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Cover
Two shafts removed from twin vertical
turbine pumps on identical shipboard
service (left, the original equipment;
right, a "newer" shaft supplied by a
bootlegger) are graphic evidence of an
ever-present danger associated with the
use of bootlegged parts. As pointed out
in the lead article in this issue, "pump
replacement parts must have more than
just dimensional identity with the parts
they replace. Although many pump
parts appear to be easily duplicated with
simple machine tools, they must also
have metallurgical identity and often
must be produced following a process
identical to that used in producing the
original part." A candid look at this of-
ten-ignored subject of concern to both
pump maker and pump user begins on
page 5.

CASING

RING

SLEEVE

SHAFT

IMPELLER

GALLANTE

The whole is greater than the sum of the parts.
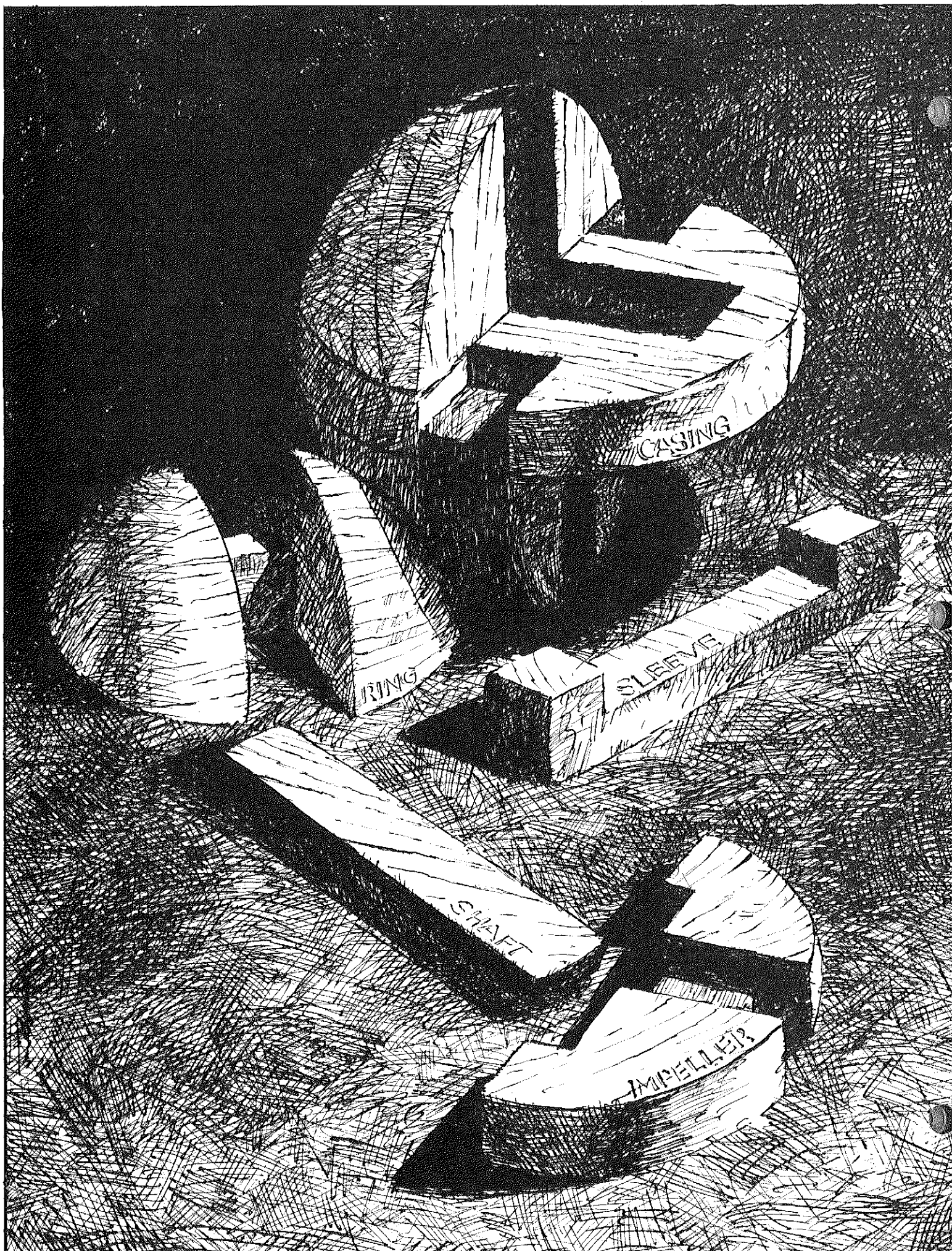
by I.J. Karassik

Most industries have their share of delicate subjects or sensitive areas which, by tacit agreement, are treated as "unmentionables." Everyone knows about them but, for one reason or another, nobody wants to talk about them. Although "silence is golden" is an admirable adage in some circumstances, there come times when for the betterment of all, someone has to be willing to speak out — blow the whistle if need be — and risk unpopularity in order to bring a serious problem out of the closet so that it can be examined in the clear, hard light of day. Igor Karassik, chief consulting engineer for Worthington Pump, went into the closet last January during the Western Gas Producers' and Oil Refiners' Association Pump Workshop at Long Beach, California, and hauled out for all to see and discuss just such an industry sore spot — "bootlegged" pump parts. The following article is adapted from his remarks.

Just so there will be no confusion, let it be clearly understood at the outset that the "parts" I am referring to in the above title are the spare and repair parts that have to be acquired periodically by pump users — and especially the parts that are too frequently obtained from suppliers other than the manufacturers of the original equipment for which the replacements are intended. This area of customer-supplier relationship is generally avoided in technical magazine articles and talks before technical societies — an evasion which is unjustifiable and even harmful. It is exactly because this subject has been so studiously avoided that it threatens to become the

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most critical of the problems that the pump industry and pump users will be facing in the future. And that is why I have chosen to let some light shine on it, even at the risk of making myself unpopular with some users and manufacturers.

The fact is that the practice of obtaining replacement pump parts from suppliers other than the original pump manufacturer is not a here-and-there, now-and-then matter. On the contrary, such acquisitions represent a significant portion of the total parts business. Through some rather complex calculations it can be shown that any pump manufacturer who has been in business for a reasonable period of time should normally expect to supply parts with a yearly volume corresponding to 75 percent of his yearly volume in new units. This is a substantial potential parts business, yet, on the average, pump manufacturers obtain only about 60 percent of it. Where, then, do the purchasers and users of pumps get the other 40 percent of the parts they need?

Exact figures are difficult to establish, but there are reasonable indications that pump users account for about one-half of that missing parts business, manufacturing the parts in their own shops or ordering them from small local machine shops. The other half is supplied by what are commonly known in the trade as "pirates," or "bootleggers." Thus, in round numbers, we can estimate that 60% of replacement parts are supplied by pump manufacturers, 20% by the users themselves, and 20% by bootleggers.

Some people believe that such terms as "pirate" and "bootlegger" are too harsh, conveying as they do the sense of illegality, and I agree. In the first place, with rare exceptions, the practice is perfectly legal. And in the second place, the practice would not have arisen at all, or would be of

little overall significance, if the pump manufacturers and pump users themselves had not created a situation which was particularly favorable for the breeding of the so-called pirates and bootleggers.

Let us examine first the legality of the practice. In this connection, a body of U.S. law was created to satisfy two somewhat contradictory desires:

1. It was considered necessary to reward the talents of inventors and protect them from imitators. This led to the establishment of our patent system, which gives the inventor a 17-year period of protection against infringement.

> **This area of customer-supplier relationship is generally avoided in technical magazine articles — an evasion which is unjustifiable and even harmful.**

2. It was also desired to protect the user or consumer against the creation of monopolistic situations; therefore, unless a product incorporates a patented feature, anyone is free to acquire that product, take it apart, measure all the relevant dimensions, and reproduce it for sale. This is generally called "reverse engineering."

