Introduction to the Design of Mobile Hydraulic Systems - *Part 2*

Course No: M02-045

Credit: 2 PDH

K. Michael Clark, P.E.

Continuing Education and Development, Inc.
22 Stonewall Court
Woodcliff Lake, NJ 07677

P: (877) 322-5800
info@cedengineering.com

ANALYSIS

Cylinders

The equations describing the performance of a double acting cylinder are reproduced here from the section on End Devices in Course 1. In this example the 'Head end' refers to the blind end or the end without the rod and the 'Rod end' is the end with the rod. In calculating the area that the pressure is acting on, accounting of the rod area must be made as indicated here. If there is a rod on both ends, simply account each end with each rod diameter. In double acting cylinders, the differential volume of hydraulic fluid in the cylinder is simply the rod area (or differential rod area) times the stroke.

Cylinder Performance Analysis

$$\text{Force} = (P_H)(A_H) - (P_R)(A_H - A_R)$$

$$\text{Flow}_{Head} = \pi(D_H/2)^2 * (\text{rod vel}) / 231 \qquad \text{gpm}$$

$$\text{Flow}_{Rod} = \pi((D_H/2)^2 - (D_R/2)^2) * (\text{rod vel}) / (231) \qquad \text{gpm}$$

$$\text{Differential Fluid Volume} = A_R\,(\text{Stroke Length})$$

$$= \pi(D_R/2)^2\,(\text{Stroke Inch})/231 \qquad \text{Gallons}$$

Where:

P ~ Lb/in^2

A and D ~ Head and Rod area and diameter, $inch^2$ and inch

rod vel ~ inch/minute

If the cylinder is Single Acting there will be no return flow in one direction of the cylinder. Flow only returns when the cylinder returns due to the load. Either end may be the active end. The other end is generally vented to the atmosphere. The above equations simplify since there is no significant force generated on the vented end, no flow returning from the vented end, and the differential volume becomes the active end area times the stroke.

Motors and Pumps

The equations describing the performance of pumps and motors are also reproduced here from the section on End Devices in Course 1. Remember there is no difference between a pump and motor except for the direction that work or energy is flowing. These equations apply to any positive displacement pump or motor used in hydraulic systems. Centrifugal pumps and motors are not typically used in hydraulic systems and are not discussed here.

Pumps and Motors (American Standard Units)

Torque = CIR * ΔP / 2π	in-lb
Flow Rate = CIR * rpm / 231	gpm
Shaft Power = Torque * rpm / 63025	hp
Hydraulic Power = gpm * ΔP / 1714	hhp or hydraulic horse power

Where:

CIR = pump or motor displacement cubic inch/revolution

rpm = revolutions / minute

gpm = gallons / minute

ΔP = pounds / inch^2

In metric units the performance becomes:

Pumps amd Motors (Metric Units)

Torque = (mL/rev) * Δbar / (20π)	N-m
Flow Rate = (mL/rev) * rpm / 1000	L/min
Shaft Power = W = π * Torque * rpm / 30	watts
Hydraulic Power = (5/3) * (L/min) * Δbar	watts

Where:

mL/rev = pump or motor displacement, mili-Liter/rev.

(KPa/1000 may be used in place of bar)

These equations are available from several vendor web sites where you can also download catalogs describing their selection of pumps and motors.

Thermal Analysis

Many hydraulic systems operate for extended periods. If the duty cycle for your system includes operating periods longer than a few minutes (depending on reservoir size) you must consider the possibility of excessive heat build-up in the hydraulic fluid. The net work put into the hydraulic system will eventually end up in the hydraulic fluid as heat. You can either sum all of the pressure loses (calculated later in this section), or take a more global view of the system and calculate 'net work.' The global approach begins with the work input to the fluid by the pump. Then subtract the work done (or add negative work done). Your system may normally lift a load that remains at a higher elevation; for example, a conveyor that carries a load up a ramp and dumps, or a system with external losses that do not feed back into the hydraulic fluid as heat. The following figure illustrates how to analyze this.

