pumpworld

*pump*world

In this issue

Did you recognize the design on our cover this month? Do you recognize the pattern shown opposite? If so, you win applause for your pump expertise, because all too few people involved in fluid handling are intimately familiar with today's rotary screw pump. The screw patterns on the cover and the casing section opposite will become better known as more and more users realize the unique merits of screw pumps in high-viscosity, high-flow, low-npsh applications—as the following pages point out.

The theme of less familiar pumps continues with a look at another breed of pump. The concrete pump, adapted to sub-zero operation in the USSR, is equally applicable to other parts of the world. If some of today's pumps seem strange, the centrifugal pump of tomorrow may be more so with changes in hydraulics, structure, materials, drivers, and applications. In this issue I. J. Karassik, the engineer responsible for many past changes in pump design and application, looks in his crystal ball and speculates about tomorrow's pumps. Finally, in a more familiar vein, our Pump Refresher installment reviews how fluid flow analysis can be used to solve common pumping problems.

A new look at rotary screw pumps for high-viscosity, high-flow, low-NPSH applications.

By Ram Javia

The rotary screw pump offers efficient handling of viscous fluids over a broad range of demanding conditions of service. These typical pumping problems can all be solved with a screw pump.
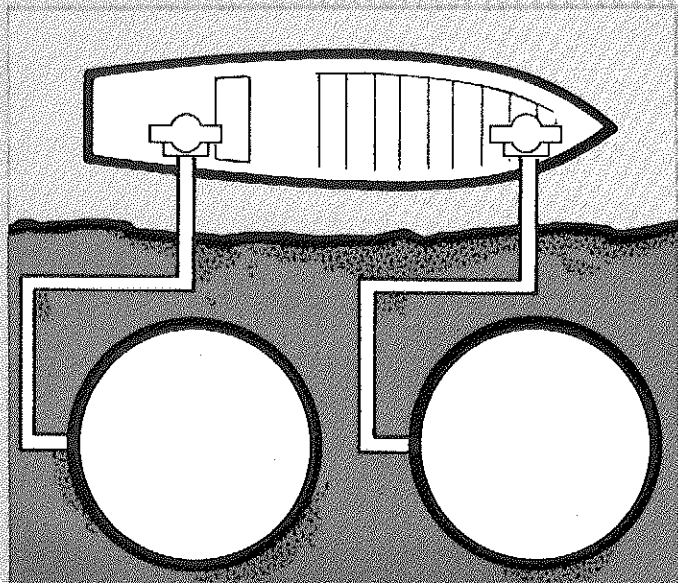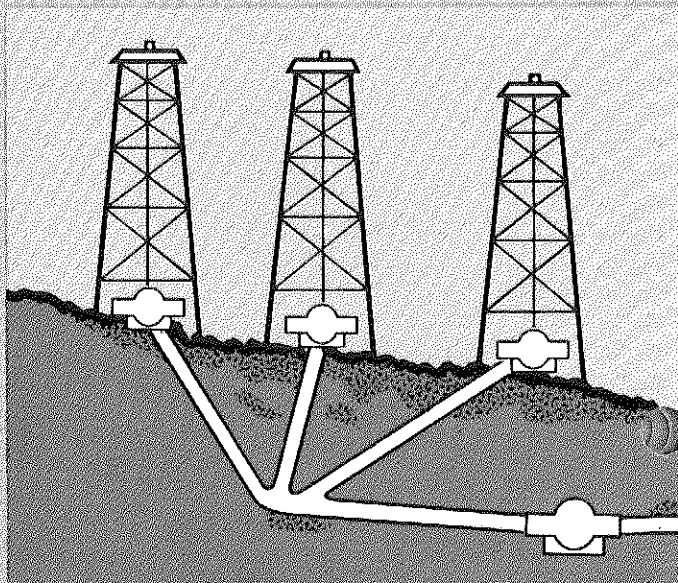
In barge unloading service, both high and low viscosity can be moved using just one pump design.

In pipeline pumping, a variety of crudes with high viscosities, high differential pressures and low npsha are handled.
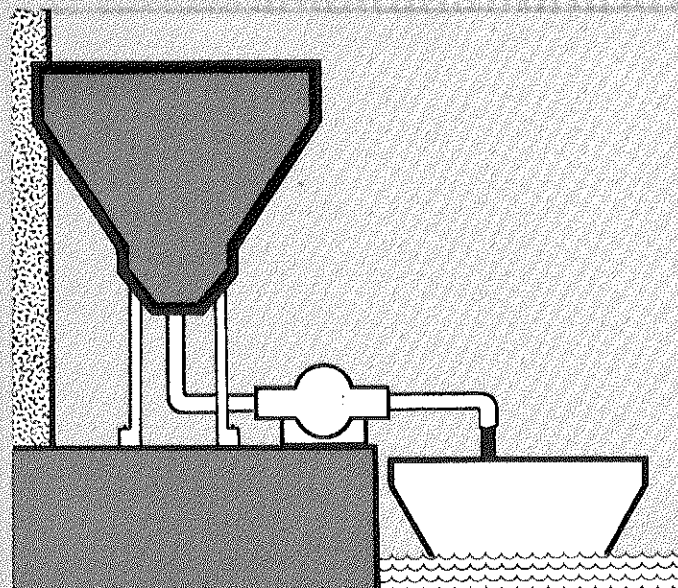
The extreme viscosities encountered in asphalt transport service are readily handled, with steam jacketing a possibility if required.
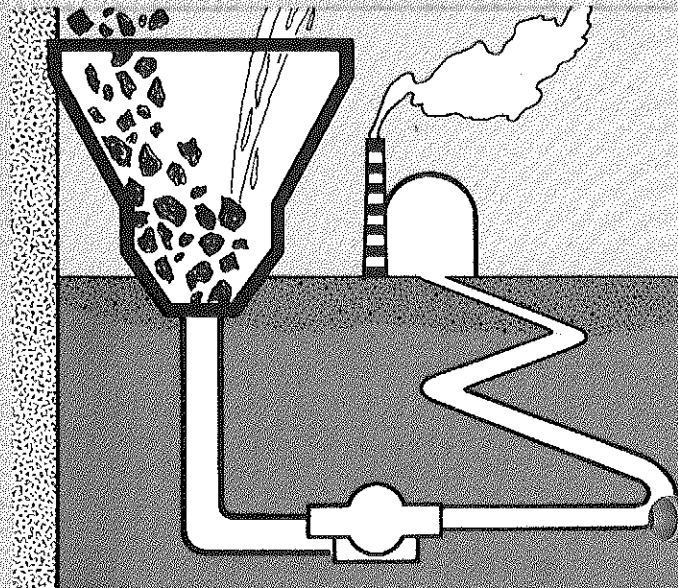
High temperature liquefied coal slurry with a high percentage of solids can be moved at low suction pressure.

Smooth flow of the screw pump suits it for extrusion and transfer of high viscosity polymers with less chance of damaging the product.
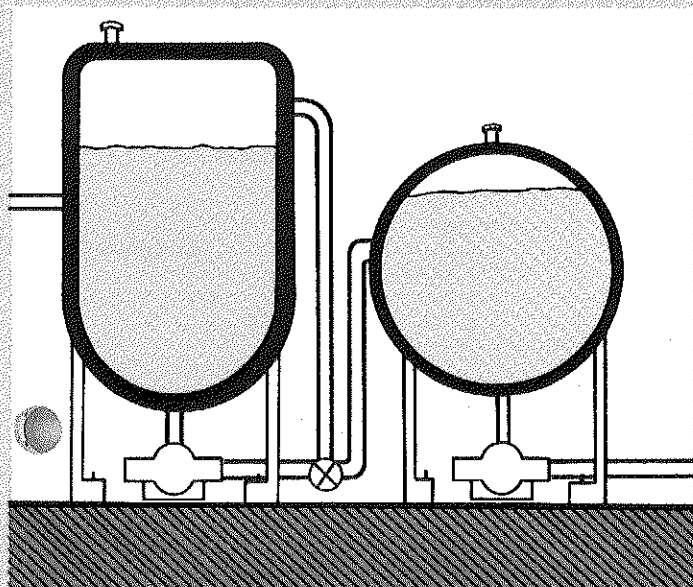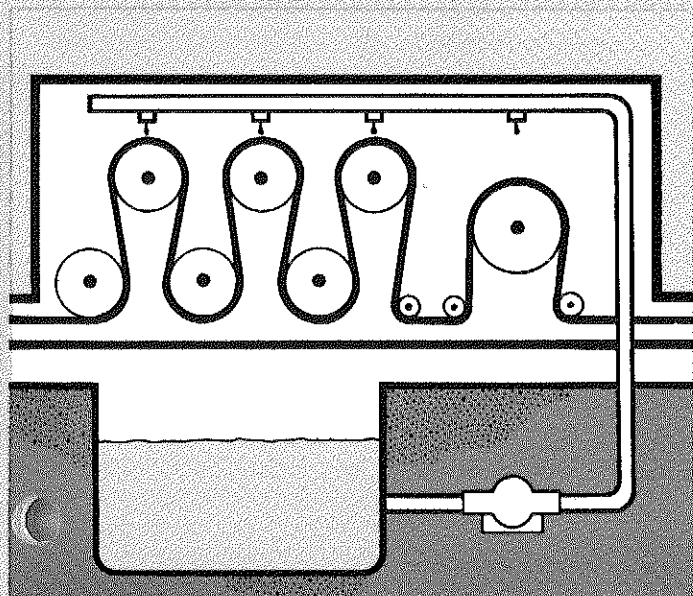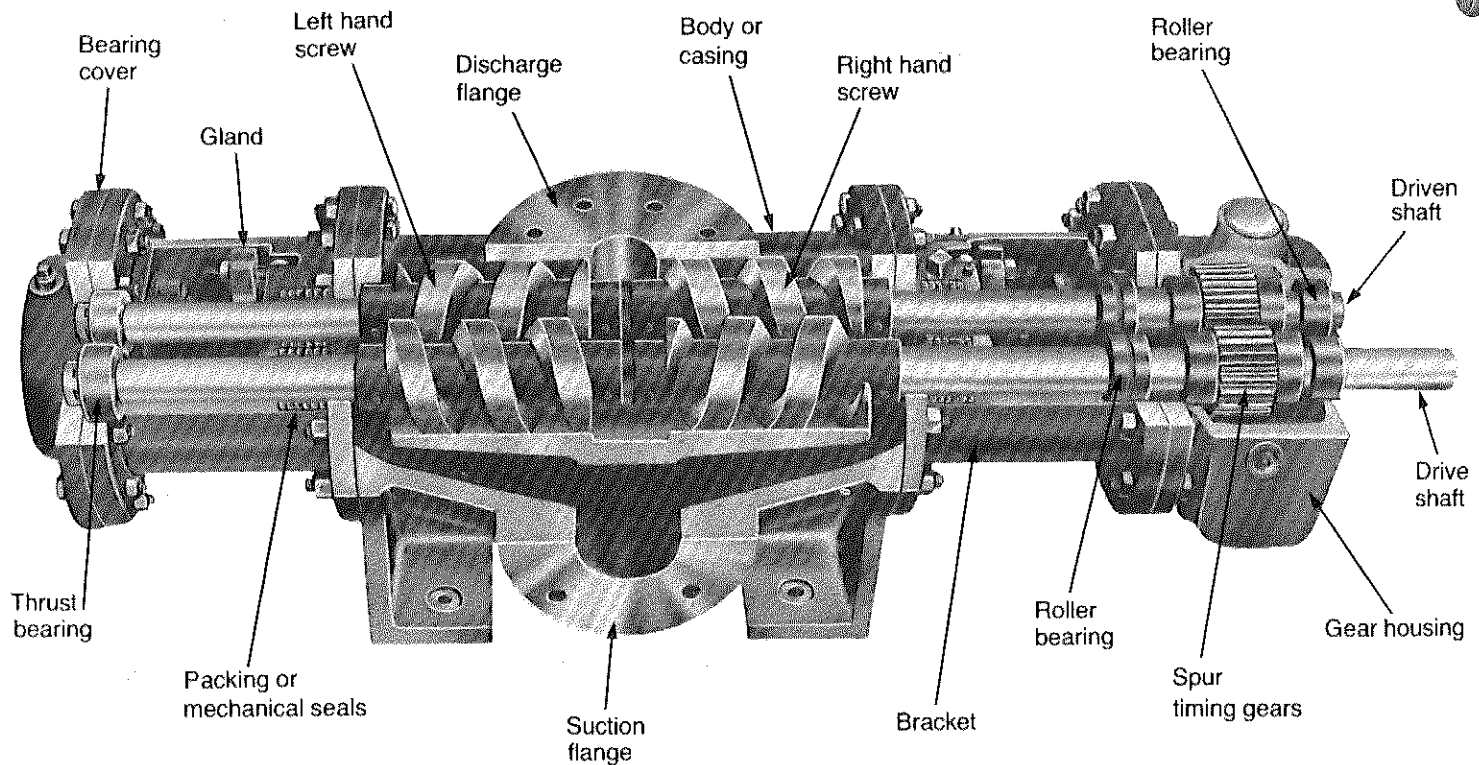

