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MODERN LUBRICATING OIL SYSTEMS

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ABSTRACT

Lubricating oil systems are vital to operation, long-term reliability, and availability of turbomachinery trains (or strings) which are strategic and expensive components of industrial plants. Sometimes, the entities that specify machinery strings for their projects do not completely specify the required features of oil systems. For example, API 614 datasheets that contain only minimum information, such as the type of lube oil pumps and their drivers, the type of coolers, and the required inspection and testing, creates an inadequate basis

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for oil system design, and time is wasted in obtaining missing or incomplete details. In some situations, the purchasers change or add scope much later into the design, which can negatively impact cost and delivery schedules. Design deficiencies in lubricating oil system results in underperformance and affects reliability, and operability not only of the oil system but also the served equipment string or train.

The scope and configuration of the lubricating oil system for a stated application should be jointly evaluated by the vendor having the unit responsibility, the oil system designer, and the specifying machinery engineer. All components of the oil system should be discussed, and the design basis finalized by these parties as much as practically possible before awarding the purchase order.

INTRODUCTION

This tutorial discusses the selection and sizing considerations for components and elements that constitute the pressurized lubricating mineral oil system for a modern turbomachinery string. The design principles explained in this text are commonly applied to oil systems that serve different types of driven and driving equipment. Hydraulic oil systems for gas turbines are not covered in this text. Note that the design aspects for systems that use synthetic oils are generally completely compatible with the design aspects considered for mineral oil systems with regard to specific gravity, viscosity characteristics, acidity, anti-foaming, elastomer compatibility, etc. The occurrence of situations utilizing synthetic oils with special considerations impacting the system designs is insignificant and not discussed in the following sections.

Auxiliary components and accessories, such as oil accumulators, overhead rundown tanks (atmospheric and pressurized), oil mist eliminators and oil conditioners are elaborated in the following sections. It also presents a sample Hazard and Operability Study (HAZOP) of API 614 oil system with typical causes, consequences, severity, and barriers.

INFORMATION REQUIRED FROM PURCHASER

Lube oil system manufacturers need application-specific information from the purchasers to configure and design the oil system. In most situations, the purchaser is the vendor who is assigned the unit responsibility for the compressor-driver string or train. The following minimum inputs are required to quote an engineered oil system:

- General description of the driven equipment including number of casings/bearings, hydraulic controllers, etc. and required pressure levels - minimum/maximum/normal for each.
- The type of oil, the total and the component flow rates and the pressure level(s). The two most commonly used mineral oils are ISO VG 32 and 46. The important properties for design are their viscosities through the range of start-up and expected operating temperatures, specific gravity or density, specific heat, pour point and flash point.
- Heat load.
- Site elevation and ambient conditions (maximum and minimum) and where the oil system will be positioned: indoors (heated or unheated) or outdoors (or in the open, such as on the steel module or under the roof) or offshore installations where special requirements might apply including floating structure movement and marinization requirements.
- Utility conditions and the cooling method (water or air).
- Electrical area classification, Nominal voltage level(s) available and frequency, and reduced voltage start requirement, if any.
- Component features and options.
- Layout and maintenance requirements.
- Any constraints on transportation to the job site (for example, limitation on height might dictate the orientation of a tall reservoir to be horizontal).
- Elevation and location of the served equipment with respect to the oil system. The flow frictional losses of the interconnecting oil supply piping and elevation head difference of the console to the equipment must be added to the required oil console outlet pressures. Note that in some applications, small elevation differences between the console and equipment can require low profile reservoirs to facilitate sloped oil return lines entry above maximum operating levels.
- Accessory equipment to be supplied with the base oil system. For example, oil mist eliminator and oil purifier.
- Commissioning and flushing access, for example, jumpers around bearings, fill and high vent points, etc.
- Special requirements from the bearing manufacturer if any.
- End user specifications.

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Figure 1 shows API 614 based oil system for a centrifugal compressor string.

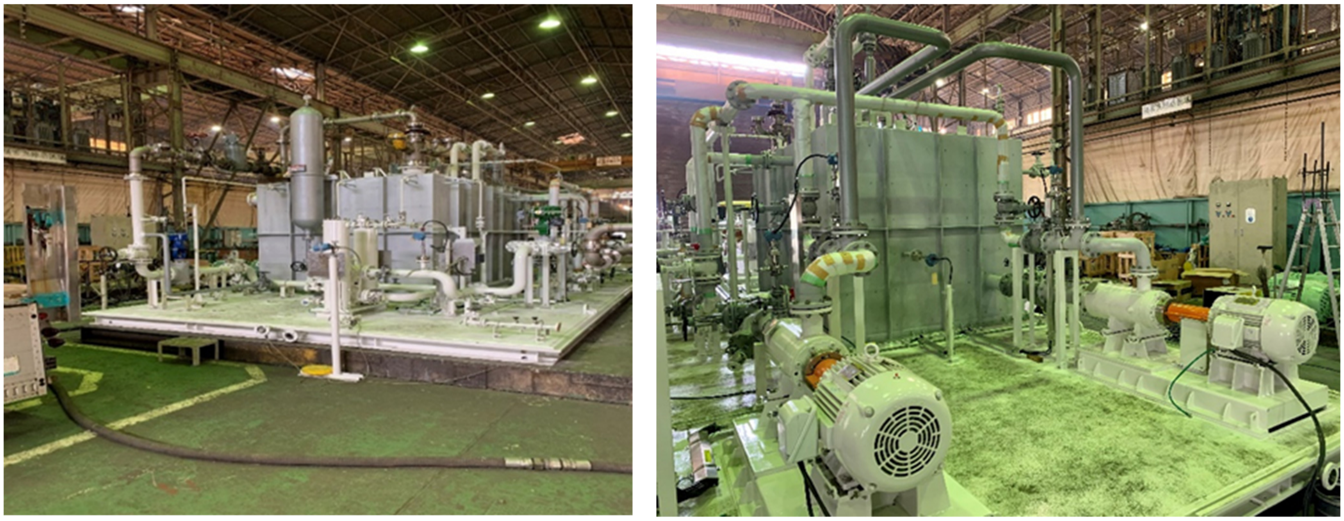

Figure 1: Lube Oil System for a Centrifugal Compressor String (Image source: Yoshimine, Japan)

TYPICAL DESIGN PROCESS

Using the Purchaser provided data, the lube oil system designer sizes the reservoir to conform with the API 614 specified oil capacities, levels, and a minimum-required free surface of oil 60 cm^2 for each litre per minute (0.25 square foot per gallon per minute) of normal flow. The plot space available at the location of installation, the layout constraints, any transportation limitations, need for internal and external bracings, and the access and maintenance requirements for the reservoir's top-mounted components are considered in the selection of the cylindrical, square, or rectangular shape of the reservoir. The orientation of the cylindrical reservoir is dictated by the space available over and around the lube oil console.

Purchasers usually specify their choice of oil pumps: either positive displacement rotary screw type; or centrifugal. If the purchaser has not stipulated the type of pump, the oil system designer selects pumps after their capacity and the required maximum discharge pressure are finalized.

The purchasers are also required to specify the type of heaters (electric or steam) for the oil reservoir. Sizing of oil reservoir heaters must consider total reservoir size and geometry, minimum ambient temperature and whether or not the insulation is being installed. API 614 minimum default sizing requirement is to be able to heat the reservoir's full-charge capacity of oil from minimum ambient temperature to minimum equipment startup temperature in a 12-hour period, based on an uninsulated reservoir with a 16 kilometers per hour (10 miles per hour) prevailing wind speed. Purchasers can specify shorter heat-up times or additional sizing safety factors.

Pump discharge pressure (differential pressure) is determined from the highest required pressure at the served equipment for bearing lube oil, hydraulic control oil actuators for steam governor and extraction valves, and inlet guide vane (IGV) positioners on dynamic compressors. System pressure losses in oil coolers, dirty oil filters, piping, fittings, valves, control valves, interconnect, and elevation difference must be included.

Additionally, for positive displacement pumps, the pump discharge (pump differential) must be suitable for the discharge relief valve (or pressure limiting valve) setpoint plus accumulation. The setpoint is established at margin above the maximum required discharge pressure to prevent opening during the pressure rise when starting the standby pump for the two-pump run.

The sizing process including determination of the rated power is identical for the steam turbine and electric motor drivers. However, sizing of drivers is different for positive displacement and centrifugal type oil pumps, discussed more later.

After these steps, oil filters, oil coolers, transfer valves, relief valves, and oil-pressure regulating valves (back-pressure and pressure-reducing valves) are sized and the results cross-checked to confirm the capacity and the required discharge pressure of the oil pumps.

The required oil system design pressure and the design temperature govern the sizes and schedules of piping and fittings. However, they are finalized only after complete evaluation of other parameters, such as the calculated maximum flow, the optimum flow velocities in supply and return lines, pressure drops and checking for no more than half full, if flowing at normal drain operating conditions at the maximum flow requirement for gravity-flow drain lines, as specified in API 614. If the purchaser has no preference, the piping materials and schedules are selected based on the minimum requirements laid out in API 614 for lubricating and control oil services.

Rundown tank and lube oil accumulator (lube oil and control oil) are sized and if included in the scope, mist eliminator, and oil

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conditioner are selected. API 614 provides typical piping and instrumentation diagrams and module options in each sub-system. The purchaser is required to specify certain details and features to allow the oil system manufacturer to completely configure and customize their lube oil system. The location, the quantity and the type of flow, pressure, temperature, and level instrumentation becomes an inherent part of this scope. Component design details are further discussed in the following paragraphs.

