UNIVERSITY OF SOUTH CAROLINA
CHEMICAL ENGINEERING LABORATORY

ECHE 460 LABORATORY PROCEDURE

ANALYSIS OF CENTRIFUGAL PUMPS AND PIPE NETWORK FLOW

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Introduction

The history of the centrifugal pump can be traced to the late 1600s. However, the modern centrifugal pump came into general use only within the past one hundred years or so. The progress of the development of the centrifugal pump essentially paralleled that of electric motors and modern steam turbines. As these drivers became bigger and more powerful, so did the centrifugal pump.

The general design and layout of the centrifugal pump has hardly changed in the last fifty years. However, improvements in containment, efficiency, reliability, and cost have been made and will continue to be made. Future developments will most likely include: the use of innovative sealing devices to reduce fugitive emissions, the use of new and innovative pump controls such as variable-speed motors and telemetry systems, the application of modern materials such as composites, ceramics and plastics, and the use of sealless pumps possibly with canned rotors and magnetic bearings. In addition, with the aid of very fast computers and modern flow visualization techniques, new and improved impeller and casing configurations are rapidly being developed.

The worldwide oil refining and petrochemical industries are the largest users of centrifugal pumps. Applications include fluids that can be (and often are) both flammable and toxic. Centrifugal pumps typically and routinely handle temperatures from –50 to over 700 ºF, pressures from vacuum to over 3000 psig, and specific gravities from as low a 0.62 to over 2.0. This wide range of pumping conditions has presented some unique challenges not only to the engineers designing the centrifugal pump (i.e., the manufacturer), but also to the engineers designing the flow system that contains the centrifugal pump (i.e., the end user).

Valves are used to regulate and throttle the flow being supplied by a pump. Many different types of valves have been designed to fit a multitude of applications. One very common type of valve is the globe valve, which has the ability to start, stop and regulate flow; not all valves can do all three tasks. Another popular valve is the ball valve, which, as the name implies, uses a ball with a bore machined through its the center to start and stop flow. When this valve is fully open, which is achieved by rotating its stem 90 degrees, the hole in the ball is in line with the flow. One advantage of this design is that there is little or no friction loss in the valve when it is fully open, since the hole is usually the same diameter as the inside diameter of the pipe. However, ball valves are not very good at regulating flow, although they can be used for this purpose.

The proper design of a piping network requires detailed knowledge of the various resistances in the system that the pump most overcome to move a fluid. These resistances are associated with the piping network itself (i.e., fittings, reducers and expanders, and entrance and exit loses), the characteristics of any valves in the network, and any elevation changes in the system. Once the design requirements for the piping network have been calculated, a pumping system is selected to meet the operational criteria of the piping network and the chemical plant, in general.

Objectives

This laboratory experience can be divided into three parts: the study of the flow characteristics of valves, the study of the performance characteristics of centrifugal pumps, and the study of an entire flow network including, valves, piping and fittings. You will first determine the valve coefficient (C_v) for a valve as a function of the valve position (*i.e.* between 0% open and 100% open). Second, you will study the performance a two different centrifugal pumps each operating alone and then in series and parallel configurations. You will compute pump characteristic curves, the work of the pumping system (which includes the pump, motor and shaft or coupling), and the corresponding efficiencies of the various components of the pumping system. You will compare all this information with the manufacturer's data. You will also calculate the actual net positive suction head (NPSH) and compare it with the required NPSH. Finally, you will perform a mechanical energy balance for the entire piping network operating through one of the centrifugal pumps. This energy balance, or theoretical system curve, will be compared with the pump curve to determine the theoretical operating point for the total system. You will then measure this operating point and compare it with the predicted one.

Design Theory

Centrifugal Pumps

The Bernoulli equation is a form of the generalized energy balance that is commonly used to analyze flow in piping networks. This equation relates all forms of hydraulic energy including static or potential energy, pressure energy, kinetic energy, frictional energy losses in piping and fittings, and pump energy. Each term in the Bernoulli equation is usually written in terms of pressure, but expressed in equivalent feet of water or *head*. The pump head (W_p) represents the work done on a unit weight of liquid in passing from the inlet or suction flange to the discharge flange. The pump must provide enough pressure head (energy) to overcome the pressure losses in the rest of the system. The pump performance is described by the extended Bernoulli equation as

$$\frac{(p_2 - p_1)}{\gamma} + (Z_2 - Z_1) + \frac{(V_2^2 - V_1^2)}{2g_c} + \left(\sum f\frac{l}{D} + \sum K\right)\frac{V^2}{2g_c} = h_p \qquad (1)$$

where p is the pressure, γ is the specific weight, V is the average liquid velocity, g is the acceleration due to gravity, Z is the elevation above or below a reference point, l is the length of tubing, D is the tubing diameter, f is the friction factor (dimensionless), h_p is the head of the pump, and K is the valve, fitting, entrance or exit coefficient. Note that in this project you <u>must use English units for all quantities</u>. For instance, pressures should be reported in psi, and lengths or heights or diameters should be reported in inches or feet. The term $(p_2 - p_1)$ is called the pressure head or flow work. The term $(V_2^2 - V_1^2)$ is called the velocity head and represents the kinetic energy of a unit weight of the liquid moving with velocity V. The term $(z_2 - z_1)$ is called the elevation head or potential head and represents the potential energy of a unit weight of liquid with respect to a reference point. The reference point on a single stage horizontal centrifugal pump is the pump shaft centerline.

The pumping system typically consists of piping, valves, fittings, and process equipment. When a particular system is being analyzed, the resistance to flow through these various components must be calculated. Fittings (elbows, tees, etc.) are rated in equivalent length of straight tubing the same size as the fitting. Valve resistances are characterized by their respective flow coefficients, i.e., C_v or K values, which are obtained from different forms of Eqn. (1) and essentially differ by a constant. The "system-head curve" or "system curve" is a plot of the total system resistance, variable plus fixed, for various flow rates. The system curve is obtained by computing and then summing all the terms on the LHS of the Bernoulli equation as a function of flow rate over the desired range of flow rates. It is not measured experimentally! The system curve is usually plotted in terms of feet of head versus flow rate, similarly to a pump curve.

A pump curve is a plot of the amount of head (or pressure) a pump can develop as a function of flow rate. Naturally, a properly designed pump must provide sufficient head to the system to attain the desired flow rate. What this means is that the pump curve must be consistent with or intersect the system curve. You will experimentally measure four pump curves in this laboratory experience and compare the results with the pump curves provided by the supplier. For one of the pumps, you will compute the system curve and determine where it intersects the pump curve. This theoretical operating point will then be compared to the experimentally measured operating point of this system.