In addition to smooth, pulseless flow, the screw pump provides maximum reliability for critical lube oil services.

The newest member of the Worthington Pump product family is the Sier-Bath rotary twin screw. The new line handles viscosities as high as 200,000,000 ssu, flows to 10,000 gpm, pressures to 3000 psi and temperatures to 800°F. The working principle of the screw pump is, of course, that of Archimedes' screw, invented by the Greek mathematician about 200 BC and still in use today in many parts of the world. Although it has been around for about 70 years, the modern screw pump came of age more recently, with the advent of sophisticated, tape-controlled machine capabilities. With high efficiencies made possible by computer-controlled, precision-cut screws and gears, today's screw pump handles a wide variety of fluids. It is often the pump of choice when conditions of service include high suction lift or high pressure over a broad range of viscosities, flow rates and temperatures. Screw pump applications include fuel, lube and crude oil service; navy and marine cargo; oil burners; slurry handling; and a variety of high viscosity materials such as polymers, copolymers and elastomers, cellulosics, syrups, fats and greases, soaps and solvents.

A basic understanding of the working principles of the twin screw pump, as well as an

Ram Javia is a chief engineer for Worthington (Canada) Ltd.

Bearing cover

Gland

Thrust bearing

Packing or mechanical seals

Left hand screw

Discharge flange

Body or casing

Right hand screw

Roller bearing

Driven shaft

Drive shaft

Gear housing

Spur timing gears

Bracket

Suction flange

Roller bearing

overview of its wide range of applications, is helpful in evaluating this candidate for your high flow, viscous applications. This article provides that overview. In future articles we will delve deeper into the specifics of screw pump selection and applications.

The screw pump is a rotary pump, with the inherent characteristics of positive displacement, pulse-free flow, and self-priming capabilities. It improves on traditional rotary pump features by giving smoother, more constant flow with varying viscosities and pressures.

Both single-rotor and multiple-rotor screw pumps are on the market. In either design, the pumping action is accomplished by progressing cavities which advance along the rotating screw from inlet to outlet. This axial flow pattern minimizes vibration, resulting in smooth flow—an excellent feature where liquid agitation or shear may be a problem.

In the twin-screw pump, two sets of screws rotate and mesh in an accurately bored casing, with precise operating clearances maintained between them. The mechanical displacement of fluid from inlet to outlet is produced by

trapping a slug of fluid in the helical cavity—called a "positive lock"—created by the meshing of the screws (Figure 1).

Actually, the term "twin screw" isn't quite correct. To balance the axial thrust exerted by the displacement of fluid, each shaft usually mounts not one, but a pair of pumping screws which split the intake so the fluid flows in opposite directions. Fluid may enter at the ends of the screws and flow inward in both directions, or enter at the center and flow outward. Either arrangement provides axial hydraulic balance. Precision clearances between screws themselves and between screws and casing substantially limit internal leakage or "slip," ensuring positive displacement and high efficiencies.

In most twin screw designs, the clearance between screws is maintained by a pair of timing gears mounted on the shafts. These gears also transmit power from the drive shaft to the driven shaft.

The body.
The body is a casing with two precision-machined bores to house the rotating screws. Fluid passes from the inlet chamber into the pumping chamber and then to the discharge chamber. Screw pump bodies are commonly made of cast iron, ductile iron, cast steel or 316

stainless, depending on such factors as pressure requirement, need for corrosion or galling resistance, pumping temperatur etc. Body bores are sometimes chrome plated to improve surface finish, anti-galling characteristics and hardness. Various body designs are adapted to a wide range of pumping applications, as shown on pages 7 and 8.

The screws.
The twin pumping screws are precision-machined to very closely controlled profiles to assure low internal leakage, or slip. Internal dimensions on both screws are selected for lowest practical internal clearances, maintaining very high efficiencies.

The twin screws must be designed to withstand hydraulic pressure within the screw channels, which exerts both a radial and an axial force on the screws. Axial force, or thrust, is balanced as we have seen, by using the matched pair of screws on each shaft. The radial force, which tends to cause shaft deflection, is resisted by the fluid film trapped betw the screws and by the body bore diameter.

Screws and shafts may be of "pinned" ("integral" design. In the pinned screw design, the screws are machined as separate pieces and mounted on the

Typical twin screw pump bodies and flow patterns.

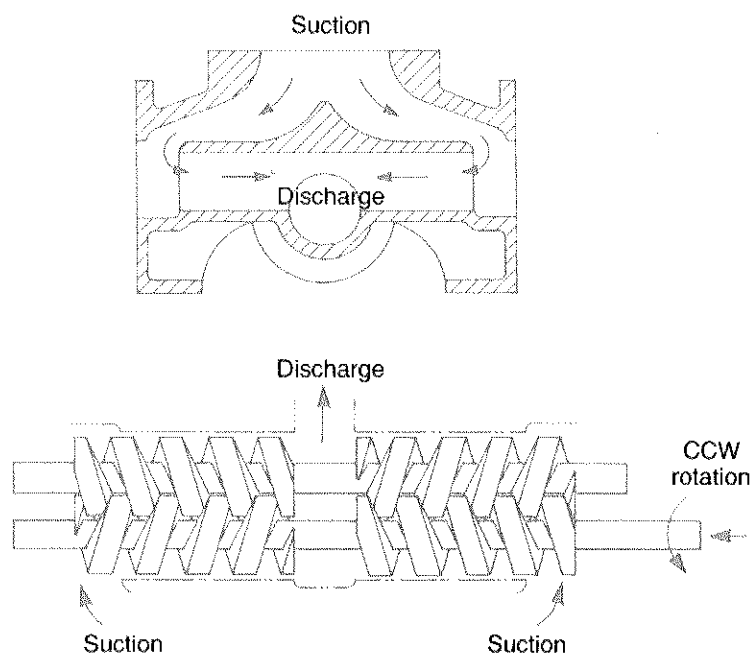The design of twin screw pump bodies and the arrangement of screw assemblies are interrelated, depending on system parameters and characteristics of the fluid to be handled. This box shows several typical body designs, and the screw configurations most commonly associated with them, although other combinations are possible.

Standard end-flow design

Discharge

Suction

Discharge

Suction   Left hand screw   Right hand screw   Suction   CCW rotation
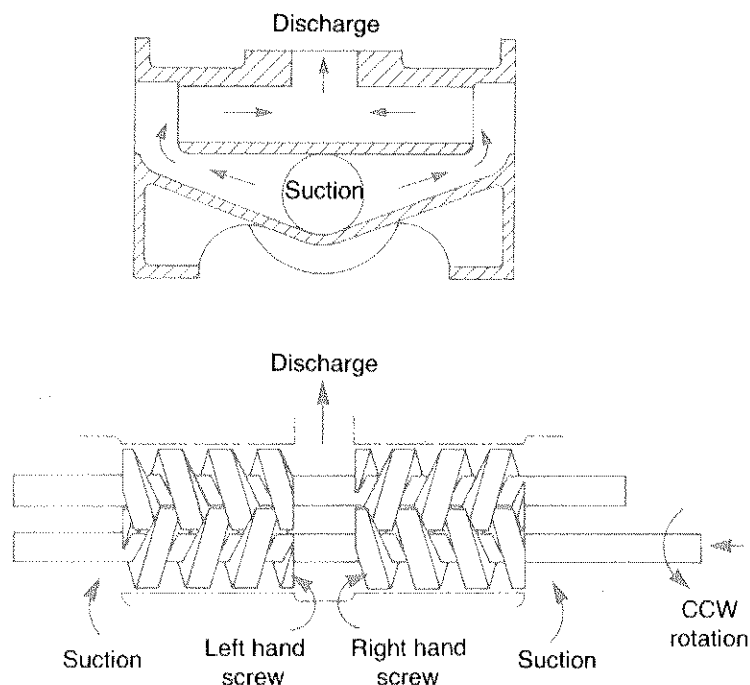
Hopper/standard end-flow design with extended screw

Suction

Discharge

Discharge

Suction   Suction   CCW rotation

Capabilities: npsha above 15 feet, moderate viscosities.

Typical fluids: fuel oils, lube oils, crude oils, low-viscosity asphalt.

In this common screw pump design, the pump body is designed for side suction, top discharge. Fluid fills a bottom sump, enters both ends of the screw and is moved to the center and discharged through the top. Stuffing boxes are under suction pressure only.

Capabilities: Low npsha, below 20 feet, at low viscosities, 1000 to 10,000 ssu. Moderate to high viscosities, above 100,000 ssu, with npsha of 20 feet or more.

Typical fluids: High-viscosity polymers, coal slurries, fuel oils.

A multi-purpose design with top suction and side discharge. Fluid enters the pump body from the top to fill an enlarged suction passage, or hopper. Fluid enters the screws at each end, is moved to the center, and is discharged at the side. Stuffing boxes are under suction pressure only. The longer pumping screws extend into the suction cavity, promoting smooth flow and improving the npsh capabilities and extending the viscosity range.

