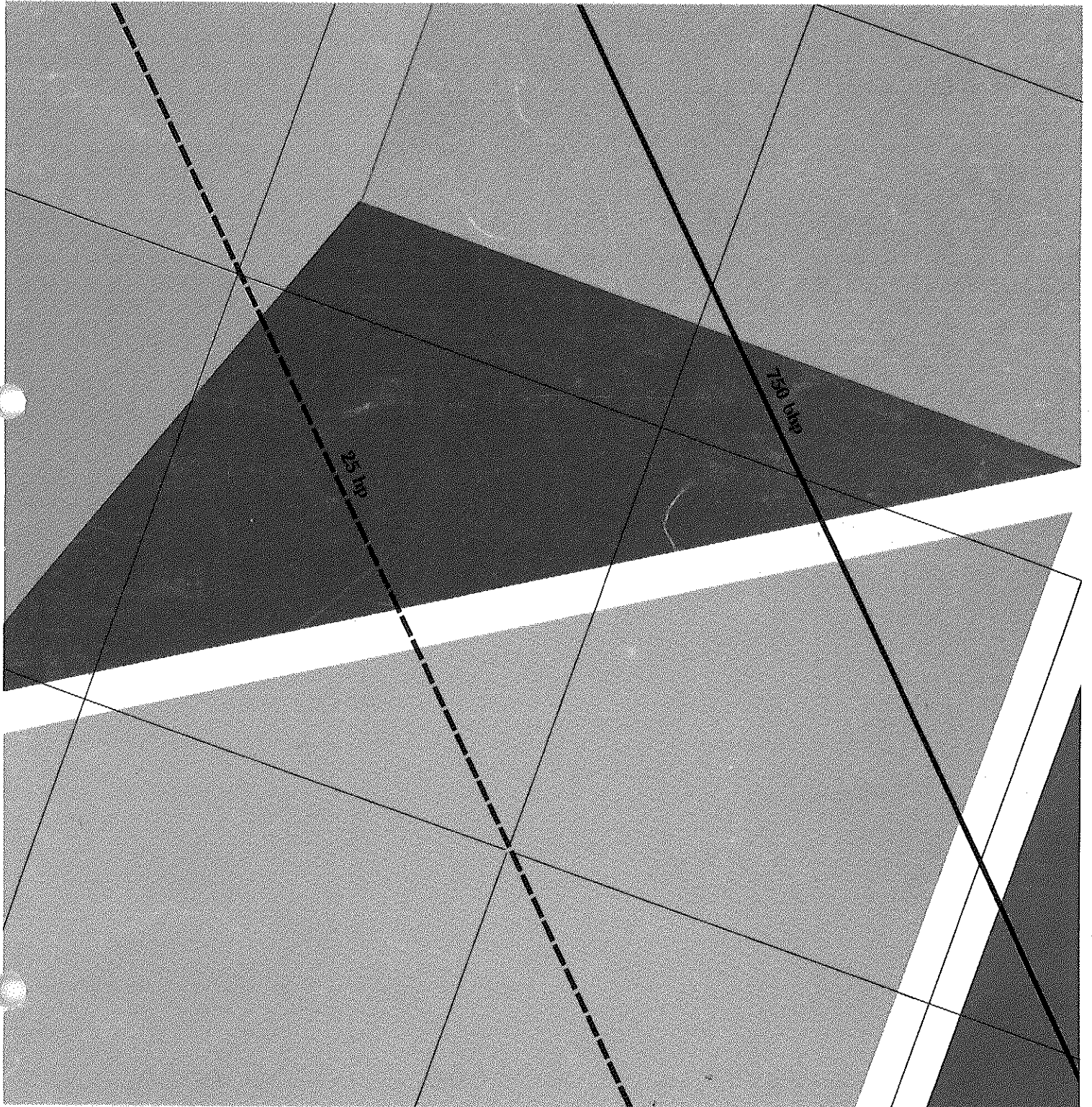
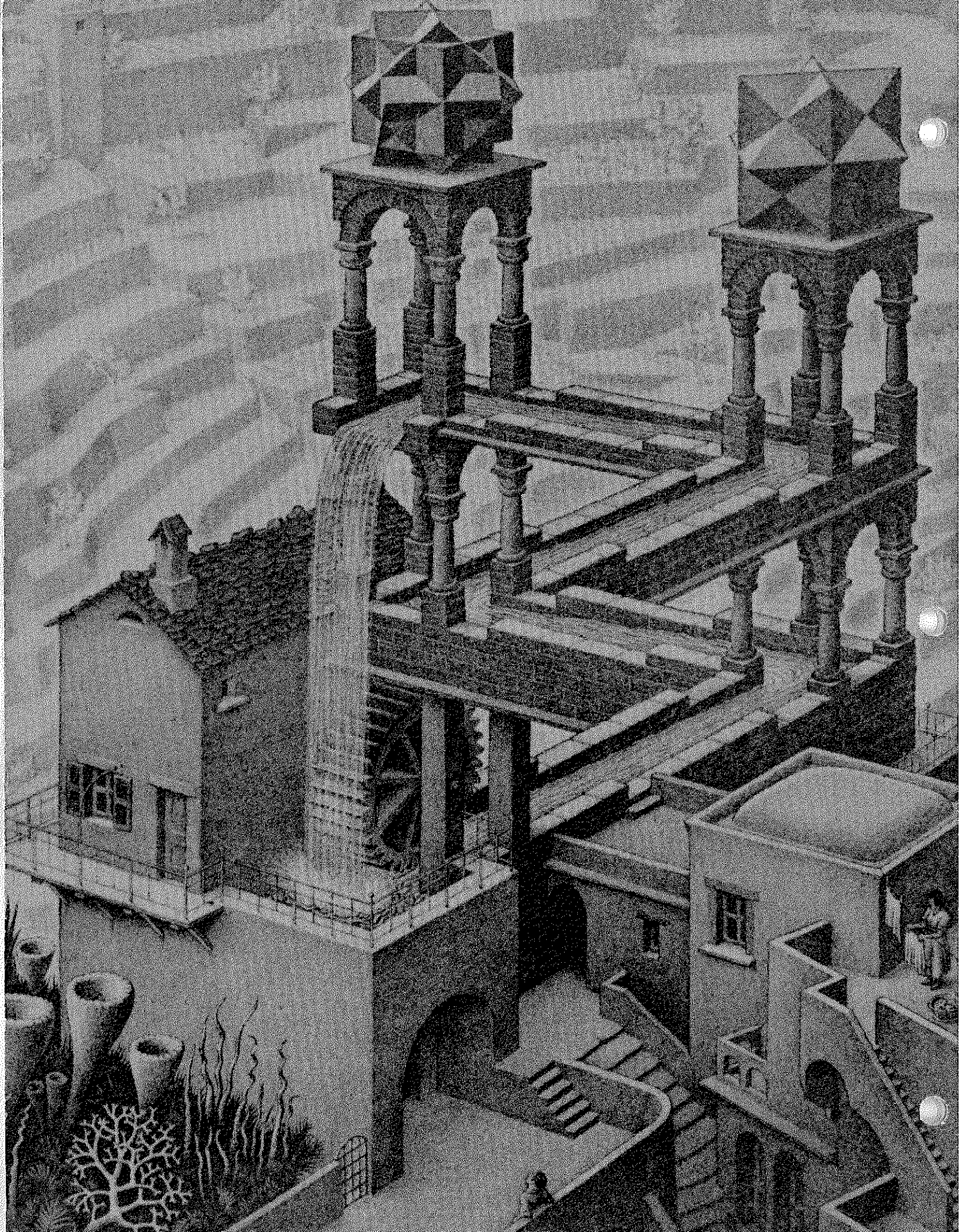


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### **New views of an age-old problem:**

#### **Improving fluid-handling efficiency.**

In this visual paradox depicting the ultimate in fluid-handling efficiency, water flows uphill and perpetual motion is a reality. Of course, the illusion created on paper must remain there, never to be translated into fact, while design and operating engineers continue their examination of real-life energy-consuming processes with an eye to improving efficiency. As energy costs rise, even small gains become significant.

#### **About the authors**

In this issue, Worthington Pump Inc. specialists analyze a variety of ways to improve over-all pump efficiency and save energy — at the drawing board and over the life of the installed system. I. J. Karassik, Director Pump Technology for WPI, discusses proper sizing, knowledgeable operation and appropriate maintenance procedures. J. H. Doolin, Director of Engineering for WPI's Standard Pump Division, develops an efficiency nomograph as a worthwhile tool in selecting the proper pump type for the job to be done. P. W. Polansky, Senior Application Engineer, reminds us of the "hidden" savings available through power recovery — one example of innovative thinking that can bring a system to its optimum efficiency. In the final article, co-authored by Austin R. Bush, Chief Engineer — Multistage Pumps, Warren H. Fraser, Chief Design Engineer, and I. J. Karassik, we get an inside look at the evolutionary process of pump progress at work — a process which leads to pumps of ever higher efficiencies, and sometimes to new problems as well.

A fine-tuned up-to-date pump, correctly selected for a well-designed system, does its essential work with minimum expenditure of energy. Take these articles as thought-starters to help achieve this optimum for your system.