Cavitation is defined as the formation and subsequent collapse of vapor-filled bubbles in a liquid usually at the eye of the impeller. Cavitation occurs when the inlet pressure to the pump is below the vapor pressure of the fluid. NPSH is a measure of the minimum suction conditions (i.e., inlet pressure head) required to prevent cavitation in a pump. The available or actual NPSH must be at least equal to (but necessarily greater) than the required NPSH to avoid cavitation. An actual NPSH that is substantially greater that the required NPSH provides a margin of safety against the onset of cavitation, which can severely damage the pump. The actual NPSH is calculated from

$$NPSH_{ACT} = \frac{p_t - p_{vp}}{\gamma} + Z - h_f \tag{2}$$

where p_t is the pressure on free surface of liquid in the tank, p_{vp} is the vapor pressure of the liquid at the pumping temperature, γ is the specific weight of the liquid at the pumping temperature, Z is the height from centerline of pump to the surface of water in the tank, and h_f is the frictional head loss from the tank outlet to the pump suction.

Technically speaking, a "pump" is just the impeller and housing (or volute) that transfers energy to the fluid. In order for the pump to work, two or more components are needed: an electric motor drive, and a mechanical coupling between the motor and the pump. Typically, the term pump is used loosely to refer to the entire system including the motor, coupling and pump, but this is technically incorrect. When specifying a large pumping system, the engineer must design the electric motor and electrical system to be compatible with the pump. In order to quantify the performance of the pump, coupling, and motor, three different efficiencies are used and defined below.

Pump power (or water power) is the rate of useful work done by the pump; *i.e.* it is the power imparted to the fluid. The pump power is calculated from the following dimensional formula in terms of work horsepower (*whp*):

$$whp = \frac{gpm \times head \times S.G.}{3960} \tag{4}$$

The units of *whp* are in Hp. The pump power is always less that the actual electric power consumed by the pump motor because of inefficiencies and losses. The electric power input to the motor gives the total power input to the pumping system. The electric power is the product of the voltage and the current input to the motor and is calculated from Eq. (5) in terms of electric horsepower (*ehp*).

$$ehp = \frac{voltage \times current}{746} \tag{5}$$

The units of *ehp* are also in Hp. The overall efficiency (η_{OA}) is the ratio of the energy imparted to the liquid by the pump (*whp*) to the energy supplied to the motor (*ehp*), and is expressed in percent. This efficiency takes into account all inefficiencies and losses in the pump, electrical motor, and mechanical couplings and is calculated from:

$$\eta_{OA} = \frac{whp}{ehp} \times 100 \tag{6}$$

The pump efficiency (η_p) is the ratio of the energy imparted to the liquid by the pump (*whp*) to the energy supplied to the coupling imput shaft, *i.e.*, the ratio of the liquid horsepower to the brake horsepower (*bhp*) expressed in percent. The brake horsepower is the power delivered to the mechanical coupling, and is smaller than the electric power input to the motor due to electrical inefficiencies in the motor. The pump efficiency η_p is thus a measure of the efficiency of the design of the pump hydraulic internals, and is given by Eq. (7).

$$\eta_p = \frac{whp}{bhp} \times 100 \tag{7}$$

The efficiency of the motor (η_m) is the ratio of the brake horsepower to the electrical input (horsepower) to the driver (or coupling) expressed in percent as

$$\eta_m = \frac{bhp}{ehp} \times 100 \tag{8}$$

Valve Coefficients

Valve coefficients (C_v) are an industry wide measure of valve size (*i.e.* ability of the valve to accommodate flow). For liquid service, C_v is the measure of the number of gallons per minute of 60 ^oF water that will flow through a valve at a 1 psi pressure differential across the valve. The maximum C_v is widely accepted as the measure of the maximum valve size. For a variable-position valve, C_v varies as the valve is opened or closed. If the C_v is known, the liquid flow rate through a valve can be calculated at any pressure differential using

$$Q = C_v \sqrt{\frac{\Delta P}{S.G.}} \tag{9}$$

where Q is the flow rate in gpm, S.G. is the specific gravity of fluid, ΔP is the pressure drop across valve in psi. In this laboratory experiment, the C_v of a valve with be measured and compared with that provided by the manufacturer. The K value of a different valve, used in the piping network to mimic the resistance of a chemical process, will also be determined and used in Eq. (1) to calculate the system curve. Recall that a K value is similar to a C_v value, as mentioned above. It fact, it is possible to derive Eq. (9) from Eq. (1) and determine a relationship between C_v and K.

Equipment and Supplies

Centrifugal Pump Experimental Module

Major pieces of equipment in this system include two different centrifugal pumps from Little Giant Pump Co. Refer to manufacturer's data for more information (attached). This system also has one 55 gal plastic drum. Numerous 4.5 in. process pressure gauges, current and voltage meters form Omega Inst. Co., and two digital flow meters from G.P.I. Co. are also included. Minor pieces of equipment in this system include numerous fittings and valves. A tape measure and micrometer are also needed and included.

Operational Procedure

Initial Conditions

Valves V-1, V-2, V-3, V-4, V-6, V-7 and V-8 are closed. The flow control valve V-5 can be in any position. All the pumps are turned off. The 55 gallon drum is filled to the 50 gallon mark with water. It is important to review the calculation requirements before you begin the experiment. There is certain data or information that must be recorded in order for you to complete the calculations correctly.

Part I: Test Valve C_v Measurement

1. Align the valves in the system so water flows from Pump #2 through the test valve and back to the tank.
2. Close the test valve completely. Then begin to open it carefully and record how many complete and then fractional turns it takes to go from being fully closed to

being fully opened. It is best to do this in terms of degrees, noting that one full turn is 360^o. Finally, convert the number of turns into % of fully open.

3. Close the test valve, open V-5 fully, and then start pump #2.
4. With the test valve fully closed, record the ΔP across the test valve and corresponding flow rate as a function of degrees open (i.e. valve position in % of full open).
5. Open the test valve in increments of 5% of full open and record the ΔP across the test valve and corresponding flow rate. Continue until the valve is 50% open.
6. Once at 50%, continue to open the test valve now in increments of 10% of full open and record the ΔP across the test valve and corresponding flow rate. Continue until the valve is 100% open.
7. Turn pump # 2 off, and close all valves opened in step 1.

Part II. Pump Characteristic Curve Measurement

1. Align flow from the 55-gallon drum through pump #1 and back to the 55-gallon drum while bypassing the chemical process.
2. Close V-5 and start the pump(s). Record all necessary information, including pressures, flows, voltages and currents. This first point gives the "shutoff head" of the pump (*i.e.*, no flow).
3. Obtain the next point at the lowest flow rate achievable with the flow meter. This is accomplished by opening V-5 approximately 1/4 turn, and then slowly closing the valve until the flow decreases to the minimum possible value. You might have to do this several times to get a feel for the response of the valve. Record all necessary information.
4. After obtaining this minimum flow rate point, increase the flow rate in approximately 1/2 gpm increments until V-5 is fully open. Record all necessary information. This should give you approximately 20 data points.
5. Perform steps 1 to 4 for pump #2.
6. Perform steps 1 to 4 for pumps 1 and 2 configured in parallel (voltages and currents do not need to be recorded).
7. Perform steps 1 to 4 for pumps 1 and 2 configured in series (voltages and currents do not need to be recorded).