Hopper/reverse-flow design with extended screw

Jacketed Design

CW rotation

Pitched Design

CCW rotation
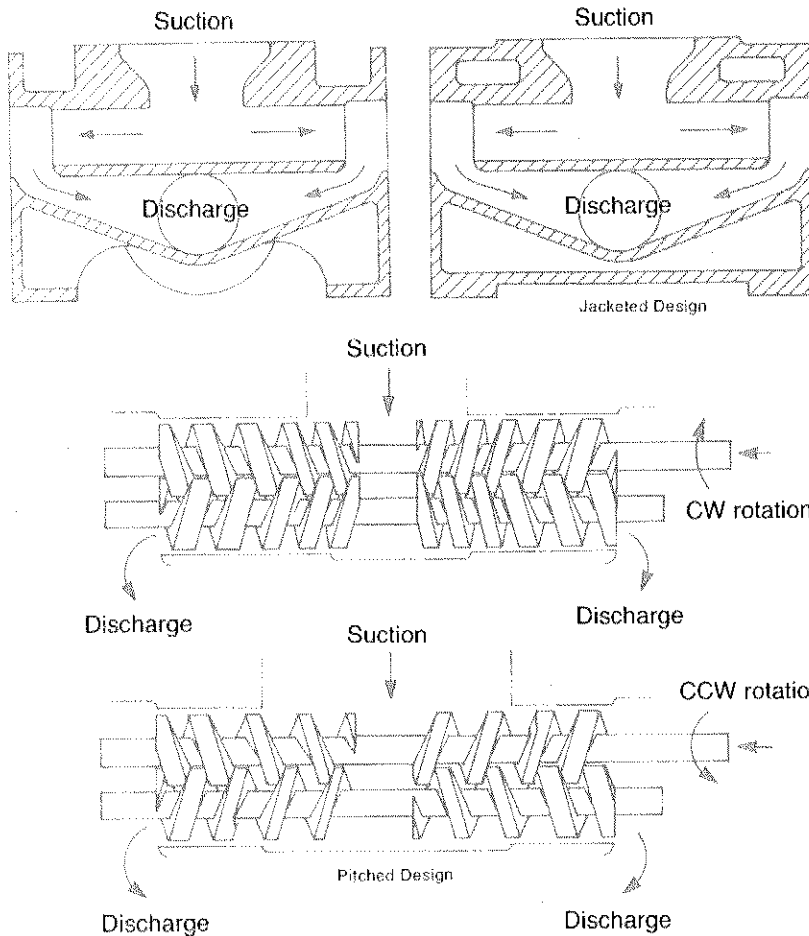
Capabilities: With regular screw: medium to high viscosities, above 100,000 ssu; low npsha, under 15 feet. With special screw: high viscosities, above 500,000 ssu; very low npsha, under 10 feet.

Typical fluids: Coal and other high-viscosity slurries, greases, rubber cement, polymers.

This design, with top suction, side discharge and reverse flow, is suitable for applications with low npsha and moderate to high viscosities. Because of the reverse flow pattern (fluid enters the center of the screw assembly and leaves at each end), stuffing boxes are under discharge pressure. This prevents air from entering the vacuum system. The jacketed design, shown at right, provides for heat-tracing process and other fluids as they pass through the pump – a wise precaution with many high-viscosity fluids. The specially-designed extended screw has a pitch designed for lower npsha and higher viscosities.

Single-entry, standard end-flow design

Discharge

Suction

Discharge

CCW rotation
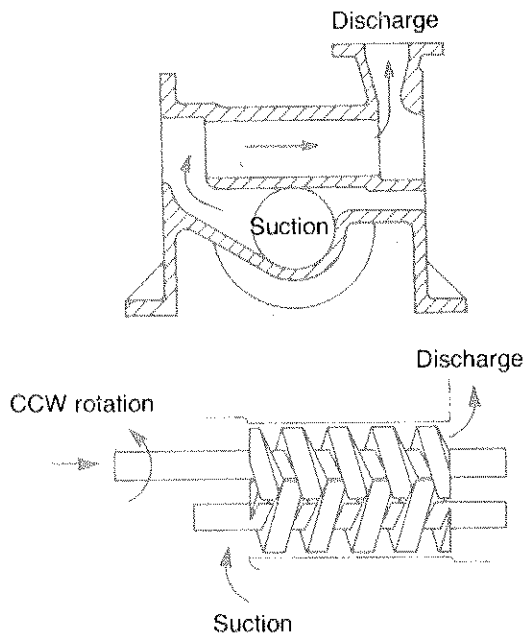
Suction

Capabilities: npsha above 20 feet; low viscosities, 32 to 20,000 ssu.

Typical fluids: Lube oils, fuel oils, crude oils, low-viscosity asphalts.

The side-suction, top-discharge design is suitable for moderate conditions of service. Fluid enters the screw from one end only, and exits from the other. This arrangement is not inherently balanced, as the double-flow screws are, and so an external balancing device is required.

shaft with pins or keys. This design is used for relatively low pressures, up to about 500 psi. The integral screw arrangement, with screws and shaft machined from a single forging or piece of barstock, provides much higher pressure, viscosity, and shaft torque capabilities.

Screws may be made of cast iron, ductile iron, or various steels and stainless steels. To provide better surface finish and anti-galling characteristics, the outside diameters may be Colmonoy or Stellite tipped, or they may be bronze coated to provide a yielding surface. Sometimes designs include replaceable screw tips.

helical or spur designs are also used, but they maintain only the angular relationship, relying on the thrust bearing to maintain the proper axial relationships between pumping screws.

Some screw pump designs work without timing gears. In these designs there is a drive screw which is used to drive the driven screws.

Timing gears are designed to American Gear Manufacturers Association (AGMA) standards and recommended quality levels; gear teeth are hardened to increase surface durability.

When the fluids to be pumped have lubricating characteristics, the timing gears may be mounted inside the pumping chamber and lubricated by the pumped fluid. This is an "internal" design. When fluids are non-lubricating, corrosive, abrasive, or at temperatures above 300 °F, the timing gears must be mounted outside the pumping chamber and independently lubricated. This is the "external" design.
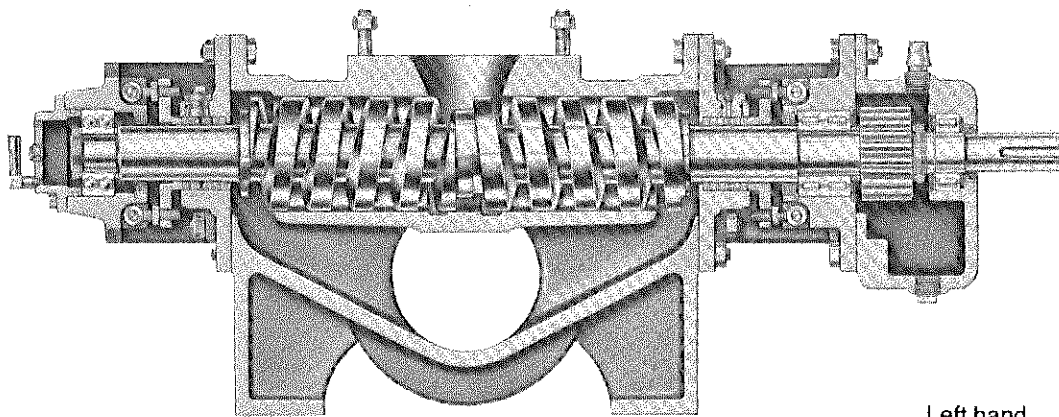

Twin-screw assembly combines right-hand and left-hand screw threads on each shaft, creating a counterflow which balances axial thrust.

The timing gears.
Timing gears are used to transmit power from one rotor shaft to the other and to maintain the proper clearance between the pumping screws. For optimum efficiency, most screw pumps are designed with relatively small internal clearances between the pumping screws. This mandates high-precision timing gears if metal-to-metal contact is to be avoided. The positioning of timing gears relative to the pumping screws is called "the timing of the screws."

Timing gears may be double helical (herringbone) type, serving to maintain the proper angular and axial relationship between the pumping screws. Single
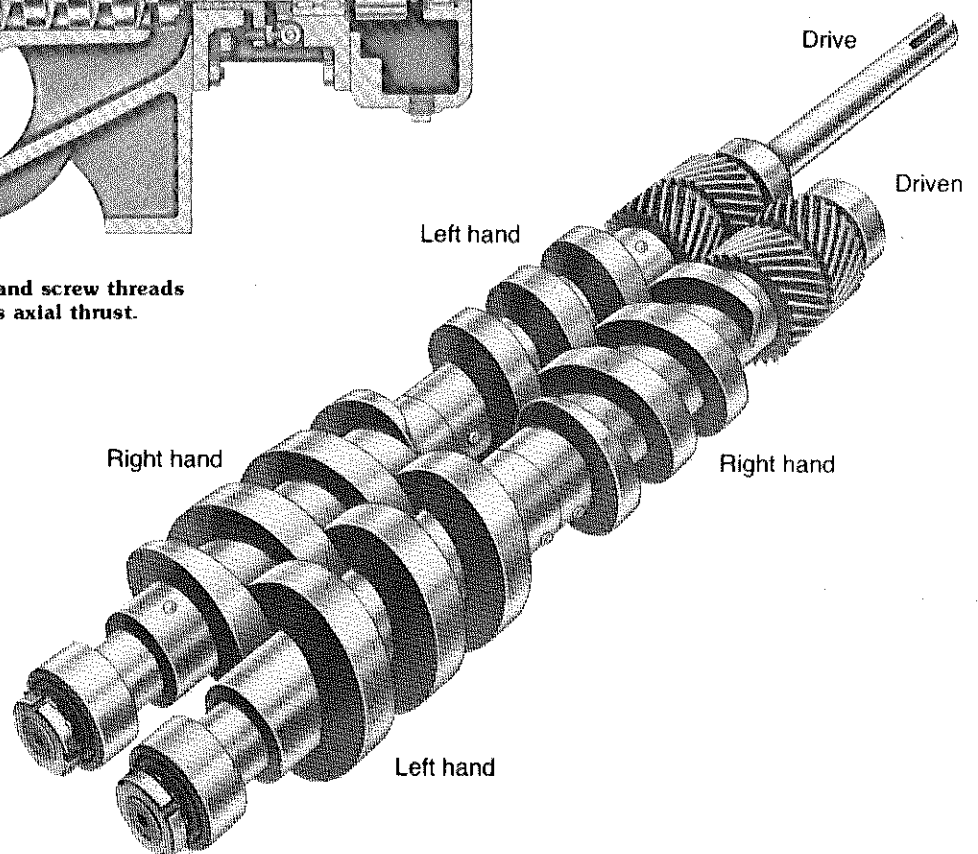
Arctic conquest: New sub-zero concrete pump is helping to pave the way.

By R. Romani

The Arctic circle stretches 4,000 miles from Finland to the Bering Strait across the roof of the Soviet Union and is already dotted with hundreds of Russian cities. Yet it is the policy of the Soviet Union to push its frontiers even farther north. In its drive to unlock vast new reservoirs of resources, the USSR is making use of the latest technologies, including a unique Worthington approach to pumping concrete at sub-zero temperatures.

Man is a born explorer. Yet, with so much modern day exploration devoted to outer space, large areas of our own globe remain a mystery. Vast reaches of our planet, although discovered and explored, remain hostile to settlement because of severe climatic conditions. One such region is the frigid Arctic wasteland that stretches more than 4,000 miles across the roof of the Soviet Union. The conquest of this land, the largest continuous Arctic territorial expanse claimed by any nation, is a continuing objective of the USSR.

Much of the success of this endeavor depends upon developing and applying new technologies—and Worthington Pump is playing a unique part in this area. Through agreements reached between Worthington S.p.A., Milan, and several ministries of the Soviet government, the plant and systems division of the Italian-based company has developed a new concrete pump capable of operating at sub-zero Arctic temperatures, permitting the casting of concrete under the most adverse conditions.

