LUBRICATION AND SEAL OIL SYSTEMS— COMMON PROBLEMS AND PRACTICAL SOLUTIONS

by

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ABSTRACT

Current technologies now make it possible to build turbomachinery without the need for a lubrication and seal oil system. However, there is an established machinery population that still requires support systems for oil wetted bearings and seals. For most companies, it is not economically practical to convert to new bearings and seals to eliminate the oil support system. Indeed, this may not be the best solution in many cases.

Lubrication and seal oil systems can be the single largest cause of spurious and nuisance trips. In most cases, such trips are due to controller degradation, improper operation, or simple neglect. The most common causes of oil system problems, and practical solutions aimed at long term operating reliability are addressed.

A chronic cause of spurious trips is oil pump switchover. Usually, such problems can be eliminated by upgrading controllers and/or control valve selection philosophy, installation of hydraulic pulsation dampeners, a design audit of the piping configuration, and, very often, operating procedures.

Oil system instability contributes significantly to nuisance trips. Again, a review of system control components and associated hardware can usually minimize this phenomenon. Additionally, improper purging of air has been known to cause similar problems, and frustrate the most experienced operator.

There are other problems which can also result in unscheduled downtime of critical rotating machinery. These include collapsing oil filter elements, improper sour seal oil drainer sizing, improper control valve sizing, lack of attention to transient conditions, controller and pressure switch physical location, and oil cooler integrity, to name a few.

There is a large installed machinery population that has been in operation for many years, but has never been upgraded to present standards. New developments in control technology lend themselves to increasing oil system reliability, in addition to other component upgrades.

Refineries and chemical plants now commonly utilize comprehensive preventive maintenance (PM) programs; however, oil systems do not seem to be included in such programs to the same level as that of their supported machinery. Oil system reliability can be increased, and nuisance trips minimized, by monitoring several key operating points which include valve position, oil temperature and chemical analysis, and filter differential pressure (to name a few) on a scheduled basis. Such data can also be useful in determining or predicting problems with the rotating equipment.

It has also become obvious in recent years that process demands, in addition to efforts to improve oil system reliability, have resulted in systems that are more complex. Such complexity has, in some cases, ironically decreased operational reliability due to the introduction of unnecessary hardware. Practical and realistic basic design guidelines will be presented, along with a comparison of the various design philosophies currently being utilized in the industry.

INTRODUCTION

The reliability of critical turbocompressors in today's petrochemical industry is typically greater than 90 percent. If and when a trip does occur, it is often one of the support systems that initiates the outage. The most common offender of such a situation is the lubrication and seal oil system.

Lubrication and seal oil systems are crucial to the long term operation and profitability of a compression string. However, such systems receive minimal attention after installation; they only become the focus after causing a spurious trip. Cursory attention to operating details on a continuous basis would virtually eliminate such nuisance trips. Unfortunately, such attention is usually not in an operating schedule.

The installation list of conventional compressors and the respective support systems is very large. What can be done to improve the reliability of existing installations, specifically, the lubrication and seal oil systems? Many of the problem areas that typically cause operational problems of oil systems are presented here, along with practical recommendations for improvements and enhancements. The authors do not attempt to conduct a rigorous analysis of each problem area; indeed, each specific subject could qualify for an article in its own right. Instead, the authors' objective is to highlight the major points of each problem area, to present practical guidelines for resolution, and to ignite reactions within the reader to chronic problems in their respective equipment areas.

BASIC SYSTEM DESCRIPTION

It is necessary to establish a basic oil system description for the purposes herein. In order to allow the reader to more closely follow the discussion that follows, a basic lubrication and seal oil system will be described as one which incorporates the following equipment:

- Combined lube and seal
- One reservoir
- One turbine driven main oil pump (MOP) and one motor driven auxiliary oil pump (AOP)
- Duplex coolers and filters
- One back pressure regulating valve (BPRV)
- One lube oil pressure reducing valve (PRV)
- One differential pressure regulating valve (DPRV)

A schematic representation of the above system is shown in Figure 1. The authors acknowledge that there are a wide variation of oil systems in service. However, the basic system described facilitates the discussion that follows.

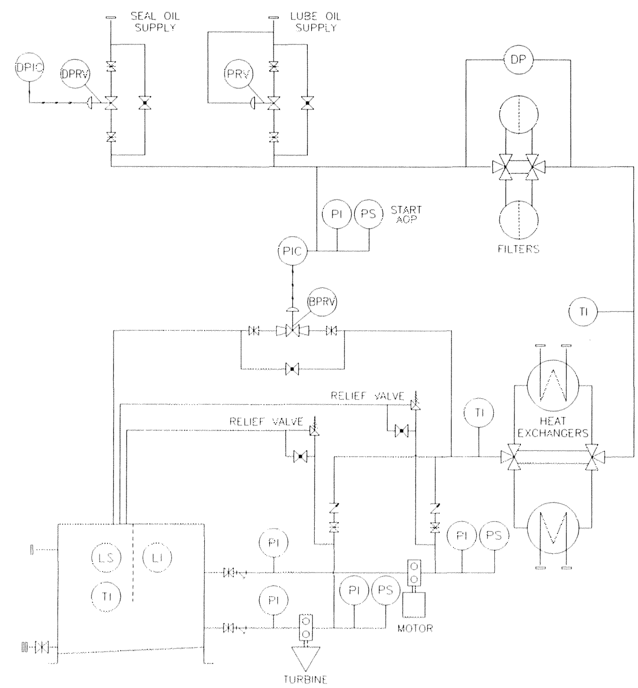

Figure 1. Basic Oil System.

MAIN PUMP TO STANDBY PUMP TRANSIENTS

Automatic starting of stand by oil pumps is the greatest contributor to spurious trips of critical machinery. It is not uncommon for a rotating string to trip on low lube oil or seal oil pressure when the AOP starts automatically for any of a number of reasons. Although the equipment does not typically suffer any significant damage, the loss of production, the potential for process upsets, and the safety considerations make such an occurrence very serious indeed.

Pump Piping

Both inlet and discharge pump piping must be properly configured to assure proper automatic transfer of pump duty without affecting oil pressure at the unit. Inlet piping must be sized to limit oil velocity to three to four ft/sec to minimize the chances of pump cavitation. Pump inlet piping must be arranged so as to preclude the collection of air under all operating conditions. Pump ingestion of air during pump startup will increase the pressure drop in the controlled system header, and surely cause a drop in oil delivery pressure at the unit, usually below the trip setting. Proper inlet piping design includes minimizing potential air pockets or traps in the external piping and internal reservoir piping, and using eccentric reducers instead of concentric reducers, as shown in Figure 2. As illustrated by this figure, the concentric reducer has the potential of trapping air in the upper portion of the pipe. Additionally, minimizing pump suction bends and fittings helps to reduce piping losses and assure smooth pump inlet fluid flow.

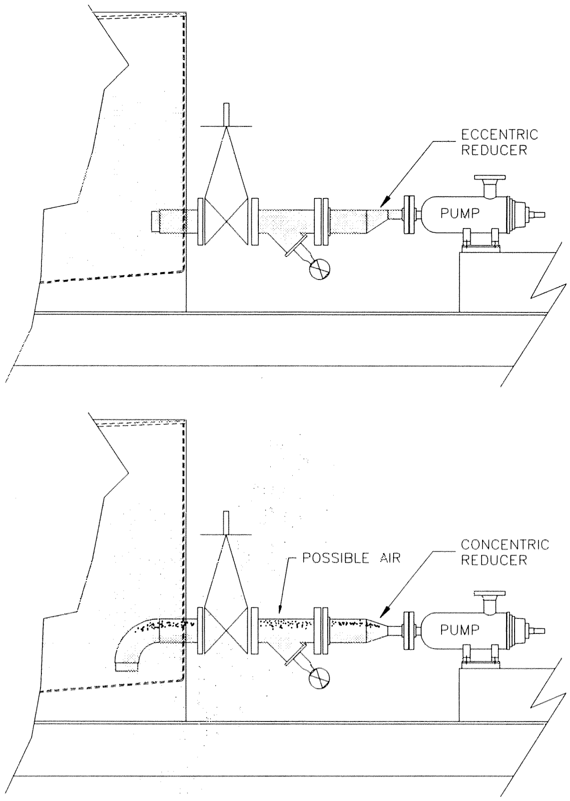

Figure 2. Concentric Reducer in Pump Suction Line.

Inlet suction strainers can affect pump performance, especially during startup. To minimize pump starvation or cavitation problems, pump suction strainer sieve opening should be no smaller than *0.0165 in* (40 mesh). Oil systems for rotating equipment can be considered a clean service; as such, the pump suction strainer is utilized only to stop large objects such as nuts, bolts, gasket material, etc. Strainers are *not* intended to prefilter the oil into the pumps to micron cleanliness. Strainers with a mesh size greater than 40 should be evaluated for pressure drop, and removed or changed if necessary. A compound pressure indicator should be located between the respective pump inlet and strainer to alert the operators to a low inlet pressure due to blockage or dirt.