Part III. Chemical Process Resistance and System Operating Point Measurements

A valve of unknown resistance is used here to simulate the resistance of a more complicated chemical process. You will obtain sufficient data to determine the resistance of this valve in terms of a K value; and then, at the same time, you will obtain information to determine the operating point of the system under these conditions.

1. Align flow from the 55-gallon drum through pump #2, then through the chemical process valve, and finally back to the 55-gallon tank.
2. Open the chemical process valve ¼ turn and open V-5 fully.
3. Start pump # 2. For the chemical process resistance determination, record the ΔP across the valve. For the operating point determination, record the ΔP across pump #2. Finally, record the flow rate, which is needed for both determinations.

4. Repeat step 3 with the chemical process valve ½ open.

Calculations and Required Graphs and Deliverables

Valve Coefficients

1. For the test valve, determine the C_v as a function of valve position in % and compare your results with the manufacturer's data (provided).
2. Make a graph of the results in terms of C_v versus % of full open. Discuss any differences between the two curves.

Pump Characteristic Curves

1. For each recorded flow rate do the following: Calculate the pump head in feet of water for pump #1, pump #2 and for the series and parallel configurations of the two pumps. Tabulate your results. Consider why the static and dynamic heads associated with the locations of the gauges and pressure taps in the lines, in relation to the impeller inlet (suction) and outlet (discharge), can be neglected when calculating the pump head.
2. For pumps #1 and #2 only, calculate the pump power in terms of work horsepower (*whp*) and the electrical horsepower (*ehp*). Then, calculate the overall, motor, and pump efficiencies in %. Finally, calculate the actual NPSH. Tabulate your results. Note that brake horsepower is obtained from the manufacturer's data sheet.
3. On the same graph, plot the pump head in feet of water vs. flow rate in gpm for pumps #1 and #2. Plot the manufacturer's pump curves on this same graph. Discuss any differences between the two curves.
4. On another graph, plot the pump head in feet of water vs. flow rate in gpm for each pump and for the series and parallel configurations. Discuss the differences between the two curves, especially those for the series and parallel configurations.
5. For pumps #1 and #2 only, make a graph of the pump power associated with each pump in horsepower (hp) versus flow rate in gpm. Plot the corresponding manufacturer's data on this same graph. Discuss the differences between the two curves.
6. For pumps #1 and #2 only, make a graph of the overall (η_{OA}), motor (η_m), and pump efficiencies (η_p) in % vs. flow rate in gpm. Discuss the differences between the two curves.
7. For pumps #1 and #2 only, make a graph of the pump efficiencies (η_p) in % vs. flow rate in gpm. Compare your results with the manufacturer's data, noting that the efficiencies reported by the manufacturer on the data sheets are pump efficiencies. Discuss any differences between the two curves.
8. Make a graph of the actual NPSH for each pump. On the same graph, plot the manufacturer's data on the required NPSH. Discuss any differences between the two curves.

System Curve and System Operating Point

This calculation is based on flow from the tank, through pump # 2, through the chemical process valve and then back to the tank. Since you took measurements at two different settings of the chemical process valve, you must calculate two different system curves, one for each setting. However, the difference between these two system curves manifests only from the difference in the K value of the chemical process valve, which you calculate for each setting.

Note that tubing size is reported in terms of the outside diameter. All calculations with tubing must be based on the inside diameter. This means that 5/8" tubing is 5/8" in diameter on the outside, but for the calculations, you must use the inside diameter of ½". Valves and flow meters are sized by a nominal size. For valves, this size can usually be found stamped on the outside of the valve body, e.g., V-5 is a 1" globe valve. The flow meter size can be found in the manufacturer's technical manual, and the chemical process simulation valve is a ¾" metering valve, but this bit of information is unnecessary as you will calculate the K value for this valve.

Constants you will need include: density of water (ρ), dynamic viscosity of water (μ.), kinematic viscosity of water (ν), roughness factor for smooth copper tubing (ε), and gravitational constant (g_c). K values you will need include those for entrance and exit losses, ½" tubing to 7/8" tubing expander (pump outlet fitting), 1" globe valve with beveled disc, full open (V-5), 7/8" tubing to 5/8" tubing reducer, 5/8" tubing to ¾ " tubing expander, chemical process simulation valve (calculated), ¾" tubing to 5/8" tubing reducer, 5/8" tubing to 7/8" tubing expander, turbine flow meter, swing check valve, and ball valves.

1. From the measured pressure differential across the chemical process valve, calculate the K-values corresponding to the two different positions of the valve.
2. Calculate the cross sectional areas for the different components and tubing. These calculations are based on internal diameter for tubing and nominal size for a component. For example, since the system flow meter is of 1" nominal size, its cross sectional area is 0.00545 ft^2.
3. Look up the needed K-values in the reference literature. For instance, V-5 is a 1" globe valve with a K value that can be found in Perry's Chemical Engineers Handbook, 7th Edition, Table 6-4.
4. In a spreadsheet, range the values of water flow rate from 0.1 to 10 gpm in increments of 0.2 gpm.
5. Calculate water flow rate in ft^3/sec.
6. Calculate the velocity [ft/sec] for each size tubing and component.
7. Calculate the Reynolds number for each size tubing and component.
8. Evaluate the Reynolds number for each size tubing and component in terms of laminar or turbulent flow regimes. The K-values used are for turbulent flow only.
9. Evaluate the friction factor for each size tubing by:
10. Evaluating the roughness factor [ε] for the tubing inside diameter.
11. Calculating ε/d.
12. Using the Moody Chart, and obtaining values of the friction factor for each point in your spreadsheet, or recognize that the values of the friction factor are very close for

each tubing size and vary linearly within the range of the Reynolds number. Enter the high and low values of the friction factor in the spread sheet and perform linear regression to obtain an equation to calculate the values of the Friction Factor for a given Reynolds number or just enter these values manually into your spreadsheet.

13. Calculate the head loss for each tubing size.
14. Calculate the head loss for each component using its K-value. Recall that the chemical process valve has two different K-values, which gives rise to two different system curves.
15. For each position of the chemical process valve, sum all the head losses for each water flow rate and tabulate the results.
16. From the measured pressure differentials across pump #2, obtained in Part III for the two different chemical process valve positions, calculate the operating point in terms of pump head in feet of water and tabulate these values with the corresponding flow rates in gpm.
17. For pump #2 only, plot the head in feet of water vs. flow rate in gpm. On this same graph, plot both theoretical system curves in terms of total head loss in feet of water vs. water flow rate in gpm. Also, on this same graph, plot both experimentally measured operating points (calculated in step 13) in terms of pump head in feet of water vs. water flow rate in gpm. Discuss the intersections of the pump curve with the two theoretical system curves, and the two experimental operating points.

