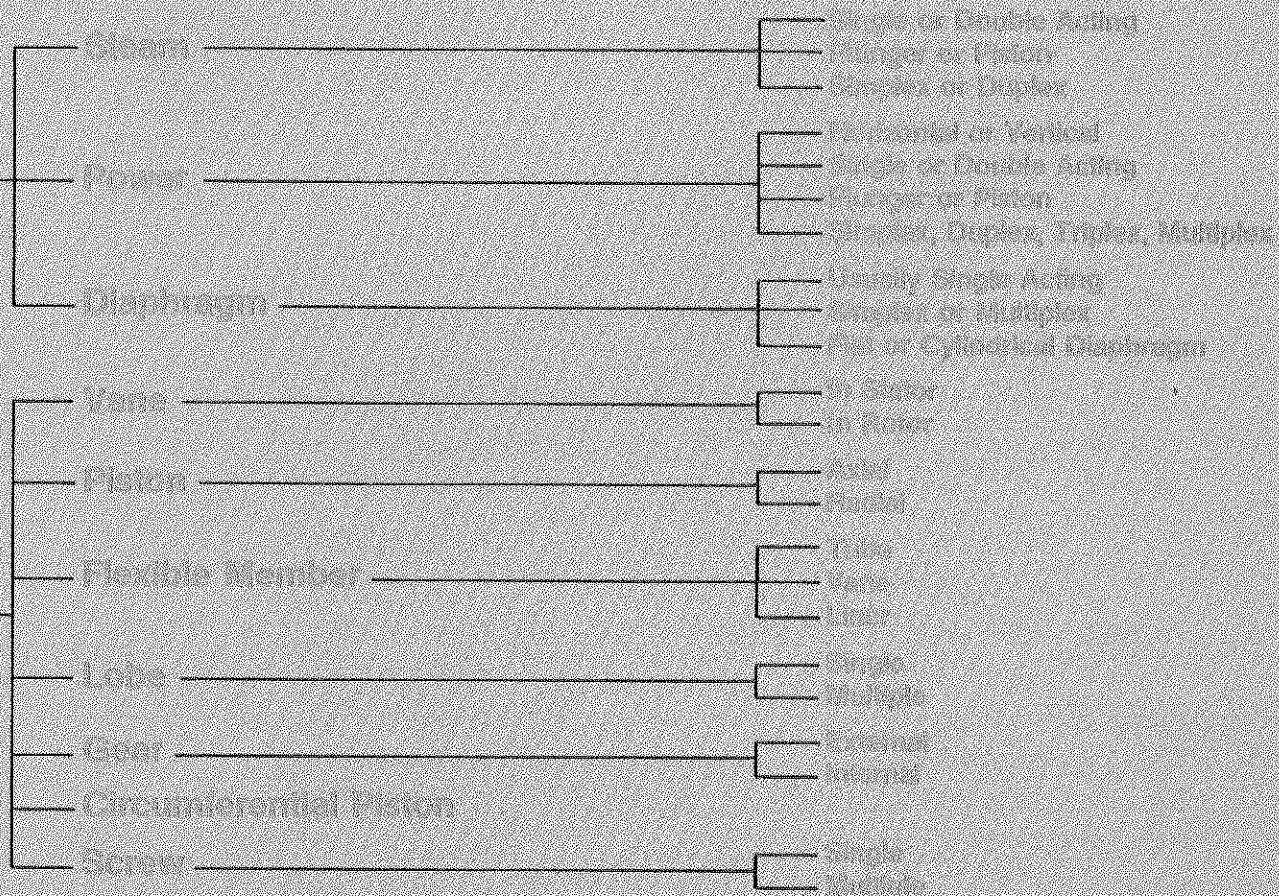
Rotary

In past issues of Pumpworld, we explored basic centrifugal pump principles and properties. The primary topics of discussion included centrifugal pump operating principles (performance and system curves) and basic hydraulics (how fluid flow analysis solves pumping problems and the characteristics of liquids). It is hoped that these refresher series have proved useful.

In future issues we will examine the basic hydraulics of displacement pumps and, subsequently, the specific functions and capabilities designed into all of the various pump types indicated on our chart. In the upcoming issue of Pumpworld, we will continue our journey into "the world of pumps." Our next topic of discussion will be reciprocating pumps.



Reversible Centrifugal
 Compressing-Casing (Piston)