Internal reservoir configuration will directly affect pump performance. The reservoir should be baffled to provide an isolation between the various returns (main equipment, relief valves, back pressure regulator, etc.) and the pump suction, as shown in Figure 3. The objective is to provide a tranquil area on the pump suction side of the baffle allowing deareation and stilling of the oil. Oil returns impinging on the pump suctions can and will affect pump performance, both during startup and normal operation, by disturbing suction flow and contributing to air ingestion. If required, internal baffles can be added during a planned outage. Such baffles can be welded in place after the proper preparation, or can be mechanically installed and fastened. Mechanically fastened baffles can be necessary for internally coated reservoirs which would suffer damage from welding or cutting. Optionally, an improperly placed return can be relocated to the furthest practical section of the reservoir.

Swirl plates utilized on the pump suction pipe entrance eliminates any fluid whirl or turbulence which will affect pump performance. It is also advisable to have all returns enter the reservoir beneath the oil level, and terminate with swirl plates. Swirl plates minimize the possibility of aeration of the oil and foaming. The typical use of swirl plates is illustrated in Figure 4.

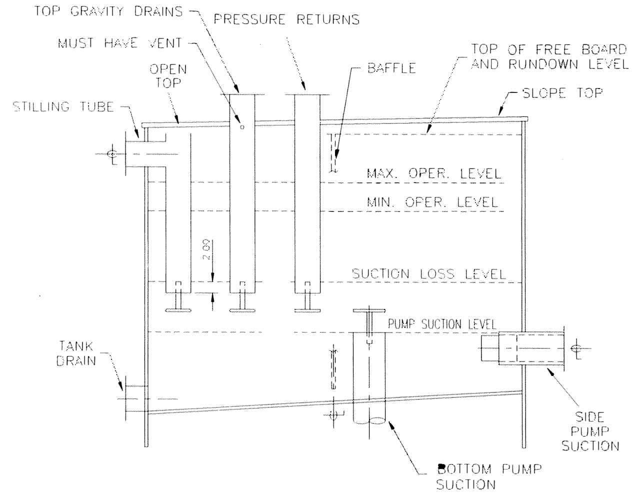

Figure 3. Oil Reservoir Layout Illustrating Baffles and Returns.

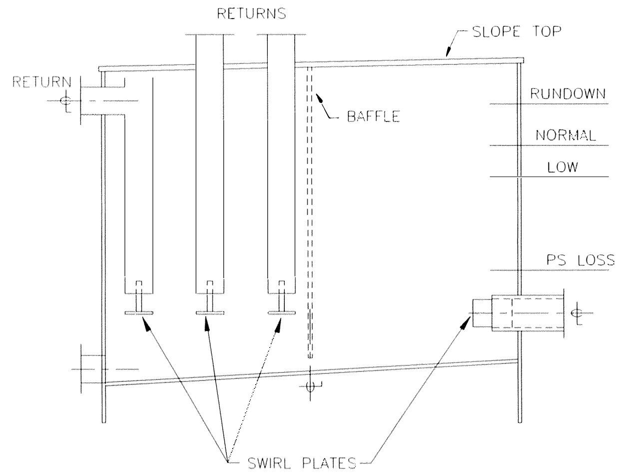

Figure 4. Detail of Swirl Plates on Pump Suctions and Returns.

Heat transfer from the hot oil reservoir to the idle pump is minimal, even with short suction piping. There have been several cases where the ambient temperature of a positive displacement standby oil pump has been much lower than that of the oil in the reservoir, such that the thermal shock on startup caused massive damage to the pump. Although such a problem will occur only in very cold climates, be attentive to such a situation, especially during a system flush, when oil temperatures can be as high as 180°F.

Pump discharge piping must be arranged to eliminate potential air pockets. This is especially critical to pump discharge check valve and RV placement (if applicable). If either or both of these components are located above the oil reservoir level, an air pocket will surely accumulate while the pump is idle, as shown in Figure 5. Any air in the pump discharge line will delay pressure accumulation during startup, and cause a trip in many cases. Such problems can be easily remedied with a constant bleed from the MOP.

The arrangement of the pump discharge header with respect to the back pressure regulator tap-in is very important in improving the response of an automatic pump start. Specifically, the back pressure regulator tap-in must be downstream of the common pump discharge connection, as shown in Figure 1. This configura-

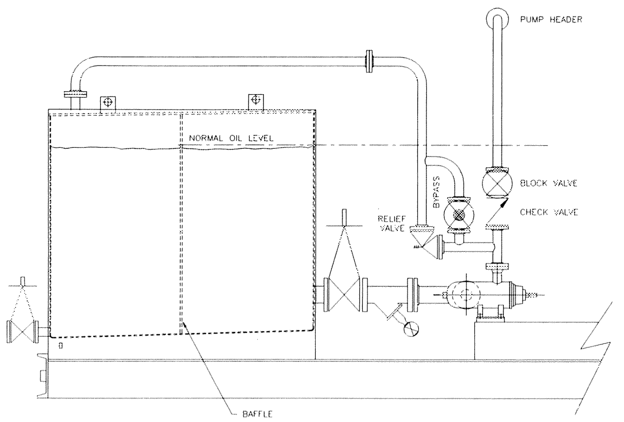

Figure 5. Location of Pump Discharge Check Valve and Relief Valve.

tion assures no time delays due to flow reversals as the pumps stop and start.

Standby Pump Startup

A reliable standby oil pump startup is of utmost importance in eliminating spurious trips of the rotating equipment. The AOP can be started in two different manners:

• automatically on loss of oil pressure or level

• manually as required by process, utility, equipment, or maintenance demands.

Automatic standby oil pump starting necessitates the installation of some type of instrumentation which senses oil system pressure, or oil level in a vessel (e.g., an overhead seal oil tank or auxiliary rundown tank). The integrity and reliability of the instrumentation, control logic, and switch setpoints directly affect the success of an automatic pump switchover.

Switch setpoints are logically dictated by the location of the respective switch in the oil system. Pressure instruments utilized for automatic startup of the standby oil pump are typically located in any of three areas:

• the controlled system header downstream of the oil filters (most common)

• the pump discharge header

• the machine in the respective delivery line (lube oil or seal oil).

Pump start pressure switches are commonly located in the controlled header downstream of the filters, as this provides a stable pressure reference, and facilitates the establishment of a workable setpoint. Generically, this is the recommended location for the start AOP switch(es).

Establishing the setpoint of the switch (on falling header pressure) is critical. If the pressure decay is too small, the pump will constantly start due to minor oil system pressure fluctuations. On the other hand, if the pressure decay is too large, the lag time between the drop in system pressure and starting of the AOP will result in a momentary pressure drop at the unit, which will invariably result in an unwanted trip.

Generally, a switch setting corresponding to a 15 psi decay works well for systems with controlled header pressures below 300 psig. Above 300 psig, it is advantageous to use a decay that is one to five percent of the controlled header pressure (or 15 psi, whichever is greater, and depending on the magnitude of the controlled pressure).

Placement of the AOP start switch in the pump discharge piping presents a new set of problems since pump discharge pressure floats with oil system resistance dictated by the cleanliness of the oil filters or flow transients. It may be difficult to assign a generic number to this particular configuration due the number of variables involved. However, a pressure decay of 35 psi can be used as a benchmark, and further tuning done during field testing of startup response. Generally, do not locate the pump start switches at a position where the pressure floats.

The AOP start switch can be placed directly at the unit, either in the lube oil supply line, seal oil supply line(s), or both. Setpoints for such switches are typically the same as the respective low oil pressure alarm setting. However, the lag time between the starting of the AOP and trip can be so small that a unit shutdown is almost guaranteed. Locating the start AOP switch(es) at the unit is not recommended.

There have been cases where a HAND-OFF-AUTO switch for the standby oil pump driver has been inadvertently left in the OFF position, thus negating a pump start when called upon. It is recommended that such switches be changed to the HAND-AUTO configuration with a separate OFF switch to prevent such mishaps.

In cases where a liquid level switch is used to start the AOP (e.g., in an overhead seal oil tank), the pump start liquid level will be set during design of the tank (based upon total rundown time), and is generally not adjustable.

When selecting or replacing the AOP start switch(es), attention must be paid to the dead band of the switch ("deadband" is defined as the range between activation and resetting of the respective switch). Obviously, a switch with a 20 psi deadband used in a controlled header with a 15 psi decay will not be satisfactory. Select a switch with a deadband which is less than the anticipated header decay pressure (a typical deadband for a start AOP switch is 10 psi; this will change, depending on the pressure range of the switch). Be certain to record deadband when calibrating the switch during routine maintenance. If erratic automatic pump starting is a problem, verify that the existing switch deadband is smaller than the header decay.

