Peerless Pump
An Indian Head Company

TECHNICAL INFORMATION BULLETIN

NUMBER TWENTY

Axial Flow Pumps

Centrifugal pumps can be divided into three general classes. Radial flow pumps are those where the fluid enters the impeller in a direction parallel with the axis of rotation and leaves the impeller in the radial tangential plane. An axial flow pump is one in which the fluid enters parallel to the axis of rotation and leaves in the axial tangential plane. A mixed flow impeller is a type which combines the properties of a radial flow and an axial flow. In a mixed flow pump, the water enters in a direction parallel to the axis of rotation, but the discharge from the impeller has velocity components in the radial, tangential, and axial direction.

The above three types have many family characteristics in common. Certain characteristics are more dominant or important in one class of pump than the other, even though common to all. In addition to these common characteristics, there are some clear-cut and non-overlapping differences, particularly between the radial flow and the axial flow units.

Axial flow pumps are normally designed for conditions where the capacity and speed are relatively high in relation to the head developed. Quantitatively, pumps with specific speeds $\left(N_s = \frac{RPM \times \sqrt{GPM}}{H^{3/4}}\right)$ in excess of 8000 represent the nominal region of application and design of axial flow pumps. For example, a pump having a capacity of 4500 GPM against 10.5 feet total dynamic head, operating at 1160 RPM, has a specific speed of 13,300. An axial flow pump would be the most economical and efficient unit to meet these conditions. A radial flow pump to meet the same conditions would operate at very much lower speed, and the driver would most likely be larger and more costly.

An axial flow pump is essentially a high capacity low head centrifugal pump. Normally, the absolute velocities of the fluid in an axial flow pump are quite low. Typical velocities for a pump described above as having a 13,300 specific speed are 12.7 feet per second for the axial velocity at both entrance and exit with a 6 feet per second tangential component at the periphery at exit and a vane peripheral speed of 68 feet per second. Under normal application conditions, the peripheral velocities generally do not exceed 5,000 feet per minute or 83 feet per second. When operated at higher velocities, local cavitation may occur on the underside of the impeller just behind the leading edge. Most axial flow designs are so proportioned that a constant head is developed at each radius of the propeller, which results in increasing the angles of the vanes toward the hub to give this condition.

The following is a table showing a general comparison between an axial flow and a radial flow pump.

Characteristic	Axial Flow	Radial Flow
Specific Speed	Over 8000	500 — 3600
Vertical pump mechanical losses	Negligible	Significant
External velocity losses	Very important	Negligible
Sump design	Very important	Moderately important
Suction lift	Poor (High NPSH)	Good (Low NPSH)
Air Handling	Good	Poor
When pump reversed	Flow reverses	Flow same direction
Backspin speed	High	Moderate
Maximum head per stage	35 ft. (Approximate)	600 ft. (Approximate)
Head capacity curve	Steep	Flat to moderately steep
S.O. Head/Design head	2 to 3	1.1 to 1.6
S.O. HP/Design HP	1.5 to 2.5	Less than unity
HP to right of design point	Always decreases	Always increases

The relative influence of mechanical and external hydraulic losses is extremely important to recognize on vertical pump applications. This is best illustrated by comparing two specific pumps.

(a) Axial flow pump for 4500 GPM, 10.5 feet head, 1160 RPM, 80% efficiency, 14.9 bowl unit horsepower.

(b) Radial flow pump for 450 GPM, 105 feet head, 1760 RPM, 80% efficiency, 14.9 laboratory horsepower.

Assuming that both of the above pumps are driven with 1" shafting and using friction losses from published charts, the axial flow pump with 10.5 feet of shafting would have a horsepower friction loss of .033 HP or .22% of the bowl unit horsepower. This would be an insignificant correction and is normally neglected on this type of pump. The identical type of loss occurs on a radial flow unit, but assuming again that it has a length of drive shaft equivalent to the discharge head of 105 feet, the loss would be .55 HP or 3.7% of the bowl unit horsepower, which is a very significant and important amount.

A similar analysis, but with exactly the reverse results comes when the external hydraulic losses are compared on a radial flow and axial flow pump. On the axial flow pump, a loss of 1′ in head at discharge would be 9.5% of the total head and would reduce the efficiency of the overall unit by this amount. On the other hand, a loss of 1′ head on the radial flow pump would be only a .95% loss. Although the identical hydraulic losses occur in each system, the importance is much greater to the low head pump and, therefore, the design of the discharge piping is extremely important wherever axial flow pumps are concerned. (See example below.)

In axial flow pumps, the shut-off horsepower is often in excess of twice the design horsepower. For this reason, it is desirable to start an axial flow pump with the discharge valves open, thus requiring the minimum head during starting. Radial flow pumps are frequently started with the discharge valve closed, as this is the minimum horsepower requirement for this type of pump.

