1600 Series In-Line Pump

Taco 1600 Series In-Line Pumps combine the ultimate in reliability with ease of installation and maintenance, for heating, air conditioning, pressure boosting, cooling water transfer and water supply applications. Quiet, dependable and proven performance makes the 1600 Series Pump the right choice. Available in 50 Hz and 60 Hz.

Do your **best work.**®

Features & Benefits

Quiet, dependable power and proven performance.

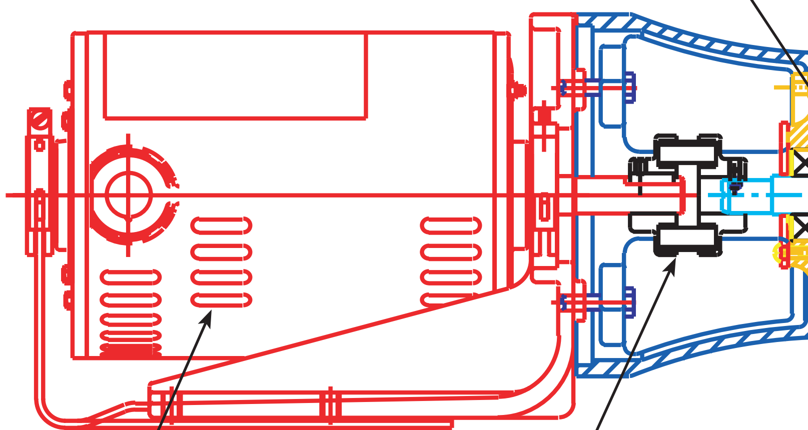
1600 Series In-Line pumps meet the latest industry standards for hydraulic performance and reliability. Each is backed by Taco Inc. a worldwide leader in the design and manufacture of heating and cooling equipment for more than eight decades. Taco "1600" Series In-Line pumps are available in 15 models ranging in size from 1 1⁄2" X 1-1⁄2" to 2" X 2" with a flow range from 10 to 235 GPM and head capacities up to 68 feet.

Taco 1600 Series In-Line pumps utilize an exclusive replaceable, cooler running bearing cartridge design. This unique design isolates the bearings from the effects of system fluid temperatures which greatly extends bearing life. The cartridge design incorporates **permanently sealed** grease lubricated ball bearings making it virtually maintenance free. All 1600 series In-Line Pumps are furnished with ceramic seals (standard) in order to meet a wide range of application requirements. The standard mechanical seal is an industry-standard Type 21 design consisting of the rotating element (SS spring & retainer, EPT elastomers, and carbon mating ring) and ceramic seat.

The standardized cartridge design simplifies maintenance with minimal parts requirements, one size cartridge and mechanical seal to service the entire product line.

Rear pull out design allows servicing of the pump without disturbing the piping.

Replaceable "cooler running" bearing cartridge extends bearing life and permanently sealed grease lubricated ball bearings makes it virtually maintenance free.