# **pumpworld**

VOLUME 1 / NUMBER 1 / SUMMER 1975

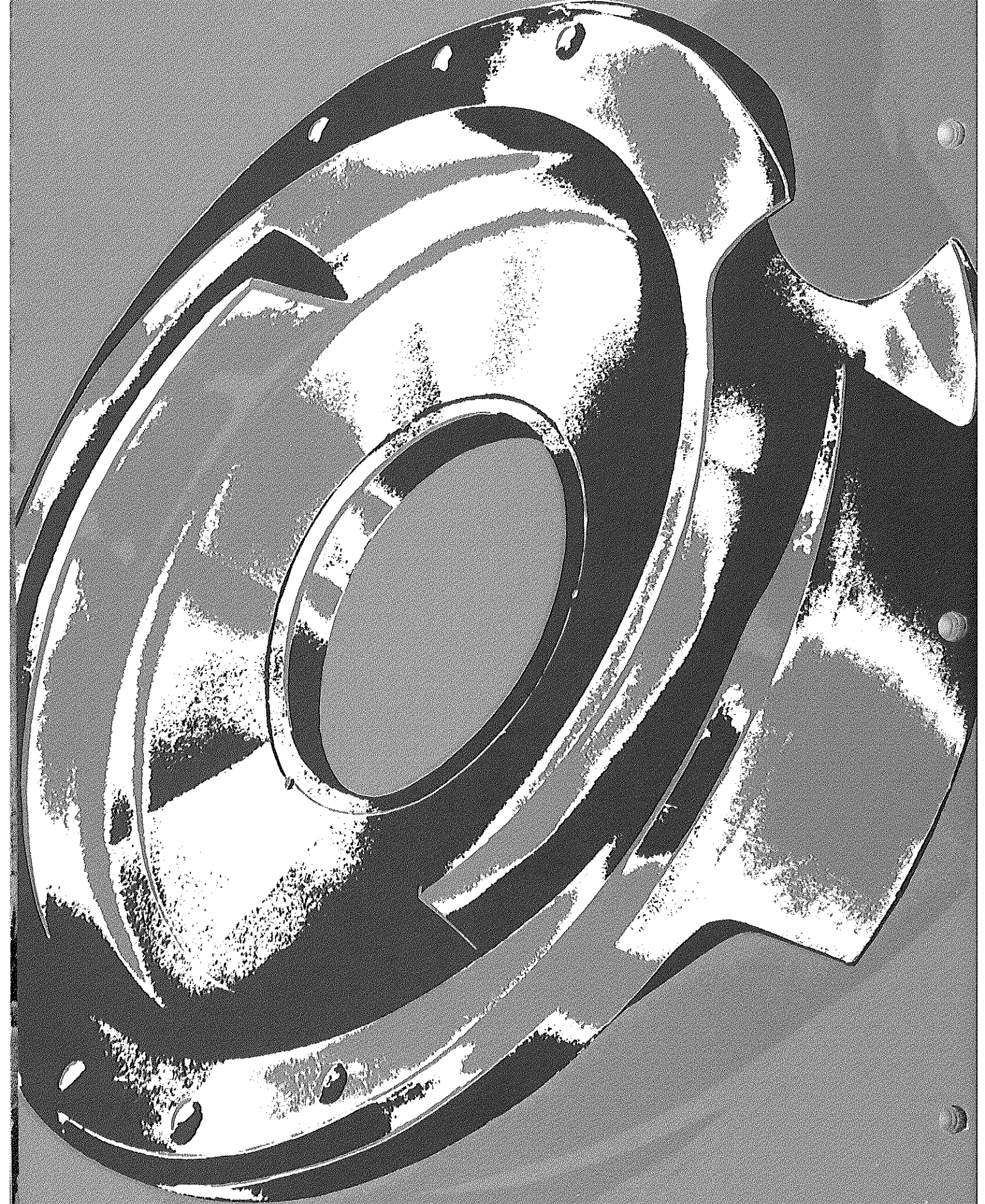
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- 5    Design and Operate your fluid system for improved efficiency.**
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Published by Worthington Pump Inc., 270 Sheffield St., Mountainside, New Jersey 07092.

Editor: Joseph A. Montana

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Cover: Saving energy by improving the efficiency of fluid-handling systems is the theme of this issue. Our cover is a section of the selection chart on page 13, a good starting point for the efficiency-minded design engineer.



# Design and operate your fluid system for improved efficiency.

By I. J. Karassik

When it comes to evaluating pumps, guaranteed or rated efficiency is only one point among many – and not necessarily the most important. Here, a leading pumping-systems expert analyzes additional factors that can result in more meaningful power savings when specifying new pumps – as well as operating and maintenance procedures that can improve performance of those already installed.

Pumps may well be the one machine essential to the well-being of our modern civilization. Every industrial process involves the transfer of liquids from one level of pressure or static energy to another. And so pumps, used to impart energy to fluids, are major consumers of energy in their own right – and a logical place to start an examination of energy-consuming processes with a view toward improving over-all efficiency.

Our most obvious reaction might be to look for pumps with higher efficiencies and favor the one which might exceed others by as little as  $1/2$  or 1 percent. All things being equal, there is some logic to this approach. But all things are seldom equal! To begin with, a small difference in guaranteed efficiency may have been obtained at the expense of reliability, either by providing smaller running clearances or using lighter shafts – a point to note when comparing specs. Moreover, the savings in power consumption obtained from a small difference in efficiency is rarely significant.

There are, however, quite meaningful savings to be realized involving other approaches to reducing or eliminating wasteful power consumption. In this article, we will analyze three major avenues to significant power savings:

1. Planning for energy conservation when designing the system – by avoiding the waste of power caused by oversized pumps.

2. Conserving energy when operating by using only one of two pumps in parallel for part-load.

3. Conserving energy by restoring internal clearances – at the right time.

### Oversized pumps waste power.

One of the greatest wastes of power involves the traditional practice of oversizing a pump by selecting design conditions with excessively “conservative” margins in both capacity and total head. This practice can lead to the strange situation in which a great deal of attention is paid to a small  $1/2$  or 1 percentage-point gain in efficiency, while ignoring a potential power savings of as much as 15 percent through an overly conservative attitude in setting the required conditions of service.

It is true that some margin should always be included, mainly to provide against the wear of internal clearances which will, in time, reduce the effective pump capacity. How much margin is a fairly complex question which we will consider when we analyze the savings in power consumption to be realized from restoring internal clearances to their original value. The point here is that, traditionally, system designers have piled margin on top of margin, “just to be safe.” Some of this margin can definitely be eliminated.

### Consider the curves.

A centrifugal pump operating in a given system will deliver a capacity corresponding to the intersection of its head-capacity curve with the system-head curve, providing the available npsH is equal to or exceeds required npsH. To change this operating point requires changing either the head-capacity curve, the system-head curve, or both. The first can be accomplished by varying the speed of the pump (Figure 1), while the second requires altering the friction losses by throttling a valve in the pump discharge (Figure 2).

In the majority of pump installations, the driver is a constant-speed motor, so throttling is the method used to change the pump capacity. Thus, if we have provided too much margin in the selection of the pump head-capacity curve, the pump will have to operate with considerable throttling at all times. In effect, we are first expending

power to develop a much higher pressure than needed, and then wasting a part of it in frictional losses in order to reduce pump delivery to the desired value – an obvious waste of power, causing additional equipment wear and tear as well.

If, on the other hand, we permit the pump to operate unthrottled, the flow into the system will increase until that capacity is reached where the system-head and head-capacity curves intersect.

### Example: an overly conservative selection requires 165 bhp.

Let's examine a concrete example, in which the maximum desired capacity is 2700 gpm, the static head is 115 ft. and total friction losses, assuming 15-year-old pipe, are 60 ft. Total head required at 2700

Figure 1 – Varying pump capacity by varying the speed.

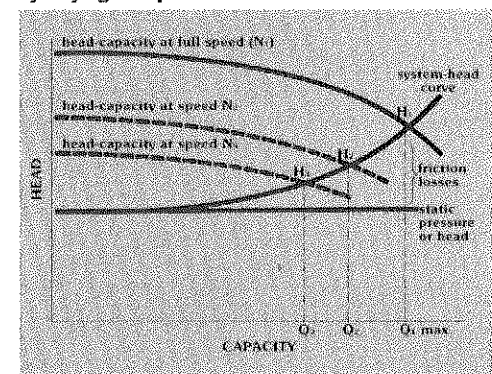


Figure 2 – Varying pump capacity by throttling the discharge.

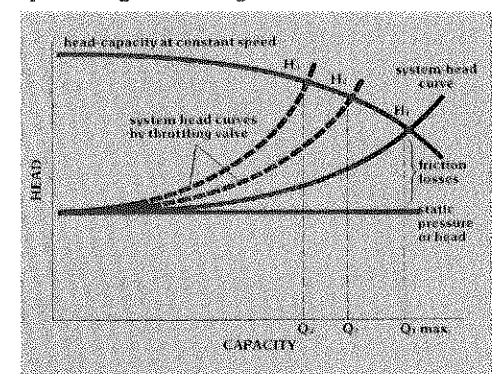
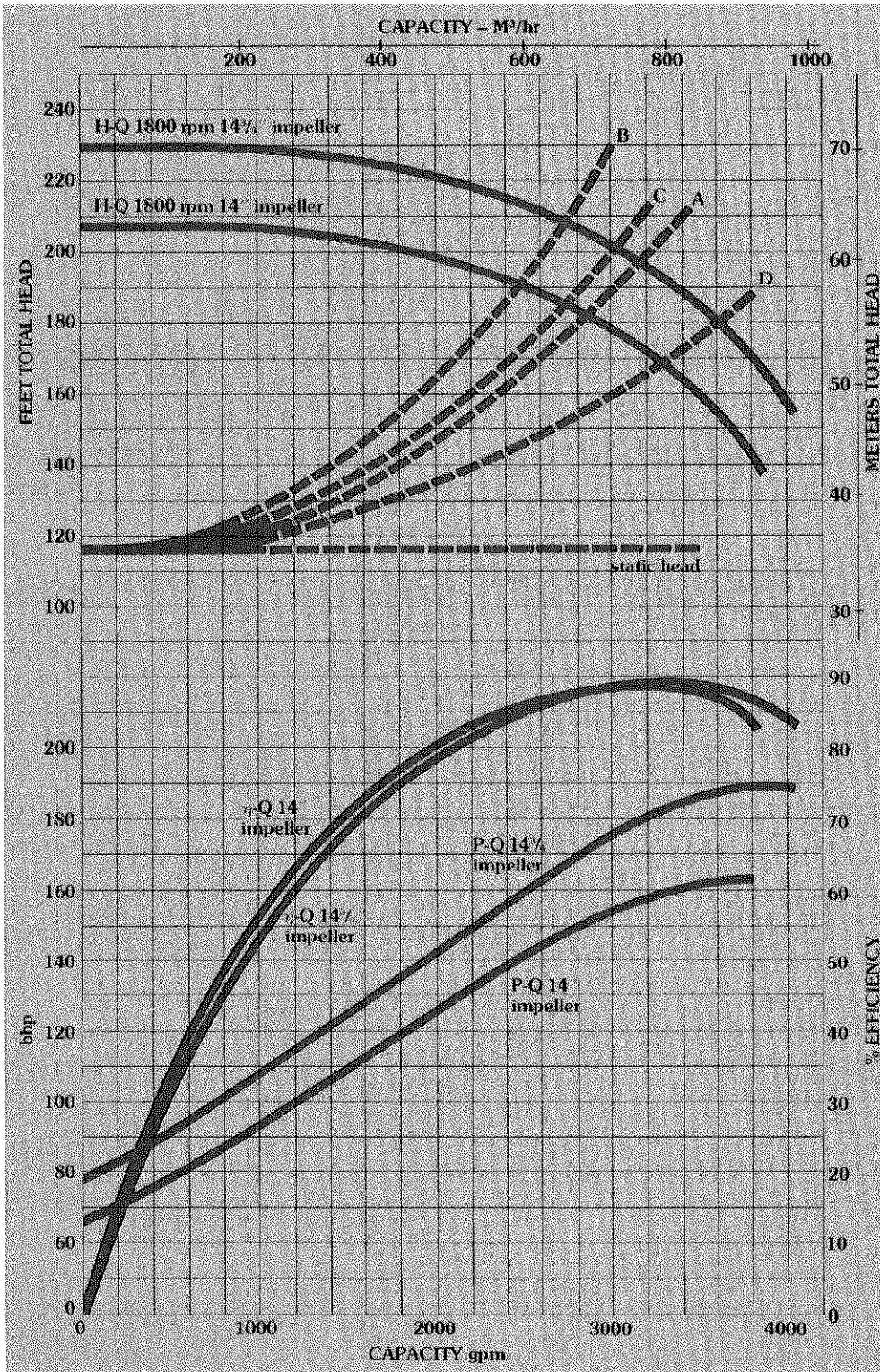