LUBE OIL PUMPS AND DRIVERS

Pumps

Lube and control systems can utilize either positive displacement or centrifugal pumps. The use of positive displacement pumps requires added components of discharge pressure limiting valves (PLVs) and a back-pressure or spill-back regulating control valve to divert the excess pump output flow to the reservoir for pressure control. Figure 2 shows a back-pressure regulating valve. Its impulse line with tapping downstream of oil filters is partially visible in this image.

Figure 2: Positive Displacement Pump Back-pressure Control Valve (Image courtesy of G.J. Oliver, Inc.)

Centrifugal pump systems are much simpler, as they do not require the pressure limiting valves or back-pressure control regulating valves. However, centrifugal pumps are considerably less efficient than positive displacement pumps. Generally, purchasers specify their preferred pump type.

API 614, 5th Edition specified that positive displacement type lube oil pumps shall be sized for lowest oil viscosity and capacity shall not be less than 120% of the normal oil flow under any operating condition plus 50% of control oil transient flow. These criteria were critically assessed in the development of API 614, 6th Edition, because 20% margin in high flow rate oil systems (over 946 litres per minute or 250 gallons per minute) results in grossly oversized pumps, drivers, piping, and can lead to wasted power and oil pressure control instability. As a result, API 614, 6th Edition stipulates that each pump shall be capable of supplying normal oil flow required by the equipment plus parentages based on a range of flow rate capacities, for example, 20% of normal oil flow for flow rates greater than 95 l/min (25 gal/min) and up to 760 l/min (200 gal/min) and 15% of normal oil flow for flow rates greater than 760 l/min (200 gal/min).

Positive displacement type rotary screw pumps are required to be capable of continuous operation at the rated flow, pressure-relief valve or pressure-limiting valve set pressure and minimum viscosity (i.e., viscosity corresponding to the highest oil temperature at the pump suction, plus heat rise in the pump). API 614 requires rotary screw pumps to be sized so that the pressure differential through the pump is not greater than 90% of the pump manufacturer's rating at the pumps rated capacity, with minimum oil viscosity and subject to the pressure-limiting valve's set point. These requirements in API 614 are to ensure that the selected pumps are suitable to operate in extreme conditions and designed to avoid rotor-to-stator contact.

In the case of centrifugal pumps, stipulating the more conventional 10% continuous head rise from rated point to shut-off can affect lube oil pressure controller stability due to relatively larger change in pump discharge pressure with a small change in flowrate. For this reason, centrifugal pumps with lower, continuous head rise to shut-off 5% are preferred and recommended by API 614 in single-pump

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as well as dual-pump operation scenarios. The continuous head rise characteristic enables determination of where a centrifugal pump is operating on its performance curve. API 614 also requires the sizing and selection of centrifugal pumps to include the ability of future increase of the discharge head pressure by 10 percent with the installation of a larger impeller.

The highest required discharge pressure of positive displacement and centrifugal pumps is established from maximum pressure at relief valve accumulation at full flow, plus the sum of static head, pressure drop in piping (pipe and fittings), pressure drop in control valves, and pressure drop in dirty oil filter condition. API 614, 5th Edition limited the pressure drop of clean filter elements to 30 kPa (5 psi) at an operating temperature of 40 ºC (100 °F) passing normal flow. In the 6th Edition of API 614, pressure drop limit is changed to 30 kPa (5 psi) for the "overall filter assembly" with clean filter elements or cartridges and including transfer valve.

The following pump discharge pressure cases are usually analysed, and the most demanding case considered for a given application. The selected pressure is checked against the highest specified coolant pressure (cooling water or glycol and water mixture) for shell-and-tube type heat exchangers and adjusted if required to satisfy API 614 requirement that the oil-side operating pressure shall be higher than the water-side operating pressure to avoid oil contamination if heat exchanger fails (such as a tube leak). Note that this may sometimes require regulation of the plant water system pressure at the inlet to the oil system.

Pump discharge operating cases are based on:

- Clean filter at the specified start-up temperature.
- Clean filter at operating temperature (lowest pressure case).
- Dirty filter at operating temperature.
- Full relief valve accumulation at the minimum start-up temperature (highest pressure case).

Pump Drivers

Steam turbine and electric motor drivers for positive displacement type oil pumps are sized, based on the most demanding condition out of:

a) Power required for the pump's cold oil start temperature operation at the pump discharge relief (or pressure limiting valve) setting plus accumulation.

b) 110% of the power required to pump the rated capacity at the normal oil reservoir temperature, typically 66 ºC (150 ºF) at maximum pump discharge pressure.

Steam turbine and electric motor drivers for centrifugal pumps are sized based on the most demanding condition out of:

a) Power required for cold oil start temperature operation at run-out capacity. Theoretically, run-out capacity corresponds to maximum flow at zero head, however pump manufacturer's head versus capacity curve ends just after the allowable operating region to prevent overload operation and higher vibrations. Note that when the system is first started, a centrifugal pump will operate temporarily at runout until the system fills and establishes flow resistance.

b) Power required at rated head and rated capacity point with oil at viscosity corresponding to normal reservoir oil temperature 66 ºC (150 ºF).

The API 614 default for pump cold oil start temperature is 10 °C (50 °F) unless a different value is specified by the purchaser.

Steam turbine drivers are typically used only for the main oil pumps. Due to the slow start up time (total time to initiate a start and have the pump come up to full speed), steam turbines are not recommended as drivers for the standby pumps.

Steam turbines are usually single stage turbines and provided in accordance with API 611.

Low voltage electric motor drivers in North America are typically National Electrical Manufacturers Association (NEMA) totally enclosed fan cooled, NEMA B design with a 1.15 service factor and selected for across the line (full voltage start) to minimize pressure dips on the automatic starting of the standby pump. Electric motors designed to the International Electrotechnical Commission (IEC) standards are more common in Europe and Asia regions.

There are mechanical and electrical differences between NEMA and IEC motors. NEMA motors utilize English inch unit dimensions while IEC motors utilize metric unit dimensions. Further, NEMA motors usually have sided mounted electrical connection boxes while IEC motors have the connection boxes located on the top or 12 o-clock position. In many instances, the bearing grease type is also different between the two types of motors and is not compatible.

IEC motors also have more operating duty type designations than NEMA motors. For lube oil systems, NEMA motors must be rated "Continuous Duty" which is "S1" duty type for IEC motors.

NEMA motors, used in lube oil systems, are required to be "Premium Efficiency" rated which corresponds to IEC's IE3 efficiency rating. Another difference between NEMA and IEC motors is some variation of the allowable tolerances of voltage/frequency combinations. The IEC tolerances are slightly tighter.

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API 614 special purpose oil systems require NEMA motors used in non-hazardous and hazardous Division 2 (Zone 2) areas to also be in accordance with IEEE-841 which addresses required features such as cast iron frames, shaft seals, oversized rotatable terminal connection boxes, and minimum routine testing report requirements. However, Division 1 (Zone 1) hazardous areas require motors to be TEFC explosion-proof or flame-proof which is not within the IEEE-841 design. Likewise IEC motors ratings are TEFC Ex nA non-sparking for Zone 2 applications and TEFC Ex d(e) Explosion or Flame Proof for Zone 1 applications.

Reduced voltage starting is sometimes stipulated in project specification(s) for larger motors based on site-specific electrical system characteristics. This condition and the service factor must be covered in selecting electric motor. Reduced voltage starting increases the startup time of a standby pump and must be considered in the system design by implementing accumulator(s) with adequate capacity.

Additional criteria to select drivers based on possible conditions for discharge pressure of lube oil pumps are listed below. Precaution should be taken to avoid unnecessary oversizing of drivers that can make them inefficient and negatively affect space, weight, and cost of the equipment.

- Maximum operating condition.
- 110% of maximum operating condition.
- Maximum relief valve setting.
- 110% of maximum relief valve setting.
- Maximum relief valve accumulation or pressure increase (accumulation shall not be greater than 110% of the system maximum allowable working pressure).
- 110% of maximum relief valve accumulation.
- API 676 basis (maximum relief valve accumulation).

For site conditions with elevations exceeding 1,000 meters (3,280 feet) above sea level, the reduced air density will negatively impact electric motor ratings. Per NEMA MG-1 rule of thumb motor ratings can be reduced as much as 3% per 500 meters (1,640 feet) above the 1,000 meters (3,280 feet) elevation. IEC motors utilize the same rating criteria as NEMA motors for insulation and elevation ratings.

Additionally, higher elevation above 1,000 meters (3,280 feet) can result in increased temperature rise. Therefore, installations with high elevation must be considered when selecting motor drivers.

RELIEF VALVES

Relief valves on pump discharge, thermal overpressure protection service and bypass relief are discussed in this section.

Pump Discharge Service

Positive displacement pump discharge pressure limiting valves (PLVs) must be liquid service modulating type to prevent chatter and excessive blowdown to close. Note that API 614 does not require these valves to be ASME code valves in accordance with API 520, 521, and 526 and differentiates them from “pressure safety” relief valves.

These valves must be sized to pass the entire maximum pump output flow with accumulation pressure over setpoint not exceeding 110% of the system design pressure. Piping and vessel design codes permit maximum operating pressure of 110% of the design pressure rating during relief service. Some non-ASME pressure limiting valves can have higher than 10% accumulation. The fully accumulated relief pressure must not exceed 10% over the design pressure of the system, which may require a lower setpoint.