It is also important to assure that the AOP will remain on line after it has been signaled to start. A latching relay, installed in the AOP motor control center (MCC), is required to assure that the pump will remain on until manually stopped.

Starting the AOP while the MOP is on line is the greatest momentary shock that the oil system will experience in any operating mode. This is due to the sudden introduction of excess oil flow into the system, and the lag time in the BPRV response. The next worst condition is automatic stopping of the MOP and starting of the AOP, due again to pressure fluctuations and lag times in valve response.

API oil systems are designed for starting and stopping oil pumps without a major system disturbance, and without a unit trip. However, at times when conditions dictate that the AOP be started at the operator's discretion (e.g,, when the MOP requires emergency maintenance), it is prudent to avoid starting the AOP automatically, due to control degradation and system sensitivity. This is especially applicable to systems utilizing positive displacement pumps. Unknown instrument integrity and controller calibration warrant reducing the risks of a unit trip by conducting a manual switchover when conditions permit.

Instead, it is recommended both oil pumps be gently valved into and out of the system, thus allowing the BPRV to respond to changes in system flow. This switchover is facilitated by the use of manual RV bypass valves, as shown in Figure 1.

The following procedure will assure a smooth transition from the main (or operating) pump to the standby pump.

• Open the RV bypass on the idle pump.

- Start the idle pump. Total pump flow will be diverted to the reservoir through the bypass line.

- Slowly close the RV bypass until completely closed. Full pump flow will now be passing through the BPRV, in addition to the excess flow from the MOP.

- Slowly open the RV bypass on the main pump; the BPRV will respond to the change in flowrate by slowly closing. Continue until the RV bypass is fully open, and full pump flow is passing through the bypass line back to the reservoir.

- At this point, the pump can be secured, and either prepared for maintenance or now used as the standby oil pump.

If a RV bypass is not available, as is common, the above transfer procedure cannot be utilized. Two transfer options exist. One option is to try manually starting the standby pump, as follows:

- manually start the standby pump
- verify stable operation of the oil system
- manually stop the main pump (and hope that valve response is sufficiently fast so as to prevent a trip)

Note that the low oil pressure trips can be bypassed for this condition only, but can be very risky. This procedure poses a general risk of a unit shutdown unless all controls and instrumentation have been inspected prior to the transfer.

The other method involves using the pump discharge block valves to selectively isolate and commission the respective oil pumps. This method, as described below, will minimize the possibility of a trip due to an oil pressure change.

- Close the isolation block valve on the idle pump completely.

- Start the idle pump. Total pump flow will now be passing through the RV back to the reservoir (verify that local regulations permit such operating modes).

- Slowly open the isolation valve until fully opened. The BPRV will respond by opening to pass the second pump flow back to the reservoir.

- Slowly close the isolation valve on the main pump, until completely closed. Total pump flow will now be passing through the RV back to the reservoir, and the BPRV will return to a normal operating position.

- Shut down the pump driver, and secure.

Field Testing

The oil system pump switchover performance should be tested on a regular basis (every one to six months, depending on site conditions) by the operations department, and definitely verified prior to start up of the unit after maintenance or an unscheduled shutdown.

Obviously, the safest time to conduct any testing is while the unit is shut down. This also allows the opportunity to investigate various "what if" scenarios, and allows operators the time to practice manual pump transfers. Off line testing is simply a matter of commissioning the MOP, preparing the standby pump, tripping the main pump and then observing system response. A pressure drop below any of the trip setpoints should be immediately annunciated by the unit alarm panel. If indeed a trip would have been detected, further testing and problem analysis can be conducted on a comfortable basis.

On line testing, however, represents a greater risk of a unit shutdown. With proper precautions, such risks can be minimized. The following procedure can be used or modified as required to initiate a safe on line pump switchover. The purpose of this test is only to verify that the pump will start automatically in the event of a system pressure loss; it is not intended to observe oil system performance. It should be noted that the unit will be temporarily without the availability of the standby pump during the test, which does indeed represent a mechanical risk to the unit itself.

- Prepare the pump for automatic startup.

- Open the standby pump RV bypass (or close the pump discharge valve).

- Install a pressure gauge at the pump start pressure switch, and isolate the switch. Slowly bleed the pressure trapped in the line, and observe the pressure at which the pump starts.

- Shut off the pump after testing is complete and close the RV bypass (or open the pump discharge valve, if applicable).

A true pump switchover can be conducted with the rotating string on line; however, unless all controls and instruments are in proper working order and calibration, such a test is inviting a trip at the very least, and damaged bearings, seals, and rotor at the worst. Some users have disabled the trips momentarily during such tests, but have had personnel positioned in strategic areas and poised to react in the event of a problem.

Transient Dampers

Transient dampers are devices (mechanical or electrical) that artificially maintain oil pressure for a short time period (accumulators), dampen the pressure signal (hydraulic snubbers), or delay actuation of unit trip circuitry during a pressure drop (electrical time delays).

Accumulators can be placed in the controlled header, lube oil supply line, seal oil supply line, or all three. Generally, however, the accumulator is placed in the controlled header, downstream of a check valve to prevent reverse oil flow during system pressure drops. Sizing of the accumulator is based upon the startup characteristics of the AOP driver.

A motor driven pump will accelerate quickly (depending on the frame size), and will thus require a smaller accumulator than required by a slower accelerating turbine driven pump. An accumulator should provide oil for three to five seconds on a motor drive application, and at least 20 seconds on a turbine drive application. There are many theories as to the precharge pressure of a bladder type accumulator. A precharge pressure of 80 to 85 percent of the start AOP switch set pressure generally provides satisfactory response during a pump switchover.

When using accumulators in the controlled header, it is very important to verify that the AOP start switch and sensing line for the BPRV are located upstream of the check valve, as shown in Figure 6. Location of the pressure switch downstream of the check valve will assure that the accumulator discharges before the AOP is called upon to start, thus rendering the accumulator ineffective. Location of the BPRV sensing line downstream of the check valve will allow the valve to remain open while the accumulator is discharging, thus dumping oil back to the reservoir, instead of diverting oil to the equipment where it is needed to maintain delivery pressures, and thus, prevent a trip.

Hydraulic dampers (snubbers) are damping devices that fit in the instrument pressure sensing line between the pressure source and the respective instrument(s). Essentially, a snubber is a hydraulic time delay. Snubbers damp the signal into the pressure switch in one or both directions. For purposes of this discussion, the oil signal must be damped *away* from the respective pressure switch. In the event of a pressure drop (during automatic starting of an AOP), the snubber does not permit the oil pressure at the switch to decay as quickly as the delivery line oil pressure. As such, a momentary pressure loss will not result in a shutdown of the rotating equipment string, where a total loss of oil pressure will still cause a shutdown, as is required in such a situation.

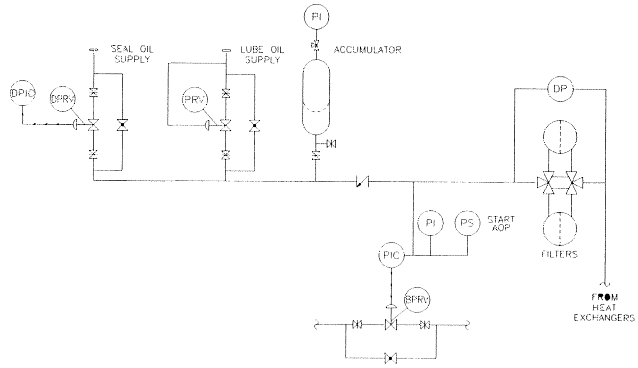

Figure 6. Accumulator Installation in Controlled Header.

An electrical time delay is placed at the low oil pressure trip switch(es) contact outputs, such that actuation of the trip switch on low oil pressure will activate a timer. The timer contacts then activate the equipment trip devices after a predetermined time interval if the oil pressure has not been restored. If, however, oil pressure recovers in a short period of time, as is typical of pump switchovers, the timer contacts will be reset by the resetting contacts of the trip switch before activation, and an unwanted shutdown is avoided. Typical time delay settings are one to three seconds; however, there have been cases where time delays have been set as high as fifteen seconds. The use of time delays includes a risk of equipment damage if not properly engineered, and must be understood by the user. If indeed this option is pursued, it is strongly recommended that the turbomachinery manufacturer be consulted before implementation.

REGULATING AND CONTROL VALVES

The words "control" and "stability" are terms that are often used equally, but incorrectly. They do, however, denote different concepts. An instrument's ability to control a setpoint does not necessarily mean that the system is stable; conversely, a stable system does not mean that design setpoints are being maintained.

Control is the ability to maintain a setpoint within a certain range. Stability is the ability to respond to a change in the setpoint, and return the system to that setpoint without undue disturbance or loss of overall control. On the basic oil system, the back pressure regulator is intended to maintain the header pressure under all operating conditions; this includes automatic switchover of the oil pumps. In this case, the controller or valve must respond quickly to prevent the overall system pressure from dropping to the point that a shutdown occurs. However, if response (or tuning) is too fast, the control will overshoot, and control of the system will be lost (the system will now be unstable). Conversely, a slow controller (or valve) will maintain stability in a system, but the setpoint could drift beyond acceptable limits (loss of control). It is the job of the instrument technician to tune the controller (or valve) to the point that both control and stability are maintained to an acceptable degree. The point to be made is that very fast response of the control system does not mean that the system will be stable.

