How to Solve and Prevent Hydraulic Problems

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Introduction

I've spent the better part of 16 years working in and running hydraulic repair shops i.e. rebuilding hydraulic components. During this time I kept seeing the same pattern: Failed hydraulic component comes into shop along with concerned customer who wants to know why it has failed after a relatively short time in service. Based on what I saw after tear-down, I would explain the cause of failure - for example, high contamination levels, wrong oil viscosity, high temperature operation, cavitation, faulty circuit protection devices and so on. Customer leaves thousands of dollars poorer with rebuilt component and a hard-learned lesson on hydraulic equipment maintenance.

For as long as there are hydraulic equipment owners, mechanics and maintenance people out there who believe that hydraulics don't require any special kind of attention, this cycle will continue. In an effort to bridge the knowledge gap on what needs to be done to get maximum life from hydraulic components, I have written Insider Secrets to Hydraulics and it's sequel, Preventing Hydraulic Failures.

Over the past 30 years, the performance, sophistication and operating pressures of hydraulic equipment have increased significantly. This is particularly true in the case of mobile hydraulic equipment. As a result, modern hydraulic equipment is not only more expensive to fix when it breaks, proactive maintenance is imperative to maximize service life and minimize operating costs. It's not realistic to expect (as many equipment owners do) to run a hydraulic machine for 10,000 hours, without checking anything more than the fluid level, and not have any problems.

Six routines must be followed in order to minimize the chances of your hydraulic equipment suffering costly, premature component failures and unscheduled downtime:

- Maintain fluid cleanliness;
- Maintain fluid temperature and viscosity within optimum limits;
- Maintain hydraulic system settings to manufacturers' specifications;
- Schedule component change-outs before they fail;
- Follow correct commissioning procedures; and
- Conduct failure analysis.

An effective, proactive maintenance program requires time, effort and some expense to implement. But it is cost-effective. The investment is quickly recovered through savings as a result of improved machine performance, increased component life, increased fluid life, reduced downtime and fewer repairs.

So if you own, operate or maintain hydraulic equipment, are serious about minimizing your running costs and your current maintenance practices are unsophisticated or non-existent, start implementing a proactive maintenance program today.

In the meantime, it is my hope that the following tutorials will assist you to solve or better still, prevent problems with your hydraulic equipment. And if you haven't done so already, you should subscribe to my 'Inside Hydraulics' Newsletter at www.HydraulicSupermarket.com to receive regular tips and articles relating to the maintenance, repair and overhaul of hydraulic equipment.

Anatomy of a hydraulic pump failure

I was asked recently to give a second opinion on the cause of failure of an axial piston pump. The hydraulic pump had failed after a short period in service and my client had pursued a warranty claim with the manufacturer. The manufacturer rejected the warranty claim on the basis that the failure had been caused by contamination of the hydraulic fluid. The foundation for this assessment was scoring damage to the valve plate (Figure 1).

Figure 1. Scoring damage to valve plate

How does contamination cause this type of damage to a hydraulic pump?

When hydraulic fluid is contaminated with hard particles that are the same size as the clearance between two lubricated surfaces, a process known as three-body abrasion occurs. Three-body abrasion results in scoring and heavy wear of sliding surfaces (Figure 2).

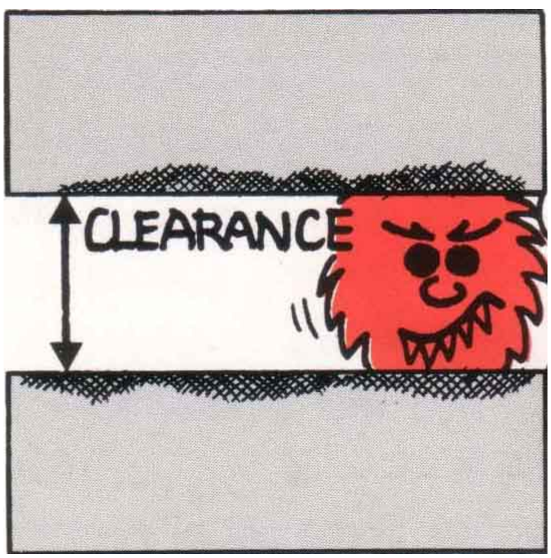


Figure 2. The process of three-body abrasion

What other explanations are there for this type of damage?

In axial piston designs, the cylinder barrel is hydrostatically loaded against the valve plate. The higher the operating pressure, the higher the hydrostatic force holding the cylinder barrel and valve plate in contact. However, if operating pressure exceeds design limits or if the valve plate is not in proper contact with the cylinder barrel, the cylinder barrel separates from the valve plate. Once separation occurs, the lubricating film is lost, the two surfaces come into contact and a process known as two-body abrasion occurs.

A major clue that the damage to the valve plate was not caused by contamination in this case, is the pattern of wear. Notice that the scoring (bright areas) is confined to the inner and outer edges of the sliding surface of the valve plate (see Figure 1). If the scoring had been caused by three-body abrasion, the damage would be more evenly distributed across the entire surface, with the areas between the pressure kidneys at the top of the picture, likely to exhibit the heaviest damage.

The pattern of wear on the valve plate is consistent with two-body abrasion resulting from uneven contact between the valve plate and cylinder barrel, caused by warping of the valve plate and/or separation. Examination of the sliding surface of the cylinder barrel (Figure 3) supports this assessment. Notice that the scoring of the cylinder barrel is heaviest top right of the picture and lightest bottom left. Examination of the head of the hydraulic pump also revealed uneven contact between the valve plate and head.

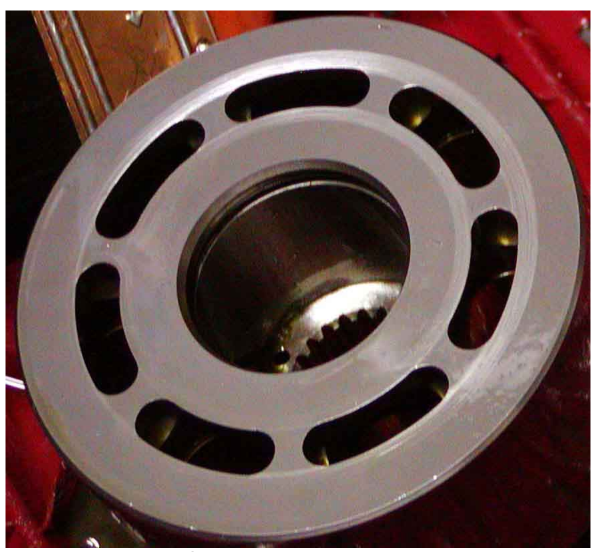


Figure 3. Scoring damage to cylinder barrel

Root cause of failure

Although the valve plate was flat, its locating dowel was holding it off the head on one side (center right of Figure 1). This in turn was causing the valve plate to be tilted against the cylinder barrel, resulting in uneven loading, separation and two-body abrasion of the two surfaces. The root cause of this hydraulic pump failure was not contamination; but rather improper assembly at the factory.

For more information on hydraulic failures and how to prevent them, read Preventing Hydraulic Failures.

Hydraulic cylinder failure caused by the 'diesel effect'

I was recently engaged by a client to conduct failure analysis on a large (and expensive) hydraulic cylinder off an excavator. This hydraulic cylinder had been changed-out due to leaking rod seals after achieving only half of its expected service life.

Inspection revealed that apart from the rod seals, which had failed as a result of the 'diesel effect', the other parts of the hydraulic cylinder were in serviceable condition.

What is the 'diesel effect'?

The diesel effect occurs in a hydraulic cylinder when air is drawn past the rod seals, mixes with the hydraulic fluid and explodes when pressurized.

How does this affect a hydraulic cylinder?

When a double-acting hydraulic cylinder retracts under the weight of its load, the volume of fluid being demanded by the rod side of the cylinder can exceed the volume of fluid being supplied by the pump.

When this happens, a negative pressure develops in the rod side of the hydraulic cylinder, which usually results in air being drawn into the cylinder past its rod seals. This occurs because most rod seals are designed keep high-pressure fluid in and are not designed to keep air out. The result of this is aeration - the mixing of air with the hydraulic fluid.

Aeration causes damage through loss of lubrication and overheating, and when a mixture of air and oil is compressed it can explode, damaging the hydraulic cylinder and burning its seals. As you have probably gathered, the term 'diesel effect' is a reference to the combustion process in a diesel engine.

In the example described above, the cause of the aeration was a faulty 'float' valve. The function of a float valve on a hydraulic excavator is to allow the boom or arm to be lowered rapidly under its own weight.

When activated, this valve connects the ports of the hydraulic cylinder together allowing it to retract under the weight of the boom or arm. The fluid displaced from the piston side of the cylinder is directed with priority to the rod side, before any excess volume is returned to the hydraulic reservoir. An orifice controls the speed with which the hydraulic cylinder retracts.

If this valve malfunctions or is set incorrectly, a negative pressure can develop on the rod side of the hydraulic cylinder, causing air to be drawn past the rod seals, leading to failure of the cylinder.

How can this type of failure be prevented?

This example highlights the importance of checking the operation and adjustment of circuit protection devices at regular intervals. As in this case, if the faulty float valve had been identified early enough, the failure of this hydraulic cylinder and the significant expense of its repair could have been prevented.

For more information on hydraulic failures and how to prevent them, read Preventing Hydraulic Failures.

Hydraulic ram leak caused by operator error

A client recently asked me to explain a seal failure on a hydraulic ram. The ram had been removed from a hydraulic lift due to a leaking rod seal, but upon inspection, both the rod seal and the surface of the rod were found to be in serviceable condition.

What is a hydraulic ram?

A hydraulic ram is a single-acting hydraulic cylinder in which fluid pressure acts on the cross-section of the rod i.e. it has no piston (Figure 1).

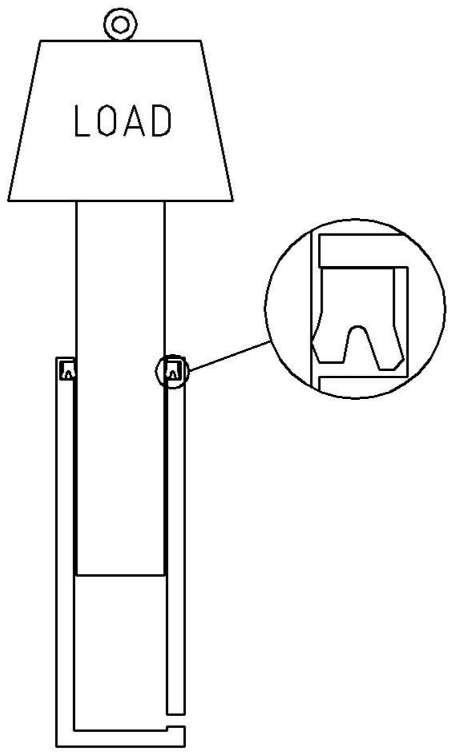


Figure 1. Hydraulic ram sectional view showing U-cup seal profile.

Failure investigation

Because inspection of the rod seal (U-cup type) and the rod's surface didn't reveal any obvious cause of failure, I asked the operator to describe the nature of the leak. He explained that during his morning inspections he had noticed that there was always a trickle of hydraulic fluid down the side of the ram.

Further investigation revealed that the current operator had only been assigned to the machine several weeks earlier. So I asked the operator to explain how he left the hydraulic lift at night. He advised that after shutting down he always relaxed the hydraulics (released the load off the hydraulic ram). This revealed the most likely explanation for the nuisance leak.

Seal energization

To seal effectively, a U-cup seal relies on hydraulic fluid pressure to energize the lips of the seal against the rod and seal groove (Figure 1 inset). Releasing the load-induced pressure from the hydraulic ram after shutdown effectively de-energizes the seal. Once the seal is de-energized, a gradual increase in the volume of fluid in the ram due to thermal expansion can result in fluid leaking past the seal. This gradual loss of fluid prevents development of sufficient pressure to effectively energize the seal, so the leak continues until the temperature, and therefore volume, of the fluid in the hydraulic ram stabilizes.

Root cause of failure

I advised my client that the practice of taking the load of the hydraulic ram after shutdown was the most likely cause of the leak. This being the case, there were two possible solutions. Discontinue the practice or change the seal profile to an energized U-cup (a U-cup that has an O-ring fitted in the 'U' to pre-energize the lips of the seal).

The root cause of the problem was confirmed when, without changing the seal profile, rod seal leakage was eliminated by discontinuing the practice of unloading the hydraulic ram. **Warning!** In certain situations, leaving loads suspended on hydraulic equipment can pose a safety hazard. For this reason, it is recommended that a safety risk assessment be carried out on a case-by-case basis before adopting this practice.

Hydraulic pump and motor case drains - filter with caution

One of our readers wrote to me recently with the following question: "We have recently been involved in designing and building a hydraulic machine. The system has three, separate circuits each with an axial-piston pump and a common reservoir. Case drain filtration was included to reduce the possibility of cross contamination if a failure occurs. After contacting the pump manufacturer I was led to believe that it isn't the norm, but if the pressure drop across the filter is kept to less than 30 PSI it will be OK. This just forces filter maintenance. What filter beta or micron rating should be used?"

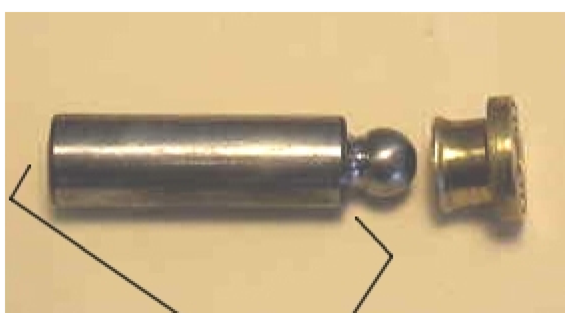
Installing filters on piston pump and motor case drain lines can result in excessive case pressure, which causes seal failure and mechanical damage.

Seal failure

High case pressure results in excessive load on the lip of the shaft seal. This causes the seal lip to wear a groove in the shaft, which eventually results in leakage past the seal. If case pressure exceeds the shaft seal's design limits, instantaneous failure can occur. The subsequent loss of oil from the case can result in damage through inadequate lubrication.

Mechanical damage

The effect of high case pressure on axial piston pumps is the same as excessive vacuum at the pump inlet. Both conditions put the piston-ball and slipper-pad socket in tension during inlet (see below). This can cause buckling of the piston retaining plate and/or separation of the slipper from the piston, resulting in catastrophic failure.



Vacuum here...
or case pressure here...
puts the piston-ball and slipper-pad socket in tension.

High case pressure can cause the pistons of radial piston motors to be lifted off the cam. This can occur in operation during the outlet cycle. The pistons are then hammered back onto the cam during inlet, destroying the motor. If residual case pressure remains high when the motor is stopped, loss of contact between the pistons and cam can allow the motor to freewheel, resulting in uncontrolled machine movement.

To avoid these problems, pump and motor case drain lines should be returned to the reservoir through dedicated penetrations. These penetrations must be higher than the unit's case port and be connected to a drop-pipe inside the reservoir that extends below minimum fluid level. For the reasons outlined above, filters are not recommended on case drain lines. While this does allow a small percentage of fluid to return to the reservoir unfiltered, in most applications the contamination risk is low and can be effectively managed using oil analysis and other condition-based maintenance practices.

Filter with caution

If a filter is fitted to a pump or motor drain line, I recommend a 125-micron screen, grossly oversized for the maximum expected flow rate. The filter housing must incorporate a bypass valve with an opening pressure lower than the maximum, allowable case pressure for the particular component (typically 5-15 PSIG). Installing a gauge or transducer upstream of the filter for monitoring case pressure is also advisable. To learn more about the failure modes of hydraulic components and how to avoid them, read *Preventing Hydraulic Failures* available here.

The value of the humble hydraulic symbol

I am regularly involved in troubleshooting problems with hydraulic equipment. In these situations, there are two things I always do before reaching for my test gear. The first is to conduct a visual inspection of the hydraulic system, checking all the obvious things that could cause the problem in question (never overlook the obvious). The second is to ask for the schematic diagram for the hydraulic circuit.

What is a hydraulic schematic diagram?

A hydraulic schematic diagram is a line drawing composed of hydraulic symbols that indicate the types of components the hydraulic circuit contains and how they are interconnected.

What makes a hydraulic schematic diagram valuable?

A schematic diagram is a 'road map' of the hydraulic system and to a technician skilled in reading and interpreting hydraulic symbols, is a valuable aid in identifying possible causes of a problem. This can save a lot of time and money when troubleshooting hydraulic problems.

If a schematic diagram is not available, the technician must trace the hydraulic circuit and identify its components in order to isolate possible causes of the problem. This can be a time-consuming process, depending on the complexity of the system. Worse still, if the circuit contains a valve manifold, the manifold may have to be removed and dismantled - just to establish what it's supposed to do. Reason being, if the function of a component within a hydraulic system is not known, it can be difficult to discount it as a possible cause of the problem. The humble hydraulic symbol eliminates the need to 'reverse engineer' the hydraulic circuit.

Where are all the hydraulic schematic diagrams?

As most hydraulic technicians know, there's usually a better than even chance that a schematic diagram will not be available for the machine they've been called in to troubleshoot. This is unlikely to bother the technician because it is the machine owner who pays for its absence.

Where do all the hydraulic schematic diagrams go? They get lost or misplaced, they don't get transferred to the new owner when a machine is bought second hand and in some cases they may not be issued to the machine owner at all. Why? Because generally speaking, hydraulic equipment owners don't place a lot of value on them.

So if you're responsible for hydraulic equipment and you don't have schematic diagrams for your existing machines, try to obtain them - before you need them. And ensure that you are issued with schematic diagrams for any additional hydraulic machines you acquire. It will save you money in the long run.

Learn how to read hydraulic symbols and interpret hydraulic schematics.

Compression and decompression of hydraulic fluid

One of our readers wrote to me recently regarding the following problem:

"I have a problem with a large garbage compactor. Each and every time the valve cycles the hydraulic cylinder, there is a loud bang. What are the possible causes?"

Assuming this noise is being generated by the hydraulics, i.e. it is not a symptom of a mechanical problem, its likely cause is uncontrolled decompression of the hydraulic fluid.

Hydraulic fluid bulk modulus and decompression

This problem arises because hydraulic fluid is not perfectly rigid. The ratio of a fluid's decrease in volume as a result of increase in pressure is given by its bulk modulus of elasticity. The bulk modulus for hydrocarbon-based hydraulic fluids is approximately 250,000 PSI, (17,240 bar) which results in a volume change of around 0.4% per 1,000 PSI (70 bar). The formula for calculating the volume change of a hydraulic fluid under pressure using its bulk modulus is available here. When the change in volume exceeds 10 cubic inches (160 cubic centimeters) decompression must be controlled.

The compression of hydraulic fluid results in storage of energy, similar to the potential energy stored in a compressed spring. Like a compressed spring, compressed fluid has the ability to do work. If decompression is not controlled, the stored energy dissipates instantaneously. This sudden release of energy accelerates the fluid, which does work on anything in its path. Uncontrolled decompression stresses hydraulic hose, pipe and fittings, creates noise and can cause pressure transients that damage hydraulic components.

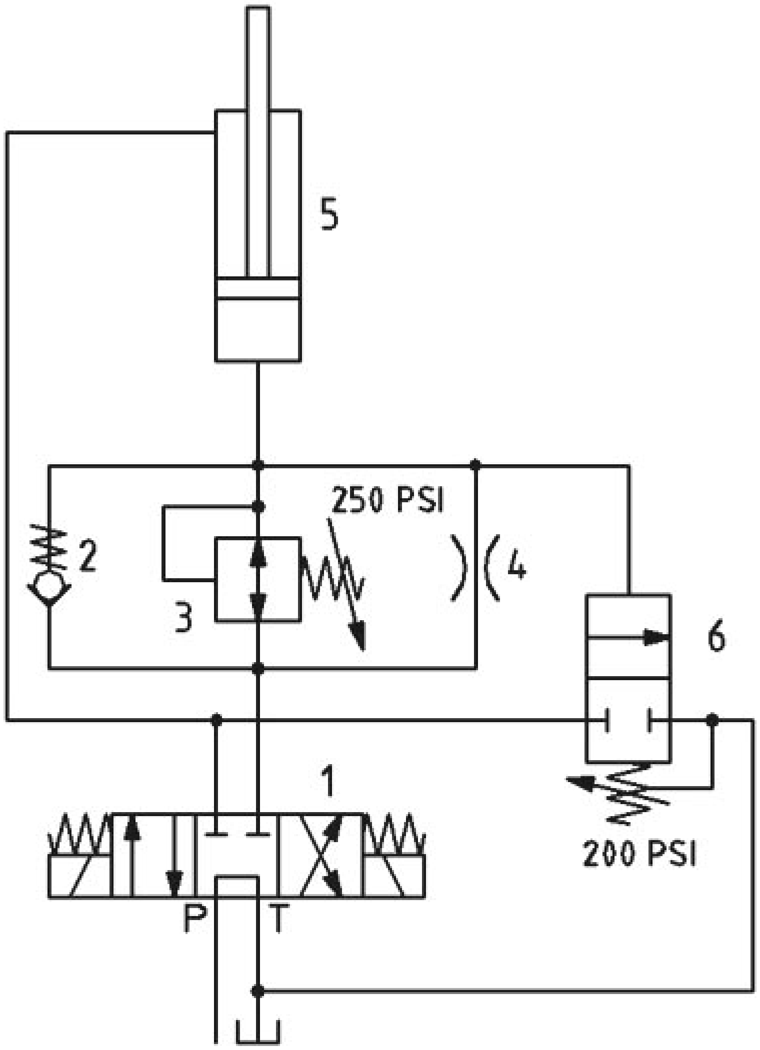
Troubleshooting decompression problems

Decompression is an inherent problem in hydraulic press applications, which have large volume cylinders operating at high pressures (a garbage compactor is effectively a press). Although hydrocarbon-based hydraulic fluids compress 0.4% - 0.5% by volume per 1,000 PSI, in actual application it is wise to calculate compression at 1% per 1,000 PSI. This compensates for the elasticity of the cylinder and conductors and a possible increase in the volume of air entrained in the fluid.