Figure 3 – Effect of oversizing a pump.



gpm is therefore 175 ft. We can now construct a system-head curve, shown as curve "A" in Figure 3.

If we add a margin of about 10 percent to the desired capacity and then, as frequently is done, add some margin to the total head above the system-head curve at this rated flow, we end up selecting a pump for 3000 gpm and 200 ft. total head. The performance of such a pump, with a 14 3/4 in. impeller, is superimposed on system-head curve "A" in Figure 3.

Of course, this pump develops excess head at our maximum desired capacity of 2700 gpm. If we wish to operate at that capacity, this excess head will have to be throttled. Curve "B" is the system-head curve that will have to be created by throttling.

At 3000 gpm the pump required 175 bhp, so we must drive it with a 200 hp motor. At the desired capacity of 2700 gpm, operating at the intersection of its head-capacity curve and curve "B", the pump will absorb 165 bhp.

This pump has been selected with too much margin!

**Example: a very adequate pump uses 145 bhp.**

For the same conditions stated above, we can safely select a pump with a smaller impeller diameter, say 14 in., and a head-capacity curve as shown in Figure 3. It will intersect curve "A" at 2820 gpm, giving us about a 4 percent margin in capacity, which is sufficient. We will still have to throttle the pump slightly to arrive at system-head curve "C". However, the power consumption at 2700 gpm is now only 145 bhp instead of the 165 bhp required with our first, overly conservative selection. *This means a very respectable 12 percent saving in power consumption!*

Furthermore, we no longer need use a 200-hp motor; a 150-hp motor will do quite well. The saving in capital expenditure is another bonus from not oversizing.

Our savings may actually be even greater than we have shown. In many cases, the

pump may be operated unthrottled, the capacity being permitted to run out to the intersection of the head-capacity curve and curve "A". If this were the case, a pump with a 14¾ in. impeller would operate at approximately 3150 gpm and take 177 bhp. If a 14 in. impeller were to be used, the pump would operate at 2820 gpm and take 148 bhp. *We could be saving over 16 percent in power consumption.*

Our real margin of safety is actually greater than the 4 percent indicated above. Remember that the friction losses we used to construct system-head curve "A" were based on losses through 15-year-old piping. The losses through new piping would be only 0.613 of the losses we have assumed. The system-head curve for new piping is curve "D". If the pump we had originally selected with 14¾ in. impeller were to operate unthrottled, it would produce 3600 gpm and take 190.5 bhp. A pump with a 14 in. impeller would intersect system-head curve "D" at 3230 gpm and take 156.5 bhp, a power saving of almost 18%.

There is obviously no question that important energy savings can be made if we avoid unnecessarily conservative margins of safety when conditions of service are determined.

**Existing installations:  
It's never too late.**

But what of existing installations in which the pump or pumps have excessive margins? Is it too late to achieve these savings? Far from it! As a matter of fact, it is possible to establish the true system-head curve even more accurately by running a performance test once the pump has been installed and operated. Once a reasonable margin has been selected, three choices become available:

1. The existing impeller can be cut down to meet the real conditions of service required for the installation.
2. A replacement impeller with the necessary reduced diameter may be ordered from the pump manufacturer. The original impeller is then stored for future use if friction losses are ultimately increased with time or if greater capacities are ever required.

3. If two separate impeller designs are available for the same pump, one with a narrower width, as is sometimes the case, a replacement may be ordered from the pump manufacturer. Such a narrower impeller will have its best efficiency at a lower capacity than the normal width impeller. It may or may not need to be of smaller diameter than the original impeller, depending on the degree to which excessive margin had originally been provided. Again, the original impeller is put away for possible future use.

**Variable speed operation offers possible savings.**

The vast majority of motor-driven centrifugal pumps are operated at constant speed, but some installations do take advantage of the possible savings in power consumption provided by variable speed operation. Wound-rotor motors were once frequently used for this purpose, but today's practice is to interpose a variable speed device, such as a magnetic drive or a hydraulic coupling, between the pump and the electric motor. It is then possible to match pump operating speed to the exact conditions of service without throttling.

For instance, consider a system-head curve for a new installation that corresponds to curve "D" in Figure 3. If we wish the pump to handle exactly 2700 gpm and 152 ft. head, we could use a 14¾ in. impeller and run it at 87.5 percent of design speed. We could also use a 14-in. impeller at 92.2 percent of design speed. In either case, the pump would take 118 bhp. Compare this with the 165 bhp taken by a constant speed pump with a 14¾ in. impeller at the same 2700 gpm.

Not all this horsepower difference is savings, however, because a variable-speed device has its own losses, as evidenced by the formula for calculating power input to the variable speed device:

$$\text{motor bhp} = \frac{\text{pump bhp}}{\text{pump speed}} \times \frac{\text{motor speed}}{\text{pump speed}} + \text{fixed hp losses of variable speed device}$$

Magnetic drives as well as hydraulic coup-

lings generally have a slip of about 3 percent at full load and additional fixed losses of about 1 percent. Thus, the efficiency of such a variable-speed device at full load is about 96 percent.

In the example given above, the power consumption at 2700 gpm with a 14¾ in. impeller would be approximately 141 bhp at the motor, so we would still be saving 24 bhp by using a variable speed device. And even in this case – an existing constant-speed installation – it is not necessarily too late to achieve these power savings. In many cases it is possible to modify an installation by installing a magnetic drive or a hydraulic coupling between the pump and its constant speed motor.

To decide whether such a modification is advisable, plot the actual system-head curve. This will give you the speed required at various capacities over the operating range, and the motor horsepower input to the variable-speed device over this range. The difference between this horsepower and the pump bhp at constant speed represents potential power savings. Next, assign a predicted number of hours of operation at various capacities and calculate the potential yearly savings in hp-hrs or kw-hrs. Finally, compare these savings to the cost of the conversion to determine whether the cost of the modification to variable-speed operation is justified. If it is, proceed with the modification. If not, the options of smaller impellers or of narrower impellers are still available.

**Run one pump instead of two.**

Many installations are provided with so-called "half-capacity" pumps: two pumps operating in parallel to deliver the required flow under full load conditions. If the service on which these pumps are installed is such that required flow varies over a considerable range, important power savings may be possible through improved operating practices. Too often, with this arrangement, both pumps are kept on the line even when demand drops to a point where a single pump can carry the load (see Figure 4).

### **Example: Parallel pumps supply a total 3200 gpm.**

Just how important are the savings resulting from operating a single pump whenever it can meet the required demand? Let us assume that full load conditions correspond to 3200 gpm and 50 ft. head, of which 20 ft. represent static head and 30 ft. the friction loss in the system. For simplicity, we'll neglect the question of capacity or pressure margins and imagine that the pumps carry their full load with throttling valves wide open, operating at constant speed.

Each pump will then be designed to meet full load conditions at 1600 gpm and 50 ft. head and their performance is shown in Figure 5. At that capacity pump efficiency is 84 percent. Each pump will take 24.1 bhp, with a total power consumption of 48.2 bhp for the installation.

Capacity that can be delivered into a given system is determined by the intersection of the head-capacity curve of the pump (or group of pumps) serving the system and of the system-head curve. Therefore, if we want to reduce the flow to 1600 gpm and still keep both pumps on the line, it will be necessary to throttle the pump discharge and create a new system-head curve. Under these conditions, each pump will deliver 800 gpm at a head of 68 ft., with an efficiency of 66 percent and using 20.9 bhp. Total power consumption will be 41.8 bhp.

### **Using one pump for half load saves 42.5 percent.**

It would be possible, however, to shut down one of the pumps and meet the capacity requirement of 1600 gpm with a single pump. The discharge would be throttled considerably less and the system-head and head-capacity curves would intersect at 1600 gpm and 50 feet. The pump's power consumption would be 24.1 bhp. The saving resulting from operating a single pump would be 17.7 bhp (41.8 minus 24.1). This is a saving of 42.5 percent over the power requirements imposed by the practice of operating two pumps at all loads!

If we were to assume that the process served by this pump installation is such that during 20 percent of the yearly operating hours it requires flows of 50 percent or less, the yearly power saving resulting from shutting one pump down whenever possible would be in the order of over 8 percent.

As a matter of fact, a single pump could carry much more than the 1600 gpm corresponding to the half-load in this case. The head-capacity curve of this pump intersects with the system-head curve at approximately 2000 gpm. In other words, we can save a considerable amount of power by shutting down one of the pumps whenever the demand falls below 2000 gpm.

### **A better practice for longer pump life.**

There are other benefits to such an operating procedure. In the first place, if we assume 8500 yearly operating hours for the process served by these pumps, 20 percent taking place at flows of 50 percent of maximum flow or lower, each pump will operate for only 7650 hours a year instead of 8500, extending the calendar life of all running parts by over 11 percent.

