CENTRIFUGAL PUMPS AS HYDRAULIC TURBINES.

By A. Agostinelli and L. Shafer

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In 1978 the United States Congress passed PURPA, the Public Utilities Regulatory Policy Act, and in doing so, caused a dramatic upsurge in U.S. hydropower interest. This has indirectly, but significantly, increased demand for pumps run as hydraulic turbines. PURPA requires that small producers of electricity be allowed to hook into utility power lines and that the utilities purchase back unused electricity at avoided cost rates (the rates at which it costs the utility to produce electricity).

Increasing energy costs have also stimulated interest in using available water resources to drive hydraulic turbines. These applications range from mammoth projects harnessing the tides and flow of major rivers and elevated lakes to very small applications by private individuals with a stream running across their property.

If the small hydropower market continues to expand as rapidly as expected, the extremely limited number of small hydraulic turbine manufacturers will create a shortfall of conventional turbine machinery. The use of pumps running in reverse as turbines is an excellent alternative to conventional hydraulic turbines and even offers many unique advantages.

Worthington has undertaken an extensive testing program to further refine what is already known about the operation of pumps as hydraulic turbines. This article discusses some basic findings resulting from this testing and lays the ground for more in-depth articles in the future.

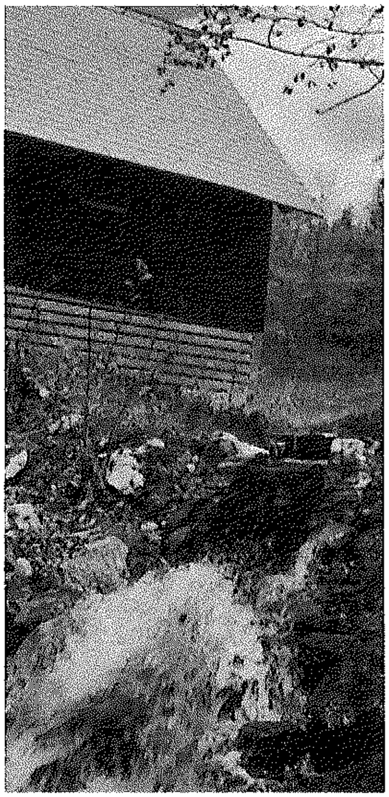
Pumps operating in reverse as turbines yield good efficiencies and can be run even more efficiently by operating multiple pumps of various sizes rather than one large conventional hydraulic turbine.

Other advantages which pumps run in reverse have over turbines are: pumps are more readily available in many sizes; they are several generations ahead of conventional hydrualic turbines in cost effectiveness; pumps are less complex, making them easier to install and maintain, and simpler to operate; and pumps are available in a broader range of configurations than conventional hydraulic turbines—wet pit, dry pit, horizontal, vertical, and even submersible, to mention a few.

Dual capability predicted.

Centrifugal pumps from radial flow to the axial flow geometry can be operated in reverse and used as hydraulic turbines. This dual capability is not just happenstance, since turbomachinery theory predicts this capability. Furthermore, because this theory is applicable, a hydraulic turbine follows the same affinity relationships as do centrifugal pumps. Consequently, the performance of a turbine can be predicted accurately from one set of operating conditions to another, and new turbine designs can be "factored" from existing designs.

Over the years Worthington has tested many pumps as turbines. From these tests it has been observed that when a pump operates as a turbine: its mechanical operation is smooth and quiet; its peak efficiency as a turbine is essentially the same as its peak efficiency as a pump; head and flow at the best efficiency point as a turbine are higher than they are as a pump; and the power output of the turbine at its best efficiency point is higher than the pump input



Small hydropower projects are on the upsurge in the U.S.

power at its best efficiency point.

Typical performance characteristics.

A comparison of the characteristics of normal pump operation with the characteristics of the same pump operated as a turbine at the same speed is shown in **Figure 1.** The curves are normalized by the values of head, flow, efficiency, and power at the pump BEP (best efficiency point). As mentioned previously, note that the location of the turbine BEP is at a higher flow and head than the pump BEP. The ratio of the turbine capacity and head at BEP to the pump capacity and head at BEP has been observed to vary with specific speed—ratios of 1.1 to 2.2 having been determined by test.

There are two other important characteristics of pumps operating as turbines shown in Figure 1. The first of these is that the turbine maximum efficiencies tend to occur over a wide range of capacity. Consequently, relatively wider ranges of turbine operating head can be accommodated without an adverse effect upon efficiency.