Additional Suggestions for Critical Thinking and Discussion

The oral and/or written reports should include the following considerations, organized appropriately:

1. A description of the major components of a centrifugal pump with a discussion of the function that each part performs.
2. Compare and contrast static head, static lift, and discharge head.
3. What effect does the magnetic drive have upon the pump efficiency?
4. What would happen if the actual NPSH fell below the NPSH required for the pump?
5. Is the required NPSH the same for both pumps?
6. How does the fluid temperature affect the NPSH?
7. How are the values of C_v and K related for the 1” globe valve, V-5? What if any are the units associated with these constants?
8. Will the system operating point ever fall off the pump curve? Explain.

References

1. Walas, S. M., *Chemical Process Equipment, Selection and Design,* Butterworth Heinemann, Boston (1990).
2. Karassik, I.J., *Pump Handbook*, McGraw-Hill Book Company, New York, (1986).
3. Perry’s *Chemical Engineer’s Handbook,* 7th Edition, Table 6-4., McGraw-Hill Book Company, New York (1999).
4. Mark’s *Standard Handbook for Mechanical Engineer’s*, 10th Edition, McGraw-Hill Book Company, Boston (1996).

Manuals Located at the Experimental Module

1. Operator’s Manual for “Omega” DP-25-VRMS/CRMS True RMS Meter
2. Operator’s Manual for “GPI. Electronic Digital Meter.

Manufacturer’s Data

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flow data
Cv Values for Valves

								VALVE SIZE							
Fig. Nos.	⅛	¼	⅜	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6
GATES															
S & T-22	.5	2	4.9	9.1	22	40	65	95	175						
S & T-180	–	5.6	10.7	17.6	32	50	95	130	220						
S & T-111-113-131-133 134-135-136-174-176	–	5.6	10.7	17.6	32	54	97	135	230	337	536	710	960	1,525	2,250
(T & F-617-619-667-669 607-609) (CS-102-103 302-303-602-603) (F-637-639-DI-102)									215	335	510	710	945	1,525	2,250
.OBES															
S & $\boxed{\text{T-211 (BWY)}}$-235Y 275Y	.61	1.16	2.2	3.64	6.65	11.1	20	28	48	70	111	–	198		
T-275-B	–	1.16	2.21	3.64	6.65	11.1	20	28	48	70	111				
F-718-(CS-132-133 332-333-632-633 (738)									45	70	105	–	195	315	465
CHECKS															
S & T-413-433-473 (Swing)	–	1.3	2.5	4.8	14.3	24	43	60	102	150	238	315	435	675	1,000
S & T-480 (Poppet)	–	–	3.70	6.86	16.3	30	49	72	130						
F-908 (Swing)									150	243	356	–	665	1,073	1,584
T & F-918-968-938 (Swing)									137	221	327	–	605	975	1,440
W-900-W (Wafer)									48	77	135	–	270	450	720
F-910-960 (Poppet)											155	–	278	350	625
W-910-960 (Poppet)									66	88	130	–	228	350	620

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Liquid Flow:

$$Q = C_v \sqrt{\frac{\Delta P}{S}} \quad \text{or} \quad \Delta P = S\left(\frac{Q}{C_v}\right)^2$$

where... Q = flow rate (gallons per minute)
ΔP = pressure drop across valve (psi)
S = specific gravity of media

This equation is good for turbulent flow and for liquids with viscosities near that of water.
(Cv is defined as the flow in GPM that a valve will carry with a pressure drop of 1.0 psi when the media is water at 60°F.) (The specific gravity of water is 1 (one).)

Gas Flow:

$$Q = 1360\, C_v \sqrt{\frac{\Delta P \times P_1}{ST}}$$

where... Q = gas flow (SCFH—std. cu. ft/hr)
S = specific gravity of gas (air = 1.0)
T = temp—degrees Rankine (°F + 460)
ΔP = pressure drop across valve (psi)
P_1 = upstream pressure (psia) absolute

NOTE: ΔP must be less than .5 P_1. (Flow is critical when ΔP is greater than .5 P_1.)

(THROTTLING FACTORS)
For throttling use with disc partially open. Multiply Cv by factor.
Note: Gate valves are not throttled.

								Percent Open										
8	**10**	**12**	**14**	**16**	**18**	**20**	**24**	**0**	**10**	**20**	**30**	**40**	**50**	**60**	**70**	**80**	**90**	**100**
'50	6,700	9,925	13,800	18,375	23,600	29,600	43,570											
								0	.35	.65	.90	.93	.96	.98	.99	1.00	1.00	1.00
								0	.03	.035	.06	.1	.16	.24	.32	.47	.68	1.00
860	1,390							0	.35	.65	.90	.93	.96	.98	.99	1.00	1.00	1.00

2,937	4,730	6,985					
2,670	4,300	6,350					
1,450	2,650	4,100	5,300	8,000	11,000	14,000	25,000
1,115	1,770	2,500	3,280	4,150	5,175	6,400	9,300
900	1,450						

WARNING
The Fluid Flow factors contained herein are calculated values. They are therefore approximations and cannot be used for highly critical flow or pressure drop calculations.
For very precise flow measurements, tests must be conducted on any valve mentioned within this catalog.

LITTLE GIANT

LITTLE GIANT® MAGNETIC DRIVE CENTRIFUGAL PUMPS

Attn: Chuck
803-777-8265
From: Gary

MD-SC Series Model 3-MD-SC

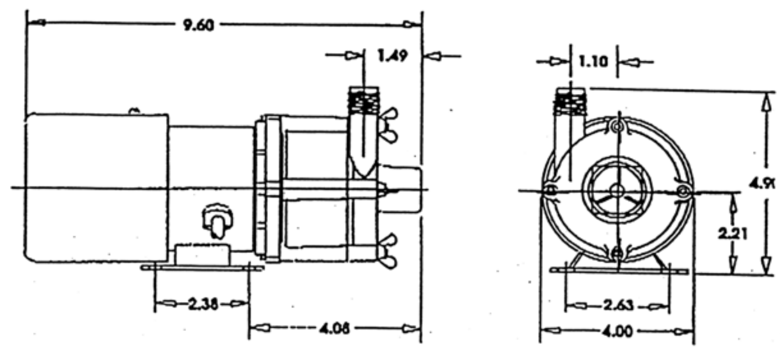
Description:

Thermally protected open fan-cooled motor with ball bearings. 6' power cord with 3-prong molded plug. (230V version has no plug.)