But more important than this straight arithmetical effect is the fact that *pumps which frequently operate at reduced capacities do not have as long a life as pumps operated more nearly to their best efficiency point.* Thus, running only one pump whenever it can handle the required flow will add much more life to each pump than just the arithmetical difference in operating hours.

### **Safety: as good or better.**

But what about safety? Operators who keep both pumps on the line at all times feel they are doing so for safety reasons: if one pump fails, the other can still supply a portion of the required flow. Then, until the failed pump can be restored to service, the process served by the pumps can still operate at reduced load. If a standby pump is available, it can be brought into service without complete flow interruption.

However, an equally good case can be

made for running the single pump at light loads. Should an accident occur when two pumps are on the line, there is a good chance that both pumps will be damaged at once. We are actually on safer ground with only one pump running, considering that most motor-driven pumps are started across the line and that full speed can be achieved in seconds.

The evaluation of any energy conservation program consists of comparing the amount of energy saved with the cost of implementing the program. The remarkable fact is that implementing this procedure costs nothing – only the time required to explain to the operators how this procedure saves power.

### **Restore internal clearances.**

The rate of wear of internal clearances depends on many factors. To begin with, it increases in some relation to the differential pressure across the clearances. It also increases if the liquid pumped is corrosive or contains abrasive foreign matter. On the other hand, the rate is slower if hard, wear-resisting materials are used for the parts subject to wear. Finally, wear can be accelerated very rapidly if momentary contact between rotating and stationary parts occurs during the operation of the pump.

As running clearances increase with wear, a greater portion of the "gross" capacity of the pump is short-circuited through the clearances and must be repumped. The effective or "net" capacity delivered by the pump against a given head is reduced by an amount equal to the increase in leakage. While in theory the leakage varies approximately with the square root of the differential pressure across a running joint (and therefore with the square root of the total head), it is sufficiently accurate to assume that the increase in leakage remains constant at all heads. Figure 6 shows the effect of increased leakage on the shape of the head-capacity curve of a pump. Subtracting the additional internal leakage from the initial capacity at each head gives a new head-capacity curve after wear has taken place.



**When is renewal worth the cost?**

To decide, we must compare the cost of restoring the internal clearances with the value of the power saved by operating a pump with original-size clearances. This cost is relatively easy to determine: we can obtain prices on new parts and estimate the cost of labor to carry out the task. But how about the savings?

The fact is that savings are not the same for every pump. Both analytical and experimental data indicate that leakage losses vary considerably with the specific speed of a pump.

$$\text{specific speed } (N_s) = \frac{\text{rpm} \sqrt{\text{capacity in gpm}}}{\text{total head in feet}^{3/4}}$$

Figure 7 shows the relation between the leakage losses of double-suction pumps and their specific speed.

Let us examine a few typical cases. The pump illustrated in Figure 3, when fitted with a 14-in. impeller, has its best efficiency at 3200 gpm and 170 ft. head. Its specific speed is:

$$N_s = \frac{1800 \sqrt{3200}}{170^{3/4}} = 2160$$

From Figure 7 we can estimate that its leakage losses—when the pump is new—are about 1.4 percent. So when internal clearances have increased to the point where this leakage has doubled, we can regain approximately 1.4 percent in power savings by restoring the pump clearances.

Now consider a pump designed for 180 gpm and 250 ft. head at 3550 rpm. Its specific speed is:

$$N_s = \frac{3550 \sqrt{180}}{250^{3/4}} = 755$$

Such a pump will have leakage losses of about 5 percent. If clearances are restored after the pump has worn to the point where its leakage losses have doubled, we can count on 5 percent power savings.

Thus, restoring clearances of pumps with lower specific speeds, gives greater returns. In addition, pumps with higher head per stage generally have lower specific speeds than lower head pumps and, all else being equal, wear increases with higher differential heads. Thus, one will generally find more reasons to renew clearances of high head pumps and more savings from doing so.

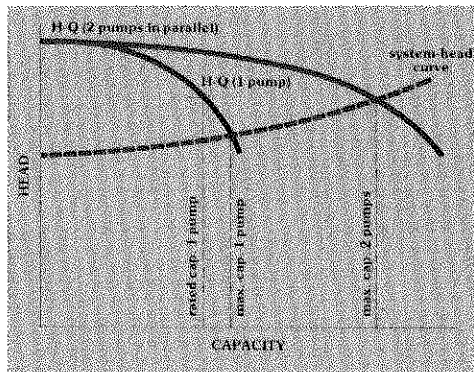
**Energy conservation: a new way of thinking.**

Never before in the history of our industrial society have we faced as pressing a need to conserve energy. For over a century, the development of energy technology has been forced by the constantly rising demand for energy. Now, we can expect new developments in technology to be inspired by the current shortage—temporary or

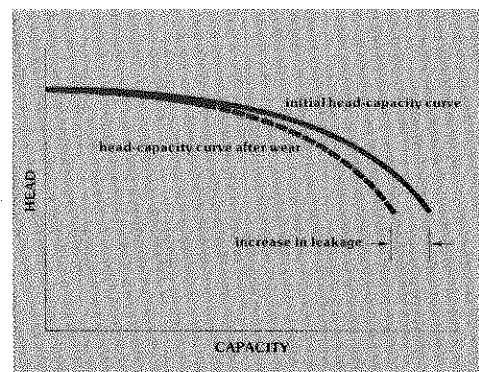
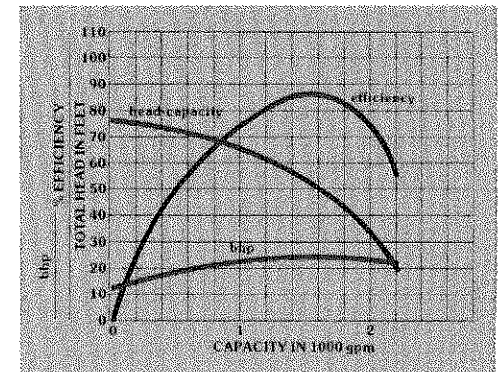
permanent—in the supply of fuels such as oil, natural gas, and even coal, as well as by new ecological constraints on the use of available sources of power.

Besides new technology, a new attitude on the use and abuses of energy is required on the part of everyone involved in industry. If this article can help design and operating engineers conserve even a small percentage of the power consumed by their pumping equipment, it will be a good beginning.

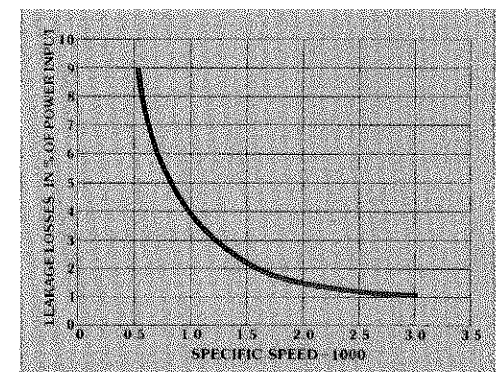
**Figure 4 – Two-pump and one-pump operation.**



**Figure 5 – Head-capacity curve of a 1600 gpm, 50 ft. head pump.**



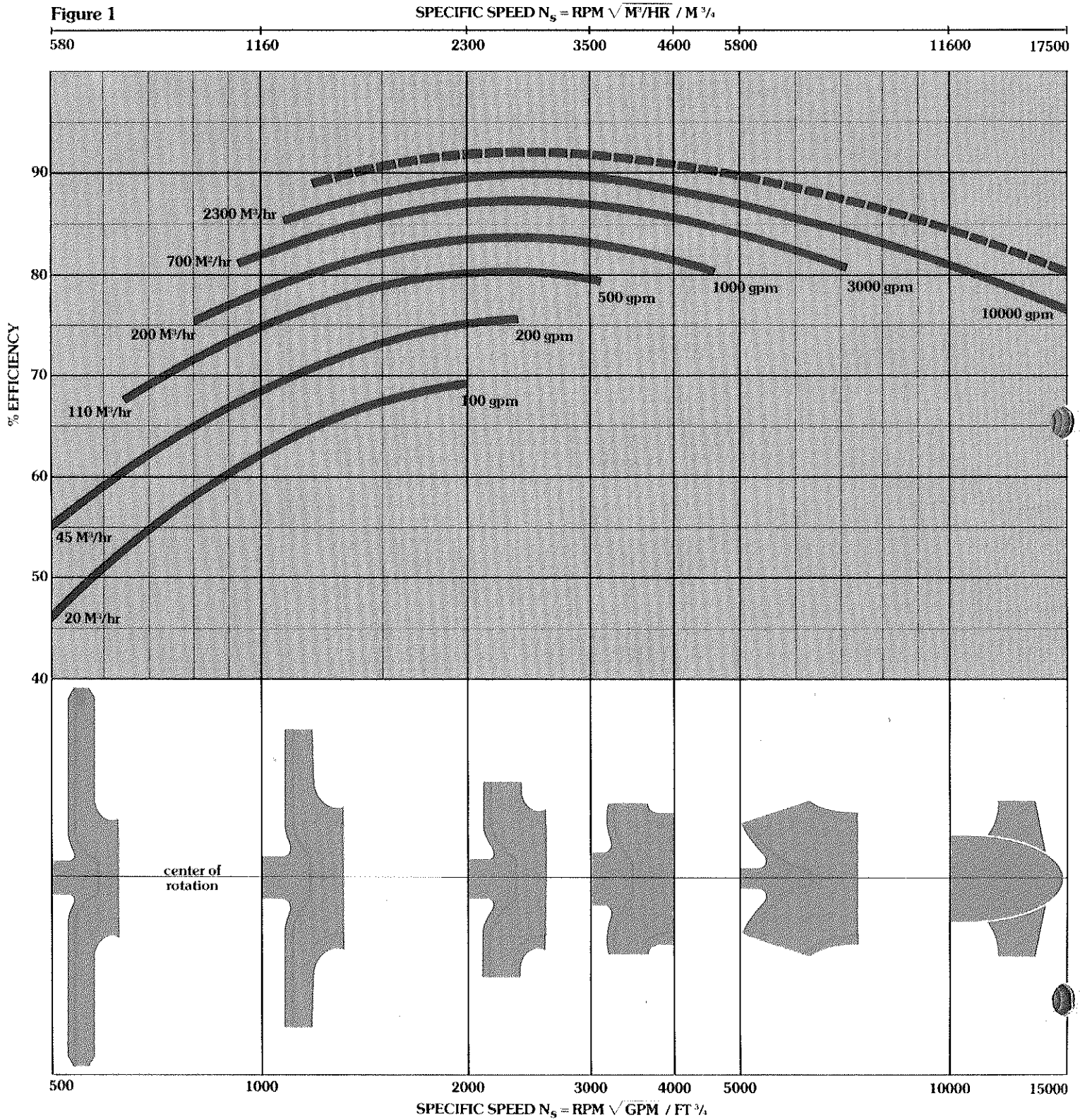
**Figure 6 – Effect of internal wear on a centrifugal pump head-capacity curve.**



**Figure 7 – Leakage losses vs. specific speed of double-suction pumps.**

# How to select pump type for best efficiency.

By J. H. Doolin



With rising energy costs, the yearly operating cost of a typical centrifugal pump might well exceed its purchase price! At current rates, for example, assuming only 2000 hours per year operation, a 10-hp pump might cost \$600 to purchase — but \$1,000 to operate. So, it becomes more important than ever to get off to a good start by properly matching your pump to the job to be done. Taking into account the many variables of pump design, a pump efficiency nomograph can be developed as a worthwhile starting point in selection. This article shows how it can be done.