The unique project began more than two years ago when the Soviet Ministries of

Energy and Civil and Industrial Building announced extensive development programs for the country's northern regions, requiring construction of roads, power stations, homes, schools, and hospitals. Worthington Italy was selected to develop and manufacture a prototype concrete pump that could overcome problems associated with casting concrete at outside temperatures of −40° Celsius.

The first prototype pumps entered service during the winter of 1976 in the Kola peninsula of western Russia, north of the Arctic circle. Three months later, Soviet project engineers placed additional orders for 10 truck-mounted concrete pumps for use at several nuclear power stations and other construction sites. The units incorporate a unique controlled-heat system adapted to a standard rotary distribution pump and delivery system.

Two major problems confronted design engineers. First, with conventional distribution procedures, concrete would freeze in minutes under Arctic conditions. Second, castings made directly by a truck mixer or boom, or both, would need to be coupled with jets of steam to allow the mixture to "cure"—or complete its chemical and thermal reaction over a prescribed time period.

Working to stringent specifications, Worthington S.p.A. began its investigation of the heat requirements for a total delivery system—from pump to boom end. The most logical starting point seemed to be modification of the multi-cylinder pumping unit with the distribution box driven by an independent engine, drawing concrete from the loading hopper and transferring it to the distribution pipe. It was also decided that the performance goal would be in the range of 65 cubic yards per hour, 260 feet of vertical and 1300 feet of horizontal discharge distance. The system would employ the standard rotary distribution pump whether mounted on truck, trailer or skid with the advantage that it contained fewer moving parts (such as sliding gates, butterfly valves, Siamese or Y connections) found in other systems. The standard Worthington rotary pump
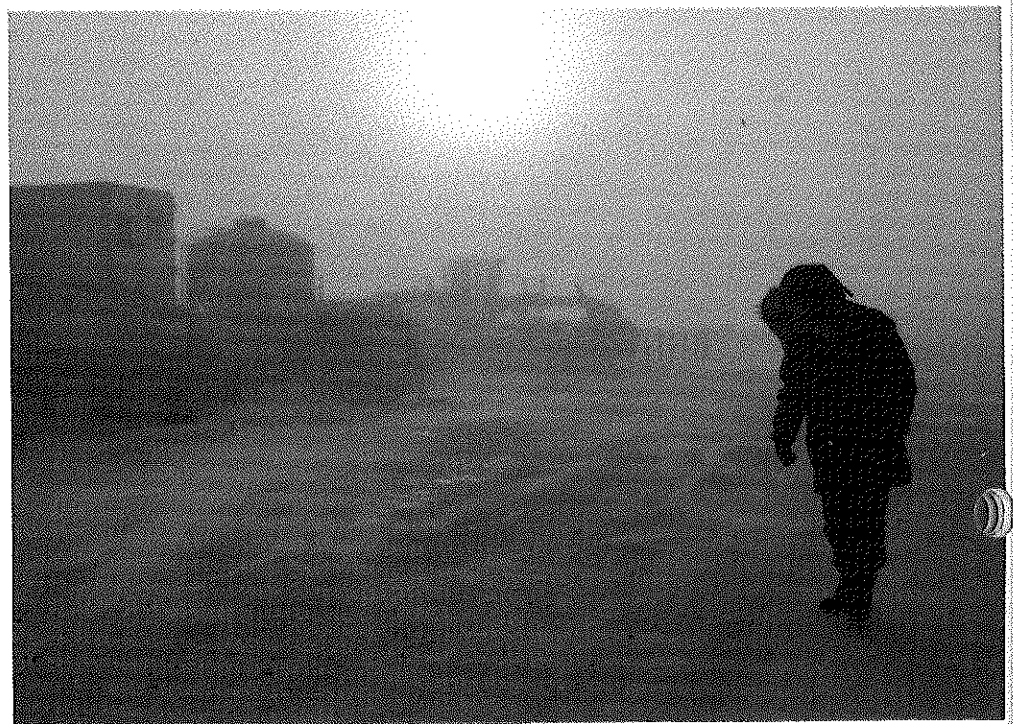
R. Romani is manager of public relations and advertising for
Worthington S.p.A., Milan

Worthington Pump's specially designed concrete pump is at home in the sub-zero temperatures north of the Arctic Circle.

utilizes no valve. Concrete flows freely from the cylinders to the pipeline by passing through a slight S-curve of large radius—the "rotary." Since the ability to pump concrete is determined by its composition, this simple, linear path enables the Worthington design to handle difficult mixes.

In developing the heating system, Worthington S.p.A. engineers focused their efforts on achieving operating temperatures of between 5 and 30

degrees Celsius, the range in which concrete can best be cast and set without problems.

Engineers also had to deal with the need to preheat all other mechanical units in addition to the pump and related piping to avoid thermal stresses. The heat generated by the diesel engines which drive the pump and truck was utilized for this purpose. Calculations by industrial refrigeration specialists led to the conclusion that a proper heat balance

could be achieved through a closed-circuit hot air system that would encompass the hopper, pump and all accessories.

Heat dissipated by the cooling air of the pump drive diesel engine maintains the temperature in the hopper, the pumping cylinders and the rotary distribution system. On leaving the engine, hot air is channeled toward the loading hopper, the distribution system controls, the exit coupling from the pumping unit and,

POSITION 1

ROTARY

cylinder A

cylinder B

POSITION 2

cylinder A

cylinder B
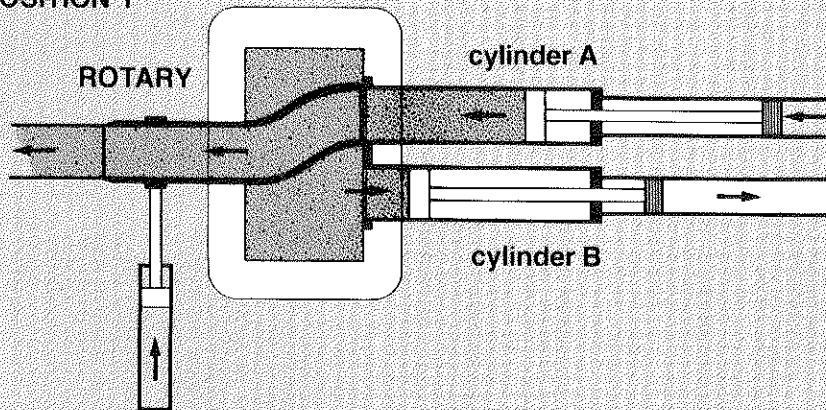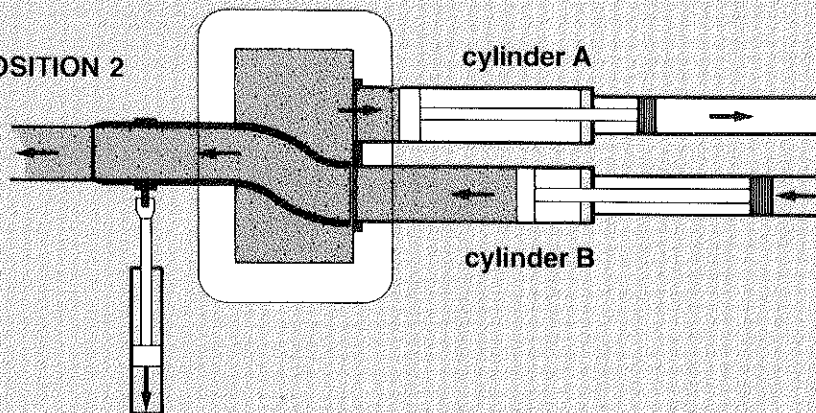
A simple, effective concrete distribution device allows the Worthington concrete pump to handle difficult mixtures with low-cement and high-sand-and-aggregate content. The large radius S-curve shape of the rotary distributor ensures a linear pumping path with no restrictions to cause clogging. No sliding gates, butterfly valves, Siamese or Y connections used. In Position 1, cylinder A is in the discharge phase, pumping concrete toward the rotary distributor. Cylinder B is filling with concrete. In Position 2, the rotary distributor swivels to a full cylinder B, providing the same linear flowpath for smooth discharge from that cylinder. Cylinder A is filling with concrete.

finally, hydraulic units housed in the bay adjoining the engine. The air returns to the suction of the diesel engine cooling system, mixed with fresh outside air. Special controls regulated from the truck cabin maintain the proper temperatures throughout the system. Temperatures ranging between 10 and 30 degrees Celsius, ideal for engine operation, are maintained by a double shutter which controls incoming outside air and the ex of hot air downstream of the engine which is channeled to the pump. A butterfly valve inside the duct directs hot air to other parts of the system. When the required temperatures are reached, the valve interrupts flow.

Separate heating and thermal protection systems were devised for the discharge pipes of the pump. When extended, the pipes have greater exposure to the biting cold and require added protection. In addition to special insulation, the pipes are protected by hot exhaust gases of the auxiliary engine piped in the opposite direction to the flow of concrete, thereby preheating the free end of the pipe at the start of the casting operation. With the combination of hot gases and insulation it is possible to deliver concrete at temperatures between 20 and 30 degrees Celsius, even under the most adverse conditions.

The last problem to be met was protection for the operator's cabin. This was achieved by a battery-powered electric heating system which, in addition to providing a comfortable atmosphere i the cabin, heats the wash water for the pump cylinders and the oil (in this case, an aeronautical grade) used for the main drive circuit of the pumping unit, the hydraulic drive of the auxiliaries, the concrete mixing screw and the boom control and stabilizer valves.

The final step involved the perfection of controls for achieving the desired temperature at various points in the pump and delivery system. This was accomplished through the installation of thermal devices. Temperatures are

Capable of operating at temperatures as low as −40° C, the unique Worthington concrete pump is the result of technological exchanges between Worthington S.p.A. Italy and the USSR.

monitored from the cab and regulated through digital instrumentation.