So if the combined captive volume of the hydraulic cylinder and conductors on our garbage compactor was 10 gallons and operating pressure was 5,000 PSI, the volume of compressed fluid would be 0.5 gallons (10 x 0.01 x 5). This equates to potential energy of around 33,000 watt-seconds. If the release of this amount of energy is not controlled, you can expect to hear a bang!

Controlling decompression

Decompression is controlled by converting the potential energy of the compressed fluid into heat. This is achieved by metering the compressed volume of fluid across an orifice. A simple decompression control circuit, which will eliminate the bang from our garbage compactor, is shown below.



When the directional control valve (1) is activated to extend the hydraulic cylinder (5), fluid enters the cylinder via the check valve (2). Pressurization of the cylinder during extension closes the pressure reducing valve (3), so that when the directional control valve (1) is activated to retract the cylinder (5) the compressed volume of fluid is metered across the orifice (4). When pressure upstream of the orifice (4) falls below the setting of the pressure reducing valve (3) the remaining fluid in the cylinder flows back to tank across the pressure reducing valve (3). The sequence valve (6) prevents pressurization of the rod side of the cylinder before the piston side has decompressed and the pressure reducing valve (3) has opened.

Hydraulic cylinder rods turn black

Black nitride is a relatively recent alternative to the hard-chrome plated hydraulic cylinder rod. With reports of achieved service life three times that of conventional chrome, longer seal life and comparable cost, black nitride rods for hydraulic cylinders are an option that all hydraulic equipment users should be aware of.

Black nitriding is an atmospheric furnace treatment developed and patented in the early 1980's. It combines the high surface hardness and corrosion resistance of nitriding with additional corrosion resistance gained by oxidation. The process begins with the cleaning and super-polishing of the material to a surface roughness of 6 to 10 Ra. The steel bars or tubes are then fixed vertically, and lowered into an electrically heated pit furnace.

The furnace sequence involves nitrocarburizing the steel at temperatures up to 1150ºF in an ammonia atmosphere. The steel's surface is converted to iron nitride to a depth of typically 0.001". Atmospheric oxidizing is employed to produce a black, corrosion resistant surface film.

The process generates a thin (0.001") uniform and extremely hard (64 to 71 Rc) iron nitride layer infused with a film of iron oxide. Beneath the iron nitride layer is a nitrogen-enriched, hardened diffusion zone. The diffusion zone functions as a lightly hardened case with a hardness gradient ranging from around 55 Rc just below the iron nitride layer to approximately 40 Rc at a depth of 0.015". Testing has verified that black nitride bar machines and welds as well as hard chrome plated stock.

Available in standard diameters up to 5" (127 mm), black nitride hydraulic cylinder rods offer the following benefits over conventional hard chrome:

- Superior corrosion and wear resistance
- Better oil retention (longer seal life)
- Dimensional uniformity
- Dent resistant - without the need for induction hardening
- No pitting, flaking or micro-cracking
- Environmentally friendly process

Determining hydraulic pump condition using volumetric efficiency

I was recently asked to give a second opinion on the condition of a variable displacement hydraulic pump. My client had been advised that its volumetric efficiency was down to 80%. Based on this advice, he was considering having this hydraulic pump overhauled.

What is volumetric efficiency?

Volumetric efficiency is the percentage of theoretical pump flow available to do useful work. In other words, it is a measure of a hydraulic pump's volumetric losses through internal leakage. It is calculated by dividing the pump's actual output in liters or gallons per minute by its theoretical output, expressed as a percentage. Actual output is determined using a flow-tester to load the pump and measure its flow rate.

Because internal leakage increases as operating pressure increases and fluid viscosity decreases, these variables should be stated when stating volumetric efficiency. For example, a hydraulic pump with a theoretical output of 100 GPM, and an actual output of 94 GPM at 5000 PSI and 120 SUS is said to have a volumetric efficiency of 94% at 5000 PSI and 120 SUS. In practice, fluid viscosity is established by noting the fluid temperature at which actual output is measured and reading the viscosity off the temperature/viscosity graph for the grade of fluid in the hydraulic system.

What is the significance of volumetric efficiency?

As a hydraulic pump wears in service, internal leakage increases and therefore the percentage of theoretical flow available to do useful work (volumetric efficiency) decreases. If volumetric efficiency falls below a level considered acceptable for the application, the pump will need to be overhauled.

Calculating the volumetric efficiency of variable hydraulic pumps

The hydraulic pump in question had a theoretical output of 1,000 liters per minute at full displacement and maximum rpm. Its actual output was 920 liters per minute at 4,350 PSI and 100 SUS. When I advised my client that the pump's volumetric efficiency was in fact 92%, he was alarmed by the conflicting results. To try and explain the disparity, I asked to see the first technician's test report.

After reviewing this test report, I realized that the results actually concurred with mine, but had been interpreted incorrectly. The test had been conducted to the same operating pressure and at a fluid temperature within one degree of my own test, but at reduced displacement. The technician had limited the pump's displacement to give an output of 400 liters per minute at maximum rpm and no load (presumably the maximum capacity of his flow-tester). At 4,350 PSI the recorded output was 320 liters per minute. From these results, volumetric efficiency had been calculated to be 80% (320/400 x 100 = 80).

To help understand why this interpretation is incorrect, think of the various leakage paths within a hydraulic pump as fixed orifices. The rate of flow through an orifice is dependant on the diameter (and shape) of the orifice, the pressure drop across it and fluid viscosity. This means that if these variables remain constant, the rate of internal leakage remains constant, independent of the pump's displacement.

Note that in the above example, the internal leakage in both tests was 80 liters per minute. If the same test were conducted with pump displacement set to 100 liters per minute at no load, pump output would be at 20 liters per minute at 4,350 PSI - all other things being equal. This means that this pump has a volumetric efficiency of 20% at 10% displacement, 80% at 40% displacement and 92% at 100% displacement. As you can see, if actual pump output is measured at less than full displacement (or maximum rpm) an adjustment needs to be made when calculating volumetric efficiency.

Time for an overhaul?

In considering whether it is necessary to have this hydraulic pump overhauled, the important number is volumetric efficiency at 100% displacement, which is within acceptable limits. If my client had based their decision on volumetric efficiency at 40% displacement, they would have paid thousands of dollars for unnecessary repairs.

Solving hydraulic system overheating problems

I was asked recently to investigate and solve an overheating problem in a mobile hydraulics application. The hydraulic system comprised a diesel-hydraulic power unit, which was being used to power a pipe-cutting saw. The saw was designed for sub-sea use and was connected to the hydraulic power unit on the surface via a 710-foot umbilical. The operating requirements for the saw were 24 gpm at 3000 psi.

Why do hydraulic systems overheat?

Heating of hydraulic fluid in operation is caused by inefficiencies. Inefficiencies result in losses of input power, which are converted to heat. A hydraulic system's heat load is equal to the total power lost (PL) through inefficiencies and can be expressed as:

PLtotal = PLpump + PLvalves + PLplumbing + PLactuators
If the total input power lost to heat is greater than the heat dissipated, the hydraulic system will eventually overheat.

Hydraulic fluid temperature - how hot is 'too hot'?

Hydraulic fluid temperatures above 180°F (82°C) damage most seal compounds and accelerate degradation of the oil. While the operation of any hydraulic system at temperatures above 180°F should be avoided, fluid temperature is too high when viscosity falls below the optimum value for the hydraulic system's components. This can occur well below 180°F, depending on the fluid's viscosity grade.

Maintaining stable hydraulic fluid temperature

To achieve stable fluid temperature, a hydraulic system's capacity to dissipate heat must exceed its inherent heat load. For example, a system with continuous input power of 100 kW and an efficiency of 80% needs to be capable of dissipating a heat load of at least 20 kW. It's important to note that an increase in heat load or a reduction in a hydraulic system's capacity to dissipate heat will alter the balance between heat load and dissipation.

Returning to the above example, the hydraulic power unit had a continuous power rating of 37 kW and was fitted with an air-blast heat exchanger. The exchanger was capable of dissipating 10 kW of heat under ambient conditions or 27% of available input power (10/37 x 100 = 27). This is adequate from a design perspective. The performance of all cooling circuit components were operating within design limits.

Pressure drop means heat

At this point it was clear that the overheating problem was being caused by excessive heat load. Concerned about the length of the umbilical, I calculated its pressure drop. The theoretical pressure drop across 710 feet of ¾" pressure hose at 24 gpm is 800 psi. The pressure drop across the same length of 1" return hose is 200 psi. The formula for these calculations is available here. The theoretical heat load produced by the pressure drop across the umbilical of 1,000 psi (800 + 200 = 1000) was 10.35 kW. The formula for this calculation is available here.

This meant that the heat load of the umbilical was 0.35 kW more than the heat dissipation capacity of the hydraulic system's heat exchanger. This, when combined with the system's normal heat load (inefficiencies) was causing the hydraulic system to overheat.

Beat the heat

There are two ways to solve overheating problems in hydraulic systems:

- decrease heat load; or
- increase heat dissipation.

Decreasing heat load is always the preferred option because it increases the efficiency of the hydraulic system. In the above example, the heat load of the umbilical alone was nearly 30% of available input power, a figure that would normally be considered unacceptable. Decreasing this heat load to an acceptable level would have involved reducing the pressure drop, by replacing the pressure and return lines in the umbilical with larger diameter hoses. The cost of doing this for what was a temporary installation meant that, in this case, the most economical solution was to install additional cooling capacity in the circuit

Continuing to operate a hydraulic system when the fluid is over-temperature is similar to operating an internal combustion engine with high coolant temperature. Damage is guaranteed. Therefore, whenever a hydraulic system starts to overheat, shut it down, identify the cause and fix it.

Hydraulic cylinders - checking rod straightness

As a product group, hydraulic cylinders are almost as common as pumps and motors combined. They are less complicated than other types of hydraulic components and are therefore relatively easy to repair. As a result, many hydraulic equipment owners or their maintenance personnel repair hydraulic cylinders in-house. An important step in the repair process that is often skipped by do-it-yourself repairers, is the checking of rod straightness.

How do bent rods affect hydraulic cylinders?

Bent rods load the rod seals causing distortion, and ultimately premature failure of the hydraulic cylinders seals.

Allowable run-out

Rod straightness should always be checked when hydraulic cylinders are being re-sealed or repaired. This is done by placing the rod on rollers and measuring the run-out with a dial gauge. The rod should be as straight as possible, but a run-out of 0.5 millimeters per linear meter of rod is generally considered acceptable.

Straightening hydraulic cylinder rods

In most cases, bent rods can be straightened in a press. It is sometimes possible to straighten hydraulic cylinder rods without damaging the hard-chrome plating, however if the chrome is damaged, the rod must be either re-chromed or replaced.

Hydraulic pump life cut short by particle contamination

I was recently asked to conduct failure analysis on a hydraulic pump that had an expected service life of 10,000 hours. The pump had been removed from its machine after achieving only 2000 hours in service.

Analysis revealed that this hydraulic pump hadn't actually failed - it had been 'worn-out' through erosive wear caused by contaminated hydraulic oil.

What is 'contaminated hydraulic fluid'?

Contaminants of hydraulic fluid include solid particles, air, water or any other matter that impairs the function of the fluid.

How does contamination affect a hydraulic pump?

Particle contamination accelerates wear of hydraulic components. The rate at which damage occurs is dependent on the internal clearance of the components within the system, the size and quantity of particles present in the fluid, and system pressure.

Particles larger than the component's internal clearances are not necessarily dangerous. Particles the same size as the internal clearances cause damage through friction. However, the most dangerous particles in the long term are those that are smaller than the component's internal clearances.

Particles smaller than 5 microns are highly abrasive. If present in sufficient quantities, these invisible 'silt' particles cause rapid wear, destroying hydraulic pumps and other components.

How can this type of hydraulic pump failure be prevented?

While the type of failure described above is unusual in properly designed hydraulic systems that are correctly maintained, this example highlights the importance of monitoring hydraulic fluid cleanliness levels at regular intervals.

As in this case, if the high levels of silt particles present in the hydraulic fluid had been identified and the problem rectified early enough, the damage to this hydraulic pump and the significant expense of its repair could have been avoided.

Learn more about contamination and hydraulic oil cleanliness.

Hydraulic press failure illustrates the importance of scheduling change-outs

A manufacturing company recently hired me to check the performance of four piston pumps operating a large hydraulic press. The hydraulic pumps had clocked over 10,000 hours in service and the customer's concern was that if pump performance was down, production would be too.

My test results revealed that the performance of all four pumps was within acceptable limits. In my report, I advised my client that there would only be a minimal increase in productivity if the pumps were replaced. I further advised that the change-out of all four pumps should be scheduled urgently.

The foundation for this recommendation was that the pumps had exceeded their expected service life and in the absence of an effective condition-based maintenance program, the probability of an in-service bearing failure was significantly increased.

When a hydraulic component fails in service, large amounts of metallic particles are generated. These particles circulate in the hydraulic fluid, often causing damage to other components before the system's filters can remove them. In extreme cases, the contamination load can clog the hydraulic filters, which results in unfiltered fluid being circulated through the system.

A component that fails in service is almost always more expensive to rebuild than a component that is removed from service in a pre-failed condition. A failure in service usually results in mechanical damage to the internal parts of the component. As a consequence, parts that would have been serviceable have to be replaced. In extreme cases, components that would have been economical to repair become uneconomical to repair, increasing the cost of component replacement by up to 40%.

The client took my advice, but unfortunately, a bearing failed in one of the pumps before all of the change-outs were completed. A piece of cage from the failed bearing found its way into the main hydraulic cylinder, causing $6,000 damage. The pump that failed cost 50% more to rebuild than the three units that were removed from service in pre-failed condition. Not to mention the downtime cost of the hydraulic press.

The additional repair costs in this case were significant and could have been avoided, if the pumps had been changed-out once they achieved their expected service life.

To minimize the chances of hydraulic components failing in service, the machine manufacturers' recommendations on expected service life should be used to schedule component change-outs. It may be possible to safely extend service life beyond that recommended through careful application of condition-based monitoring techniques, such as oil analysis (wear debris analysis). But unless an effective, predictive maintenance program is in place, running hydraulic components beyond their expected service life is false economy.

Hydraulic motors - how dry starts damage them

I was asked recently to conduct failure analysis on a hydraulic motor that was the subject of a warranty claim. The motor had failed after only 500 hours in service, some 7,000 hours short of its expected service life.

Inspection revealed that the motor's bearings had failed through inadequate lubrication, as a result of the hydraulic motor being started with insufficient fluid in its case (housing).

A common misconception among maintenance personnel with limited training in hydraulics, is that because oil circulates through hydraulic components in operation, no special attention is required during installation, beyond fitting the component and connecting its hoses. Nothing could be further from the truth.

After this hydraulic motor was installed, its case should have been filled with clean hydraulic oil prior to start-up. Starting a piston-type motor or pump without doing so, is similar to starting an internal combustion engine with no oil in the crankcase - premature failure is pretty much guaranteed.

Some of you may be thinking that the case should fill with hydraulic fluid through internal leakage. In most cases it will, but not before the motor or pump has been damaged. In many cases, this damage may not show itself until the component fails prematurely, hundreds or even thousands of service hours after the event.

In this particular example the warranty claim was rejected on the basis of improper commissioning and the customer was lumbered with an expensive repair bill.

How can this type of failure be prevented?

This example highlights the importance of following proper commissioning procedures when installing hydraulic components. As in this example, if the case of the hydraulic motor had been filled with fluid prior to start-up, the failure of this motor and the significant expense of its repair could have been prevented.

For more information on hydraulic failures and how to prevent them, read Preventing Hydraulic Failures.

Hydraulic valve failure caused by cavitation

A client recently asked me to advise them on the possibility of repairing a large hydraulic valve off a 400 ton excavator, used in open-cut mining.

The hydraulic valve in question was a spool-type directional control. It had been badly damaged as a result of cavitation, which had occurred over a long period in service.

What is cavitation?

Cavitation occurs when the volume of hydraulic fluid demanded by any part of a hydraulic circuit exceeds the volume of fluid being supplied.

This causes the absolute pressure in that part of the circuit to fall below the vapor pressure of the hydraulic fluid. This results in the formation of vapor bubbles within the fluid, which implode when compressed.

Cavitation causes metal erosion, which damages hydraulic components and contaminates the hydraulic fluid. In extreme cases, cavitation can result in major mechanical failure of pumps and motors.

While cavitation commonly occurs in the hydraulic pump, it can occur just about anywhere within a hydraulic circuit.

In the hydraulic valve described above, the metal erosion in the body of the valve was so severe that the valve was no longer serviceable. The valve had literally been eaten away from the inside, as a result of chronic cavitation.

In this particular case the cause of the cavitation was faulty anti-cavitation valves, which are designed to prevent this type of damage from occuring.

How can this type of failure be prevented?

This example highlights the importance of checking the operation and adjustment of circuit protection devices, including anti-cavitation and load control valves, at regular intervals.

As in this case, if the faulty anti-cavitation valves had been identified and replaced early enough, the damage to this hydraulic valve and the significant expense of its replacement could have been avoided.

For more information on hydraulic failures and how to prevent them, read Preventing Hydraulic Failures.

Hydraulic hose failure costs and prevention

Hydraulic hose has a finite service life, which can be reduced by a number of factors. From a maintenance perspective, little or no attention is usually paid to the hoses of a hydraulic system until a failure occurs.

Hydraulic hose failures cost more than the replacement hose. Additional costs can include:

- Clean up, disposal and replacement of lost hydraulic fluid.
- Collateral damage to other components, e.g. a hose failure on a hydrostatic transmission can result in loss of charge pressure and cavitation damage to the transmission pump and/or motor.
- Possible damage caused by the ingression of contaminants.
- Machine downtime.

What causes hydraulic hose failures?

Focus on the following points to extend hydraulic hose life and minimize the costs associated with hydraulic hose failures:

External damage

Hydraulic hose manufacturers estimate that 80% of hose failures are attributable to external physical damage through pulling, kinking, crushing or abrasion of the hose. Abrasion caused by hoses rubbing against each other or surrounding surfaces is the most common type of damage.

To prevent external damage, ensure all clamps are kept secure, pay careful attention to routing whenever a replacement hose is installed and if necessary, apply inexpensive polyethylene spiral wrap to protect hydraulic hoses from abrasion.

Multi-plane bending

Bending a hydraulic hose in more than one plane results in twisting of its wire reinforcement. A twist of five degrees can reduce the service life of a high-pressure hydraulic hose by as much as 70% and a seven degree twist can result in a 90% reduction in service life.

Multi-plane bending is usually the result of poor hose-assembly selection and/or routing but can also occur as a result of inadequate or unsecure clamping where the hose is subjected to machine or actuator movement.

Operating conditions

The operating conditions that a correctly installed hydraulic hose is subjected to will ultimately determine its service life. Extremes in temperature, e.g. high daytime operating temperatures and very cold conditions when the machine is standing at night, accelerate aging of the hose's rubber tube and cover.

Frequent and extreme pressure fluctuations, e.g. rock hammer on a hydraulic excavator, accelerate hose fatigue. In applications where a two-wire braid reinforced hydraulic hose meets the nominal working pressure requirement but high dynamic pressure conditions are expected, the longer service life afforded by a spiral reinforced hydraulic hose will usually more than offset the higher initial cost.

Hydraulic system troubleshooting - check the easy things first

In Part II of *Insider Secrets to Hydraulics*, I outline a logical approach to hydraulic system troubleshooting that begins with checking and eliminating the easy things first. The benefits of this approach are clearly illustrated by a troubleshooting situation I was involved in recently.

The machine in question had a complex hydraulic system, the heart of which comprised two engines driving ten hydraulic pumps. Six of the pumps were variable displacement pumps and four of these had electronic horsepower control.

The symptoms of the problem were slow cycle times in combination with lug-down of the engines (loss of engine rpm). The machine had just been fitted with a new set of pumps.

The diagnosis of the mechanic in charge was that the hydraulic system was tuned above the power curve of the engines i.e. the hydraulics were demanding more power than the engines could produce, resulting in lug-down of the engines and therefore slow cycle times. The other possible explanation of course, was that the engines were not producing their rated horsepower.

Due to the complexity of the hydraulic system, I knew that it would take around four hours to run a complete system check and tune-up. So in order to eliminate the easy things first, when I arrived on site I inquired about the condition of the engines and their service history. The mechanic in charge not only assured me that the engines were in top shape, he was adamant that this was a "hydraulic" problem.

Four hours later, after running a complete check of the hydraulic system without finding anything significant, I was not surprised that the problem remained unchanged. After a lengthy discussion, I managed to convince the mechanic in charge to change the fuel filters and air cleaner elements on both engines.

This fixed the problem. It turned out that a bad batch of fuel had caused premature clogging of the engine fuel filters, which were preventing the engines from developing their rated horsepower.

If the relatively simple task of changing the engine fuel filters had been carried out when the problem was first noticed, an expensive service call and four hours of downtime could have been avoided.

Learn more about hydraulic troubleshooting techniques.

Proactive maintenance for hydraulic cylinders

Damaged hydraulic cylinder rods and wiper seals are an eternal problem for users of hydraulic machinery. Dents and gouges on the surface of hydraulic cylinder rods reduce seal life and give dust and other contaminants an easy path into the hydraulic system. These silt-sized particles act like lapping compound, initiating a chain of wear in hydraulic components.

In response to this problem, a protective cylinder rod cover called *Seal Saver* has been developed and patented. *Seal Saver* is not a typical bellows boot you may already be familiar with. It is a continuous piece of durable material, which wraps around the cylinder and is closed with Velcro. It is then clamped onto the cylinder body and rod end. This makes installation simple with no disassembly of hydraulic cylinder components required.

Seal Saver forms a protective shroud over the cylinder rod as it strokes and prevents buildup of contaminants around the wiper seal - a common cause of rod scoring, seal damage and contaminant ingress. Research has shown that the cost to remove contaminants is ten times the cost of exclusion. This, combined with the benefits of extended hydraulic cylinder rod and seal life, makes *Seal Saver* a cost-effective, proactive maintenance solution.