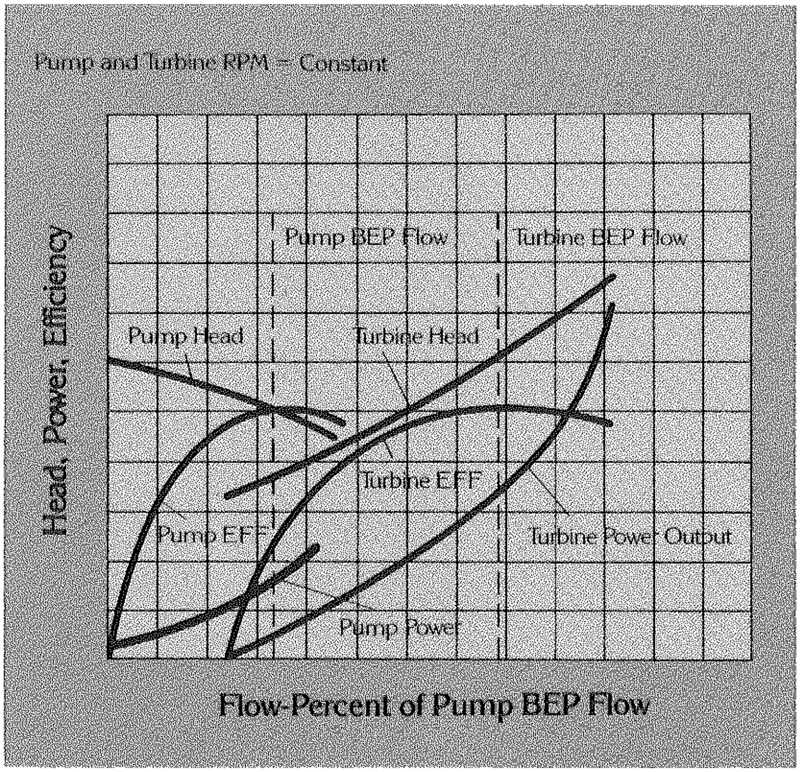


Figure 1—Normalized performance characteristics for a pump operating in the normal pump mode and in the turbine mode.

Secondly, note that there is a value of head at which the turbine power output is zero even though there is flow through the unit (this point is called the runaway speed). Further reduction in head below this value causes the turbine to begin absorbing power, assuming the connected load is capable of providing the power. The flow corresponding to the head at zero power varies from about 40 to 80 percent of the flow at turbine BEP, depending upon specific speed.

The turbine performance, or rating curve, normally supplied to a customer is either the one shown in **Figure 2** or **3**, whatever his preference. Figure 2 is a plot at constant speed with capacity as abscissa, while Figure 3 is a plot at constant head with speed as abscissa. Given the performance test in either format, the other can easily be obtained by use of the affinity relationships.

Runaway speed.

Note that the runaway speed can be read directly from the curves of Figure 3. The runaway speed could also be calculated using Figure 2 and the affinity laws, i.e., by taking the product of the value of speed and the square root of the ratio of the head for which runaway needs to be determined to the head at zero power output.

As illustrated, the magnitude of the runaway speed can easily be determined for any operating condition, provided its value is known for

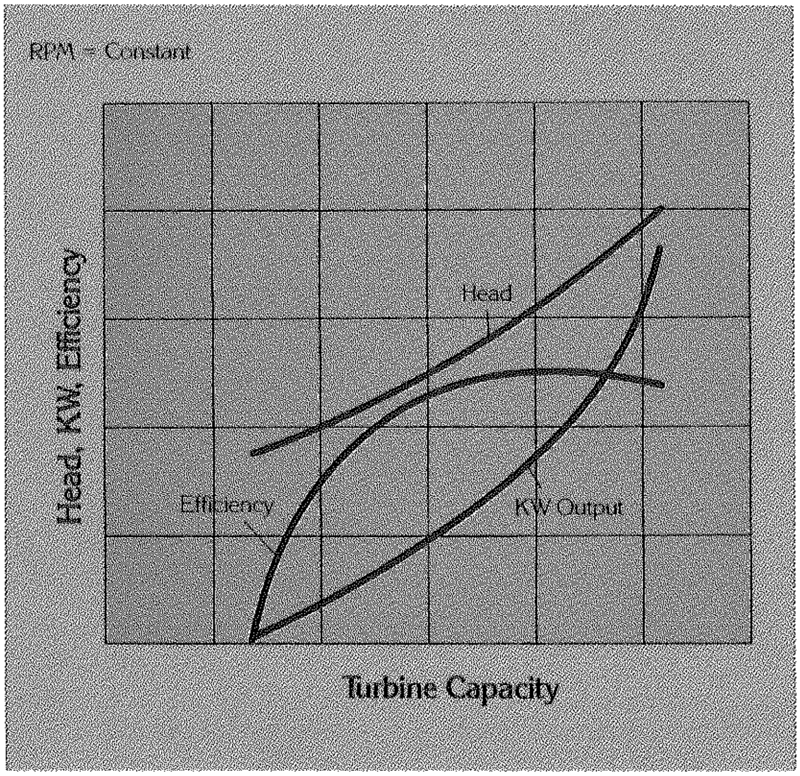


Figure 2—Typical turbine performance curve for constant speed operation.