ΔT = (Work) / (mass flow rate)(fluid C_v)

$\Delta T = Q * \Delta P / (Q * \rho * C_v) = \Delta P / (\rho * C_v)$

The issue now is resolving units

$\Delta T = \Delta P*(144\ in^2/ft^2)/((62.4\ lb/ft^3)*SpGr*C_v*\eta_{pwr}*(778\ ft\text{-}lb/BTU))$

Included is the density of fresh water, 62.4 lb/ft^3, fluid specific gravity, pump power efficiency, and units of pound, foot, BTU, and $^\circ$F.

$\Delta T = 0.002966 * \Delta P / (SpGr*C_v* \eta_{pwr})$

For typical hydraulic oil, and a high efficiency pump this reduces to:

$\Delta T = 0.0103 * \Delta P$ (or about 10°F per 1000 psi)

Where: ΔT is

ΔP is pounds per square inch differential, psid

SpGr is fluid specific gravity = 0.8

η is pump overall efficiency = 0.9

C_v is oil specific heat = 0.4 BTU/(lb-°F)

The easiest way to calculate the effect of this temperature increase on the system is to mentally reduce the system to three elements. These are: the pump, the net loads, and the reservoir. Then divide the duty cycle into approximately constant power increments of time or time increments that flow a maximum of about $1/10^{th}$ of the reservoir liquid volume. In each time increment about $1/10^{th}$ of the reservoir volume is heated and returned in the remaining fluid mass (averaged). If the inside oil wetted surface of the reservoir is large, you can assume a heat transfer coefficient of about 0.5 to 0.75 BTU/(°F ft^2 hour) for natural convection in still air to calculate the equilibrium temperature. Don't over estimate this heat transfer coefficient because accumulated dirt on the external surface will lower the effective convection coefficient. If the hydraulic fluid temperature goes too high during the worst duty cycle in the worst environmental condition, you need active cooling of some kind. The limit high hydraulic fluid temperature is dependent on the particular fluid in use and on the minimum pressure in the system. For example, if you are using water for the hydraulic fluid and a vented reservoir, your maximum fluid temperature will be less than 200°F (95°C).

If a heat exchanger is required, the following calculations as well as the heat exchanger vendors will help you size this device.

- Size a heat exchanger based on total energy that needs to be dissipated.

  $$\text{BTU/Min} = 42.4 * \text{hp}$$

  $$= 42.4 * (\text{Hydraulic Power})/\eta_{pwr}$$

  $$\mathbf{BTU/Min = 0.02474*gpm*\Delta P/\eta_{pwr}}$$

- To specify a heat exchanger, specify hydraulic fluid flow rate(s), maximum acceptable outlet and maximum expected inlet temperature, and ambient temperature (oil/air exchanger) or other heat sink fluid temperature.

Thermal Expansion Effects

There are two considerations that must be given to thermal expansion. First is a consideration in sizing the system reservoir. You don't want to fill a system cold, then while operating and heating the fluid, discover that fluid is overflowing onto the ground. The reservoir must accommodate the difference in fluid volume between minimum storage temperature and maximum operating temperature. The reservoir must also accommodate changes in volume in the system from components like cylinders. For example, if the system includes a single acting cylinder that at maximum extension contains 5 gallons more fluid than at storage, that 5 gallons of additional fluid has to come from (or return to) the reservoir.

While on the subject of reservoir volume, it must also contain additional fluid to accommodate routine leakage and evaporation between routine servicing of the system.

The second consideration of thermal expansion of the fluid has to do with system design. Any system will experience temperature extremes while operating and while 'off.' For example, while the system is off it may be stored outside with cold nights and direct sun in the days that heat part of the system. Changes in temperature of about 200°F are possible. The problem is that typical hydraulic fluids have a coefficient of thermal expansion that is one to two orders of magnitude higher than aluminum or steel. The coefficient of thermal expansion of typical hydraulic fluids is 0.0008/°C. The coefficient for aluminum and steel are 0.00003/°C and 0.000004/°C, respectively. If your system has trapped fluid that cannot return to the reservoir, you have a problem. As the fluid heats, it will expand more than the metal and pressure will increase to an equilibrium between fluid compressibility and thermal expansion coefficients (pressure may go very high). When the temperature gets high enough, the pressure will increase sufficiently to yield and deform the metal components. The manifold or tube may not totally fail the first cooling-heating thermal cycle, but after repeated cycling, the metal will eventually yield enough to cause a leak, either internal to a manifold, or external.