Hydraulic fitting selection the key to leak-free hydraulic plumbing

Hydraulic fitting leaks are often considered to be an inherent characteristic of hydraulic machines. While this may have been true 30 years ago, advances in sealing technology and the development of reliable connection systems means that today, leak-free hydraulic plumbing is readily achievable.

Reliable Connections

Leak-free reliability begins at the design stage, when the type of hydraulic fitting is selected for port, tube-end and hose-end connections.

Ports - Connectors that incorporate an elastomeric seal such as UNO, BSPP and SAE 4-bolt flange offer the highest seal reliability. NPT is the least reliable type of connector for high-pressure hydraulic systems because the thread itself provides a leak path. The threads are deformed when tightened and as a result, any subsequent loosening or tightening increases the potential for leaks. In existing systems, pipe thread connections should be replaced with UNO or BSPP for leak-free reliability.

Tube and Hose Ends - ORFS tube and hose end connections feature the high seal reliability afforded by an elastomeric seal but, due to its cost, ORFS is not as widely used as compression fittings and JIC 37-degree flare.

Flared connections have gained widespread acceptance due to their simplicity and low cost. However, the metal-to-metal seal of the flare means that a permanent, leak-free joint is not always achieved, particularly in the case of tube-end connections.

Leaking flare joints can be eliminated using a purpose-built seal developed by Flaretite. The Flaretite seal is a stainless steel stamping shaped like a JIC nose, with concentric ribs that contain pre-applied sealant. When tightened, the ribs crush between the two faces of the joint, eliminating any misalignment and surface imperfections. The combination of the crush on the ribs and the sealant ensure that a leak-free joint is achieved.

Incorrect Torque

A common cause of leaks from flare joints is incorrect torque. Insufficient torque results in inadequate seat contact, while excessive torque can result in damage to the tube and fitting through cold working. The following is a simple method to ensure flare joints are correctly torqued:

1. Finger tighten the nut until it bottoms on the seat.
2. Using a permanent marker, draw a line lengthwise across the nut and fitting.

3. Wrench tighten the nut until it has been rotated the number of hex flats listed in the following table:

Tube Dash Size	Hex Flats
4	2.5
5	2.5
6	2.0
8	2.0
10	1.5 - 2.0
12	1.0
16	0.75 - 1.0
20	0.75 - 1.0
24	0.5 - 0.75

Vibration

Vibration can stress plumbing, affecting hydraulic fitting torque and causing fatigue. Tube is more susceptible than hose. If vibration is excessive, the root cause should be addressed. Ensure all conductors are adequately supported and if necessary, replace problematic tubes with hose.

Seal Damage

Having outlined the benefits of hydraulic fittings that incorporate an elastomeric seal, it is important to note that their reliability is contingent on fluid temperature being maintained within acceptable limits. A single over-temperature event of sufficient magnitude can damage all the seals in a hydraulic system, resulting in numerous leaks.

Conclusion

A leak-free hydraulic system should be considered the norm for modern hydraulic machines - not the exception. But the proper selection, installation and maintenance of hydraulic plumbing are essential to ensure leak-free reliability.

Converting 37-degree hydraulic fittings into zero-leak connections

The 37-degree hydraulic fitting is the world's most commonly used hydraulic connection. This popularity is due to its ease of fabrication, wide size range, adaptability to metric tubing and worldwide availability. For over fifty years, it has been the hydraulic fitting of choice for nearly hydraulic system manufacturers. As technology has improved and hydraulic system pressures have steadily increased, the flared fitting has become prone to leaks and drips, which results in dirty, sludge-covered systems that give hydraulics a bad name. The cost of hydraulic oil leaks to industry and the environment is only now being fully considered.

Alternative hydraulic fittings are gaining acceptance, most notably the 'O-Ring Face Seal' (ORFS). The ORFS fitting has significantly improved the integrity of hydraulic connections, but at a cost. It is larger in size, offers fewer adapter options, is more difficult to install (alignment must be perfect or O-ring extrusion occurs), has limited worldwide availability, is twice the price of a flared connection, and the O-ring is susceptible to environmental failure.

A new sealing device has propelled the old, flared connection into the new "no-leak" world demanded by modern hydraulic systems. The Flaretite Seal is a stainless steel stamping, designed with multiple, concentric rings. The entire seal is impregnated with a baked-on Loctite coating. When inserted into a flared hydraulic fitting, the concentric rings form multiple seals down the face of the flare and the Loctite coating fills minor imperfections. The sealing rings prevent environmental debris and aggressive cleaning solvents from attacking the sealing face and also protect the face from fretting, galling and over-tightening. And unlike the ORFS fittings, it will not fail during a fire. This prevents atomized hydraulic oil from fueling combustion.

Flaretite Seals are easy to install. They snap onto the male end of the flare and the hydraulic fitting is tightened in the usual way. The result is a permanent, zero-leak connection which exceeds the performance of today's ORFS fittings, at significantly less cost.

Hydraulic filters that do more harm than good - Part 1

Given that particle contamination of hydraulic fluid reduces the service life of hydraulic components, it would seem logical that a system can never have too many hydraulic filters. Well... not exactly.

Some hydraulic filters can actually do more harm than good and therefore their inclusion in a hydraulic system is sometimes misguided.

Pump inlet (suction) filters fall into this category. Inlet filters usually take the form of a 140 micron, mesh strainer which is screwed onto the pump intake penetration inside the hydraulic reservoir.

Inlet filters increase the chances of cavitation occurring in the intake line and subsequent damage to, and failure of the hydraulic pump. Piston-type pumps are particularly susceptible.

If the reservoir starts out clean and all fluid returning to the reservoir is filtered, inlet filters are not required since the hydraulic fluid will not contain particles large enough to be captured by a coarse mesh strainer.

What does this mean?

I generally recommend removing and discarding inlet filters where fitted. The one possible exception to this rule is charge pump intakes on hydrostatic transmissions. If in doubt consult the hydraulic pump manufacturer.

If you are involved in the design of hydraulic systems, think twice before fitting hydraulic filters to pump intake lines.

Hydraulic filters that do more harm than good - Part 2

In response to my previous article on hydraulic filters and the negative effects of suction strainers, one of our readers wrote the following:

"The one thing a suction strainer does that's worthwhile is to keep out the trash that gets dropped into the tank during service. We lost pumps to things like bolts that we know were not in the tank when it got built. The process of adding hydraulic fluid to the tank often doubles as the trash-installation function. The screens that are often installed in the fill neck usually get a hole poked through them so that oil will go in faster..."

A couple of years ago, I was involved in a case where the seals failed in the swivel on a hydraulic excavator. This allowed the automatic greasing system to pump grease into the hydraulic reservoir.

The grease clogged the suction strainers, which subsequently failed. The wire mesh from the suction strainers destroyed all four hydraulic pumps and several other components.

Had suction strainers not been fitted, it is likely that the grease would have eventually dissolved in the hydraulic fluid with minimal damage to any components.

My point is, I don't use this example as an argument **against** fitting suction strainers - because grease should **not** be in the reservoir.

Likewise, I do not consider trash exclusion to be a valid argument **for** fitting suction strainers - because nuts, bolts or similar debris should **not** be in the reservoir.

The sloppy operators that allow trash to drop into the reservoir are the same operators that never drain and clean the reservoir, and change the suction strainer. So the suction strainer clogs eventually and the pump fails through cavitation. Therefore, with or without the suction strainer, the pump is destined to fail prematurely.

The correct solution is not to allow trash to get into the reservoir. And this is fundamental to my recommendation to remove and discard suction strainers, where fitted.

Excessive vacuum at the pump inlet caused by suction strainers is a bigger threat to pump life in the long run, than trash that shouldn't be in the reservoir in the first place.

Hydraulic filters that do more harm than good - Part 3

In a previous article on hydraulic filters, I pointed out that all fluid returning to the reservoir should be filtered.

The one exception to this rule is the case drains of hydraulic piston pumps and motors. Connecting case drain lines to return filters can cause excessive case pressure, which has a number of damaging effects.

High case pressure results in excessive load on the lip of the shaft seal. This causes the seal lip to wear a groove in the shaft, which eventually results in a leaking shaft seal.

The effect of high case pressure on in-line piston pumps is the same as excessive vacuum at the pump inlet. Both conditions put the piston ball and slipper-pad socket in tension during intake.

In severe cases this can result in buckling of the piston retaining plate and/or separation of the bronze slipper from the piston, causing major failure.

Under certain conditions, high case pressure can cause the pistons of radial piston motors to be lifted off the cam during the outlet cycle. When this happens the pistons are hammered back onto the cam during inlet, destroying the motor.

What does this mean?

The case drain line of piston pumps and motors should be returned to the reservoir through a dedicated penetration below minimum fluid level. For the reasons described above, hydraulic filters are not recommended on case drain lines. However, if a filter is fitted it should be generously oversized to minimize back pressure in the pump or motor case. If in doubt, consult the hydraulic pump or motor manufacturer.

Hydraulic pump control problem highlights the value of testing

A client recently engaged me to design and build a hydraulic power unit for a specific application. The unit comprised a diesel engine driving an axial piston pump fitted with load sensing, power limiting and pressure limiting control.

What is a hydraulic power unit?

A hydraulic power unit comprises a prime mover (usually an electric motor or combustion engine), hydraulic pump, tank, filters and valves.

What is hydraulic pump load sensing control?

Load sensing control is so called because the load-induced pressure downstream of the directional control valve is sensed and hydraulic pump flow adjusted to maintain a constant pressure drop (and therefore flow) across the valve.

For example, let's say we have a hydraulic pump driving a winch thru a manual, directional valve. The operator summons the winch by moving the spool in the directional valve 20% of its stroke. The winch drum turns at five rpm. For clarity, imagine that the directional valve is now a fixed orifice. Flow across an orifice decreases as the pressure drop across it decreases. As load on the winch increases, the load-induced pressure downstream of the orifice (directional valve) increases. This decreases the pressure drop across the orifice, which means flow decreases and the winch slows down.

The load sensing control senses the load-induced pressure downstream of the orifice and adjusts hydraulic pump flow so that pressure upstream of the orifice increases by a corresponding amount. This keeps the pressure drop across the orifice (directional valve) constant, which keeps flow constant and in this case, winch speed constant.

Because the hydraulic pump only produces the flow demanded by the actuators, load sensing control is energy efficient (fewer losses to heat) and as demonstrated in the above example, provides more precise control.

What is hydraulic pump power limiting control?

A constant power or power limiting control operates by reducing the displacement, and therefore flow, from the hydraulic pump as pressure increases, so that the power rating of the prime mover is not exceeded. The advantage of this type of control is that more flow is available at lower pressures, so that the actuators can operate faster under light loads. This results in better utilization of the power available from the prime mover. The power limiting control overrides the load sensing control.

What is hydraulic pump pressure limiting control?

Pressure limiting control limits the maximum operating pressure of the hydraulic pump. Also referred to as a pressure compensator or pressure cut-off. The pressure limiting control overrides both the load sensing and power limiting controls.

Hydraulic pump control problem

A new hydraulic pump was ordered for the project from a leading manufacturer. When the hydraulic power unit was commissioned, the power limiting control was not functioning.

When advised of the problem, the manufacturer maintained that the pump had been tested prior to delivery and that the cause of problem therefore must be elsewhere in the circuit. Possible external causes were quickly checked and eliminated. While waiting for the manufacturer to respond to the problem, I checked the schematic diagram of the pump's control and noticed that a vital part was missing.

Plug-in hydraulic pump controls

The power limiting control on this particular hydraulic pump is a modular, screw-in cartridge fitted to the standard pump with load sensing and pressure limiting control. The power limiting cartridge is a relief valve with a link to the swash plate that increases spring bias as swash angle decreases. This relief valve limits load signal pressure depending on swash plate position. When the allowable power setting is reached, the relief valve intervenes to reduce the load pressure signal to the load sensing control. This results in a decrease in swash angle and therefore flow. The lower the swash angle and therefore flow, the higher the load signal pressure and therefore operating pressure permissible. Because power is a product of flow and pressure, this limits the power draw of the hydraulic pump.

If you examine the two hydraulic pump schematic diagrams closely, you will notice that other than the addition of a power limiting relief cartridge, there is a second difference. An orifice is shown just below the load sensing signal connection or X port. Without this orifice to limit the flow from the load sensing line, the power limiting relief valve cannot effectively limit the load pressure signal. This means that the power limiting control cannot function.

I checked the pump fitted to the hydraulic power unit and it did not have this orifice fitted. I advised the manufacturer and requested that they dispatch one of these orifices urgently. I was astonished by the manufacturer's reply - the required part was on back order. To minimize any further downtime, I manufactured an orifice, fitted it to the pump and handed the hydraulic power unit over to the customer.

Test for success

Thoroughly testing new or rebuilt hydraulic components prior to dispatch ensures that the component will work the way it should and will perform within its design parameters. It is possible that the manufacturer tested the hydraulic pump discussed above - but only its load sensing and pressure limiting controls. Had the functionality of the power limiting control been tested, the pump would not have been dispatched without the necessary orifice. This would have avoided an embarrassing mistake for the manufacturer and many hours of downtime for the customer.

For more information on variable pump controls and how they work, go to: www.IndustrialHydraulicControl.com

Adding hydraulic oil - without the dirt

Hydraulic fluid straight from the drum, has a typical cleanliness level of ISO 4406 21/18.

A 25 GPM pump operating continuously in hydraulic oil at this cleanliness level will circulate 3,500 pounds of dirt to the hydraulic system's components each year!

To add hydraulic oil, and not the dirt, always filter new oil prior to use in a hydraulic system.

This can be accomplished by pumping the oil into the hydraulic reservoir through the system's return filter. The easiest way to do this is to install a tee in the return line and attach a quick-connector to the branch of this tee. Attach the other half of the quick-connector to the discharge hose of a drum pump.

When hydraulic oil needs to be added to the reservoir, the drum pump is coupled to the return line and the oil is pumped into the reservoir through the return filter. As well as filtering the oil, spills are avoided and the ingress of external contamination is prevented.

The benefits of carrying out this simple modification are well worth the minor cost involved.

Learn more about contamination and hydraulic oil cleanliness.

Hydraulic fluid - getting the viscosity right

Most hydraulic systems will operate satisfactorily using a variety of fluids, including multi-grade engine oil and automatic transmission fluid (ATF), in addition to the more conventional anti-wear (AW) hydraulic fluid - provided the viscosity is correct.

Viscosity is the single most important factor when selecting a hydraulic fluid. It doesn't matter how good the anti-wear, anti-oxidization or anti-corrosion properties of the fluid are, if the viscosity grade is not correctly matched to the operating temperature range of the hydraulic system, maximum component life will not be achieved.

Defining the correct fluid viscosity grade for a particular hydraulic system involves consideration of several interdependent variables. These are:

- starting viscosity at minimum ambient temperature;
- maximum expected operating temperature, which is influenced by maximum ambient temperature; and
- permissible and optimum viscosity range for the system's components.

Once these parameters are known, the correct viscosity grade can be determined using the viscosity/temperature curve of a suitable type of fluid - commonly AW hydraulic fluid defined according to ISO viscosity grade (VG) numbers.

Automatic transmission fluid, multi-grade engine oil and anti-wear, high VI (AWH) hydraulic fluid are commonly used in hydraulic systems that experience a wide operating temperature range. These fluids have a higher Viscosity Index (VI) than AW hydraulic fluids due to the addition of VI improvers. The higher the VI a fluid has, the smaller the variation in viscosity as temperature changes.

In simple terms, this means that if you are running ATF(46) in your skid-steer loader, you can operate the hydraulics with a higher fluid temperature before viscosity falls below optimum, than you could if you were running ISO VG46 AW hydraulic fluid.

When selecting a high VI fluid, the component manufacturer's minimum permissible viscosity value should be increased by 30% to compensate for possible loss of viscosity as a result of VI improver sheardown.

VI improvers can have a negative effect on the demulsification and air separation properties of the fluid and for this reason some hydraulic component manufacturers recommend that these types of fluids only be used when operating conditions demand.

As far as fluid recommendations go, for commercial reasons relating to warranty etc, I always advise following the machine manufacturer's recommendation. But in equipment that has a history of satisfactory performance and component life, there is usually no compelling reason to change the type of fluid being used.

High hydraulic fluid temperature - how it causes premature failures

I was asked recently to conduct failure analysis on two radial piston hydraulic motors that had failed well short of their expected service life. Inspection revealed that the motors had failed through inadequate lubrication, as a result of low fluid viscosity caused by excessive hydraulic fluid temperature.

How does this happen?

As the temperature of petroleum-based hydraulic fluid increases, its viscosity decreases. If fluid temperature increases to the point where viscosity falls below the level required to maintain a lubricating film between the internal parts of the component, damage will result.

The temperature at which this occurs depends on the viscosity grade of the fluid in the system. Hydraulic fluid temperatures above 180°F (82°C) damage seals and reduce the service life of the fluid. But depending on the grade of fluid, viscosity can fall to critical levels well below this temperature.

How can this type of failure be prevented?

The above example highlights the importance of not allowing fluid temperature to exceed the point at which viscosity falls below the optimum level for the system's components.

Continuing to operate a hydraulic system when the fluid is over-temperature is similar to operating an internal-combustion engine with high coolant temperature. Damage is pretty much guaranteed.

Therefore, whenever a hydraulic system starts to overheat, shut down the system, find the cause of the problem and fix it!

For more information on hydraulic failures and how to prevent them, read Preventing Hydraulic Failures.

Biodegradable hydraulic fluid - its application and use

A client recently engaged me to advise them on an application that required the use of biodegradable hydraulic fluid. This client was tendering on an earth-moving project located in environmentally sensitive wetlands. A condition of the contract was that the hydraulic systems of all equipment employed on the project use biodegradable fluid to minimize pollution in the case of oil leaks, especially hydraulic hose failures.

Biodegradable or biobased hydraulic fluids use vegetable oils such as canola, rapeseed, sunflower or soybean as the base oil. The properties of these fluids can be equivalent to that of mineral oil based, anti-wear hydraulic fluids. But due to limited application testing, some hydraulic component manufacturers recommend reducing maximum permissible operating pressure (load) when using biodegradable hydraulic fluids, to ensure no reduction in component life.

After reviewing the available technical data on the hydraulic components fitted to the machinery being employed, a reduction in operating pressure to 80% of that permissible for mineral oil was considered prudent.

The extraordinary costs that the contractor needed to consider in their bid included not only the expense of the fluid, and draining and flushing the hydraulic system to convert from mineral oil to vegetable oil and back again, but also the costs associated with derating the machinery.

A reduction in system operating pressure means a reduction in actuator force. This means that a hydraulic excavator that has its operating pressure reduced by 20% will experience a 20% reduction in "break-out" force.

The commercial implication of this meant that the contractor needed to cost the job allowing for the use of bigger machinery than they otherwise would have.

Initiatives such as renewable energy and non-food uses for agricultural production have driven advances in biobased fluid technology. Once these fluids can compete with mineral oils on price and performance, their usage will increase and more data relating to hydraulic component life will become available.

When this point is reached, biodegradable hydraulic fluids will no longer be relegated to special applications and the extraordinary costs associated with using them, as illustrated in the above example, will no longer apply.

Hydrostatic transmissions - making sense of case drain flow Part 1

One of our readers recently wrote to me regarding the following problem:

"I tried one afternoon and evening to determine what was wrong with a hydrostatic transmission by monitoring case drain flow and was confused by the readings I was seeing. There was a flow meter in the transmission pump outlet and another in its case drain that always showed charge pump flow, even though the motor was bypassing profusely. The motor case drain went through the transmission pump case to tank."

What is a hydrostatic transmission?

A hydrostatic transmission consists of a variable-displacement pump and a fixed or variable displacement motor, operating together in a closed circuit. In a closed circuit, fluid from the motor outlet flows directly to the pump inlet, without returning to the tank.

As well as being variable, the output of the transmission pump can be reversed, so that both the direction and speed of motor rotation are controlled from within the pump. This eliminates the need for directional and flow (speed) control valves in the circuit.

Because the pump and motor leak internally, which allows fluid to escape from the loop and drain back to the tank, a fixed-displacement pump called a charge pump is used to ensure that the loop remains full of fluid during normal operation. The charge pump is normally installed on the back of the transmission pump and has an output of at least 20% of the transmission pump's output.

In practice, the charge pump not only keeps the loop full of fluid, it pressurizes the loop to between 110 and 360 PSI, depending on the transmission manufacturer. A simple charge pressure circuit comprises the charge pump, a relief valve and two check valves, through which the charge pump can replenish the transmission loop. Once the loop is charged to the pressure setting of the relief valve, the flow from the charge pump passes over the relief valve, through the case of the pump or motor or both, and back to tank.

What is the significance of case drain flow?

When a pump or motor is worn or damaged, internal leakage increases and therefore the flow available to do useful work decreases. This means that the condition of a pump or motor can be determined by measuring the flow from its case drain line (internal leakage) and expressing it as a percentage of its theoretical or design flow.

How does this apply to hydrostatic transmissions?

When applying this technique to a hydrostatic transmission, charge pump flow must be considered. In most transmissions, the charge pump relief valve vents into the case of either the pump or the motor.

This means that in the circuit described by our reader, where the motor case drain flushed through the transmission pump case to tank, you would expect to see the flow meter in the transmission pump case drain line reading design charge pump flow. Here's why:

Say charge pump flow was 10 GPM, of which 4 GPM was leaking out of the loop through the motor's internals (case drain) and 2 GPM was leaking out of the loop through the pump's internals. The balance of 4 GPM must therefore be going over the charge relief - but still ends up in either the pump or motor case, depending on the location of the relief valve. In this particular circuit, because the motor case drain flushed through the transmission pump case to tank, you would expect to see the flow meter in the transmission pump case drain line reading the sum of these three flows (10 GPM).

Before any meaningful conclusions can be drawn, the case in which the charge pump relief is venting (motor or pump) must be determined and the two case drain lines (motor and pump) must be isolated from each other. If the charge relief vents into the case of the pump, then it is possible to determine the condition of the motor by measuring its case drain flow, but not the pump. If the charge relief vents into the case of the motor, then it is possible to determine the condition of the pump by measuring its case drain flow, but not the motor.

