Peerless Pump Company
2005 Dr. M.L. King Jr. Street, P.O. Box 7026, Indianapolis, IN 46207-7026, USA
Telephone: (317) 925-9661 Fax: (317) 924-7338
www.peerlesspump.com www.epumpdoctor.com

TECHNICAL INFORMATION
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THRUST IN VERTICAL TURBINE PUMPS

By Melvin Mann

Vertical turbine-type* pumps are being applied more and more on industrial and commercial applications for a variety of reasons. Their application is straightforward; and operation, maintenance, and servicing features recommend them in many situations. Here are some pointers to watch for in their selection.

Problems Arising from Down thrust

Single suction turbine pump impellers are subjected to hydraulic axial forces (thrust) which must be properly accounted for in pump selection and design, driver selection and design, and pump operation. Failure to do so has resulted in a wide variety of costly field problems, such as motor radial bearing failure, thrust bearing failure, mechanical seal failure, excessive column shaft vibration and bearing wear, and impellers severely rubbing in bowls. Axial forces due to thrust need not result in failures, but the problem is somewhat unique in that the pump application engineer, the pump design engineer, the driver design engineer, and user operational procedures can each contribute to the success or failure of a given installation. Problems encountered can affect either pump assembly or driver or both. However, successful application must consider not only characteristics of pump and driver, but also operational requirements. Avoidance of problems reflects true "application engineering," as well as proper design. This article will discuss the basic causes of thrust, how thrust can be calculated, and then the application of this information as related to vertical turbine pumps.

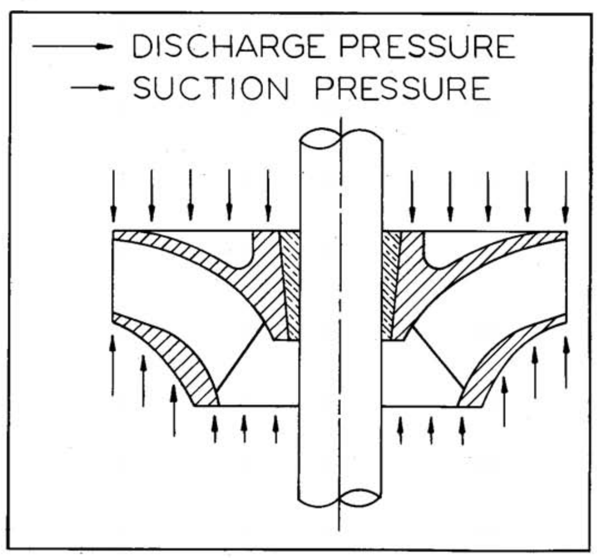

Figure 1

The forces causing thrust on a vertical turbine pump impeller can be pictured as in Figure 1.

Note that both upper and lower shrouds are subjected to discharge pressures, whereas the eye is only subjected to the suction pressure.

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*"Turbine-type lineshaft" pumps here refer only to diffuser-type centrifugal pumps-lineshaft driven.

The value of the familiar thrust constant can be approximated by the following formula:

lbs./ftr. Head = .433 (net eye area in sq. in.)

This formula is only an approximation, because the actual pressures imposed on the various areas of the impeller shrouds are not exactly equal to the discharge pressure. These pressures are influenced not only by the rotation of the impeller, but also by the external shape of the impeller and the surrounding bowl structure. Probably the most accurate way to determine thrust is by measurement in the laboratory, and most Peerless thrust data is obtained by such measurements. It is interesting to note from this formula that a large eye area will produce relatively high thrust. For example, 12" bowls show the following:

Bowl	Eye Area Sq. Inches	Downthrust in pounds per Foot of head
12 LB (Low Capacity)	14.1	6.0
12 MA (Medium Capacity)	17.9	7.5
12 HXB (High Capacity)	27.4	8.5

It should be noted that for a given diameter of bowl, the larger capacity bowl will have a bigger eye area, and therefore higher thrust constants. Another facet of this characteristic is that higher speeds for the same selection will usually reduce thrust. For example, a pump selected for 200 GPM can be a 6LB at 3460 RPM or an 8LB at 1760 RPM. The thrust constants for these bowls are 1.5 and 2.6 pounds per foot of head. Thus, for the same head, the 6LB will produce considerably less thrust.