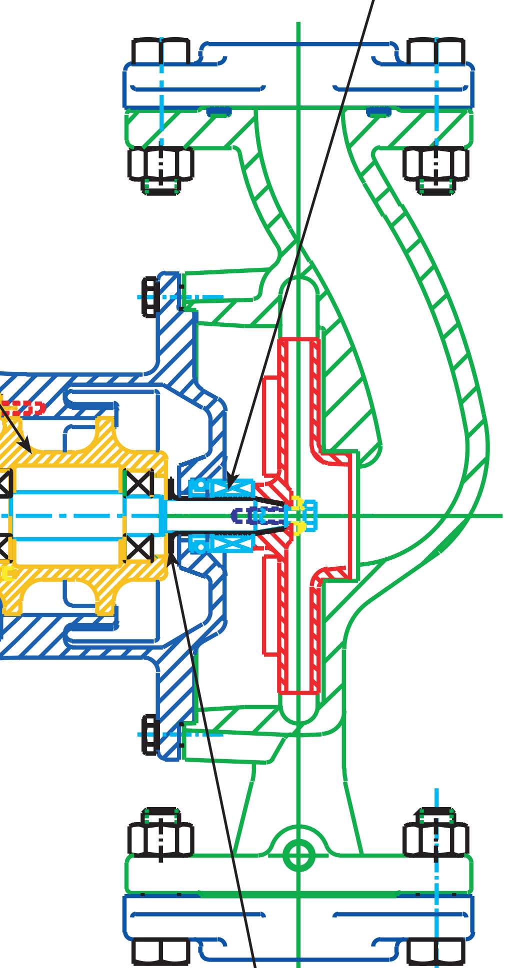


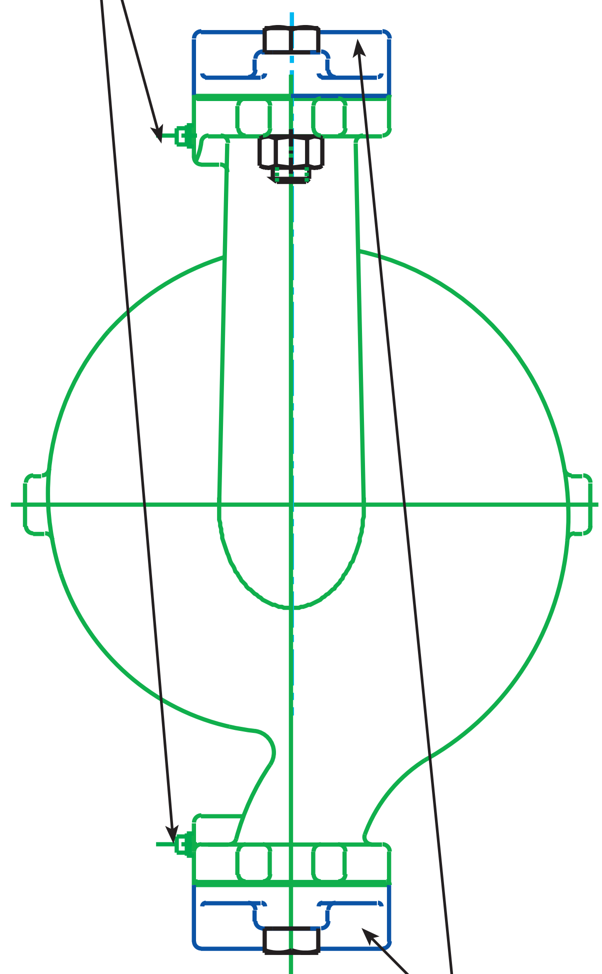
Exceptionally quiet, smooth running resilient mounted motors.

Flexible coupler absorbs shock, vibration, and misalignment that could be transmitted to the cartridge and motor bearings while also isolating and preventing any motor related noise or vibrations from being transmitted to the system.

Standard ceramic seal meets the demands of a wide range of application requirements, and the Type 21 design facilitates quick and easy replacement simplifying maintenance.

1/4 NPT pressure tappings on suction and discharge connections.





Replaceable corrosion resistant shaft sleeve incorporates a "built in" slinger to deflect water away from the bearing cartridge in the event of a seal leak.

Companion flanges included.

Useful Definitions

Flow is a volume measure to establish pump capacity per unit of time, usually as GPM.

Head is a pressure measurement represented by how high the pump can lift a column of liquid, usually in feet. To convert the popular pressure expression P.S.I. to feet of water, multiply P.S.I. X 2.31.

Horsepower (H.P.) is the amount of power available to drive the pump.

Brake Horsepower (BHP) is the amount of power required to drive the pump.

Net Positive Suction Head Required (NPSHR) is a pressure measure – in absolute units – expressed in feet, and indicates the pressure required at the pump suction to prevent cavitation. Reducing the pressure at the pump flange below the vapor pressure of the liquid can cause formation of vapor pockets in the impeller passes. This condition (cavitation) will interfere with pump performance, and is usually accompanied by noise as the vapor pockets collapse. NPSHR can be thought of as the amount of pressure in excess of vapor pressure required to prevent the formation of vapor pockets.

Net Positive Suction Head Available (NPSHR) is the pressure available at the pump suction flange. If NPSHA is less the NPSHR, cavitation problems should be expected.

Pump efficiency represents the portion of brake horsepower converted into useful work. Pump efficiency, along with flow, head, and liquid specific gravity affect the power required to drive the pump. The more efficient the pump, the less power required to drive it.

Specific Gravity (S.G.) is the relative weight of a liquid when compared with water (water = 1.0 S.G.)

R.P.M. is the rotational speed of a pump.

Shut-Off Head is the head developed by a pump at zero flow.

Static Head is the pressure at the pump discharge which the pump must overcome before it can produce flow. Static head is a difference in elevation and can be computed for a variety of conditions surrounding a pump installation.

System Resistance is the pressure on the pump discharge resulting from the resistance to flow created by friction between the fluid and the piping system. This value will vary with flow rate.

Suction Pressure is the pressure observed at the pump suction connection. This may be a positive pressure or a negative pressure.

Discharge Pressure is the pressure at the discharge connection. This will always be a positive pressure.

Differential Pressure is the algebraic difference between the discharge and suction pressures. This value represents pump head.

Service Factor is the reserve power available from an electric motor when operating under normal conditions.

System Curve is a graphical representation of the hydraulic characteristics of a piping system. When the pump performance curve is laid over the system curve, the intersection indicates the flow and head pressure of the pump when coupled to the hydraulic system.

Constant Speed is the RPM of a pump upon which a published pump curve is based.

Part I – Fundamentals

A centrifugal pump operated at constant speed delivers any capacity from zero to maximum depending on the head, design and suction conditions. Pump performance is most commonly shown by means of plotted curves which are graphical representations of a pump's performance characteristics. Pump curves present the average results obtained from testing several pumps of the same design under standardized test conditions. For a single family residential application, considerations other than flow and head are of relatively little economic or functional importance, since the total load is small and the equipment used is relatively standardized. For many smaller circulators, only the flow and pressure produced are represented on the performance curve (Fig. 1-1).

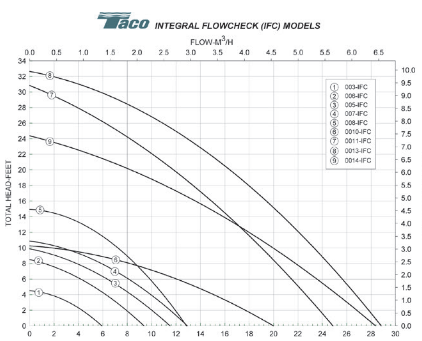


Fig. 1-1

For larger and more complex buildings and systems, economic and functional considerations are more critical, and performance curves must relate the hydraulic efficiency, the power required, the shaft speed, and the net positive suction head required in addition to the flow and pressure produced (Fig. 1-2).