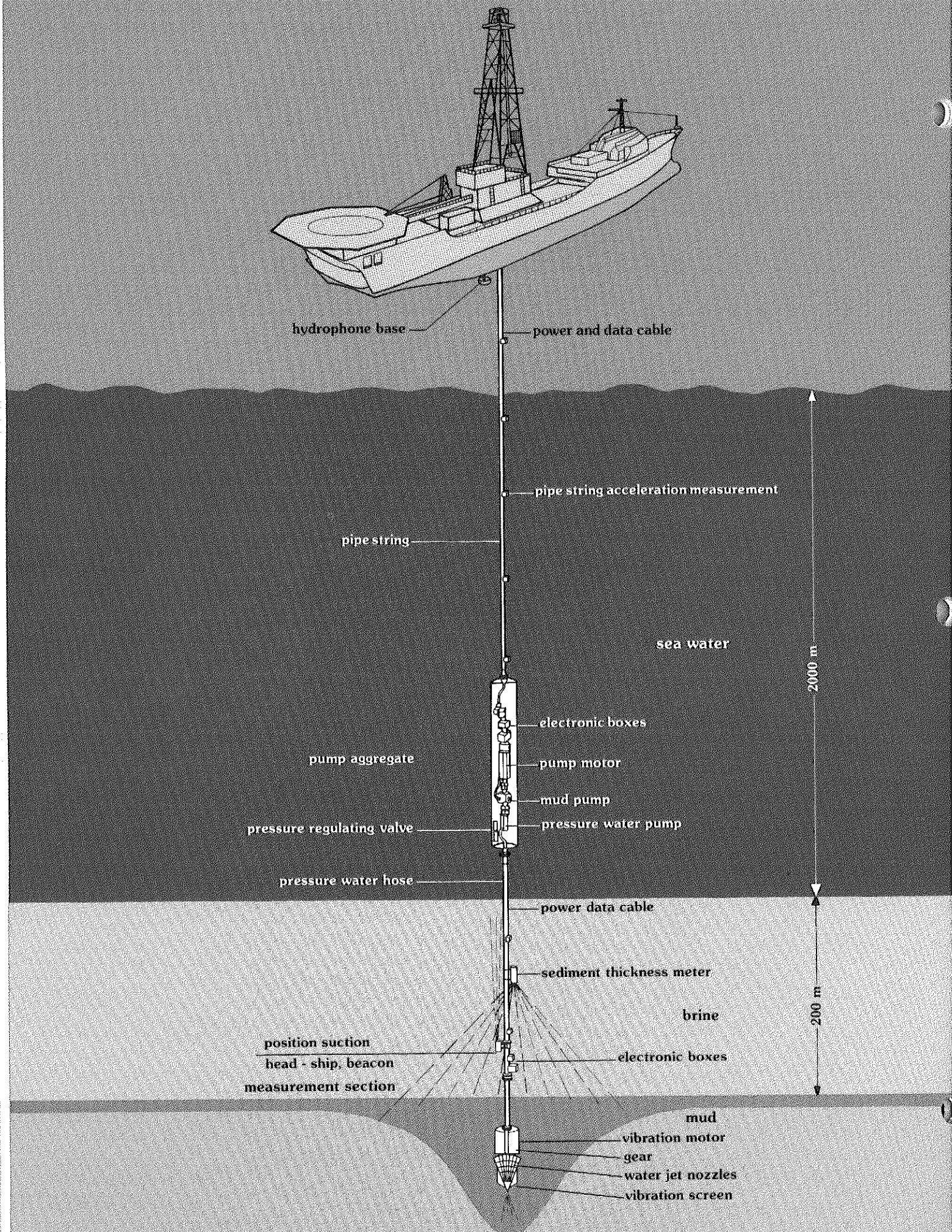


Fig. 1. Diagram of a 2000-m pipe string assembly.

Treasure from the ocean depths: deep-sea mining comes of age in the Red Sea.

By Joachim Knippenberg and Klaus Luck

Like a drowned Aladdin's cave, the Atlantis II Deep region of the Red Sea is piled high with treasure—not precious gems or gold but ore muds rich with millions of tons of copper, zinc, silver, and iron. It's there for the taking, but the task will not be an easy one. The Atlantis II Deep lies nearly 2200 meters beneath the surface of the Red Sea. Now, a deep-sea mining plant is in the making to tap this treasure trove of metal-rich ore. The heart of the system is a specially designed Worthington pipeline pump. Its mission: move the ore-laden sediments from the sea floor, 2200 meters up to the surface, where an offshore processing plant will reclaim the valuable metals.

The Atlantis II Deep is a depression within the central trough of the Red Sea. At its lowest point, 2200 meters below sea level, hot brines with high metal content are extruded from sources beneath the ocean floor. As the flow cools and contacts the oxygenated seawater, dissolved metals precipitate out to settle on the sea floor, literally filling the Deep with ore.

During exploratory cruises, research vessels determined that these metal-rich sediments cover the bottom of the Atlantis II Deep with a deposit ranging from a few meters to as much as 30 meters, the height of a 10-story building. In an area of about 60 square kilometers lie some 30,000,000 tons of iron, 2,500,000 tons of zinc, half a million tons of copper, and 9,000 tons of silver.

Joachim Knippenberg is managing engineer for Deutsche Worthington G.m.b.H., Hamburg, Germany; Klaus Luck is an engineer in the marine technology department of Preussag A.G., Hannover, Germany.

Once discovered, such riches could not remain unclaimed for long. The Saudi-Sudanese Red Sea Joint Commission appointed Preussag A.G. of Hannover, West Germany, to find a way to tap this natural resource in a prepilot mining plant. In December, 1977, Deutsche Worthington was awarded a contract by Preussag to develop a pump package capable of conveying the sediments from the sea floor to surface.

The first functional test of the mining plant was accomplished with good results in 100 meters of water off the Scottish coast. Next, the plant performed successfully in a series of tests in the Red Sea.

Design of the deep-sea mining plant.

The plant is incorporated in a conventional drill vessel. The 2200-meter-long mining pipe assembly, integrated suction head, electric motor-driven pump train, and measuring instruments hang in the derrick well. In operation the mining assembly is lowered through the ship's "moon pool," or drilling bay, length by length to the ocean floor.

At a water depth of about 1950 meters, just above the upper brine level of the Atlantis II Deep, a 15-ton pump/instrumentation assembly is suspended from the 5-inch diameter drill pipe. From the measuring devices, mining data will be sent to a process computer on the deck of the vessel.

The specially-developed vibration screen suction head has been designed to penetrate into the layers of sediment. A flush pump cycles water to the suction head where the vibration screen fluidifies the thick mud for transport by the main mud pump. Other major components of the system include driver, a submersible electric motor, the water regulating system, and electronic control equipment. A deep-sea power cable, which supplies the pump driver and also serves for data transmission, is attached outside the pipe assembly, along with the water hose from the flush pump down to the suction head. Power for the submersible motor, instruments, and regulation components is

provided by a 1000-kw, frequency-regulated generator package and transformer on the deck of the drill ship.

Processing the ore.

Once brought to the surface, the mined mud is stored in tanks aboard the drill vessel and processed to ore concentrate in a parallel operation. Residues of the processing plant are discharged back to the sea at a water depth of 400 meters. A separate environmental research program is under way to minimize the impact of these residues.

Pump/instrumentation assembly design.

Components of the 15-ton pump/instrument unit are integrated in a steel frame with flanges on either end to connect with the suction and discharge pipe sections. The mud pump, flush pump, and their common driver are mounted in the lower half of the steel frame. On the upper end of the frame, watertight, oil-filled, pressure-compensated boxes contain connections for the power cable as well as signal cables from transducers sensing temperature, bearing vibration, flow density, and other parameters.

The Worthington mud pump.

In order to convey the valuable sediment from the bottom of the sea, Worthington engineers modified a 6-stage, horizontally-split pipeline-type pump. The pump provides a flow rate ranging from 33 to 70 m³/hr at a head of 660 meters. Rotation speed is variable from 2850 to 3560 rpm, averaging 3382 rpm. The average specific gravity of the flow mixture is 1.25.

The highly abrasive, metal-bearing mud has a consistency ranging from muddy to greasy, like shoe polish or peat. Most of the solids are smaller than two microns in diameter, and only 10% are over 20 microns. Nevertheless, the pump must be able to handle solid particles up to 10 mm in diameter, since small basalt stones will be encountered in the mud at the borders of the central trough. For positive protection from the abrasive mud, the pump's bearings are oil filled and pressure compensated.

The driver is a 3000-v, 60-c, 535-kw submersible motor, coupled to the pump by a water-lubricated, toothed coupling. The motor is filled with fresh water and pressure compensated. The flush pump is an 8-stage submersible, coupled to the end of the mud-pump shaft and driven by the same motor.

Operation of the mining plant.

A standard drill ship was specially modified to house the deep-sea mining plant. When the ship is under way, the pump unit is stored vertically in the derrick, ready to be lowered along with a deep-sea cable for power supply and data transmission. To operate the mining plant, a 200-meter

suction pipe (approx. 22 pipe sections @ 9m each) is attached to the lower end of the pump unit. With a drill pipe bolted to the upper end, the assembly is gradually lowered as pipe length after pipe length is connected, until the cutter head reaches the ore mud.

As demand increases, the race to tap more and more of the ocean's mineral resources accelerates. Increasingly, we must look farther, dig deeper . . . produce better yield from known reserves.