Upthrust Also Important

In addition to the downthrust force caused by differences in pressure surrounding the impeller, there is also a force commonly known as upthrust. In the normal operating range of the pump, this upthrust (which is always present at any flow) is small in magnitude compared with the downthrust. However, at very high capacities, the actual value in pounds of upthrust force can be very much greater in value than the downthrust. Thus, the upthrust force, while always present, only becomes a factor when very high capacities occur. To help understand this more clearly, the following analysis shows how upthrust occurs.

We have all seen a garden hose whip around like a wild snake if the nozzle is released while water is flowing with a relatively high velocity at the nozzle. The forces acting on the hose are the same forces which act on an elbow or in the eye of an impeller. These forces illustrate the "impulse momentum principle," a fundamental principle of hydraulics. The forces on any elbow can be shown to act as follows:

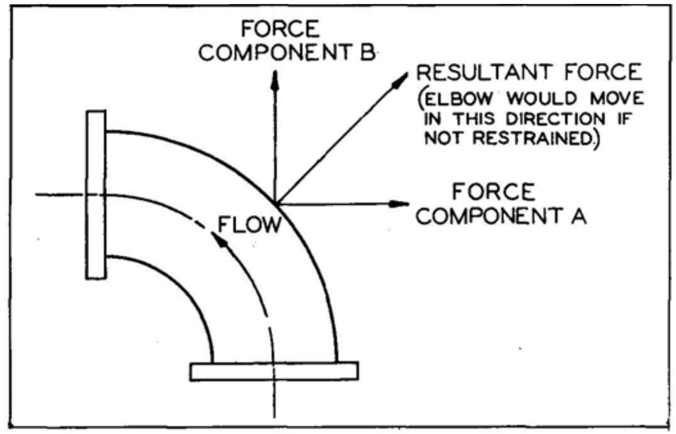

Figure 2

If the elbow is thought of as the eye of an impeller where the water turns in direction, it can be seen from Figure 3 that force component "A" will be cancelled out and the component "B" remains.

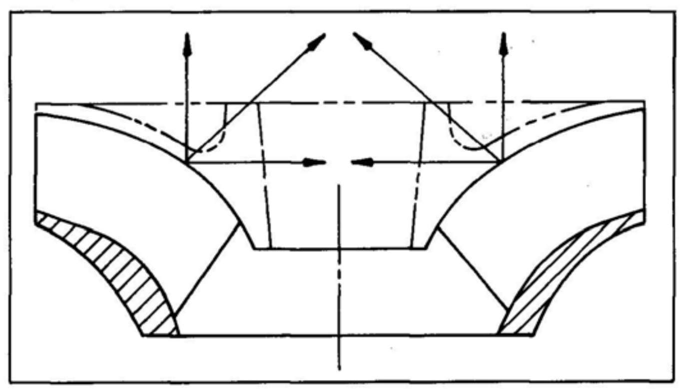

Figure 3

For an impeller with a 90° turn, the approximate value of this upthrust force (F) in pounds is:

$$F = \frac{Ve^2}{2g} \times \frac{(\text{net eye area})}{2.31}$$

Where Ve = velocity in eye of impeller
Units: V is ft./sec.
Area in sq. inches.

It can be seen that the force (F) goes up as the square of the eye velocity, which helps explain why this upthrust force reaches high values at high capacities. Although values or upthrust can be calculated for a given pump, experience indicates that laboratory test data is probably more reliable than calculated data.

A typical thrust curve is shown in Figure 4. This curve was obtained by a laboratory test in which the thrust loads were accurately measured. There are some general observations which can be made about this thrust curve, a curve that is typical of our vertical turbine pumps.

By dividing the thrust in pounds by the head in feet, we obtain the familiar value of pounds thrust per foot of head, which we publish in our Sales Manual, Note, however, that this value is by no means a constant value.