The root cause of poor oil system performance is often due to the pressure regulating and control valves. Poor sizing, application, or response can mean the difference between the oil system absorbing transients versus an unwanted turbomachinery shutdown.

Sizing and Arrangement

During normal service, the operating position of the oil system pressure regulating valves should be between 40 and 60 percent of total design travel. Valves operating significantly outside of this range can be unstable if operating too close to the valve seat, or unable to control large transient conditions if too far open.

Back pressure regulator sizing is critical to acceptable oil system performance. This valve must to be of a fail closed design to insure turbomachinery oil supply should failure occur. Flow sizing input must include:

- Excess of one pump running minus minimum oil supply flow requirements.
- Excess of one pump running minus normal oil supply flow requirements.
- Excess of one pump running minus maximum oil supply flow requirements.
- Excess of two pumps running minus normal oil supply flow requirements.

Valve sizing criteria for establishing the pressure operating range must include the following:

- *Minimum pressure* differential buildup based upon the design header pressure, in addition to component pressure drops.
- *Maximum pressure* differential based upon the design header pressure, in addition to component pressure drops.

Normally, the back pressure regulator branches from the pump discharge header as is shown in Figure 1, to facilitate system component sizing for only the turbomachinery demand. If oil system heat loads are predicted to be high due to high pump horsepower, the back pressure regulator may be branched downstream of the heat exchangers. The pressure sensing point for the back pressure regulator should always be located at the section where it is best to maintain a steady state pressure. In most cases, this sensing point is upstream of the supply control valves to minimize the pressure regulating valve inlet pressure variations.

Although the sizing parameters for PRVs are usually not as rigorous as those for the BPRV, the PRVs must be capable of responding quickly to upstream and downstream transients. The PRVs should be fail open action. Flow sizing criteria must include the following:

- Minimum oil flowrate.
- Normal oil flowrate.
- Maximum oil flowrate.

The differential pressure across the valve body should be determined by subtracting the upstream pressure (usually the controlled header pressure) from the pressure immediately downstream of the valve body. This downstream pressure is determined by the following criteria:

- Minimum oil pressure required by the turbomachinery, plus an allowance for static and dynamic pressure losses (see the section on interconnecting piping).
- Normal oil pressure required by the turbomachinery, in addition to the static and dynamic pressure losses.
- Maximum oil pressure required by the turbomachinery, in addition to the static and dynamic pressure losses.

As shown in Figure 1, the pressure sensing points for the PRVs are preferably located immediately downstream of the respective valve. This configuration is intended to minimize response time by maintaining short piping or tubing lengths.

Improper actuator valve travel adjustment or bench calibration will often masquerade as incorrect valve sizing. Improper travel adjustment can have the valve operating at mid stroke, when the valve plug should actually be at a full open or closed position, as dictated by the real time operating requirements. Consequently, valve control is affected, since the valve is not synchronized with the control signal. By verifying that the actual valve travel is per design specifications, it is simple to determine if further adjustments are required. Improper bench settings (the pressure range of the valve actuator through the full range of travel) will also impair valve performance, similar to that described previously. In both cases, calibration must be conducted with the valve off line, depressurized, and per the manufacturer's recommendations.

If regulator sizing is questionable, or outside of the above design guidelines, it is recommended that the valve station arrangement (i.e., upstream and downstream piping) be audited. Line and block valve sizing can influence control valve performance by compromising (i.e., reducing) the design operating differential pressure across the valve. If the control valve selection dictates a valve size which is less than the calculated line size, the block valves must *not* be arbitrarily reduced to match the control valve selection without a pressure drop evaluation.

Direct vs Pneumatic

Most, if not all, existing and new oil systems utilize conventional direct acting pressure regulators or pneumatic spring diaphragm actuated control valves. The choice of either type is dependent upon the required sizing parameters for each application.

Direct acting valves have a spring opposed diaphragm which senses the pressure to be controlled through an opening in the diaphragm case. The direct acting valve is a simple design, extremely reliable, and very responsive to transient conditions. However, by the nature of its design, rangeability is limited due to "droop" or valve "offset." Valve droop (or offset) is essentially the difference in pressure at different flowrates due to the proportional relationship between the valve plug travel and the spring constant. To counter the effects of droop, designers usually limit valve plug travel and employ quick opening characteristic plugs.

Droop typically manifests itself when the unit is started or shutdown. For example, the PRV in the oil system shown in Figure 1 is designed to maintain 20 psig at the unit bearings in normal operation. However, when the unit is shutdown, it is noted that the pressure increases to 22 psig. At this shutdown condition, lube oil flowrate will decrease, thus allowing the pressure to increase due to the actuator spring movement. The difference in the operating and shutdown pressures is actually the droop. As such, the direct acting valve will require adjustment to neutralize the effects of droop at design operating conditions (off design conditions may require a temporary compromise in operating pressures). This phenomenon is typical of direct acting valves, and generally must be accounted for during system design and valve selection (valve manufacturers can provide the data necessary to calculate valve offset), and tolerated during operation. Other than changing valve springs or the entire valve assembly, attempts to reduce valve droop will yield limited results.

Once in service, some direct acting valves can experience instability due to valve sensitivity. "Tuning out" direct acting valve instability can be accomplished by installing an orifice in the signal line, or by using a directional flow control valve in the same signal line. The use of the flow control valve in the signal line, as shown in Figure 7, facilitates damping an unstable control valve, and yet can accommodate maximum response speed in one direction.

An unconventional method used to combat instability or valve noise is to reverse the flow direction of the valve. By changing the fluid flow path through the valve body, the valve stability or noise problem can be eliminated. However, in some cases, it may be necessary to verify the valve actuator sizing with the valve manufacturer prior to implementation.

Pneumatic control valves offer greater rangeability, due primarily to increased valve travel (average 3/4 in for pneumatic versus 1/4 in for self acting). Typically, pneumatic control valves in pressure control service are equipped with equal percentage char-

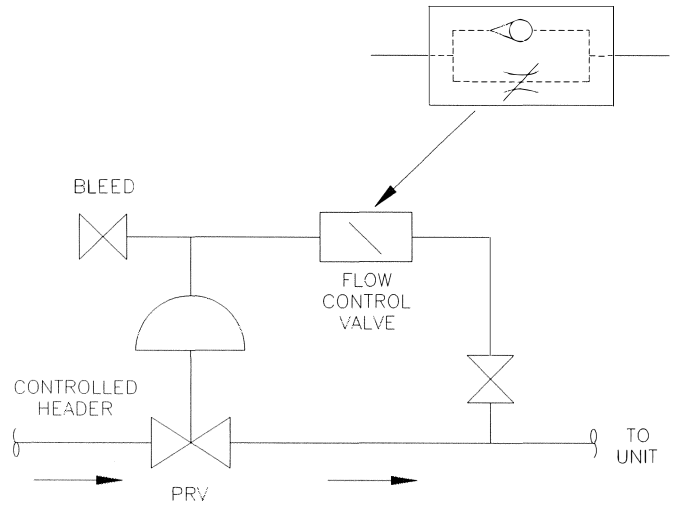

Figure 7. Hydraulic Dampener on Self Acting Regulator.

acteristic valve plugs, with the capability of operating reliably and stability between 10 and 90 percent of the design travel. However, for oil system applications, some considerations regarding the use of pneumatic spring opposed diaphragm actuated control valves are notable. A pneumatic control valve, with typical accessories for enhancing response, is shown in Figure 8.

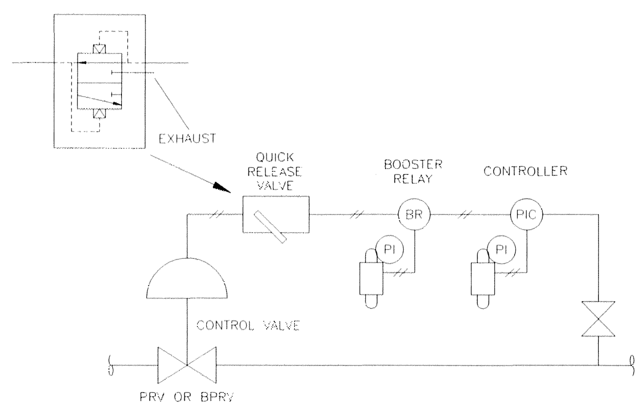

Figure 8. Pneumatic Control Valve with Accessories.