The total system, including pump and complete delivery mechanism, becomes operational in minutes, even at −40° Celsius—the exact requirement spelled out by Soviet engineers when they turned to Worthington S.p.A.
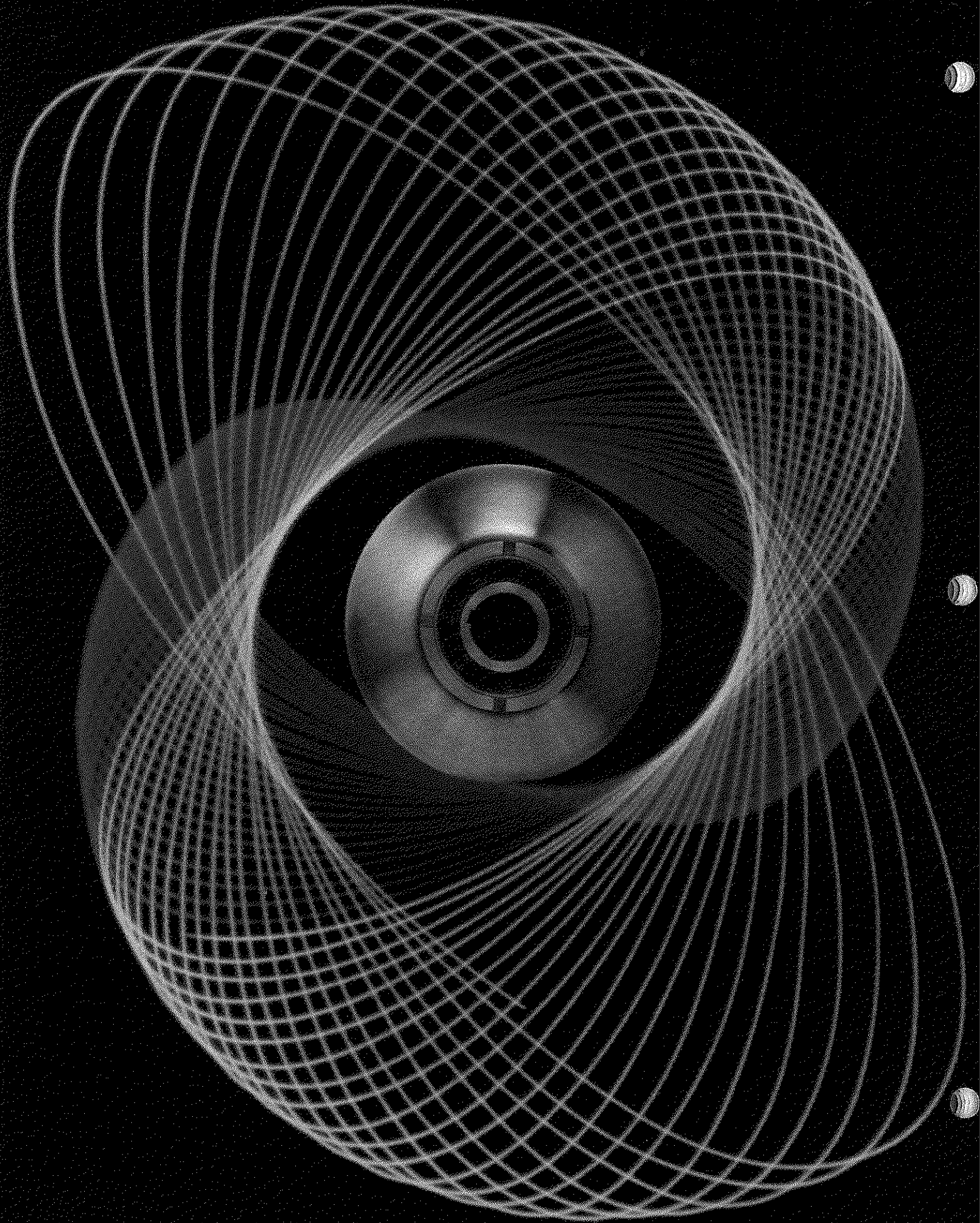
The centrifugal pump of tomorrow.

By I. J. Karassik

Few machines or tools have had a longer history in the service of man than the pump. Every single industrial process which underlies our modern civilization involves the transportation of fluids. Thus the pump is an essential part of industrial processes, and the growth and development of these processes is linked most intimately with the development and the improvements of pumping equipment.

From its first practical applications a century ago, the centrifugal pump has steadily become more efficient, more reliable, and more adaptable to an expanding range of services. What new developments can be expected during the next few decades?

It is difficult to imagine the discovery of any new principle which would supersede the centrifugal pump entirely. Man will always need to move liquids from one point to another, and the centrifugal pump will continue to render this service most readily. But, will this pump be much different from the centrifugal pump of today? And wherein will be the differences?

Trends we can expect to emerge in the development of the centrifugal pump in the years ahead can be roughly classified into five categories: hydraulic characteristics, structural characteristics, materials, drivers, and applications.

Hydraulic characteristics.

The great recent strides in the field of aerodynamics and aerofoil design are being applied more and more widely in

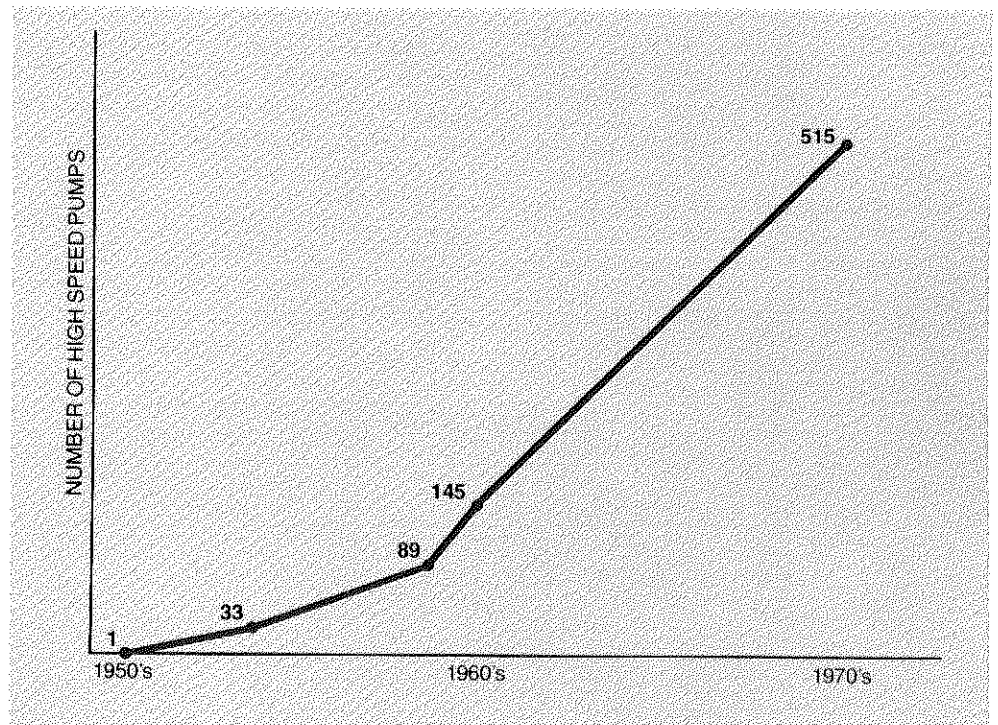Igor Karassik is chief consulting engineer for Worthington Pump Inc.

Figure 1—High-speed boiler-feed pumps have proliferated in central station application.

the hydraulic design of pumps. Already we can see higher head per stage, wider application of inducers, and a greater understanding of the stability of hydraulic performance. These trends will continue until we reach levels where further improvement can no longer be achieved economically, as the centrifugal pump nears the practical optimum in performance for whatever service conditions must be met.

Higher head per stage permits the use of single-stage pumps for conditions that otherwise would require multistage units, or at least reduces the number of stages required. As a result, pump shafts can be shorter and more rigid, with reduced deflection increasing pump reliability.

Higher operating speeds are intimately linked with higher head per stage. The centrifugal pump came into its own with

the availability of high-speed drivers, especially electric motors. Its growth is closely related to improvements in motor design.

Today, heads up to 2800 feet per stage are available commercially. This corresponds to impeller tip speeds of about 440 feet per second. It is probable that head per stage will not exceed 3000 feet in the near future, but the average head per stage will continue to increase. As head per stage continues to grow, and therefore to increase the effect of radial hydraulic loads, there will be more tendency to use double volutes to balance this load.

As the size of 2-pole induction motors operating at 3550 and 2950 rpm (60- and 50-cycle current) increased, so did the speeds for which centrifugal pumps could be built. Finally, not even 3550

rpm was high enough. Where the use of a steam turbine, which is not limited by a maximum synchronous speed, was not desirable, step-up gears began to be applied in greater and greater quantities.

It may be interesting to examine the growth of high-speed pumps—over 4500 rpm—in the specific field of high-pressure boiler feed. Figure 1 shows this growth. A high-speed boiler feed pump was first used in 1954; there were 515 such pumps by 1971. It is probable that in a few more years the 3550-rpm high-pressure boiler feed pump will have disappeared from the scene.

Whether the present 9000-rpm plateau of commercial application will be exceeded in the next few years is problematic. It is certain, however, that just as in the case of heads per stage, the average pump speed will continue to rise.

An inducer is a low-head axial-type impeller with few blades, located in front of a conventional impeller (Figure 2). By design, it requires considerably less npsh than a conventional impeller, and so it is used to reduce npsh requirements of a pump, or let it operate at higher speeds with a given available npsh (Figure 3). This is not a new concept. The first

patent covering inducers was issued to O. H. Dorer of Worthington in 1926. But because of special requirements in the field of liquid-fuel propellent rockets, missiles, and aerospace satellites, a tremendous amount of research has bee devoted to the inducer in recent years. I is now coming into broad commercial application, and the years to come will see its ever-growing use. Development new, exotic materials with greater resistance to cavitation damage will greatly help this trend.

Hydraulic stability problems, involving hydraulic pulsations resulting from the interaction of impeller vanes and collector, or from suction and discharge recirculation within the impeller, are beginning to be better understood. The incidence of these pulsations, and resultant problems of piping vibration, will be eliminated or reduced in the future.

Variations of the basic hydraulic design are to be expected. In the future we w see more unconventional centrifugal pump designs, such as the vortex or recessed-impeller pump, the rotating-casing or "pitot" pump, and other innovations.
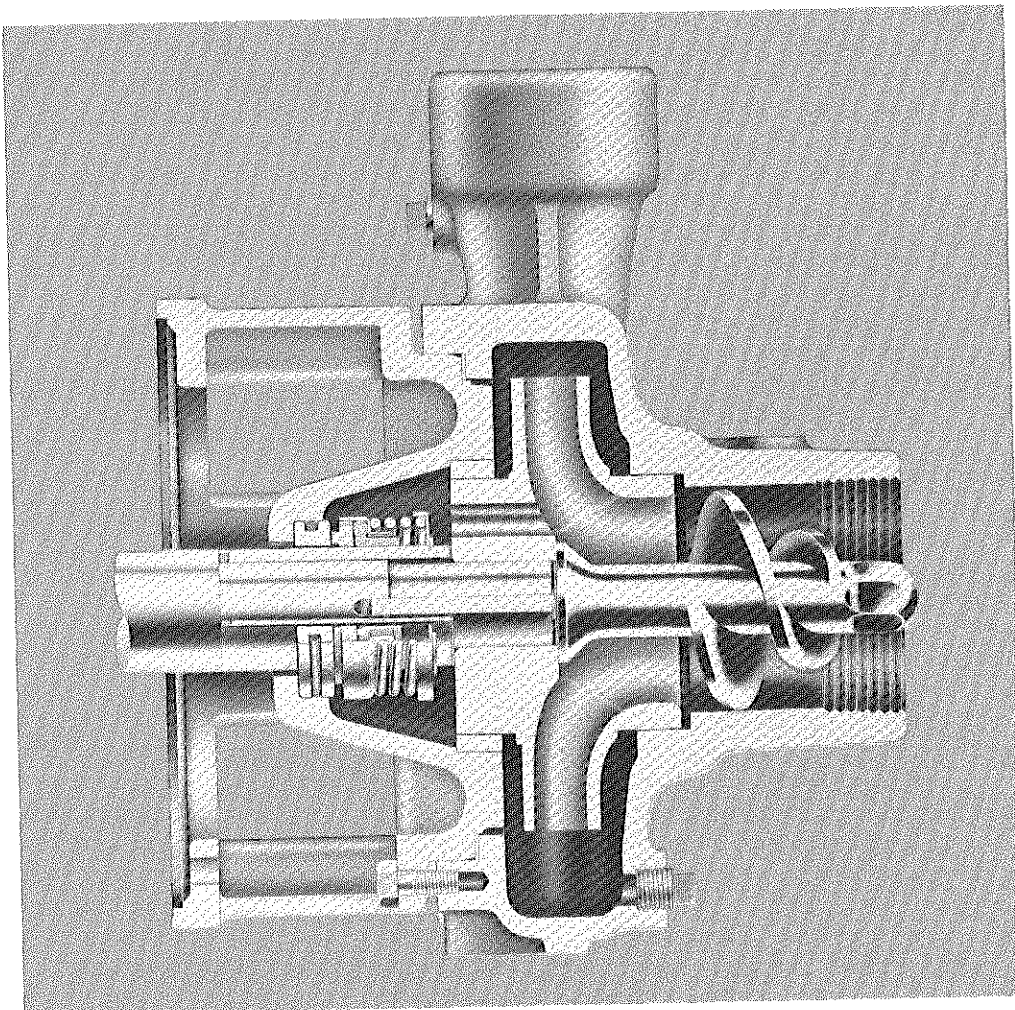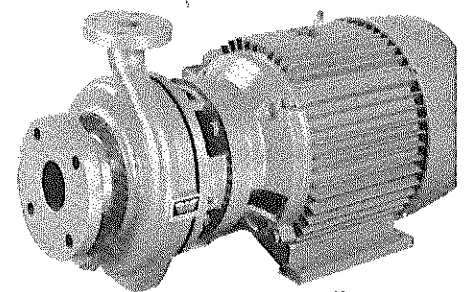

Simple end-suction pumps will take over more sizes and services traditio assigned to axially-split designs.