Our sales book shows a fixed value for ease of application, and is normally a vale picked slightly to the left of peak efficiency. For the 10MA example shown in Figure 4, the sales book value is 5.5 lbs./ft. head. Actually, the thrust factor of pounds per foot of head increases toward shutoff and gradually decreases at high capacities. In the case of the 10MA in Figure 4, the factor at shut off is 9 lbs./ft. head. Note that at very high capacities, the thrust actually changes from downthrust to upthrust. The shape of this curve is typical of many vertical turbine pumps. A good understanding and working knowledge of the curve shape is the best tool for avoiding any kind of field trouble due to downthrust or upthrust loads.

The typical deep well irrigation lineshaft pump probably gives the least amount of thrust troubles because the range of operation is not only narrow, but also the pump usually is operating around peak efficiency. The pump is never operated with a closed valve and the weight of the lineshaft is usually sufficient to overcome any staring upthurst problems. However, should the irrigation pump be applied where there is surface pressure (sprinkler system) or should the unit be a submersible, the installation begins to become more susceptible to thrust trouble. An irrigation pump with surface pressure can be thought of as a pump where the developed head significantly exceeds the setting. Close-coupled industrial turbine pumps more vividly illustrate this general class of application. A submersible can also be thought of as a close-coupled pump. In general each application of a vertical turbine pump, except the standard deep well turbine pump, must be carefully checked in terms of range of operation versus thrust versus driver design. By range of operation, we must include not only the running range, but also how the pump is started and stopped.

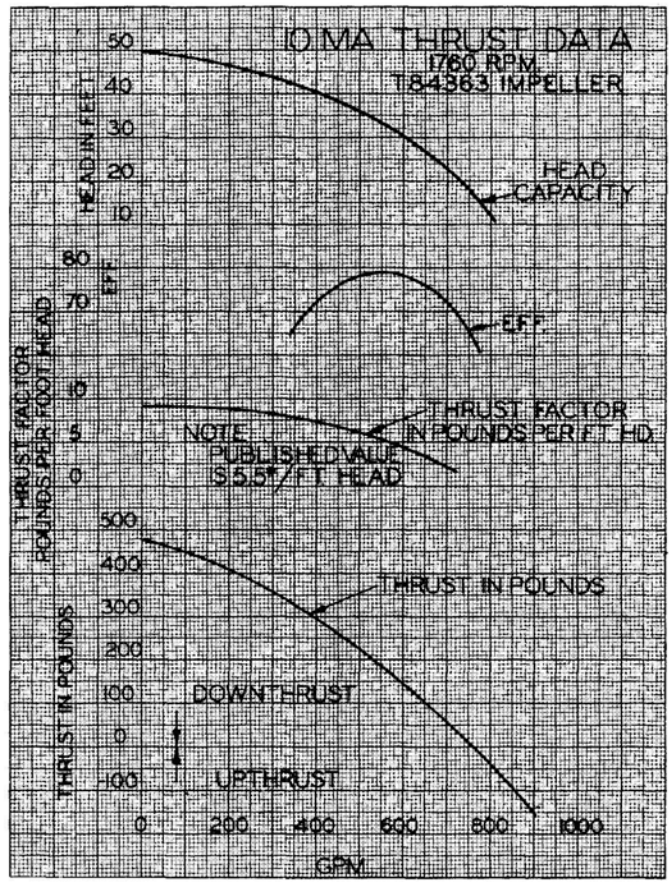

Figure 4

Start-Up Considerations

When a pump is first turned on, the pump usually operates at very high capacities, because the motor gets up to speed I just a few seconds, and it may take somewhat longer for the head to build up. The pump is therefore likely to operate in the very high capacity range where upthrust occurs. In most installation, the head builds up almost immediately, so that the upthrust is only "momentary." However, even though it is momentary, *it is present,* and the equipment must be designed to take this upthrust. If there are hundreds of feet of lineshaft, and the head equals the setting, the dead weight of the shaft will normally "absorb" the upthrust. If the unit is "close-coupled," or if it is a submersible, then the upthrust load will be transmitted to the driver and adequate provision must be made. The following rules will avoid momentary upthrust problems:

Drivers must have 30%* momentary upthrust capacity if any of the following conditions are obtained:
A. Where the setting is 75 feet or less
B. Where the head at the surface is 25% or more of the total head, regardless of setting (sprinkler irrigation deep will pumps, vertical fire pumps, industrial pumps)
C. Any submersible
* 30% refers to 30% of the downthrust capacity.