At this moment vast areas of the ocean's floor remain unproductive. However, in the Deep II region of the Red Sea, deep-sea mining has truly come of age.

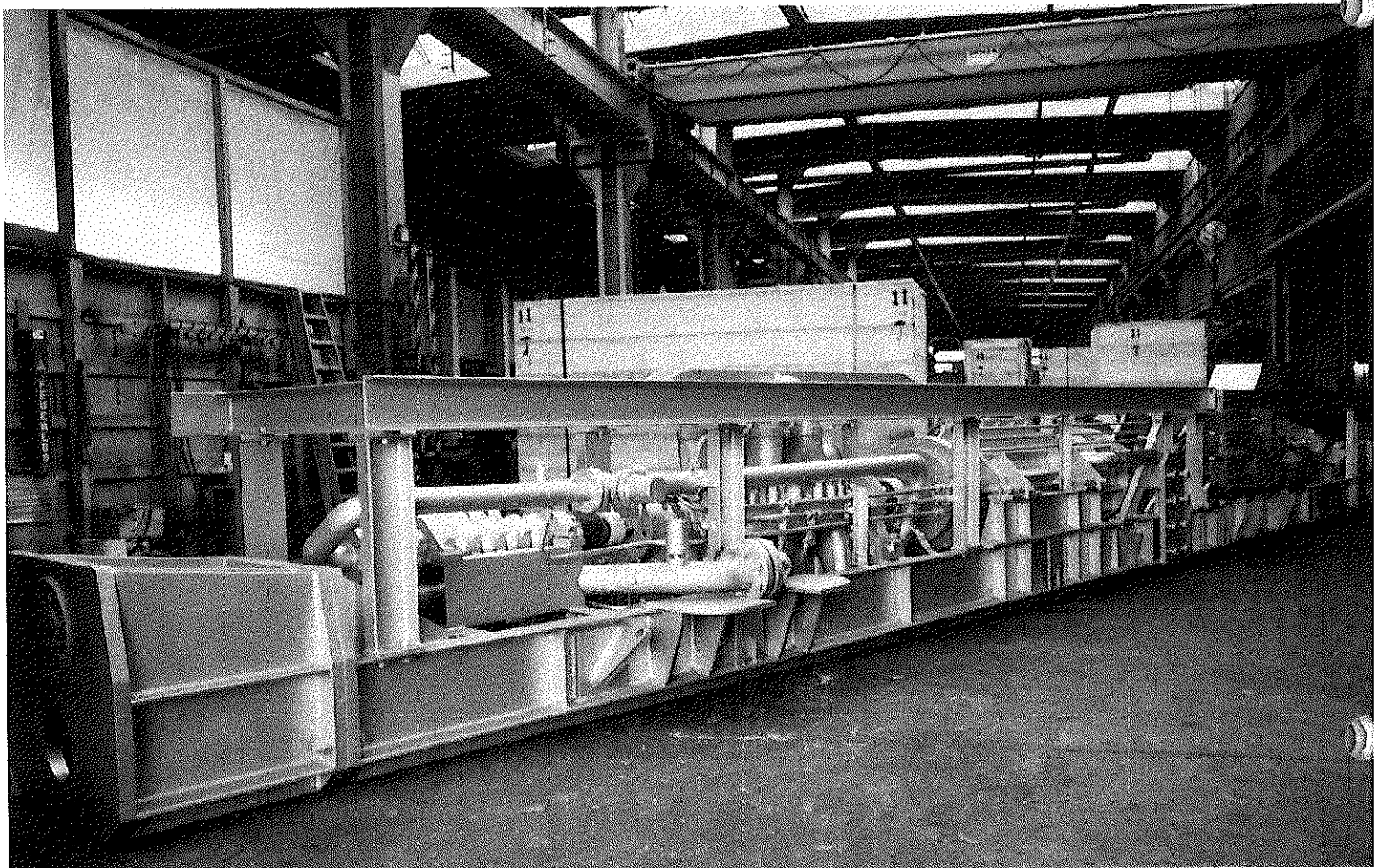


Figure 2-4 UX-1 pump unit with submersible motor drive for deep sea mining application. Unit will operate at 2000 m (6500 feet) submergence.

Mechanical seal covers eliminate flushing, prolong seal life.

By Ed Simin

Even though popularity of the mechanical end face type seal has increased substantially in recent years, the influence of the stuffing box is still inherent in most industrial pump designs. In fact, most mechanical seal manufacturers have had to design their seal to fit into the pump manufacturer's stuffing box specification designed primarily for the tight annular space that is more practical for packing (see "What pump design does to seals"). This has caused compromises in some seal designs and has created the need for additional hardware when a mechanical seal is to be used (circulating and flush piping plans, different gland styles, throat bushings, orifices, heat exchangers, cooling jackets, etc.). Although the need for such items will always exist in the most special applications, the majority could be simplified by use of the mechanical seal cover.

While most pump manufacturers have begun to standardize the stuffing box, enabling it to accept both packing and mechanical seals, the cost of such adaptability, along with added requirement for hardware, is passed on to the user. The basic end face mechanical seal cover design can be functionally more simple, less costly, and will create a better sealing environment, resulting in longer seal life.

A mechanical seal housing or casing as shown in Figure 1 will satisfy the basic requirements of housing the seal and containing the stationary seat,

Ed Simin is a development engineer for Worthington Pump Corporation.

serving as a removable casing closure, providing means for circulating liquid around the seal faces to dissipate heat (one of the most frequent causes of seal failures), and providing adequate space around the seal where solids and particles in suspension settle out far away from the seal faces.

How the mechanical seal cover works.

Briefly, let's examine how such a mechanical seal cover works. Figure 1 illustrates a closed impeller pump with wearing ring. The clearance between the impeller hub diameter and the wearing ring bore breaks the casing pressure down to a much lower pressure in the cavity where the seal is housed. The pressure in the cavity is further relieved by the impeller balance holes which reduces pressure to within about one atmosphere above suction pressure. This greatly reduces the sealing pressure requirements of the mechanical seal and results in a lighter face loading, less heat generation, and is, therefore, more suitable for most unbalanced and usually less costly seals. Balanced mechanical seals reduce this face loading even further and are recommended for higher sealing pressures.

How the mechanical seal cover works.

As liquid flows from the wearing ring to the cover, a continuous, evenly distributed flow pattern is directed toward the mechanical seal which performs an internal flushing, cooling, and self-cleaning function automatically whenever the pump is running. Compared to external flushing means (Figure 2), the internal flushing pattern created by the wearing ring minimizes the erosion and damage which can be caused by a concentrated jet of flush liquid carrying particles or abrasives. Furthermore, the impeller hub is continuously rotating, minimizing clogging. Flush lines, particularly when used with a line orifice, do not afford us

this luxury. In fact, wear increases simultaneously with the flushing action because of increased clearances created by the wear ring. Internal flushing from the rear wearing ring also has the advantage of keeping out large, unwanted debris which can damage or interfere with the more delicate parts of the seal. In effect, it acts as a large, built-in dynamic flushing orifice.