There are various other laws which impinge on these two areas, including those which protect trade secrets. It is certainly

> **The practice of obtaining parts from suppliers other than the original pump manufacturer is not a here-and-there, now-and-then problem.**

illegal to steal a manufacturer's drawings or other documents which may be required to produce exact copies of an original product — and by exact copies I mean not only duplication of dimensions, tolerances and clearances, but also duplication of the metallurgical properties of the components and of the complete performance of the copied product or part. Some may also feel that "stealing" is too strong a word to use in this context, so I'll put it this way: I mean the acquisition by illegal means of drawings, bills of particulars, lists of vendors, process data, and any other information which can materially contribute to reducing the time and cost of doing reverse engineering. Unfortunately, the burden of proof that trade secrets exist is placed on the injured party, with the result that the situation can become very complicated.

This is the legal background against which we need consider the supplying of parts by others than the original equipment manufacturers. Before continuing, however, I would like to offer two comments. First, the fact that an action or activity is legal does not necessarily make it either desirable or beneficial. Second, I am not entirely in sympathy with these laws, mainly because they tend to produce certain unhealthy and undesirable situations. Of course, they are part and parcel of our free enterprise economy, and the advantages of such an economy are so tremendous that they overbalance its disadvantages. But this should not blind us to such an extent that we fail to recognize the disadvantages; nor should it hold us back from pointing them out and trying to reduce their impact.

Now we come to a very important question: Why do some pump users buy parts from bootleggers rather than from the

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pump manufacturers? There can be but two reasons — price and availability. Sometimes the purchaser can get a better price from a bootlegger, but it is my experience that price is not the major factor. What turns most users towards bootleggers is the simple fact that usually they can get quicker delivery.

How did this state of affairs come about? In the early days of pump manufacturing, because they were a novel product, pumps and their construction were usually covered by patents. Today, pumps are mature products; changes and improvements in

R&D, for example, is carried on for the bootlegger *gratis* by the entire pump industry.

construction are not likely to be spectacular. In fact, it is difficult to foresee any changes that would be so drastic in their departure from present-day practice that the manufacturer could obtain patent protection. And even if patents were to be involved, there would still be a significant number of component parts which could

be produced without patent infringement.

At any rate, the very nature of the equipment for which bootleggers furnish parts is such that reverse engineering became an easy means of creating a lucrative market. In time, many small local machine shops became proficient in duplicating replacement pump parts, and did so with little or no concern over possible patent infringement problems. This was particularly true of parts which are easily machined from stock material and require no investment in patterns. Some of these shops grew both in size and ambition, and eventually became large independent parts manufacturers, with repetitive usage parts available for immediate shipment from stock. Today, some parts bootleggers offer specifically indentified parts at published prices for an imposing list of original pump lines. Consider, by way of illustration, this statement from a pump user: "We have experienced poor delivery on spare parts from original pump manufacturers, but have recently found a supplier who carries a large supply of parts on the shelf and can even make some parts with little delay. This supplier has offices and warehouses in a number of cities throughout the United States."

Often lost sight of is the undeniable fact that the parts bootlegger enjoys several marked advantages over the original equipment manufacturer. He need not engage in research and development programs, for example. These are carried on for him, *gratis*, by the entire pump industry. The amount of money spent by individual pump manufacturers varies enormously, but taken together, I would guess that the yearly investment in R&D by all the manufacturers in the United States to be perhaps $40 million.

Like pump users, pump manufacturers are in business to make money. When they lose business to parts bootleggers, they are deprived of legitimate profit.

Similarly, the parts bootlegger does not have to make the significant investments in large machine tools which are required by the pump manufacturer. By the very nature of the parts he supplies, relatively small machine tools are required. He does not require the services of specialists in a variety of disciplines, such as hydraulic and structural design engineering, metallurgical research, elaborate quality assurance procedures, laboratory testing and the like. Nor will he need to employ an elaborate

What turns most users towards bootleggers is the simple fact that usually they can get quicker delivery.

market research program; this, too, is supplied to him *gratis*. Finally, he will not need to concern himself with the question of whether a given pump line has outlived its usefulness and must be replaced by a more modern line, possibly before the profits from the older line have finished paying for the cost of its development. To the contrary, he will be delighted that a pump line is being obsoleted, since this will guarantee that repair parts for the obsolescent pumps may take longer to procure from the pump manufacturer and that more business will be thrown his way.

All these factors enable the parts bootlegger, if he wishes, to undersell the original equipment manufacturer. He does not, however, always choose to do so. I have in files an interesting price comparison, dating from 1971. It involves a large group of repair parts for a $10 \times 7 \times 10$ vertical duplex piston steam pump. Worthington had quoted a price of $2,978 for the parts; supplier X quoted $1,906, but supplier Y came up with a figure of $3,258.

As I said previously, however, quicker delivery rather than price is really the major

factor in deciding where repair parts will be obtained. The main problem for pump manufacturers with regard to deliveries is that without extremely close cooperation with the user — a cooperation which exists only in rare instances — it is practically impossible to satisfy the user's needs in an acceptable time span. To do so, the manufacturer would have to carry in inventory an impossibly large number of parts, well beyond that which would be economically practical. For instance, an analysis of Worthington Pump's records indicates that a total of above 200,000 part numbers is used for the pumps we produce in the United States. Of these, about 30,000 unique part numbers are critical for parts service. It is obviously impossible to carry that many parts in stock; it is only for those parts with a high degree of usage — both for new units and for repair orders — that a stocking program can be justified.

Furthermore, in many cases, especially in the petrochemical and process industries, customers insist on using their own specifications for new pumps, with the result that the pump manufacturer ends up having to provide equipment which incorporates minor, and sometimes major, deviations from his standard construction. In such cases, it becomes impractical to maintain finished replacement parts in stock without prior arrangement with the customer.

I should also point out that, in many cases, quicker deliveries by bootleggers may be more imaginary than real. In nearly all emergency breakdown situations, a variety of parts will be required, all of them necessary to complete the repair. The bootlegger

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can usually supply most of these parts, but very often at least one can be obtained only from the original manufacturer. If 90 percent of the parts arrive in a day or two from the local supplier and the 10 percent from the manufacturer take several weeks or even months, it is the latter who incurs the wrath of the user. Yet, if the bootlegger had to supply the missing part, he would have to make patterns, thus lengthening his delivery time considerably. He might, of course, take heroic measures, working 24 hours around the clock, but his charges would soar accordingly.

Like pump users, pump manufacturers are in business to make money. Obviously, then, when they lose business to parts bootleggers, they are deprived of legitimate profit. And by the nature of the machinery business, parts do, in reality, represent a significant part of the manufacturer's profit.

But the manufacturer loses more than just profit. Frequently, a customer will become quite upset when he compares the delivery performance of the pump manufacturer with that of the bootlegger. He either takes the manufacturer off his bidders' list, or accepts his bids with no intention of buying

In this particular play with a cast of three characters — the pump user, the pump manufacturer and the parts bootlegger — there are neither heroes nor villains. There are, however, two victims, for the pump user is hurt just as badly by the situation as the manufacturer.

In order to perform as it should perform, a part must have more than just dimensional identity with the part it replaces.

from him. Of course, the new pump manufacturer he selects will very likely not perform any better in supplying parts, so he also, in turn, will be toppled from the ranks of acceptable suppliers. In the long run, I suppose, it all balances out, if each pump manufacturer picks up someone else's disgruntled customer every time he loses one of his own. But it is hardly a solid, efficient way of doing business.

In this particular play with a cast of three characters — the pump user, the pump manufacturer and the parts bootlegger — there are neither heroes nor villains. There are, however, two victims, for the pump user is hurt just as badly by the situation as the manufacturer.

At first, the user may on occasion obtain lower prices on the repair parts that he

In many cases, the pump manufacturer can set up repair parts stocking procedures for a particular customer, based on the history of parts usage.

buys. But eventually the prices for new units will have to rise to a level at which manufacturers can obtain a reasonable return on their investment. Otherwise they will be forced out of business. This will become, and in fact is already becoming, a reality. But even before this happens manufacturers will be forced to reduce the amount of money that they can invest in

research and development, and the users will lose the benefit of constantly modernized lines of pumps. Manufacturers will likewise have to reduce their efforts in market and metallurgical research, as well as in such peripheral areas as participation in seminars and conferences.