Viscous Pressure Loss

Pressure loss calculations are divided into two groups. First are the losses associated with fluid viscosity. Sometimes viscous losses are referred to as Major or primary losses. Second are those losses due to dynamic pressure in corners, rapid expansions or contractions, and valves. Dynamic pressure losses are sometimes referred to as Minor or secondary losses. The relative importance is dependent on details of the system and is not related to the names or order of consideration. Typically pressure losses in machinery are either trivial or critical. Since viscosity of hydraulic fluids increase exponentially as temperature drops, critical pressure loss is found on cold start-up or severe cold operations. Viscosity vs. temperature is linear on an ASTM log-log plot as seen in the figure below. The temperature scale is the log of absolute temperature (°R) and has been labeled with the equivalent °F.

The flow in hydraulic tubes and hoses will either be laminar or turbulent. The Friction factors are found in most classical boundary layer theory text books for each as follows:

Laminar resistance coefficient, or Friction Factor, for smooth tubes:

$FF = 64 / R_d$ Schlichting 5.11

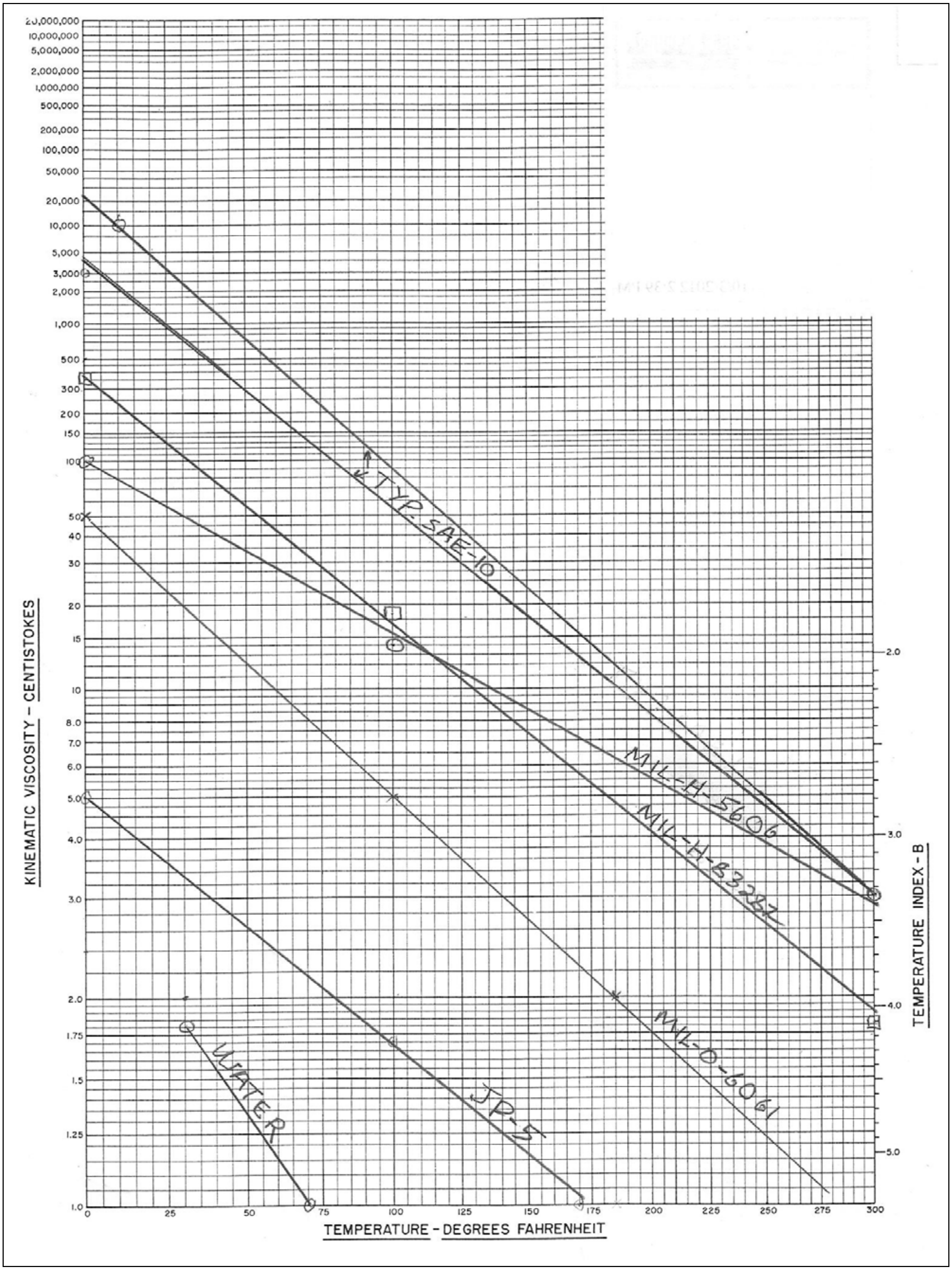
Turbulent Friction Factor (Blasius correlation):