Other advantages.

A mechanical seal gland is not normally required because there is no opening to seal as with a stuffing box cover. Along with the absence of a flush line, it minimizes the potential for leakage and seal misalignment due to improper assembly. However, for liquids that crystallize or solidify on atmospheric contact, an auxiliary quenching arrangement is recommended as illustrated in Figure 3. This arrangement will prolong seal life considerably by rinsing and dissolving particles away which could otherwise build up and work their way into the seal faces from the atmospheric side of the seal. The quenching arrangement is usually water, steam, or some other solvent circulated at very low pressures and flows, just enough to avoid the buildup of solidified product. Usually a low-pressure sealing device, such as a lip seal, packing ring, or throttle bushing, serves to act as the secondary seal for the quenching liquid. In the event of a mechanical seal failure, the secondary seal can serve as a temporary backup system until a seal replacement can be made.

When sealing liquids that solidify close to pumping temperatures, a temperature drop is more likely to occur with a long stuffing box containing a small volume of liquid as opposed to the generous quantity of liquid surrounding and flushing a seal in the mechanical seal cover at maximum product temperatures. The temperature drop in a stuffing box must be overcome by the added requirement of stuffing box heating or heat tracing.

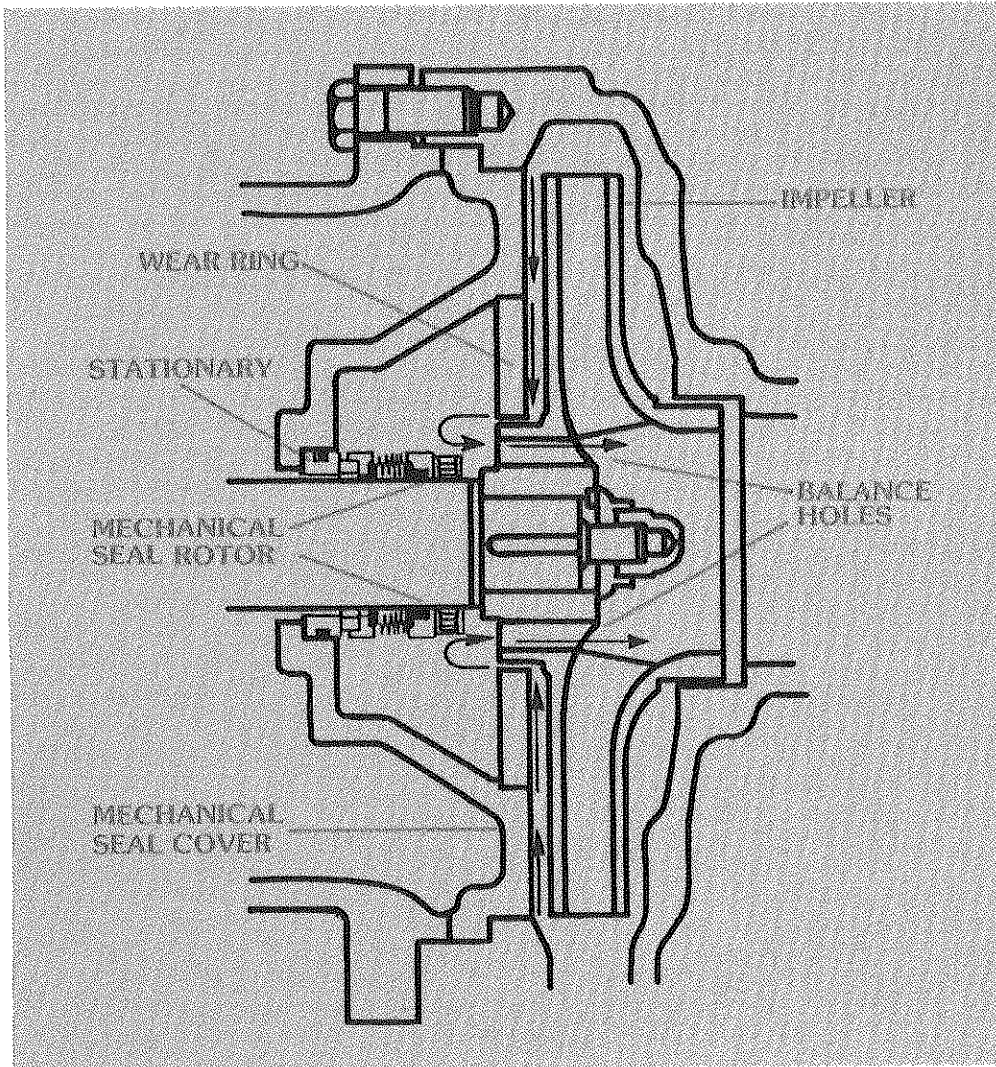


Figure 1—The mechanical seal cover design provides a better seal environment for greater dependability. The self-flushing action keeps the seal running cooler and cleaner, ensuring longer seal life. It also eliminates the need for a recirculation line and mechanical seal gland.

A pump which uses a self-flushing mechanical seal cover can provide higher product temperatures. This is not the case with a pump using a stuffing box (i.e., without external flushing).

The mechanical seal cover eliminates most air vortexing or air entrapment. It is essentially self-venting as opposed to most stuffing box cover arrangements fitted with double mechanical seals or throat bushings, which are not. Even when mounted in a vertical arrangement,

air removal is almost immediate and complete at start-up by the flushing action created by the wearing ring.

Most abrasive applications can be better handled by using a mechanical seal cover installed with a single mechanical seal, and by upgrading seal faces to harder materials such as tungsten and/or silicon carbides. Such an arrangement is more reliable and can be more economical than a double sealing arrangement because it does not require

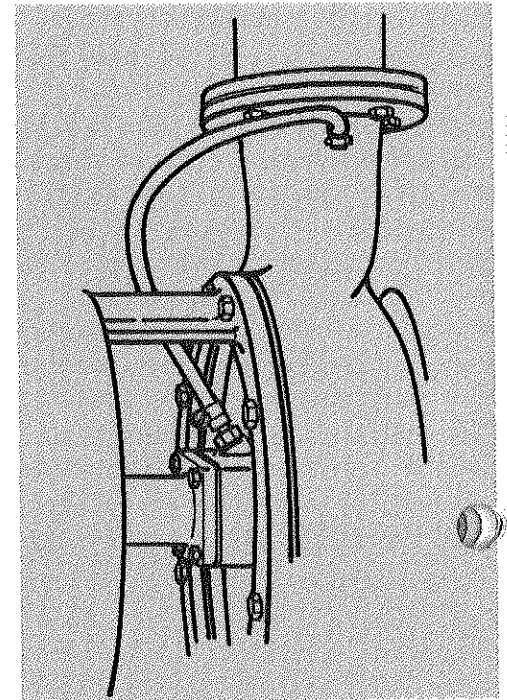


Figure 2 — External flushing lines and attendant hazards are eliminated by the mechanical seal cover.

barrier fluid to be maintained above product pressure and circulated in adequate quantity.