But, you say, these are purely theoretical conclusions; besides, pump manufacturers are forced to maintain these programs for competitive reasons. There is an element of truth in that. Let us therefore consider shortcomings of a more concrete nature which arise from obtaining parts from sources other than the original manufacturer.

As the title of this work points out, "the whole is greater than the sum of the parts." There are many reasons why this is true. In order to perform as it should perform, a part must have more than just dimensional identity with the part it replaces. It must also have metallurgical identity, and it must be produced following an identical process to that followed in producing the original part. Reverse engineering, especially when it involves no drawings or other documents that have been obtained illegally, has two major drawbacks: It does not provide any information on optimum tolerances, and it does not tell whoever copies the part exactly how it was produced in the first place. And it is this information which frequently constitutes that narrow margin between brilliant success and utter failure. Let me cite you a few typical examples of the problems that can arise because of this lack of information:

Case 1
Engineers at one utility customer decided to buy shafts for their multistage boiler feed pumps from a local supplier, since they could get much faster delivery and would

realize a considerable savings over the price quoted by the pump manufacturer. After all, they reasoned, a shaft is probably the easiest pump part to duplicate dimensionally. And as to the metallurgy, both the proposal and the instruction books spelled out exactly what the shaft was made of, including chemical and physical characteristics. They didn't appreciate the fact, however, that there was more to the matter than dimensions and metallurgy. There was also a process that the pump manufacturer had developed through years of experience to insure a shaft that could survive successfully in its environment. This process included a complex sequence of operations such as normalizing and tempering the bar, rough machining, heat treatment, further machining, stress relieving, still more machining, cutting threads where necessary, grinding, and a few other operations. The manufacturing procedure document included all of the sequential steps, including the temperatures at which the various heat treatments are to take place. None of this information, of course, could be deduced from merely measuring the shaft or determining its hardness.

The local supplier produced the shafts without access to this information, and within a short time the shafts ran out of true beyond permissible limits. Other component parts were damaged and the pumps had to be dismantled, which increased the cost of the overhaul by several times and delayed the return of the units to service.

Case 2
A major steel company was purchasing repair parts for barrel type descaling pumps from a parts bootlegger. The metallurgy of these parts was inappropriate and the customer started experiencing the need for major repairs every three months. The fact is that the supplier of bootleg parts generally has no idea of what is a reasonable life expectancy for a given part in a given service. He is therefore unable to guide the user in his diagnosis of what causes a shorter life than would normally be expected.

Case 3
The inability of the parts bootlegger to

If we accept the fact that there are advantages in purchasing repair parts from the pump manufacturer, we must next ask ourselves what can be done to improve the situation.

diagnose the real source of a difficulty or a failure is what ultimately creates problems for the user. The following incident is a typical illustration of this situation. One of our customers thought that he was experiencing excessive plunger wear in several of his power pumps. A local parts supplier was only too happy to supply him with plungers of a different metallurgy. Unfortunately, this failed to solve the problem. When our service people finally were called in, they discovered that the wear was being caused by inadequate alignment and improper packing procedures. When these two problems were corrected, the life of the plungers returned to its normal expected span.

Case 4
This final example involves a problem which is encountered quite frequently. The pump in question was a two-stage axially-split casing unit, with back-to-back mounted impellers for axial thrust balance. The service was rather severe in that the water handled by the pump contained an appreciable quantity of very fine abrasive particles. This particular circumstance had not been mentioned to the pump manufacturer at the time of the purchase. In a few months, the thrust bearing failed and was replaced locally. It failed again — and again. What the customer and his bearing supplier failed to diagnose was that the interstage bushing was experiencing an excessive rate of wear. Because of the increased interstage leakage, the two impellers were operating at different flows. This, in turn, caused the two stages to operate at different heads, upsetting the axial balance. Replacing the thrust bearing without attending to the interstage bushing and sleeve was equivalent to curing a symptom

without attacking the disease itself. Ultimately, the pump manufacturer was contacted, more wear-resistant materials were used for the interstage parts, and the life of the bearing returned to normal.

If we accept the fact that there are advantages in purchasing repair parts from the pump manufacturer, but that certain factors, such as quicker deliveries, sometimes interfere with the user's desire to do so, we must next ask ourselves what can be done to correct this situation. It would be possible to spell out a complete and definitive program for improving the handling of parts orders to such a point that the bootleggers would go out of business. I shall not do so, however, because such a program can be neither patented nor copyrighted. I can, however, list a few thoughts which would form the basis of any such program.

The user must develop a conscious understanding of the detrimental effects of parts purchased from sources other than the original pump manufacturer. And he must understand that carrying an inventory of parts costs money, whether the inventory is carried by the manufacturer or by the user himself. This cost ultimately has to be absorbed in the price of the parts.

In many cases, the pump manufacturer can set up repair parts stocking procedures for a particular customer, based on the history of parts usage. But to do so requires fairly complete and accurate information from the user. This will permit the manufacturer to set up the program with certain parts carried in stock at the factory, certain other parts in local or regional centers, and the rest carried in stock by the user. This kind of cooperation with the pump manufacturer is necessary not only to develop parts

programs which provide adequate inventories, but to establish appropriate lead times for items which are impractical to inventory.

Customers who institute a regular scheduled preventive maintenance program should either carry the necessary parts in their inventory or, at least, order them in ample time to eliminate most emergency situations. In turn, with a reduction of emergency situations, the manufacturer must gear himself to handle with greater speed the few emergencies that will still inevitably arise.

If this understanding and cooperation are developed between the manufacturer and the user, we will truly be able to say that "the whole is greater than the sum of the parts."
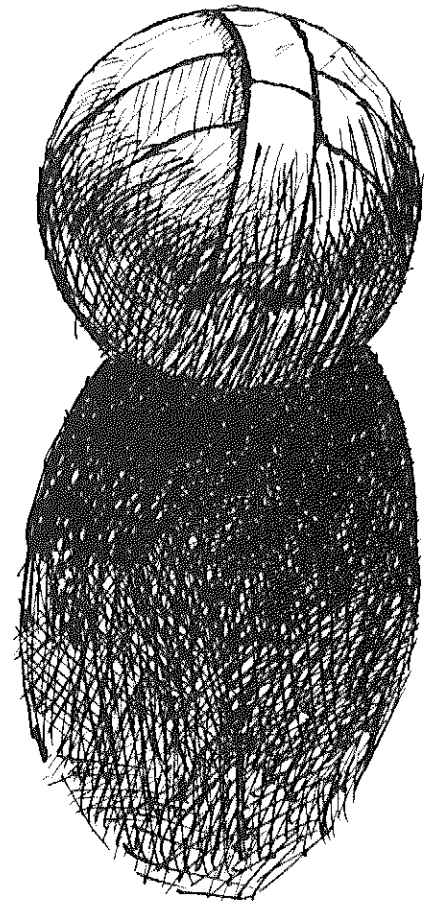
The effect of rotative speed on centrifugal pump performance.

E.J. Serven

Do high-speed pumps need more maintenance? Are low-speed pumps relatively more expensive? What speed is most efficient for a particular application? Does rotative speed affect the noise level very much? Why is a certain pump designed to run at 3550 rpm instead of 1770 rpm? The effect of rotative speed on initial and operating costs and on such factors as reliability and mechanical performance needs to be clearly understood, since it plays an important part in both pump design and pump selection. This article examines the most important factors that may influence the choice of speed, including hydraulic design and efficiency, installed cost and cost of replacement parts, mechanical performance, maintenance considerations, and noise generation.

To understand the relationship between rotative speed and such factors as hydraulic performance and efficiency, we should first review the term Specific Speed (N_s), to which attainable efficiency and rotative speed are both related. In the formula for specific speed

$$N_s = \frac{n\sqrt{Q}}{H^{3/4}}$$

Q represents the capacity (in gpm) for which the pump was originally designed — the capacity at best efficiency. H is total head (in feet) for which the pump was designed, based on the maximum diameter impeller at the design capacity. And n represents the rotative speed (rpm) at which the above Q and H were measured.