Most axial flow pumps operate on installations where suction lift is not required. Since it is usually necessary to submerge the impeller of an axial flow pump, most units of this hydraulic design are mounted vertically with the pumps submerged beneath the fluid they are handling. It is quite possible to mount axial flow pumps in a horizontal position, and this is frequently done in process work where high volumes and low heads are required, and ample submergence above the pump is available.

Many axial flow pumps with the shaft mounted in a vertical direction, using sleeve bearings with a single elbow on the discharge side are in service. The bearings between the motor and the pump unit can be lubricated by drip-oil lubrication, flooded oil lubrication, open lineshaft construction where the bearings are lubricated by the fluid pumped; and under extremely abrasive conditions, enclosed tubing construction where clean water is used to flush the bearings to prevent the entrance of abrasives. Some specifications request grease lubricated lineshaft bearings. They are highly recommended for installations where the pump must operate instantly, after having been dry and inoperative for extended periods. For example in underpass service and flood control applications. Grease lubricated pumps in these type services require a thorough and continuous lubrication program at all times. The following shows a tabulation of various materials which may be used in the construction of vertical axial flow pumps. Many other combinations are available, and the tabulation is merely a suggested one.

Part	Standard Const.	Mild Corrosion	Severe Corrosion
Bowl & Suction	C. I.	Bronze	Monel; stainless steel
Impeller	Bronze	SAE 63 bronze	Monel; stainless steel
Bearings	Bronze	Graphite; Teflon; bronze	Graphite; Teflon; bronze;
Shaft	416 Stainless	18-8 Stainless	Type 316; monel
Column and Elbows	Mild Steel	mild steel coated	Mild steel, with coating; stainless.

In order to minimize external losses, it is often desirable to taper the discharge line at exit.

If the piping on pumps used for flood control or drainage work is to pass over a dike, advantage of a siphon is often gained by dropping the discharge lower than the maximum point in the piping. Whenever this is done, care must be taken to maintain either sufficient velocity in the piping to sweep out the air or to provide a vacuum pump at the top of the suction piping. When siphons are used, it is important that the characteristic curve of the pump be examined to make certain that there is ample capacity at the priming condition to sweep the air out of the pipe and also that the motor and particularly the starting equipment is ample to take care of the load during priming. Provision should be made to stop the pump if, for some reason, the priming of the pump is not accomplished in a given length of time. This is one of the most important factors to consider in axial flow pump application.

On drainage service, there is a back-flow of water when the pump is stopped, if check valves or vacuum breakers are not used. If a siphon system exists in the unit, a good vacuum breaker will prevent backflow and also protect the large pipes which have thin walls in relation to the diameter of the piping. In all cases, care should be taken that large negative heads are not developed which would collapse the relatively thin wall piping. There are many types of vacuum breakers which are ordinarily used in the highest point of the discharge line; a rubber ball float type, an electrical solenoid type, and a velocity type have been tried. Usually, the most reliable is a vacuum breaker which opens when there is reverse flow in the piping and closes when the pumps are in operation. This avoids any problem of sticking of floats or failure of electrical equipment, and is relatively simple in design and operation.

Good engineering practice dictates careful attention to the above consideration, as proper application of axial flow pumps depends upon them.

RELATIVE IMPORTANCE OF VELOCITY HEAD ON AXIAL FLOW AND RADIAL FLOW PUMPS

Obviously in a low-head pump, a given velocity-head loss constitutes a larger percentage of total dynamic head than in a high-head pump. This is extremely important in the selection of drivers for axial-flow pumps, as shown in the two following examples:

$$H_v = \frac{V^2}{2g}$$

H_v = head loss due to velocity effect

V = velocity fluid ′/sec.

g = acceleration/gravity ft. per sec. per sec.

1) Assuming a 1000′ setting radial flow pump with a capacity of 100 GPM; efficiency = 80%

$$HP = \frac{1000 \times 100}{3960 \times .80} = 31.55 \text{ HP}$$

$$V = 8\ \text{′/Sec.},\ H_v = \frac{V^2}{2g} = \frac{64}{64.4} = 1' = H_v \text{ approx.}$$

$$H_v = \frac{1}{1000} / \text{T.D.H.}$$

$$HP_{hv} = \frac{1 \times 100}{3960 \times .80} = 0.03155 \text{ HP} = 1/10\% \text{ Total HP}$$

2) Assume a 10′ setting axial flow pump with a capacity of 10,000 GPM, efficiency = 80%