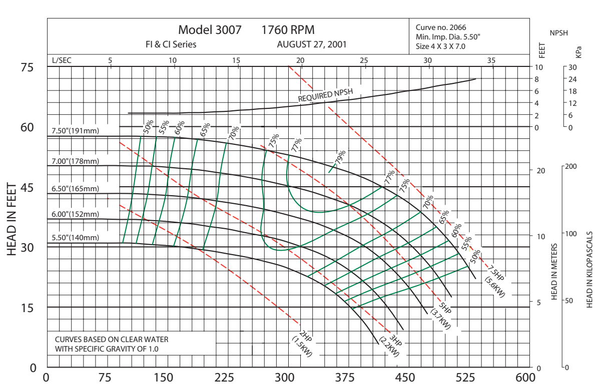


Fig. 1-2

Pump performance curves show this interrelation of pump head, flow and efficiency for a specific impeller diameter and casing size. Since impellers of more than one diameter can usually be fitted in a given pump casing, pump curves show the performance of a given pump with impellers of various diameters. Often, a complete line of pumps of one design is available and a plot called a composite or quick selection curve can be used, to give a complete picture of the available head and flow for a given pump line (Fig. 1-3).

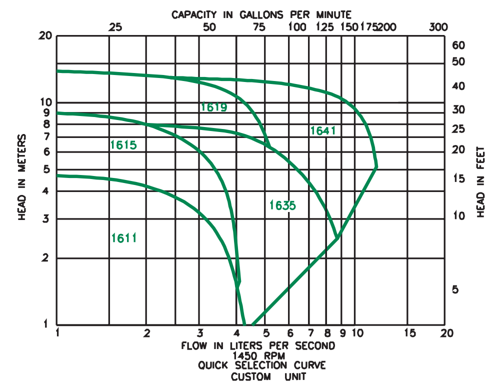


Fig. 1-3

Such charts normally give flow, head and pump size only, and the specific performance curve must then be referred to for impeller diameter, efficiency, and other details. For most applications in our industry, pump curves are based on clear water with a specific gravity of 1.0.

Part II – The System Curve

Understanding a system curve, sometimes called a system head curve, is important because conditions in larger, more complex piping systems vary as a result of either controllable or uncontrollable changes. A pump can operate at any point of rating on its performance curve, depending on the actual total head of a particular system. Partially closing a valve in the pump discharge or changing the size or length of pipes are changes in system conditions that will alter the shape of a system curve and, in turn, affect pump flow. Each pump model has a definite capacity curve for a given impeller diameter and speed. Developing a system curve provides the means to determine at what point on that curve a pump will operate when used in a particular piping system.

Commercial Hydronic Application Information

Pipes, valves and fittings create resistance to flow or friction head. Developing the data to plot a system curve for a closed Hydronic system under pressure requires calculation of the total of these friction head losses. Friction tables are readily available that provide friction loss data for pipe, valves and fittings. These tables usually express the losses in terms of the equivalent length of straight pipe of the same size as the valve or fitting. Once the total system friction is determined, a plot can be made because this friction varies roughly as the square of the liquid flow in the system. This plot represents the SYSTEM CURVE. By laying the system curve over the pump performance curve, the pump flow can be determined (Fig. 2–1).

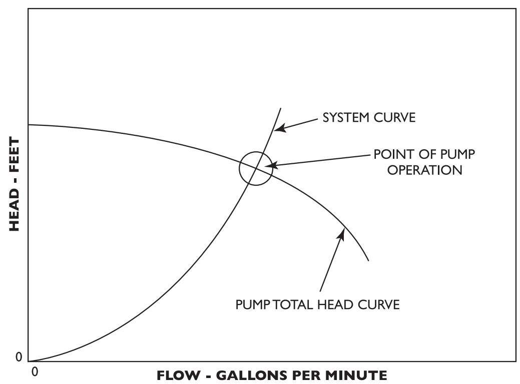


Fig. 2-1

Care must be taken that both pump head and friction are expressed in feet and that both are plotted on the same graph. The system curve will intersect the pump performance curve at the flow rate of the pump because this is the point at which the pump head is equal to the required system head for the same flow.

Fig. 2–2 illustrates the use of a discharge valve to change the system head to vary pump flow. Partially closing the valve shifts the operating point to a higher head or lower flow capacity. Opening the valve has the opposite effect. Working the system curve against the pump performance curve for different total resistance possibilities provides the system designer important information with which to make pump and motor selection decisions for each system. A system curve is also an effective tool in analyzing system performance problems and choosing appropriate corrective action.

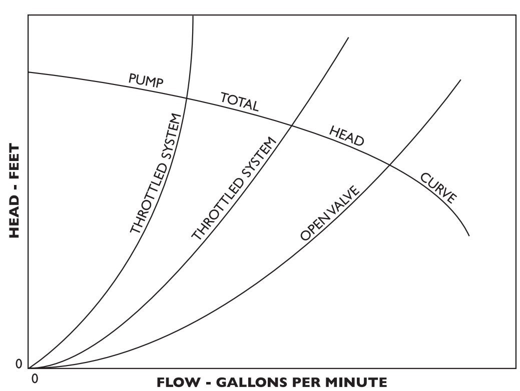


Fig. 2-2

In an open Hydronic system, it may be necessary to add head to raise the liquid from a lower level to a higher level. Called static or elevation head, this amount is added to the friction head to determine the total system head curve. Fig. 2–3 illustrates a system curve developed by adding static head to the friction head resistance.