$FF = 0.3164 / (R_d^{\,0.25})$ Schlichting 20.5

for $R_d < 100{,}000$

Where R_d is based on tube inside diameter

Reynolds number here is based on the wetted inside diameter of the conduit. You will need viscosity from the figure. It is suggested that you look up the viscosity for your selected fluid. If you find the viscosity at two temperatures more than 100°F apart, they can be plotted on the figure and extended / interpolated graphically as a straight line. Favor the lowest temperature point in drawing your line. Multiply Centistokes from the curve by 0.00001076 to convert viscosity to Ft^2/Sec.


KINEMATIC VISCOSITY - CENTISTOKES
TEMPERATURE - DEGREES FAHRENHEIT
TEMPERATURE INDEX - B
TYP. SAE-10
MIL-H-5606
MIL-H-83282
MIL-O-6061
JP-5
WATER

Reynolds number is defined as:

$$R = (\text{velocity} * \text{wetted diameter} * \text{density}) / (\text{viscosity})$$

$$= (\text{velocity} * \text{wetted diameter}) / (\text{kinematic viscosity})$$

Where:

Velocity	= flow rate / flow area (Ft / Sec)
Diameter	= Fluid wetted diameter (Feet)
Kinematic viscosity	= Centistokes from curve * 0.00001076 (to convert viscosity to Ft^2/Sec)

Transition from laminar to turbulent flow occurs between Reynolds numbers of 1500 and 2500. However the curves of friction factor cross at a Reynolds number of 1167. A reasonable and slightly conservative (higher loss) assumption is to use the larger of laminar or turbulent friction factor.

In some hydraulic hose and most steel tubing, roughness is not an issue. In rougher pipe or hose the effect of roughness will usually be to limit the decline in friction factor with increasing Reynolds number to 0.015 to 0.020 or above. For further details, see Moody Diagram or see Schlichting Figure 5.18.

From the friction factor, the calculation of pressure loss is outlined here (in American Standard units).

$$\Delta h = FF * (L/D) * (V^2/(2*g))$$

$$\Delta P = (FF * \rho * L/D * V^2)/(144*2*g)$$

$$=62.4*SpGr*FF*L/D*(V^2)/(144*2*32.174)$$

$$\mathbf{\Delta P = 0.00673 * FF * SpGr * L/D * V^2} \qquad \text{psid}$$

Where:

ρ	Fluid density = 62.4 Lbs/ft^3 (water)
SpGr	Specific Gravity of liquid of interest
L/D	segment Length / Diameter ratio
g	gravity constant = 32.174 ft/sec^2
V	Velocity = flow rate / area

In the attached spread sheet work book, this series of calculations is presented on the sheet tab labeled 'VISCOUS.'

Dynamic Pressure Loss

Pressure is the measure of energy in a fluid. The Bernoulli equation describes the relationship between kinetic and potential energy in a flowing fluid. The kinetic energy portion is referred to as the dynamic pressure and the potential energy as the static pressure.