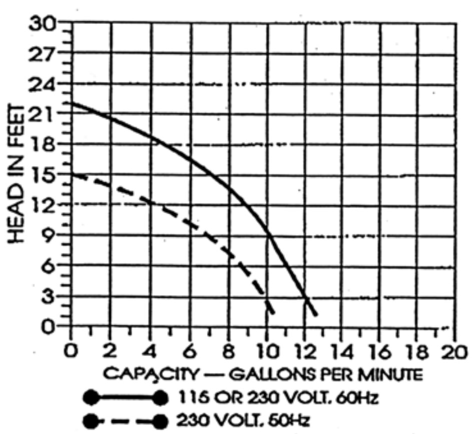
Fluid System Design Parameters:

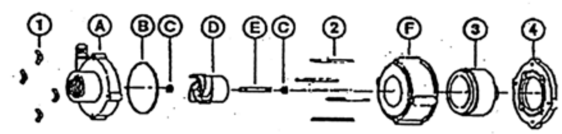
- Specific Gravity - Up to 1.1.
- Ambient Air Temperature - Up to 77°F (25°C).
- Fluid Temperature - Up to 150°F (66°C).

NOTE: Consult your local distributor or the factory for applications with higher ambient temperatures, specific gravities, and viscosities.



NOTE: Designs and dimensions may vary for various reasons (i.e. type of motor). This information should be used as general guide rather than an unqualified guarantee. Specifications are subject to change without prior notice.





Wetted Parts	Materials of Construction
A	Glass-filled Polypropylene Volute
B	Buna-N® (Nitrile) O-ring
C	Ceramic Thrust Washers
D	Glass-filled Polypropylene Impeller Assembly
E	Ceramic Shaft
F	Glass-filled Polypropylene Magnet Housing

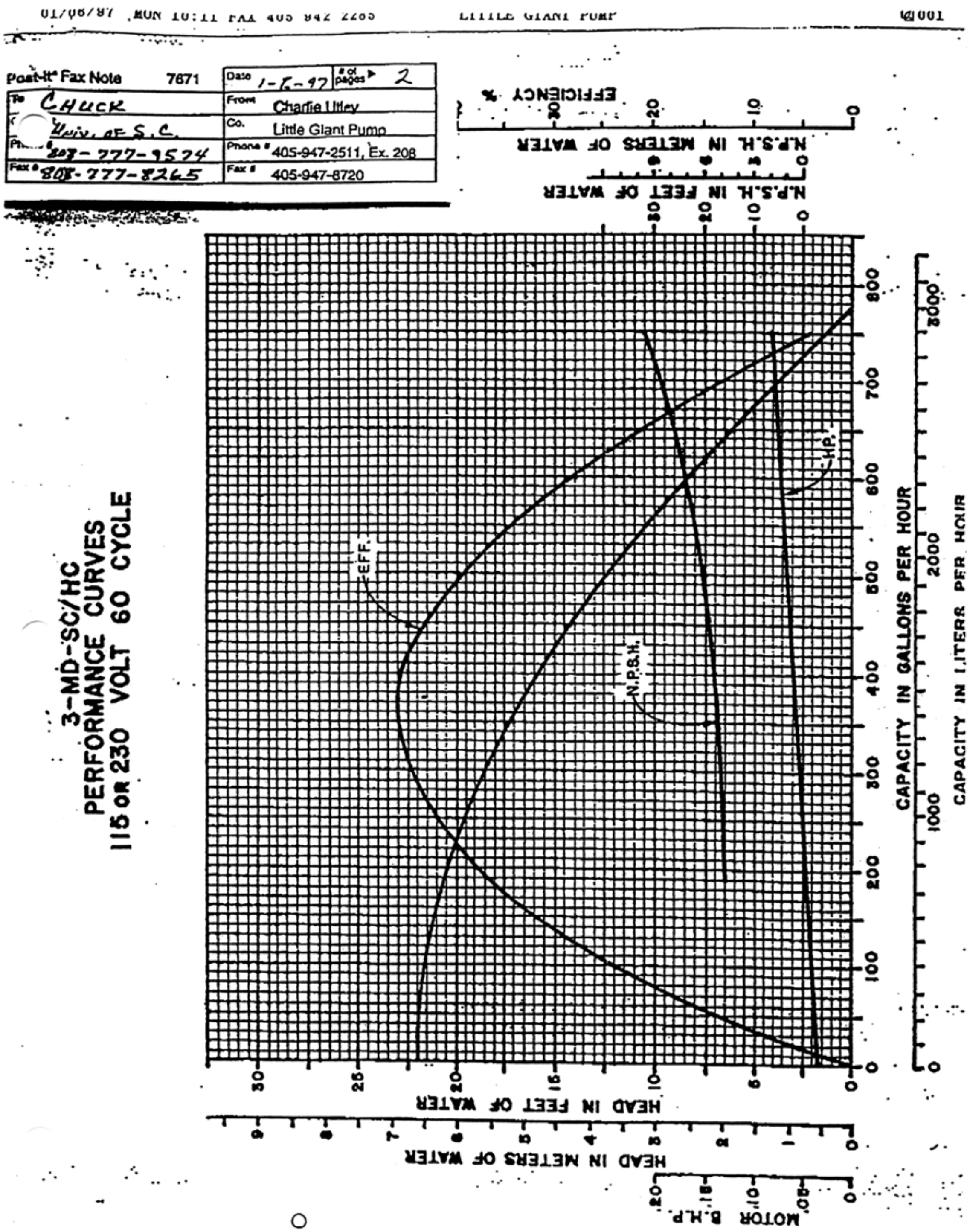
Item No.	Part Description
1	Nut
2	Stud, Collared
3	Drive Magnet Assembly
4	Mounting Bracket

Model No.	Catalog No.	Wgt. (lbs.)	Intake Size (In.)	Disch. Size (In.)	MOTOR									
					HP	RPM	Volts	PH	Hertz	Watts	Amps	Type	UL	CSA
3-MD-SC	581503	8.2	3/4 FNPT	1/2 MNPT	1/12	3100	115	1	60	180	2.4	Open FC	UR	C-CS
3-MD-SC	581513	8.2	3/4 FNPT	1/2 MNPT	1/12	3000	220-240	1	60	181	1.2	Open FC	UR	C-CS
3-MD-SC	581576	7.7	3/4 FNPT	1/2 MNPT	1/12	2600	220-240	1	50	205	1.7	Open FC	UR	NO
3-MD-SC	581597	2.0	3/4 FNPT	1/2 MNPT	Pump Head Less Motor									

FORM 995404 2/95 Rev. New

Little Giant Pump Company
P.O. Box 12010
Oklahoma City, OK 73157-2010
Phone:(405) 947-2511
Fax: (405) 947-8720

Little Giant Pump Company
SUBSIDIARY
TECUMSEH PRODUCTS COMPANY


LITTLE GIANT PUMP
Post-it® Fax Note 7671
Date 1-6-97
# of pages 2
To CHUCK
From Charlie Utley
Co. Univ. of S.C.
Co. Little Giant Pump
Phone 803-777-9574
Phone 405-947-2511, Ex. 208
Fax 803-777-8265
Fax 405-947-8720
3-MD-SC/HC
PERFORMANCE CURVES
115 OR 230 VOLT 60 CYCLE
EFFICIENCY %
N.P.S.H. IN METERS OF WATER
N.P.S.H. IN FEET OF WATER
EFF.
N.P.S.H.
H.P.
CAPACITY IN GALLONS PER HOUR
CAPACITY IN LITERS PER HOUR
HEAD IN FEET OF WATER
HEAD IN METERS OF WATER
MOTOR B.H.P.