The designer of a new centrifugal pump is required to meet a certain capacity and head based on the maximum impeller

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diameter. Referring to the formula above, it can be seen that once the design conditions have been established, the specific speed of the resulting pump will depend upon, and in fact will be directly proportional to, the rotative speed.

Figure 1 shows the relationship between rotative speed and specific speed in the range we are considering — that is, end-suction centrifugal pumps for process and general industrial service up to a discharge nozzle size of 6 inches. The chart takes the place of the slide rule calculation. The example shown on the chart is for a pump designed to deliver approximately 350 gpm at a total head of approximately 110 feet. If this pump had been designed to run at 1770 rpm to produce these conditions, the specific speed would be a little over 1000. If, on the other hand, 3550 rpm had been selected as the design speed, the specific speed would be just over 2000.

Below the chart in Figure 1, there are half-sections of various impeller configurations. They show the approximate shape of an impeller that has the specific speed indicated directly above. It can be seen that an impeller of low specific speed (about 500) is very narrow and has a relatively large diameter. As the specific speed of a pump increases, the impeller becomes wider with respect to its diameter. Figure 2 shows the relationship between the practical attainable efficiency and various specific speeds in a centrifugal pump.

Hydraulic design and efficiency.
Since the small end-suction centrifugal pumps we are discussing can seldom be designed to have a specific speed higher than 3000, we can ignore the part of Figure 2 that falls to the right of $N_s = 3000$. We can now make a very important observation about these pumps: Their attainable efficiency increases with an increase in specific speed and, therefore, increases with an increase in rotative speed. This means that for a given set of head and capacity design conditions, the pump designed to operate at the higher rotative speed will have a higher efficiency. Referring back to Figure 1, it can be seen that

this relationship generally favors 3550 rpm as the design rotative speed, provided that the selection of this rotative speed does not mean that the specific speed will exceed 3000.

Of course, since pump designers are very conscious of this relationship, the purchaser of standard models is often relieved of any choice in regard to rotative speeds for given conditions of head and capacity. If he does have a choice, he will find that the pump operating at the lower speed will have a reduced efficiency.

From Figure 2, it can be seen that about the only time efficiency will be at a standoff is when the two pumps of different rotative speeds have specific speeds that are both in the 2000-3000 range. For instance, if the design conditions for the pump are to be 3000 gpm at a head of 100 feet, the specific speed of a pump designed to run at 1770 rpm would be about 3000. For a pump designed to run at 1150 rpm, the specific speed would be just under 2000. The 3000-gpm curve in Figure 2 shows the attainable efficiency for these two specific speeds to be about the same, so that other considerations will govern the choice in this case.

To conclude this discussion of efficiency, let us consider a situation involving a choice between two commercially available pumps of different rotative speeds. Since no two pumps are normally designed to run at exactly the same conditions, we will use two that are as close as possible. Worthington's $3 \times 2 \times 6$ D-1000 pump operates at a speed of 3550 rpm and is designed for 325 gpm at a total head of 150 feet. Referring to Figure 1, this condition would correspond to a specific speed of approximately 1600. Worthington also has a $3 \times 2 \times 13$ D-1000 pump designed for 350 gpm at a total head of 170 feet, which was designed to run at 1770 rpm. These conditions would correspond to a specific speed of about 740.
Comparing the performance of these two pumps using Figure 2, we find that the 3550-rpm pump has a maximum operating efficiency of 77%, while the 1770-rpm
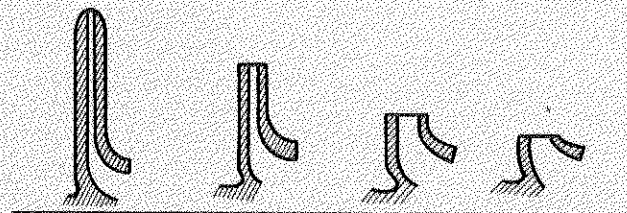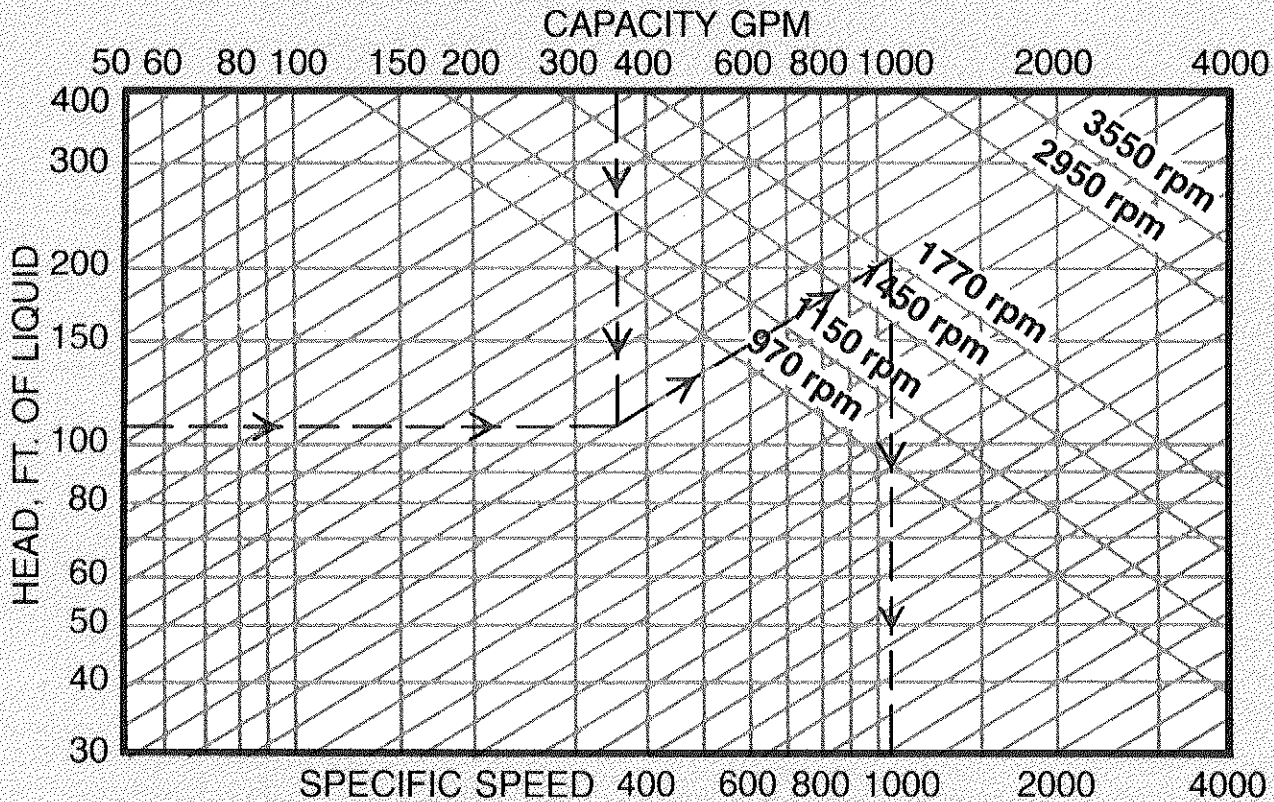
Figure 1—Relationships between specific speed, rotative speed, and impeller proportions.

pump has a maximum operating efficiency of 67%. In other words, for a given set of h' and capacity conditions, the 1770-rpm pump will require about 15% more power than the 3550-rpm model. Furthermore, the original motor will be more expensive.

Net positive suction head

A factor other than efficiency that can be affected by specific speed is the net positive

suction head (npsh) required by the pump. It is easier for the pump designer to achieve low npsh requirements on low specific speed pumps than on high specific speed pumps. However, it is not possible to make the general statement that, for a given set of head and capacity conditions, the pump designed for the lower rotative speed (consequently, lower specific speed) will always have a lower npsh requirement. Other factors enter into this, and in cases where

npsh is critical, the purchaser should examine the potential pump selections to make sure that the npsh available is at least equal to that required by the pump.

Costs: initial, installation, and parts.

The head that a single-stage centrifugal pump can develop, and the capacity it can deliver, are based on the impeller diameter and the rotative speed of the pump. For a given rotative speed, the larger the impeller

diameter, the greater the head and capacity capability of the pump. Similarly, for a given impeller diameter, the greater the rotative speed, the greater the head and capacity. Or, to put it another way, when comparing various pumps for a given set of conditions of service, the pump with the higher rotative speed will have a smaller impeller diameter.