Gummy or muddy-type liquids, if they have sufficient face-lubricating qualities, lend themselves better to a nonlogging-type mechanical seal cover. Here, the large volume around the seal is critical for keeping solids and sediments from settling out on the working parts of the seal.

Adaptability.

The mechanical seal cover as described above can be adapted to a variety of pumping configurations: horizontal frame-mounted, horizontal close-coupled, vertical close-coupled, and vertical frame-mounted.

The frame-mounted vertical and horizontal arrangements have the advantages and flexibility of providing greater freedom of driver selection, casing, or drive replacement without

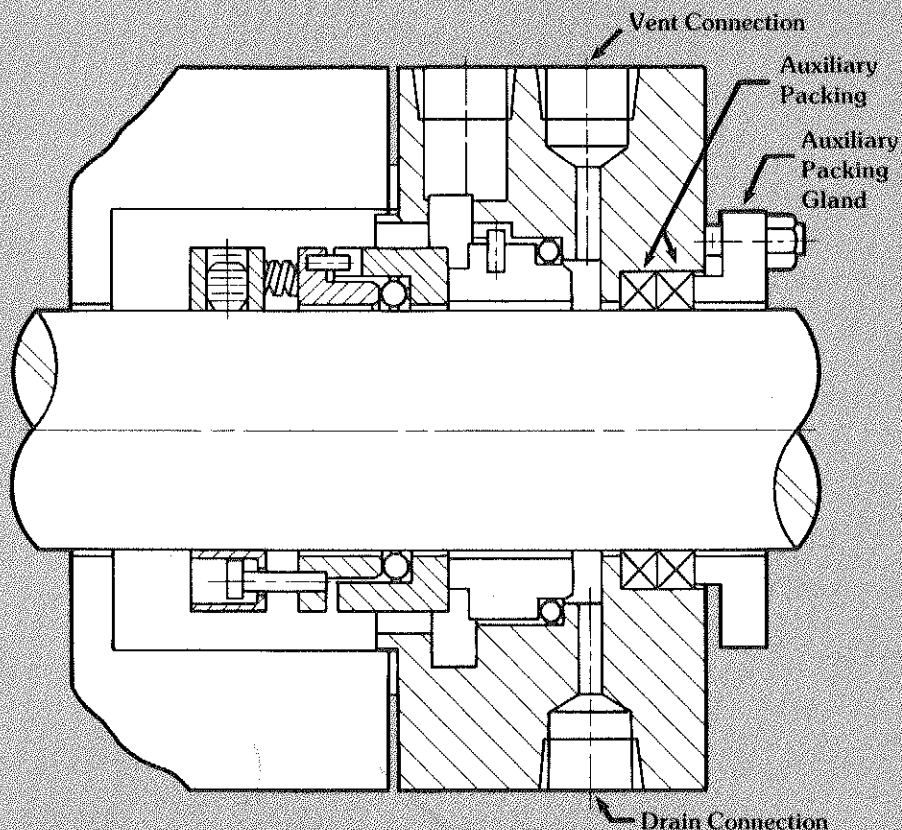


Figure 3—Auxiliary quenching arrangement.

disturbing the pump. The close-coupled horizontal and vertical arrangements have the advantages of compactness, usually lower initial cost, and few, if any, misalignment problems.

Reduce problems and costs.

By eliminating the external flush lines and using the mechanical seal cover's inherent design advantages, pumping problems and pumping costs can be reduced in many applications:

Energy is conserved by not recirculating additional product for flushing. Use of existing internal recirculating does not

result in added efficiency penalties.

Auxiliary pumping systems are eliminated, as are associated operating expenses required to maintain flush liquid at controlled pressures and flows.

Product is conserved by elimination of leakage caused by packing.

Overall process efficiency and purity is improved by adding flush water (if it must be removed later in the process) to the product.

Maintenance costs are reduced because of less downtime resulting from better and more reliable sealing.

Initial cost for overall equipment is reduced due to simplified pump design, elimination of flushing hardware, and supporting systems.

Most pumping applications dealing with non-hazardous, non-toxic liquids (below 300°F and below 275 psi) can be handled without external flushing lines. Utilizing a mechanical seal cover designed to be self-flushing with a cooler and cleaner seal environment will ensure longer seal life. It can also satisfy most pumping applications being handled by a packed stuffing box which will, by design, leak fluid.

What pump design does to seals.

J.H. Doolin

The most significant part of any pump is the impeller, which has a strong influence on the stuffing box pressure. The single-stage end-suction type pump is a frequent user of seals (see Figure 1). Here the seal is usually mounted on the side of the impeller opposite the suction. The pressure on the seal can vary anywhere between suction and discharge pressure. In the case of the closed-type impeller with a full back shroud, the area close to the shaft is open directly to discharge pressure, and the resultant pressure is very high. However, the high velocity in the volute and the rotational effect of the water reduce the pressure at the shaft to approximately seven tenths of the total head plus suction pressure.

The high discharge pressure at the shaft can be reduced near suction pressure by adding a close-fitting wearing ring on the back of the impeller and hydraulic balance holes. Here, the final pressure attained is dependent upon the relative area of the wearing ring clearance and impeller balance holes. For best results, the balance holes area should be at least twice the ring clearance area. However, as the impeller wears, the clearance increases, and the stuffing box pressure will increase. For example, Figure 2 shows the effect of ring clearance wear based on an initial clearance of .015". This curve is based on the simple relationship that $V=C(2gH)^{1/2}$ where V is the velocity through the clearance, H is the head across it, and C is an empirical coefficient.