LITTLE GIANT® MAGNETIC DRIVE CENTRIFUGAL PUMPS

ENGINEERED FOR EXCELLENCE

MD-SC Series Model 5-MD-SC

FOR SEMI-CORROSIVE SOLUTIONS AND CHEMICALS

LITTLE GIANT PUMP QUALITY ENGINEERED FOR PERFORMANCE

Description:

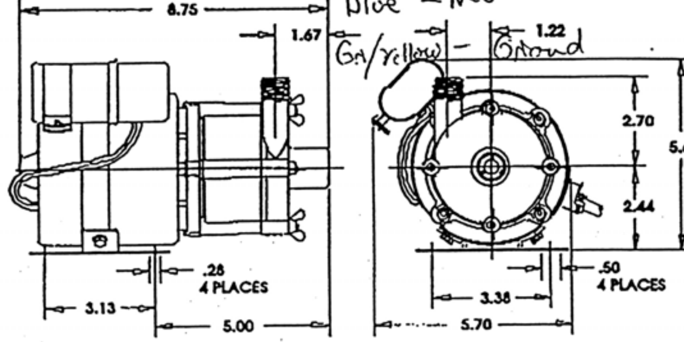
Thermally protected open fan-cooled permanent split capacitor motor with sleeve bearings. 6' power cord with 3-prong molded plug. (230V version has no plug.)

Fluid System Design Parameters:

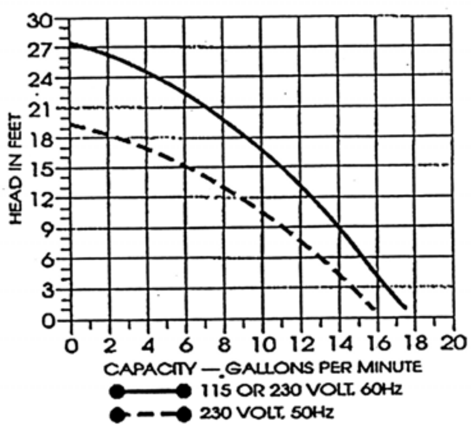
- Specific Gravity - Up to 1.1.
- Ambient Air Temperature - Up to 77°F (25°C).
- Fluid Temperature - Up to 150°F (66°C).

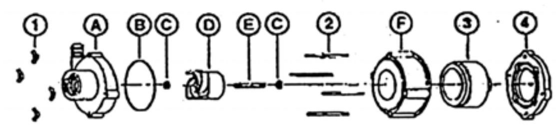
NOTE: Consult your local distributor or the factory for applications with higher ambient temperatures, …ific gravities, and viscosities.

Brown – Hot
Blue – Neu
Gr/Yellow – Ground



NOTE: Designs and dimensions may vary for various reasons (i.e. type of motor). This information should be used as general guide rather than an unqualified guarantee. Specifications are subject to change without prior notice.





Wetted Parts	Materials of Construction
A	Glass-filled Polypropylene Volute
B	Buna-N® (Nitrile) O-ring
C	Ceramic Thrust Washers
D	Glass-filled Polypropylene Impeller Assembly
E	Ceramic Shaft
F	Glass-filled Polypropylene Magnet Housing

Item No.	Part Description
1	Nut
2	Stud, Collared
3	Drive Magnet Assembly
4	Mounting Bracket

Model No.	Catalog No.	Wgt. (lbs.)	Intake Size (in.)	Disch. Size (in.)	MOTOR									
					HP	RPM	Volts	PH	Hertz	Watts	Amps	Type	UL	CSA
5-MD-SC	583503	9.25	1 FNPT	1/2 FNPT	1/8	3000	115	1	60	208	2.0	Open FC PSC	UR	C-CSA
5-MD-SC	583513	9.25	1 FNPT	1/2 MNPT	1/8	2400/2500	220/240	1	50/60	200/240	1.0/1.1	Open FC PSC	UR	C-CSA
5-MD-SC	583598	2.0	1 FNPT	1/2 MNPT	PUMP HEAD LESS MOTOR									

FORM 995406 2/95 Rev. New

Little Giant Pump Company
P.O. Box 12010
Oklahoma City, OK 73157-2010
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Fax: (405) 947-8720

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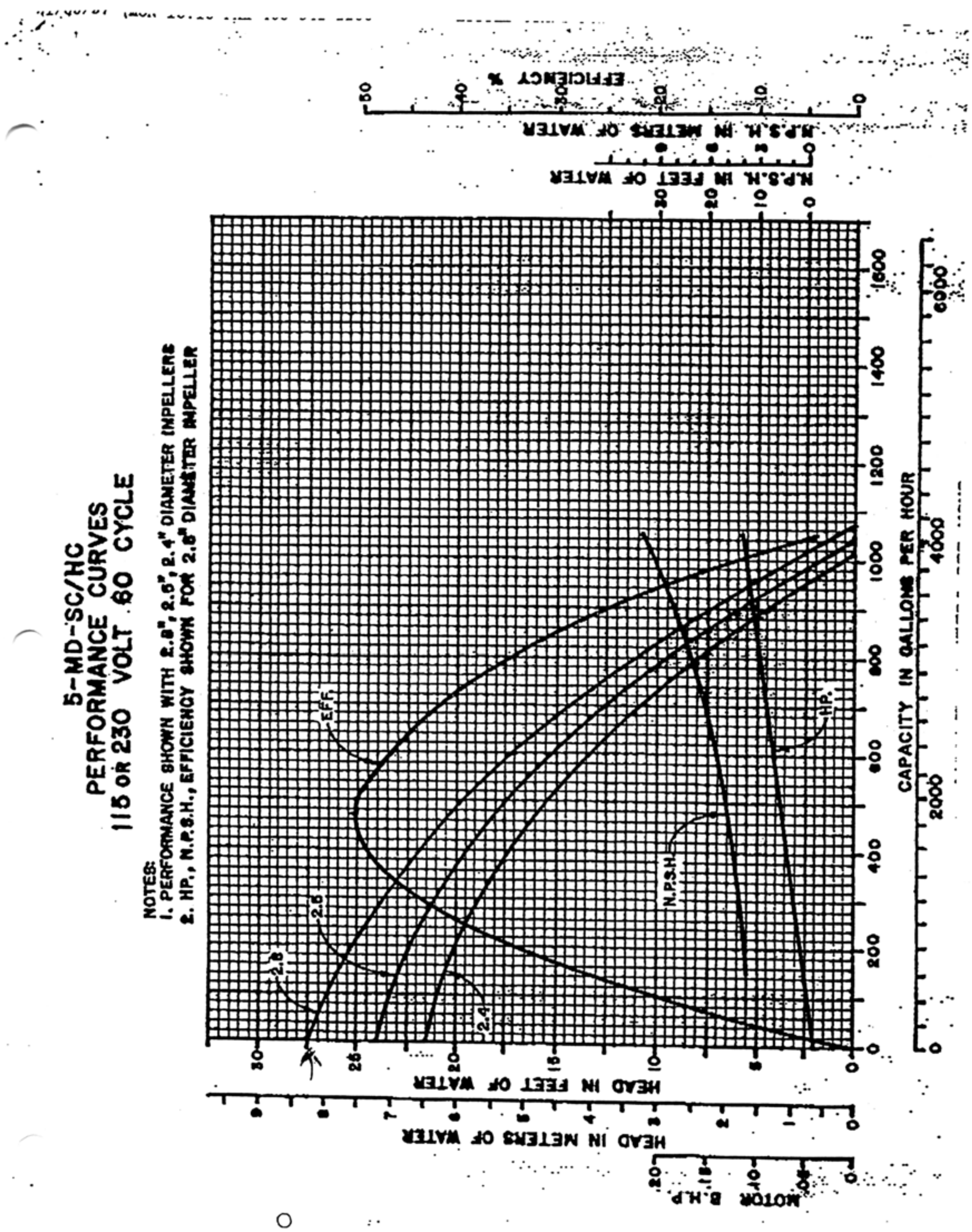
5-MD-SC/HC
PERFORMANCE CURVES
115 OR 230 VOLT 60 CYCLE
NOTES:
1. PERFORMANCE SHOWN WITH 2.8", 2.6", 2.4" DIAMETER IMPELLERS
2. HP., N.P.S.H., EFFICIENCY SHOWN FOR 2.8" DIAMETER IMPELLER
EFFICIENCY %
N.P.S.H. IN METERS OF WATER
N.P.S.H. IN FEET OF WATER
HEAD IN FEET OF WATER
HEAD IN METERS OF WATER
MOTOR B.H.P.
CAPACITY IN GALLONS PER HOUR
EFF.
H.P.
N.P.S.H.
2.8
2.6
2.4