For general industrial service, the centrifugal is the largest single category of pumps. Reciprocating pumps are usually limited to low-capacity, high-pressure applications, and rotary pumps such as the gear and vane type are more or less reserved for the viscous fluids. When it comes to clear, low-viscosity fluids such as water, the obvious choice is a centrifugal.

But just specifying a centrifugal pump leaves many variables still unclarified. Limit yourself to a single-stage pump and you reckon with questions such as driver speed and single versus double suction. Or would a multistage pump be better, or a pump with a built-in speed increaser? And how about inducers?

**Best efficiency is a function of impeller geometry.** Most of these variables can be put in

proper perspective for logical evaluation if we begin by looking at the effect of each on pump efficiency.

The centrifugal pump is a hydrodynamic machine, with an impeller designed for one set of conditions of flow and total head at any given speed. Impeller geometry or shape runs the gamut from very narrow, large-diameter impellers for low flows, through much wider impellers for higher flows, to the specialized propeller for highest-flow, low-head conditions. Unfortunately, not all designs can have equally good efficiency. In general, medium flow pumps are most efficient; extremes of either low or high flow will drop off in efficiency, as shown by the chart in Figure 1.

The best attainable efficiency is a function of impeller geometry, or the dimensionless factor called *specific speed*:

$$\text{specific speed } (N_s) = \frac{\text{rpm} \sqrt{\text{capacity in gpm}}}{\text{total head in feet}^{3/4}}$$

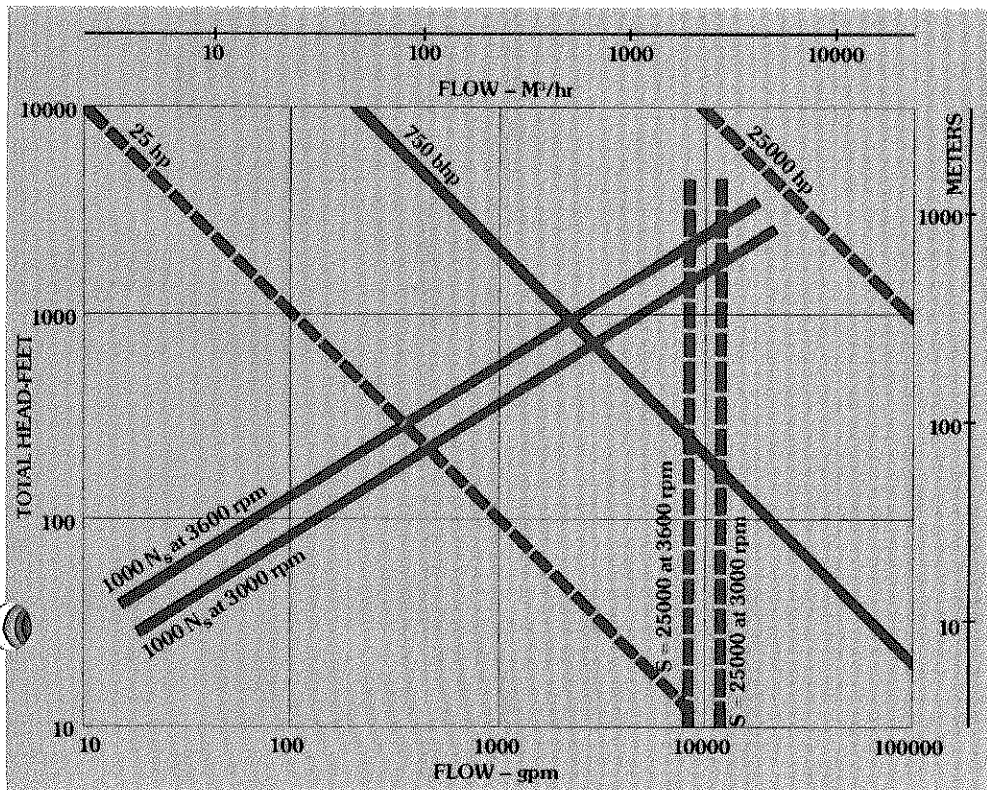
In the English system, flow is measured in gpm (gallons per minute) and total head is measured in feet. In the metric system, it's m<sup>3</sup>/hr (cubic meters per hour) and meters.

While efficiency tends to drop off at high specific speed, the greater difficulty is at specific speeds below 1000 (English system). In Figure 1, the slope of the efficiency curve below 1000 becomes quite steep and efficiency falls off rapidly. And so, other factors being equal, for good efficiency it's best to avoid pumps designed for specific speeds below 1000.

#### Plotting a line for single stage efficiency.

Since 2-pole motor speeds of 3600 or 3000 rpm are normal for many pumps, we can plot a line for each speed on a field of total head versus flow which represents all the design points for  $N_s = 1000$  (see Figure 2). In this chart, all conditions to the right of the diagonal line have a specific speed greater than 1000; a single-stage centrifugal pump selected for these conditions will run at good efficiency. All applications to the left will have a specific speed

Figure 2



less than 1000; efficiency of a single-stage centrifugal pump for these conditions will be poor.

Immediately to the left of the  $N_s = 1000$  line we have two alternatives that can operate efficiently: multistage pumps with two or more impellers, or higher-speed pumps.

Another factor which must be considered is suction specific speed. This is similar to  $n_s$ , but net positive suction head (npsH) is used in place of total head, so:

$$\text{suction specific speed (S)} = \frac{\text{rpm} \sqrt{\text{capacity in gpm}}}{\text{npsH}^{3/4}}$$

While normal centrifugal-pump impellers are designed so  $s$  is about 10,000 in the English system, with today's technology a suction specific speed of 25,000 is readily attainable. If we assume that npsH for many water applications is 30 ft. or more, we can locate a line around 10,000 gpm which is the limit for 25,000  $s$ , 30 ft. npsH, and 3500 or 3000 rpm (Figure 2). Beyond 10,000 gpm we can still use single-stage centrifugal pumps with good efficiency, but the driver speed must be reduced to 1750 or 1450 rpm or even lower.

#### An interesting phenomenon.

At this point an interesting phenomenon occurs. If we combine the formulas for  $N_s$  and  $S$  and solve for total head we get:

$$\text{total head}^{3/4} = \frac{S \times \text{npsH}^{3/4}}{N_s}$$

Substituting our limits of  $S = 25,000$ ,  $\text{npsH} = 30$  and  $N_s = 1000$  we get total head = 2200 ft.

In Figure 2, the diagonal lines for  $n_s = 1000$  intersect the 10,000 gpm line at about 2200 ft. head. The phenomenon is that *any diagonal line for  $N_s = 1000$ , at any speed, will always intersect the vertical capacity limit of 25,000  $S$  at 2200 ft. head.*

In other words, it is impractical to design a single-stage pump for more than 2200 ft. head, regardless of speed, without either

loss in efficiency or increase in npsH beyond 30 ft.

In Figure 3 we have added these additional limits, plus a minimum size limit. While this line can't be defined as precisely as the others, in general there are limitations at speeds over 20,000 rpm, or where impeller inlet diameter is greater than the outlet.

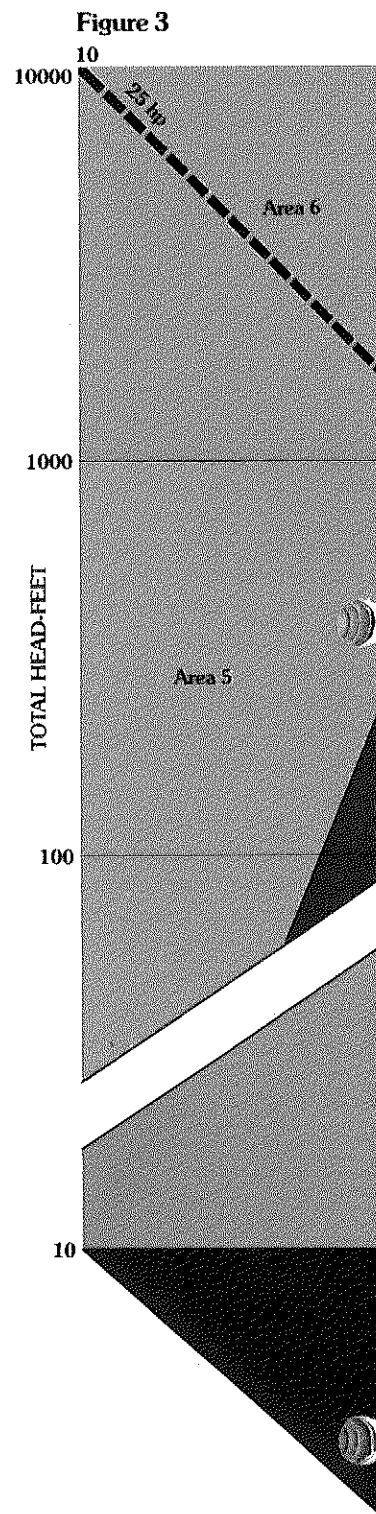
What about the two other variables, inducer and double-suction impeller? The inducer was included in the analysis when we selected a value of 25,000 for suction specific speed, because this speed cannot be attained without an inducer. If inducers are not used, the  $s$  value must be reduced to 10,000 and the maximum flow for 2-pole speeds is reduced to 1300-1800 gpm.