Maximum flow situation in positive displacement pump occurs at the cold pump starting temperature when the oil viscosity is maximum. Pressure limiting valves on oil pump discharge lines are shown in Figure 3.

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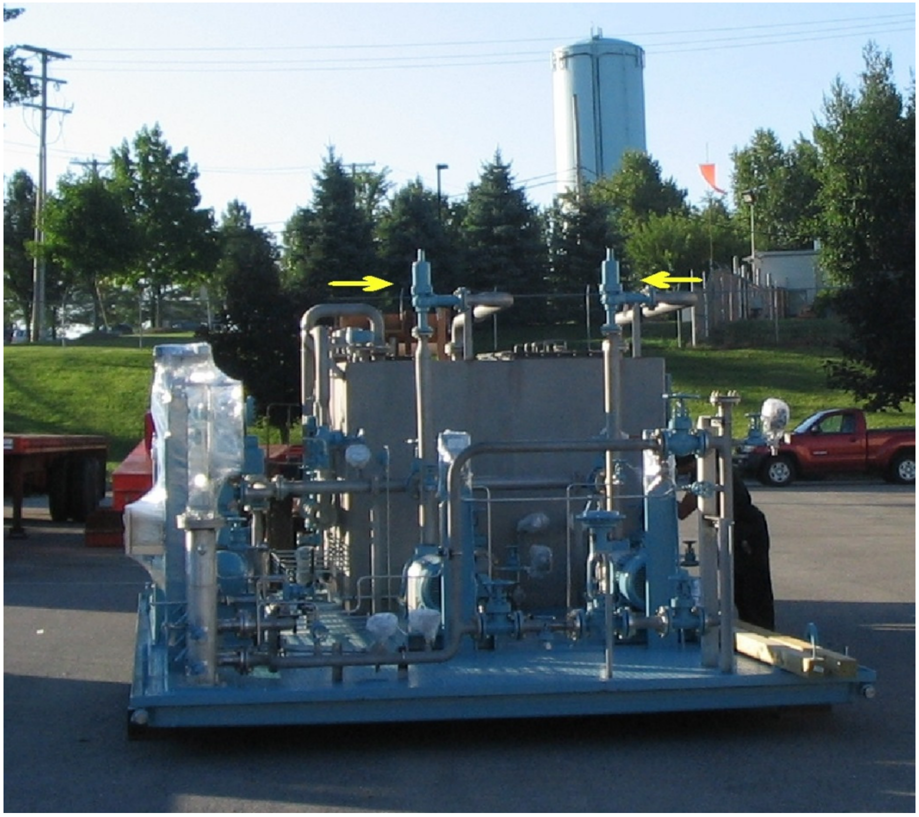

Figure 3: Pressure Limiting Valves (Image courtesy of G.J. Oliver, Inc.)

Thermal Overpressure Protection Service

Standby vessels, such as coolers and filters, which can be blocked in, can require thermal relief valves. Note that for duplex cooler and filter arrangements, API 614 does indicate the orificed fill equalizing lines can be used to protect against oil side thermal overpressure of the standby vessels provided the orifice is correctly sized for thermal relief and the equalizing valve is locked open.

Shell and tube coolers can also require thermal overpressure protection on the cooling water side if the water side of the coolers can be blocked in.

The sizing of thermal overpressure protection devices must consider the blocked in volumes and worst-case thermal rise conditions from a fire situation. API-520 and 521 provide sizing guidelines for thermal relief due to fire.

Bypass Relief Service

Relief valves can also be required to protect from overpressure due to a pressure regulating control valve failure, such as the fail-open scenario of a lube oil supply pressure regulating valve.

In this application, relief valves must limit the lube oil supply pressure to the unit and must be sized to divert the excess flow back to the reservoir during the regulating valve failure. The required relief valve capacity must be determined by considering the failed open regulating valve flow capacity, at the relief pressure, less the flow to the main equipment.

Generally, liquid service modulating pressure limiting valves (PLVs) are recommended for this application.

LUBE OIL HEAT EXCHAGERS

Lube oil coolers (heat exchangers) must be sized to provide total cooling of the oil. The total cooling heat load duty consists of heat rejection into the oil from the served equipment and also includes the heat energy input added by the oil pumps.

The oil coolers can be water cooled or air cooled as specified by the purchaser, which is normally based on whether or not a suitable liquid cooling medium, such as cooling water, is available.

Water Coolers

Water coolers can be of the shell, tube type, or plate, and frame type as specified. Plate and frame type coolers are a compact design and

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offer significant real estate space savings which is especially a significant advantage for offshore applications.

API 614 specifies that a water side fouling factor of 0.35 m^2 K/kW (0.002 hr-ft^2 °F/Btu) shall be included to ensure cooling capability with some water side fouling. To minimize water fouling with reasonable water flow, the water velocity is required to be 1.5 m/s to 2.5 m/s (5 ft/s to 8 ft/s). Ideally, coolers are sized to obtain a minimum water temperature rise of 10 K (20 ºR), but this can often be difficult while maintaining the water velocity, which takes priority.

Generally, for critical machinery applications, water coolers are configured as a duplex arrangement of 100% capacity exchangers connected with continuous flow transfer valves to enable on-line switching or parallel operation.

Shell-and-Tube Heat Exchangers

Shell and tube coolers per API 614 default are to be TEMA C design type. Generally, the coolers are configured to have oil on the shell side with water on the tube side with the oil pressure higher than the water pressure to prevent the oil from being contaminated if the heat exchanger fails. Note that the purchaser can specify the oil pressure to be lower than the water pressure, due to overriding environmental situations if the heat exchanger fails.

The purchaser may specify more robust TEMA R design coolers and other requirements depending on the facility's experience. API 614 require removable bundles and does not permit U-bend tubes unless approved by the purchaser. Different types of TEMA head and shell configurations are shown in Figure 4.

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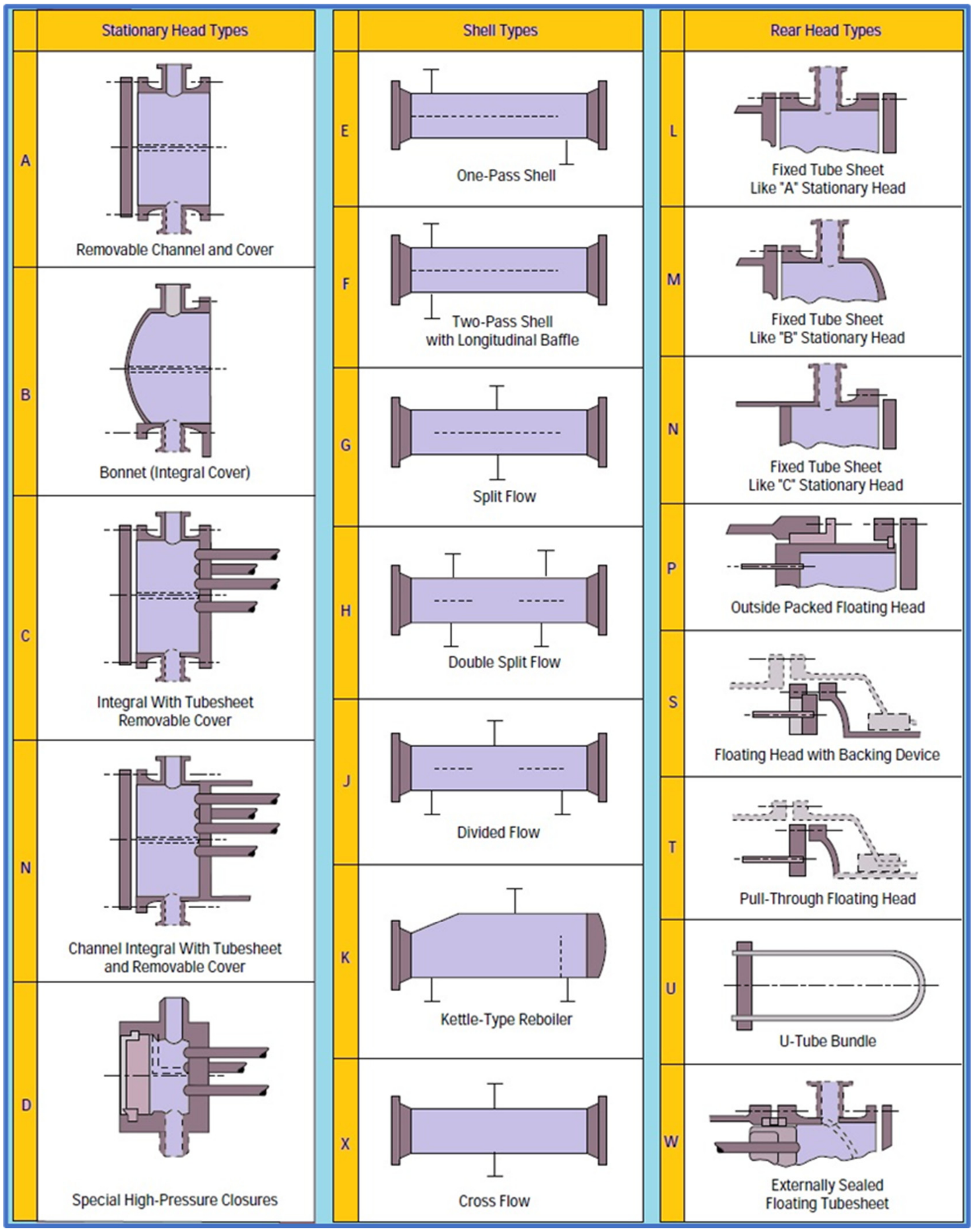

Figure 4: TEMA Exchanger Head and Shell Configurations (Image courtesy of G.J. Oliver, Inc.)