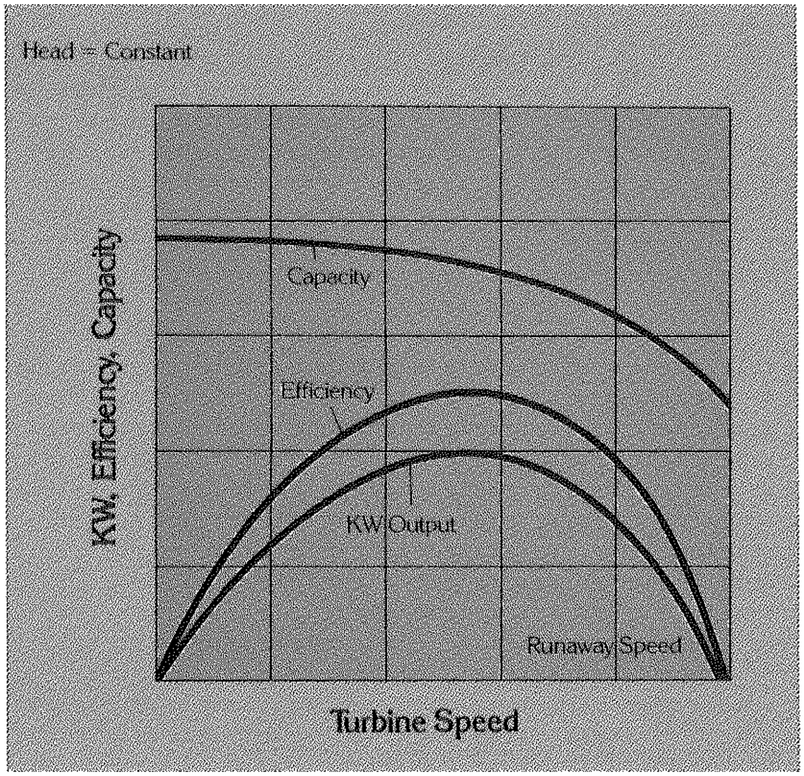


Figure 3—Typical turbine performance curve for constant head operation.

a given condition. This is important data because the magnitude of the runaway speed could affect the structural integrity of the rotating equipment, making it necessary to incorporate overspeed protection in the control system.

Cavitation.

Just as in a pump, at any point in the machine where the local pressure drops to the vapor pressure of the liquid, vapor is formed and cavitation damage can occur. Sufficient outlet or backpressure must be maintained to prevent cavitation, just as adequate suction pressure must be maintained on a pump. The value of the available backpressure is TAEH (total available exhaust head), and the value of the backpressure required for proper turbine operation is TREH (total required exhaust head).

Design changes.

In most instances no design changes or modification need to be made for a pump operating as a turbine. When a selection is made, a design review is required, however, because when operating as a turbine the rotation is reversed and operating heads and power output are generally higher. Consequently, a design review would include items such as: checking that threaded shaft components cannot loosen; evaluating the adequacy of the bearing design; shaft stress analysis; and checking the effect of increased pressure forces.☐

EFFECT OF NOZZLE LOADS ON PROCESS PUMPS.

By John H. Doolin

A long-standing problem for both plant designers and pump manufacturers has been the effect of nozzle loads on process pumps. Loads or forces on pump nozzles result from expansion in connecting pipes and are unavoidable due to large changes in temperatures as a process system is brought up to operating temperature. On the other hand, a high-temperature process pump is a good example of precision machinery with close-running clearances, delicate mechanical seals, and precision ball bearings—and is not intended to be a pipe anchor.

Although this problem has not changed over the years, plant designers have learned more about prediction and control of pipe expansion and have done much to minimize expansion forces. In addition, pump manufacturers have learned more about the capability of pumps to operate with large nozzle loads. It is now possible to quantify the magnitude of pipe forces, and the amount of such forces the pump can carry, so that serious problems can be avoided. This article, which appeared in Hydrocarbon Processing, discusses some of the factors relating to the effect of nozzle loads on process pumps.

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The Hydraulic Institute, an association of U.S. pump manufacturers, appointed a committee to study the problem of nozzle load effects on process pumps and make recommendations. The committee reported that there are four factors to be considered in determining the effect of nozzle loads: material stress in pump nozzles due to forces and bending moments; distortion of internal moving parts affecting clearances in precision parts; stresses in pump hold-down bolts; and distortion in pump supports and baseplate resulting in driver coupling misalignment.

Stress in pump nozzles.

These stresses should be calculated as the combined stress resulting from external forces and bending moments from piping, plus the influence of internal fluid pressures. A report to the committee by A. T. Ganzon offered the following solution to the determination of stress levels.

The discharge nozzle was assumed to be the weakest portion of the casing under pipe loads. Furthermore, the throat area was considered to be the weakest section of the nozzle section. These assumptions must be verified by actual tests.