The most predictable design exists in pumps where side suction leads directly to the stuffing box side of the impeller. The most common example of this arrangement is the horizontal split-case, double-suction pump (see Figure 3). Here the stuffing box pressure is almost equal to suction pressure. In addition, some single and two-stage overhung designs are

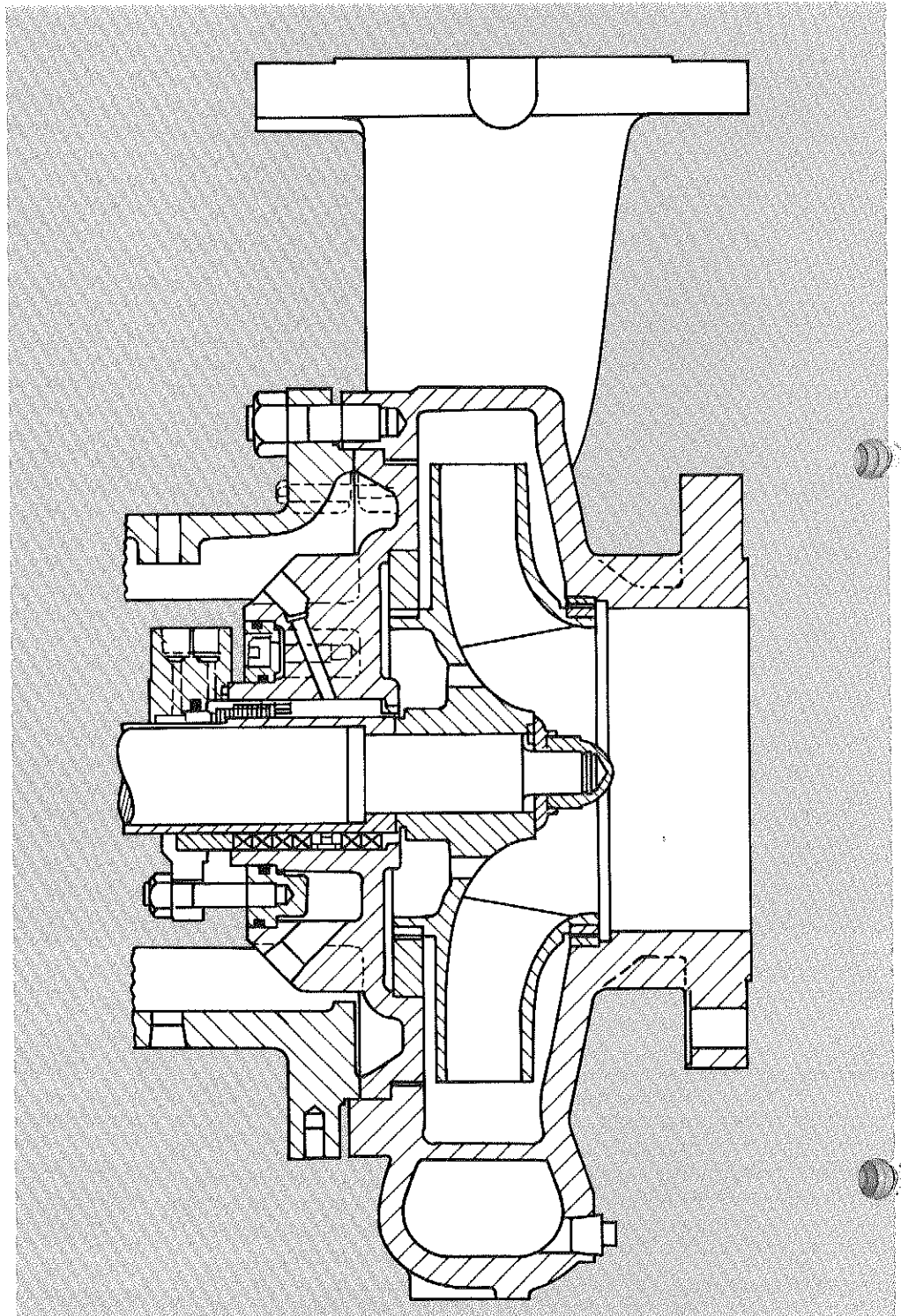


Figure 1—D-1000 closed impeller liquid end.

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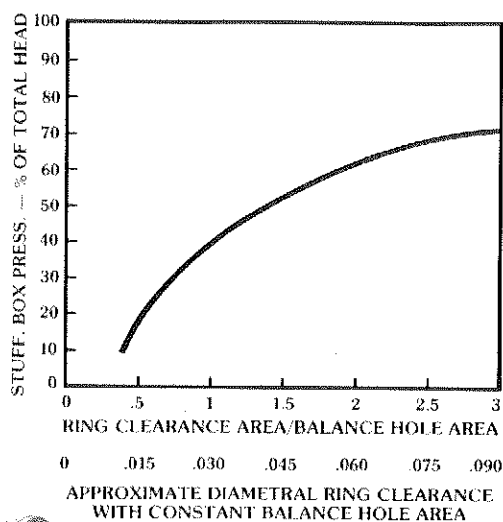


Figure 2—Stuffing box pressures versus wearing ring clearance.

arranged with this type of inlet configuration. Another common type of impeller to consider is the open or semi-open type. The semi-open impeller is designed with a full back shroud and no front shroud (see Figure 4). Consequently, from the stuffing box it is no different from the closed impeller. Like the closed impeller, a back wearing ring and balance holes are usually used. The full open impeller has no shroud surrounding the vanes, but usually has a partial web or shroud between the blades for support (see Figure 5). With this configuration and ample balance holes in the web, the pressure on the back of the impeller is very close to the suction pressure. In fact, the pumping action of the blades on the back can result in stuffing box pressures below the suction pressure. Tests show a very close relationship between stuffing box pressure and suction pressure.

Mechanical considerations.

prime sources of difficulty which end-face seals encounter are end play in the shaft mounting and thrust reversals in the pump, which causes shaft movement. The most common shaft mounting is two deep-groove ball bearings, one of which is locked to the shaft and housing to take