The Bernoulli equation describes the conservation of energy within a flowing fluid as follows:

Total Pressure = Static Pressure + Dynamic Pressure

$$P_t = P_s + q = P_s + w*V^2/(2g)$$

$$= P_s + (SpGr)*62.4*(V^2)/(2*32.174)$$

$$\mathbf{P_t = P_s + 0.9697 * SpGr * V^2} \qquad \mathbf{Lb/ft^2}$$

$$\mathbf{P_t = P_s + 0.00673 * SpGr * V^2} \qquad \mathbf{Lb/in^2}$$

Where:

P_t	Total pressure of flow	
P_s	Static Pressure of flow at that location	
SpGr	Specific Gravity of fluid	
62.4	density of fresh water in Lb/ft^3	
V	velocity (volume flow rate/Area	ft/sec

In the integration of the hydraulic system into the vehicle there often will be tight spots where locally the tube or hose diameter must be smaller, resulting in fluid velocities greater than 10 ft/sec. Until specific data becomes available, the following table is provided as guidance for initial calculation of system pressure losses. If you know early in the design that you have a problem in a specific area of the vehicle, you may have options that are not available later in the design process.

Feature	Loss Multiple of Dynamic Pressure
Mitered Elbow	1
Expansion 2 sizes	0.5
Expansion 4 size or more	1
Contractions	Same as expansion and based on Upstream average velocity
Radius Elbow	0.1-0.5
Quick Disconnect	2 (use vendor data ASAP)
Valve	2-3 (use vendor data ASAP)
Filter (clean)	3 (use vendor data ASAP)

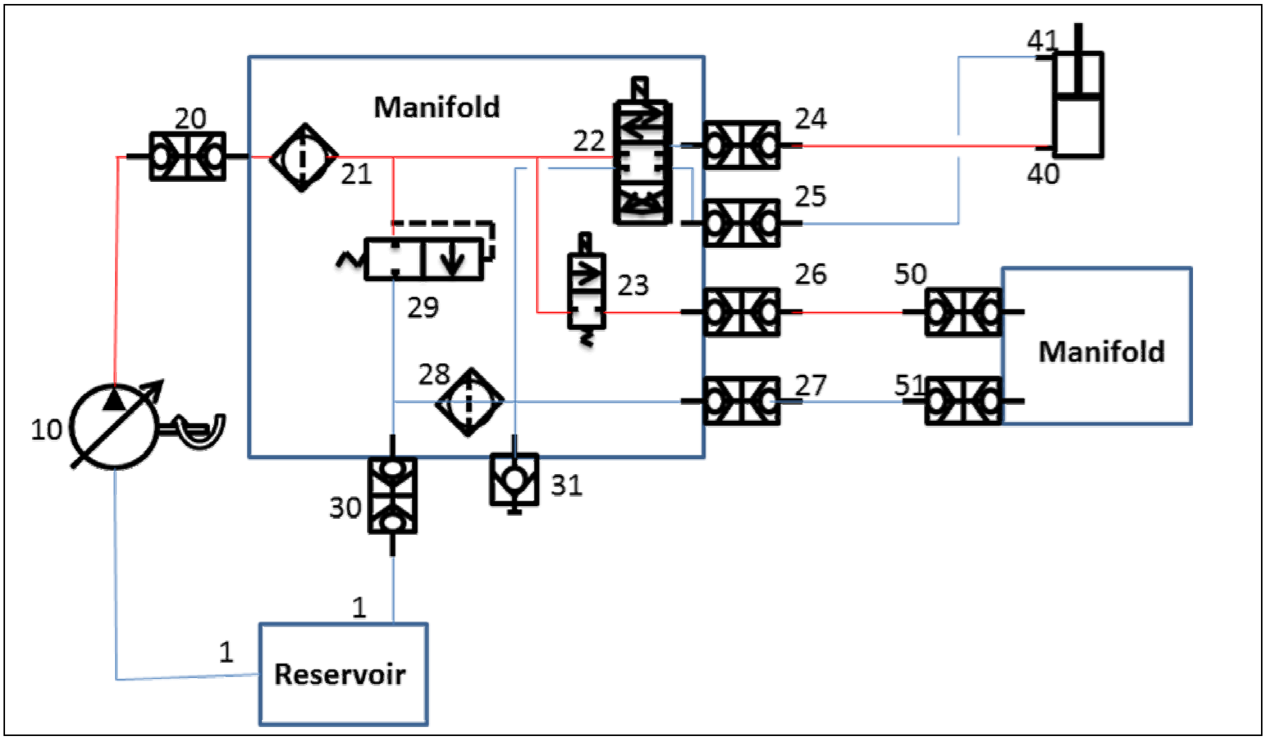
In addition valves and fittings often result in the loss of all or some of the fluid dynamic pressure. The pressure loss through a manifold should be identified by a test conducted by the vendor, including the valves contained in the manifold. If these losses are to be defined by tests, these tests must be identified and agreed to prior to signing the purchase contract. Testing costs money and you must be fair to yourself and your vendor. The table above lists some guidelines for early estimation of dynamic pressure loss.