Figure 5 depicts a common typical configuration for TEMA lube oil exchangers is the AEW configuration, which incorporates removable channels and cover stationary heads with single pass shell flow (once through) with externally sealed floating tube sheet return or rear heads.

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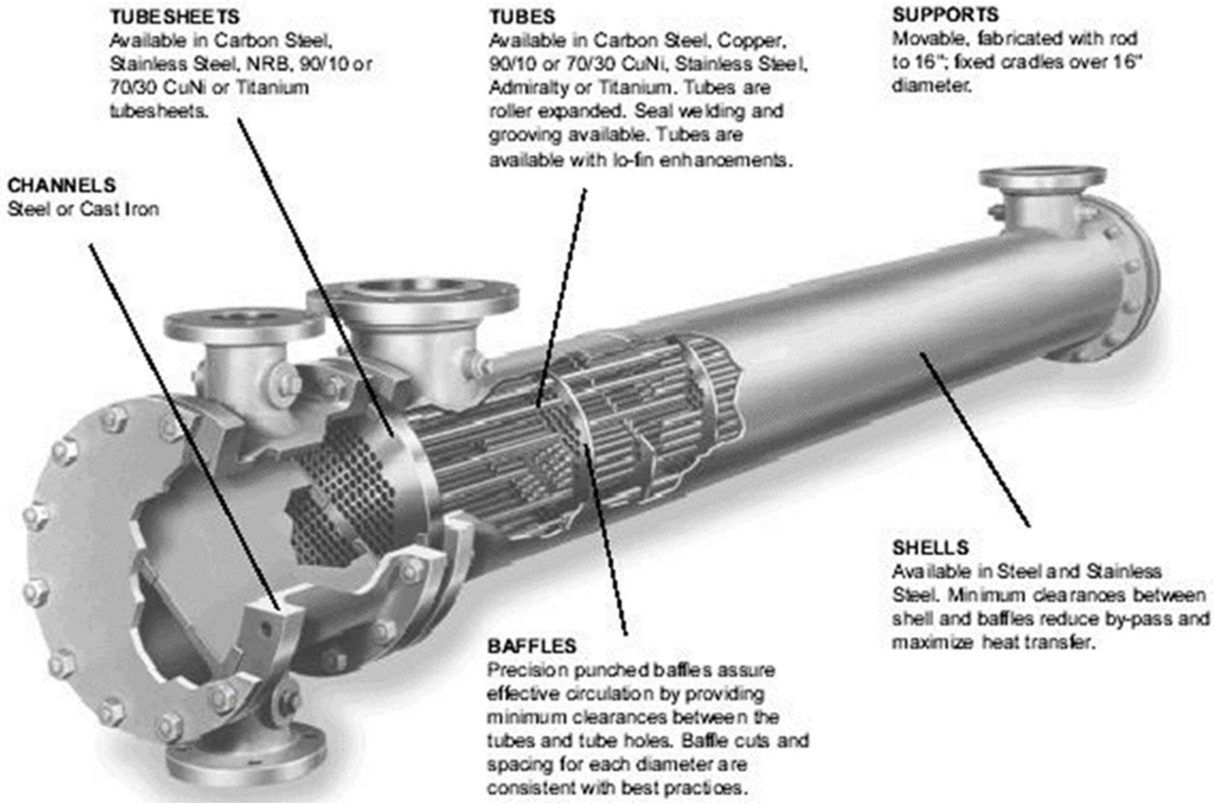

Figure 5: TEMA AEW Heat Exchange Configuration (Image courtesy of API Basco)

U-bend tube exchangers can only be used with purchaser's approval. Due to the U-bend tubes, this design can be more difficult to clean the tube interiors.

API 614 defines the default cooler construction materials. However, the purchaser can also specify other materials.

Plate-and-Frame Heat Exchangers

Plate-and-frame heat exchangers, as pointed out previously are compact and save space. API 614 requires austenitic stainless steel material for the heat transfer plates. The purchaser can specify other materials. These heat exchangers are sized for 100% duty and are also required to have the capability of future installation of at least 20% additional plates. Figure 6 shows a typical cutaway or exploded view of a plate-and-frame heat exchanger.

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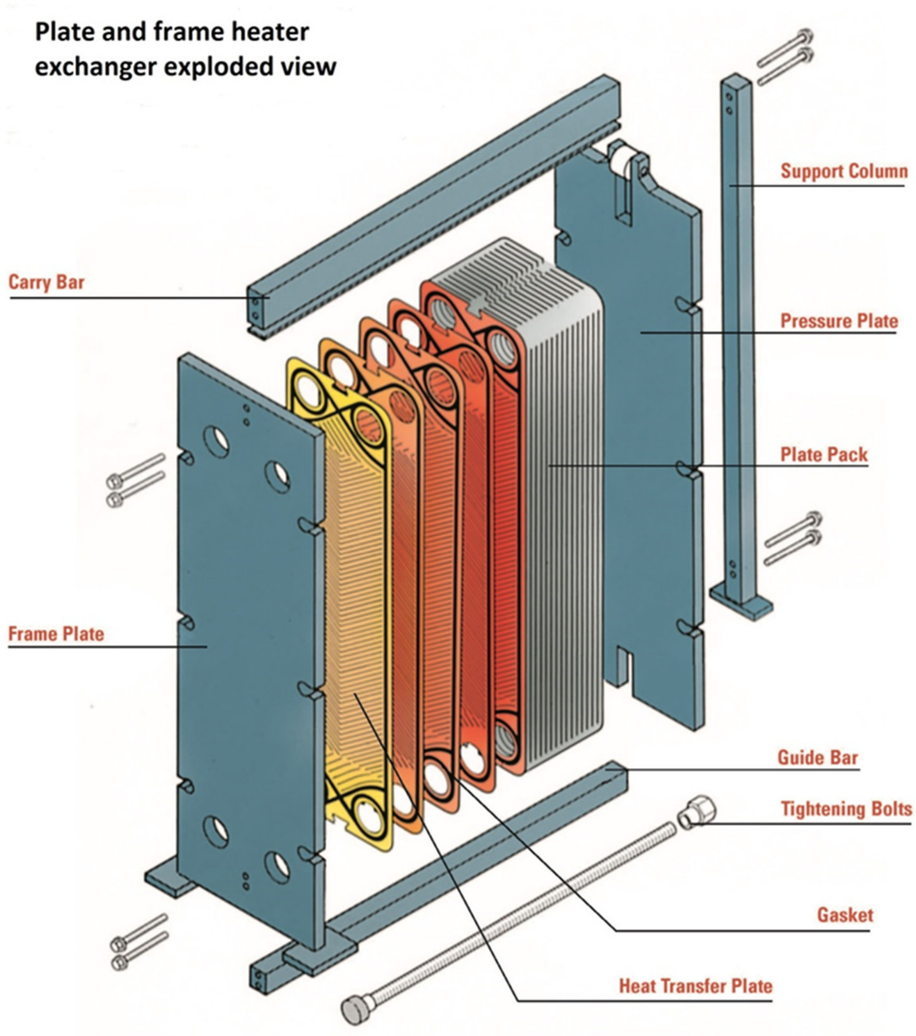

Figure 6: Plate-and-Frame Heat Exchanger Arrangement (Image courtesy of Standard-Xchange)

Plate-and-frame heat exchangers use a series of chevron-style heat transfer plates sequenced between two frame plates. Fluid inlet and outlet connections are located in the four corners of the front frame plate and each of the heat transfer plates. The oil and cooling water flows are directed to opposite sides of each heat transfer plate in typically a counter flow direction. Fluid flow path in plate-and-frame heat exchanger is shown in Figure 7.