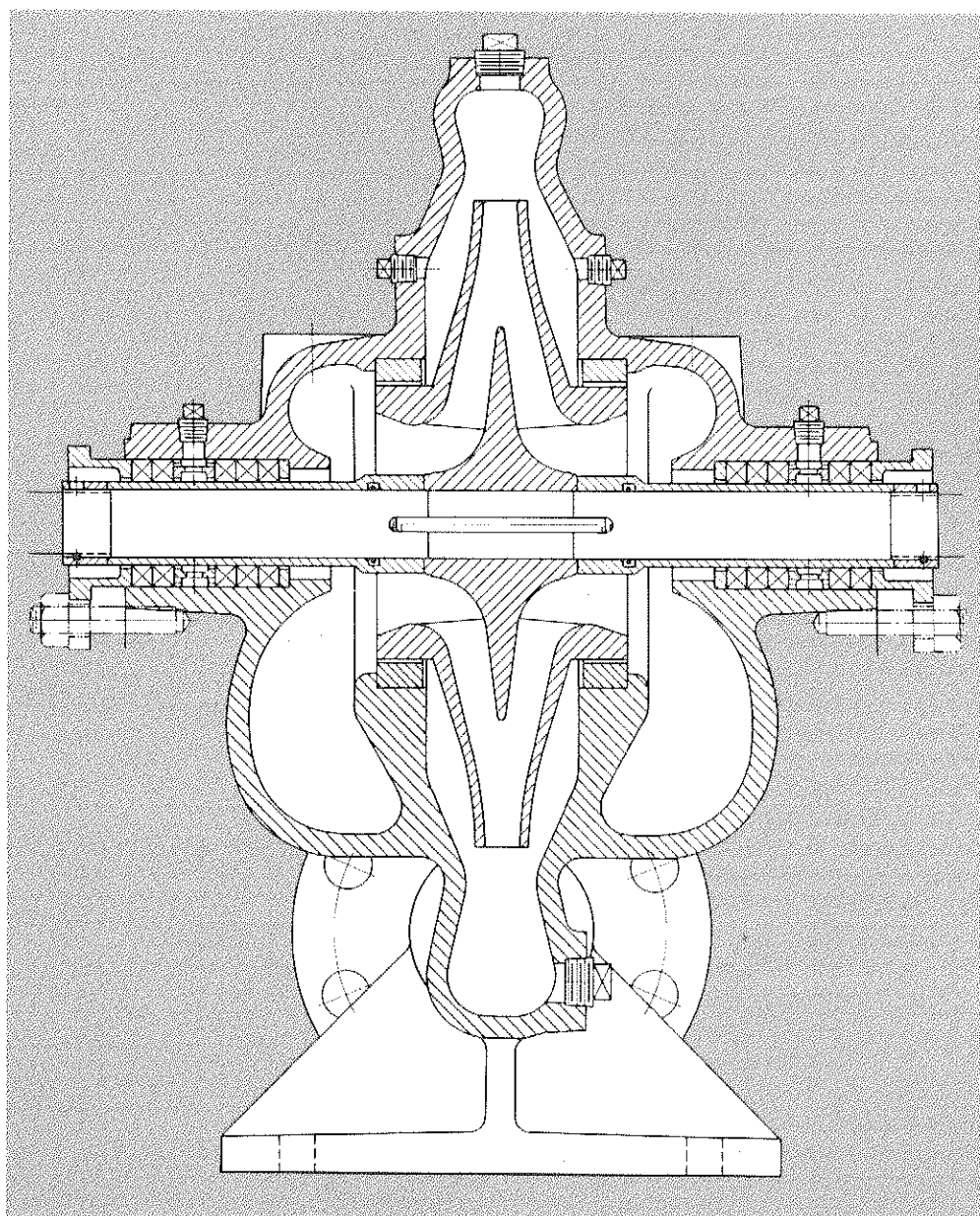


Figure 3—Double suction pump.

thrust in either direction. In this arrangement, end play does not exceed that allowed by the internal clearance of the bearings (this does not exceed .010" for a new bearing). This maximum end play could be greater if both bearings are allowed a limited float, as in some designs. On the other hand, it can be reduced if

angular contact bearings or other, more limited end-play bearings are used. Thrust reversal on bearings occurs when the net result of the acting forces changes direction. Generally, these forces come from the following causes: (A) The impulse force resulting from the change in momentum of the fluid with the velocity

change from axial to radial. This force is equal to the mass rate of flow in pounds per second times the velocity at the impeller eye divided by g , (B) The force obtained from suction pressure acting on the end of the shaft toward the thrust bearing and in the same direction as A, (C) The unbalance of pressure on the impeller shrouds which is somewhat unpredictable but usually acts toward the pump suction. When $A + B = C$, the net force is zero. Any change in conditions could allow the net force to change magnitude and direction. Figure 6 illustrates the variation of these forces with flow. It also illustrates how this thrust reversal takes place. If there is end play in the bearings, this means there is limited axial movement and sometimes an opening of the seal faces.

Shaft deflection due to hydraulic radial unbalance on an impeller also contributes to seal difficulties. This radial unbalance results from the uneven pressure distribution around the impeller when the pump is operated at other than its design point. The radial load becomes greatest at the shut-off or zero-flow condition. Generally, the radial load on the impeller is determined from the formula $W = kpdb$ where:

- W = radial load, pounds
- k = empirical factor
- p = pump total head, psi
- d = impeller diameter, inches
- b = impeller width, inches

The factor K varies between zero and about .36 and is determined by impeller and casing configuration, and the point of operation on the curve. Generally, well-designed pumps can withstand these loads without undue distortion of parts. However, even in properly designed pumps, this load tends to deflect the shaft radially — and deflection of the shaft at the stuffing box will take place. This causes radial movement of the running faces and can cause leakage in a seal due to the realignment of the wear pattern. Radial shaft deflection is also accompanied by

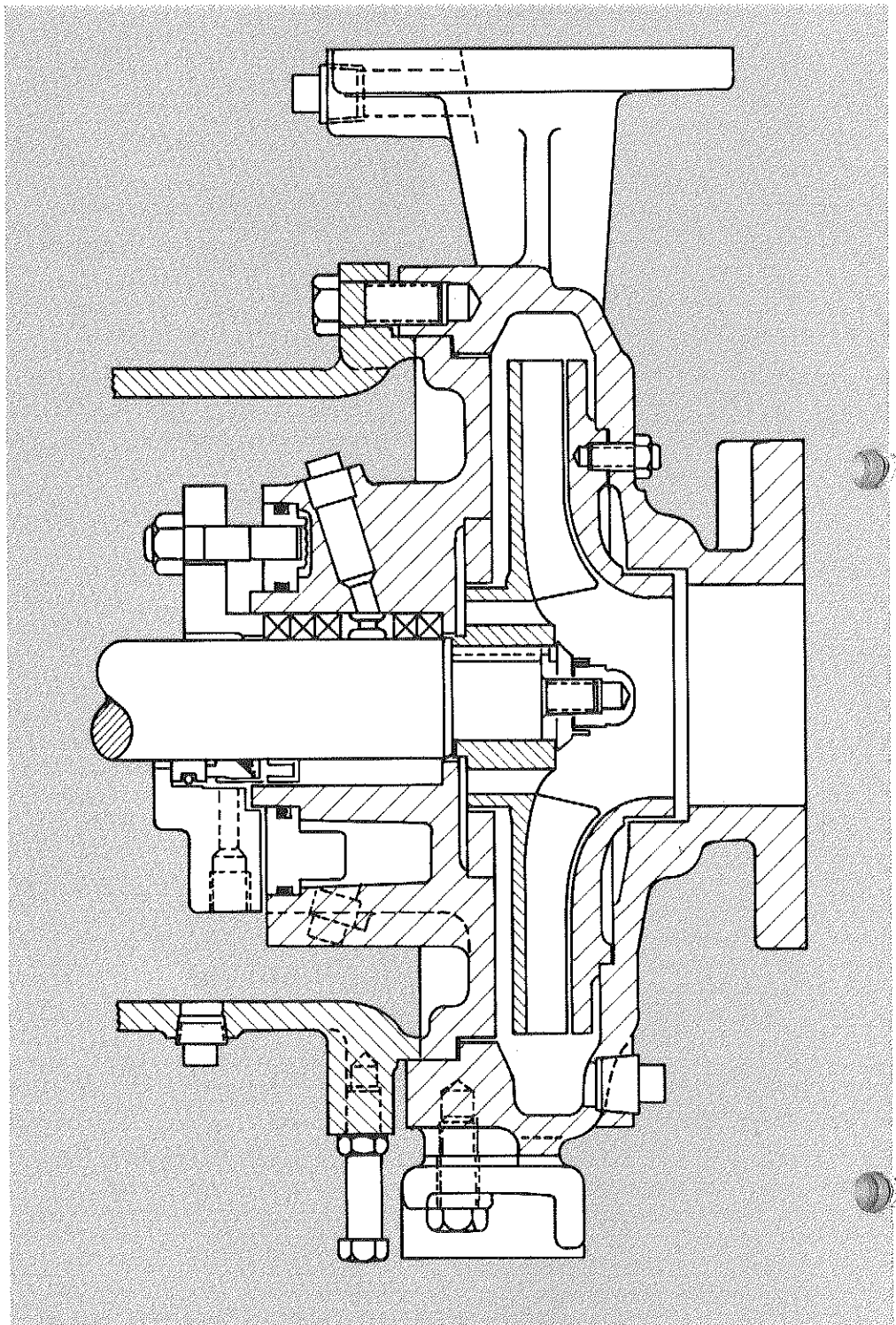


Figure 4—Semi-open impeller pump.