Structural characteristics.
More mechanical seals will be used. Greater standardization of sizes and increase in the volume of seals purch have made them more economical. T trend will be reinforced by the contin improvement of seal life, broadening

Figure 2—The inducer pressurizes flow, enabling the impeller

applications for seals, and increase in pressures, temperatures and rubbing speeds they can withstand. Better and longer-life packing will also be available.

Ball and roller bearings will continue to be the most commonly used type for centrifugal pumps, even at speeds exceeding 3550 rpm. We can also expect developments which will considerably reduce the amount of cooling water to the bearings in all cases, and raise the level of severity of service below which cooling water may be eliminated. More bearings will be lubricated by the fluid being pumped.

As trends to larger plants and consequently larger sizes of individual pumps continue, journal bearings will also be used, and tilting-pad bearings will find their way into some of the larger pumps. Some special pumps will be built with air or magnetic bearings.

End-suction pumps, both close-coupled and frame-mounted, will continue to displace the classical axially-split double-suction design. It is quite conceivable that close-coupled pumps will reach sizes up to 500 hp, and, in the frame-mounted construction, up to 3000 hp. Vertical in-line pumps will remain popular.

Standardization of dimensions.
In another 10 or 20 years there may well be a worldwide standardization of pump dimensions for a whole series of services —along the lines of the standardization which has existed for a long time among electric motor manufacturers in the U.S. The API standards of the petroleum industry, ANSI standards for the chemical industry, and European ISO standards for chemical pumps are forerunners. While the subject is still quite controversial, the trend, spurred by the increasingly international outlook of both suppliers and users, is inescapable. It will be favorably assisted by worldwide application of the metric system.

Materials.
There will be a trend away from cast iron, to nodular iron for higher strength and shock resistance. More "cast-weld"
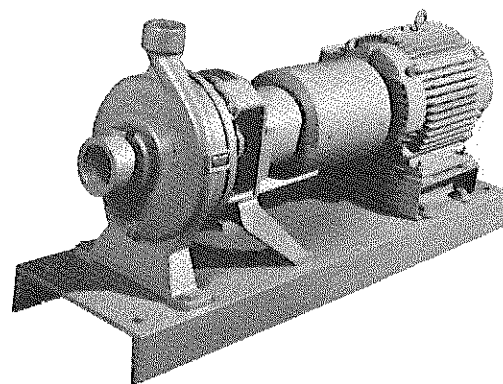
and fabricated construction will be seen for large pumps and nonrepetitive designs.

As the petrochemical industry handles more and more corrosive fluids, demand for high-grade alloys will increase. There will be a greater use of the more exotic metals, such as titanium, for special applications. New plastics with better strength and corrosion resistance will frequently be substituted for metals. Nonmetallic coatings such as neoprene, Teflon, glass, ceramics and carbides will frequently be coated over inexpensive base metals.

Material X.
We all know that regardless of what may trigger the event—hydraulic or mechanical vibration, pump distortion caused by unequal thermal expansion, flashing of the liquid pumped or the presence of hard foreign matter—the majority of pump failures originate with internal contact between stationary and rotating parts.

Figure 3—A well-matched inducer can optimize suction performance throughout the normal capacity range of the pump.

This is why so much effort has been directed towards reducing the probability of internal contact, and why pump designers who wish to increase pump reliability tend to use shorter shaft spans, larger shaft diameters, and larger internal clearances. It is also the reason for the efforts to provide uniform warmup conditions, to protect the installation against flashing and to ensure the cleanliness of the piping system.

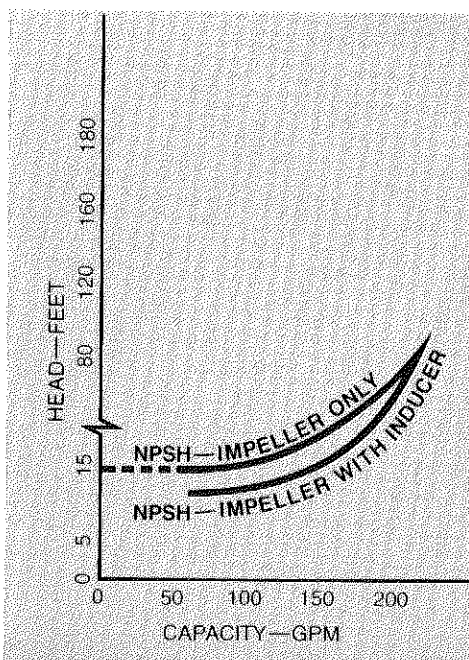**Frame-mounted standard centrifugal pumps may reach 3,000 hp.**

We know that these efforts cannot always be successful. There remains one more approach: a search for materials to withstand the effects of accidental contact. The goal is a "material X," which would have the same coefficient of expansion as the major components of the pump; corrosion resistance at least equal to the major components of the pump; very high resistance to erosion; complete resistance to galling; and the ability to run dry occasionally. Material X would be used for such parts as wearing rings, inter-stage bushings, and balancing devices: all parts with close running clearances.

At first glance these specifications appear utopian in nature. But when we consider the amount of research being done in the field of missiles and aerospace, the goal becomes more plausible. Already, flame-plated surfaces have been used as bearings in the exhaust stream of jet engines, where they have operated
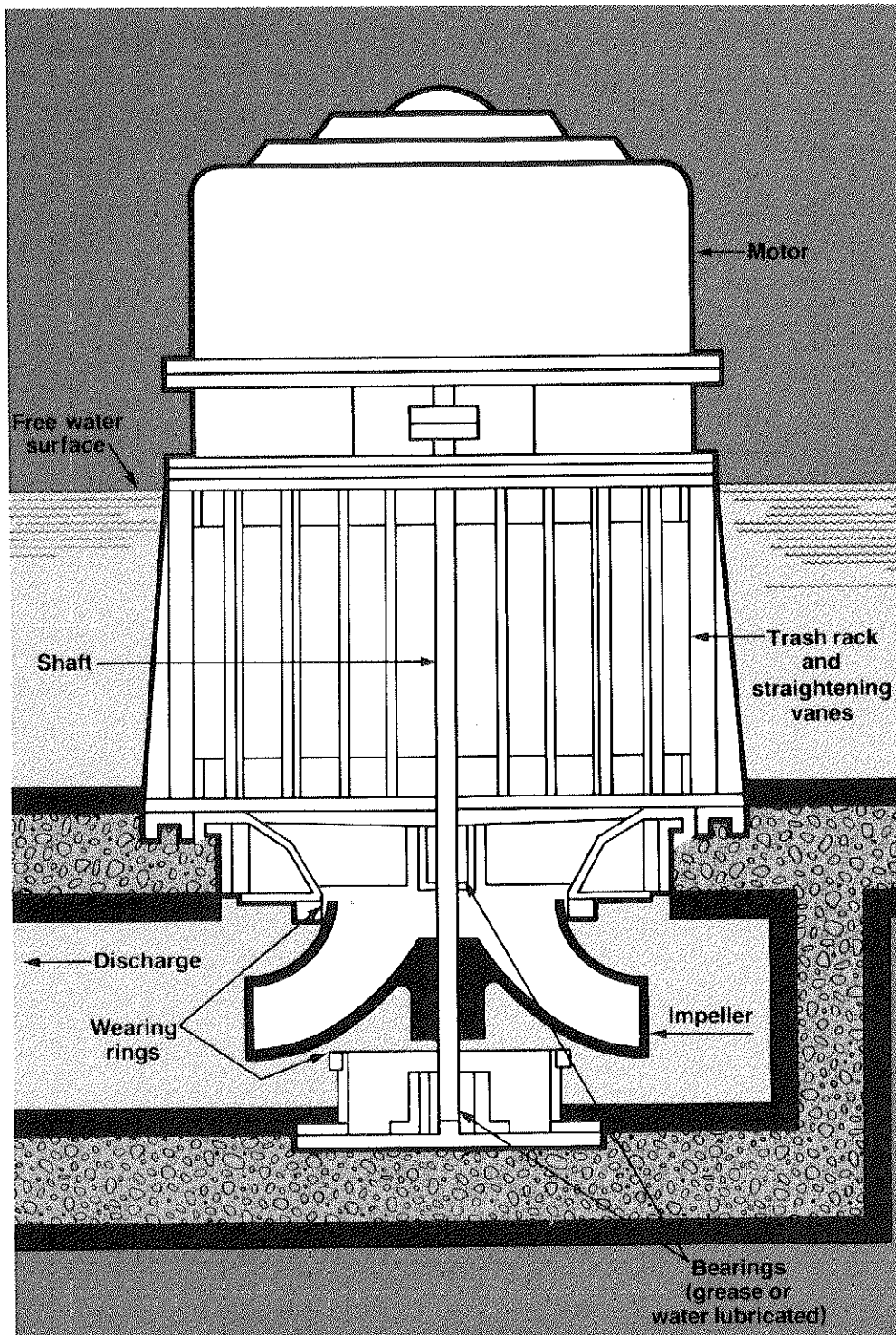
Figure 4—Concrete-cased pumps can be precisely matched to conditions of service. This concrete pump is designed for cooling-tower supply.

without lubrication for periods over 1000 hours; certain exotic materials, metals as well as cermets, have been used for similar purposes in missiles.

Material X will ultimately be found, as a homogeneous material or a coating that can be permanently bonded to the basic pump components. The result will be a dramatic reduction of unscheduled outages and pumping problems for the pump user of the future.

Concrete pumps.

Pump casings made of concrete are not new—they have been used in Europe for many years. What is new is the impetus this type of construction will receive from the fact that in one specific application, central-station condenser circulating service—individual pump sizes have grown to a point tending to make other solutions prohibitively costly. Pump capacities of 300,000 to 500,000 gpm are being encountered, with heads ranging up to 150 feet.