The relationship is basically linear. For instance, in the comparison between two D-1000 pumps for 350 gpm at a head of 150 feet, the 3550-rpm pump would require a 6.5-inch impeller, and the 1770-rpm pump, a 12.6-inch impeller. Both pumps will have the same inlet and outlet size, since they are both handling the same capacity. But because the lower speed pump has a much larger impeller and, therefore, a much larger casing, its initial cost will be considerably more. This is particularly significant in the more expensive alloys where the difference in the cost of the pumps can amount to 70% or more. Furthermore, since we have seen that the efficiency of the higher speed unit is greater, a lower horsepower motor can be used in many cases. Even in those instances where the motor horsepower remains the same, the NEMA frame size will be lower for the higher speed unit, so that the cost of the motor to drive the higher speed pump will always be less. In some cases, this differential cost can be very substantial.

The cost of installation of the higher speed unit is also lower. Being smaller, the unit is easier to handle and requires a smaller foundation. When the cost of plant floor space is at a premium, this can become an important item. In addition, since the higher speed unit will probably use a lower horsepower motor, the cost of wiring and starters can be reduced.
The cost of individual liquid-end parts for the pump is related to the size of the part, and since the higher speed unit has smaller parts, the cost of replacement parts is greatly reduced.

Pumps handling severely abrasive slurries fall into a separate category. Here, the fact that the impeller will be eroded faster at
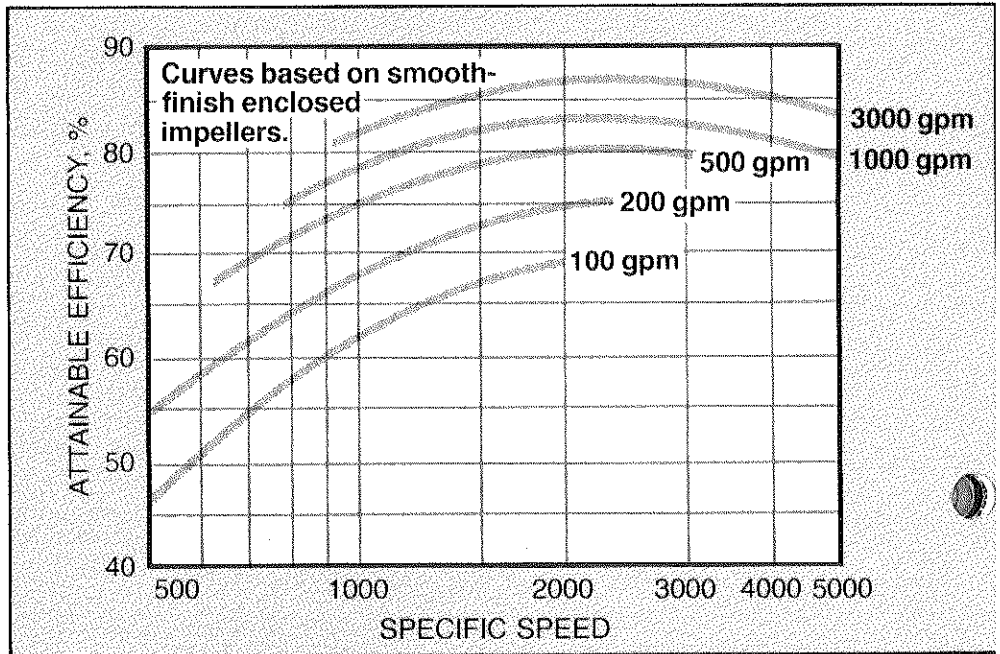

Figure 2—Attainable efficiency as a function of specific speed.

higher speeds must be evaluated.

Performance, reliability, and maintenance.

Even though we are considering the relative merits of various rotative speeds, it is important to realize that a well-designed, accurately machined, and carefully assembled pump will operate satisfactorily regardless of the rotative speed. Conversely, changing the rotative speed alone will not cure the ills associated with a poorly designed or poorly manufactured pump.

Generally speaking, the two most important maintenance items on small centrifugal pumps are the bearings and the stuffing box or mechanical seal (in severely abrasive services, the pump liquid-end itself may also be attacked). In examining problems involving bearings and stuffing boxes, the forces acting on the shaft of the centrifugal pump must be kept in mind.

The shaft in a centrifugal pump must transmit the power from the driver to the impeller, so that the impeller can do the work of pumping the liquid. The pump shaft must be adequately sized to transmit the horsepower. Since the selection of a

shaft diameter depends upon the torque that the shaft is transmitting, and since a lower speed pump would be transmitting more torque in order to transmit the same horsepower, it is obvious that the lower the rotative speed of the pump, the larger the shaft diameter must be for a given set of head and capacity conditions.

Furthermore, we have already demonstrated that the efficiency of the lower speed pump is generally lower, and therefore, the amount of horsepower to be transmitted is generally higher. This further increases the torque that must be transmitted by the lower speed unit and tends to increase the shaft diameter requirements even more. There is also a bending force on the pump shaft. This force is exerted on the impeller and tends to bend the pump shaft. It also loads the pump bearings.

When the centrifugal pump is operating, pressure distribution around the circumference of the impeller varies and a resulting unbalanced force occurs. The term "radial reaction" is applied to this force because it acts in a radial direction on the impeller. The direction of application and the force's

magnitude will depend on the pump design and the capacity at which the pump is operating. This force may be 10 pounds on a small pump, or over 500 pounds on a pump with a 6-inch discharge nozzle.

The load exerted on the shaft is proportional to the diameter of the impeller, width of the impeller, head produced by the pump, and a constant designated K. The load is also proportional to the specific gravity of the liquid — but we will assume this to be a constant factor.

Since we are analyzing the difference in performance of pumps designed for the same head and capacity, the width of the impeller will also be about the same. The load on the impeller and shaft can thus be considered as proportional to the diameter of the impeller and the constant K. This means that if we are considering a 3550- and a 1770-rpm pump for the same job, the load on the shaft would be twice as high for the 1770-rpm unit — if the constant K were the same for both pumps. Actually, the constant for the lower speed pump is generally somewhat less than for the high speed pump. In fact, for situations where the higher speed pump has a specific speed under 1500, the constant K for the lower speed unit is sufficiently less, to essentially compensate for the larger impeller diameter. But when the specific speed of the higher speed pump is above 1500, the constant K begins to approach the K for the lower speed unit, so that the load on the shaft and impeller tends to become proportional to the ratio of the impeller diameters. In these cases, the shaft and bearings for the lower speed unit must be made to withstand this higher load without bending.

While most of the difficulties associated with the maintenance of packed stuffing boxes or mechanical seals stem from improper operation or adjustment, even properly adjusted packings or seals cannot give satisfactory service if the shaft is bending through the stuffing box area due to the load imposed on the impeller. It is therefore extremely important that the pump designer adequately size the shaft through the stuffing box area so that, regardless of the rotative speed of the pump, deflection of the shaft through the stuffing box is limited to a specific maximum value. Determining the "best" shaft diameter is an optimization problem; exceeding the optimum will increase the replacement cost not only of the shaft itself but also of the shaft sleeve and mechanical seal parts. Because of all these considerations, the soundness of the pump manufacturer's design philosophy is much more significant than the pump's rotative speed as far as seals and packings are concerned.

The picture is about the same for anti-friction bearings. Theoretically, the life of an anti-friction bearing under a given load is inversely proportional to the rotative speed. However, we have already demonstrated that the load in the 1770-rpm pump is higher. On the other hand, the shaft diameter in the 1770-rpm pump is larger, and this generally means that larger bearings are used. It is extremely difficult to put all these factors together in one analysis and come up with a general statement.