Little Giant®

Pump Specifications

MODEL	CAT. NO.	INTAKE SIZE	DISC. SIZE	MOTOR VOLTS	MOTOR HZ	MOTOR RPM	MOTOR PH	MOTOR HP	MOTOR WATTS	MOTOR AMPS	MOTOR TYPE	MAX. PSI	SHUT OFF HEAD (FT.)	GALLONS PER MINUTE 1'	GALLONS PER MINUTE 5'	GALLONS PER MINUTE 10'	GALLONS PER MINUTE 15'	DIMENSIONS HT.	DIMENSIONS WD.	DIMENSIONS LG.	WT. PKD. LBS.
MD SERIES SPECIFICATIONS																					
1-AA-MD	566001	1/2" O.D.	1/2" O.D.	115	60	3250	1	1/200	19	.4	OPEN	2.35	6.4	2.7	1.7	-	-	3.72	2.4	4.73	2.33
1-AA-MD	566002	1/2" O.D.	1/2" O.D.	230	60 50	3000 2900	1	1/200	20	.12 .14	OPEN	2.35 1.8	6.4 4.2	2.7 2.5	1.7 1.2	-	-	3.72	2.4	4.73	2.33
1-MD	569002	1/2" O.D.	1/2" O.D.	115	60	3450	1	1/70	60	.79	OPEN FC	4.6	10.6	3.0	2.8	0.3	-	5.0	4.0	6.8	4.0
1-MD	569012	1/2" O.D.	1/2" O.D.	230	60 50	3000 2700	1	1/70	100	.64 .66	OPEN FC	4.6 3.2	10.6 7.5	3.0 3.6	2.8 3.2	0.3 -	-	5.0	4.0	6.8	4.0
2-MD	560002	1/2" FNPT	1/2" MNPT	115	60	3100	1	1/30	100	1.6	OPEN FC	6.3	14.6	8.5	7.8	5.6	-	5.0	4.0	6.9	6.25
2-MD	560012	1/2" FNPT	1/2" MNPT	230	60 50	3000 2700	1	1/30	100 105	.64 .65	OPEN FC	6.3 4.5	14.6 10.5	8.5 7.8	7.8 7.0	5.5 3.3	-	5.0	4.0	6.9	6.25
3-MD	561002	1/2" FNPT	1/2" MNPT	115	60	3300	1	1/20	105	1.4	OPEN FC	7.1	18.3	9.8	9.1	6.5	2.5	5.0	4.0	6.9	7.6
3X-MDX	561030	5/8" O.D.	5/8" O.D.	115	60	1550	1	1/50	67	.84	OPEN FC	3.0	7.0	5.4	5.0	-	-	5.25	3.8	7.7	6.2
3-MDX	561031	5/8" O.D.	5/8" O.D.	115	60	3000	1	1/25	126	1.65	OPEN FC	8.2	19.0	7.6	7.2	6.0	3.0	5.25	3.8	7.7	7.0
TE-3-MD	561012	1/2" FNPT	1/2" MNPT	230	60 50	3200 2750	1	1/20	115 135	.8 1.1	TEFC	7.1 5.0	18.3 11.5	9.6 8.2	9.1 7.5	6.5 3.8	2.5 -	5.0	4.0	8.6	8.0
4-MD	562002	1/2" FNPT	1/2" MNPT	115	60	3250	1	1/12	213	2.4	OPEN FC	10.2	23.7	11.5	11.0	9.1	6.5	5.0	4.0	8.9	8.25
4-MD	562012	1/2" FNPT	1/2" MNPT	230	60	2500	1	1/12	212	1.4	OPEN FC	10.2	23.7	11.5	11.0	9.1	6.5	5.0	4.0	8.9	8.25
5-MD	563002	1/2" FNPT	1/2" MNPT	115	60	3250	1	1/8	200	2.1	PSC OPEN FC	15.0	34.6	15.1	14.6	12.5	10.6	5.7	5.7	8.6	9.25
5-MD	563012	1/2" FNPT	1/2" MNPT	230	60 50	2500 2400	1	1/8	240 200	1.1 1.0	PSC OPEN FC	15.0 9.7	34.6 22.4	15.1 15.8	14.6 15.1	12.5 11.9	10.6 6.4	5.7	5.7	8.6	9.25
MD – SC SERIES SPECIFICATIONS																					
1.5-MDI-SC	569201	3/4" MNPT 3/8" FNPT	1/4" MNPT	115	60	3100	1	1/35	86	1.1	OPEN FC	5.4	12.5	5.4	5.0	2.7	-	4.5	3.5	7.5	4.5
PE-1.5-MDI-SC	569204	1/2" MNPT	1/4" MNPT	115	60	3000	1	1/35	63	0.9	ENCAP.	5.5	12.8	5.4	5.0	2.7	-	3.75	4.02	7.5	5.0
2-MD-SC	560503	1/2" FNPT	1/2" MNPT	115	60	2900	1	1/25	109	1.43	OPEN FC	6.3	14.6	8.5	7.4	5.5	-	5.0	4.0	6.9	6.25
2-MD-SC	560513	1/2" FNPT	1/2" FNPT	230	60 50	3000 2700	1	1/30	100 105	0.64 0.65	OPEN FC	6.3 4.5	14.6 10.5	8.5 7.8	7.8 7.0	5.5 3.3	-	5.0	4.0	6.9	6.25
2-MDIX-SC	560509	1" FNPT	1" MNPT	115	60	3000	1	1/25	110	1.5	OPEN FC	3.5	8.1	12.8	10.7	-	-	5.25	4.0	9.3	6.25
3-MDI-SC	561504	3/4" FNPT	1/2" MNPT	115	60	3200	1	1/15	90	0.8	TEPSC	7.8	18.0	11.5	10.8	8.3	4.0	4.9	4.0	11.4	9.2
3-MDIX-SC	561509	1" FNPT	1" MNPT	115	60	3200	1	1/15	100	0.9	TEPSC	8.1	14.0	18.3	16.7	11.5	-	5.21	4.0	11.73	9.2
3-MD-SC	561503	3/4" FNPT	1/2" MNPT	115	60	3100	1	1/12	180	2.4	OPEN FC	9.5	21.9	12.5	11.8	10.0	7.1	5.0	4.0	9.6	8.2
3-MD-SC	561513	3/4" FNPT	1/2" MNPT	230	60	3000	1	1/12	181	1.2	OPEN FC	9.5	21.9	12.5	11.8	10.0	7.1	5.0	4.0	9.6	8.2