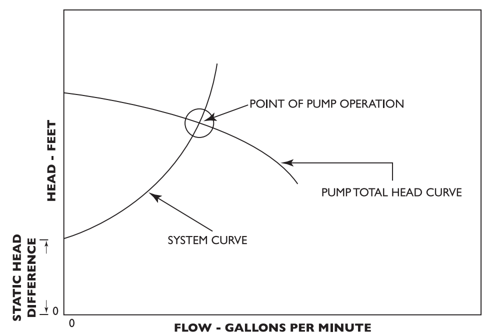


Fig. 2-3

Part III – Stable Curves, Unstable Curves and Parallel Pumping

One of the ways in which the multitude of possible performance curve shapes of centrifugal pumps can be subdivided is as stable and unstable. The head of a stable curve is highest at zero flow (shutoff) and decreases as the flow increases. This is illustrated by the curve of Pump 2 in Fig. 3 – 1.

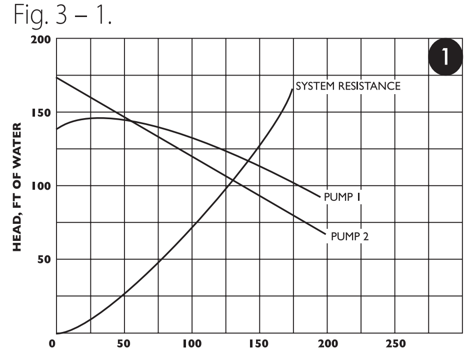


Fig. 3-1

So-called unstable curves are those with maximum head not at zero, but at 5 to 25 percent of maximum flow, as shown by the curve for Pump 1 in Fig. 3 – 1.

The term unstable, though commonly used, is rather unfortunate terminology in that it suggests unstable pump performance. Neither term refers to operating characteristic, however. Each is strictly a designation for a particular shape of curve. Both stable and unstable curves have advantages and disadvantages in design and application. It is left to the discretion of the designer to determine the shape of his curve.

In a vast majority of installations, whether the pump curve is stable or unstable is relatively unimportant, as the following examples of typical applications show.

Single Pump In Closed System

In a closed system, such as a Hydronic heating or cooling system, the function of the pump is to circulate the same quantity of fluid over and over again. Primary interest is in providing flow rate. No static head or lifting of fluid from one level to another takes place.

All system resistance curves originate at zero flow any head. Any pump, no matter how large or small, will produce some flow in a closed system.

For a given system resistance curve, the flow produced by any pump is determined by the intersection of the pump curve with the system resistance curve since only at this point is operating equilibrium possible. For each combination of system and pump, one and only one such intersection exists. Consequently, whether a pump curve is stable or unstable is of no consequence. This is illustrated in Fig. 3 –1.

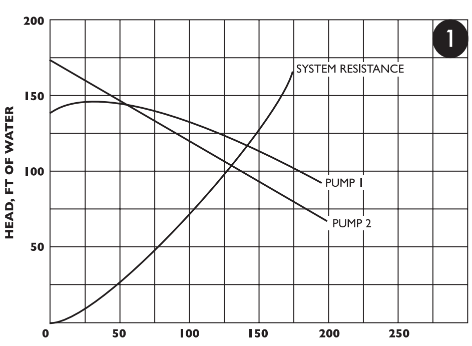


Fig. 3-1

Single Pump In Open System With Static Head

In an open system with static head, the resistance curve originates at zero flow and at the static head to be overcome. The flow is again given by the intersection of system resistance and pump curves as illustrated for a stable curve in Fig. 3–2.

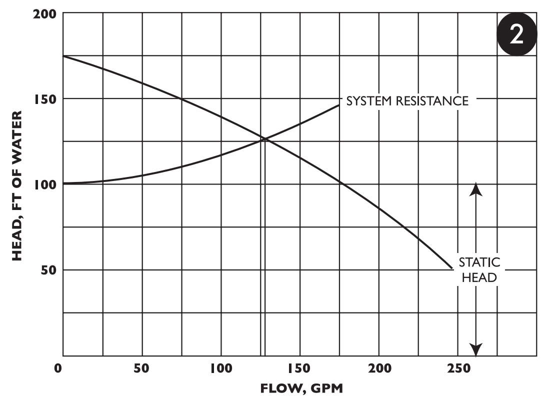


Fig. 3-2

It has been said that in an open system with static head a condition could exist where an unstable curve could cause the flow to "hunt" back and forth between two points since the system resistance curve intersects the pump curve twice, as shown in Fig. 3–3. The fallacy of this reasoning lies, in the fact that the pump used for the system in Fig. 3–3 already represents an improper selection in that it can never deliver any fluid at all. The shutoff head is lower than the static head. The explanation for this can be found in the manner in which a centrifugal pump develops its full pressure when the motor is started. The very important fact to remember here is that the shutoff head of the pump must theoretically always be at least equal to the static head.