If we take a small elementary cube out of the section of the nozzle in question, we will find three main stresses acting on it **(Figure 1)**. In general, these stresses will come about from the following loads:

σ_1 (normal)—longitudinal due to pressure, pipe forces, and moments

σ_2 (normal)—hoop forces

σ_3 (normal)—internal pressure (only present in thick-walled vessels)

σ_s (shear)—torsion about nozzle axis

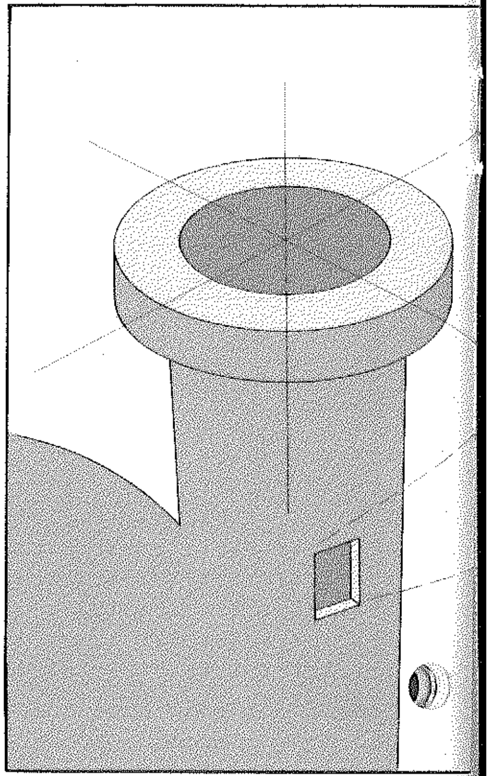


Figure 1—Three principal stresses act

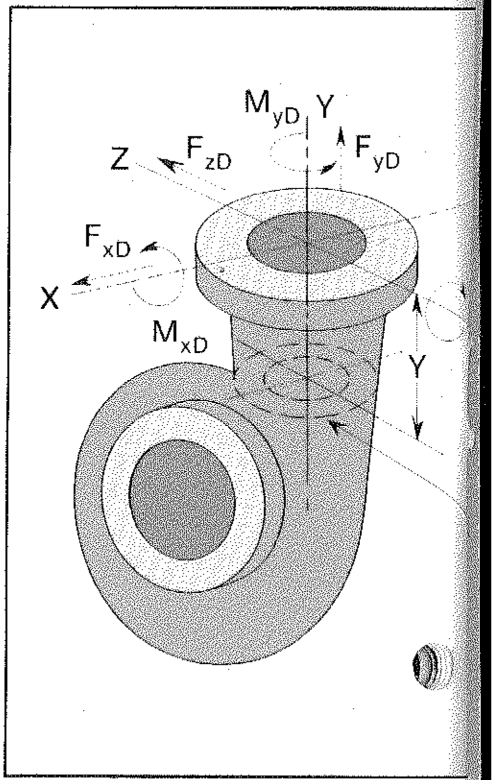
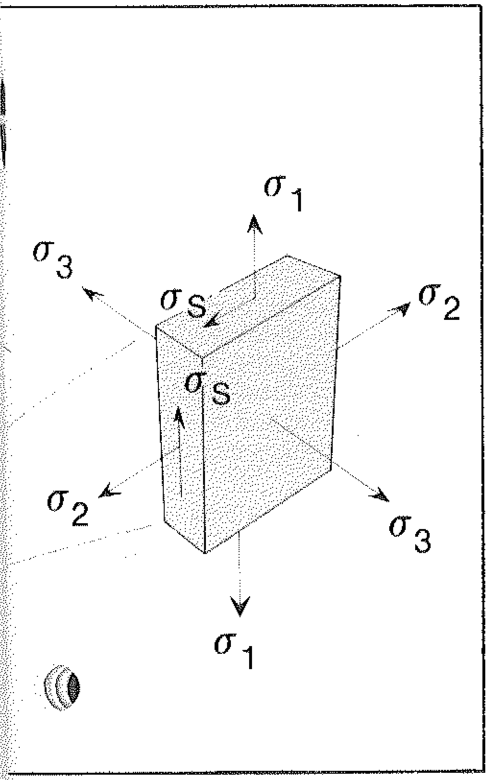
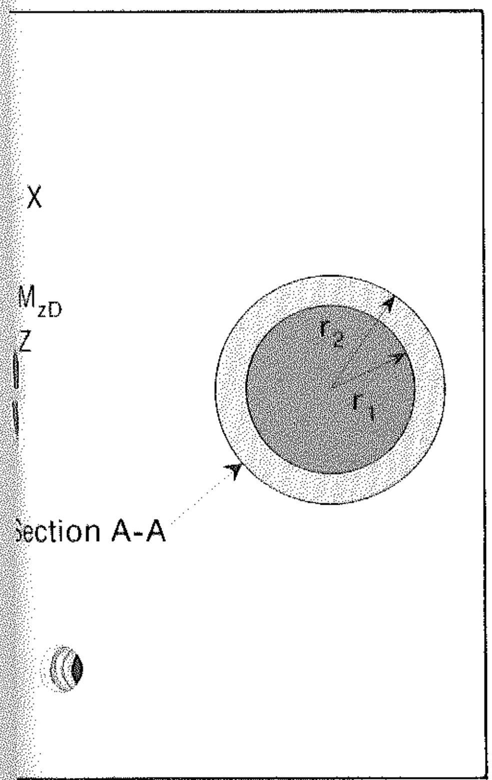


Figure 2—Forces and moments acting



ng on a cube section.



n a nozzle section.