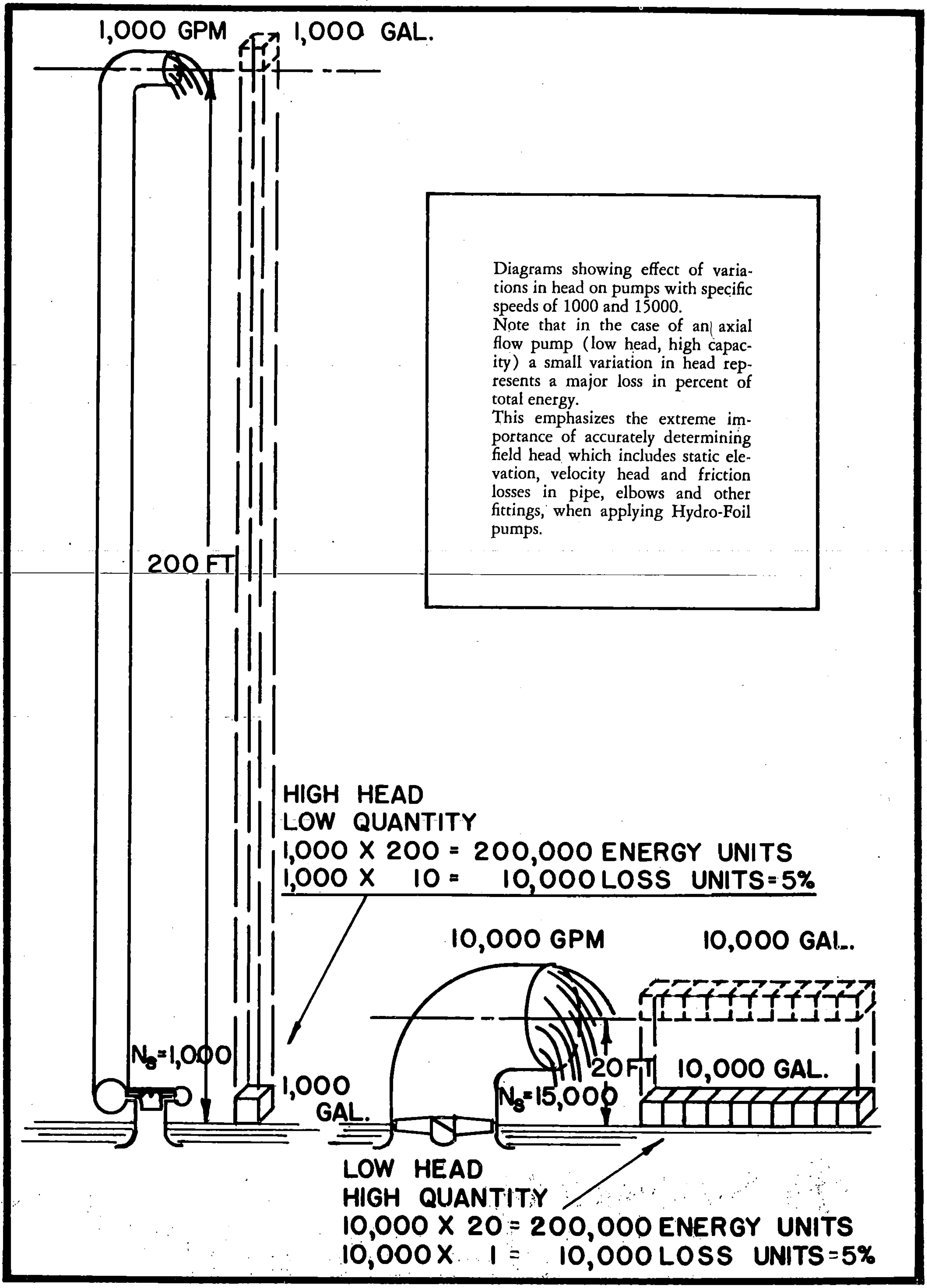
$$HP = \frac{10 \times 10000}{3960 \times .80} = 31.55 \text{ HP}$$

$$V = 8'/\text{sec.},\ H_v = \frac{V^2}{2g} = \frac{64}{64.4} = 1'\ H_v \text{ approx.}$$

$$H_v = \frac{1}{10} / \text{T.D.H.}$$

$$HP_{hv} = \frac{1 \times 10000}{3960 \times .80} = 3.155 \text{ HP} \doteq 10\% \text{ of total horsepower}$$

Hence, it is seen that accurate and complete appraisal of *all* head losses *must* be considered in higher specific speed pumps, since each element of head constitutes an important part of the total, in axial flow pumps applications.



Diagrams showing effect of variations in head on pumps with specific speeds of 1000 and 15000.
Note that in the case of an axial flow pump (low head, high capacity) a small variation in head represents a major loss in percent of total energy.
This emphasizes the extreme importance of accurately determining field head which includes static elevation, velocity head and friction losses in pipe, elbows and other fittings, when applying Hydro-Foil pumps.

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