The overall indication is that the slower speed unit would probably have a longer statistical bearing life. There is a fallacy in stopping at this point, however. Bearings generally do not fail because of old age, since they do not normally operate under such controlled conditions that statistical failure is obtained. Rather, contamination, over-lubrication, under-lubrication, or other problems normally cause the bearings to fail long before their useful life has been achieved. In such circumstances, the consideration of rotative speed becomes unimportant, since rotative speed normally does not have any relationship whatsoever to the factors causing premature bearing failure. Of course, it is still important to be sure that the pump is designed with adequately sized bearings, regardless of the rotative speed, and the pump purchaser should carefully analyze the bearing lubrication system to make sure that the pump is designed to minimize the possibility of premature bearing failure.

In a properly operated stuffing box, the amount of leakage is determined by the small gap between the packing and the shaft, by the pressure differential between the outside of the stuffing box and the inside, by the length of the stuffing box, and by the circumference of the shaft. If we assume that the gap between the shaft and the packing is the same regardless of rotative speed and that the length of the stuffing box is approximately the same, we are left with the circumference of the shaft, since the pressure differential would obviously be the same. Since the lower rotative speed shaft has a larger diameter and, therefore, a larger circumference, it follows that the leakage rate will be greater.

Noise generation

Once it was common to link noise level with rotative speed. More recently, however, the factors contributing to noise generation in centrifugal pumps have been more clearly defined through extensive research. While rotative speed is *one* of the factors that may contribute to the noise level, there are many others that also have an effect. Actually some of these are related to rotative speed in such a way that an increase in the latter will result in a decrease in the noise level.

The subject of noise level in centrifugal pumps is extremely complex. Fortunately, centrifugal pumps are basically quiet machines, and consideration of noise level only comes into the picture when extremely quiet operations must be assured, such as in the air conditioning or heating of office buildings, hospitals, and so on. The important thing is that the pump designer, knowing how factors other than rotative speed affect the general noise level, can now make a very quiet pump even though he uses the higher rotative speed.

Rotative speed does have a significant effect in many key areas of pump design and selection — however, there are no universal answers to the many questions it raises. When faced with a specific application, we must examine and evaluate each factor that might influence our choice of speed and weigh the relative importance of each to the case at hand.

Quick changeout: An inside look at high-pressure barrel pump cartridge replacement.

By P. Filan
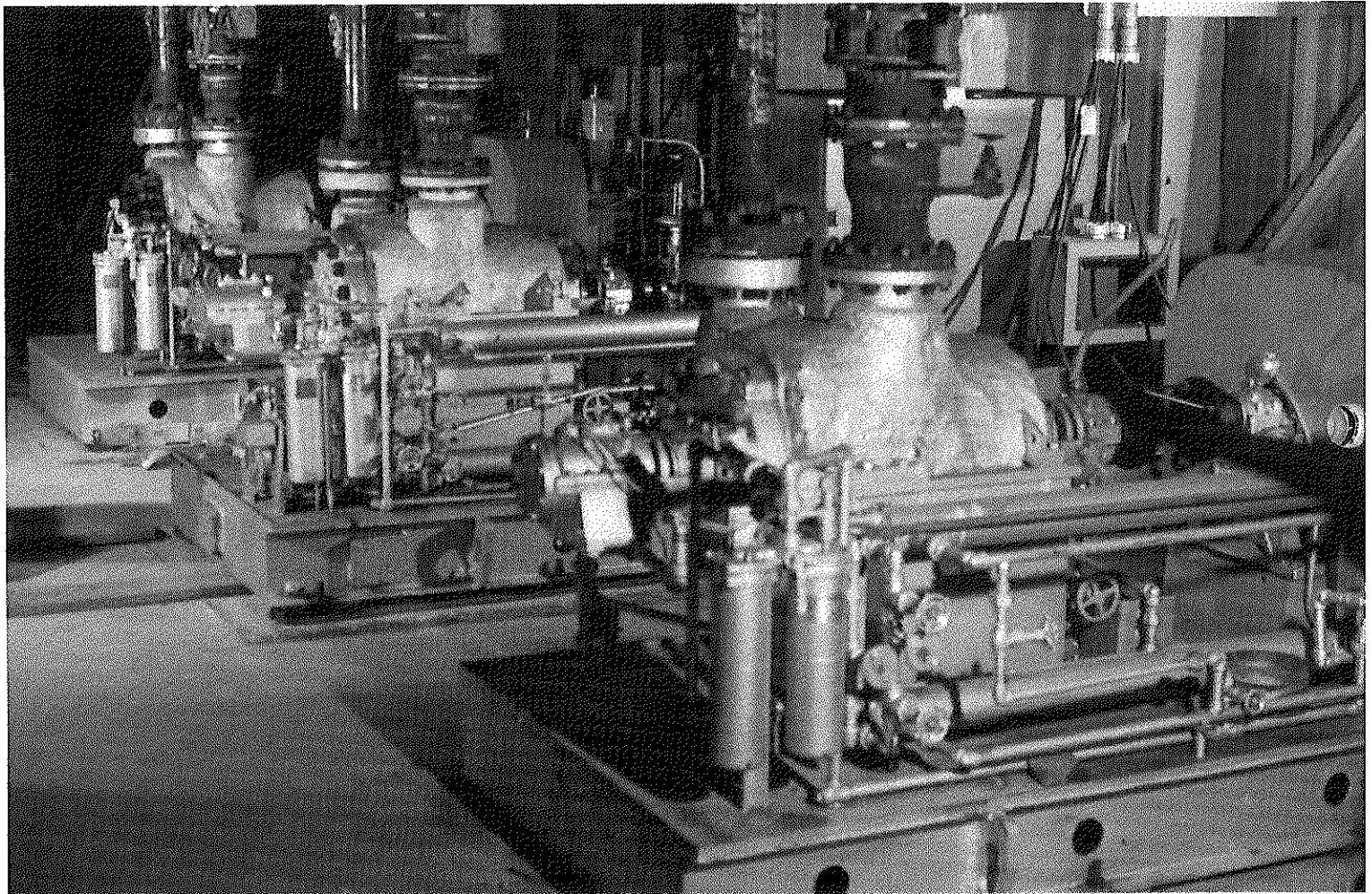
The Worthington Type W is a high-pressure multistage cartridge pump designed for demanding services such as industrial boiler feed, high-pressure process applications, steel mill descaling and accumulator service, hydraulic press, and waterflood. All working parts of the pump — including impellers, diffusers, shaft, bearings, and seals — are incorporated in a removable cartridge, a unique design which results in important benefits to the user.

The Type W cartridge pump is specially designed for the job of containing high pressure. The cartridge itself forms a separate inner casing which is completely contained inside a rugged, three-part weldment, which forms the outer casing. The radial

joints of the containment casing seal pressures up to 3200 psig. As a result, the Type W pump is remarkably free of the bolting and gasketing problems that plague pumps with less positive sealing arrangements.

When service is required, another major benefit of the cartridge design proves its worth. Since all working parts are contained in the single cartridge element, the cartridge is simply removed intact and serviced — or replaced on the spot with a spare. Only minor lube piping and instrumentation needs to be disconnected. The casing and main system piping remain in place, and the pump can be restored to operation quickly.

The entire changeout can be accomplished

by two men in one shift or less. A special carriage facilitates cartridge withdrawal and replacement; however, an overhead crane can also be used.

Although Worthington's Type W pump instruction manual gives a detailed, step-by-step procedure, and should be carefully studied before an actual changeout is begun, the following sequence of photos taken from a training film, affords a rare, inside look at highlights of the quick-changeout procedure.

Here's how two men can change a Type W pump cartridge in one shift or less.

Pat Filan is field engineer for WPC(USA).
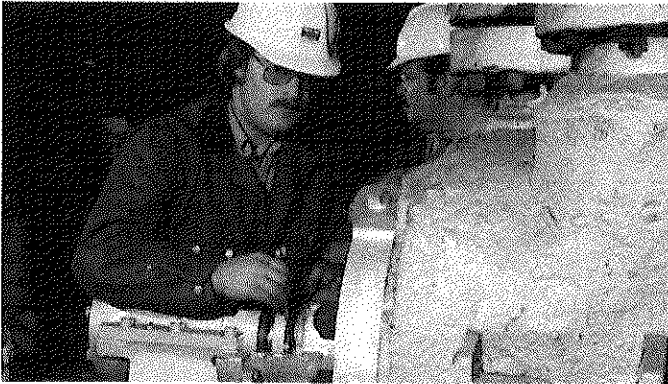
Step 1 The pump rotor is locked to the cartridge using a special clamping plate. Two clamps, one for each end, are supplied with the pump. The clamped shaft acts as a tie-bolt, holding the cartridge securely together after it has been removed from the casing.