Incidentally, Peerless submersibles are protected against *momentary* upthrust (not continuous) by a special bearing arrangement in the interconnector. Also, most submersible motors have upthrust protection built into the motor. Vertical sufface motors with ball bearing design can be obtained (with no price addition) with provision for 30% momentary upthrust protection. On large vertical motors with Kingsbury-type bearings, momentary upthrust protection can be furnished, but such units are almost always special and factory quotations should be obtained.

Starting upthrust problems have caused mechanical seal malfunction. This occurs where the shaft moves upward an excessive amount, thereby changing the fine adjustment between the stationary face and the rotating face of the seal. The ability of the seal to accommodate vertical movement of the shaft varies with each seal design. However, as a general rule, no seal trouble will be encountered if the vertical shaft movement is limited to .015". In order to limit the movement, it is necessary that the end play in the driver rotating assembly be adjusted to a maximum of .015".

Where a pump is to be operated at very high capacities on a continuous basis, it is likely it will have a *continuous* upthrust. The operating condition may present unusual design problems for both the pump and the driver, and all such applications must be referred to the factory for detailed study. Where a pump operates at very high capacities, and in a region of continuous upthrust, cases have occurred where damage was evident in one or more of the following ways:

A. Lineshafts bend (buckle) due to compression load and cause vibration, and rapid bearing wear.
B. Leakage of mechanical seals due to shaft vibration and/or due to excessive axial movement (upward) of shaft.
C. Impellers rub on top of bowl.
D. Driver radial bearings undergo upthrust loads and fail rapidly.
E. Driver thrust bearings (such as angular contact) fail since they can take thrust in only one direction.
F. Motor rotor rubs against stator causing severe electrical and mechanical damage.
G. Ultimate destruction of motor and/or pump may occur due to one or a combination of the above.

As a general rule, no pump should be operated continuously at a capacity greater than 130% of the full diameter peak efficiency capacity in order to avoid continuous upthrust loads.

Downthrust Maximum at Shut-Off

Looking at the other extreme, we know that on industrial applications, it is not uncommon to operate a pump against a closed valve, resulting in very high downthrust loads. This can occur as part of start and stop procedure (pump operating in parallel, pumps on long pipe lines), or it may occur on single units in certain loading and process operations. It may even occur during a field test procedure. The thrust at shutoff is very high due not only to the higher head, but also (as previously noted) due to the increased value of the thrust factor. If the driver is equipped with an anti-friction type thrust bearing (ball bearing, spherical roller), there should be no problem even if the thrust bearing is overloaded, because such bearings are capable of high overloads for a short duration without significantly affecting the life of the bearing.

However, where the driver is equipped with a plate-type bearing (Kingsbury), then the driver thrust capacity must be designed for the shutoff thrust. Plate-type thrust bearings are used on all submersible motors, and also on large vertical motors in the range of about 1000 horsepower and larger. When related to pump thrust applications, the basic difference in the two thrust bearing types is that the plate type has no ability to sustain overloads. Since the actual thrust load at shutoff can be more than twice the load at peak efficiency, it can be seen that the driver thrust bearing must be designed for shutoff thrust where a Kingsbury-type bearing is used. Most submersible motor manufacturers solve this by providing thrust capacity greater than the published values. However, such is not necessarily true in large surface motors.