It is not possible to determine the condition of the component that has the charge relief valve venting into it because there is no way of telling what proportion of the total case drain flow is due to internal leakage - unless of course the charge relief can be vented externally while the test is conducted. While it is possible to do this on most transmissions, it's not usually a simple exercise.

Using case drain flows to determine the condition of the components of a hydrostatic transmission, without a thorough understanding of closed circuits, can result in incorrect conclusions and the costly change-out of serviceable components.

Learn more about hydrostatic transmissions.

Hydrostatic transmissions - making sense of case drain flow Part 2

In my previous article on hydrostatic transmissions, I outlined the theory and technique for using case drain flow to determine the condition of the components of a hydrostatic transmission.

In response to this article, some readers thought that the function of the flushing valve warranted discussion, while others were still confused about the influence of the charge pump when determining case drain leakage. Let's consider flushing valves first.

What is a flushing valve?

A closed circuit flushing valve (also called a transmission valve or replenishing valve) usually comprises a pilot operated directional valve and a low pressure relief valve. When the hydrostatic transmission is in neutral, the directional valve is centered and the gallery to the low pressure relief valve is blocked. When the transmission is operated in either forward or reverse, the high pressure side of the loop pilots the directional valve. This opens the low pressure side of the loop to the relief valve gallery.

What does a flushing valve do?

In a closed circuit, fluid from the motor outlet flows directly to the pump inlet. This means that apart from losses through internal leakage, which are made up by the charge pump, the same fluid circulates continuously between pump and motor. If the transmission is heavily loaded, the fluid circulating in the loop can overheat.

The function of the flushing valve is to positively exchange the fluid in the loop with that in the reservoir. A flushing valve is most effective when it is located at the motor, assuming the charge check valves are located in the transmission pump, as is the norm.

When the hydrostatic transmission is in neutral, the flushing valve has no function and charge pressure is maintained by the charge relief valve in the transmission pump. When the transmission is operated in either forward or reverse, the flushing valve operates so that charge pressure in the low pressure side of the loop is maintained by the relief valve incorporated in the flushing valve. This relief valve is set around 30 psi lower than the charge pump relief valve located in the transmission pump.

The effect of this is that cool fluid drawn from the reservoir by the charge pump, charges the low pressure side of the loop through the check valve located close to the transmission pump inlet. The volume of hot fluid leaving the motor outlet, that is not required to maintain charge pressure in the low pressure side of the loop, vents across the flushing valve relief into the case of the motor and back to tank, usually via the pump case.

How does a flushing valve influence the process of using case drain leakage to determine the condition of a transmission?

The technique is the same as that outlined in the last issue. As explained above, if a flushing valve is fitted to a transmission, it acts as the charge pump relief valve once the transmission is operated in forward or reverse. So if the flushing valve vents into the case of the motor, then it is possible to determine the condition of the pump by measuring its case drain flow, but not the motor. If the flushing valve vents into the case of pump, then it is possible to determine the condition of the motor by measuring its case drain flow, but not the pump.

This reinforces the point that using case drain flows to determine the condition of the components of a hydrostatic transmission, without a thorough understanding of the circuit in question, can result in incorrect conclusions and the costly change-out of serviceable components.

Hydrostatic transmissions - making sense of case drain flow Part 3

In my previous articles on hydrostatic transmissions, I described the technique for determining the condition of a hydrostatic transmission using case drain flow, and discussed the role and influence of a flushing valve when doing this.

In response to these articles, some readers were still confused about the influence of the charge pump when determining case drain leakage.

One reader held the view that, assuming the charge pump relief vents into the case of the motor and the motor case drain line is isolated from the pump, then transmission pump leakage is determined by subtracting charge pump flow from the total flow from the pump case. For example, if total charge pump flow was 10 GPM and the flow-meter in the pump case drain line was reading 15 GPM then transmission pump leakage would be 5 GPM (15 - 10 = 5).

This is incorrect because it suggests that a hydrostatic transmission can leak more than the total available flow from its charge pump. It cannot. That is, it is impossible for the flow meter in the pump case drain line to read 15 GPM when the total available flow from the charge pump is only 10 GPM, as in the above example.

The reason is simple. Because the function of the charge pump is to make up losses from the loop through internal leakage, if total losses exceed available charge pump flow, the transmission will cavitate. If in the above example, the transmission was leaking 5 GPM more than the total available flow from the charge pump, there would be a serious deficit of fluid in the loop. In practice, the transmission would destroy itself through cavitation before it got to this point.

Let me explain this another way. Let's assume we have a transmission that has a volumetric efficiency of 100%, that is, the pump and motor have no internal leakage. The loop has a total volume of two gallons and is full of fluid. Because there is no internal leakage there is no need for a charge pump.

The pump is stroked to maximum displacement, which circulates the two gallons of fluid in the loop at a rate of 50 GPM. Because it's a closed loop, with no leakage, the flow from pump to motor is 50 GPM and the flow from motor to pump is 50 GPM.

Now let's introduce internal leakage of 0.5 GPM in both pump and motor. The result is that, with no charge pump to replenish the loop, after one minute there will only be one gallon of fluid left in the loop (the other gallon will have leaked back to tank). Within a second of the transmission starting to leak, the transmission pump will start to cavitate and the severity of this cavitation will increase with each passing second until the transmission destroys itself.

Now let's install a charge pump with a flow rate of 1 GPM in the circuit. Problem solved, temporarily at least. With 1 GPM leaking out of the loop and 1 GPM being replenished by the charge pump the status quo is maintained... until wear causes the internal leakage of the transmission to exceed 1 GPM.

As you can see, it's not possible for the internal leakage of a hydrostatic transmission to exceed the flow rate of its charge pump. Charge pump flow rate is typically 20% of transmission pump flow rate. This means that volumetric efficiency can drop to 80% before the transmission will cavitate and destroy itself. The trick is to overhaul the transmission before this point is reached.

Learn more about hydrostatic transmissions.

Temperature shock of hydraulic components and how to avoid it

A client recently asked me to investigate and solve a recurring problem on a diving bell launch and recovery system. The system comprised of a hydraulic power unit, a bell winch, an umbilical winch and a guide-wire winch.

To launch the bell, the guide-wire winch is used to lower a clump weight to the seabed (the guide wires prevent the bell from spinning during launch and recovery) and then the bell and its umbilical are launched using their respective winches.

After the divers have finished their shift on the seabed (usually 6-8 hours) the bell and its umbilical are recovered, followed by the clump weight.

The problem was occurring during recovery of the clump weight with the guide-wire winch. The hydraulic motor on this winch was of radial piston design. When the winch was summoned to haul up the clump weight, the distributor shear pin (designed to prevent torque from being applied to the distributor valve), was frequently shearing, rendering the winch unserviceable. Once this pin has sheared, the distributor must be removed from the motor and the pin replaced. Apart from the obvious inconvenience, this was resulting in costly downtime. The cause of this problem was temperature shock.

What is temperature shock?

When there is a significant difference between the temperature of a hydraulic component and the fluid being supplied to it, rapid, localized heating of the internal parts of the component can occur. This causes individual parts of the component to expand at different rates, resulting in interference between parts that normally have fine clearances.

How does this happen?

Temperature shock occurs when part of a hydraulic circuit is operated for long enough for the hydraulic fluid in the system to reach operating temperature, and then a previously idle part of the circuit is functioned. This results in hot fluid being delivered to cold components.

In the example described above, due to the length of time between the launch of the bell and its recovery, the hydraulic system was at ambient temperature at the start of the recovery operation.

By the time the bell and its umbilical had been hauled up 450 feet and were safely on deck, the hydraulic fluid was at operating temperature. But the guide-wire winch, which had been idle during this time, was still at ambient temperature.

When the guide-wire winch was operated to recover the clump weight, the hot fluid entering the cold motor was causing the distributor valve to expand and bind in its housing, resulting in failure of the shear pin and rendering the motor unserviceable.

How can this be prevented?

The solution to this problem and the fix in the above example is quite simple. To prevent temperature shock of hydraulic motors, the motor's case must be continuously 'flushed' (positive circulation of a relatively small volume of fluid through the case). This ensures that the motor is always at the same temperature as the fluid in the system.

Configuring mobile hydraulic valves using power beyond

I regularly receive questions from owners of mobile hydraulic equipment in relation to the correct installation of an additional directional control valve using the power beyond facility on an existing directional control valve.

What is power beyond?

Power beyond - also called high-pressure carry over (HPCO), is a facility on a mobile hydraulic directional control valve that enables the pressure gallery to be isolated from the tank gallery and be carried over to an additional valve - usually another directional control valve. The valve being fitted must be sized to handle the rated flow from the pump.

The arrangement of the power beyond facility varies with valve type and manufacturer. However the most common arrangement is a facility to install a threaded plug or sleeve that blocks the drilling between the pressure and tank galleries inside the valve. The power beyond port is then used to supply pump flow to the additional directional control valve. If the existing directional control valve has an alternative tank port, this allows the tank line from the additional valve to be connected to tank via the existing valve (Figures 1 & 2).

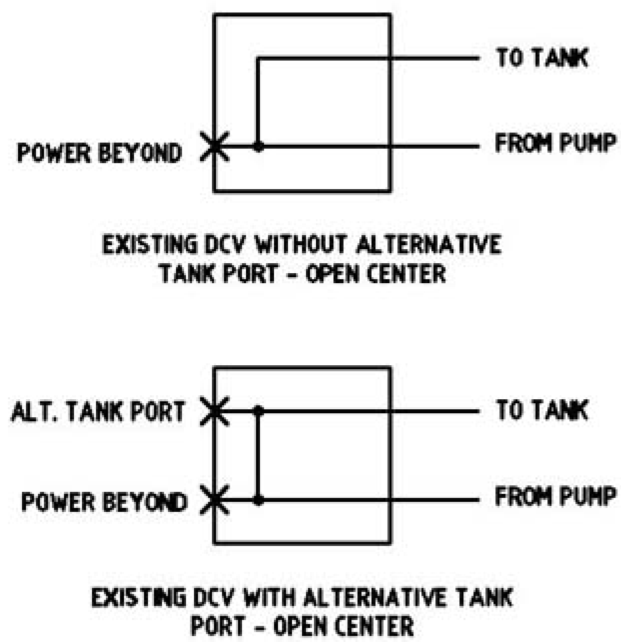


Figure 1. Simplified line drawing showing pressure and tank galleries of a directional control valve (DCV) in open center arrangement.

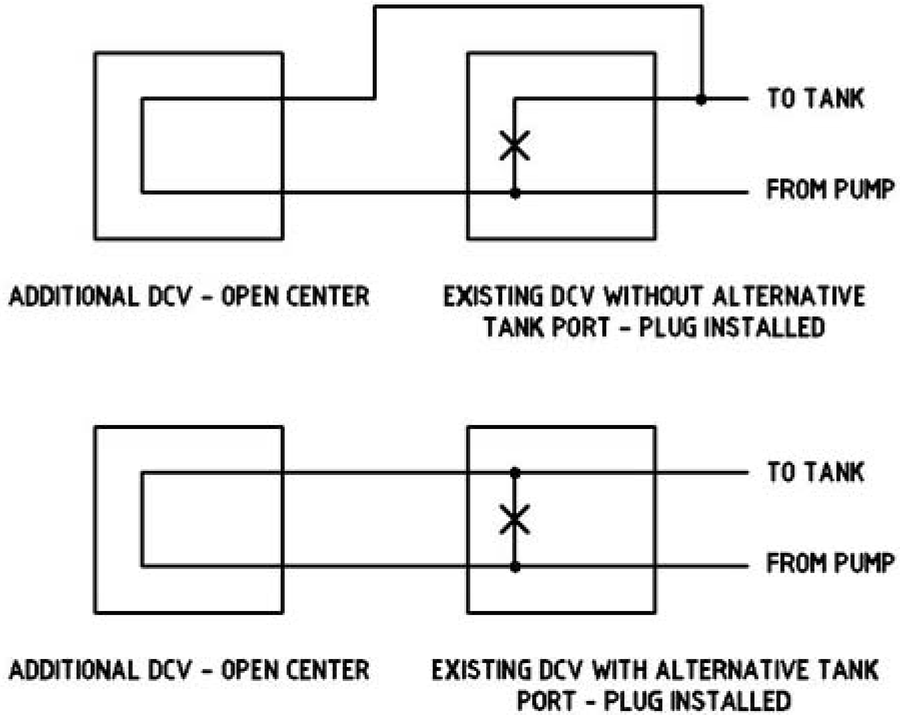


Figure 2. Connection of an additional open center DCV using the power beyond facility.

Most mobile directional control valves can be made closed center by plugging the drilling between the pressure and tank galleries and leaving the power beyond port plugged (Figure 3). This means that if the existing valve is closed center, supplying pump flow to the additional valve only requires the connection of its pressure line to the existing valve's power beyond port (Figure 4).

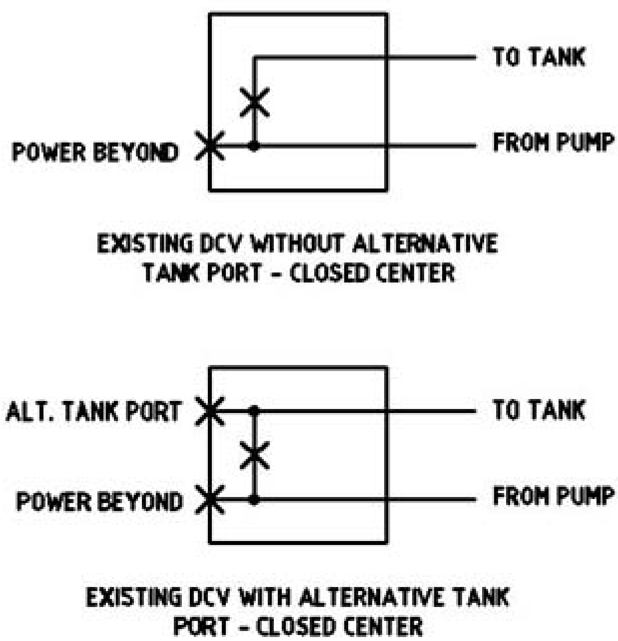


Figure 3. Simplified line drawing showing pressure and tank galleries of a DCV in closed center arrangement.

It is important to note that if the existing valve is closed center, the power beyond plug or sleeve must be installed in the additional valve to make it closed center also (Figure 4).

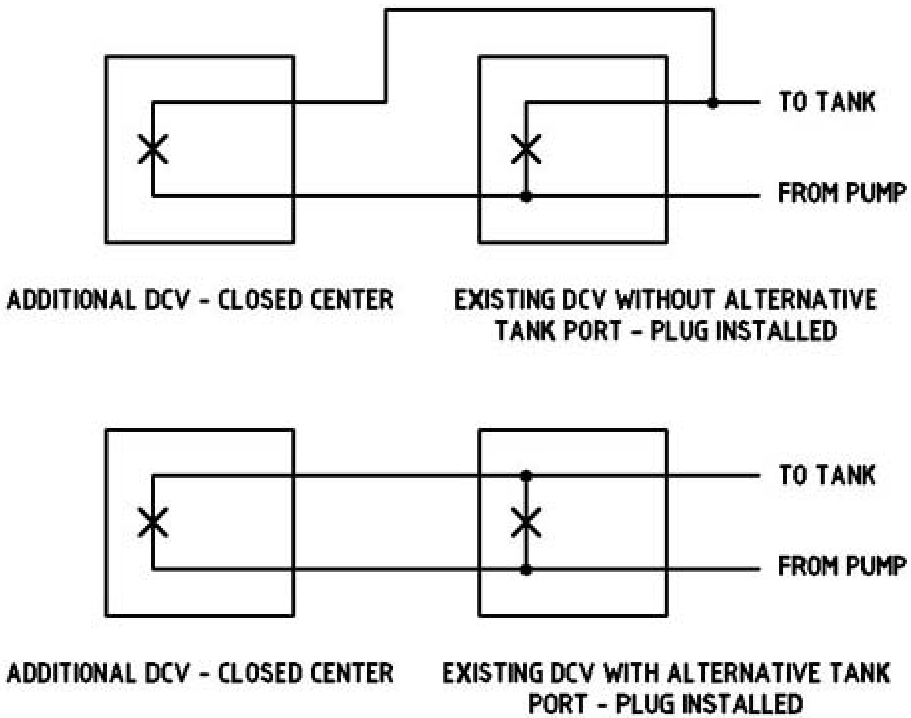


Figure 4. Connection of an additional closed center DCV using the power beyond facility.

The true cost of hydraulic oil leaks

Hydraulic systems are often considered perennial consumers of oil and in turn, make-up fluid an inherent cost of operating hydraulic equipment. But what is the real cost of one or more "minor leaks" on your hydraulic equipment? To answer this question, the costs associated with all of the following factors need to be considered:

- Make-up fluid.
- Clean-up.
- Disposal.
- Contaminant ingress.
- Safety.

Make-up fluid

The cost of make-up fluid should be the most obvious cost of hydraulic system leaks. I say 'should be' because many hydraulic equipment users fail to consider the accumulative effect on the cost of one or more slow leaks over time.

Consider a leak from a hydraulic fitting that produces six drops of oil per minute. Hardly worth your attention, right? If the volume of each drop was half a milliliter, over 24 hours the loss is nearly half a liter - perhaps not a significant amount. But over a month this equates to 15 liters and 180 liters over the course of a year. Assuming a fluid cost of $2 per liter, this "minor leak" is costing $360 per annum in make-up fluid alone.

Clean-up

Where there are oil leaks there is almost always a clean-up cost to consider. Clean-up costs include:

- labor;
- equipment required to empty sumps and drip trays, and degrease machine surfaces; and
- consumables such as detergents and absorbent material.

Assuming it costs $10 per week in labor, equipment and consumables to clean up the minor leak discussed above, the annual clean-up bill totals more than $500.

Disposal

I can remember a time, not so long ago, when waste oil companies used to pay for the privilege of emptying waste hydraulic oil tanks. These days they bill you for the privilege. Environmentally acceptable disposal of waste oil and absorbent material containing waste oil costs money.

Assuming a disposal cost of $0.60 per liter, the annual disposal costs attributable to the minor leak discussed above amounts to $110.

Contaminant ingress

Where oil leaks out, contaminants such as air, particles and water can get in. Costs to consider here include:

- hydraulic component damage and fluid degradation as a result of contaminant ingress;
- hydraulic system reliability problems; and
- removal of ingested contaminants.

Safety

In many situations, oil leaks can pose a safety hazard. Like the costs associated with contaminant ingress, the costs associated with the safety risk posed by oil leaks are difficult to quantify - short of a lost time accident actually occurring. In addition, the cost of minimizing the safety risk can be obscured. An example would be more frequent clean-up than may otherwise be required. This hides what is essentially a safety cost in clean-up expenses.

Conclusion

The annual cost of one slow leak, similar to that discussed above, amounts to nearly $1,000 per year in make-up fluid, clean-up and disposal costs alone. If you have multiple pieces of hydraulic equipment with several leaks on each one, the accumulative cost over an extended period of time should alarm you. Inspect your hydraulic equipment today and tag all leaks for corrective action during the next available maintenance outage. It could save you a lot of money!

Hydraulic system oil types and manufacturers' recommendations

I recently came across the following story posted on a message board by a heavy equipment mechanic:

" I have a customer who recently bought a new service and operator's manual for his old Komatsu PC150 hydraulic excavator… He went to the lubrication guide for the hydraulic system and chose what the chart said was a 10W-40 engine oil and got what Texaco showed as being Komatsu's recommendation. Here is where the problems begin... First you never want to use a detergent oil in a hydraulic system - it can cause anything from foaming in the oil which will cause cavitation in the hydraulic pump and result in premature pump failure, to it picking up waste/contaminants in the oil and depositing it as sludge throughout the hydraulic system. But according to Texaco EVERY multi-grade oil available in the US is gonna be a detergent oil... Second, and I actually called Komatsu America on this, they DO NOT recommend a detergent oil in the hydraulic system...What is supposed to be in the hydraulic system is either a 10W or a 30W non detergent oil... This tells me that there are a lot of people out there that are blissfully going by the book and slowly killing their machine's hydraulic system. Hopefully at least some of them have asked before they wasted a lot of money and filled their hydraulic system with the wrong oil. My customer is a long-time hydraulic equipment owner, so didn't ask me before he changed his hydraulic oil... the book has to be right, right?... Now he's stuck with a 50 gallon system that he can only change 30 gallons of at a time so it's going to cost him big to fill and flush enough times to get all of the detergent oil out of the hydraulic system..."

When the OEM doesn't say what they mean, it can certainly make it difficult to follow their lube recommendations. But is a detergent oil, such as a multi-grade diesel engine oil, really detrimental to a modern hydraulic system?

Detergent additives

DIN 51524; HLPD fluids are a class of hydraulic fluids which contain detersive and dispersive additives. The use of these fluids is approved by most major hydraulic component manufacturers and can be advantageous in many applications, including mobile, to prevent build-up of sludge and varnish deposits, which can lead to valve stiction and other reliability problems. The main caution with these fluids is that they have excellent water emulsifying ability, which means that if present, water is not separated out of the fluid. Emulsified water reduces lubricity and filterability, can cause corrosion and cavitation and reduces the life of the oil. These problems can be avoided by maintaining water content below 0.1% - which is not a low water content target for any high-performance hydraulic system. A hydraulic fluid that has the ability to emulsify small amounts of water can be beneficial in mobile applications.

Modern lube chemistry means that detersive and air-separation/anti-foaming properties aren't necessarily mutually exclusive. At least one major, mobile equipment manufacturer specifies heavy-duty diesel engine oil to API classifications CD, CF-4, CG-4 or CH-4 as an acceptable alternative to a single-grade, anti-wear, hydraulic oil in their equipment.