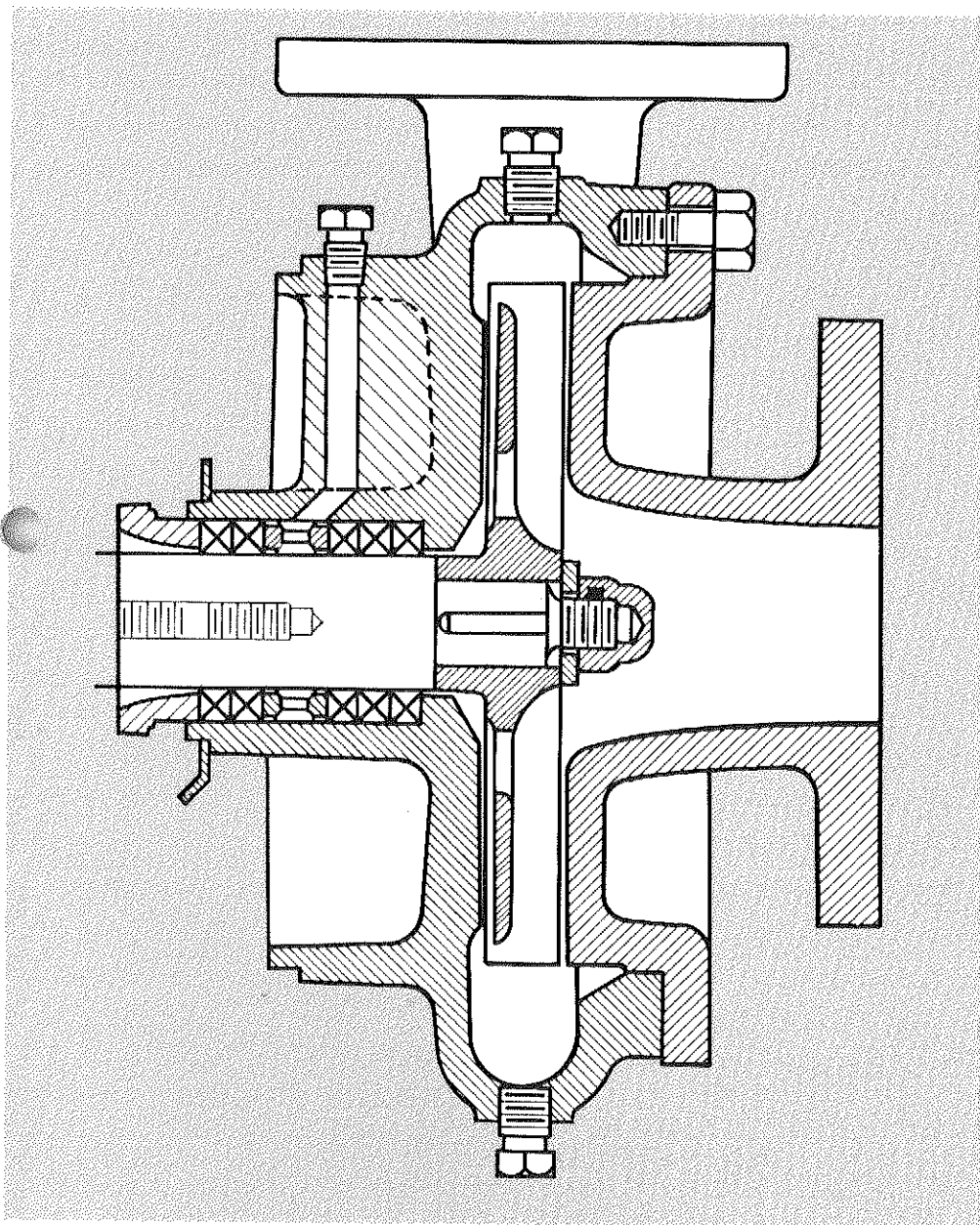


Figure 5—Open impeller.

angular deflection, which causes further problems. Manufacturing tolerances also play an important part in seal reliability. Even in the most rugged designs, alignment between the shaft and stuffing box is not assured. The seal is usually mounted on a sleeve which goes on a shaft, through bearings and in a bearing

housing or frame, to which the stuffing box is mounted. Each step results in further misalignment. This condition is minimized in the pump that has the fewest number of steps or fits between the sleeve and seal seat. It is not unusual to have an eccentricity between the sleeve and the stuffing box of .005",

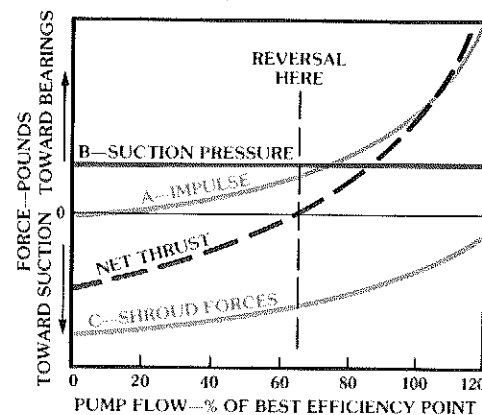


Figure 6—Axial force versus percent flow.

and face squareness out by .0005" per inch of diameter. In other words, a 4" diameter face could be out by .002". Because of the problem in maintaining squareness between the face and shaft, it is desirable to have the stationary member of the seal flexible and the rotating member fixed, but square with the shaft. In this configuration, the rotating face will always rotate square with the axis regardless of problems in shaft deflection, bearing wear, or manufacturing tolerances. Consequently, they should operate with less leakage and wear. In fact, during testing of high-speed seals, the seals with flexible seats were the most successful.

Continued consideration of pump design problems will further aid in the improvement of seal performance.

Pressure conditions due to impeller design are important considerations, as are deflection, misalignment, and tolerances. The circulation systems must be designed to suit varying conditions, and the list goes on. In the end, it is the seal designer who must create a seal to fit the pump... not an enviable task.

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