Although normal-thrust-capacity vertical motors are not ordinarily used because of their low thrust capacity, such motors are usually furnished with “deep groove” type thrust bearings, which are capable of taking some thrust on a continuous basis in *either* direction. However, the pump must be specifically designed to accommodate such an operation. The main reason for treating the pump design in a special manner is to be sure the shafting is adequately supported by bearings, because under an upthrust condition; the shaft is in compression. Bearing spacing in relation to shaft diameter and thrust load must be carefully analyzed for such a condition. We seldom recommend that a pump be operated in a continuous upthrust condition. Occasionally, such a con-dition is difficult to avoid where high suction pressure exists. This can occur in a Hydro-Line® (“can”) type unit, where the differential head is low compared to the suction pressure. Perhaps this can be best illustrated graphically (Figure 5) to show that the high suction pressure acts on the bottom of the shaft, and results in a significant upward force on the entire rotating element.

Thrust Balancing

Occasionally, it is desirable to balance-out the axial downthrust on a vertical turbine pump. This is done on close-coupled units in order to require less downthrust in the motor. In such a case, the savings in the driver cost (because of lower thrust capacity) must be compared with the added cost of balancing the thrust. Also, a hydraulically balanced pump will be less efficient (about 1.2%) than the same pump not hydraulically balanced, so that the operating power costs should also be considered.

Finally, the thrust characteristics of a pump must be carefully studied in relation to the

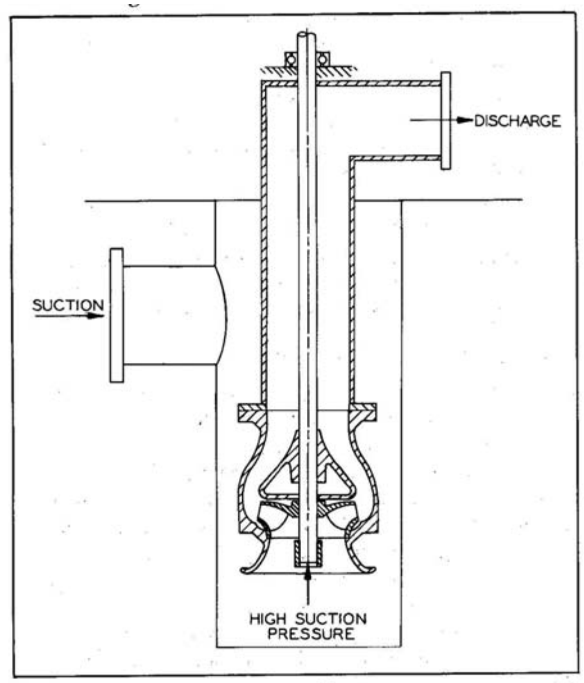

Figure 5

operation of the pump. Figure 6 shows the thrust curve in a pump before and after balancing. In this case, 75% of the thrust was balanced out at shutoff and almost 100% was balanced out at the peak efficiency capacity (550 GPM). Note that the capacity where upthrust starts is 580 GPM for the balanced impeller, and 760 GPM for the standard impeller. This change in the “zero thrust” capacity is generally true so that it becomes important to know at what maximum capacity the pump will operate, particularly if it is balanced, in order to avoid, or design for, operation in the continuous upthrust area.

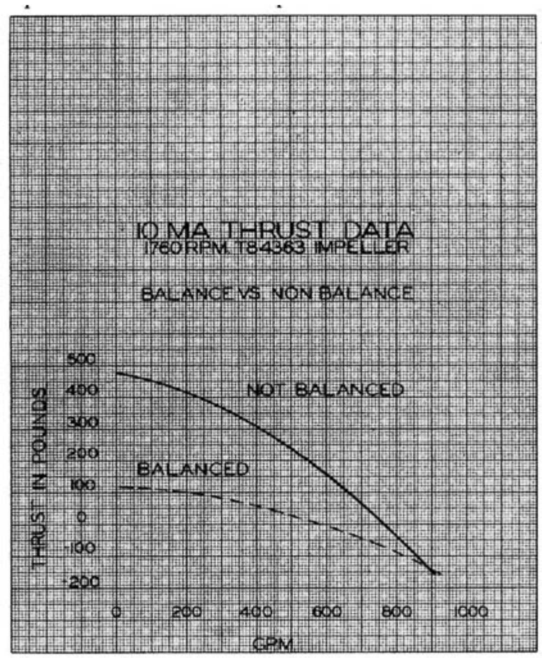

Figure 6