Viscosity index improvers

This is not to say that mutli-grade engine oils are suitable for every hydraulic system. Multi-grade engine oil, automatic transmission fluid and anti-wear, high VI (AWH) hydraulic fluid are commonly used in hydraulic systems that experience a wide operating temperature range. These fluids have a higher Viscosity Index (VI) than standard anti-wear hydraulic fluids due to the addition of VI improvers. The higher the VI a fluid has the smaller the variation in viscosity as temperature changes.

In simple terms, this means that if you are running a 10W-40 engine oil in your excavator hydraulics, you can operate the hydraulics with a higher fluid temperature before viscosity falls below optimum, than you could if you were running a single-grade, anti-wear, hydraulic fluid.

When selecting a high VI fluid, it is wise to increase the hydraulic component manufacturer's minimum permissible viscosity value by 30% to compensate for possible loss of viscosity as a result of VI improver sheardown. VI improvers can also have a negative effect on the air-separation properties of the fluid and for this reason some hydraulic component manufacturers recommend that high VI fluids only be used when operating conditions demand.

Conclusion

As far as hydraulic oil recommendations go, for commercial reasons relating to warranty etc, it is always advisable to follow the machine manufacturer's recommendation. But on the other hand, if you were to realize retrospectively that the hydraulic system of your excavator or dozer had been inadvertently charged with a multi-grade, diesel-engine oil, it's not necessary to press the panic button either. Discussing the application with a technical specialist from the oil manufacturer, in conjunction with the OEM, would however be advisable.

Which hydraulic oil?

The May 2004 issue of *Inside Hydraulics* carried an article that examined the use of multigrade, detergent engine oil as a hydraulic fluid in mobile equipment.

Following the publication of this article, I received a lot of feedback. Some folk thought my recommendations were not definitive enough. But when it comes to hydraulic fluid, it is not possible to make one, definitive recommendation that covers all types of hydraulic equipment in all applications.

In an effort to clarify this matter, I decided to pose (and answer) the following question:

What type of oil would I use in my excavator hydraulics and why?

Assuming that my hydraulic excavator is a late model machine fitted with variable displacement piston pump(s) which operate at pressures up to 5,000 PSI or higher, my selection would be based on the following factors.

Multigrade versus monograde

The operating temperature range of the excavator is the factor that will determine whether I use a multigrade or monograde oil. If my excavator is required to operate in freezing temperatures in winter and tropical conditions in summer, then multigrade oil will be required to maintain oil viscosity within optimum limits across this wide operating temperature range.

If my excavator has a narrow operating temperature range and it is possible to maintain optimum fluid viscosity using monograde oil, I wouldn't chose a multigrade. This is because the viscosity index (VI) improvers used to make multigrade oils can have a negative effect on the air separation properties of the oil. This is not ideal in a mobile hydraulic machine like an excavator, which typically has a relatively small reservoir with correspondingly poor de-aeration characteristics. In addition, by not using a multigrade oil when it's not required, I don't need to concern myself with loss of viscosity as a result of VI improver sheardown.

Detergent versus non-detergent

Detergent oils have the ability to emulsify water, and disperse and suspend other contaminants such as varnish and sludge. This keeps components free from deposits but means that contaminants are not precipitated out - they must be filtered out. These are desirable properties in a mobile hydraulic machine like an excavator which, unlike an industrial hydraulic system, has little opportunity for the settling and precipitation of contaminants at the reservoir, due to its relatively small volume.

I'll be using an oil with detersive/dispersive additives in my excavator and closely monitoring water content on the fluid condition reports. Water accelerates aging of the oil, reduces lubricity and filterability, reduces seal life and leads to corrosion and cavitation. Emulsified water can be turned into steam at highly loaded parts of the system. I'll be avoiding these problems by maintaining water content below 100 ppm.

Anti-wear versus no anti-wear

The purpose of anti-wear additives is to maintain lubrication under boundary conditions. The most common anti-wear additive used in engine and hydraulic oil is Zinc dialkyl dithiophosphate (ZnDTP).

The presence of ZnDTP is not always seen as a positive, due to the fact that it can chemically break down and attack some metals. However, it is an essential additive in oil used in any high-pressure, high-performance hydraulic system, such as a modern excavator fitted with piston pumps and motors. I'll be selecting an oil containing ZnDTP at a concentration of at least 900 ppm for use in my excavator hydraulics.

Fill 'er up

In summary, if optimum fluid viscosity can be maintained using monograde oil, then I would select and use one that contains detersive/dispersive additives and at least 900 ppm of ZnDTP. If multigrade oil were required, I would use either multigrade hydraulic fluid (with detergent additives) or heavy-duty diesel engine oil (API classification CD, CF-4, CG-4 or CH-4) with a minimum ZnDTP content of 900 ppm.

Reducing noise emission from hydraulic systems

Many industrialized countries have regulations that restrict noise levels in the workplace. The high power density, and corresponding high noise emission of hydraulic components means that industrial hydraulic systems are often the target of efforts to reduce mean noise levels in the workplace.

The dominant source of noise in hydraulic systems is the pump. The hydraulic pump transmits structure-borne and fluid-borne noise into the system and radiates air-borne noise.

All positive-displacement hydraulic pumps have a specific number of pumping chambers, which operate in a continuous cycle of opening to be filled (inlet), closing to prevent back flow, opening to expel contents (outlet) and closing to prevent back flow.

These separate but superimposed flows result in a pulsating delivery, which causes a corresponding sequence of pressure pulsations. These pulsations create fluid-borne noise, which causes all downstream components to vibrate. The pump also creates structure-borne noise by exciting vibration in any component with which it is mechanically linked, e.g. tank lid. The transfer of fluid and structure induced vibration to the adjacent air mass results in air-borne noise.

Reducing fluid-borne noise

While fluid-borne noise attributable to pressure pulsation can be minimized through hydraulic pump design, it cannot be completely eliminated. In large hydraulic systems or noise-sensitive applications, the propagation of fluid-borne noise can be reduced by the installation of a silencer. The simplest type of silencer used in hydraulic applications is the reflection silencer, which eliminates sound waves by superimposing a second sound wave of the same amplitude and frequency at a 180-degree phase angle to the first.

Reducing structure-borne noise

The propagation of structure-borne noise created by the vibrating mass of the power unit (the hydraulic pump and its prime mover) can be minimized through the elimination of sound bridges between the power unit and tank, and the power unit and valves. This is normally achieved through the use of flexible connections i.e. rubber mounting blocks and flexible hoses, but in some situations it is necessary to introduce additional mass, the inertia of which reduces the transmission of vibration at bridging points.

Reducing air-borne noise

The magnitude of noise radiation from an object is proportional to its area and inversely proportional to its mass. Reducing an object's surface area or increasing its mass can therefore reduce its noise radiation. For example, constructing the hydraulic reservoir from thicker plate (increases mass) will reduce its noise radiation.

The magnitude of air-borne noise radiated directly from the hydraulic pump can be reduced by mounting the pump inside the tank. For full effectiveness, there must be a clearance of 0.5 meters between the pump and the sides of tank, and the mounting arrangement must incorporate decoupling between the power unit and tank to insulate against structure-borne noise. The obvious disadvantage of mounting the hydraulic pump inside the tank is that it restricts access for maintenance and adjustment.

If hydraulic system noise remains outside the required level after all of the above noise propagation countermeasures have been exhausted, encapsulation or screening must be considered.

Hydraulic repair using aftermarket hydraulic parts

Repairing a hydraulic component involves reworking or replacing all of the parts necessary to return the component to 'as new' condition, in terms of performance and expected service life. In many cases, repairing a hydraulic pump, motor or cylinder can result in significant savings when compared with the cost of purchasing a new one.

The economics of proceeding with any hydraulic repair is ultimately dependent on the cost of the repair, relative to the cost of a new component. As a rule, the more expensive the new component is in absolute dollar terms, the more likely it is that a repair will be cost effective.

The cost of a hydraulic repair is determined by a number of factors including:

- extent of wear or damage to the component;
- facilities and expertise of the hydraulic repair shop; and
- the repair techniques employed.

When a hydraulic component is repaired, there are usually some parts that can be successfully re-used after they have been reworked using processes such as machining, honing, lapping, grinding and hard-chrome plating. The skilled application of these techniques can reduce the number of new parts required, reducing hydraulic repair costs.

Aftermarket hydraulic parts

It is sometimes possible to reduce hydraulic repair costs further, by using non-genuine or aftermarket hydraulic parts. Some aftermarket spare parts for hydraulic components originate from the same factories that supply the genuine article and are therefore of the same quality. In many cases however, aftermarket hydraulic parts originate from niche manufacturers and their quality can vary from poor to excellent.

With this in mind, it is important to distinguish between a repair using aftermarket parts of known quality that will save you money, and a repair using parts of unknown quality that is likely cost you twice as much in the long-run. You can do this by asking the repair shop two questions:

- Has the quality of the parts been proven in terms of performance and achieved service life?
- Is the repair covered by warranty?

If the aftermarket parts are of known quality and the hydraulic repair shop is prepared to back them, the decision carries minimal risk. If the repair shop hasn't used the aftermarket parts before and their quality is unknown, you need to base your decision on how much money you will save if the parts live up to expectations; what it will cost you if they don't; and whether the repair shop is willing to carry some of the risk involved in finding out.

If the quality of the parts is unproven, or the repair shop is not willing to back the repair with a warranty, it's wise to think twice before proceeding. You need to make an informed decision based on the amount of money the parts will save you if they live up to expectations, versus what it will cost you if they don't. The hydraulic repair shop may be willing to share some of the risk involved in finding out, because if the parts prove to be successful, they can offer the same solution to other customers.

Dealing with water in hydraulic fluid

If you have worked with hydraulic equipment for any length of time, it's likely that you've come across a hydraulic system with cloudy oil. Oil becomes cloudy when it is contaminated with water above its saturation level. The saturation level is the amount of water that can dissolve in the oil's molecular chemistry and is typically 200 - 300 ppm at 68°F (20°C) for mineral hydraulic oil. Note that if hydraulic oil is cloudy it indicates that a **minimum** of 200 - 300 ppm of water is present. I recently audited a hydraulic system with cloudy oil that was found to contain greater than 1% (10,000 ppm) water.

Why is water in hydraulic fluid bad?

Water in hydraulic fluid:

- Depletes some additives and reacts with others to form corrosive by-products which attack some metals.
- Reduces lubricant film-strength, which leaves critical surfaces vulnerable to wear and corrosion.
- Reduces filterability and clogs filters.
- Increases air entrainment ability.
- Increases the likelihood of cavitation occurring.

How much water is too much?

A number of factors need to be considered when selecting water contamination targets, including the type of hydraulic system and reliability objectives for the equipment. It's always wise to control water contamination at the lowest levels that can reasonably be achieved, ideally below the oil's saturation point at operating temperature.

Water removal methods

Methods for removing free (unstable suspension) and emulsified (stable suspension) water include:

- polymeric filters;
- vacuum distillation; and
- headspace dehumidification.

Vacuum distillation and headspace dehumidification also remove dissolved water.

Polymeric filters - These look like conventional particulate filters, however the media is impregnated with a super-absorbent polymer. Water causes the polymer to swell, which traps the water within the media. Polymeric filters are best suited for removing small volumes of water and/or maintaining water contamination within pre-determined limits.

Vacuum distillation - This technique employs a combination of heat and vacuum. At 25 inches Hg, water boils at 133°F (56°C). This enables water to be removed at a temperature that does not damage the oil or its additives.

Headspace dehumidification - This method involves circulating and dehumidifying air from the reservoir headspace. Water in the oil migrates to the dry air in the headspace and is eventually removed by the dehumidifier.

In the case of small systems with high levels of water contamination, changing the oil may be more cost-effective than using any of the above methods of water removal.

Prevention is better than cure

Like all other forms of contamination, preventing water ingress is cheaper than removing it from the oil. A major point of water ingression is through the reservoir headspace. Many hydraulic system reservoirs are fitted with breather caps that allow moisture (and particles) to enter the reservoir as the fluid volume changes through either thermal expansion and contraction, or the actuation of cylinders.

Replacing the standard breather cap with a hygroscopic breather will eliminate the ingression of moisture and particles through the reservoir's vent. These breathers combine a woven-polyester media that filters particles as small as 3 microns, with silica gel desiccant to remove water vapor from incoming air. The result is relative humidity levels within the reservoir headspace that make condensation unlikely, therefore reducing water contamination of the oil.

Hydraulic troubleshooting basics

One of our readers wrote to me recently regarding the following problem:
"We have a hydraulic system that operates two cylinders. The maintenance staff recently reported that the pump (piston-type) had failed - for reasons unknown at this time. The tank, valves and cylinders were cleaned and a replacement pump installed. The new pump is delivering a maximum pressure of 1,000 PSI and appears to be creating heat. Can you suggest some tips to find a solution to this problem?"

In any troubleshooting situation, no matter how simple or complex the hydraulic system, always start with the basics. This ensures that the obvious is never overlooked. In order for the 'obvious' to be obvious, the fundamental laws of hydraulics must be kept in mind:

- Hydraulic pumps create flow - not pressure.
- Resistance to flow creates pressure.
- Flow determines actuator speed.
- Pressure determines actuator force.
- Fluid under pressure takes the path of least resistance.
- When fluid moves from an area of high pressure to an area of low pressure (pressure drop) without performing useful work, heat is generated.

Theory is great, but it always makes more sense when put into practice. So let's apply these fundamentals to the above situation in a way that ensures the obvious things are not overlooked.

"The new pump is delivering a maximum pressure of 1,000 PSI..."

We know that a hydraulic pump can only produce flow (pressure is created by resistance to flow). It follows that if the pump can't get oil it can't produce flow. So check that the reservoir is filled to the correct level, the breather is not clogged, the suction strainer or filter (if fitted) is not clogged, the pump intake isolation valve is fully open and the pump intake line is otherwise unrestricted.

If the pump is producing flow, then an absence of pressure indicates an absence of resistance to flow. Knowing this, and that fluid under pressure always takes the path of least resistance, the task now is to find the point at which pump flow is escaping from the circuit. If you're skilled in reading and interpreting hydraulic symbols, the system's schematic diagram (if available) can be useful in identifying possible locations.

"The new pump... appears to be creating heat."

Because heat is generated when there is a pressure drop, using an infrared thermometer to check the temperature of individual components will quickly lead us to the hottest part of the system - and the probable location of the internal leakage. Note that in a properly functioning system fitted with a piston pump, it is not unusual for the pump case to be the hottest part of the circuit.

The above checks should have taken less than 10 minutes. If nothing conclusive was revealed, I would continue the process of elimination using a flow-tester to conduct a direct pump test.

Conclusion

The type and variation of problems a hydraulic system can encounter are infinite. But as you can see from this example, a solid understanding of the fundamental laws of hydraulics can be applied in any situation, and is the foundation of effective troubleshooting.

Dealing with air contamination of hydraulic fluid

Contaminants of hydraulic fluid are broadly defined as any substance that impairs the proper functioning of the fluid. Air fits this definition and therefore when air becomes entrained in the hydraulic fluid, corrective action is required to prevent damage to both the fluid and system components.

Air can be present in four forms:

- Free air - such as a pocket of air trapped in part of a system.
- Dissolved air - hydraulic fluid contains between 6 and 12 percent by volume of dissolved air.
- Entrained air - air bubbles typically less than 1 mm in diameter dispersed in the fluid.
- Foam - air bubbles typically greater than 1 mm in diameter that congregate on the surface of the fluid.

Of these four forms, entrained air is the most problematic. Pre-filling components and proper bleeding of the hydraulic system during start-up will usually eliminate free air. Small amounts of foam are cosmetic and generally do not pose a problem. However, if large volumes of foam are present, sufficient to cause the reservoir to overflow for example, this can be a symptom of a more serious air contamination and/or fluid degradation problem.

Why is entrained air bad?

Negative effects of entrained air include:

- Reduced bulk modulus, resulting in spongy operation and poor control system response.
- Increased heat-load.
- Reduced thermal conductivity.
- Fluid deterioration through increased oxidation and thermal degradation (dieseling).
- Reduced fluid viscosity, which leaves critical surfaces vulnerable to wear.
- Cavitation erosion.
- Increased noise levels.
- Decreased efficiency.

Gaseous Cavitation

As pointed out above, hydraulic fluid can contain up to 12 percent dissolved air by volume. Certain conditions can cause this dissolved air to come out of solution, resulting in entrained air.

When fluid temperature increases or static pressure decreases, air solubility is reduced and bubbles can form within the fluid. This release of dissolved air is known as gaseous cavitation.

Decrease in static pressure and subsequent release of dissolved air can occur at the pump inlet, as a result of:

- Clogged inlet filters or suction strainers.
- Turbulence caused by intake-line isolation valves.
- Poorly designed inlet (diameter too small, length excessive, multiple bends).
- Collapsed or otherwise restricted intake line.
- Excessive lift (vertical distance between pump intake and minimum fluid level).
- Clogged or undersized reservoir breather.

Other causes of decreased static pressure include changes in fluid velocity through conductors and orifices, flow transients and faulty or incorrectly adjusted anti-cavitation or load control valves.

External Ingestion

Air entrainment can also occur through external ingestion. Like gaseous cavitation, this commonly occurs at the pump as a result of:

- Loose intake-line clamps or fittings.
- Porous intake lines.
- Low reservoir fluid level.
- Faulty pump shaft seal.

Other causes of air ingestion include faulty or incorrectly adjusted load control valves, which can result in air being drawn past the gland of double-acting cylinders, and return fluid plunging into the reservoir (drop-pipes extending below minimum fluid level should be fitted to all return penetrations).

Prevention is better than cure

Like other hydraulic problems, proper equipment maintenance will prevent the occurrence of most air contamination problems. As in all troubleshooting situations, when air contamination does occur, an understanding of the problem and a logical process of elimination are required to identify the root cause.

Understanding hydraulic load sensing control

Load sensing is a term used to describe a type of pump control employed in open circuits. It is so called because the load-induced pressure downstream of an orifice is sensed and pump flow adjusted to maintain a constant pressure drop (and therefore flow) across the orifice. The 'orifice' is usually a directional control valve with proportional flow characteristics, but a needle valve or even a fixed orifice can be used, depending on the application.

A load sensing circuit typically comprises a variable displacement pump, usually axial-piston design, fitted with a load sensing controller, and a directional control valve with an integral load-signal gallery (Exhibit 1). The load-signal gallery (LS, shown in red) is connected to the load-signal port (X) on the pump controller. The load-signal gallery in the directional control valve connects the A and B ports of each of the control valve sections through a series of shuttle valves. This ensures that the actuator with the highest load pressure is sensed and fed back to the pump.

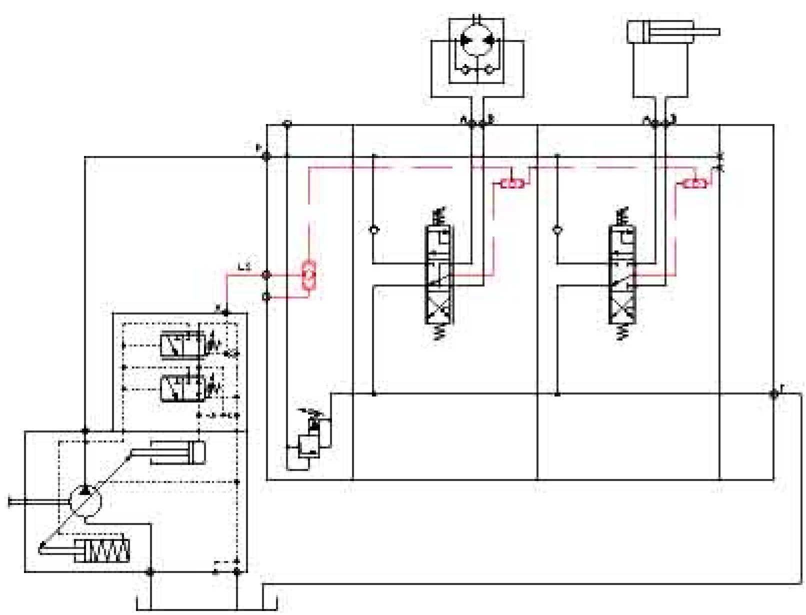


Exhibit 1. Typical load sensing circuit. Enlarge

To understand how the load-sensing pump and directional control valve function together in operation, consider a winch being driven through a manually actuated valve. The operator summons the winch by moving the spool in the directional valve 20% of its stroke. The winch drum turns at five rpm. For clarity, imagine that the directional valve is now a fixed orifice. Flow across an orifice decreases as the pressure drop across it decreases. As load on the winch increases, the load-induced pressure downstream of the orifice (directional valve) increases. This decreases the pressure drop across the orifice, which means flow across the orifice decreases and the winch slows down.

In a load sensing circuit the load-induced pressure downstream of the orifice (directional valve) is fed back to the pump via the load-signal gallery in the directional control valve. The load-sensing controller responds to the increase in load pressure by increasing pump displacement (flow) slightly so that pressure upstream of the orifice increases by a corresponding amount. This keeps the pressure drop across the orifice (directional valve) constant, which keeps flow constant and in this case, winch speed constant. The value of the pressure drop or delta P maintained across the orifice (directional valve) is typically 10 to 30 Bar (145 to 435 PSI). When all spools are in the center position the load-signal port is vented to tank and the pump maintains 'standby' pressure equal to or slightly higher than the load sensing controller's delta P setting.

Because the pump always receives the load signal from the function operating at the highest pressure, high-end load sensing directional control valves feature a pressure compensator (not shown) at the pressure inlet to each section. The section pressure compensator works with the spool-selected orifice opening to maintain a constant flow, independent of the pressure variations caused by the operation of multiple functions at the same time. This is sometimes referred to as 'sensitive load sensing'.

A load sensing pump only produces the flow demanded by the actuators - this makes it energy efficient (fewer losses to heat) and as demonstrated in the above example, provides more precise control. Load-sensing control also provides constant flow independent of pump shaft speed variations. If pump drive speed decreases, the load-sensing controller will increase displacement (flow) to maintain the set delta P across the directional control valve (orifice), until maximum displacement is reached.

Load sensing pump controls usually incorporate a pressure limiting control, also referred to as a pressure cut-off or pressure compensator. The pressure compensator limits maximum operating pressure by reducing pump displacement to zero when the set pressure is reached.