Pneumatic actuators should be selected to match the respective pneumatic controller output. Use of a pneumatic positioner to provide additional air pressure to meet the required actuator forces is satisfactory in other applications. However, for oil system pressure control service, which requires a fast response, positioners have proven to be a liability. The positioner, being an additional component, adds a lag into the loop that cannot be easily attenuated by the controller. In some cases, valve stability is degraded by a positioner, to the point that controller tuning cannot overcome the effects of the positioner. On the other hand, pneumatic volume boosters installed in the signal line (between the controller and actuator) can increase the speed of response. Normally, volume boosters are installed when pneumatic signal lines are in excess of 100 ft. However, the volume capacity of a pneumatic booster relay is generally greater than that of a pneumatic controller. Therefore, the controller need only work with the short signal line between the controller and booster relay, instead of a large diaphragm case on the actuator air head. Such a volume booster relay can be installed so as to increase the speed of response of a sluggish BPRV, DPRV, or PRV.

Pneumatic actuator diaphragm vent porting can also affect valve response. For example, a pneumatic BPRV must close quickly during a pump switchover transient, as discussed in the previous section on main pump to standby pump transients. To increase the actuator closing speed, the vent porting must be large. Additionally, a device referred to as a "quick exhaust valve" can be installed between the pneumatic pressure controller and valve actuator, as shown in Figure 8. When the controller output decreases rapidly, as does occur when a pump is shut down, the quick exhaust valve senses the pressure difference and exhausts the diaphragm of the BPRV actuator. The quick exhaust valve is simple, reliable, and easy to retrofit on existing oil system pneumatic control valve assemblies.

In all pneumatic valve instrumentation tubing must be smooth, with no kinks or restrictions of any kind. Tubing size should be 3/8 in OD. Smaller tubing sizes can cause restrictions, which can affect valve performance. Larger tubing sizes can increase response time by unnecessarily increasing control air volume. The 3/8 in OD is self supporting, and provides a satisfactory flow path for pneumatic control signals.

CONTROLLERS

New or replacement pneumatic controllers should include the following features or capabilities for the best overall oil system performance.

- Process variable indication.
- Bumpless auto/manual switching. This feature is useful at startup, and for troubleshooting.
- A wide range of proportional band adjustment, from five to 500 percent.
- Reset capability of 100 repeats per minute.
- Sensing element calibration twice that of the set point, for direct pressure sensing pneumatic controllers.
- The process scale calibrated to match the set point parameters.
- A high degree of accuracy and repeatability characteristics.

Normal control modes include proportional (gain) plus reset (integral). The rate (derivative) mode is considered unnecessary for oil system applications.

The controller mounting configuration is crucial for performance and system reliability. Although many controllers are mounted to actuator yokes for convenience, constant valve vibration can lead to premature wear, and affect controller response. The best location for mounting of a pneumatic controller is remote from the control valve, and mounted to a rigid fixture. The proximity of the controller to the control valve should be as close as practical so as to avoid long tubing runs.

Many of the pneumatic controllers that were provided on older oil systems have limited proportional band and/or reset. Therefore, tuning capabilities are limited, as well as overall valve response. Replacing such outdated pneumatic controllers is cost effective, and can greatly improve overall oil system performance and reliability.

Control loop sensitivity can be selectively modified by reviewing or changing the pressure sensing element in the pneumatic controller. For example, a BPRV which is controlling a header pressure of 200 psig can have a pressure sensing bourdon tube with a range of 0-600 psig, and a pneumatic output signal of 3-15 psig. The overall controller calibration is as follows:

Controller Input (psig)	*Controller Output (psig)*
0 (zero)	3
300	9
600	15

In the preceding example, every 50 psig change in input pressure results in a one psig change in controller output pressure. To make the loop more sensitive, the above pressure sensing bourdon tube can be replaced with one with a lower range, e.g., 0-300 psig. The new valve calibration would be as follows:

Controller Input (psig)	*Controller Output (psig)*
0 (zero)	3
150	9
300	15

Thus, every 25 psig change in input pressure now results in a one psig change in controller output pressure. As can be seen from the above examples, the procedure can be reversed so as to make a sensitive controller more stable. In general, the range of the pressure sensing element should be at least twice that of the control setpoint (i.e., a setpoint of 200 psig would require the selection on a sensing element with a range of at least 0-400 psig).

Generally, the majority of oil system valve controllers are pneumatic. However, there is a slow trend towards upgrading oil system control systems by replacing these pneumatic controllers with electronic controllers. Traditionally, the arguments against electronic controllers has been high associated costs, and suitability to local field mounting (one of the main reasons for maintaining local installation of oil system controllers is the ease with which the oil system can be commissioned and tuned). Until lately, electronic controllers did not meet electrical qualifications for hazardous areas without installation in a suitable enclosure. Some users have, therefore, started using the plant (or unit) distributive control system (DCS) to control oil pressure control valves. However, there are several manufacturers who are now marketing local field mountable electronic controllers, which meet area classifications for hazardous service.

Electronic controllers offer the benefits of versatility and hardware reliability. To be sure, the expanded capabilities of most electronic single loop controllers will be wasted on an oil system application. The replacement of pneumatic controllers with electronic controllers in oil system service is typically not made for the sake of change, but rather where a user wishes to incorporate the oil system control into a DCS. Additionally, installation of an electronic controller can cost approximately two to three times that of a comparable pneumatic controller. For this reason, retrofits of electronic controllers on existing systems, especially those that have historically been reliable, are few at this time, and are usually driven by a user preference for such equipment. It is recommended that consideration for upgrading to electronic controllers on an oil system be evaluated on an individual basis.

RELIEF VALVES

Safety relief valves are required on oil systems at the discharge of positive displacement oil pumps, on certain vessels as required by the code or design, at valve stations with low delivery pressures (to protect the serviced equipment), and on any motive gas supply lines (e.g., accumulator motive gas supplies).

Relief valves for oil system service can be classified as either hydraulic or ASME type relief valves. Hydraulic relief valves can generally be described as spring opposed piston/cylinder assemblies, which modulate flow, with no tight shutoff characteristics. Conversely, the ASME type relief valves conform to code, have spring opposed disc assemblies, do not modulate flow, and will shut off tight.

The pump relief valve description in API 614 second edition is that of a hydraulic type relief valve. The hydraulic type valves are generally a standard construction, which do not follow a specific code. Acceptance of the hydraulic valve has been due to favorable user experiences, and are generally utilized in low pressure applications (less than 1100 psig). The constant leakage feature of such valves minimize oil system instability due to chattering, and have been used to resolve oil system stability problems, by assuring a smooth lift and reseat. Note, however, the constant oil leakage through the valve (which can be significant, with some valves passing as much as 12 gpm during normal service) with respect to pump sizing must be considered on any new system or retrofit.

Conversely, the ASME valves are indeed built to a recognized standard. The ASME type safety relief valve is used when local codes do not permit a hydraulic relief valve (no matter how low the operating pressure level), where operating pressures exceed the ratings for the hydraulic valves, and for all gas services.

Pump discharge relief valves must operate smoothly and without chatter to avoid component or piping damage. The set pressure is to be 110 percent of the highest expected pump discharge pressure or a minimum of 15 psig. Normal relief valve sizing should be such that overpressure (the pressure at which full design flow is passing through the relief valve) is 110 percent of set pressure. Reset pressure (the pressure at which the valve closes) should be no less than 90 percent of set pressure.

Improper setting of the pump relief valves can lead to unwanted bypassing, and affect performance of the BPRV. Setting of pump relief valves should be carried out on the oil system, with the oil at normal operating temperature. The relief valve is manually lifted by utilizing the pump discharge block valve and/or RV bypass valve. As the respective valve is closed, the pump discharge pressure should be monitored at the discharge pressure indicator.

The set pressure of ASME type relief valves is audibly discernible when reached. However, the setpoint of the hydraulic valve can be very difficult to confirm, due to the smooth lifting characteristics of this type of valve. Therefore, it is more convenient and reliable to set the hydraulic relief valve at the overpressure setting (recall that the overpressure setting will be a function of pump flow and the set pressure).

Corrosion of the relief valve internals can be a problem, especially in hot and humid climates, or where water contamination of the lube oil is excessive. It is, therefore, advantageous to keep the relief wetted with oil at all times. Hydraulic valves can be mounted in any orientation, and can, therefore, be mounted in a flooded attitude (below the normal operating level of the reservoir), as shown in Figure 5. ASME type relief valves however must be mounted in a vertical stem position. As such, the valve can be mounted below the normal operating level of the reservoir (if the oil system layout permits), or purged via an orificed constant oil bleed from the running oil pump.

Air in relief valve lines will also cause poor performance during automatic startup of the AOP. Such poor performance manifests itself as a sharp drop in pressure during AOP acceleration, and can be accompanied by loud banging or instability of the oil system. Such an event will almost always cause a low oil pressure trip of the unit. Air in the relief valve line can be eliminated by utilizing either of the above recommendations. The minor disadvantage of flooding the relief valve(s) by locating them below the normal oil reservoir level is the necessity of draining the oil reservoir to service the RV (unless of course a block valve is installed upstream of the RV, which might not be permitted by code).