Using a double-suction impeller without inducer, the maximum flow for 2-pole speeds would be about 2600 to 3600 gpm.

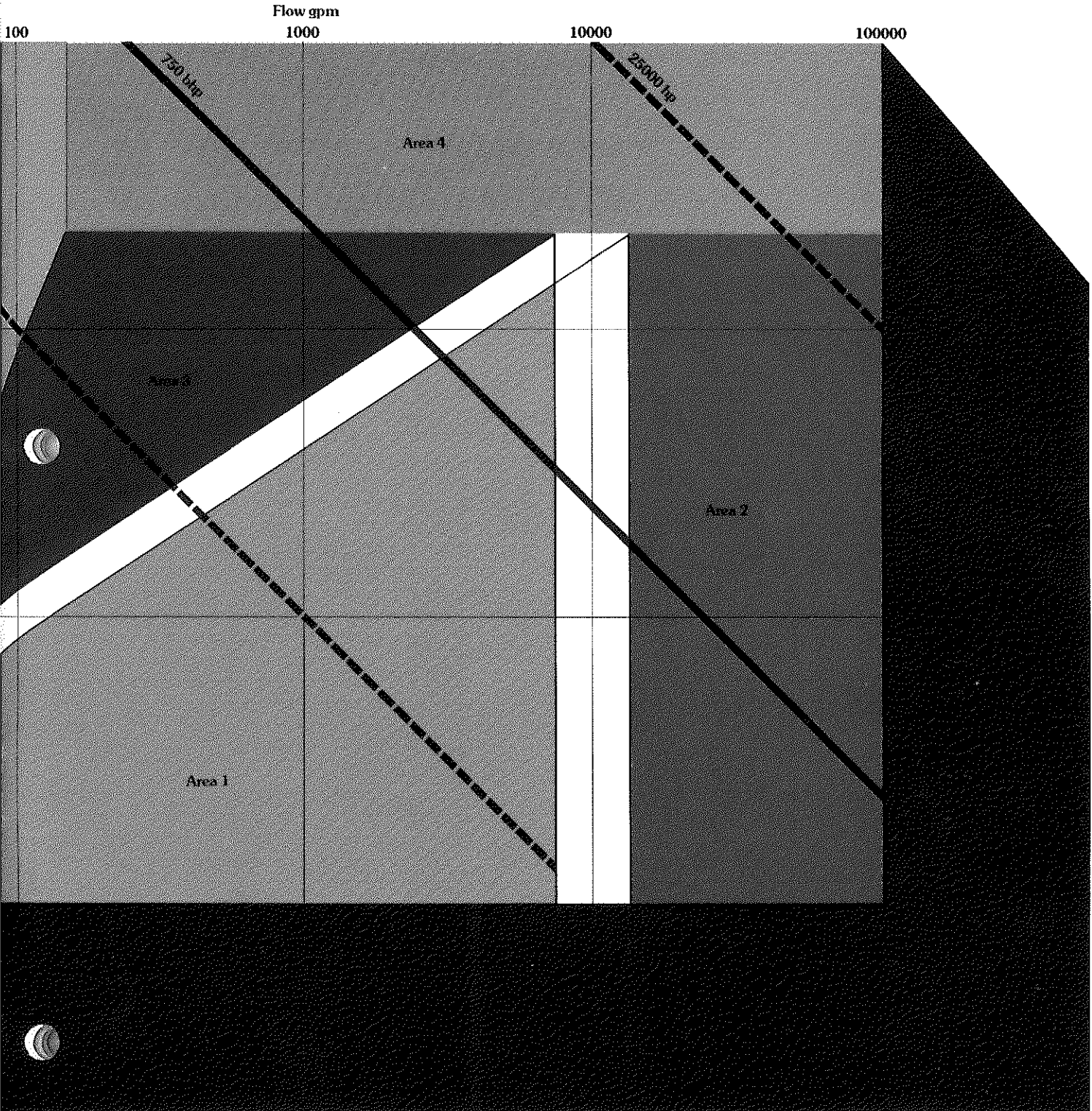
#### A good beginning.

Now, let us look at the final results in Figure 3 — and the clues it provides to pump selection depending on conditions of service. Conditions falling in Area 1 are appropriate for standard, single-stage pumps at 2-pole speeds. Area 2 is also covered by single-stage pumps, but at speeds lower than 2-pole. Area 3 gives us a choice: either high-speed single-stage pumps, or multistage pumps at 2-pole speeds. Area 4 is reserved exclusively for multistage pumps. Area 5 is best served with small multistage pumps with as many as 25 stages. Finally, there is Area 6, the domain of the reciprocating pumps.

Of course, our nomograph is an indication of probabilities — not the last word in pump selection! The boundaries of each area should be considered as overlapping, since small changes in the assumed limits will alter the boundary lines. Take Figure 3 as a good starting point — a design aid — when weighing the alternatives in selecting pumps to operate at best efficiency for your conditions of service.

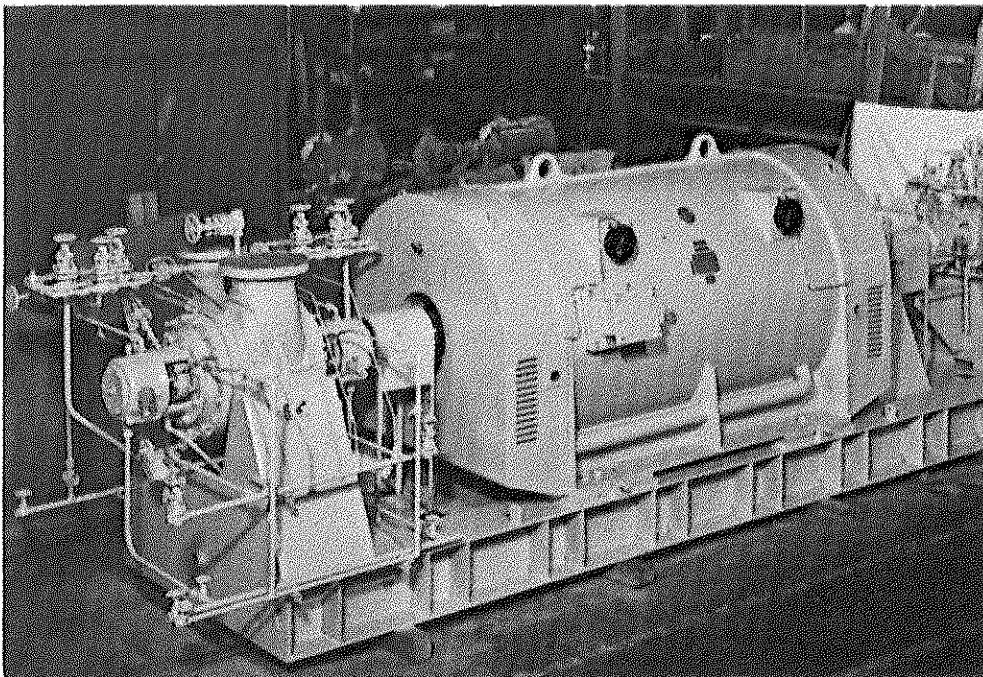


- 2 pole single stage
- single stage
- multi-stage
- multi-stage
- reciprocating
- high speed single stage or multi-stage

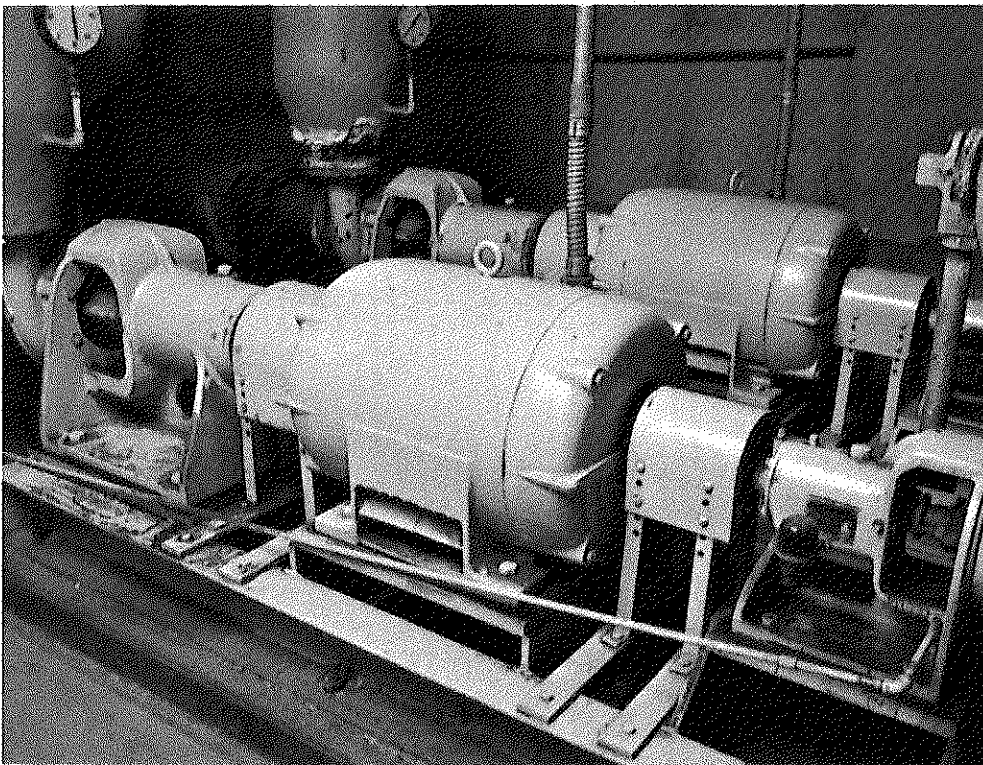


# The pump as water turbine

By P. W. Polansky



Worthington 6HDS-152 pump with 6HNN-124HT pump used as water turbine.



Worthington 3CNE-72 pump with 3CNE-52 pump used as water turbine.

In our quest for energy conservation, an unconventional approach may pay dividends in increased system efficiency. One such imaginative approach is to use a centrifugal pump as a water turbine when the situation presents itself. This concept has excellent potential for power recovery where good quantities of moderate-pressure water are available to drive the "turbine" – and a number of other advantages as well. Although not a new concept, it is one that should be receiving more attention as an adjunct to today's prime power sources.