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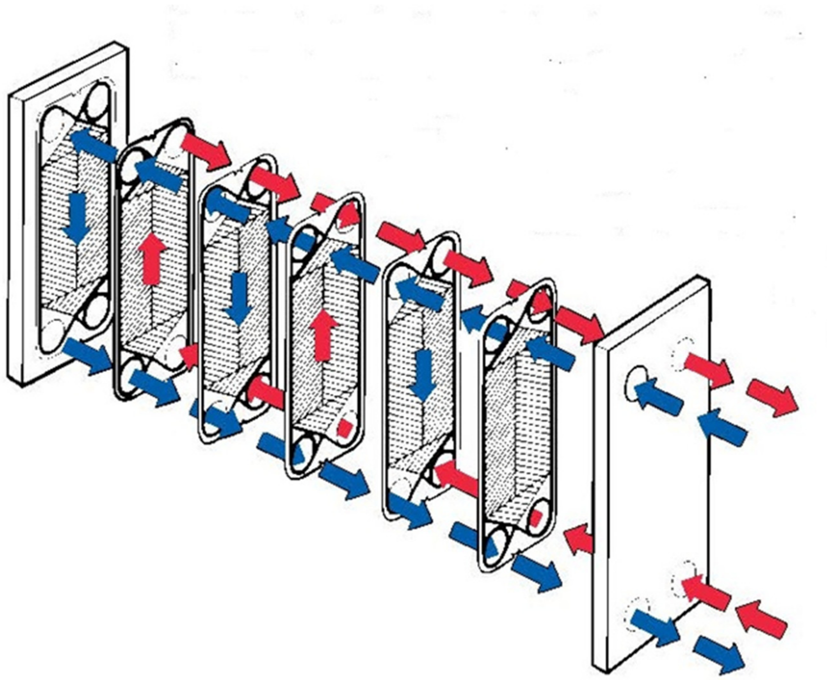

Figure 7: Fluid Flow Path within the Plate-and-Frame Heat Exchanger (Image courtesy of Standard-Xchange)

Air-cooled Heat Exchangers

Air-cooled lube oil heat exchangers are generally used where an adequate cooling fluid system is not available. API 614 requires air coolers to incorporate two-(2) 100% capacity fans by default. Further, air-cooled heat exchangers are provided with straight finned tubes with a header box on each side with 10% spare tubes. Except for small systems, air coolers are normally mounted with the bundles horizontally with a structural galvanized steel support. Since the force draft fans are mounted underneath, the cooler bundles are usually quite elevated to provide maintenance access for the fans and drives. As a result of the bundle elevations, elevated manway access platforms and ladders are required.

Carbon steel is the default material for the tubes and header boxes; however, the purchaser can specify stainless steel.

Air-cooled heat exchangers can incorporate flow turbulators, within the tubes, to improved heat transfer efficiency, thereby reducing the heat transfer area required. However, the use of turbulators must be approved by the purchaser.

Different methods of temperature control can be implemented on air-cooled heat exchanger, such as adjustable frequency drive or AFD (also known as variable frequency drive; VFD) powered fan motors for fan speed control, louvers for air flow control, or temperature control by oil bypass.

Hail guards are generally necessary and a good idea to protect the tube fins from being bent by hailstorms.

If using air-cooled heat exchangers, the oil volume in the exchanger bundle assemblies, along with interconnecting piping, must be included in the oil reservoir rundown capacity. The distance and size of the interconnecting piping not only influences the oil volume but also the system frictional pressure losses, alone with the bundle flow pressure loss, which must be included in determining the required pump discharge pressure.

Figure 8 shows a lube oil system with an air-cooled heat exchanger mounted directly on top of the oil console due to unavailable space to mount the exchanger elsewhere.

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Figure 8: Air Heat Exchanger Mounted on the Lube Oil System (Image courtesy of G.J. Oliver, Inc.)

OIL TEMPERATURE CONTROL

Oil temperature control is accomplished by directing all of the oil through or some portion of the oil around (bypass) the oil heat exchanger to maintain the desired oil supply temperature.

For water cooled heat exchangers, it is not recommended to throttle the water flow. Reducing water velocity through the exchanger for temperature control can result in accelerated fouling of the water side heat transfer surfaces (inside the tubes of typical configured shell and tube exchangers).

Three-(3) basic types of oil temperature control valve arrangements are discussed in the following paragraphs.

Thermostatic Temperature Control

The first is a self-contained thermostatic 3-way mixing temperature control valve or regulator. This device has two-(2) inlet ports and a common temperature controlled oil outlet port. One inlet port is connected to the cooled oil from the heat exchanger outlet. The other inlet port is connected to the uncooled oil taken from upstream of the heat exchanger, which is known as cooler bypass oil.

The thermostatic valve contains a modulating valve plug element that opens and closes the cooler bypass oil port with temperature. The element contains thermally expanding wax material, which forces the valve plug to move in response to the temperature. The valve provides the ability to mix the two-(2) oil streams (hot bypass and cooled flow) to satisfy the desired temperature setpoint, which is fixed for a given element.

Thermostatic valves can be provided with manual overrides to move the valve plugs to the fully extended hot oil position, closing off the hot oil bypass valve port, in the event that the thermostatic element should fail. Thermostatic valves can also incorporate block and bypass valve arrangements, if desired, to permit online maintenance if necessary.

Two-way Cooler Bypass Temperature Control

The second is utilizing a 2-way temperature control valve, which provides a parallel hot oil bypass flow path around the heat exchanger allowing hot oil to mix with cooled oil to maintain the desired temperature. The control valve is normally a pneumatically operated valve, which is modulated via a temperature controller with the temperature sensing device, located downstream of the bypass and cooled oil mixing tee.

Since the 2-way bypass flow path is parallel to the heat exchanger flow path, the total 100% of the flow cannot be diverted around the heat exchanger. To provide the capability of the bypassing approximately 55% to 65% of the flow, the 2-way bypass control valve must be sized so that the total pressure loss of the bypass flow path, with the control valve fully open with 100% flow, is 15% to 25% of the total pressure loss of the heat exchange flow path with 100% flow.

The 2-way temperature control valves are normally provided with block and bypass valves to permit online maintenance. Additionally, the control valve can be fitted with manual override hand wheels.

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Three-way Temperature Mixing Control

The third and best temperature control arrangement incorporates a 3-way control valve, which is similar in function to the self-contained thermostatic control valve in that it mixes the cooled and the hot bypass oil streams to obtain the desired set point. However, this arrangement relies on a temperature controller sensing oil temperature downstream of the mixing valve versus a thermostatic element.

The advantages of this arrangement are:

a) Oil can be directly fully through the heat exchanger circuit or fully through the bypass circuit and any ratio of mixture as required for maintaining the oil set point or for other diagnostic procedures. The temperature controller can be placed in manual to drive the valve to a specific position.

b) The oil temperature set point can be changed, as desired, via the controller's entered set point value. In some situations, the main equipment rotor systems may behave better at a slightly higher viscosity than the specified design value, which can be achieved by lowering the oil supply temperature.

A temperature-control valve is installed on a bypass line around oil cooler to control and maintain the oil supply temperature. Figure 9 shows views of a three-way temperature control valve, its impulse line sensing oil outlet temperature and bypass piping to facilitate maintenance of the valve.

Figure 9: Oil Temperature Control Valve (Image source: Yoshimine, Japan)

LUBE OIL FILTERS

Duplex (2 x100% capacity) filters with on-line changeover capability are used in lube oil and hydraulic oil services for turbomachinery trains. Filters protect bearings in the served equipment by removing contaminants, reducing oil consumption and disposal, and extending life of the lubricant. Typical contaminants in lubricating oil can include particulates, wear metals, acids, moisture, varnish, or gases.

The information required for sizing and selecting the type of filters include: type of oil (mineral or synthetic); oil viscosity and the associated minimum and maximum operating temperatures; degree of filtration; types of contaminants; system volume or capacity; required and maximum design flow rates; design and operating pressures and temperatures; type of filter element; maximum allowable pressure drop and the associated temperature; and Code construction (for example, ASME Section VIII, Division1).

Filters are sized to meet clean filter "element" pressure drop of less than 30 kPa (5 psi) at maximum flow at 40 °C (100 °F) as per API 614. The 6th Edition of API 614 has changed the 30 kPa (5 psi) drop limit to the "overall filter assembly" including transfer valve with clean elements as opposed to just the element(s).

At maximum flow and dirty filter pressure drop condition, typical element change out differential pressure is 103 kPa to 172 kPa (15

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psi to 25 psi). Pressure drop values are determined from consideration of worst-case flow and cold start viscosity. A safety factor of 3 to 4 is recommended on margin between collapse differential pressure and maximum pressure drop. API 614 requires a minimum collapsing differential pressure rating on a filter element of 500 kPa (70 psi).

API 614 specifies the default flow direction through filter elements as flowing from the outside inward towards the filter element center unless approved or specified. It is important to note that coalescing filter elements, used in lube oil conditioners and purifiers, typically flow in the opposite direction from the center of the elements toward the outside to maximize the coalescing efficiency.

Once filter sizing is completed, application-specific requirements and configuration options, such as the following can be easily integrated in the filter system, to make it a custom-engineered product:

- The type of inlet and outlet connections: Flanged or threaded (FNPT).
- Cover lifter: API 614 specifies lifters for covers weighing in excess of 16 kgs (35 lbs.).
- Instrumentation: Differential pressure gauge or differential pressure transmitter across filters.
- Dedicated filter for control oil, if required.
- Valved vents and drains: for example, ¾" 300# ANSI raised face flanged vent and clean and dirty-side shell drain connections.
- Code Stamp and Registration: for example, ASME National Board, Canadian Registration Number (CRN) or Pressure Equipment Directive (PED).
- Gasket material: must be compatible with oil type.
- Transfer valve for duplex filters: for example, integral, straight plug type, diverter, ball, two-way, six-ported, continuous-flow transfer valve and equalizing line (or thermal overprotection relief valves for the blocked in out-of-service filter, if thermally-sized orificed equalizing line is not provided).
- Controlled pleat radius for increased filtration area, dirt retention, and resistance to deformation of filter element.
- Some purchasers apply pressure vessel and welding specifications that can require additional NDE, specific corrosion allowances, types of flanges (i.e., weld neck), flanged vents and drains, etc.

API 614 defines the default minimum particle filtration level to be 10 μm with 90 percent efficiency, which is defined as $\beta 10 \geq 10$.