FILTERS

The integrity of the oil filters, and especially the elements, is an important consideration in assuring reliable operation of the unit. Neglected or improperly maintained oil filtration systems will result in worn bearings and seals, higher seal oil leakage rates, shortened operating intervals, and major rotor repairs.

Types

Numerous filter element types are available today which can be retrofit into existing filter housings. Substitutions for the OEM filter elements must be equal to, or better than, the original elements.

- *Pleated Paper*: Paper (cellulose) filter elements offer the advantage of having very high flowrate capabilities, due to the extended surface area of the pleated paper design. However, paper elements are sensitive to water, and will swell if exposed to water, unless treated with a resin.

- *Pleated Fiberglass/Polyester*: These elements are a crossover application from the hydraulic industry. Fiberglass and polyester elements are not sensitive to water, and offer high flow capability without media fiber migration.

- *Wound*: Wound filter elements (typically cotton material) offer reliable service. However, cotton fibers are sensitive to water, and will swell when exposed to water in the oil, often to the point that the filter collapses. Replacement aftermarket cotton elements are readily available, however, the quality of many such elements is very poor.

- *Phenolic Resin*: The phenolic resin element is not sensitive to water, and is, therefore, used in many cases where cotton elements swell. However, the phenolic resin element can be sensitive to temperature, and can rupture when thermally shocked.

- *Cleanable*: Cleanable elements offer the ability of being manually cleaned when fouled, and then returned to service. Cleanable elements are typically metallic, and can retrofit into existing filter housings. The disadvantage to cleanable elements is that improper handling during cleaning, or damage, will void the filtration rating.

- *Others*: Other types of filter elements include nylon, rayon, and polypropylene, to name a few. Such elements are designed for other industries (e.g., food, drugs, and paint), and see limited use in turbomachinery applications.

Ratings

Filtration rating is the single most important criteria to be used when evaluating the selection of a filter element, and is specified in MICRONS. The filtration standard adopted for critical turbomachinery is 10 microns (0.00039 in). This rating has been selected by the turbomachinery manufacturers based upon minimum oil film thickness of journal and thrust bearings. Use of any filter element rated higher than that specified by the turbomachinery manufacturer will compromise the integrity of the unit, even though the element may be less expensive or decrease filter housing pressure drop.

Cartridge efficiency is a measure of how well the element will remove dirt particles greater than the particle size stated, and is expressed as a percent. Generally, the higher the efficiency, the better the element. There are three terms used by the filter manufacturers in specifying a cartridge efficiency, which are described as follows:

- *Absolute*: Absolute efficiency is the amount of dirt particles at the filter's rating that will be captured and held during one pass. Typical values are 98 or 99 percent.

- *Nominal*: Nominal efficiency is the amount of dirt particles at the filter's rating that will be captured and held during continuous duty, or through a closed loop. Such a rating is more representative of the operation of an oil system. Typical values are 94 or 95 percent.

- *Beta Ratio*: Beta ratio is the efficiency of element at a specific particle size, but is expressed in whole numbers, based upon a logarithmic performance curve. For example. a Beta ratio $B_{10} = 50$, indicates that an element is 98 percent efficient in removing particles larger than 10 micron. Beta ratios are calculated from actual test data of particle counts upstream and downstream of the element, taken during the test.

Whereas there is not a turbomachinery industry standard in place for testing and evaluating filter elements, there are filtration industry standards for such tests. However, the exact test details and data expressed from such tests may vary between filter manufacturers. A filter element comparison should be based upon Beta ratio, flowrate, pressure drop, dirt holding capacity, materials, overall quality, and cost.

Sizing

Correct filter sizing will assure proper filtration and prevent unnecessary collapsing of the filter elements. This is the responsibility of the oil system manufacturer. However, sizing is ultimately a very important consideration to the user, since different filter elements can have significantly different recommended maximum flowrates. If this value is exceeded, the possibilities for problems increase.

Collapsing of the filter elements while the unit is on line can result in a spurious trip, due to a rapid drop in oil pressure downstream of the filter housings. Under certain conditions, bearing and seal damage could be experienced. Therefore, it is imperative that the chances of a filter collapse be minimized or eliminated.

All reputable filter manufacturers will provide collapse ratings of their filter elements, in addition to the efficiency ratings previously discussed. This rating should be at least 70 to 80 psid for turbomachinery service. The cartridge separator plate should be rated for a collapse rating greater than the filter element. Numerous factors influence the collapse potential of a filter element, including:

- *Flow*: Verify that the actual oil flow (per element) does not exceed the manufacturer's rating. For example, a cotton element might be rated at 4.0 gpm per cartridge, whereas a paper element may be rated at 10 gpm per cartridge. Obviously problems will arise if cotton elements are installed in the filter housing that was designed for paper elements. Be especially cautious on systems where the BPRV was not sized for two pump capacity.

- *Viscosity*: The pressure drop through an element with cold oil will be greater than that pressure drop at design operating temperature. If the oil is too cold, the filter can collapse.

- *Water*: As mentioned previously, water can cause certain types of filter media to swell, which can lead to collapse.

- *Air*: The presence of air in the filter housing can cause filter collapse, due to liquid slugging.

- *Pressure Surges*: Surging of the oil system, due to either instability or the starting of a second pump, can cause a momentary pressure rise and lead to element collapse.

- *Direction of Flow*: In almost all cases, direction of flow is outside of the element to inside. If flow is reversed for any reason, the element(s) will essentially explode at a pressure drop that is lower than the collapse rating. In most cases, this will send filter debris into the unit bearing, seal, and control housings.

The best method of controlling filter collapse is elimination of the causes. If water contamination of the oil is a problem, use only that filter media that is not sensitive to water. Always heat the oil charge to 70°F minimum before starting the oil pumps. If this is not practical, open the pump or BPRV bypasses, and recirculate the oil before slowly valving the oil into the system and through the filters. Do not force cold oil through the filter housings if the differential pressure approaches the collapse rating; throttle the oil until warm enough to continue safely. Always bleed the filters and coolers during startup, and prior to a filter switchover.

Maintenance

Proper filter maintenance will assure uninterrupted periods of reliable operation. Cleaning of the filter housing during a filter change is of major importance. The filter housing is separated into two distinct chambers by a diaphragm, as shown in Figure 9; the "dirty sump" above the diaphragm, and the "clean sump" below the diaphragm. The dirty sump must be wiped down with cleaner and lint free rags during an element change. However, the clean sump is very often overlooked, which can lead to bearing and seal wear problems. When the old filters are lifted out of the housing for disposal, the dirty oil that drips from the elements drops into the clean sump. Unless this trash is removed, the contaminated oil will be sent directly to the unit when the filter is placed on line. It is therefore imperative that the clean sump be thoroughly washed with a wand and solvent during maintenance. This is especially critical after a unit flush or oil pump failure.

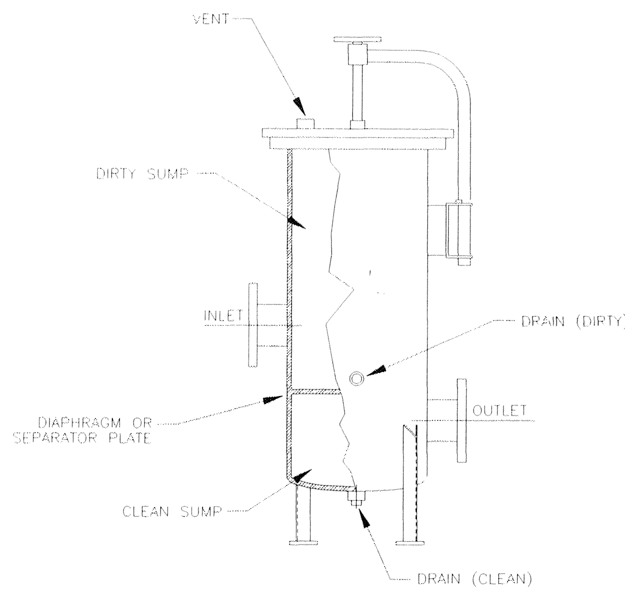

Figure 9. Filter Clean and Dirty Sumps.

Filter element selection is at the discretion of the user, based upon plant stocking policy, OEM requirements, operating experiences, and the above discussions on this subject. It is quite normal today for paper elements to replace the original cotton elements where collapsing and high filter differential pressure has been a problem. The availability of multiple length cartridges facilitate maintenance and reduce the risk of internal bypassing at the stacked joints (internal bypassing can contribute to damaged bearings and seals). The original hardware, which includes the guide rods, seal plates, spacers, and springs, must be reused, or changed if damaged (note that some of the multiple length elements eliminate the need for a guide rod and spacers).