For more information on load-sensing and other types of variable pump controls, go to: www.IndustrialHydraulicControl.com

Flushing hydraulic systems

One of our readers wrote to me recently with the following question:

"Aside from replacing oil in the reservoir, what is the best way to purge contaminated oil from hydraulic system plumbing and components?"

Techniques for flushing hydraulic systems vary in cost and complexity. Before I discuss some of these methods, let's first distinguish between flushing the fluid and flushing the system.

The objective of *flushing the fluid* is to eliminate contaminants such as particles and water from the fluid. This is usually accomplished using a filter cart or by diverting system flow through an external fluid-conditioning rig.

The objective of *flushing the system* is to eliminate sludge, varnish, debris and contaminated or degraded fluid from conductor walls and other internal surfaces, and system dead spots. Reasons for performing a system flush include:

- Fluid degradation - resulting in sludge, varnish or microbial deposits.
- Major failure - combined with filter overload disperses debris throughout the system.
- New or overhauled equipment - to purge 'built-in' debris.

Common methods for flushing hydraulic systems include:

- Double oil and filter change.
- Mechanical cleaning.
- Power flushing.

The technique or combination of techniques employed will depend on the type of system and its size, reliability objectives for the equipment and the reason for the flush.

Double oil and filter change

This technique involves an initial oil drain and filter change, which expells a large percentage of contaminants and degraded fluid. The system is then filled to the minimum level required and the fluid circulated until operating temperature is reached and the fluid has been turned over at least five times. The oil is drained and the filters changed a second time. An appropriate oil analysis test should be performed to determine the success of the flush. To maximize the effectiveness of this technique, the system should be drained as thoroughly as possible and the reservoir mechanically cleaned.

Mechanical cleaning

Although not technically a flushing technique, the selective use of mechanical cleaning may be incorporated in the flushing strategy. This can involve the use of a pneumatic projectile gun to clean pipes, tubes and hoses (see exhibit 1), and disassembly of the reservoir and other components for cleaning using brushes and solvents. Mechanical cleaning is labor intensive and therefore costly. It carries with it reliability risks associated with opening the hydraulic system and intervention by human agents.

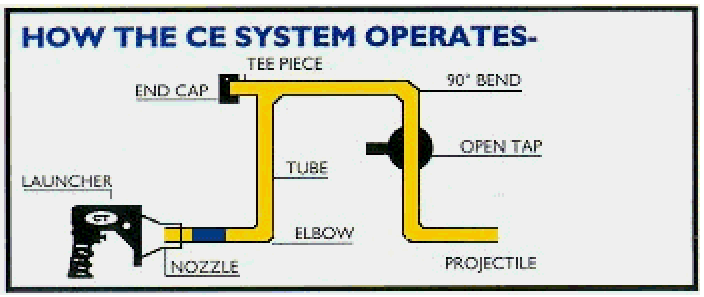


Exhibit 1. Pipe/hose cleaning projectile (Compri Technic).

Power flushing

Power flushing involves the use of a purpose-built rig to circulate a low viscosity fluid at high velocities to create turbulent flow conditions (Reynolds number > 2000). The flushing rig is typically equipped with a pump that has a flow rate several times that of system's normal flow, directional valves, accumulators, fluid heater and chiller and of course, a bank of filters. The directional valves enable the flushing direction to be changed, the accumulators enable pulsating flow conditions and the heater and chiller enable the fluid temperature to be increased or decreased, all of which can assist in the dislodgment of contaminants. Analysis of the flushing fluid is performed regularly during the flushing operation to determine the point at which the system has been satisfactorily cleaned.

What about components?

The question of how to deal with system components arises when contemplating a system flush. Plumbing should be flushed first in isolation from pumps, valves and actuators. Once the conductors have been flushed clean, valves and actuators can be gradually included in the flushing circuit. The decision to disassemble and mechanically clean components will depend on the type of equipment, your reliability objectives and the reason for the flush.

Prevent or cure?

With the exception of new or overhauled equipment, the need to flush a hydraulic system generally represents a failure of maintenance. If you follow an effective proactive maintenance program like the one I outline in Insider Secrets to Hydraulics, it's likely that you'll never need to flush.

The anatomy of hydraulic vane pump failure

One of our readers wrote to me recently regarding the following problem:

"Recently, we bought a used hydraulic power unit (15HP electric motor directly coupled to a vane pump). A high pitched, clicking noise is generated when the unit runs. We have checked the following:

1. We thought it was a motor bearing, so we detached pump from motor, no noise heard.
2. Pressure line was connected to tank line (to simulate low pressure < 100 psi), very little noise heard.
3. As pressure is increased, noise gets louder and louder, very intolerable.
4. Measured current draw of motor - no overload.

What do you think could be causing the excessive noise?"

Given that the symptoms described above are consistent with a restriction at the pump inlet, I inquired if there was a suction filter in the circuit. Our reader replied:

"The system has a 40 micron suction filter but I haven't checked it because I have to drain the oil and take off the access hatch to get to the filter."

The restriction caused by a suction filter, which increases at low fluid temperatures (high viscosity) and as the element clogs, increases the chances of a partial vacuum developing at the pump inlet. Excessive vacuum at the pump inlet causes cavitation erosion and mechanical damage.

Cavitation erosion

When a partial vacuum develops in the pump intake line, the decrease in absolute pressure results in the formation of gas and/or vapor bubbles within the fluid. When these bubbles are exposed to elevated pressures at the pump outlet they implode violently. When bubbles collapse in proximity to a metal surface, erosion occurs. Cavitation erosion contaminates the hydraulic fluid and damages critical surfaces.

Mechanical damage

When a partial vacuum develops at the pump inlet, the mechanical forces induced by the vacuum itself can cause catastrophic failure. In vane pump designs, the vanes must extend from their retracted position in the rotor during inlet. As this happens, fluid from the pump inlet fills the void in the rotor created by the extending vane. If excessive vacuum exists at the pump inlet - it will act at the base of the vane. This causes the vanes to lose contact with the cam ring during inlet, and they are then hammered back onto the cam ring as pressurized fluid acts on the base of the vane during outlet (figure 1). The impact damages the vane tips and cam ring, leading rapidly to catastrophic failure.

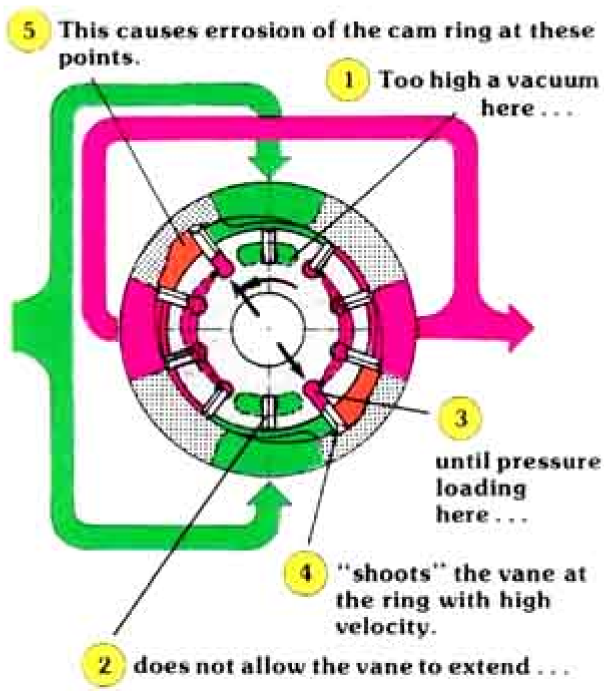


Figure 1. Vane pump section (Bosch Rexroth Corp).

The intolerable noise our reader is referring to is symptomatic of cavitation bubble collapse and the vanes being hammered against the cam ring. Both of these conditions are intensified by increasing system pressure.

The solution to our reader's problem is simple: replace the suction filter or better still, discard it completely. If suction filtration must be installed, follow these precautions to prevent pump damage:

- A filter located outside of the reservoir is preferable to a suction strainer. The inconvenience of servicing a filter located inside the reservoir is a common reason why suction strainers go unserviced - until after the pump fails.
- If a suction strainer is installed, opt for 250 microns rather than the more common 150 microns.
- The filter should be grossly oversized for the pump's flow rate to ensure that pressure drop is minimized, even under the most adverse conditions.
- Regardless of the type of filter employed, it must incorporate a bypass valve to prevent the element from creating a pressure drop that exceeds the safe vacuum limit of the pump.
- A gauge or transducer should be installed downstream of the filter to enable continuous monitoring of absolute pressure at the pump inlet.

For more information on hydraulic failures and how to prevent them, read Preventing Hydraulic Failures.

Pressure intensification in hydraulic cylinders

A question that I'm asked regularly is "What is the best way to test the integrity of the piston seal in a double-acting hydraulic cylinder?"

There is a simple bench-test for doing this but it involves the intensification of pressure in the cylinder. While the test procedure is safe if you understand the concept of intensification in a hydraulic cylinder - it is inherently dangerous if you don't.

In this article I will explain the dangers of intensification in a double acting cylinder and in a future article I will explain the test procedure.

Cylinder force

Force produced by a hydraulic cylinder is a product of pressure and area (F = p x A). In a conventional double-acting cylinder the effective area and therefore force produced by the piston and rod sides of the cylinder are unequal. It follows that if the rod side of the cylinder has half the effective area of the piston side, it will produce half the force of the piston side for the same amount of pressure.

Pressure intensification

The equation F = p x A can be transposed as p = F/A that is, pressure equals force divided by area. If the rod side of the cylinder has to resist the force developed by the piston side, with only half the area, then it needs double the pressure. This means that if the piston side is pressurized to 3,000 PSI a pressure of 6,000 PSI will be required on the rod side to produce an equal force. This is why pressure intensification can occur in a double-acting cylinder. Note that pressurizing a cylinder rated at 3,000 PSI, to 6,000 PSI can have devastating consequences.

If, for any reason, the piston side of a double-acting cylinder is pressurized and at the same time fluid is prevented from escaping from the rod side, pressure will increase (intensify) in the rod side of the cylinder until the forces become balanced or the cylinder fails catastrophically. Consider the following scenario one of our newsletter readers described to me recently:

"It was minus 36 degrees here the other day and we had a cylinder at about minus 10 degrees. The boss was attempting to press out a pin. He turned on the pump and moved the lever, and the gland end of the cylinder blew out. It was a 7.5" cylinder with a 2,500 PSI operating pressure."

The gland on this cylinder blew out as a result of pressure intensification due to a blockage between the rod side of the cylinder and tank. Possibly due the cold conditions, that is the ambient temperature had fallen below the pour point of the hydraulic fluid.

Safety is paramount

As you can see, pressure intensification in a double-acting cylinder is a dangerous phenomenon and the concept must be thoroughly understood when testing hydraulic cylinders.

Testing Hydraulic Cylinders

In a previous article, I described the danger associated with the intensification of pressure in double-acting hydraulic cylinders. In this article, I will explain how to use the intensification effect to test the integrity of the piston seal in a double-acting cylinder. Before attempting this test procedure, it is absolutely essential that the danger associated with pressure intensification in a cylinder is fully understood. Therefore, read this article first!

The conventional way of testing the integrity of the piston seal in a double-acting cylinder is to pressurize the cylinder at the end of stroke and measure any leakage past the seal. This is commonly referred to as "end-of-stroke bypass test" (demonstrated here).

The major limitation of the end-of-stroke bypass test, is that it generally doesn't reveal ballooning of the cylinder tube caused by hoop stress as a result of under designed cylinder wall thickness or reduction of wall thickness through excessive honing. The ideal way to test for ballooning of the cylinder tube is to conduct a piston-seal bypass test mid-stroke. The major difficulty with doing this is that the force developed by the cylinder has to be mechanically resisted, which in the case of large diameter, high-pressure cylinders is impractical.

However a mid-stroke bypass test can be conducted hydrostatically using the intensification effect. The necessary circuit is shown in Figure 1 below.

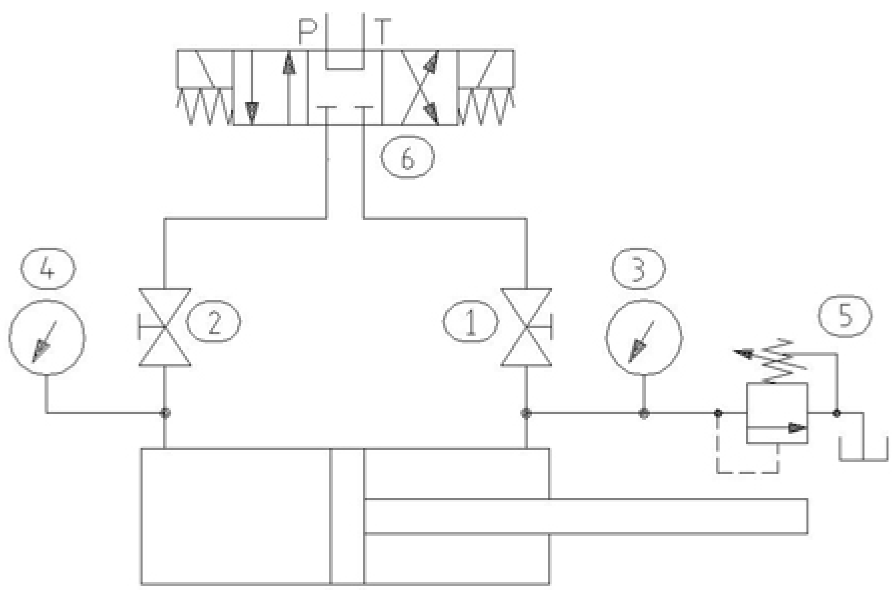


Figure 1. Hydraulic cylinder test circuit.

Test procedure

The procedure for conducting the test is as follows:

1. Secure the cylinder with its service ports up.
2. Fill both sides of the cylinder with clean hydraulic fluid through its service ports.
3. Connect ball valves (1) and (2), gauges (3) and (4), relief valve (5) and directional control valve (6) as shown in Figure 1.
4. With ball valves (1) and (2) open, stroke the cylinder using the directional control valve (6) multiple times to remove all remaining air from both sides of the cylinder - take care not to 'diesel' the cylinder.
5. Position the piston rod mid-stroke and close ball valve (2).
6. With the adjustment on the relief valve (5) backed out, direct flow to the rod side of the cylinder.
7. Increase the setting of relief valve (5) until the cylinder's rated pressure is seen on gauge (3).
8. Close ball valve (1) and center directional control valve (6). **Note:** it is assumed that the hydraulic power unit used to conduct the test has its own over-pressure protection - not shown in Figure 1.
9. Record the respective pressure readings on gauges (3) and (4) and monitor any change over time.

If the ratio of effective area between the piston and rod side of the cylinder is 2:1, then if the rod side of the cylinder has been pressurized to 3,000 PSI, gauge (2) on the piston side should read 1,500 PSI. If the differential pressure across the piston is not maintained, this indicates a problem with the piston seal or tube.

Safety is paramount

Under no circumstances should flow be directed to the piston side of the cylinder with ball valve (1) closed. Failure of the cylinder and personal injury could result. When conducting this or any other hydrostatic (pressure) test, always wear appropriate personal-protection equipment.

Reducing hydraulic cylinder repair costs

As a product group, cylinders are almost as common as pumps and motors combined. So if you operate a lot of hydraulic equipment, it's likely that cylinder repair expenses are a significant portion of your total maintenance costs.

It is often stated that up to 25% of mechanical equipment failures are failures of design. If we extrapolate this to hydraulic cylinders, as many as one in four hydraulic cylinders are not adequately designed for the application they are operating in. This doesn't mean that the cylinder won't do the job asked of it, it will - but not with an acceptable service life. If you have a particular cylinder that requires frequent repair, you may need to address one or more of these design-related problems:

Bent Rods

Bent rods are a common cause of rod seal failure. Bending of cylinder rods can be caused by insufficient rod diameter or material strength, improper cylinder mounting arrangement or a combination of all three. Once the rod bends, excessive load is placed on the rod seal resulting in premature failure of the seal. The permissible rod loading for a cylinder in an existing application can be checked using the Euler formula. A detailed explanation of how to do this is contained in Industrial Hydraulic Control.

Ballooned Tubes

Ballooning of the cylinder tube is usually caused by insufficient wall thickness and/or material strength for the cylinder's operating pressure. Once the tube balloons, the correct tolerance between the piston seal and tube wall is lost and high-pressure fluid bypasses the seal. This high velocity fluid can erode the seal and localized heating caused by the pressure drop across the piston reduces seal life.

Insufficient Bearing Area

If the internal bearing areas in the gland and at the piston are insufficient to carry the torsional load transferred to the cylinder, excessive load is placed on the rod and piston seals. This results in deformation and ultimately premature failure of the seals.

Rod Finish

The surface finish of the cylinder rod can have a dramatic effect on the life of the rod seal. If the surface roughness is too low seal life can be reduced through inadequate lubrication. If the surface roughness is too high, contaminant ingression is increased and an unacceptable level of leakage can result.

In the context of extending cylinder service life, consider the surface of the cylinder rod as a lubricated wear surface and treat it accordingly. In some applications, the use of an alternative rod surface treatment with superior mechanical properties to conventional hard chrome plating, such as black nitride or High Velocity Oxygen Fuel (HVOF) metal spraying, can increase the service life of the rod and its seals. The installation of a shroud to protect the rod surface and seals from impact damage and contaminants can afford similar life extension benefits.

Repair or Redesign?

Not all hydraulic cylinders are made equal. So if you have hydraulic cylinders that suffer recurring failure, it's likely that modifications to the cylinder are required to break the vicious circle of failure and repair.

Dealing with bent hydraulic cylinder rods

In last month's issue we examined ways to reduce the recurring cost of hydraulic cylinder repairs. If you missed this article or for a recap, it's available here. One of the topics discussed was cylinder rod buckling loads. In response to this article, many or our readers wanted to know how to deal with bent rods in a repair situation.

Checking rod straightness

Rod straightness should always be checked when a hydraulic cylinder is being re-sealed or repaired. This is done by placing the rod on rollers and measuring the run-out with a dial gauge (Figure 1). Position the rod so that the distance between the rollers (L) is as large as possible and measure the run-out at the mid-point between the rollers (L/2).

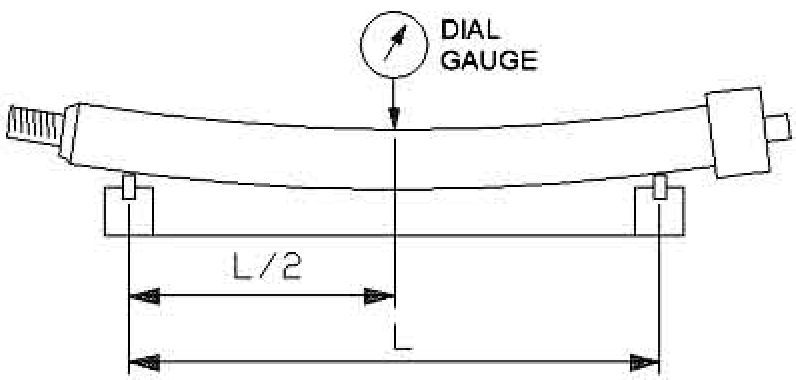


Figure 1. Checking rod straightness.

Allowable run-out

The rod should be as straight as possible, but a run-out of 0.5 millimeters per linear meter of rod is generally considered acceptable. To calculate maximum, permissible run-out (measured at L/2) use the formula:

Run-out max. (mm) = 0.5xL/1000
Where: L equals distance between rollers in millimeters.

For example, if the distance between the rollers was 1.2 meters, then the maximum, allowable run-out measured at L/2 would be given by 0.5 x 1200 / 1000 = 0.6mm.

Dealing with bent rods

In most cases, bent rods can be straightened in a press. It is sometimes possible to straighten hydraulic cylinder rods without damaging the hard-chrome plating, however if the chrome is damaged, the rod must be either re-chromed or replaced.

If a rod is bent, then it is wise to check actual rod loading against permissible rod loading based on the cylinder's mounting arrangement and the tensile strength of the rod material. The formulas and procedure for doing this are explained in detail in Industrial Hydraulic Control. If actual road load exceeds permissible load then a new rod should be manufactured from higher tensile material and/or the rod diameter increased to prevent the rod from bending in service.

Understanding Hydrostatic Balance

Hydraulic components are unique in that it is often possible to offset or balance hydrostatic forces to reduce loads on lubricated surfaces. By reducing surface loading, the maintenance of full-film lubrication is improved and therefore boundary lubrication conditions are less likely to occur.

Hydrostatic force is the product of pressure and area. Expressed mathematically: $F = P \times a$. The balancing or offsetting of hydrostatic force is achieved by exposing opposing areas to the same pressure. The double-acting cylinder in Figure 1 illustrates this concept.

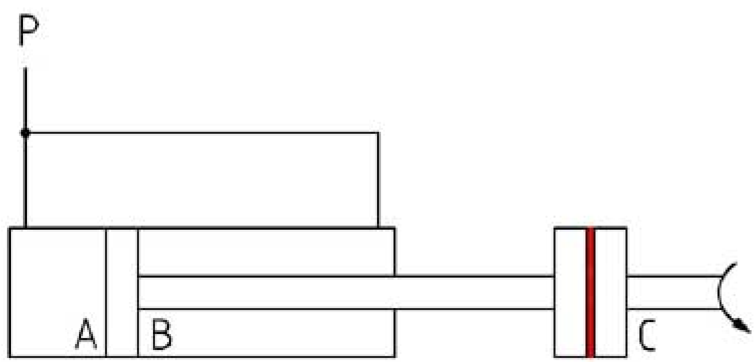


Figure 1. Hydrostatically balanced cylinder loading two lubricated surfaces.

The rod-side area of the piston, area B, is 80% of area A. This means that the force exerted on the lubricated surfaces at the end of the cylinder rod is 20% of the force developed by the pressure acting on area A. This is due to the balancing or offsetting force developed by the same pressure acting on area B. Assuming the speed of the rotating surface (C) and fluid viscosity are adequate, full-film lubrication of the sliding surfaces is achieved.