Filter efficiency is expressed by Beta Ratio βx, which is the ratio of the number of particles greater than a given size upstream of the filter, divided by the number of particles greater than the same size downstream of the filter (see Equation 1). Using the filter Beta Ratio is a precise method of specifying filter performance ratings because different filter manufactures can define "nominal" ratings with different efficiency values. Beta Ratio thus eliminates the need for nominal filter ratings that can be misleading when comparing performance of filters of different types from different manufacturers.

$$\beta x \geq Nu / ND \tag{1}$$

where;

x = Particle Diameter μm

Nu = Number of Particles greater than X μm upstream of the filter

ND = Number of Particles greater than X μm downstream of the filter

The following two examples illustrate how the Beta Ratio of filters is calculated.

a) If $\beta 6 \geq 15{,}600/78$, then $\beta 6 \geq 200$

b) If $\beta 5 \geq 15{,}982/213$, then $\beta 5 \geq 75$

Particle removal efficiency of filter is determined from $\eta = \{[1-(1/\beta x)] \times 100\}$. Table 1 shows filter efficiencies for various Beta Ratios. API 614 specifies as a default, minimum efficiency of 90% for 10 μm particles ($\beta 10 \geq 10$), and minimum efficiency of 99.5% for 15 μm particles ($\beta 15 \geq 200$), when tested as per ISO 16889 standard (Hydraulic fluid power—Filters—Multi-pass method for evaluating filtration performance) to a minimum end of test run differential pressure of 345 kPa (50 psi).

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Table 1: Beta Ratio and Efficiency, Courtesy of The Hilliard Corporation

Beta Ratio	Particle Removal Efficiency (%) of Filter Element
1	0
2	50
4	74
10	90
20	95
75	98.7
100	99
200	99.5
1000	99.9

Cellulose paper cartridge filter elements are hygroscopic (absorb water) and need frequent replacement and are seldom used in modern oil systems. Reliability-focused plant operators have long moved away from this type to non-hygroscopic synthetic filters for lube, hydraulic and control oil services. Figure 10 illustrates higher pressure drop of cellulose filter element (PL718) with flow in comparison to three types of synthetic filters (PH718).

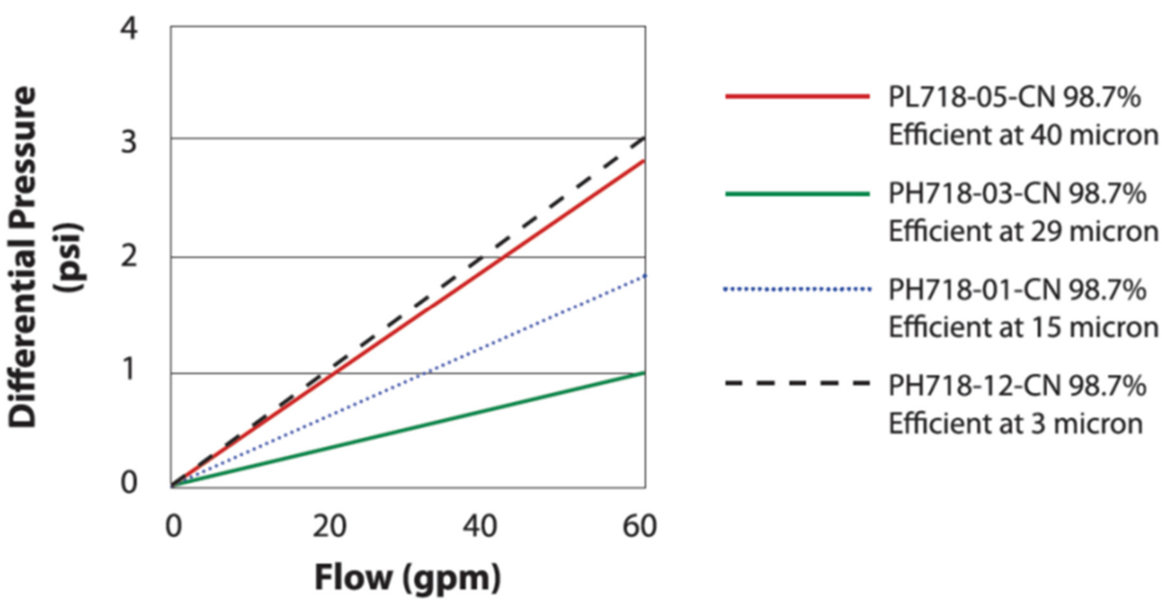

Figure 10: Flow vs. Pressure Drop of Cellulose (PL) and Synthetic (PH) filters passing 220 SSU oil (Image courtesy of The Hilliard Corporation)

Figure 11 shows a duplex filter with equalizing line and transfer valve. Figure 12 represents filter elements with center support tube.

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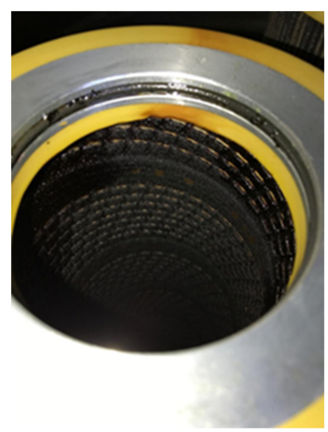
Figure 11: Duplex Filter Figure 12: Filter Elements
(Both images are courtesy of The Hilliard Corporation)

Static charges can build on the non-conductive components of the lube oil system, such as the filter elements if the lubricant has low electrical conductivity (less than 50 pS/m, a common unit of conductivity is a pico Siemens/Meter (pS/M).). When a sufficient charge is accumulated, static discharge occurs between two points of differing potentials. Over time, static discharge contributes to oil degradation and varnish build up throughout the system. The severity of the varnish formation can be aggravated by the specific oil chemistry, hot spots in the lube oil system, and static discharge.

Anti-static filters help to minimize potential for varnish formation. A particular design of anti-static filter elements incorporates conductive fibers, which are pleated in between two layers of microglass filter media. It dissipates static charge buildup before discharge can occur. Fluid conductivity can be determined by utilizing ASTM standard D2624 (Standard Test Methods for Electrical Conductivity of Aviation and Distillate Fuels). Laboratory and field testing has revealed that fluid conductivity affects the number and severity of static discharge. Oils having a conductivity < 35 pS/M are prone to high intensity static discharges while oils > 200 Ps/M exhibit no static discharge issue. The progressive effect of static discharge on lube oil filter element is shown in Figure 13.

New Filter

After 2 Months Run



Severe Varnish Deposit

Figure 13: Static Discharge Effect on Lube Oil Filter Element (Image courtesy of The Hilliard Corporation)

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TRANSFER VALVES

Transfer valves are used in duplex oil filters, duplex shell, and tube type heat exchangers and duplex plate, and frame type heat exchangers. Many modern transfer valves have equalization ports or a pressure equalization line to allow uninterrupted flow transfer and low turning moment when switching from online to standby filter or exchanger. Some valves feature metal-to-metal sealing and sealing pads coated with low friction materials while the other type has elastomeric material embedded in sealing pads. Both design permit low turning torque.

Full-port valve design provides reduced pressure drop. API 614 allows use single-body six-port straight plug type transfer valves or two three-way ball or plug valves connected by single operating lever. Three-way, six-port transfer valves can have oil inlet and outlet ports on the same side or on the opposite sides as illustrated in Figure 14. Split type three-way transfer valves and Pressure Drop versus Flow characteristics of same side and split type valves are shown in Figures 15 and 16.

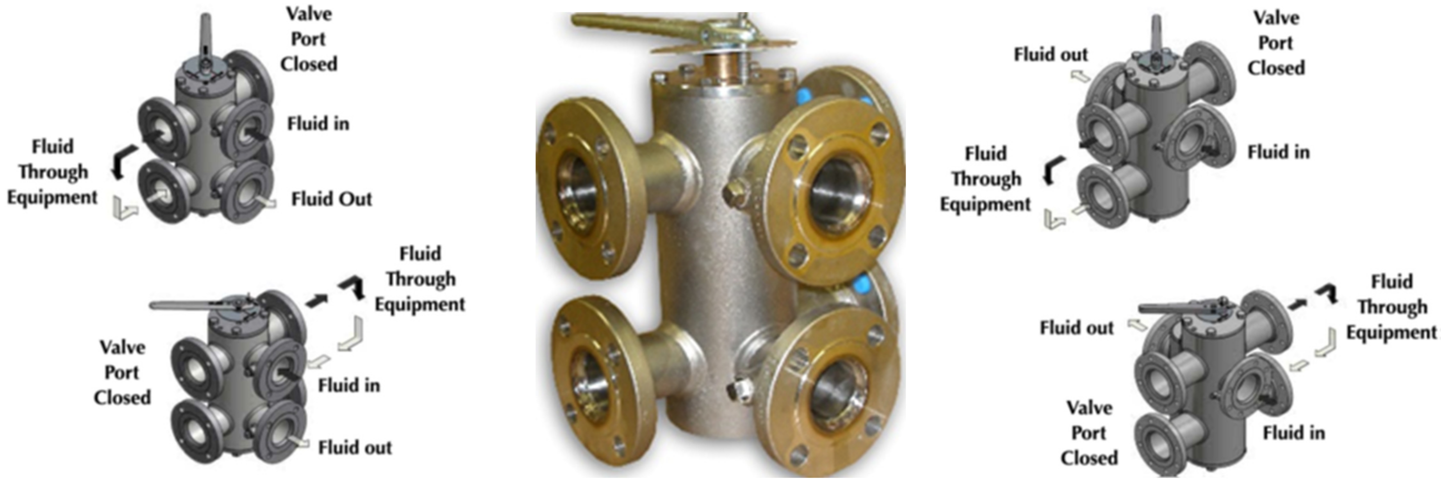

Figure 14: Six-port Transfer Valves (Image courtesy of The Hilliard Corporation)