As mentioned above, internal bypassing can result in damage to unit bearings and seals. To prevent this bypass from occurring, it is recommended that the single length elements be changed to the correct respective multiple length element (this minimizes handling, and decreases filter turnaround time as well). When this is not practical, spacers should be used, along with Buna N or Viton seals. Cork seals should be avoided because of limited shelf life.

Filter replacement should take place when filter differential pressure has risen 15 psid above the original clean value. This is a conservative figure, which is intended to maintain a comfortable margin between reliable operation and the collapse rating of the elements. Note that depth type elements (e.g., cotton and nylon) suffer from a phenomenon termed "cartridge erosion," where oil velocity enlarges or erodes the filter passages over time, which effectively invalidates the filtration rating of the element. Such a phenomenon will allow larger dirt particles to pass to the bearings and seals, and can cause extensive damage to the rotor. Since the element flow path is essentially increased, pressure drop will remain the same, and can therefore go undetected. To prevent unit damage from cartridge erosion, it is conservatively recommended to change filter elements every *six months*, or after the pressure drop from clean increases by *15 psid* (unless recommended otherwise by the element manufacturer).

INTERCONNECTING PIPING

Interconnecting piping is probably one of the most overlooked aspects of oil system design and installation. However, this area deserves critical attention, since improperly designed piping can lead to oil system performance deficiencies, and is almost impossible to change after the unit has been installed and commissioned. For this reason, interconnecting piping should be discussed between the purchaser, engineering contractor, and vendor during the design phases of the project.

However, what should be investigated on the unit which has been installed and commissioned for years, and what can be done if a problem is identified?

Layout

The physical arrangement of the rotating machinery, oil system, and support systems play a key role in determining the interconnecting piping configuration. Items to consider include:

- *Arrangement*: Will the rotating string be mounted on a mezzanine or at grade? If grade mounted, attention must be given to adequate elevation differences to allow for adequate fall of the drain piping, along with correct location of intermediate support equipment (e.g., sour seal oil drainers and degassing tanks). If mezzanine mounted, oil system design must account for the interconnecting static and dynamic pressure losses, including those losses in long supply headers on large multibody strings of equipment. Other items such as personnel access, overhead equipment (e.g., vessels, cranes, lights, sprinklers), and laydown areas or openings must be properly considered.

- *Piping Slope*: As mentioned above, proper slope is required to assure positive drainage of the oil from the unit back to the oil reservoir. Improper drain line slopes will create traps, cause oil leaks at bearing housings and coupling guards, and could potentially cause flooding of bearing housings. Generally, all drain piping should be given a slope of 1/2 in/ft of piping run. The exception is pressurized drains (where process pressure can drive drain oil uphill to an atmospheric vessel, as in the case of sour seal oil drainers into a degassing vessel). Inadequate or uphill drain slope from sour seal oil drainers have been known to fill compressor casings and empty reservoirs while the compression string is idle for extended periods.

- *Instrument Location*: Alarm and trip switches and seal oil differential pressure transmitters should be located at the unit if at all possible. If such devices are located with the oil system at grade, compensation for pressure due to static and dynamic losses must be accounted for in the calibration of the instruments.

Obviously, the relative elevations of existing installations cannot be changed easily or economically. However, it is a simple matter to relocate pressure switches to the unit during a scheduled shutdown if problems lie in this area. Items such as piping slope must be dealt with and corrected as the existing configuration permits.

Piping

Proper design and attention to interconnecting piping details can make the difference between a machine that is reliable and one that is an operator's headache. It is very important to be aware of the allowable interconnecting piping losses, and then arranging the piping within these guidelines. Interconnecting piping pressure drop can be divided into two categories:

- *Static*: Static pressure loss is the pressure difference between the oil system and the unit due simply to the oil head. Obviously, the oil pressure at the outlet of the pressure regulator at the oil console will be greater than that at the unit bearings, seals, or controls. Static head is fixed by the relative geometry of the equipment, and cannot be changed without major changes in the unit layout. Generally, 32 in oil column is equivalent to one psi.

- *Dynamic*: Dynamic pressure loss is the oil pressure loss that will occur as the oil flows from the oil console pressure regulator up to the unit. This pressure loss is a function of the length of oil piping, pipe size (diameter and schedule), and number of piping fittings. Generally, oil system designers allow for a total dynamic loss of 10 psi.

The respective oil system connection size should always be considered the minimum required piping size to be used for construction of interconnecting piping. After the line has been laid out by a piping designer, but before construction begins, the pressure drop at design and maximum flowrates should be calculated. If dynamic loss is predicted to be greater than that allowed by the oil system designer, then the next larger pipe size should be used, or the pipe rerouted and the total length shortened.

Piping fittings also contribute to the overall dynamic pressure losses in the interconnecting piping. Such losses are expressed in equivalent ft (K value); a fitting with a lower K value will have a lower pressure loss, where a higher K value represents a higher loss, or greater resistance to flow.

Among the biggest contributors to interconnecting piping problems are socket weld elbows and tees. Such fittings are very popular, due to their ease of fitting and welding. However, these fittings also have the greatest resistance to flow, or K value. A socket weld fitting can have a K value that is two or three times greater than that of a long radius butt welded fitting, due essentially to the tortuous path inherent in this design. The difference in cross sections between a socket weld and butt weld long radius elbow is illustrated in Figure 10. Excessive use of such fittings could mean reduced delivery pressure at the unit, and equally important, increased response time at the unit trip switches during an upset.

If spurious trips or other mechanical problems are found to be related to inferior interconnecting piping design, obviously the best solution is to replace the piping with an improved layout. However, timing and costs may make this modification impractical (remember that breaching oil delivery piping in any manner requires an oil flush prior to startup). If the problem is marginal, the PRV can compensate automatically by opening more. However, as dynamic line losses increase due to piping inadequacies, control valve modifications or pressure setpoint adjustments may be necessary. Changes in switch setpoints must be reviewed by the turbomachinery supplier.

Overhead rundown vessels, which rely simply upon the difference in height between the vessel and the centerline of the unit, are very sensitive to interconnecting piping losses. Excessive pressure losses can render an OHRDT or seal oil tank useless. Since such vessels rely solely upon gravity as the motive flowing force, the options for compensating for poor piping design are limited: either change the piping, or raise the vessel. In almost all cases, modification of the piping is the easier modification to make. Generally, overhead seal oil rundown piping pressure drop should not exceed

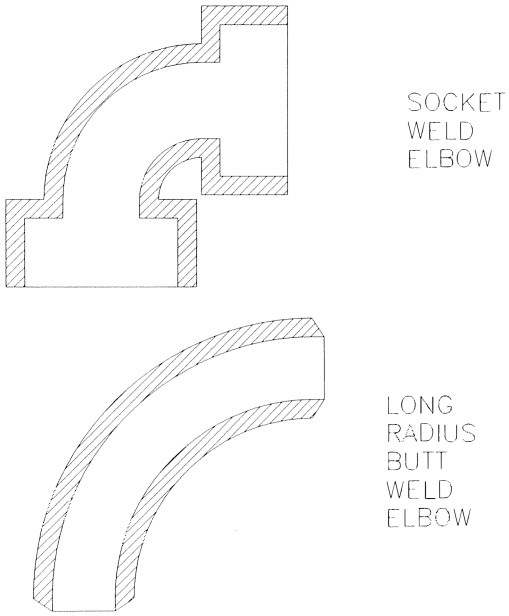

Figure 10. Socket Weld and Butt Weld Elbows.

one psi at the design oil flowrate; OHRDT piping pressure drop should not exceed two psi at the design oil flowrate.

In summary, minimize the use of piping fittings, and maximize the pipeline size. It is never a problem to make the pipe size larger than the oil system supply connection; it is always a problem if it is smaller. Never attempt to make the interconnecting piping size smaller than the connection on the oil system (recall that the oil system designer has selected that connection for a certain pressure drop at the design oil flowrate).

Piping material is another important consideration in the overall design of the oil system, and especially the interconnecting piping. Stainless steel oil supply and drain piping will provide the greatest reliability and the least problems, as will be quickly appreciated on a unit that has a large steam turbine driver. When a steam turbine is a part of the rotating string, there is always the potential that excessive steam leakage from the packing cases may cause water contamination of the lube oil. This water will manifest itself as corrosion on all bare internal carbon steel surfaces which happen to be wetted by the oil. This naturally includes the oil supply and drain piping. Corrosion in the supply piping compromises the integrity of the bearings and seals of a unit; corrosion of the drain piping compromises the integrity of the oil pumps. The extra cost of stainless steel oil piping at the beginning of a project will pay for itself many times over throughout the operating life of a plant.

Many oil systems designed and purchased today are almost 100 percent stainless steel construction, some units down to the control valves and block valves. As a minimum, all piping downstream of the oil filters should be stainless steel. If an existing unit with carbon steel oil piping has a history of bearing, seal, and pump wear, serious consideration should be given to changing the carbon steel delivery and drain piping to stainless steel. Additionally, never consider using carbon steel interconnecting piping on a unit that was purchased and built with stainless steel piping.