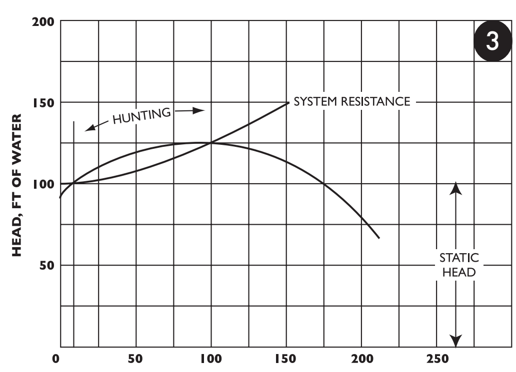


Fig. 3-3

Commercial Hydronic Application Information

From a practical point of view, the shutoff head should be 5 to 10 percent higher than the static head because the slightest reduction in pump head (such as that caused by possible impeller erosion or lower than anticipated motor speed or voltage) would again cause shutoff head to be lower than static head. If the pump is properly selected, there will be only one resistance curve intersection with the pump curve and definite, unchanging flow will be established, as shown in Fig. 3–4.

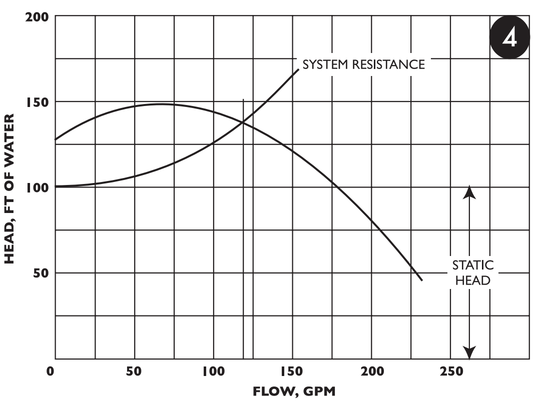


Fig. 3-4

Pumps Operating In Parallel

In more complex piping systems, two or more pumps may be arranged for parallel or series operation to meet a wide range of demand in the most economical manner. When demand drops, one or more pumps can be shut down, allowing the remaining pumps to operate at peak efficiency. Pumps operating in Parallel give multiple flow capacity against a common head. When pumps operate in series, performance is determined by adding heads at the same flow capacity. Pumps to be arranged in series or parallel require the use of a system curve in conjunction with the composite pump performance curves to evaluate their performance under various conditions.

It is sometimes heard that for multiple pumping the individual pumps used must be stable performance curves. Correctly designed installations will give trouble-free
service with either type of curve, however.

The important thing to remember is that additional pumps can be started up only when their shutoff heads are higher than the head developed by the pumps already running.

If a system with fixed resistance (no throttling devices such as modulating valves) is designed so that its head, with all pumps operating (maximum flow) is less than the shutoff head of any individual pump, the different pumps may be operated singly or in any combination, and any starting sequence will work. Fig. 3–5 shows and example consisting of two dissimilar unstable pumps operating on an open system with static head.

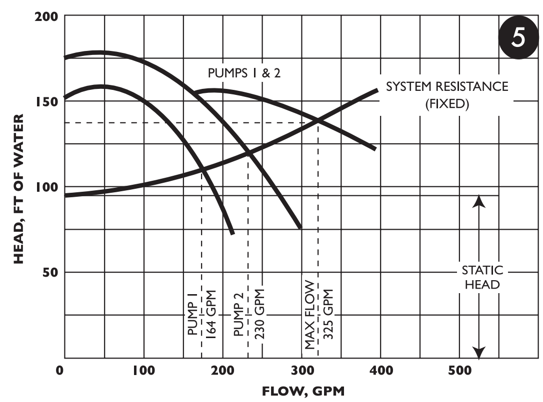


Fig. 3-5

It is also important to realize that stable curves do not guarantee successful parallel pumping by the mere fact that they are stable. Fig. 3–6 illustrates such a case. Two dissimilar pumps with stable curves are installed in a closed system with variable resistance (throttling may be affected by manually operated valves, for example).

With both pumps running, no benefit would be obtained from Pump 1 with the system resistance set to go through A, or any point between 0 and 100 GPM, for that matter. In fact, within that range, fluid from Pump 2 would flow backward through Pump 1 in spite of its running, because pressure available from Pump 2 would flow backward through Pump 1 in spite of its running, because pressure available from Pump 2 is greater than that developed by Pump 1.

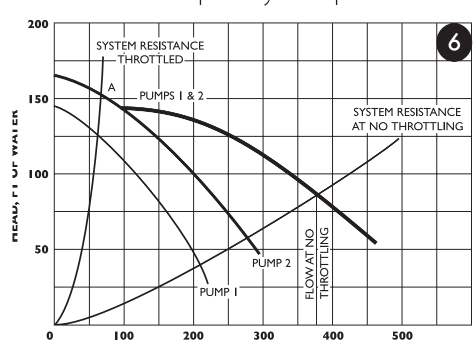


Fig. 3-6

In other words, Pump 2 overpowers Pump 1. For this reason, with Pump 2 running alone, Pump 1 should not be started unless Pump 2 operates to the right of the point where the curve of Pump 2 and the curve of Pumps 1 and 2 diverge (100 GPM) in Fig. 3–6.