A cooling system using water pumped at 500 gpm and 230 ft. head through cooling coils for air-conditioning fans located on the 11th floor of a building; Hardly an unconventional installation, but one which presented the opportunity to realize a significant energy saving. The pump was a moderate-sized Worthington 3CNE-72. Rather than just "dump" the water back to the well located below basement level, losing the advantage of its head, the system was designed so water discharged from the coils is first circulated through a smaller pump acting as a water turbine. The pump/turbine, a Worthington 3CNE-52, is coupled to the same dual-shaft 40-hp motor used to power the main pump, as shown in Figure 1. In this manner, more than 8 hp is recovered to help pump the well water back to the 11th floor.

## Calculating recoverable hp

Here is the basic approach to calculating horsepower recovered:

$$\text{from hydraulics, hp} = \frac{Q \times H \times E}{3960}$$

where hp = horsepower output, Q = capacity in gpm, H = head in ft. of water, and E = efficiency.

For the above example,  $Q = 500$  gpm;  
 $H = 230 - 130 = 100$  ft.;  $E = 68$  percent  
 (assuming the unit has the same efficiency  
 applied as a turbine or a pump). So

$$\text{HP} = \frac{500 \times 100 \times .68}{3960} = 8.6$$

### System requirements

The main requirement is a large amount of water available at moderate pressure to drive the pump/turbine. No special piping arrangements are required other than what is necessary for a well-installed pump, except that if only one valve is used, it must be put at the pump suction (turbine discharge) so the pump can't run dry.

### Selection data

To determine pump selection, the application engineer must know available head and capacity, operating speed range and horsepower required within the speed range. For instance, if the speed range is 1200 to 1800 rpm for the driven machine, horsepower required at both these speeds as well as several intermediate speeds is needed, since the unit will operate at some

intersection between power available and power required. Temperature is also needed and, should you be considering energy conservation via some process fluid, specific gravity and viscosity.

Whether the machine is applied as a pump or a turbine, its capacity, head and efficiency are considered essentially equal. This has been generally verified through a series of test programs and practical field applications. However, variations may exist, so it is recommended that both capacity and head available to the unit as a turbine be somewhat in excess of what it would develop when used as a pump. This gives a "ballpark" selection to see if the application makes sense.

### Typical applications

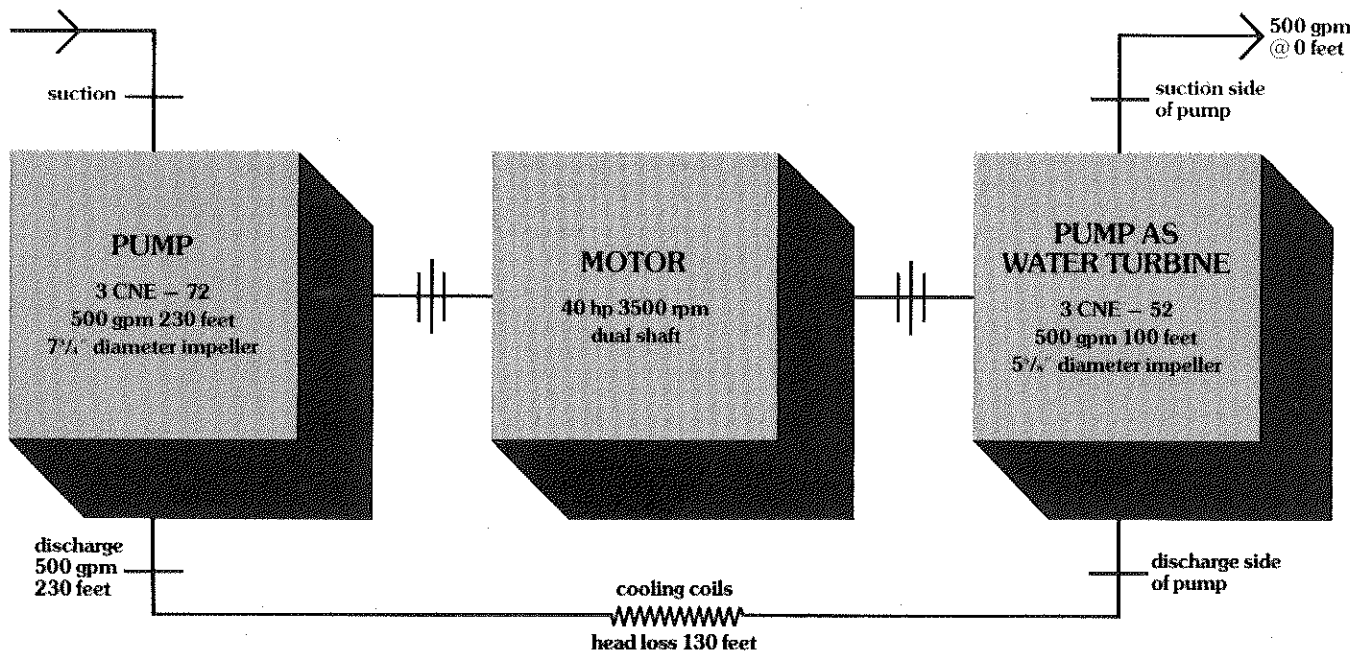
The obvious application is cooling-tower work. In systems where water is returned to a basement sump, a water-turbine driven booster pump can reduce horsepower required to return water to the roof by 20 to 40 percent.

Water or waste treatment facilities may find

the concept useful, too, for a variety of auxiliary services.

Besides saving energy, there are other good uses for a pump/turbine in which water would be pumped to a storage tank specifically to drive the turbine. For example, a water turbine might be desirable for safety reasons in an explosive atmosphere. It's also a good possibility where the need for reliability is greater than the power system can offer; in remote locations where it may not be economical to install a 3-phase electric system; in dusty or other contaminated areas where an electric motor would be short-lived; or where a high degree of speed adjustability is required, since turbine speed is infinitely variable, not tied to the speed of a 60-cycle motor.

This pump/turbine concept – essentially the same as full-scale "pumped-storage" hydro – is hardly new, but too often overlooked in times of plenty. Today's conservation-minded engineer may well find it a useful addition in the "plus" column of his plant's energy balance.



# Coping with pump progress: the sources and solutions of centrifugal pump pulsations, surges and vibrations.

By Austin R. Bush, Warren H. Fraser and Igor J. Karassik

Today's centrifugal pumps run faster, operate with lower values of NPSH and have higher efficiencies than their predecessors.

This is the good news. The bad news is that some of these pumps have exhibited severe hydraulic pulsations and surges, with impellers showing premature wear in their suction areas and sometimes near their discharge tips as well. In some extreme cases, catastrophic failures have occurred, including destruction of impellers, diffusers and volutes.

It is the nature of technological progress that the good news and the bad news are related. Technology in general tends to be cyclical; steaming ahead forcefully until difficulties arise, then pausing for retrenching, until the problem is finally resolved and the particular difficulties disappear. Then the limits – of pressure, temperature, speed, stress, what-have-you – are pushed ahead further, until new difficulties are met and a new problem solved.

This is our present situation with centrifugal pumps. This inquiry explores the relationship of progress to problem, and offers guidelines for coping with this contradictory aspect of pump progress.

A centrifugal pump is designed for just one capacity at a given head and speed, and the designer derives his selection of inlet and outlet impeller areas from the energy output of the pump (total head and capacity) for which peak efficiency is desired.

At other flows, because the geometry of the pump is no longer ideal, turbulence

results in reduced efficiencies. What has not been well understood until recently is the way this wasted energy creates various undesirable effects, both in the pump itself and, in some cases, in the piping system.

To some degree, undesirable effects such as noise, vibration and pressure pulsations have always been present in centrifugal pumps. In fact, they were proportionately worse in many earlier designs because of an ignorance of optimum design parameters. They were not, however, particularly destructive because these early pumps handled relatively lower energy levels (total head) per stage.

The introduction of higher speed, or more correctly of pumps with higher head per stage, led to the discovery that these effects were sometimes associated with unacceptably high noise levels and even with destruction of pump components such as the impeller and collector. Corrective efforts were successful, and pump operation continued peacefully until pump design flows increased further – to several times their previous levels. Then there came a second rude awakening to the need for further design changes – at a time when each pump had become even more essential, because it was associated with larger sized processes than ever before.

## Unwelcome difficulties arise

As design flows increase, symptoms begin to appear in the form of noise, vibration, and pressure pulsations within the pump. They also appear elsewhere in the system as noise and intense piping vibration. But regardless where the symptoms appear, their source is the pump and their solution must be within the pump as well.

While problems are often most acute in multi-stage boiler-feed pumps for fossil-fueled steam-electric plants, and single-stage nuclear-reactor feed pumps, equally unwelcome difficulties may be encountered with pumps applied to less severe service.

## Three signs of trouble

Various hydraulic phenomena are associ-

ated with centrifugal pump problems: pulses resulting from (1) interaction of impeller vanes and collector (2) suction recirculation within the impeller (3) discharge recirculation within the impeller. These problems and their remedies are the subject of this article.

Very similar phenomena may occur which are not traceable to the pump. Breakdown of very high pressures, as in a recirculation bypass orifice, may lead to noise and vibration in the system: a "pipe-organ" effect in the piping, or flashing due to insufficient backpressure downstream of the orifice. Pulsations and surges may also occur when air is trapped in the pump or system because of inadequate venting. In these cases the symptoms may be very similar, but the problem and its remedy lie in the system or the method of operation, not in the design of the pump.

## Pressure pulsations caused by vane passing frequency

Designers and users of centrifugal pumps now realize that unsteady flow, caused by the wake of vortices shed from the vanes, persists for a considerable distance and sets up an interaction between moving and stationary parts of a pump. In other words, hydraulic shock occurs when impeller vane tips pass volute tongues or diffuser vanes, with the magnitude of shock (and resulting pressure fluctuations) increasing at higher impeller tip velocities and as pump size increases. While the probability of phase coincidence can be predicted with reasonable accuracy, its absolute magnitude and resulting stress levels can only be approximated.

Since these pulsations were affected by impeller tip velocities, various pump manufacturers became acquainted with this problem at different times, depending on the moment in history when they began to produce high-head-per-stage pumps.