Analysis involving triaxial stresses is complicated, and the improved accuracy of the results is hardly worth considerable effort. The levels of external forces and moments involved indicate that σ_3 can be conveniently disregarded, since its magnitude is quite small, thus eliminating one of the normal stresses. Another stress that can be disregarded, though not listed in the report, is the longitudinal shear arising from an external force. Under these assumptions, sufficiently accurate analysis only involves two normal stresses—biaxial.

Using the maximum principal stress theory for failure analysis, we arrive at the following formulas listed below.

These are also based on certain assumptions: there is no discontinuity effect where the volute joins with the nozzle; the strength of the nozzle is constant for all directions on the *X-Z* plane; the weakest point of the nozzle is at the throat; and σ_3 is disregarded. Axis convention will be as shown in **Figure 2.**

Distortion of internal moving parts.

Under ideal conditions, the rotor of a centrifugal pump operates so that only forces imposed by the liquid end are supported by the shaft and bearings. These forces are: radial hydraulic forces on the impeller, often referred to as radial reaction; axial hydraulic forces on the impeller; thrust on the shaft equal to suction gage pressure times shaft area in the stuffing box; and torque required to drive the impeller.

These forces are readily determined by pump designers, and shafts and bearings are properly sized accordingly. However, external forces imposed on nozzles can add to this load.

Nozzles loads can cause suf-

Shear stress,

$$S_{S\,MAX} = \left[\left(\frac{\sigma_1 - \sigma_2}{2}\right)^2 + \sigma_S^2\right]^{1/2}$$

$$= \left[\left(\frac{\dfrac{M_R r_2}{I_{AA}} + \dfrac{F_R}{\pi(r_2^2 - r_1^2)} - P\left(\dfrac{r_1^2}{r_2^2 - r_1^2}\right)\left(\dfrac{r_2^2}{r_1^2} + 1\right)}{2}\right)^2 + \left(\frac{M_T r_2}{2I_{AA}}\right)^2\right]^{1/2}$$

Tensile stress,

$$S_{T\,MAX} = \frac{\sigma_1 + \sigma_2}{2} \pm S_{S\,MAX}$$

$$= \left[\frac{\dfrac{M_R r_2}{I_{AA}} + \dfrac{F_R}{\pi(r_2^2 - r_1^2)} - P\left(\dfrac{r_1^2}{r_2^2 - r_1^2}\right)\left(\dfrac{r_2^2}{r_1^2} + 1\right)}{2}\right] \pm S_{S\,MAX}$$

$$M_R = \overrightarrow{(M_{xD} + F_{zD} \cdot Y)} + \overrightarrow{(M_{zD} - F_{xD} \cdot Y)}$$

$$F_R = F_{yD}$$

P = maximum working pressure
I = moment of inertia at throat

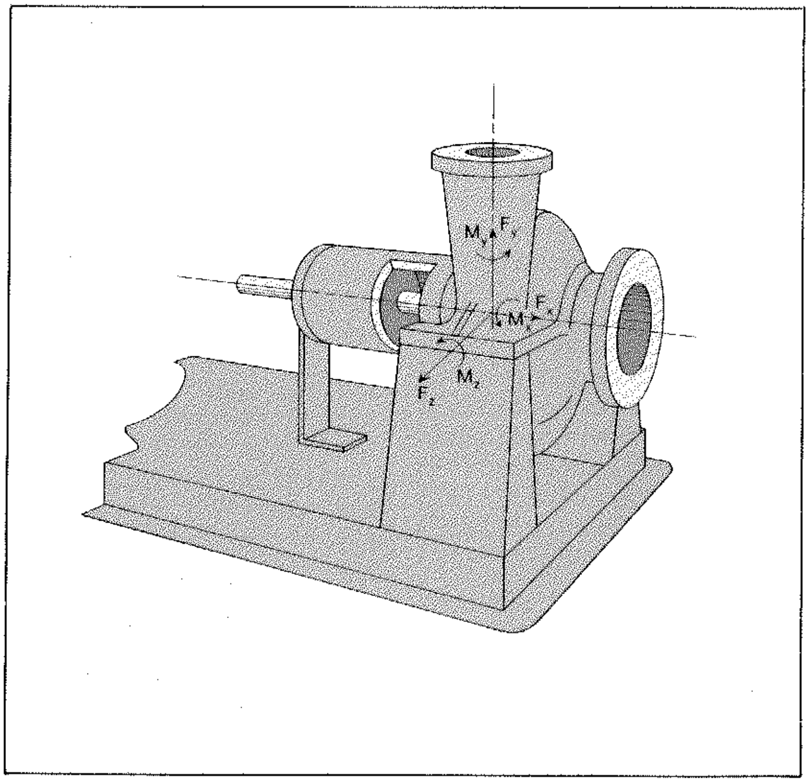


Figure 3—Forces and moments acting on a pump.