Parallel pumping is often an excellent way to obtain optimum operating conditions and to save energy. To be successful, however, systems and operating conditions must be understood. This applies to both stable and unstable pump curves.

Part IV – NPSH and Pump Cavitation

The net positive suction head (NPSH) is an expression of the minimum suction conditions required to prevent cavitation in a pump. NPSH can be thought of as the head corresponding to the difference between the actual absolute pressure at the inlet to the pump impeller and the fluid vapor pressure. An incorrect determination of NPSH can lead to reduced pump capacity and efficiency, severe operating problems and cavitation damage.

It is helpful to define separately two basic NPSH considerations; required NPSH (NPSHR) and available (NPSHA).

The required or minimum NPSH is dependent on the design of a particular pump and is determined by the manufacturer's testing of each pump model. The pump manufacturer can plot this required NPSH for a given pump model on performance curve and this value, expressed as feet of the liquid handled, is the pressure required to force a given flow through the suction piping into the impeller eye of the pump. Required NPSH can also be defined as the amount of pressure in excess of the vapor pressure required by a particular pump model to prevent the formation of vapor pockets or cavitation. Required NPSH, then, varies from one pump manufacturer to the next and from one manufacturer's model to another. The required NPSH for a particular pump model varies with capacity and rapidly increases in high capacities.

The available NPSH, on the other hand, is dependent on the piping system design as well as the actual location of the pump in that system. The NPSH available as a function of system piping design must always be greater than the NPSH required by the pump in that system.

The NPSH available as a function of system piping design must always be greater than the NPSH required by the pump in that system or noise and cavitation will result. The available NPSH can be altered to satisfy the NPSH required by the pump, if changes in the piping liquid supply level, etc., can be made. Increasing the available NPSH provides a safety margin against the potential for cavitation. The available NPSH is calculated by using the formula:

NPSHA = ha +/- hs - hvpa – hf

where:

ha = atmospheric pressure in feet absolute
hs "+" = suction head or positive pressure in a closed system, expressed in feet gauge
hs "-" = suction lift or negative pressure in a closed system, expressed in feet gauge
hvpa = vapor pressure of the fluid in feet absolute
hf = pipe friction in feet between pump suction and suction reference point.

Cavitation can be defined as the formation and subsequent collapse of vapor pockets in a liquid. Cavitation in a centrifugal pump begins to occur when the suction head is insufficient to maintain pressures above the vapor pressure. As the inlet pressure approaches the flash point, vapor pockets form bubbles on the underside of the impeller vane which collapse as they move into the high-pressure area along the outer edge of the impeller. Severe cavitation can cause pitting of the impeller surface and noise levels audible outside the pump.

The Taco pump performance curve below (Fig. 4–1) includes a plot of the required NPSH for a Taco Model 1506. If a pump capacity of 105 GPM is used as an example capacity requirement, reading vertically from that GPM rate shows a required NPSH of 4 feet. An available system NPSH greater than 4 feet would, therefore, be necessary to ensure satisfactory pump performance and operation.

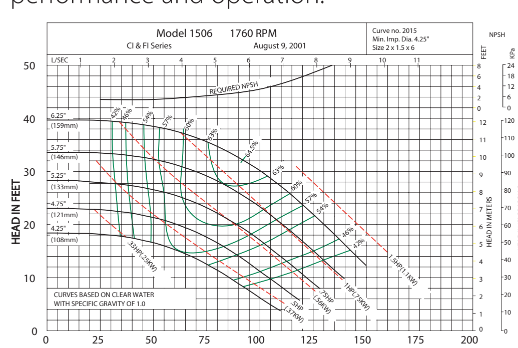


Fig. 4-1

Commercial Hydronic Application Information

Features / Benefits

Rugged Casing Design
The 1600 Series has a maximum operating pressure of 175psi, and a maximum operating temperature of 300°F. The 1600 Series is available in cast-iron stainless steel fitted construction or all stainless steel construction.

Cartridge Assembly
Replaceable "cooler running" bearing cartridge extends bearing life, and with permanently sealed greased lubricated ball bearings it is virtually maintenance free.

Pressure Tappings
Pressure tappings allow for differential pressure readings to be taken across the pump.

One Piece Enclosed Impeller
Dynamically balanced stainless steel impeller assures long life and higher pump efficiencies.

Cupro-Nickel Shaft Sleeve
Non corrosive shaft sleeve protects the shaft by preventing contact between the shaft and system fluid eliminating the need for more expensive corrosion resistant shaft materials.

Standard Mechanical Seal
1600 Series In-Line pumps utilize a Type 21 seal design which facilitates quick and easy replacement. Available in ceramic (standard) or the new "Sealide C" (for more aggressive system fluids) ensures the flexibility to meet a wide range of application requirements. One size seal fits all models.

Flexible Coupler
Flexible coupler absorbs shock, vibration and misalignment that could be transmitted to the cartridge and motor bearings while also isolating and preventing any motor related noise or vibrations from being transmitted to the system.

Resilient Mounted Motor
Resilient mounted motor insures quiet reliable vibration free operation of the pump.