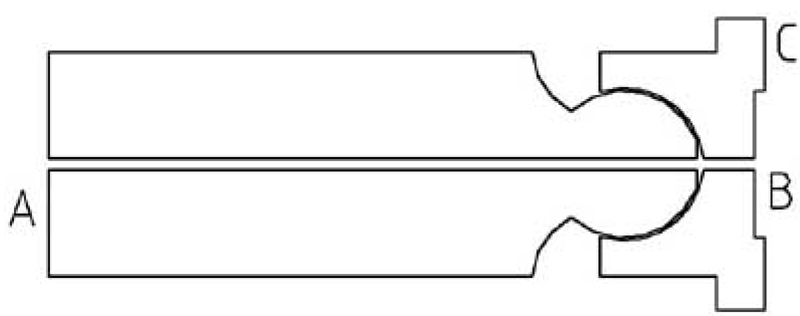


Figure 2. Typical cross-section of an axial design piston.

The same principle applied to a typical axial design piston is illustrated in Figure 2. Area A is exposed to system pressure during outlet (pump) or inlet (motor) and the force developed is transmitted to the lubricated surfaces of the slipper and swash plate. System pressure also acts on area B, the balancing area of the slipper, via the drilling through the center of the piston. Area C is the sliding (lubricated) area of the slipper. While the ratio of these three areas varies, in this particular piston, area B is 50% of area A and area C is 140% of area A. This means that the force transmitted to area C is half that developed by area A and is spread over 1.4 times the area, further reducing the load on the lubricated surfaces.

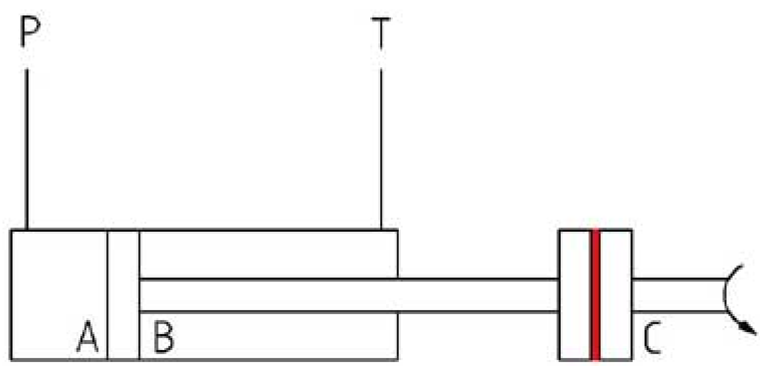


Figure 3. Loss of hydrostatic balance increases load on the lubricated surfaces.

If the hydrostatic balancing force is lost, that is there is no pressuring acting on area B (Figure 3), the force exerted on the lubricated surfaces at the end of the cylinder rod will be 100% of the force developed by the pressure acting on area A. If full-film lubrication was dependent on the hydrostatic balance of the cylinder, boundary lubrication conditions will eventuate and two body abrasion is likely. To learn more about the construction of hydraulic components, their modes of failure and how to prevent them, visit www.preventinghydraulicfailures.com.

Hydraulic filter condition monitoring

Continuous monitoring of the filter elements in a hydraulic system can provide valuable clues to the performance of the filter and the condition of the system. Before I discuss this, let's consider some of the advantages and disadvantages of common filter locations.

Pressure filtration

Locating filtering media in the pressure line provides maximum protection for components located immediately downstream. Filtration rates of two microns or less are possible, due to the positive pressure (in comparison to an intake line filter) available to force fluid through the media (in comparison to an intake line filter). Filter efficiency may be reduced by the presence of high flow velocities, and pressure and flow transients, which disturb trapped particles. The major disadvantage of pressure filtration is economic. Because the housings and elements (high-collapse type) must be designed to withstand system operating pressure, pressure filtration has the highest initial and ongoing cost.

Return filtration

The rationale for locating filtering media in the return line is this: if the reservoir and the fluid it contains start out clean and all air entering the reservoir and returning fluid is adequately filtered, then fluid cleanliness will be maintained. The other advantage of the return line as a filter location is that sufficient pressure is available to force fluid through fine media - typically 10 microns, but pressure is not high enough to complicate filter or housing design. This combined with relatively low flow velocity, means that a high degree of filtering efficiency can be achieved at an economical cost. For these reasons, return filtration is a feature of most hydraulic systems.

Off-line filtration

Off-line filtration enables continuous, multi-pass filtration at a controlled flow velocity and pressure drop, which results in high filtering efficiency. Filtration rates of two microns or less are possible, and water absorbent filters and heat exchangers can be included in the circuit for total fluid conditioning. Off-line filtration has a high initial cost, although this can often be justified on a life-of-machine cost basis.

Filter condition monitoring

Warning of filter-bypass is typically afforded by visual or electric clogging-indicators. These devices indicate when pressure drop (delta P) across the element is approaching the opening pressure of the bypass valve (where fitted).

In the case of a return filter for example, if the bypass valve opens at a delta P of 3 Bar, the clogging indicator will typically switch at 2 Bar.

Advanced filter condition monitoring

Replacing standard clogging-indicators with differential pressure gauges or transducers enables continuous, condition monitoring of the filter element. This permits trending of fluid cleanliness against filter element pressure-drop, which may be used to optimize oil sample and filter change intervals. For example, the optimal change for a return filter in a particular system could be higher or lower than the clogging indicator switching pressure of 2 Bar.

Continuous monitoring of filter pressure drop can also provide early warning of component failures and element rupture. For example, if the delta P across a pressure filter suddenly increased from 1 to 3 Bar (all other things equal), this could be an indication of an imminent failure of a component upstream. Similarly, a sudden decrease in delta P could indicate a rupture in the element - something that a standard clogging indicator will not warn of.

How to avoid hydraulic troubleshooting mistakes

Troubleshooting hydraulic systems can be a complex exercise. It involves a lot of science and sometimes, a bit of art. Incorrect diagnosis prolongs downtime and can result in the unnecessary repair or replacement of serviceable components. To avoid these costly mistakes, the correct equipment and a logical approach are required.

Assess the problem and eliminate the obvious

Before you incur the expense of hiring a technician, assess the problem and eliminate all of the obvious, possible causes. I have lost count of the number of times that I've been called to a problem and found that the cause was something quite simple. A wire broken off a solenoid valve, a pin fallen out of a mechanical linkage, an isolation valve that had vibrated closed, a blocked heat exchanger… and so the list goes on.

Your oversight won't bother the technician, because his hourly rate is the same, regardless of how easy or difficult the problem is to find. But you may be annoyed with yourself for not checking something so obvious, knowing that you could have easily saved the cost of the call-out.

Quality is more important than quantity

Paying for a technician's time when it is not required is certainly not desirable. But it is nowhere near as costly as paying for the unnecessary repair or replacement of serviceable components, as a result of incorrect diagnosis of a problem. Incorrect diagnosis in a troubleshooting situation is usually the result of the technician's incompetence, insufficient investigation of the problem or a combination of both.

Unfortunately it is not possible to determine a technician's competency from the badge on his shirt or his charge-out rate. While charge-out rates may be a factor in deciding whose technician you hire, from an overall cost perspective it is far more important to evaluate the technician and his diagnosis, so that you don't end up paying for his mistakes.

Let me illustrate how this can happen with an example. Several years ago, I was asked for a second opinion on the condition of a set of pumps operating a processing plant. The customer had called in a technician to check the performance of these pumps and was alarmed when the technician advised that all four pumps were in need of repair.

The pumps in question were variable-displacement units fitted with constant power control. The power required to drive a hydraulic pump is a product of flow and pressure. A constant power or power limiting control operates by reducing the displacement, and therefore flow, from the pump as pressure increases, so that the power rating of the prime mover is not exceeded.

Pump performance is checked using a flow-tester to load the pump and measure its flow rate. As resistance to flow is increased, pressure increases and the flow available from the pump to do useful work decreases because of internal leakage. The difference in the measured flow rate between no load and full load determines the volume of internal leakage and therefore pump performance.

I tested all four pumps, recording flow against pressure from no load through to maximum working pressure. In my report I explained to the customer that the tests revealed that pump flow did decrease significantly as pressure increased, but that this is a normal characteristic of a pump fitted with constant power control. I further advised that apart from the constant power control requiring adjustment on two of the pumps, the performance of all four pumps was acceptable.

The first technician's assessment can only be explained by fraud or incompetence. I suspect it was the latter, with the technician failing to either establish or understand that the pumps he was testing were fitted with constant power control. This ignorance led to an incorrect interpretation of the test results. Whatever the explanation, the customer could have paid thousands of dollars for unnecessary repairs, if they had not sought a second opinion.

Conclusion

When you have a problem with your hydraulic equipment, carry out an informed assessment of the problem and eliminate the obvious before you call for a technician. And if you do need to hire a technician, be sure to evaluate the technician and his diagnosis so you don't end up paying for his on-the-job-training or worse, his mistakes!

Power saving with hydraulic load sensing control

Load sensing is a term used to describe a type of variable pump control used in open circuits. It is so called because the load-induced pressure downstream of an orifice is sensed and pump flow is adjusted to maintain a constant pressure drop (and therefore flow) across the orifice. The 'orifice' is usually a directional control valve with proportional flow characteristics, but a needle valve or even a fixed orifice can be employed, depending on the application.

In hydraulic systems that are subject to wide fluctuations in flow and pressure, load-sensing circuits can save substantial amounts of input power. This is illustrated in Exhibit 1. In systems where all available flow (Q) is continuously converted to useful work, the amount of input power lost to heat is limited to inherent inefficiencies. In systems fitted with fixed displacement pumps where 100 percent of available flow is only required intermittently, the flow not required passes over the system relief valve and is converted to heat. This situation is compounded if the load-induced pressure (p) is less than the set relief pressure - resulting in additional power loss due to pressure drop across the metering orifice (control valve).

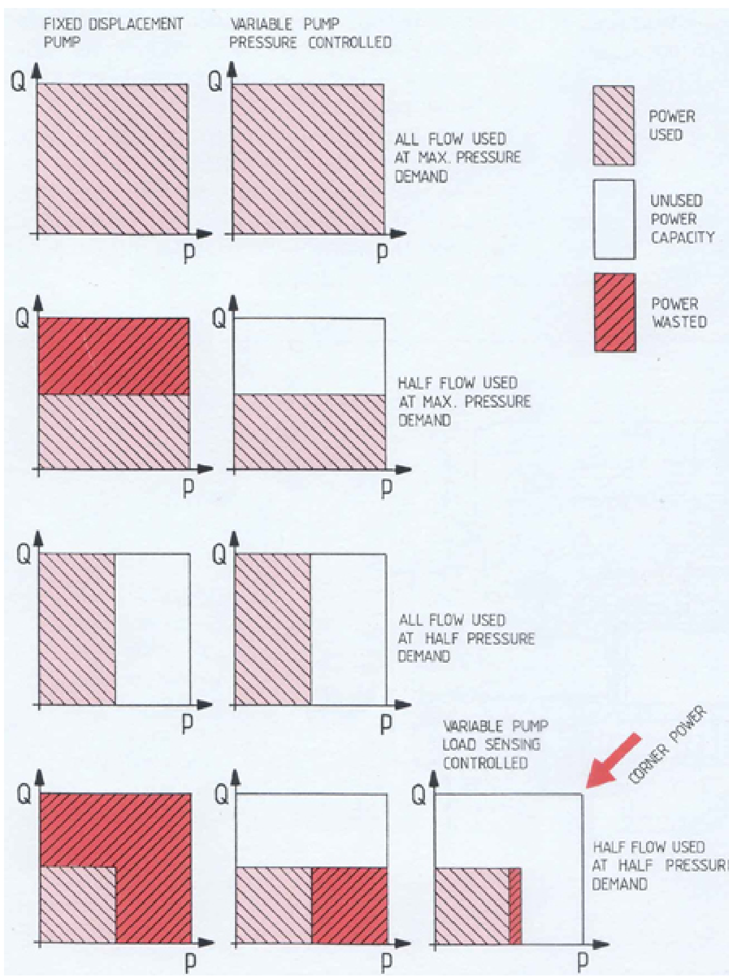


Exhibit 1. Flow-pressure-power diagrams for fixed, variable and load sensing controlled hydraulic pumps (Peter Rohner).

A similar situation occurs in systems fitted with pressure controlled (pressure compensated) variable pumps, when only a portion of available flow is required at less than maximum system pressure. Because this type of control regulates pump flow at the maximum pressure setting, power is lost to heat due to the potentially large pressure drop across the metering orifice.

A load sensing controlled variable pump largely eliminates these inefficiencies. The power lost to heat is limited to the relatively small pressure drop across the metering orifice, which is held constant across the system's operating pressure range (see bottom of Exhibit 1).

A load sensing circuit typically comprises a variable displacement pump, usually axial-piston design, fitted with a load sensing controller, and a directional control valve with an integral load-signal gallery (Exhibit 2). The load-signal gallery (LS, shown in red) is connected to the load-signal port (X) on the pump controller. The load-signal gallery in the directional control valve connects the A and B ports of each of the control valve sections through a series of shuttle valves. This ensures that the actuator with the highest load pressure is sensed and fed back to the pump control.

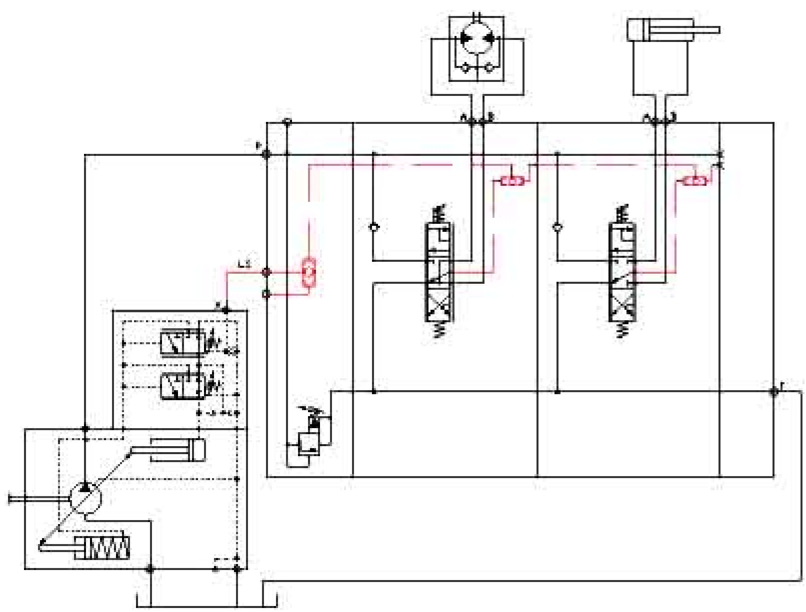


Exhibit 2. Typical load sensing circuit. Enlarge

To understand how the load-sensing pump and directional control valve function together in operation, consider a winch being driven through a manually actuated valve. The operator summons the winch by moving the spool in the directional valve 20 percent of its stroke. The winch drum turns at five rpm. For clarity, imagine that the directional valve is now a fixed orifice. Flow across an orifice decreases as the pressure drop across it decreases. As load on the winch increases, the load-induced pressure downstream of the orifice (directional valve) increases. This decreases the pressure drop across the orifice, which means flow across the orifice decreases and the winch slows down.

In a load sensing circuit the load-induced pressure downstream of the orifice (directional valve) is fed back to the pump control via the load-signal gallery in the directional control valve. The load-sensing controller responds to the increase in load pressure by increasing pump displacement (flow) slightly so that pressure upstream of the orifice increases by a corresponding amount. This keeps the pressure drop across the orifice (directional valve) constant, which keeps flow constant and in this case, winch speed constant. The value of the pressure drop or delta p maintained across the orifice (directional valve) is typically 10 to 30 Bar (145 to 435 PSI). When all spools are in the center or neutral position the load-signal port is vented to tank and the pump maintains 'standby' pressure equal to or slightly higher than the load sensing control's delta p setting.

Because the variable pump only produces the flow demanded by the actuators, load-sensing control is energy efficient (fewer losses to heat) and as demonstrated in the above example, improves actuator control. Load-sensing control also provides constant flow independent of pump shaft speed variations. If pump drive speed decreases, the load-sensing controller will increase displacement (flow) to maintain the set delta p across the directional control valve (orifice), until displacement is at maximum. To further your knowledge on load sensing and other variable pump controls, visit
http://www.IndustrialHydraulicControl.com

Dealing with hydraulic and pneumatic leaks

In a previous 'Inside Hydraulics' article I discussed the true cost of hydraulic oil leaks. In the case of oil leaks, the cost areas that need to be considered include:

- Make-up fluid;
- Clean-up;
- Disposal;
- Contaminant ingress; and
- Safety.

But what about hydraulics' fluid power cousin - pneumatics? One of the advantages that pneumatics has over hydraulics is its cleanness. Air leaks are much easier to ignore than oil leaks because they don't draw attention to themselves in the same way. You don't need to worry yourself with clean-up and disposal costs. Contaminant ingression is possible, but is generally not a major concern. And unless the leak is significant, safety is not usually a big issue either. So that leaves make-up fluid (air).

Make-up air

While air is free - clean, dry compressed air is not. In considering the cost of make-up air for a pneumatics system the following need to be considered:

- Depreciation (wear and tear) of the compressor;
- Conditioning costs - filtration, drying and lubrication; and
- Energy cost of compression.

The ideal leakage rate is of course zero, but when calculating the free air delivery (FAD) required by a pneumatic system a rule of thumb is to allow for leakage of 10% of total flow rate. Consider a 10 cubic meter/minute system leaking one cubic meter/minute. The power required to compress one cubic meter (35.3 cubic feet) of air per minute to a pressure of 6 bar (90 PSI) is approximately 5.2 kW. At an electricity cost of $0.10/kWh this leakage is costing over 50 cents per hour in electricity consumption alone. In a 24/7/365 operation that amounts to $4500 per year!

Quantifying losses

While a leakage rate of 10% of flow rate may sound high and would be unsustainable in a hydraulic system, air leakage rates as high as 25% are not unheard of - even in apparently well maintained pneumatic systems. The actual leakage rate of a system can be calculated using the following formula:

$Q_L = Q_C * t / (T+t)$

Where:
Q_L = System leakage rate (cubic meters/minute)
Q_C = Compressor FAD (cubic meters/minute)
T = Leakage time - time between compressor cut-out and cut-in (minutes)
t = Charging time - time between compressor cut-in and cut-out (minutes)
Note: this formula assumes that all system consumption is suspended while the leakage test is conducted.

Conclusion

As demonstrated by the above example, the annual cost of air leaks in pneumatic systems can be significant - in power consumption alone. Conduct regular leakage tests on your pneumatic systems and take necessary action to locate and rectify leaks as required. For more on the operation and maintenance of compressed air systems and pneumatic equipment visit:
http://www.IndustrialPneumaticControl.com

Sourcing aftermarket hydraulic components

One of our readers wrote to me recently about the following concerns:

"Are there any specific considerations I should be aware of prior to replacing pumps and motors on an excavator with aftermarket units? If I order a pump from a supplier other than the OEM, how can I be certain that the pump I'm getting is precisely the same as the one I'm replacing?"

This is a good question. In fact, I devote a whole Chapter to this topic in my book *Insider Secrets to Hydraulics*.

Aftermarket units

A machine owner will usually get a better price on a replacement hydraulic component if they are able to buy it 'aftermarket', from a fluid power distributor. However, this is not always as easy as it might seem. Original equipment manufacturers (OEM's) usually do all they can to control the distribution and sale of spare parts for the machines they build. This is particularly true in the case of mass-produced hydraulic machines. The OEM knows that if you can identify the make and model of the hydraulic pump fitted to your excavator, you will be able to shop around for the best price on a replacement pump and as a result, there is a good chance they will lose the business.

In an effort to prevent this from happening, OEM's usually identify hydraulic components fitted to their machines with their own part numbers. In most cases these numbers are meaningless to anyone else. Therefore the first thing you need to do is to identify the component's manufacturer and model code.

If you know another operator or company that owns the same machine you need the replacement component for, you could ask them if they have replaced this component with an aftermarket unit. If so, they will probably be happy to give you the model code from the aftermarket component's identification tag.

If you have a good idea of what you're looking for, you can identify the component yourself, using information that is available on the Internet. This involves first measuring and identifying the component's physical attributes such as shaft type, mounting flange, ports, displacement and control. You then need to match these variables to the dimensional and technical data contained in the manufacturer's product catalog. The catalog will show you how to compile a model code (sometimes called an order code) that corresponds to the component you want to replace. Sounds easy, but if you don't know who the manufacturer is or what the component does, this can be a difficult task. In this case, it is easier to let a fluid power distributor do the work and earn their profit margin in the process.

OEM specials

It is not always possible to source aftermarket hydraulic components for OEM equipment. OEM's sometimes use components that are manufactured with a unique difference, known as 'OEM specials'. This means that even if you do identify the make and model of the component, the only way you can buy an identical unit is through the machine dealer.

The difference may be something obvious, such as the shaft type or the orientation of the ports - or not so obvious, such as the control set-up in a variable displacement pump or motor. I can think of at least one example where the main pump for a particular excavator was, to the casual observer, a standard unit. However, if a pump from the component manufacturer's standard product line was installed on the machine, it would cause the engine to stall. Reason being, the standard pump was fitted with a hydraulic displacement control with a control range (minimum to maximum displacement) of 10 bar, whereas the OEM pump was fitted with a special control range of 35 bar. With a standard pump installed, the excavator's electronic power management system could not effectively control the pump's displacement and therefore power draw.

Obvious or not, these differences are usually enough to make it either impossible or uneconomic to adapt a unit from the component manufacturer's standard product line. Some large OEM's also manufacture their own hydraulic components. As with OEM specials, these components are a captive market for the machine dealer.

Nailing hydraulic logic element leakage

One of our readers wrote to me recently with the following question:

"In one of our applications we are using NG 40 cartridge valves (sleeve, poppet and logic cover). With the valve closed and the inlet port pressurized to 315 bar, we are seeing a leakage from the outlet port in the order of half a liter per minute. Is this level of leakage acceptable?"