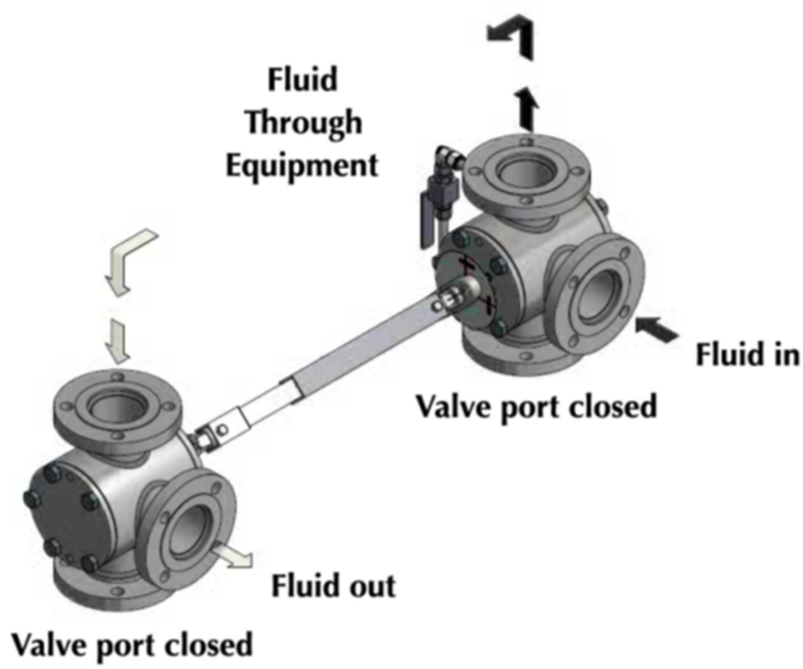

Figure 15: Split Three-way Transfer Valve (Image courtesy of The Hilliard Corporation)

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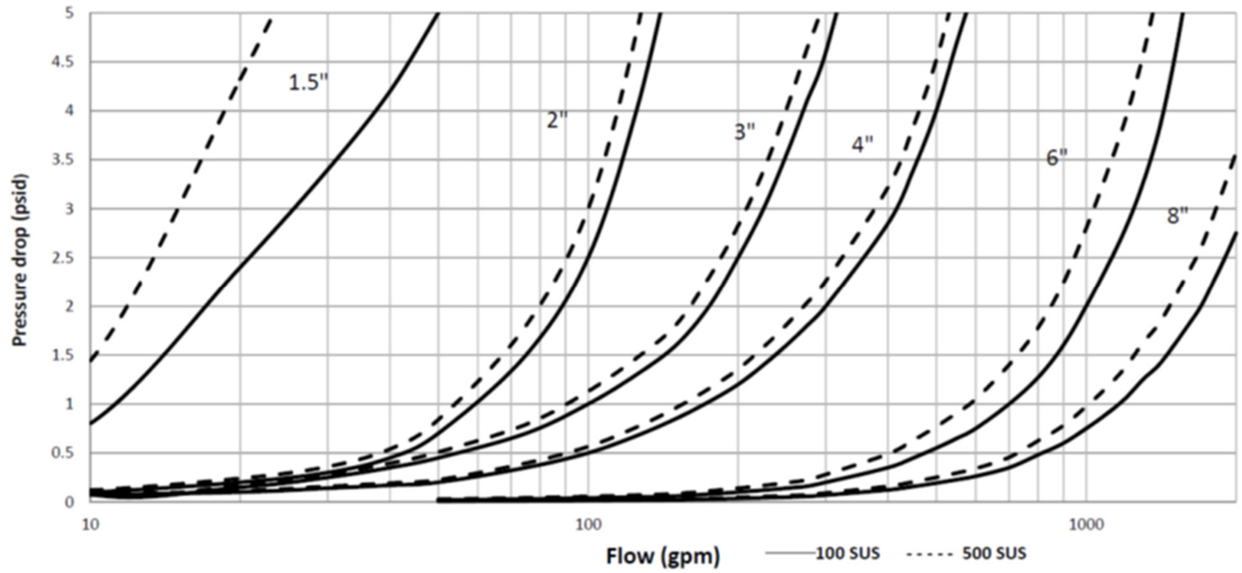

Figure 16: Pressure Drop vs. Flow of Split Type and Same Side Transfer Valves (Image courtesy of The Hilliard Corporation)

LUBE OIL AND CONTROL OIL PRESSURE REGULATORS

Back Pressure Regulator and Pressure Reducing Valve are described in the following paragraphs.

Back Pressure Regulator

Positive displacement pumps have relatively constant discharge flow, which is higher than the required system output flow based flow margin requirements defined by API 614 pump sizing criteria. As a result, all of the excess flow must be recirculated back to the reservoir via the back-pressure regulator valve. Back-pressure valves must also be capable of recirculating the excess oil associated with running both the main and standby pumps simultaneously, which adds 100 percent of the second pump's discharge flow to the recirculated oil flow.

Back-pressure valves must be of "fail close" type and able to maintain the pump discharge pressure based on all operating conditions without pressure limiting valves lifting when starting a second pump.

The following is an example for determination of the required valve flow coefficient (Cv) range for a back-pressure valve on system with motor driven main and standby positive displacement pumps with the following conditions:

- Lube Oil Reservoir is at atmospheric pressure.
- Fluid: ISO VG32 with Sg = 0.87, Viscosity = 650 SSU @ 10°C (50°F) and 60 SSU @ 66°C (150°F)
- Lube Oil Supply Flow: 322 l/min @ 172 kPa @ (85 gal/min @ 25 psig) [Includes 34 l/min (5 psi) combined frictional and head loss to the equipment train)
- Turbine Governor Control Oil: 19 l/min (5 gal/min) Normal [57 l/min (15 gal/min) Transient] @ 793 kPa (115 psig)
- Required Pump Output (based on including control oil transient): 454 l/min (120 gal/min)
- Actual Pump Discharge Flow: 466 l/min @ 1034 kPa & 66°C (123 gal/min @ 150 psig & 150°F) Hot , 473 l/min @ 1034 kPa & 10°C (125 gal/min @ 150 psig & 50°F) Cold
- Pump Pressure Limiting Valve Setting is 1207 kPa (175 psig)

Note that the back-pressure valve will be subject to a pressure drop equal to the full pump discharge pressure since the outlet of the valve is returned to an atmospheric reservoir, which could result in flashing for some fluids. However, for most lube oils, the vapor pressure is near 0 kPaa (0 psia), and hence, flashing will not occur and is not a concern.

Minimum Cv is determined from the minimum recirculation flow, based on one pump running hot with lube oil and control oil at transient flow (see Equation 2).

$$Q(Min) = 466 - (322 + 15) = 57 \text{ l/min} \qquad \text{(Equation 2a SI Units)}$$

$$Q(Min) = 123 - (85 + 15) = 23 \text{ gal/min} \qquad \text{(Equation 2b USC Units)}$$

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To be conservative, the minimum flow sizing Cv is considered at the highest back-pressure valve inlet pressure, which is the PLV set pressure of 663 kPa (175 psig).

Minimum Cv = 1.86

The maximum Cv is determined from the maximum recirculation flow based on two-(2) pumps running with cold oil with minimum system output flow of lube oil and normal control oil (see Equation 3).

$$Q(Max) = 2 \times 473 - (322 + 19) = 606 \text{ l/min} \quad \text{(Equation 3a SI Units)}$$

$$Q(Max) = 2 \times 125 - (85 + 5) = 160 \text{ gal/min} \quad \text{(Equation 3b USC Units)}$$

To be conservative, the maximum flow sizing Cv is considered at the normal back-pressure valve inlet pressure, which is 1034 kPa (150 psig).

Maximum Cv = 13.89

Note that the Cv calculations above are general computations using Equation 4 below:

$$Cv=Kf*Q*SQRT\ [Sg/((P1-P2)\ )] \quad (4)$$

where:

Cv = Valve flow coefficient

Kf = Valve factor incorporating recovery factor, viscosity correction, etc.

Q = Flow, *l/min (gal/min)*

Sg = Specific gravity of fluid

P1 = Upstream pressure, *kPa (psig)*

P2 = Downstream pressure, *kPa (psig)*

Cv is typically computed using the valve manufacturer's sizing equations, recovery, and viscosity correction factors and/or software.

The back-pressure valve selection should be to place the required Cv operating range within the 10 to 90 percent of the valve's available operating Cv.

Back-pressure valves can be either direct-acting regulator valves or pneumatically operated control valves.

Direct regulator type valves are generally faster than pneumatic control valves but have proportional only action, which requires pressure buildup or accumulation to open them for additional flow capacity. Consideration must be given to the actuator diaphragm and spring sensitivity (psi / inch of valve travel) and the amount of pressure buildup required for the two pump run condition. The set-pressure pump of the pump pressure limiting valves must be set at a margin higher than the maximum pressure determined during the two-(2) pump run to avoid lifting of the pressure limiting valves when starting a second pump. Some overshoot of the steady state pressure for two pump run will be experienced on the startup as a result of the back-pressure valve loop response delay as compared to the second pump output response time on starting.