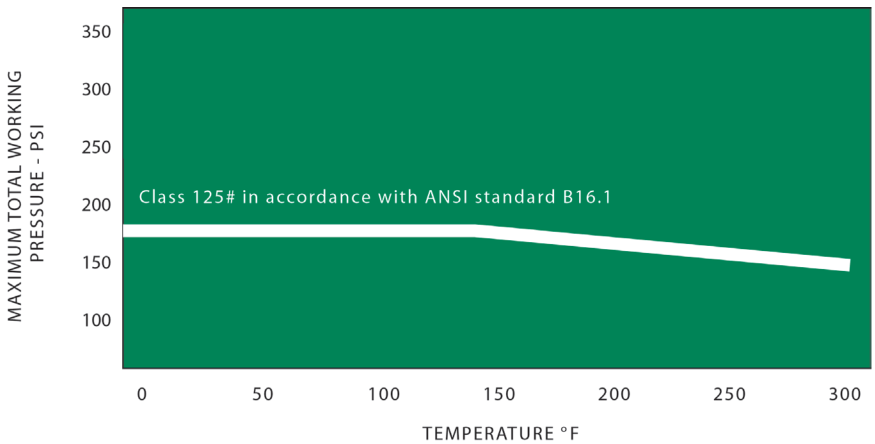
Parts Flexibility
Superior parts flexibility: one cartridge, one seal, and two motor frames fit all pump models.

Factory Tested
All 1600 series pumps are factory tested, and are built in accordance with Hydraulic Institute Standards.

Operating Specifications

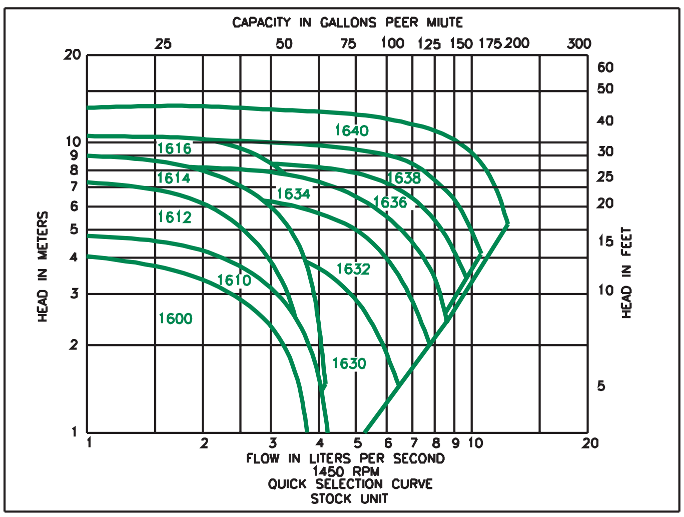
Description	Standard	Optional
Pressure	175psi Maximum Operating Pressure (125psi Flanges Standard)	---
Temperature Mechanical Seal	250°F	300°F
Motors	Nema Standard, Resilient Mounted	Special Enclosures
Coupler	Flexible Type	---
Pump Bearings	Ball Bearing Replaceable Cartridge	---
Pressure Tappings	Tapped Suction & Discharge Ports Provided as Standard	---
Pump Flanges	Available with the Pump	---

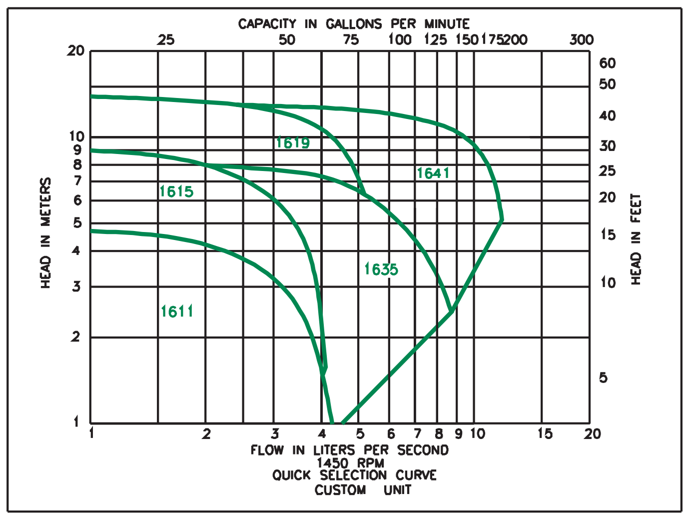
Pressure Temperature Ratings



1600 Series Performance Field 1450 RPM 50 Hz

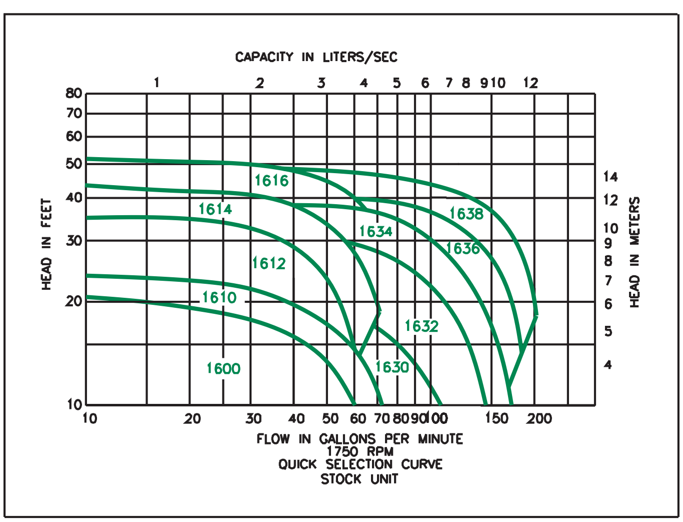
Curves also available on TacoNet®.