The first thing to consider is whether the logic element has been configured for leakless operation. If the direction of flow is from A to B this is referred to as base flow. If flow is from B to A this is know as annulus flow (see figure 1). A logic element can be configured for flow in either or both directions.

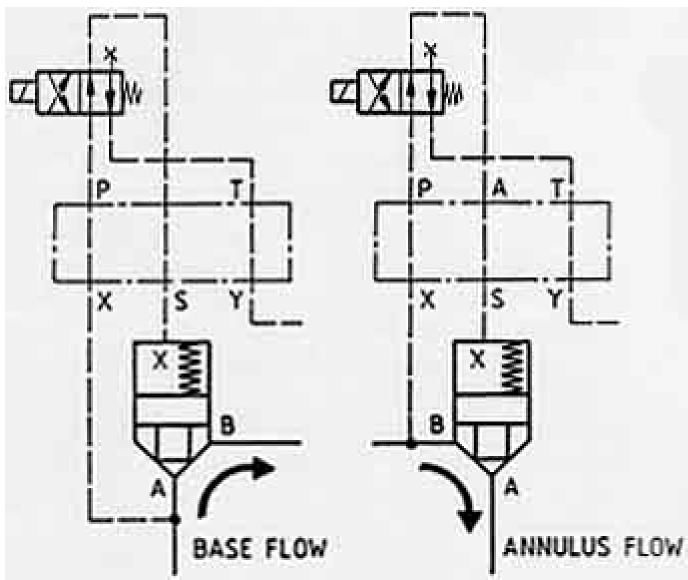


Figure 1. Logic element base and annulus flow configurations (*Industrial Hydraulic Control*).

To establish whether a logic element is configured for zero leakage, it is necessary to consider the direction of pressure drop across the poppet when it is closed. Consider a logic element configured for check valve function in both base and annulus flow directions. When configured as a check valve for base flow (A to B) see figure 2, the direction of pressure drop across the poppet when it is closed is from B to A. In this configuration the logic element is leakless.

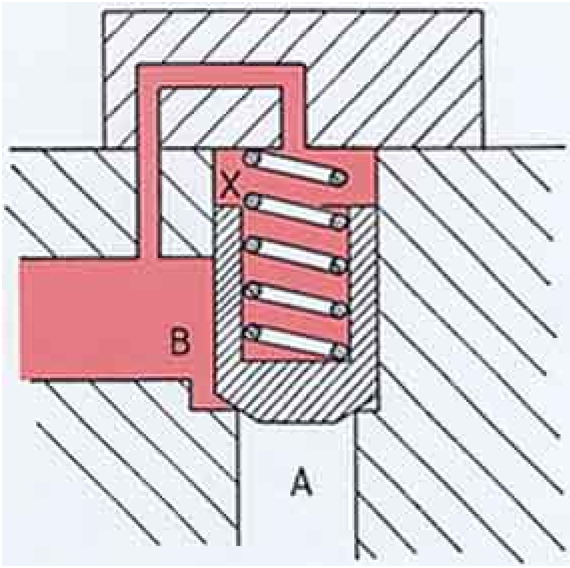


Figure 2. Logic element; check valve function; base flow (*Industrial Hydraulic Control*).

When configured as a check valve for annulus flow (B to A) see figure 3, the direction of pressure drop across the poppet when it is closed is from A to B. In this configuration the clearance between the poppet and its sleeve results in leakage from A to B. The magnitude of this leakage may increase over time as a result of wear between the poppet and sleeve.

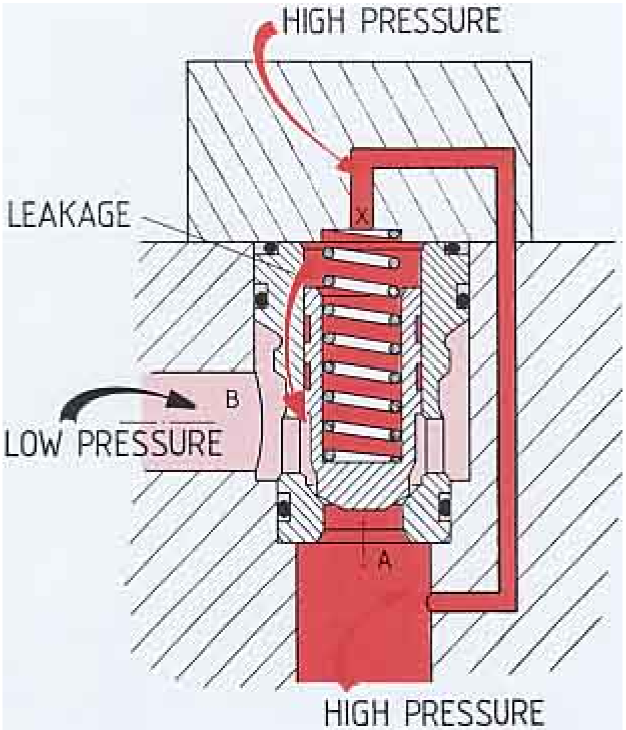


Figure 3. Logic element; check valve function; annulus flow (*Industrial Hydraulic Control*).

Assuming our reader's logic elements **have** been configured for leakless operation, other possible explanations for the leakage include:

- damage to the poppet and/or its seat
- degradation or damage to the elastomeric seal at the base of the sleeve
- incorrect machining tolerance in the logic housing

In this, and all other troubleshooting situations, the first place to look for guidance is the machine's circuit diagram and your reference library. From there on, it is a logically process of elimination.

Putting the brake on water hammer in hydraulics

One of our newsletter readers wrote to me recently about the following problem:

" We have a hydraulic power unit that runs a calendar roll on a paper machine. The calendar roll has multiple zones inside it to vary the pressure on the paper to maintain uniform thickness. Moog servo-valves control the adjustments to the different zones. The problem we have is high vibration in the pumps, lines and tank. Lines have broken and we had to disconnect one bank of filters from the tank to keep it from breaking off. This has been going on for some time. Do you have any ideas?"

As always, there are a number of possibilities and issues to consider. Some background reading on vibration and noise in hydraulic systems is available in this tutorial. One explanation that jumps to the top of the list in this application is water hammer.

Coining a phrase

Water hammer is the term used to describe the effect that occurs when the velocity of the fluid moving through a pipe suddenly changes. Sudden change in fluid velocity causes a pressure wave to propagate within the pipe. Under certain conditions, this pressure wave can create a banging noise, similar to that you would expect to hear when beating a pipe with a hammer. Hence the phrase. Not surprisingly, common symptoms of this problem are high noise levels, vibration and broken pipes.

Hitting the wall

When a moving column of fluid hits a solid boundary - when a directional control valve closes suddenly for example, its velocity drops to zero and the fluid column deforms, within the rigid cross-sectional area of the pipe, to absorb the (kinetic) energy associated with its motion - similar to a car hitting a concrete wall. However unlike a car, the fluid is almost incompressible so the deformation is small and a store of energy accumulates in the fluid - similar to compression of a spring. The magnitude of the pressure rise that results from the subsequent release of this stored energy can be expressed mathematically as follows:

$$P_r = P + u\,p\,c$$

where P is initial pressure, u and p are initial fluid velocity and density respectively and c is the speed of sound through the fluid.

In our reader's application, uniform paper thickness is dependent on the constant adjustment of the calendar roll zones by the servo-valves. Under certain conditions, rapid switching of these valves could result in something that resembles peening a pipeline with a thousand hammers.

Speed kills

Accumulators and other damping devices are sometimes installed in an effort to deal with this problem. However, the significance of the pressure rise equation shown above is that fluid velocity is the only variable that can be altered to address the **root cause**. Put simply, reducing the velocity of the fluid column that hits the solid boundary, reduces the magnitude of the subsequent pressure rise. Returning to the traffic crash analogy - the slower the car is travelling when it hits the wall, the less damage is caused.

In hydraulics, the easiest way to do this - on paper at least, is to increase the diameter of the pipe. This reduces fluid velocity for a given flow rate. The other alternative is to control deceleration of the fluid column by choking valve switching time to the point where the pump's pressure compensator and/or system relief valve reacts fast enough to reduce flow rate through the pipe and therefore velocity of the fluid.

Priority number one in hydraulics maintenance

I presented a workshop on minimizing hydraulic equipment operating costs at a local University recently. During that presentation, I shared with the attendees what I consider to be THE most important proactive maintenance routine for hydraulic equipment.

No, it's **not** contamination control. These days, best-practice contamination control is an accepted precondition for reliability. And given contemporary advances in technology for excluding and removing contaminants, it could be said that failure to control contamination is a failure of machine design - rather than a failure of maintenance.

The maintenance routine that I believe ranks above contamination control in order of importance these days - largely due to its neglect, is: **maintaining fluid temperature and viscosity within optimum limits**. This involves:

1. Defining an appropriate fluid operating temperature and viscosity range for the ambient temperature conditions in which the hydraulic machine operates;
2. Selecting a hydraulic fluid with a suitable viscosity grade and additive package; and
3. Ensuring that both fluid temperature and viscosity are maintained within the limits defined.

In order to determine the correct fluid viscosity grade for a particular application, it is necessary to consider:

- starting viscosity at minimum ambient temperature;
- maximum expected operating temperature, which is influenced by system efficiency, installed cooling capacity and maximum ambient temperature; and
- permissible and optimum viscosity range for individual components in a system.

For example, consider an application where the minimum ambient temperature is 15°C, maximum operating temperature is 75°C, the optimum viscosity range for the system's components is between 36 and 16 centistokes and the permissible, intermittent viscosity range is between 1000 and 10 centistokes.

From the temperature/viscosity diagram exhibit 1, it can be seen that to maintain viscosity above the minimum, optimum value of 16 centistokes at 75°C, an ISO VG68 fluid is required. At a starting temperature of 15°C, the viscosity of VG68 fluid is 300 centistokes, which is within the maximum permissible limit of 1000 centistokes at start up.

Having established the correct fluid viscosity grade, the next step is to define the fluid temperature equivalents of the optimum and permissible viscosity values for the system's components.

By referring back to the temperature/viscosity curve for VG68 fluid shown in exhibit 1, it can be seen that the optimum viscosity range of between 36 and 16 centistokes will be achieved with a fluid temperature range of between 55°C and 78°C. The minimum viscosity for optimum bearing life of 25 centistokes will be achieved at a temperature of 65°C. The permissible, intermittent viscosity limits of 1000 and 10 centistokes equate to fluid temperatures of 2°C and 95°C, respectively (see exhibit 2).

Viscosity Value	cSt	Temperature (VG68)
Min. Permissible	10	95°C
Min. Optimum	16	78°C
Opt. Bearing Life	25	65°C
Max. Optimum	36	55°C
Max. Permissible	1000	2°C

Exhibit 2. Correlation of typical operating viscosity values for a piston pump with fluid temperature, based on fluid viscosity grade.

Going back to our example, this means that with an ISO VG68 fluid with a viscosity index similar to that shown in exhibit 1 in the system, the optimum operating temperature is 65°C. Maximum operating efficiency will be achieved by maintaining fluid temperature in the range of 55°C to 78°C. And if cold start conditions at or below 2°C are expected, it will be necessary to pre-heat the fluid to avoid damage to system components. Intermittent fluid temperature in the hottest part of the system, which is usually the pump case, must not exceed 95°C.

Having defined the parameters shown in exhibit 2 for a specific piece of hydraulic equipment, damage caused by high or low fluid temperature (low or high fluid viscosity) can be prevented, and recurring PM tasks in respect of this routine can be virtually eliminated, by installing fluid temperature monitoring instrumentation with alarms and shutdowns.

Make sure YOU know the 'animal' you're dealing with

On September 4, 2006 Steve Irwin, also know as "The Crocodile Hunter", lost his life in a freak accident.

He was the blonde Australian guy you may have seen on Discovery Channel or Animal Planet, wrestling crocodiles or hugging sharks. One report said he was known to 500 million people around the world.

Steve was snorkelling will bull rays, in waters off Northern Australia. These 250 pound (100 kg) beasts, with wing spans of three feet (1 meter) or more are not normally aggressive to man.

Apparently though, one of these sting rays, feeling threatened by the close attention of Steve and his cameraman, turned and struck-out, plunging its 8-inch (20 cm) long, venomous barb into Steve's chest. All attempts to revive him were in vain.

It's a reminder of brevity of life. It is also a reminder that we should know the 'animal' we are dealing with. Consider this report on a freak hydraulics accident I came across recently:

An operator was using a high-pressure hydraulic tool, when the hose ruptured at the ferrule. As a result, high-pressure fluid came into contact with the operator's hand.

On presenting at Emergency, the initial prognosis was "keep clean and rest". By chance, a specialist doctor observed and intervened.

The mineral oil had already started to "eat away" fatty tissues in the hand and was travelling up the arm. The injured person had five operations to cut away oil deposits and at one point faced the prospect of losing his arm.

To fully appreciate the damage that hydraulic fluid under pressure can cause, you need see this photo of the injury. If you are squeamish, **BE WARNED** it is very graphic.

Steve Irwin was a professional. He would have known what he was dealing with. Make sure you know the 'beast' you're dealing with. Educate yourself. Make full use of the resources available to you at www.hydraulicsupermarket.com And if you ever have any doubts when working on hydraulic equipment, consult a qualified engineer or technician.

If you have friends or colleagues who work with hydraulics, you'll be doing them a good turn by passing this along to them.

Solving hydraulic cylinder squeal

In last month's newsletter, we set the scene for an unusual hydraulic troubleshooting story. If you missed it, it's available here.

The outcome of this story follows, as told by Doug Lien, founder and former owner of Cylinder City Inc.

"The squeal was present at all engine speeds - so flow rate / stroke speed was not an issue. Through a process of elimination, I isolated or changed out all cylinder valves. No difference.

I then removed the cylinder from a machine and placed it on a bench. I connected the cylinder to the same new machine with quick disconnects. I removed the rod seal and under no load I stroked the cylinder and it still squealed. I then removed the piston seal and the squeal remained. So it continued to squeal with no seals installed.

I reinstalled the seals and connected the cylinder to an older machine, which was referred to as the "yard machine", while the cylinder was still on the bench. The squeal was gone! I drained the oil from the cylinder and reconnected it to the new machine -- the squeal was back! This eliminated the cylinder.

The General Manager at the OEM was still insisting that I contact my factory and get a crew of men to his plant and change out all 80 cylinders (he was what you might call a duck hunter). By now though I was confident that the cylinders were not the cause of the problem. I asked him to give me another day in order to determine the root cause.

Earlier that day I noticed two things that put me on the right trail. One was the color of the oil that was draining from the cylinder when I disassembled it. The oil looked like ivory soap. The other was a slight hissing sound when I removed the head gland from the cylinder. I knew at this point that aeration was causing the squeal.

But the challenge was trying to convince the customer's engineers. I had to prove where the air was coming from and why. So the big question was: What was different about the new model machine? Were the pumps changed? Was the design of the hydraulic tank changed in any way? How about the suction lines or suction strainer? The answer to all these questions was the same: NO.

When I had connected the cylinder to the new machine the cylinder squealed, when I connected it to the older machine it was quiet. When switching the cylinder from new machine to old machine I had made sure that the cylinder was completely drained of hydraulic fluid.

So now back to the questions. Have you changed oil viscosity, brand, supplier or formula? Again the answer was NO. I insisted that the oil was the source of air in the system and that air was causing the squeal. The oil vendor was called and he stated that to the best of his knowledge the oil was the same. I insisted that the oil vendor contact the blender to ask if the additive package had been changed. They agreed and the next morning a chemist from the oil blender was on site.

In the meantime I went to the nearest CAT dealer and got some of CAT's Anti-Chatter hydraulic oil additive. I added a quart to a machine that was squealing and within 5 minutes the squeal was gone.

Needless to say at this point all eyes turned to the oil blender and their chemist. It turned out that the formula of the oil had been changed and the amount of air-release additive had been reduced. All 80 machines had the hydraulic fluid replaced and their shipment to customers was back on schedule."

Storing hydraulic cylinders - safely

A question I've been asked several times in recent months by equipment owners, is the procedure for storing spare hydraulic cylinders for an extended period. So here's what I recommend:

- Always store fully retracted.
- Store indoors in a clean, dry area.
- Smear the internal surfaces of eye/clevis bushes or bearings with grease - particularly if they're steel.
- Protect any exposed chrome on the rod. Oil-impregnated tape such as Denso tape can be used for this purpose. Before applying, make sure the rod is fully retracted. If a product like Denso tape is applied to the rod when the rod is not fully retracted, subsequent retraction of the rod can result in damage to the rod seal.
- Plug the service ports with steel - not plastic, plugs or blanking plates.
- Consider filling the cylinder with clean hydraulic oil through rod-end service port. Particularly if it is an expensive, large diameter or high pressure cylinder. I say "consider" because there are a few issues to understand before you do this.

If the cylinder is not filled with oil it will obviously be filled with air. If this air is not perfectly dry, then as ambient temperature decreases the air can reach due point. This results in moisture forming on the inside of the cylinder tube. This can cause spot rusting and pitting of the tube surface, which will reduce the volumetric efficiency of the cylinder, the service life of the piston seal, and ultimately, the life of the tube itself.

Completely filling the cylinder with clean hydraulic oil prevents this from occurring, however there's a **major caution** with this. It's best illustrated by an example:

Say a cylinder is prepared for storage during the winter months. When the cylinder is filled with oil, the ambient temperature is 10 degrees Celsius. A year and a half later, during the middle of summer, the same cylinder is set down beside the machine to which it is to be installed. In the heat of the midday sun the temperature of the cylinder rises to 40 degrees Celsius. Assuming an infinitely stiff cylinder, the pressure of the oil in the cylinder resulting from the rise in temperature can be approximated by the formula:

$p \text{ (bar)} = 11.8 \times (T2 - T1)$

So the theoretical pressure of the oil in the cylinder is now: $11.8 \times (40 - 10) = 354$ bar or 5134 PSI! When it comes time for the unsuspecting mechanic to crack loose the blanks on the service ports … well let's just say that's more excitement that he signed up for.

That said, cylinders CAN be safely filled with oil for storage provided you:

1. Check that the worst-case temperature rise in storage won't result in a static pressure that exceeds the cylinder's working pressure.
2. Only fill the cylinder when fully retracted and ONLY through the rod-end port. This avoids potentially dangerous pressure intensification.
3. Use service port plugs or blanks that are rated for the cylinder's working pressure.
4. Attach appropriate warning tags to BOTH service ports.
5. Provide a means to check and vent any pressure before each of the service port blanks is removed. A simple way to do this is to fit each port blank with a pressure test-point. This enables the quick attachment of a pressure gauge to check the pressure in the cylinder. And if necessary, the pressure can be safely vented into a drum using a test-gauge hose.

As you can see, this procedure is somewhat involved and so the decision to fill a cylinder with oil is something you have to weigh up based on the value of the cylinder and how long you expect it to be in storage.

Oh, and the moral to the above story is: if you get involved in installing hydraulic components, when it comes time to remove blanking plates or plugs - always assume there's a possibility the component contains oil under pressure. And take the necessary precautions.

Storing hydraulic cylinders - Part 2

In a previous newsletter article, I discussed the procedure I use when preparing hydraulic cylinders for storage. If you missed it, it's available here.

In response to this article, one of our members sent in this question:

"*One issue I feel you left out of the cylinder storage issue is the orientation question. How should a cylinder be orientated for short term or long term storage?*

Our company repairs and evaluates cylinders and their associated failures. We try to provide solutions for breakdowns as well as repairing those that have failed. We evaluate and repair approximately 1000 cylinders annually. One common issue has been seal failure particularly in large pneumatic and hydraulic cylinders. We have found that allowing cylinders to lay flat has a direct effect on piston and rod seal failures. We have instituted a cylinder storage standard that adheres to your recommendations as far as ports plugged, rods wrapped but in addition mandates that all cylinders are stored vertically - in such a position as not to distort or place the weight of the cylinder on the rod seals. This verticality also helps, we feel, the piston seals.

I wouldn't mind hearing your argument on this issue."

Hmmm. I intentionally didn't mention it because I didn't want to do anything to perpetuate the myth. Because based on my experience, that's exactly what it is. Two cases I was indirectly involved in come to mind. In both cases the cylinders in question were off 400 ton mining-size hydraulic excavators. We are talking here about cylinders that weigh between two and three tons. The piston rod typically weighs well over a ton by itself.

So you have a situation where big, expensive, high-pressure cylinders are suffering premature seal failures. In both cases, the machine operators sought the advice of "seal experts". The recommendation of these supposed experts was to store the cylinders vertically.

Let's consider the reality of this nonsense:

Someone drops a three ton cylinder with a closed length of four meters at your feet and tells you to store it vertically - so it doesn't fall over and destroy itself - or worse still, kill someone. Not a five-minute job, but possible I suppose.

A truck arrives to transport the cylinder to a remote mine-site. The route consists of 1,000 miles of rough, unsealed road. Given you have gone to the trouble of storing the cylinder vertically in the warehouse, surely you must insist that it is transported in the same orientation? I mean, if it can't be stored horizontally in a shed, then surely the pounding it is going to get if it's laid down on the back of a truck will turn the seals into mush, right? The truck driver thinks you're crazy but he doubles his rate and obliges anyway.

The cylinder arrives at the mine-site in the mandated vertical position. Trouble is it's a stick cylinder so it's orientation on the machine is horizontal. If the bearing bands on the piston and in the gland can't adequately support the piston rod and prevent it from distorting the seals when the cylinder is sitting in a shed or bouncing around on the back of a truck, how on earth will it cope with the thrust developed when it goes into service on a 400 ton excavator?

Common sense would tend to suggest that if the bearing bands have sufficient area and are correctly tolerenced to adequately support load-induced thrust without distorting the seals, then surely they will cope with the static weight of the piston-rod in storage and any dynamic loading that may occur during transport?

Whether you agree with this assessment or not, you know troubleshooting is a process of elimination. So when seal failures continue to occur even after the cylinders have been stored vertically ... well it's safe to say that's not the root cause of the problem. And that was the outcome in the two situations I mentioned above. No surprise to me.