The included Excel work book includes a tab labeled 'DYNAMIC PRESSURE.' This sheet includes an area for input of flow parameters on the left side and calculates the dynamic pressure on the right. The intent is that the VISCOUS or DYNAMIC PRESSURE line be copied to your own work book in sequence as you build a model of your system to calculate pressure loss. The common inputs between VISCOUS and DYNAMIC PRESSURE sheets are in the same column. Common outputs are in the same column to make trouble shooting easier.

Between a variable load or valve and the reservoir, the calculation logic must change because the pressure drop across the valve/load is unknown. However, if the system is running, the fluid is flowing back to the reservoir and the reservoir pressure is known. Beginning with reservoir pressure and working back to the load the losses can be added to the reservoir pressure (or segment just downstream) to work back to the load or valve.

Bookkeeping

With the analysis outlined above, you are now ready to calculate your way through your system. In order to keep track of the pressures at each location in the system and especially at the loads, it is suggested that each valve or component be identified with a number or letter.



Begin with 1 at the reservoir. Assign a number to each valve or component in sequence in a manifold. It is suggested that you leave a block of numbers unused at the end of each manifold or end device for later addition of components you don't yet know you need. I have found it very helpful to build a spread sheet or data base sheet identifying each component by number, manufacturer and part number, and rated or expected performance.

When specific pressure loss performance becomes available, it is recommended the specific spread sheet line be updated (re-programmed) with that specific data. In a large system it is suggested that the source and date of update be noted on one end of the line so you can track the status of updates.

How to use these tools

Maximum pressure loss in a system most likely will be on cold start-up and/or cold operation. The spread sheet is designed to be used to answer the question; "Do I have a pressure loss problem?" To use the spread sheet, assume a fixed flow rate (required speed of load). If the ΔP calculated at the end device is sufficient to move the load, you are done. If the calculated

ΔP is low or negative, the end device in actuality will move more slowly than desired. The spread sheet is not set up to iterate to find a reduced flow rate. If the ΔP is only slightly low, the system will warm up and quickly achieve required speed due to the heat added by work.

If the ΔP is near zero or negative, the load will not move or will move very slowly. If the speed (flow rate) is a requirement, you have work to do. The pressure loss must be reduced by reducing velocity (bigger hoses and tubes) or reducing the fluid viscosity. Looking at the viscosity curve, it is clear that heating the fluid from 0°F to 50°F reduces viscosity by a factor of about 4. This presents the possibility of using work and circulation to warm the fluid.

As seen in the thermal analysis section, power input to the fluid will eventually end up as heat in the fluid. A 'circulation valve' added to the system to circulate fluid should be sized to about ½ of the pump maximum flow rate. In this instance it is desired that the pump operate at full pressure. This approach will require a thermo-couple or RTD in the system so it can be determined when suitable temperature is achieved. A circulation valve should be located to circulate as much of the system as possible.

If this is not sufficient, possibly a change in hydraulic fluid will help. Fluids have different slope of viscosity vs. temperature. One candidate cold weather hydraulic fluid is Mil-H-5606. The down side to 5606 is that it is more flammable than 'fire resistant' alternatives. A side note -- 'fire resistant' hydraulic fluids will burn; however, they do not generate enough energy to sustain combustion from a liquid without additional heat (i.e. a hot exhaust pipe or similar).

SAFETY

Safety considerations are discussed in the areas of system design safety issues and personal safety while working on or around hydraulic systems.

System Design Safety Considerations

Underlying design safety is the objective to never lose control of a load. Consider possible failures and design alternatives so no single failure will result in dropping or losing control of the load. For example, if the system includes a cylinder that supports a load, consider what will happen if a hydraulic line is severed while the load is 'up.' The answer may be to include a rod brake that is spring loaded 'on' and hydraulic pressure released. If there is not enough room for a rod brake, maybe a pilot operated check valve can be located in the cylinder head block. Pressure to retract the cylinder opens the check valve letting the fluid out of the load supporting side. Brakes or pilot operated check valves (POCV) can also be used to control motors that may support loads.