3-MD-SC	561576	3/4" FNPT	1/2" MNPT	230	50	2600	1	1/12	205	1.7	OPEN FC	6.5	15.0	10.4	9.9	7.0	-	5.0	4.0	9.6	7.7
4-MD-SC	562503	1" FNPT	1/2" MNPT	115	60	3000	1	1/10	185	1.7	OPEN FC PSC	10.5	24.3	14.2	13.5	11.8	9.2	4.9	5.5	9.75	9.0
4-MDIX-SC	562509	1" FNPT	1" MNPT	115	60	3280	1	1/10	145	1.3	OPEN FC PSC	7.4	17.0	22.1	20.4	15.0	5.2	5.21	5.13	8.92	9.0
TE-4-MD-SC	562514	1" FNPT	1/2" MNPT	230	60 50	3000 2750	1	1/10	150 120	0.7 0.5	TEFC PSC	10.5 6.9	24.3 16.0	14.2 12.8	13.5 12.0	11.8 9.0	9.2 3.8	5.6	5.5	10.8	9.75
TE-5-MD-SC	564504	1" FNPT	1/2" MNPT	115	60 50	3450 2850	1	1/6	325 220	3.0 2.3	TEFC PSC	12.7 8.8	29.3 20.4	20.0 16.2	19.2 15.2	17.0 12.2	14.3 8.0	7.3	5.2	11.2	19.0
5-MD-SC	563503	1" FNPT	1/2" FNPT	115	60	3000	1	1/8	208	2.0	OPEN FC PSC	11.9	27.5	17.5	16.5	14.0	10.9	5.64	5.7	8.75	9.25
5-MD-SC	563513	1" FNPT	1/2" FNPT	230	60 50	2500 2400	1	1/8	240 200	1.1 1.0	OPEN FC PSC	11.9 8.5	27.5 18.5	17.5 15.8	16.5 14.8	14.0 11.0	10.9 6.1	5.64	5.7	8.75	9.25
TE-5.5-MD-SC	565504	1" FNPT	3/4" MNPT	115 230	60 60 50	3450 3450 2850	1	1/3	625 525 340	5.0 2.5 2.8	TEFC	18.0 18.0 12.6	42.0 42.0 29.0	- - -	33.0 33.0 27.0	30.5 30.5 24.5	28.0 28.0 21.0	7.0	8.71	14.4	28.0
TE-6-MD-SC	566504	1" FNPT	3/4" MNPT	115 230	60 60 50	3450 3450 2850	1	1/2	835 835 640	9.4 4.7 5.6	TEFC	22.0 22.0 15.2	51.0 51.0 35.0	- - -	38.0 38.0 33.0	36.5 36.5 30.0	34.0 34.0 26.5	7.0	8.71	14.4	29.0
MD – HC SERIES SPECIFICATIONS																					
2-MD-HC	560603	1/2" FNPT	1/2" MNPT	115	60	3100	1	1/30	105	1.7	OPEN FC	6.3	14.6	8.5	7.8	5.5	-	5.0	4.0	8.0	7.0
2-MD-HC	560613	1/2" FNPT	1/2" MNPT	230	60 50	3000 2700	1	1/30	100 105	0.64 0.65	OPEN FC	6.3 4.5	14.6 10.5	8.5 7.8	7.8 7.0	5.5 3.3	- -	5.0	4.0	8.9	7.0
3-MD-HCT	561603	3/4" FNPT	1/2" MNPT	115	60	3100	1	1/12	180	2.4	OPEN FC	9.5	21.9	12.5	11.8	10.0	7.1	5.0	4.0	9.6	8.6

APPENDIX (SECTION A)

FLOW & PRESSURE CHARTS & CONVERSIONS

- Flow & Pressure Charts
- Heads and Equivalent Pressure Conversions
- Pressure Conversions
- Flow Rate Conversions

DETERMINING INSIDE DIAMETER OF TUBING

The I.D. of tubing is set by flow requirements, permissible pressure drop and maximum allowable velocity. To aid in selecting the proper I.D. of tubing for liquid flow, Charts 1 through 10 are provided on the following pages. Charts 11 through 20 are provided for sizing tubing for gas flow.

These charts give pressure drop for 100 feet of tubing for both water and air flow. By using the formula provided, it is also possible to obtain the pressure drop of fluids other than water and gas.

To allow for pressure drops in bends and fittings, the equivalent lengths in Table A can be used when obtaining equivalent length of tubing for pressure drop calculations. To obtain equivalent length of tubing, total all straight lengths and then add lengths for each bend, elbow or tee from Table A.

EQUIVALENT FEET OF STRAIGHT TUBE

Tubing O.D. (In.)	90° Elbow (ft.)	90° Bend (ft.)	180° Bend (ft.)	45° Bend (ft.)	Tee Branch (ft.)
1/4	1	1/2	1	1/2	1
3/8	1-1/2	1/2	1	1/2	1-1/2
1/2	2	1/2	1	1/2	2
5/8	2-1/2	1	2	1/2	2-1/2
3/4	3	1	2	1/2	3
1	4-1/2	1	2	1/2	4-1/2
1-1/4	5	2	4	1	5
1-1/2	6	2	4	1	6
2	9	2	4	1	9

Table A