Step 2 Discharge head nuts are removed.

Step 3 The specially designed carriage is rolled into position.

Step 4 Carriage height is adjusted using leg leveling bolts on each side, assuring proper positioning of the cartridge during withdrawal from the casing.
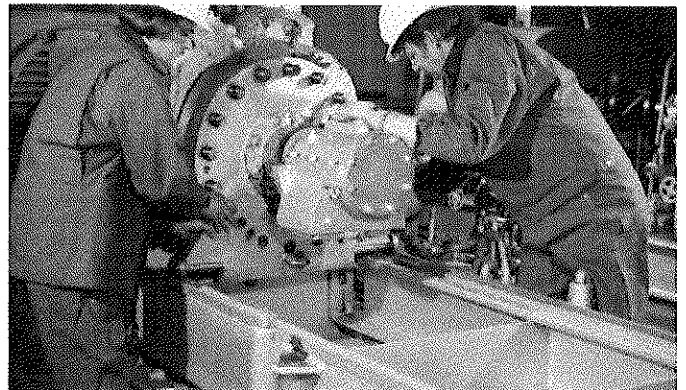
Step 5 A cartridge end support plate is bolted to the discharge head. The plate, with roller bearing wheels riding on tracks in the carriage, will carry the weight of the cartridge during withdrawal, protecting internal sealing surfaces.

Step 6 The carriage is then bolted to the base of the pump.
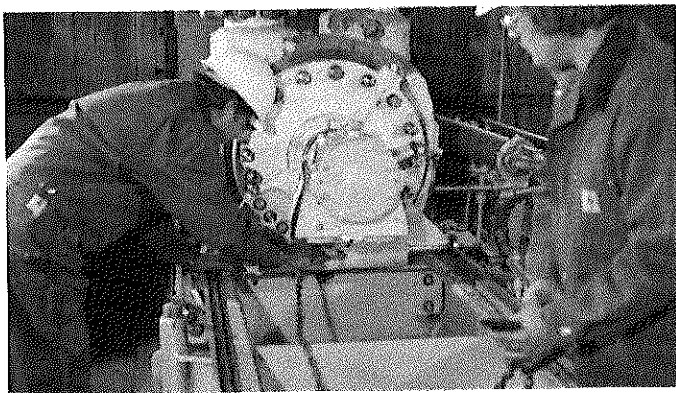
**Step 7** A pulling device is attached to the end support plate to facilitate cartridge removal.

Step 8 Equal turns on two "Jacking bolts" in the discharge head begin the removal of the cartridge from the casing.

**Step 9** During withdrawal, the cartridge is supported and positioned by skids (shown in Figure 14) in the lower quadrant of the casing.
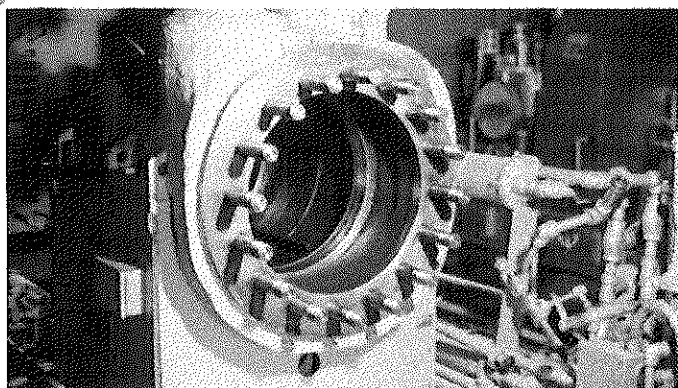

Step 10 It is important to keep the cartridge centered in the casing during the entire removal procedure, in order to protect the sealing surfaces.

Step 11 After the cartridge has been withdrawn slightly more than halfway, a center support is installed.
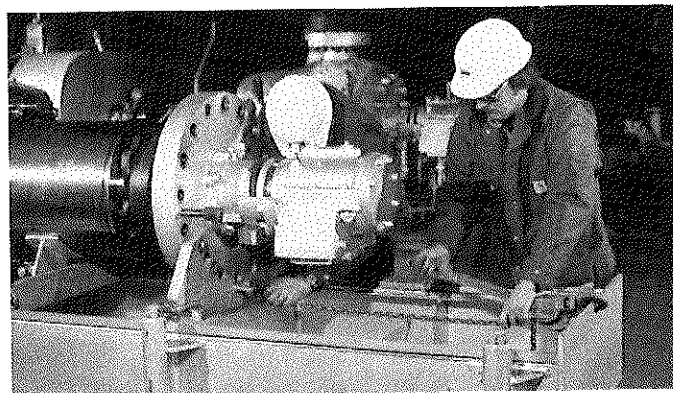
Step 12 The additional wheels will allow the cartridge to roll out smoothly, and the pulling device is disconnected. Skids prevent contact with the sealing surface at the last stage diaphragm.
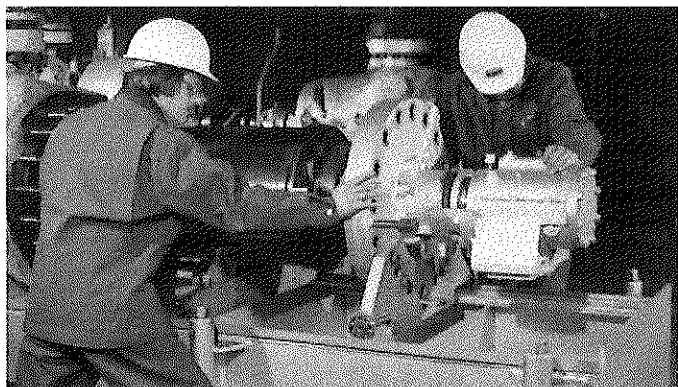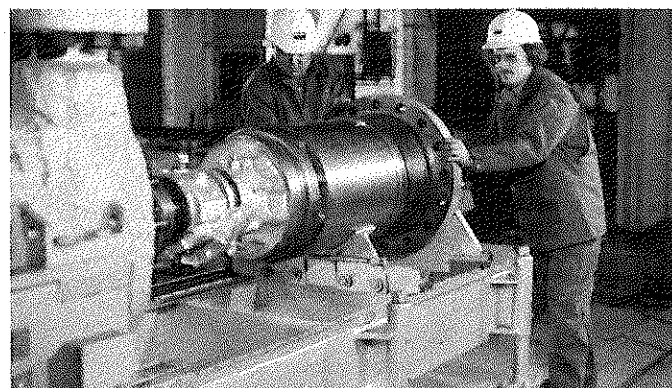
Step 13 The cartridge is withdrawn until it is completely out of the casing and clear of the studs.
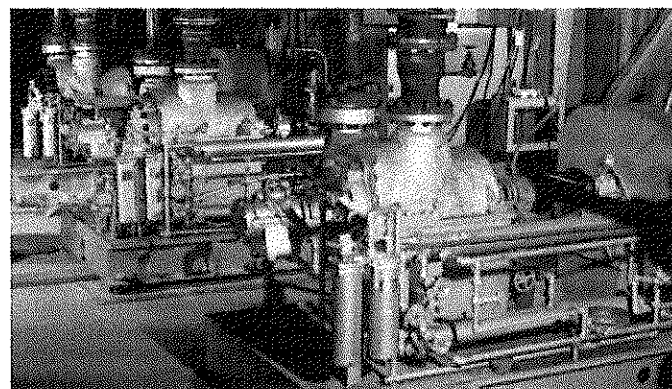

Step 14 When the cartridge is completely free of the casing, it is locked in position on the carriage and rolled away to the service area.

Step 15 Before installing the spare cartridge, the sealing surfaces inside the pump casing are inspected to make sure they are in good shape.