Hydraulics 'push, they do not pull.' This is a cute way of saying that you cannot control a load by shutting off the supply of hydraulic fluid. If the load is over-running the end device, that load must be controlled by controlling the hydraulic fluid flowing out of the device. For example, assume a 3000 psi system with 2500 psid available at the end device to control the load. Further, assume someone has tried to control that load by restricting the fluid supply side. Absolute pressure cannot be negative so in this case supply side pressure is limited to zero or -14 psig (gage). In actuality, the pressure will only go down to the vapor pressure of the hydraulic fluid at the temperature at that time. Comparing -14 psig to available 2500 psig we see that available force is 14/2500 or 0.005 or 0.5% of the force normally available, i.e., essentially zero load carrying capability by controlling the suction side of an end device.

To simplify assembly and later maintenance, use different size ports, fittings, and lines for supply and return sides of each end device. Typically higher pressure lines are smaller than return lines. If the device may have high pressure on either side, simply pick one side to make larger and the other side smaller. This reduces the chance of miss-connecting a motor or cylinder after repairs and getting an opposite motion from what is commanded. If it is impossible to use different sized lines, consider routing and clamping hoses in such a way that each hose will only reach the proper port when the end device is properly installed, or different hose or line finish or color for one side with color coded ports on the device.

Most hydraulic fluids are flammable. Even 'flame resistant' fluids will burn under some circumstances. For example fire resistant fluids will burn when sprayed on a hot exhaust manifold or muffler. In general fire resistant fluids will not continue to burn from a liquid without additional heat. In the design process the possibility of a leak resulting in a fire can be reduced by locating hydraulic devices, hoses, and tubes away from heat and ignition sources. Often high pressure hydraulic leaks will result in a fine spray or fog of hydraulic fluid. If a hydraulic component must be located near a potentially hot manifold, consideration should be given to shielding either the hot surface or hydraulic component. In general one may take care to minimize connections since most leaks occur at connections. Specifically, don't plumb a system with a stack of adapters. If the required adapter is not available, consider asking a vendor to provide a special adaptor.

From a design stand point, you have to repeatedly ask and answer several questions throughout the design process. Will the load always move under positive control? Is the speed of movement of the load acceptable under all circumstances? Hot fluid? Cold fluid? Fully loaded? Unloaded? What will happen if the movement of a load is blocked? Do I need to limit maximum force exerted by the end device? What will happen when a critical hydraulic line breaks? Etc.

Be careful not to trap fluid that is not vented when pressure builds up due to thermal expansion. This type of problem rarely causes a catastrophic failure; however, it can cause recurring leaks.

Somewhere between Design Safety and maintenance Personal Safety is the design of trouble shooting and maintenance equipment. To determine why something is not moving properly the maintainer will need to know pressure and flow rate at various points in the system. Include critical test ports for field attachment of diagnostic equipment in your design. For example if something is not moving, or is moving slowly, it is necessary to know pressure and flow rate at the pump discharge and at the end device motor or cylinder. If the equipment may operate remote from electrical power, the designer of this maintenance equipment needs to consider how the diagnostic information is to be obtained and recorded. If pump output is below expectations, be sure to verify the pump is supplied with adequate fluid (is fluid level and pressure in the reservoir at the proper level). If pump or motor volumetric efficiency has significantly decreased from the manufacturer's specifications, it needs to be replaced. As an example, we once replaced a pump on a prototype and remote piece of equipment. When we started the equipment up, the problem was little changed. Several different end devices would move slowly, indicating inadequate pump output flow rate; however, we just replaced the pump. With only minimum instrumentation available that did not require electrical power, we replaced several end devices but the problem remained. After a day and most of a night, we finally concluded the replacement pump was also bad. We secured another pump and replaced it and all was well. Had we had flow meters available, both the original and replacement pump problems would have been quickly and accurately diagnosed, saving hours of labor in front of our customer.