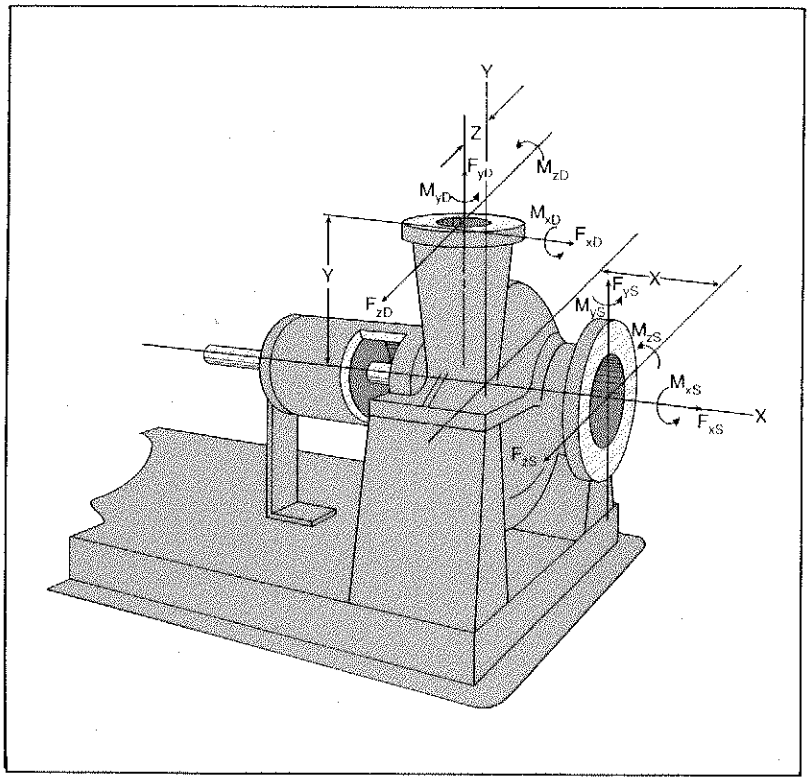


Figure 4—Resultant forces and moments at the pump centerline.

ficient bending in the bearing housing to produce misalignment between the bores for the bearing outer race. Resultant loads are imposed on the bearings. In addition, wearing ring bores can be distorted, and alignment between mechanical seal parts can be disturbed. There is no simple determination of the relationship between these loads and the point where distortion becomes significant. This can best be determined by experience or laboratory tests.

Stresses in pump hold-down bolts.

The strength of bolts which attach the pump casing to the baseplate or support must be considered also. However, before this can be done, separate forces on suction and discharge nozzles must be resolved into a single set of resultant forces and moments at the pump centerline. **Figure 3** shows the separate forces and moments acting on the pump. These are resolved into resultants at the pump centerline as shown in **Figure 4**:

$$F_x = F_{xs} + F_{xD}$$
$$F_y = F_{Ys} + F_{yD}$$
$$F_z = F_{zs} + F_{zD}$$
$$M_x = M_{xs} + M_{xD} + F_{zD}(Y) - F_{yD}(Z)$$
$$M_y = M_{ys} + M_{yD} - F_{zs}(X) + F_{xD}(Z)$$
$$M_z = M_{zs} + M_{zD} + F_{ys}(X) - F_{xD}(Y)$$

Stress due to forces is readily determined as $S = F/A$. Assuming these bolts are vertical, total vertical force F_y will cause a tensile stress

$$S_T = F_y / A_B$$

where A_B = *area of bolts = number of bolts times thread root area*.

The horizontal forces, F_x *and* F_z, can be combined to cause a shear stress in the bolts. This is represented by the formula:

$$S_S = (F^2_x + F^2_y)^{1/2} / A_B .$$

Moments from connecting pipes also add to bolt stresses. The moment about the X axis, M_x, must be counterbalanced by the moment from the bolt force times the distance between them. Where F_B = *total force on bolts and* L = *distance between casing support bolts on Z axis:*

$$M_X = F_B L$$

Bolt stress,

$$S_T = F_B/A_B = \frac{M_X/L}{\frac{\text{(no. bolts)}}{2} \text{ times root area}}$$

$$M_y = F_B L$$

Bolt stress,

$$S_S = F_B/A_B = \frac{M_y/L}{\frac{\text{(no. bolts)}}{2} \text{ times root area}}$$

M_z is counterbalanced by the rear support at the coupling and is assumed to have no effect on bolt strength.