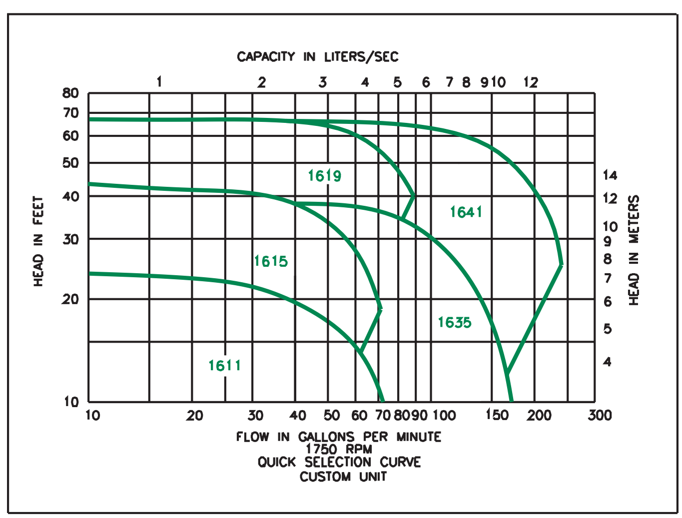




1600 Series Performance Field 1750 RPM 60 Hz

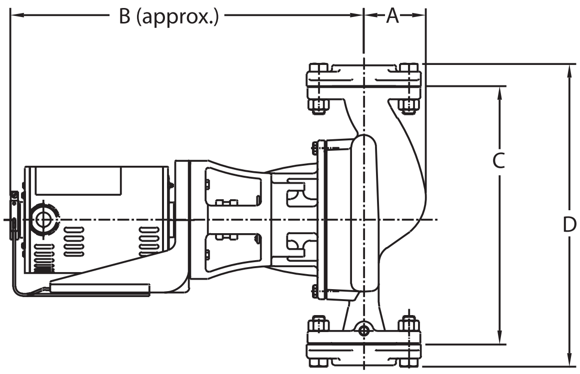
Curves also available on TacoNet®.





Applications

LoadMatch® Systems	Cooling Towers
Air Conditioning Systems	Golf Courses
Recirculation	Dry Cleaning Plants
Booster Service	Livestock Watering
Heating Systems	Bottle Washers
Laundry Equipment	Lawn Sprinklers



Materials of Construction

Description	Standard	Optional
Casing	Cast Iron	SS (304)
Impeller	Cast SS (304)	
Shaft	Hardened Alloy Steel	SS (416)
Shaft Sleeve	Cupro-Nickel	
Bracket	Cast Iron	Cast Iron with SS Face Plate

Pump Dimensions & Weights

Model No.		Flange Size	Power		Dimensions			
Custom	Stock		60Hz HP(KW)	50Hz HP(KW)	A	B	C	D
1611	1600	1-1/2 (38)	1/4* (.19)	1/6* (.17)	3 (76)	16-1/2 (419)	10-1/4 (260)	12-7/8 (327)
	1610		1/3* (.25)	1/4* (.19)	3 (76)	16-1/2 (419)		
			1/2 (.37)	1/3* (.25)	3 (76)	17.00 (432)		
1615		1-1/2 (38)	1/3* (.25)	1/4* (.19)	3-1/8 (79)	18 (457)	13-1/2 (343)	16-1/8 (410)
	1612		1/2 (.37)	1/3* (.25)	3-1/8 (79)	18-1/2 (470)		
	1614		3/4 (.56)	1/2 (.37)	3-1/8 (79)	19 (483)		
			1 (.75)	3/4 (.56)	3-1/8 (79)	19-1/2 (495)		
1619		2 (51)	3/4 (.56)	1/2 (.37)	3 (76)	18-1/2 (470)	14-1/2 (368)	17-3/8 (441)
	1616		1 (.75)	3/4 (.56)	3 (76)	19 (483)		
			1-1/2 (1.1)	1 (.75)	3 (76)	21 (533)		
			2 (1.5)	1-1/2 (1.1)	3 (76)	23 (584)		
1635	1630	2 (51)	1/2 (.37)	1/3* (.25)	3-1/2 (89)	18 (457)	13-1/2 (343)	16-1/8 (410)
	1632		3/4 (.56)	1/2 (.37)	3-1/2 (89)	18-1/2 (470)		
	1634		1 (.75)	3/4 (.56)	3-1/2 (89)	19 (483)		
			1-1/2 (1.1)	1 (.75)	3-1/2 (89)	21 (533)		
1641	1636	2 (51)	1-1/2 (1.1)	1 (.75)	3-5/8 (92)	21 (533)	16-1/2 (419)	19-1/2 (495)
	1638		2 (1.5)	1 1/2 (1.1)	3-5/8 (92)	23 (584)		
			3 (2.37)	2 (1.5)	3-5/8 (92)	24 (610)		

- English dimensions are in inches. Metric dimensions are in millimeters.
- Do not use for construction purposes unless certified.
- Metric data is presented in ( ).

* 1/4 HP and 1/3 HP AVAILABLE IN 1 PHASE ONLY.

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