The details of the system design determine the necessity of bleeding air out of the system on initial assembly or after repairs. If the system flows fluid entirely through the system, air will end up in the reservoir where it is vented or may be bled. A flow through system has the direction control valve located on or very close to the end device. When the valve changes the direction of the device, more fluid can be made to flow into and out of the device than the device contains. If maximum amount of fluid that can be made to flow through the device is substantially less than it contains, then the line and device must be bled. If an end device must be bled to remove air, you should provide a port or manual valve to accomplish this. Trapped air in a system will result in uncommanded motion in the system, and undesirable heat when it is compressed. Therefore air must be eliminated from the system one way or the other.

Personal Safety

When observing equipment operations, do not stand or allow others to stand between a moving load and fixed objects. Consider the intended motion of the load, as well as possible unintended motion. For example, after working on a hydraulic system, the hoses may have been miss-connected or full of air, and the system may not move as you intend. Keep yourself and others safe. If someone else suggests caution, reconsider the motion possibilities and maybe move your observation point. Similarly, if you see someone about to do something unsafe, speak up. It only takes a few minutes to reconsider, but a serious injury may take months to get over, or may be permanent.

If you find a leaking fitting, do not try to tighten it while it has pressure on it. At high pressure, the pressure stresses in the fitting added to the torque stresses may fail the fitting. If the fitting fails, your attempt to save a few minutes has just made a minor problem a very major one. In addition, if your system uses stainless steel fittings that are more prone to galling, the combined pressure and torque stresses may result in galling the thread, and again, a minor problem has become a major problem requiring, possibly, a new fitting and hose or tube. The point being made here is “Don’t tighten a fitting with pressure on the line.”

A high pressure jet of hydraulic fluid can penetrate the skin. Always approach a system with caution, looking for leaks or other problems. Do not look for a leak by feeling with your hand. Use a mechanic’s mirror or some other method of locating a leak. Once the leak is found, remove pressure from the line before working on it as suggested by the preceding paragraph.

Never ‘ride a load’ unless the system has been man rated. A hoist may be rated for thousands of pounds, but if it is not man rated, it is not safe to ride. Similarly, after you have worked on a rated system, it is not again man rated until it is completely checked out by cycling through all functions sufficiently to assure proper operation.

Some hydraulic fluids are aggressive to tender skin and membranes. Do not touch eyes, mouth, nose, or other tender areas with hydraulic fluid on your hands. Be sure to wash hands before eating. Be sure to wash hands before bathroom breaks. Violation of these suggestions may be very painful.

Resources

- Crane Co. [www.craneco.com/Category/200/Purchase-Flow-of-Fluids.html]
- Eaton Vickers
  www.eaton.com/Eaton/ProductsServices/Aerospace/Hydraulics/index.htm
- Parker *www.**parker**.com/*
- Rexroth pumps/motors *www.bosch**rexroth**-us.com/*
- Western filters *www.**westernfilterco**.com/*
- Pal filters *www.pall.com/main/Home.page/*
- Parker O-ring source locally or:
  www.parker.com/literature/ORD%205700%20Parker_O-Ring_Handbook.pdf
- Eaton Aeroquip hose/tubes/fittings:
  www.eaton.com/Eaton/ProductsServices/Aerospace/Hydraulics/PCT_249117
- Shell Fluids source locally or:
  www.milspecproducts.com/Brands/Royco?gclid=CMP0iNXq0bMCFayPPAod_AsAHQ
- Hydraulics magazine *www.**hydraulicspneumatics**.com/*
- Kepner *www.**kepner**.com/*
- Lee *www.theleeco.com/LEEWEB2.NSF*
- RTL *www.real-timelabs.com/*
- *Canyon Engineering* *www.canyonengineering.com/*
- Crissair: *www.crissair.com/*
- Moog servo-valves *www.**moog**.com/*
- Abex Servo-valves
  www.parker.com/portal/site/PARKER/menuitem.7100150cebe5bbc2d6806710237ad1ca/?vgnextoid=f5c9b5bbec622110VgnVCM10000032a71dacRCRD&vgnextfmt=EN&vgnextdiv=&vgnextcatid=1537927&vgnextcat=SERVOVALVES&Wtky=VALVES
- Sun Valve
  www.sunhydraulics.com/cmsnet/sun_homepage.aspx?lang_id=1
- Ausco Valves *www.auscoinc.com/index.html*
- Circle Seal *www.circle-seal.com/*