Combining the effect of forces and moments, assuming a total of four hold-down bolts, and where A_R = root area of bolt threads:

Tensile stress,

$$S_T = \left(\frac{F_y}{4A_R} + \frac{M_x/L}{2A_R} \right)$$

Shear stress,

$$S_S = \frac{(F^2{}_X + F^2{}_y)^{1/2}}{4A_R} + \frac{M_y/L}{2A_R}$$

		Coupling deflection (inches)	
		Vertical	Horizontal
M_x	2,680 ft. lb.	0.000	0.001
M_y	2,040 ft. lb.	0.000	0.003
M_z	1,370 ft. lb.	0.003	0.000

Figure 5—Worthington test results on a 6x4x10 pump.

Coupling misalignment.

Like the effect on the internal distortion of the pump, the effect of nozzle loads on coupling misalignment is best determined experimentally. One such test was run and reported in an article titled "Allowable pump piping loads" in the June, 1972, issue of Hydrocarbon Processing. The specific test data presented was not clearly defined. However, according to the author, the nozzle loads specified in API-610 fifth edition are reasonable and can be withstood by a process pump with no more than 0.010 of an inch coupling misalignment.

The API-610 sixth edition allowable moments for a 6x4 pump are:

M_x = 2,680 ft. lb.;
M_y = 2,040 ft. lb.; and
M_z = 1,370 ft. lb.

Worthington conducted experimental tests on a 6x4x10 pump. The results of these tests are shown in **Figure 5.**

Although all of these numbers are well within the 0.005 allowable, the effects of M_z are most serious. To meet the API limit, a direct restraint at the coupling is necessary. The moment M_x and M_y can be restrained by the casing alone. This secondary support at the coupling causes internal distortion on pump parts and should be avoided.

In conclusion, be sure to use the guidelines suggested in API-610 for limiting the magnitude of nozzle loads on pumps. Also, whenever possible, design pipe systems and supports so that the value of M_z is minimized. □

INSTALLATION, OPERATION, AND MAINTENANCE OF COMPRESSORS, PUMPS, AND TURBINES.

As man's industrial processes grow in complexity, so do the tasks that face plant operators and maintenance personnel. In fact, they are faced with a dual problem. On one hand, this growth in complexity results in the use of more and more sophisticated equipment and in greater interaction between the different components of a plant. On the other hand, the reliability and uninterrupted service of each piece of equipment becomes more and more vital to the continued service and productivity of the entire plant. And, as plant complexity increases, so do downtime costs.

Little wonder, therefore, that so much emphasis is being placed on the proper preventive and corrective maintenance of plant equipment. Accordingly, this article addresses itself to the subject of the proper installation, operation, and maintenance of plant mechanical equipment—pumps, compressors, and steam turbines.

Proper maintenance does not start with repairs or replacement of worn parts, but right at the time of equipment selection. Operating demands to be placed on the equipment over its projected life must be adequately anticipated, and the equipment must be properly designed for the system in which it must operate.

If proper selection is important, so is adequate installation. Most of us—be we manufacturers or plant operators—have too often seen the best possible equipment fail prematurely because some fundamental precautions were neglected at the time it was installed.

And finally, good maintenance depends on good operation. All the efforts on the part of a maintenance department can be wasted if there is no equal effort on the part of production personnel to operate the equipment as it was designed to be operated.

Similar maintenance rules.

It is remarkable how little difference there is between the rules that should be followed for the proper maintenance of such different pieces of mechanical equipment as a centrifugal pump, a power or steam pump, a steam turbine, a compressor, or even an engine. It is obviously true that each one of these ma-

Table I—Selection
Advise the manufacturer of the exact nature and characteristics of the liquid/gas to be handled, including temperature range.
Check into required capacities; check required power and speed for turbines.
Analyze suction or inlet conditions.
Analyze discharge conditions.
Advise the manufacturer whether service is continuous or intermittent.
Determine what type of power is best suited for the drive.
Advise any space, weight or transportation limitations involved.
Advise any significant effect of location of installation (elevation above sea level, geographical location and immediate surroundings).
Be sure that sufficient spare or standby equipment is available.
Keep sufficient spare parts on hand.

Table II—Installation
Install equipment in light, dry and clean locations whenever possible.
Foundations should be rigid.
Bed plate should be grouted.
Equipment and driver alignment must be checked under operating conditions.
Piping should not impose excessive strains on equipment.
Use as direct piping as possible, especially at inlet.
Provide vent valves at high points for pumps, drain connections for pumps, compressors, and turbines.
Provide warm-up and by-pass connections for centrifugal pumps, relief valves for positive displacement pumps, compressors, and turbines.
Provide a suitable source of cooling water.
Install suitable gages, flowmeters and thermometers.

chines requires a different set of diagnostic instructions to determine why it may not be performing as intended. But when it comes to preventive maintenance rather than troubleshooting, it would seem that all these different machines are equal before the "Great Engineer."

As stated earlier, these fundamental rules can be broken down very readily into four separate areas: selection of the equipment, installation, operation, and maintenance. And to underline the equal importance of these four areas, a total of 40 basic rules breaks down equally into ten rules for each of the areas. These four groups of rules are presented in Tables I through IV.

Different diagnostic techniques.