Step 16 Installing the replacement cartridge in the casing shell involves the same basic procedure in reverse. Once again the carriage is rolled up to the pump and bolted on. This time the cartridge is already in place on the carriage.


Step 17 With the carriage and end support plate bolted in position, the cartridge is carefully inserted into the casing. Here, as in cartridge removal, it is important to keep the cartridge centered in the casing to protect the seals and sealing surfaces.

Step 18 With the cartridge in position in the casing, the discharge head is bolted back on and lube lines and instruments reconnected. Clamp plates are removed and a check is made for free shaft rotation. When all checks are made, the Worthington cartridge pump is ready to go back to work – an entire changeout performed in just one shift.
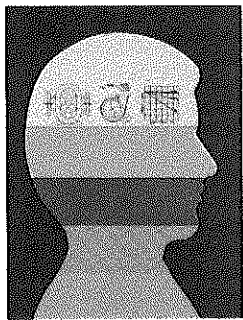
pump refresher

Basic hydraulics: The characteristics of liquids.

Part 5

By John H. Doolin

In the last issue we reviewed forms of energy in liquids, looking at how Bernoulli's Equation and the Continuity Equation can be used to analyze problems associated with pumping systems. So far, we have worked only with water, a fluid which is relatively simple in its characteristics. However, many commonly handled fluids have properties or characteristics which are markedly different from water and which may have a significant effect on a pumping system. Water, too, will have different characteristics under certain conditions.

Some of the most significant characteristics of a fluid are viscosity, specific gravity, vapor pressure, entrained solids, entrained gas and corrosive properties.

Viscosity.

Viscosity is the property of a fluid which causes it to offer resistance to shear stress such as that caused by fluid flow, primarily in the area of the pipe wall. This is illustrated in Figure 1, which shows the velocity of a fluid flowing relative to a static boundary surface.

At the static boundary surface or wall, the velocity of the fluid is zero. As the distance increases from the static surface, the velocity of the fluid increases. This change takes place only when a force is exerted on the fluid causing it to flow with a velocity. The force is usually a function of the velocity gradient V/d: that is, the maximum velocity of shear rate of the fluid, V, divided by the distance from the static surface, d. Absolute viscosity μ is the quotient of the shear stress, or force per unit area, divided by the shear rate:

$$\mu = \frac{F/A}{V/d}$$

Kinematic viscosity.

In most fluid flow problems, both the viscosity and the density of the fluid must be considered. For this reason, it is common to combine viscosity and density by dividing one by the other. This quotient is known as the kinematic viscosity:

$$V = \frac{\mu}{P}$$

One of the most common units of measure of kinematic viscosity is Saybolt seconds. This refers to the length of time it takes for a measured quantity of fluid at a specific temperature to drain from a container with a measured orifice in the bottom, as shown in Figure 2. Water has a viscosity of approximately 31 Saybolt seconds universal (SSU) at 60°F. By comparison, light lubricating oils my have a viscosity of 100 or 200 SSU. More viscous lubricating oils have viscosities in the
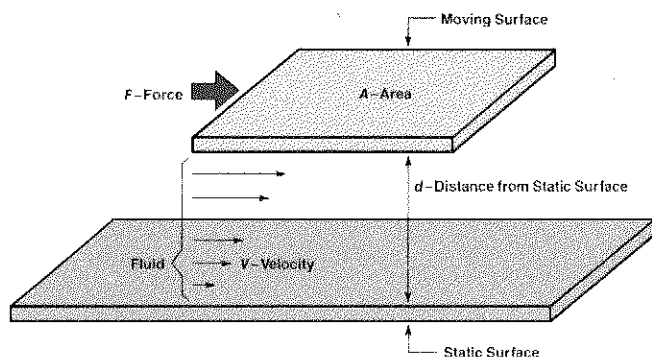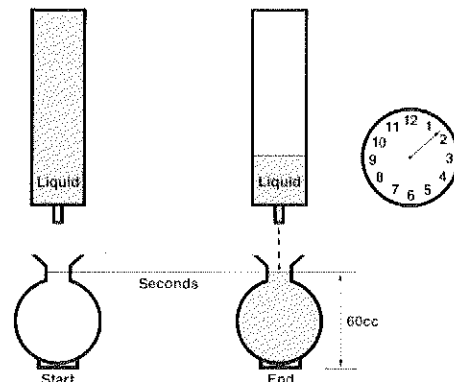

Figure 1—Viscosity.

Figure 2—Saybolt viscometer.

Jack Doolin is director of product development for the Standard Pump Division of Worthington Pump Corporation (USA).

thousands of SSU, and extremely viscous fluids — heavy tar, for example — have viscosities as high as 1,000,000 SSU.

Kinematic viscosity is also commonly expressed in metric units as stokes or centistokes. Centistokes can be derived from Saybolt seconds by the formula:

$$\text{Centistokes} = (.22 \times \text{SSU}) - (180/\text{SSU})$$

For example, the kinematic viscosity of water is 31 SSU, or 1 centistoke:

$$(.22 \times 31) - (180/31) = (6.82 - 5.81)$$
$$= \text{approximately}$$
$$1 \text{ centistoke}$$

The subject of viscosity is further complicated by the fact that the viscosity of some fluids is not constant under all circumstances. In an ideal fluid, the viscosity remains constant regardless of the rate of shear. That is, if the velocity gradient increases, the shear force necessary to provide this increased velocity increases proportionately, so that the quotient of the shear force and the velocity gradient remains constant, as shown in Figure 3. This is called a *Newtonian* fluid.

On the other hand, the viscosity of some fluids decreases as the rate of shear increases. That is, when the viscosity is increased, the force necessary to provide that increased shear rate is proportionately less than the increase in shear rate. If the viscosity is measured at various shear rates, the result will be the characteristic curve of Figure 4. Such a fluid is called *thixotropic;* paint, grease, starch, molasses and tar are examples of thixotropic fluids.

If the viscosity of a fluid increases with the rate of shear, as shown in Figure 5, the fluid is called *dilatant.* Examples of such fluids are mineral slurries and some candy compounds. Dilatant fluids are difficult to pump because the increased velocities associated with pumping action increase the viscosity of the fluid and thereby its resistance to being pumped.

In this *pump refresher* we have looked at viscosity, perhaps the most significant characteristic of a fluid in terms of its effect on a pumping system. In the next issue we will review other influential characteristics of fluids.
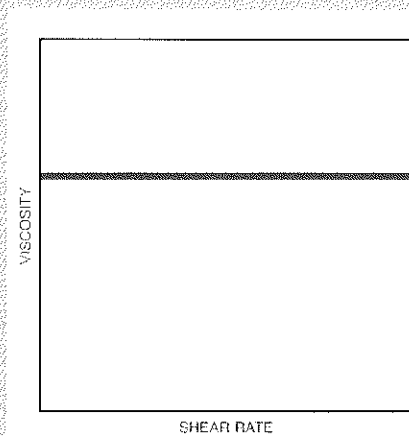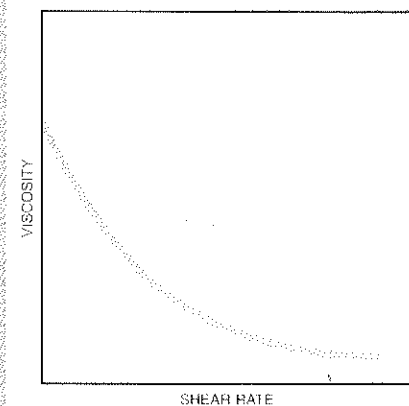

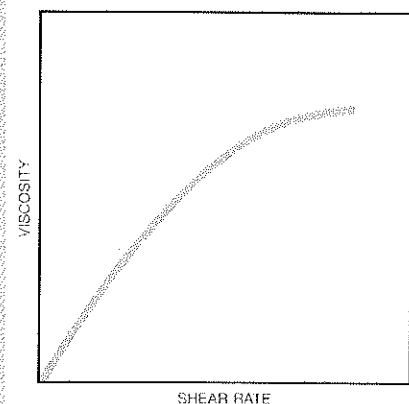**Figure 3 — Newtonian Fluid.**

Figure 4 — Thixotropic fluid.

Figure 5 — Dilatant fluid.

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