Early models of concrete pumps limited themselves to substituting concrete for the metal casing, without departing from conventional design configurations. Future designs, such as the approach shown in Figure 4, will take maximum advantage of the change of materials to reduce cost, eliminate dry pit construction, reduce excavation, eliminate stuffing boxes and permit exact optimum casing sizing for individual conditions of service.

Drivers.

Regardless of how energy may be produced in the future, it will continue to be supplied in the form of electricity. Thus, the electric motor can be expected to outlast the waterwheel, the steam turbine, the internal combustion engine and the gas turbine. In looking at the future, we must try to see what the electric motor of tomorrow will be like.

As the motor became the predominant driver, most lines of centrifugal pumps were designed to suit it. Their speeds were tied to the speed of two-pole motors: 3600 rpm for 60-cycle areas an

3000 rpm where 50-cycle current is used. This practice introduced a divergence between designs for 60-cycle and 50-cycle geographical areas, one difficulty in the worldwide application of standardized pump lines.

Standardization on one single worldwide current frequency is an unlikely situation for the near future. The best solution, as far as centrifugal pump needs are concerned, is electric current at higher frequencies than 60 or 50 cycles. For instance, with 150 cycles, a two-pole motor can operate at 9000 rpm; if we raise the frequency to 300 cycles, this speed can be met with a four-pole motor. Here, there is a bright future for the solid-state frequency converter.

Silicon-controlled rectifiers are already available commercially for ratings up to 100 kw. They are used principally to provide efficient speed variation for the driven equipment. In the future, it is likely that the SCR will be used both to multiply frequency and to vary it to meet variable conditions of pump service. While it is difficult to predict how large an adjustable-frequency drive can become while remaining economically competitive with established technology such as hydraulic couplings and magnetic drives, the maximum economical size will continue to grow with time.
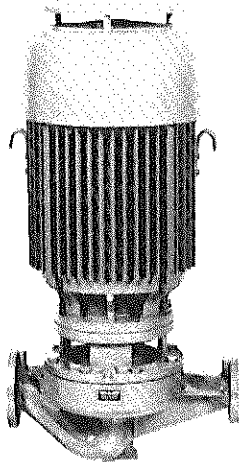
The vertical in-line end-suction pump will continue to gain in popularity.
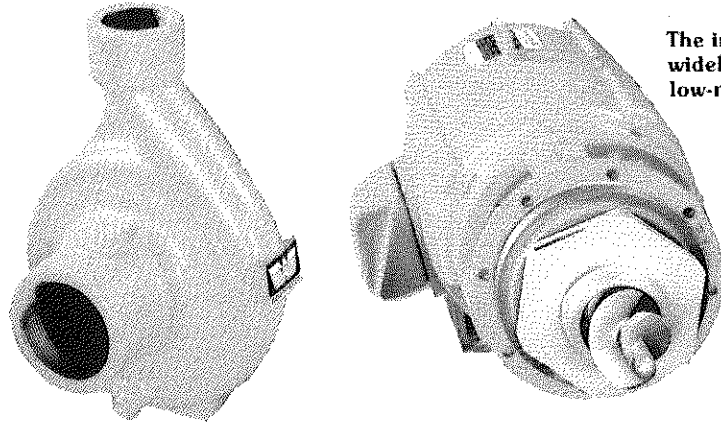
The inducer will be widely used in low-npsh services.

Application trends.

The demand for reliability is increasing. A growing number of installations will completely eliminate spare pumping equipment. To meet these stringent reliability requirements, there will be more monitoring of pump operation, with automatic interconnects, relays, protective devices, etc.

Growing awareness of environmental impact will lead to greater insistance on low noise levels, through hydraulic improvements, greater use of attenuating devices, insulation, etc.

A wider range of liquids will need to be pumped: liquid metals, coal/water slurries, iron-ore concentrate slurries, rare metals in liquid salt form, wood chips carried by water, etc. In some cases the water/solids mixtures will be handled by centrifugal pumps in an "as is" state. In others, novel pumping systems will be developed, utilizing centrifugal pumps which handle the carrier liquid while bypassing the solids themselves. There will be a growing demand for pumps to handle gas-laden liquids in ever-greater concentrations. More and more cryogenic pumps will also be required.

Some of the changes described here are already well on the way, some may not arrive for decades, if at all—but change is inevitable. What will not change into the foreseeable future is the preeminence of electric motor-driven centrifugal pumps for moving fluids. While their look may be new, their materials of construction strange, and some of the fluids handled only dreamed of by present technology, there can be no doubt that centrifugal pumps will be with us for a long time to come.
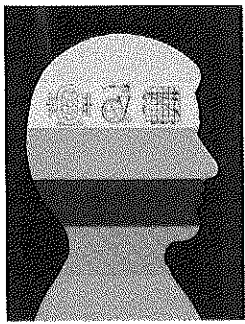
pump refresher

Basic hydraulics:
How fluid flow analysis
solves pumping problems.

Part 4

By John H. Doolin

A pumping system of any kind would be useless if the friction loss through the pipe were not properly determined for the flow that was ultimately required. In addition, the proper size pump has to be selected and matched to system requirements. In the last issue we analyzed the two basic equations of fluid flow (Bernoulli's Equation and the Continuity Equation), given friction losses. This installment of Pump Refresher explains how to determine friction losses in a pipe system.

The determination of friction loss in a pipe system is very often the most significant part of a fluid flow problem. Losses can be caused by any number of factors, not the least of which is the friction in perfectly straight pipe. This is most significant where there are extensive lengths of such pipe involved. Changes in fluid velocity also cause losses, whether the change is a reduction in velocity as the fluid goes from a smaller diameter pipe to a larger diameter, or the reverse, an increase in velocity from a large diameter pipe to a small one.

Fittings such as elbows, tees, and other bends also cause losses in pipe systems, while control valves are another very significant factor. The Hydraulic Institute friction loss tables, showing friction loss in common types of pipe material, are based on the Darcy-Weisbach Formula: $H_f = f \times \dfrac{L}{D} \times \dfrac{V^2}{2g}$. That is, the friction loss H_f is equal to the empirical factor f times the length of pipe L divided by the pipe diameter D, times the velocity head factor $\dfrac{V^2}{2g}$.

In addition to the pipe friction tables, the standards of the Hydraulic Institute also include convenient charts for determining resistance coefficients for all types of valves and fittings. These are based on the formula $H_f = \dfrac{KV^2}{2g}$; that is, the friction loss H_f is equal to coefficient K times the value for velocity head.

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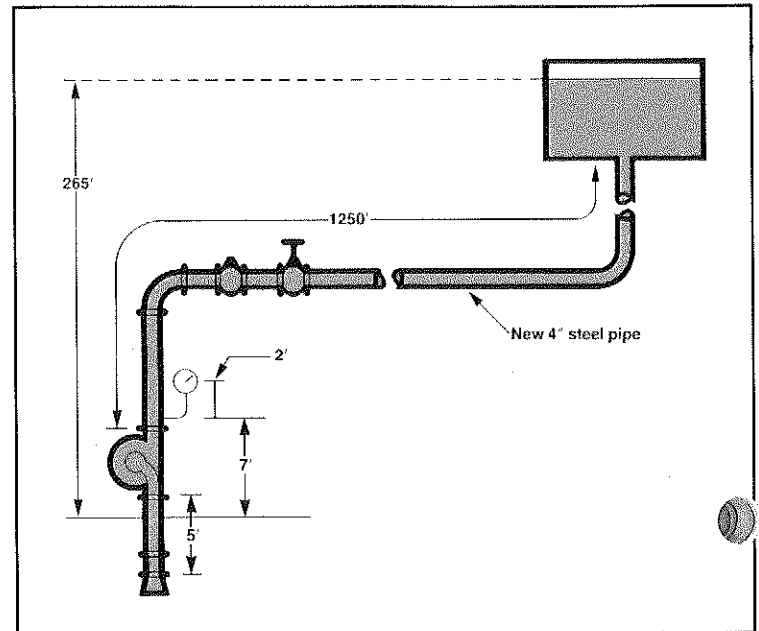
Figure 1—Pipe friction problem.

The tables supply coefficient K, which is then substituted into this formula.

Friction loss in a complex system.
Figure 1 shows a typically complex pipe system. 200 gpm of water enters a system of 4-inch steel pipe through a foot valve, is picked up by the pump and discharged upward through an elbow, a swing check valve, a gate valve, another elbow and finally a sudden enlargement to an open tank. The total length of pipe above the pump is 1250 feet.

In determining the friction loss, the first element to be considered is loss in the total length of pipe; that is, 1250 feet above the pump plus 5 feet of pipe below the pump, or 1255 feet.

The Hydraulic Institute table for 4-inch steel pipe gives the head loss as 2.27 feet per 100 feet of pipe with 200 gpm flowing. The total friction loss then the pipe alone is 2.27 divided by 100 x 1255, or 28.49 feet.

The other losses in this pipe friction problem are determined by the friction loss formula, $H_f = \dfrac{KV^2}{2g}$.

The friction loss tables give $\dfrac{V^2}{2g}$ as .395 feet for 200 gpm and 4-inch steel pipe, and the resistance coefficient tables provide the value of coefficient K for the various fittings. Substituting into the friction loss formula and adding up all losses, we find a total friction loss through all pipes and fittings of 30.22 feet.

Now we can go back to the pipe system problem illustrated in Figure 1 and find the pump energy needed to put 200 gpm through this system. Rearranging Bernoulli's Equation (Pumpworld, Volume 3, Number 3) to solve for pump energy, we get

$$E_p = \frac{P_2}{\gamma} - \frac{P_1}{\gamma} + Z_2 - Z_1 + \frac{V_2^2}{2g} - \frac{V_1^2}{2g} + H_f$$

In this problem, the gage pressure on the surface of the water at the inlet and the outlet is equal to 0, because both the sump and the tank are open. Therefore, $\dfrac{P_2}{\gamma}$ and $\dfrac{P_1}{\gamma}$ are both equal to zero. The static elevation of the upper tanks is 265 feet compared with a static level of 0 if the datum line is the lower surface of water. The velocity of the water at the surface in both chambers is zero. The friction loss we have just calculated as 30.22 feet. Substituting these numbers in our formula,

$$E_p = 0 - 0 + 265 - 0 + 0 - 0 + 30.22$$

we end up with a required total pump head of about 295 feet.

We may wish to know what pump head is required for capacities other than 200 gpm. We could, of course, go back to the Hydraulic Institute tables and look up all the pipe, valves and fittings for the new flow-rate. But we can arrive at the same result without redoing all that work.

In the calculation for friction loss, we found that the general relationship H_f is equal to some function of $\dfrac{V^2}{2g}$:

$$H_f = f \frac{V^2}{2g}$$

This means that friction loss varies as the square of

the velocity in the pipe, or as the square of the flow in gpm. Friction loss at 200 gpm equals about 30 feet, and since 100 gpm is half of 200 gpm, friction loss at 100 gpm is the square of ½, or ¼, of 30 feet; that is, 7.5 feet.