Cleaning

Cleaning of all oil piping, and in particular interconnecting piping, is critical in assuring a history of trouble free operation. All new or repaired interconnecting piping must be properly prepared before final installation. This preparation includes dressing all welds, and proper cleaning of the piping.

All interconnecting oil piping should be arranged such that *all* welds can be visually inspected and cleaned. Socket weld fittings should not be used, as they tend to accumulate trash and debris, which will ultimately extend the length of the oil flush and reduce unit reliability. It is also good practice to install blind flanges at the ends of long horizontal runs of pipe to facilitate cleaning. Main piping lines should use flanged joints only; use of threaded connections anywhere except at instrumentation connections is unacceptable.

All piping should be cleaned to remove dirt, mill scale, and weld scale prior to installation and flushing. Cleaning at this time will reduce flushing time in the later stages of the project or overhaul. Generally, new carbon steel piping should be both alkali cleaned (to remove oils, paints, and grease) and acid cleaned (to remove mild rust and scale), and then neutralized and preserved. This is a tedious and dangerous procedure, and is best left to firms that specialize in this service. Stainless steel piping on the other hand can be cleaned using vapor degreasers, steam, and/or an alkali wash. In some cases, piping can be grit blasted. This should preferably occur prior to fitting and welding, to assure that no grit is inadvertently left in the lines. After cleaning, all piping should be immediately tested and installed (preferred), or capped and sealed at the ends until ready for assembly.

After the piping is complete, hydrostatic testing should be conducted with oil, not with water. Water can and will be trapped in low spots and voids, and will cause corrosion in carbon steel pipe, leave mineral and chemical deposits in the pipe, and contribute to flushing problems later.

Flushing of oil piping, whether new or existing, is a universal cleaning problem. An unorganized flush can persist for weeks with no visible progress, and can frustrate users, contractors, and vendors alike. The key to a successful field flush is organization.

First, temporary jumpers must be fabricated to bypass the unit bearings and seals. These jumpers must be clean, and should be hard piping, not hose. All obvious blind spots and potential pockets or "dead spots" must be flagged and piped to assure flow. Such spots will accumulate dirt during a flush and prolong the procedure indefinitely. Flushing screens made of 100 mesh stainless steel should be located at the terminus of the oil delivery lines as targets to allow constant evaluation of the cleaning process. Note these screens are not intended to be used as filters (a common misconception). It is also common practice to use milkpads on the 100 mesh screen to aid in the detection of dirt and debris.

Thoroughly clean the filter housings, including the clean sump. Install new filter elements, which are the same recommended filtration rating as that of the equipment manufacturer. Never flush an oil system without oil filter elements installed.

Flush the oil system using all available oil pumps to maximize oil velocity through the piping system. Selectively flush individual delivery lines by using the control valve isolation valves. This procedure cannot be overemphasized; high oil flow and high oil velocities clean systems. Oil filter differential pressure will increase due to the abnormally high oil flowrates; this is acceptable as long as the differential pressure is kept below that the collapse rating of the elements. Utilize heat and vibration to loosen internal debris and accelerate the flush.

Some non-API oil systems, in addition to older oil systems, can have filter assemblies with high differential pressure bypasses. Such filters must be closely monitored during the flushing procedure to prevent bypassing of dirty oil (note that the use of automatic filter bypasses or relief valves is not recommended for turbomachinery service).

Aftermarket cleaning services equipped to perform field flushes in a minimum time are available. These firms utilize very high flow oil pumps mounted on portable trailers, with an auxiliary reservoir that can contain special flushing oils. Such services can significantly reduce flushing time, not to mention the wear that is imposed on the unit oil pumps during this operation (oil flushes can and will damage positive displacement oil pumps if proper precautions are not taken). Note that flushing oils which are not the same grade as the operating oil must be drained from the system prior to final reservoir charging and startup.

Such cleaning services are also equipped to flush through vessels such as overhead seal oil tanks, rundown tanks, and accumulators. These vessels are not generally easily cleaned using only the system oil pumps, due to the high flowrates typically required for this duty.

After the piping flush is complete, the bearing housings can be flushed if desired, again using target screens. However, it is not recommended to flush through seal housings, servomotors, controls, or gearboxes.

Insulation and Heat Tracing

Insulation and heat tracing are required on oil lines and vessels that do not see a continuous flow of oil. This criteria pertains specifically to overhead seal oil and lube rundown tanks that are exposed to ambient conditions, although it can apply to the entire oil system, depending on the area of installation.

The sour seal oil drain lines on refrigeration compressors can also require insulation, heat tracing, and in some cases, internal endwall heaters. This is due to the very cold process gas that can be exposed to these areas in normal operation. In some cases, the sour seal oil can be cooled to a grease, which will prevent flow of the oil to the drainer, and result in oil migration to the process.

Insulation of exposed rundown vessels will be required in geographical areas where the ambient temperature can drop below 32°F for extended periods of time. The primary purpose of insulation is to keep the oil at an elevated temperature such that it will flow readily during a rundown. ISO 32 oil at 32°F has a viscosity of 1400 SSU; when oil is this viscous, the flowing line losses are significant, resulting in greatly reduced oil pressure and flow at the unit during rundown. In such a scenario, viscous oil will render the rundown system ineffective.

Climatic areas harsher than described above will require insulation and heat tracing of the vessel(s). Heat tracing assures that the oil will be maintained at a constant temperature approaching the unit design operating temperature. This then guarantees satisfactory operation at design rundown conditions. It is acknowledged some installations resolve this problem by enclosing the entire unit in a climate controlled enclosure.

If indeed a particular application falls within one of the above categories, the rundown performance of the respective vessel should be tested while the unit is down, and lube oil supply pressure and/or seal oil differential pressure be observed. If the supply pressure drops significantly below the normal design values during the test, then insulation and heat tracing should be considered.

It should be noted that heavy hydrocarbons in contact with the seal oil in an OHST can condense, even at ambient temperatures above 70°F (e.g., in the case of natural gas reinjection compressors, or in flowthrough overhead seal tank designs). As such, insulation and heat tracing could be required for such installations, despite the favorable ambient conditions. Hydrocarbon contamination of the seal oil will lower the viscosity and flash point of the oil, resulting in a very dangerous situation, both mechanically and operationally.

MISCELLANEOUS

Although not discussed in detail, several other areas do warrant cursory review.

- *Machinery Upgrades*: When rerating or modifying existing machinery, the oil system should be audited for the new service. Such an audit reviews the oil system for undercapacity, overcapacity, and component worthiness.

- *Liquid Level Transmitters*: The calibration of liquid level transmitters installed in pressurized vessels can shift at higher operating pressures, due to changes in gas density (this is especially significant if the gas MW is high). To prevent such shifts in calibration, and the obvious control problems that can result, all liquid level transmitters should be calibrated at design operating pressure, or at least checked at this condition.

- *Addition of Rundown Vessels*: Always verify that reservoir sizing has been considered when adding any type of rundown vessel after the original installation.

- *Transfer Barriers*: Monitor transfer barriers (used with overhead seal oil tanks to isolate the seal oil from the process) for accumulation of condensate, which can create a false level signal and lead to a seal failure, due to low seal oil pressure. Verify that the installation of transfer barriers allows easy access for maintenance, and reliable precharging.

- *Oil Coolers*: *Never* use an oil cooler as a heater by injecting live steam directly into the cooler tubes; inject the steam into the cooling water flow upstream of the cooler. Clean the tube bundles manually after a pump failure or a major equipment wreck.

- *Transfer Valves*: Never turn a tapered plug transfer valve without first lifting the plug out of the seat (if equipped with a lifting jack). If valve leakage is excessive, consider replacing the valve with a straight plug valve (some retrofits are now available) or two three-way ball valves (may require piping modifications).

CONCLUSION

Oil system designs have improved greatly over the years. The operation of many existing oil systems can be improved by design audits and hardware upgrades, especially those units designed before or without the benefits of API 614.

Oil system operation must be challenged, either by testing or by off design operation, to confirm suitability to service. Unit reliability and availability can be significantly increased if the oil system is attended, either in the form of testing, maintenance, or upgrades on a regular basis. Pump switchover performance, which in itself is probably the single largest contributor to spurious trips, can be improved with minimal effort, to the point that such trips are minimized. Other items, such as controls, piping modifications, and filter selection, can prove to be cost effective upgrades on older units, and help increase reliability significantly.

NOMENCLATURE

AOP	Auxiliary Oil Pump, or Standby Oil Pump
BPRV	Back Pressure Regulating Valve
DBPRV	Differential Back Pressure Regulating Valve
DCS	Distributive Control System
DPRV	Differential Pressure Reducing Valve
MCC	Motor Control Center
MOP	Main Oil Pump
OHRDT	Overhead Rundown Tank
OHST	Overhead Seal Oil Tank
PRV	Pressure Reducing Valve
RV	Relief Valve