There are differences in the diagnostic techniques to be applied to these three different pieces of equipment. For the purpose of providing as complete a guide as possible to the preventive and corrective maintenance of pumps, compressors, and steam turbines, we have appended Check Charts V through VII which are useful in locating the source of trouble in this machinery.

Experience shows that any equipment fares considerably better if its operator has confidence in this equipment, and if the operator has a clear understanding of how it is made, why it is made as it is, and what constitutes its proper installation, proper operation, and proper maintenance.□

Table III—Operation

Observe instruction book start-up and shut-down procedures.

Operate equipment within range of flows, pressures and temperatures specified by manufacturer.

Do not throttle suction to reduce pump capacity; throttle inlet to vary speed and power for turbines.

A pump handles liquids—keep air out; a compressor handles gases—keep water out. For turbines, avoid wet steam conditions.

Do not use excessive lubricant or excessive cooling water.

Avoid shocks from sudden temperature changes.

Make hourly observations.

Do not run equipment if excessive noise or vibration appears.

Run spare equipment occasionally to check its availability.

Set up scheduled semi-annual and annual inspection.

Table IV—Repair and Maintenance

Don't open equipment for general inspection unless diagnosis indicates the need.

Great care is required in dismantling equipment; follow instruction book procedures.

Special care is needed in examination and reconditioning of metal-to-metal fits.

Clean internal surfaces thoroughly and repaint where indicated.

Use new gaskets for complete overhaul.

Examine parts for corrosion, erosion and other damage.

Check concentricity of parts.

Restore areas subject to packing wear to proper service condition.

Exercise great care in mounting anti-friction bearings or in restoring journal bearing surfaces.

Keep a complete record of inspections and repairs.

Chart VI
Water-Cooled Compressor Troubles

Symptoms	Possible causes
Failure to deliver air	Restricted suction line.
	Dirty air filter.
	Worn or broken valve strip, loose valves.
	Defective capacity control.
Insufficient capacity	Restricted suction line.
	Dirty air filter.
	Worn or broken valve strip, loose valves.
	Defective unloaders.
	Excessive system leakage.
	Incorrect speed.
	Worn piston rings.
	Defective capacity control.
Insufficient pressure	Worn or broken valve strip, loose valves.
	Defective unloaders.
	Excessive system leakage.
	Speed incorrect.
	Worn piston rings.
	System demand exceeds compressor capacity.
	Defective capacity control.
Compressor overheats	Incorrect speed.
	Worn piston rings.
	Defective capacity control.
	Inadequate cooling water quantity.
	Excessive cooling water temperature.
	Excessive discharge pressure.
	Inadequate running gear lubrication.
Intercooler pressure below normal	Worn or broken valve strip, loose valves.
	Defective unloaders.
	Worn piston rings.
	Defective capacity control.
Compressor knocks	Worn or broken valve strip, loose valves.
	Loose unloader.
	Excessive discharge pressure.
	Inadequate running gear lubrication.
	Loose flywheel or pulley.
	Excessive bearing clearances.
	Loose piston rod unit.
	Loose crosshead shoes.
Discharge air temperature high	Defective unloaders.
	Defective capacity control.
	Inadequate cooling water quantity.
	Excessive cooling water temperature.
	Excessive discharge pressure.
	Dirty intercooler.
	Dirty cylinder jackets.
Motor fails to start	Defective unloaders.
	Defective capacity control.
	Incorrect electrical characteristics.
	Motor too small.
	Voltage abnormally low.
Motor overheats	Speed incorrect.
	Defective capacity control.
	Excessive discharge pressure.
	Inadequate running gear lubrication.
	Incorrect electrical characteristics.
	Motor too small.
	Voltage abnormally low.
	Excitation incorrect.

Chart VII

Single-Stage Steam Turbine Troubles

Symptoms	Possible causes
Lack of power	Hand nozzle valves open insufficiently. Load is greater than turbine rating. Steam pressure at the throttle is low, or the exhaust pressure is high. Some nozzles plugged. Steam strainer is obstructed.
Excessive steam consumption	Load greater than realized. Too many hand nozzle valves open, steam pressure low, or exhaust pressure too high. Steam is wet, or the superheat low. Worn or damaged nozzles and blades.
Vibration	Misalignment with driven shaft. Unbalance. Rubbing. Sprung shaft. Loose wheels. Glands fitted too tightly.
Excessive gland leakage	Badly worn or broken carbon rings. Leak-off line not freely open. Excessive back pressure. Low steam temperature.

Symptoms	Possible causes
Bearing, heating and wear	Misalignment. Unbalance. Thrust from driven shaft transmitted through coupling. Rough or untrue thrust collars. Heavy slugs of water in the stream. Inadequate lubricant.
Units do not stay in alignment	Excessive stream pipe stresses. Turbine casing supporting members are hot due to poor insulation. Foundations of driver and driven machine move.
Over/under responsive trip valves	Improper adjustment or poor condition of tripping mechanism, springs, or latches. Excessive friction in trip valve spindle packing; scaling, wear, or mechanical damages in trip valve or its supports.