Similarly, 300 gpm is 1.5 times the original 200 gpm. To find the friction loss we multiply 1.5 squared times 30, and find the friction loss is 67.5 feet:

H_f at 200 gpm = 30 ft.
H_f at 100 gpm = $(\tfrac{1}{2})^2$ = ¼ x 30 ft. = 7.5 ft.
H_f at 300 gpm = $(1.5)^2$ = 2.25 x 30 ft. = 67.5 ft.
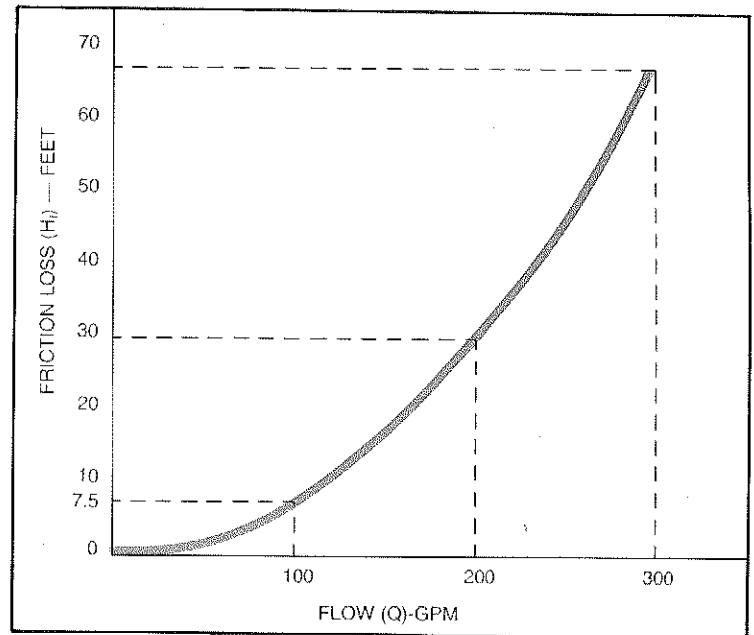

Figure 2—Friction loss.

System head curves.

With these three points we can draw a curve of friction loss versus flow, as shown in Figure 2. Then, since the change in static elevation of 265 feet is a constant regardless of flow, we can add 265 feet to each of the three values and draw the *system head curve* shown in Figure 3. This curve depicts the pump energy or total head required for any capacity.

All system head curves are not alike, of course. Various types of systems present differing system head curves. One extreme is a system which is all
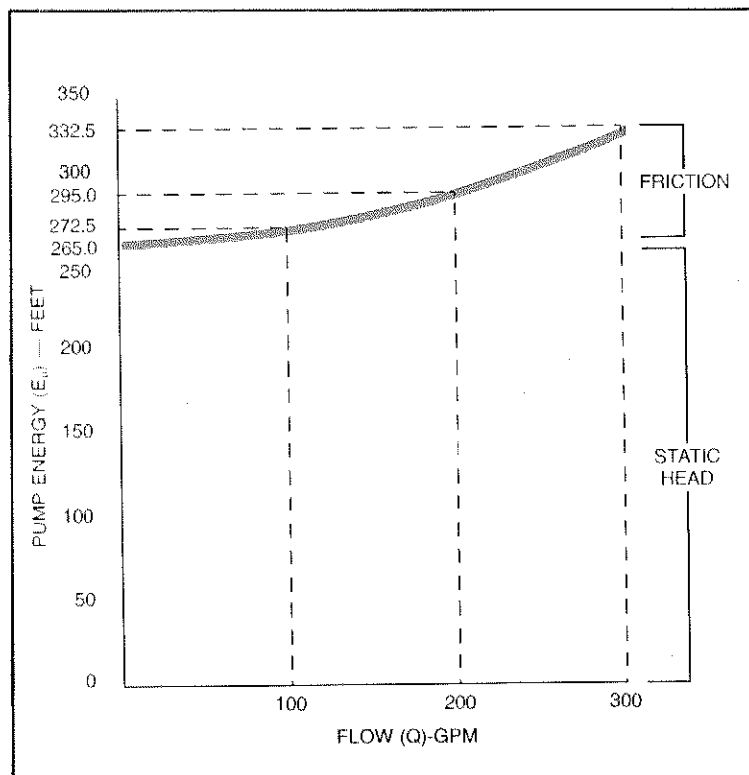
Figure 3—System curve.

Figure 4—Typical system curve for sprinkler system (all friction).

friction, with no change in static elevation and no change in pressure, such as a sprinkler system or a system of spray nozzles. The system head curve for such an installation would look like Figure 4.

The other extreme is the system curve for a boiler-feed system. In this case, the boiler pressure is the predominant factor to be overcome by the pump. The effects of friction losses and change in static elevation are usually minor, and consequently the system head curve for a boiler feed application is practically a horizontal straight line, as shown in Figure 5.

The most typical system head curve, however, is like the one we worked out in Figure 2, a combination of static elevation or pressure differential and friction losses in the system. It should also be noted that while a system head curve can be worked out for most systems, its accuracy may not always be precise. Distortions do affect the real system head curve in many ways.
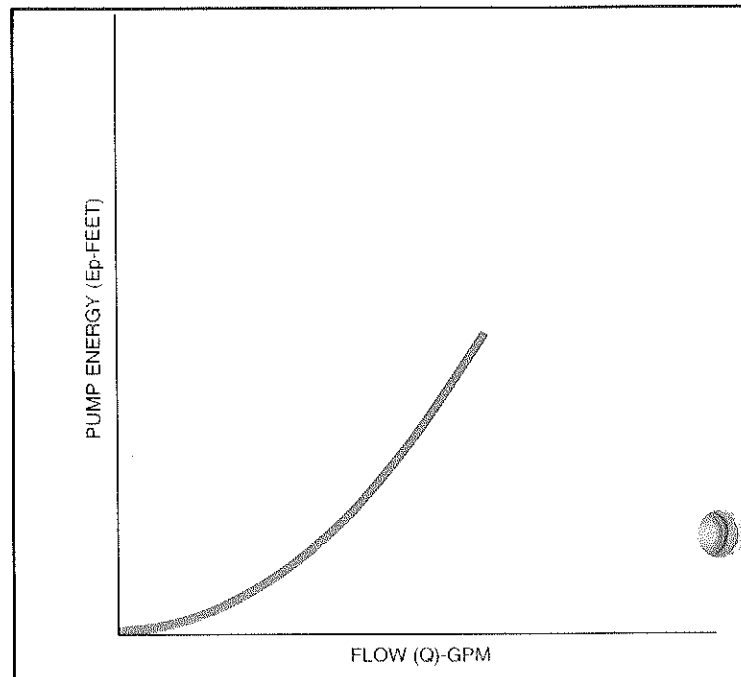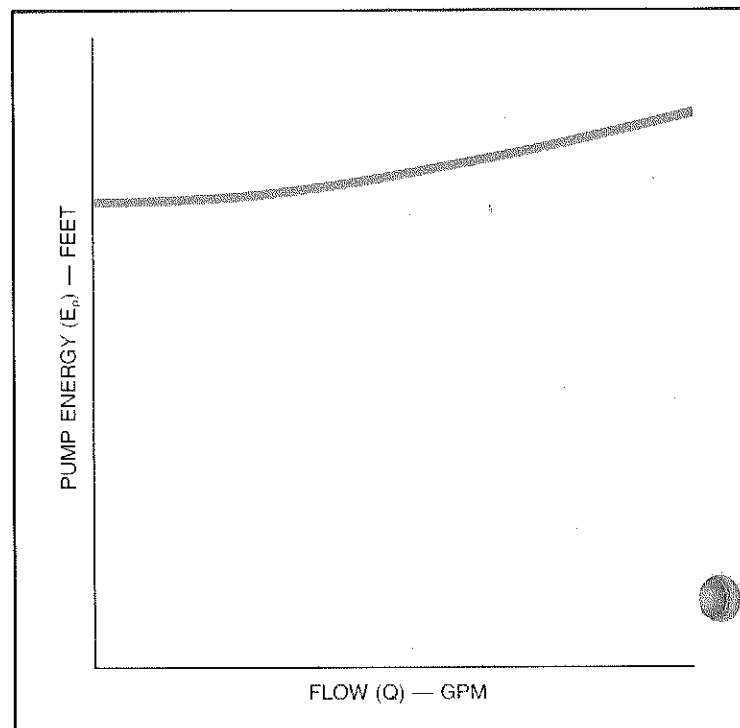

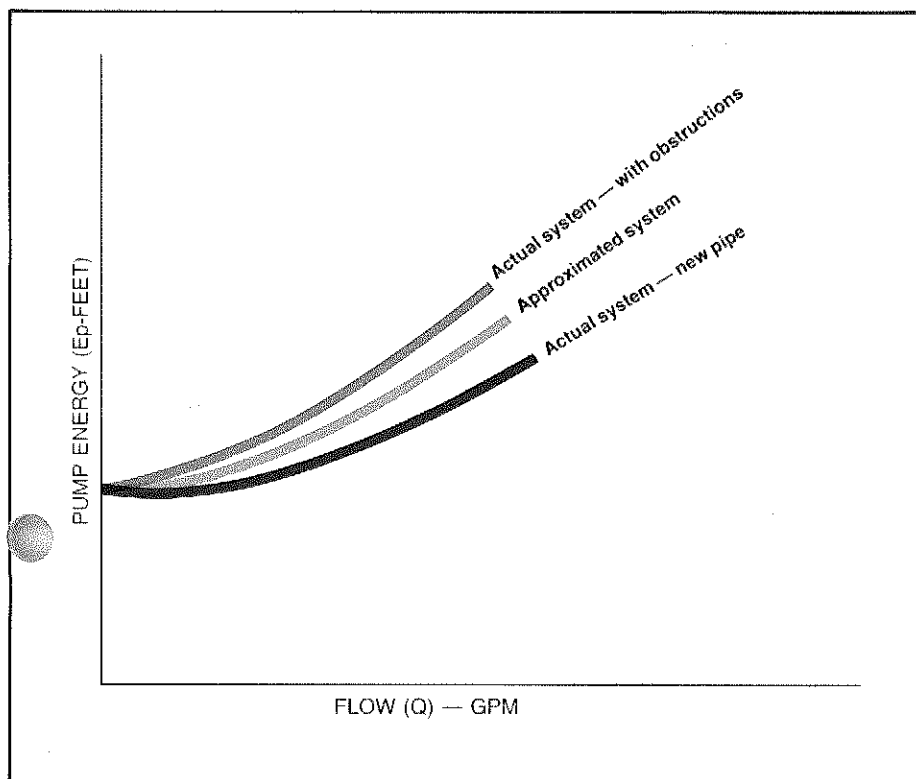Figure 5—System curve for boiler feed (mostly pressure difference).

Figure 6—System curve distortions.

The engineer may calculate the friction losses in the system and arrive at an approximate curve. However, there may be undetermined obstructions in the system which he has not taken into consideration. For example, the gate valve may be partially closed, the suction check valve may be clogged, or there may be strainers in the line which are clogged. Gaskets between flanges may not be properly cut so that there is an obstruction due to the small inside diameter of the gasket. Extra heavy pipe may be used which has a smaller inside diameter, resulting in greater pipe losses. The pipe roughness may be greater than anticipated.

Again, Hydraulic Institute tables refer to relatively clean, plain steel pipes. Scale buildup, special coatings to protect against corrosion, or any other variation which adds roughness or reduces the pipe diameter would increase the friction in the pipe and in the system. So while the theoretical system head curve may be calculated as shown by the

"approximated" line in Figure 6, the real system head curve may be more like the one labeled "obstructions."

On the other hand, the real system head curve may be lower than calculated, due to more conservative estimates on the part of the engineer. For example, he may have calculated his losses on old pipe with scale buildup, when in fact the system uses new pipe.

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