Worthington's first high-speed boiler-feed pump was built in 1954. Designed for utility service, the pump was to run at a maximum speed of 9000 rpm, developing 1635 feet per stage, at about 350 feet per



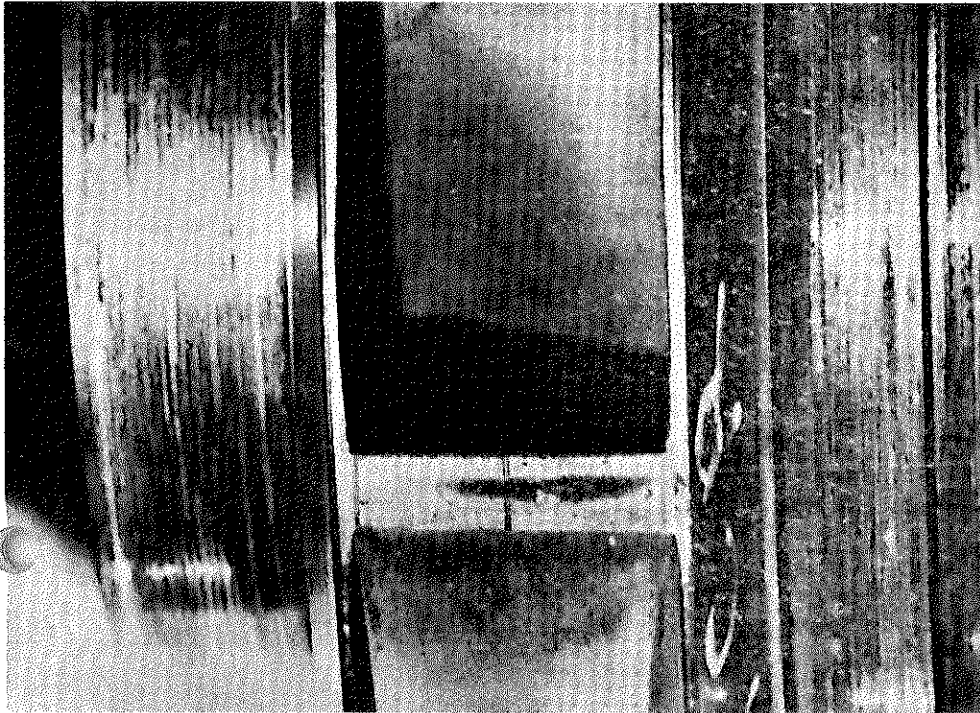


Figure 1 – Erosion of first stage impeller vane tips when running with  $1/32$ " clearance

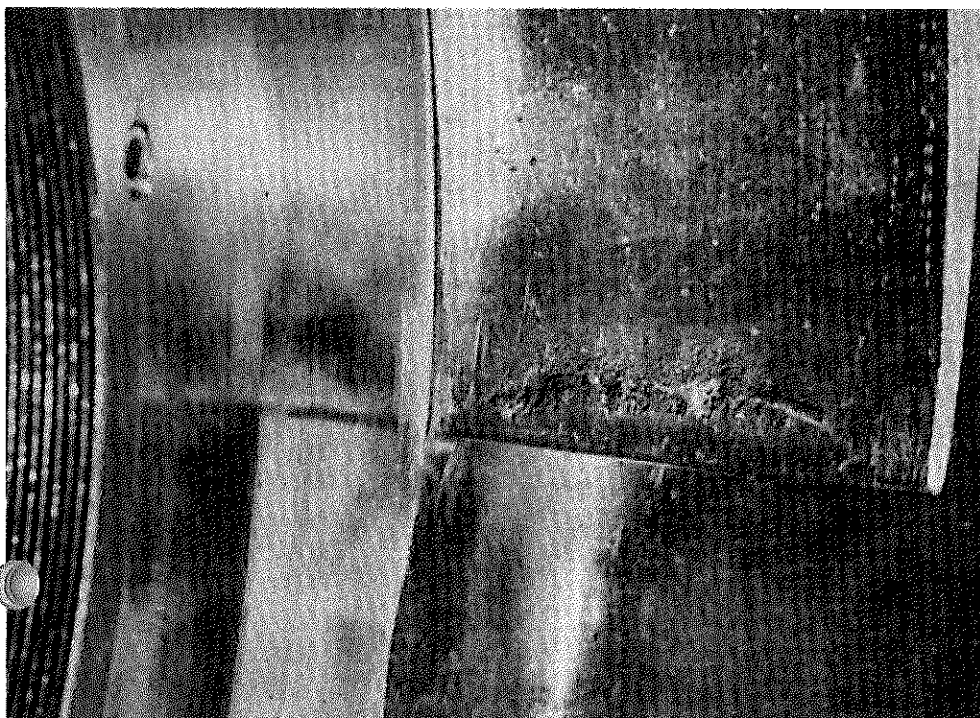


Figure 2 – Erosion of first stage twin volute tongue when running with  $1/32$ " clearance

second peripheral velocity. It turned out a little strong on test and the impellers were cut down, thus increasing vane tip clearance. This pump never developed pulsation or noise problems.

Our second set of high-speed pumps was shipped with what was then standard minimum clearance between impeller vane tips and twin-volute tongues. Early in 1957, before the plant went on the line, while the boiler pressure was being raised and the pumps were still operating at a slightly reduced speed, it became evident that they were developing excessive noise. Sound level readings taken 4.5 feet from a pump running at about 7100 rpm (85% of maximum) showed an over-all 101 decibels with very marked peaks at 1, 2, 3, 4 and 5 times vane frequency.

Figures 1 and 2 show the first-stage impeller and twin volute of one of these pumps after a rather short period of operation. At a point where the pressure should be 500 to 600 psi above vapor pressure, marked cavitation erosion indicates that the shock pressure of the impeller vane tip passing close to the twin volute tongues was momentarily reducing local pressure to less than vapor pressure. This cavitation erosion occurred only at the first stage. Apparently, pressures reached at the higher stages prevented the shock pressure from reducing the local pressures to the value of the vapor pressure.

Modifying the relative configuration of impeller and twin-volute vanes and increasing the radial clearance between them reduced the over-all noise level by 7 db with the pump running at 7900 rpm. The changes brought the noise down to an acceptable level even when the pumps operated at full load and about 8300 rpm.

#### Treating the symptoms

It became apparent in the years following these occurrences that a number of companies were encountering difficulties with their pumps from pressure pulsations and surges that could be definitely traced to vane passing phenomena. Some pump designers began to recommend that reso-

nant frequencies of the piping system be made to avoid correspondence with vane passing frequencies of the pump. This solution requires a rigorous analysis of the piping system, and modifications if it shows resonant frequencies in the ranges to be avoided.

It seems to us that such recommendations are unrealistic. To begin with, a mathematical model of the piping system requires numerous assumptions that can only be guesses. Control of the exact value of piping resonance is difficult, the solution complicated and costly, and error almost inevitable. Furthermore, since a pump might have to operate over a range of speeds, there is a corresponding range of resonant frequencies to avoid, making the problem still more complex.

#### **Attacking the problem at its source**

The point is, eliminating harmonic frequencies from the piping only removes symptoms. It does nothing to reduce the intensity of impulses created within the pump! The only true solution of the problem is to design pump hydraulics to reduce intensity of the vane passing phenomenon.

The ideal collector, of course, would have no vanes, since every time an impeller exit vane passes a stationary volute or diffuser vane a pressure pulse is generated. Since it is not physically practical to build a vaneless collector, interaction between impeller and collector vanes must be reduced by increasing their separation, or modifying the flow pattern, or both. The latter is influenced by number, orientation, and contour of the collecting vanes.

For many years, designers avoided using a number of impeller vanes divisible by the number of volute or diffuser vanes. The idea was to avoid multiplying the intensity of the pressure pulses generated by the passage of each individual impeller vane past the stationary collector vanes.

Other theoretical approaches were also suggested, leading the designer to esoteric combinations such as 5, 7 or 9 vanes for the impeller and as many as 11 or 13 vanes

for the collector. Implicit in all such juggling was the idea that the use of two vanes for the collector was certain to lead to trouble.

#### **Flying in the face of "theory"**

And yet, ever since 1937, all high-pressure boiler-feed pumps built by Worthington have used twin volutes. Since hundreds of such pumps have operated satisfactorily over an extended period beginning 38 years ago, it would certainly appear that the relative number of impeller and collector vanes must be less important than some other factors.

One well-known method for reducing the pressure pulse is to increase the gap between the impeller and the stationary collector. This tends to decrease the intensity and abruptness of the pulse, smoothing it out so it appears as a ripple rather than a wave, but it does not eliminate it completely.

Worthington arrived at the ultimate solution following a rather severe failure of twin-volute diffusers at the Bull Run plant of TVA in the mid-60's. Since the energy levels of the pulses were inordinately high, as shown by their destructive effect and the high observed noise level, it appeared essential to reduce them at their source rather than treat the symptoms by strengthening the elements which had failed. Our solution was a judicious selection of the gap between rotating and stationary components, and a careful choice of the relative configuration of the two.

The results were absolutely dramatic! The sound level, which is proportional to the intensity of the pulses, was reduced from 108 to 90 db. Since sound is measured on a logarithmic scale, this represents a ratio of 63:1, or a 98.4 per cent reduction! This not only reduced energy and noise levels to safe values, but also enabled us to follow the preferred practice of balancing radial forces by purposely using an even number of impeller vanes with twin volutes – a practice one would not dare use without effectively reducing the energy of the pressure pulses.

We now have nearly nine years' field ex-

perience following these design practices without a single failure caused by pulses induced by vane passing frequency, with one less serious exception caused by a shop machining error. After the shop error was corrected, the noise level was again dramatically reduced and no further problems have occurred.

#### **Next issue: Part II, Treating internal recirculation.**

At certain flows, centrifugal pumps are subject to internal recirculation in the suction and discharge areas of the impeller – with a resulting increase in pressure pulsations.

Internal recirculation is a fairly mysterious phenomenon. Only recently have pump designers become aware of it and analyzed its effect. Only more recently have they learned to predict and control it.

Better understanding among pump users is important so that they can appreciate the designer's dilemma and avoid specifying certain operating conditions. This is the subject of the second half of *Coping With Pump Progress* – in the next issue.

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