Pressure sensing lines should be self-purging of trapped air by incorporating bleed return line orifices. The compressibility of trapped air slows the response of the valves.

The trim characteristics of Quick Opening/Linear/Equal Percentage should be evaluated to maintain a constant loop gain response for system stability. It is noted that some direct acting back-pressure valves incorporate actuators with less stem travel than the valve body trim travel, which makes the Quick Opening characteristic desirable since only a reduced percent the valve's capacity is utilized.

If pneumatic valves are used, the combined loop response time of the pressure transmitter, pressure controller, valve positioner, and control valve have to be very quick, and at the same time, stable. Generally, a properly tuned 2-mode proportional plus integral (reset) pressure controller function is implemented in this application for achieving the best steady state and dynamic control response.

Pressure Reducing Valve (Lube Oil Supply Header Pressure Control)

Pressure reducing valves are to be fail open valve and have a much smaller Cv operating range since the oil flow and outlet pressure is relatively constant with the only variable parameter being the inlet pressure to the valve. The variation of the inlet pressure is due factors such as the filter pressure drop based on element cleanliness, drop in pressure permitted before imitating the standby pump start, and the maximum pressure associated with the pump operating just below the pressure limiting valve set-pressure.

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For the example system given above, determination of the required valve flow coefficient (Cv) range for the pressure reducing valve would be:

Minimum Cv is determined from the normal lube oil flow of 322 l/min (85 gal/min) with the valve inlet pressure associated with the pump operating near the PLV set-pressure of 1207 kPa (175 psig) with clean filters [34 kPa (5 psid)] and normal pressure losses for other parts of the system which in this case is determined to be 1103 kPa (160 psig).

Minimum Cv = 7.16

The maximum Cv is determined from the normal lube oil flow of 322 l/min (85 gal/min) with the minimum valve inlet pressure associated with the pump operating at 60 kPa (10 psi) below the 1034 kPa (150 psig) normal discharge pressure (for standby pump start), with dirty filters [@ 103 kPa (15 psid)], and other normal pressure losses for other parts of the system, which in this case is determined to be a valve inlet pressure of 793 kPa (115 psig).

Maximum Cv = 8.35

The same considerations should be evaluated for selection of trim characteristics (Quick Opening/Linear/Equal Percentage) based on the loop gain of the valve for system stability.

Another point to consider is to ensure over-pressurization of the equipment train lube oil supply does not occur if the valve should fail open. If the valve cannot be selected to ensure that the failed open condition will not result in an overpressure situation, the valve should be provided with an open travel stop to limit the valve's output. Alternately, a pressure limiting valve installed downstream of the lube oil supply pressure reducing valve can be provided to recirculate excess oil flow back to the reservoir on failure of the pressure reducing valve.

As with the back-pressure control, pressure sensing lines should be self-purging of trapped air by incorporating bleed return line orifices. The compressibility of trapped air slows the response of the valves.

Also, similarly if pneumatic valves are used, the same loop response characteristics are required as for the backpressure valve. Figure 17 shows a pressure-reducing control valve on oil supply header and its sensing line downstream of the valve.

Figure 17: Lube Oil Pressure-reducing Control Valve (Image courtesy of G.J. Oliver, Inc.)

OIL SYSTEM PIPING DESIGN

The lube and control oil system mechanical design rating must be applied to the pressure piping, vessels, and components and it must be equal to at least the pump discharge pressure limiting valve set point. The set point plus accumulation, of the pump discharge limiting

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valves, establish the maximum pressure the system must be good for. ASME, Canadian, and PED pressure vessel codes allow operation up to the vessel design rating for maximum normal operation and up to 110% of the system design rating relief service. Note that pressure vessel codes permit operating pressure up to 110% of design during relief service.

Some non-code pressure limiting valves can have higher than 10% accumulation at full relief flow, which can require their set-pressures be adjusted downward so that the 110 percent of the vessel ratings is not exceeded during fully accumulated relief service.

The system piping must be rated to at least the pressure limiting valve set-pressure at the maximum operating temperature. Default minimum pipe schedules (wall thicknesses) for given pipe sizes are defined by API 614. However, the pipe schedules must comply with the wall thickness calculations per ASME B31.3 or other specified piping code.

Further, some project specifications may impose additional requirements such as restricting the use of slip-on flanges requiring only weld neck flanges, larger minimum radius requirements for pipe bending, material source restrictions and specific non-destructive examination (NDE) and extent for piping (i.e., radiograph, liquid penetrant, and positive material inspections).

Piping is sized to limit frictional pressure losses, hydro-dynamic noise, and to maintain flowing velocities under 2 m/sec to 3 m/sec (7 ft/sec to 10 ft/sec) for reasonable friction losses, unlike high-pressure hydraulic system design criteria, which utilizes much higher flow velocities.

API 614 does not permit the use of the following pipe, fitting, and valve sizes: 30 mm (1-1/4 in); 65 mm (2-1/2 in); 90 mm (3-1/2 in); 125 mm (5 in); 175 mm (7 in); or 225 mm (9 in).

Pipe flange pressure temperature ratings are provided in ASME B16.5 or other specified flange code. The majority the API 614 lube and control oil systems utilize ASME 150# rated flanges except for some lube and control oil applications with higher required control oil pressures, which often require 300# rated flanges.

Note that systems with seal oil are not covered in this discussion. They can require much higher flange ratings. Additionally, control oil systems operating at hydraulic pressures higher than 300 PSIG are typically not combined with the lube oil systems and are stand-alone systems often utilizing other hydraulic system standards.

Atmospheric oil drain return lines from the main equipment are required to be sized to run half full based on a 40 mm/m (½" per ft) minimum continuous slope towards the reservoir. Keeping the lines no more than half full ensures the bearing and gear housings can adequately breathe and are sufficiently drainable to prevent over flooding, which can result in leaks. Additionally, the half full oil drain lines allow air and nitrogen flows from separation seals, coupling windage and air ingress from bearing and gearbox breathers to flow towards the reservoir for vent (or extraction if blower assisted powered vent demisters are used).

ACCUMULATORS

Bladder, diaphragm, and piston type accumulators are used for many applications, such as energy storage, emergency, and safety functions, damping of vibrations, fluctuations, pulsations, shocks, and flow stabilization. Bladder and direct-contact type accumulators are included and described in API 614. Bladder type accumulators are most commonly applied in lubrication and control-oil systems. In some systems where large store oil volumes are required, direct contact type accumulators are implemented.

The compressibility of a gas is utilized in hydraulic accumulators for storing fluids. Bladder accumulators are based on this principle, using nitrogen as the compressible medium. A bladder accumulator consists of a fluid section and a gas section with the bladder acting as the gas-proof separation element. The oil on the outside of the bladder is connected to the hydraulic circuit so that the bladder accumulator draws in oil when the oil pressure increases, and the gas is compressed. When the oil pressure drops, the compressed gas expands and forces the stored oil into the circuit.

Dynamic analysis can be used to confirm need for accumulator(s) in API 614 lubrication oil systems. Accumulators perform a number of functions in the lubrication and control-oil circuits of the served equipment. In lubrication circuits the accumulator's primary function is to provide a buffer or additional source of oil flow. In other words, accumulator on lube oil service maintains the system pressure above low-low oil pressure trip setpoint during manual pump changeover or when the standby oil pump is ramping up on speed from idle to its rated speed.

Lube oil accumulator is installed on oil console and located on the equipment oil supply header downstream of the oil filters. It is sized to discharge oil into the system to maintain the system pressure above low-low oil pressure trip setpoint for at least four seconds for electric motor driven oil pump as required by API 614. The connection between equipment oil supply header and accumulator includes a block valve and bypass line with orifice or a drilled check valve. Accumulator's drain (bleed) line is piped back to oil reservoir or connected to oil return header. It includes a block valve and a restriction orifice.

On the control side, the purpose of the accumulator is similar. Flow of hydraulic oil is needed to control the equipment but also to minimize the size of the pump. Here the accumulator is used to help the pump when fast, high-flow inputs are needed. Since these are usually only of short durations, an accumulator can provide this while still being able to use a small pump for normal operation. Control oil accumulator for steam turbine driven equipment train maintains oil pressure in the control oil header above the minimum pressure

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specified by turbine manufacturer for all operating conditions, including servo motor transients. For quick response, control oil accumulator is located as close as possible to hydraulic actuator in the turbine governing system.

Typically, the purchaser is required to provide the following information to size an accumulator:

- The pressure range in which the system is required to operate. When referring to accumulators, this is always a range; the maximum pressure in which the system operates and a lower limit, or the minimum pressure required to do the job (either to maintain bearing lubrication or move an actuator). These are frequently referred to as P2 (maximum pressure) and P1 (minimum pressure).
- How much fluid the accumulator is required to deliver. This will need to be calculated as the difference between the pump flow and maximum demand of the system. This can be a specified volume or a specified flow over a given length of time.
- Corresponding to flow, the required discharge rate and discharge time (duty cycle or the length of time the accumulator must deliver a given flow in a transient situation and how often this is expected).
- The expected ambient temperature range the system will be working in, as this affects the temperature exchange between the accumulator and its environment.
- Where the accumulator is to be installed to ensure that the vessel design is registered correctly. Every country and many regions have their own rules and regulations governing pressure vessels.
- The type connections on fluid and gas ports.

The above data is used by accumulator designer to determine the size of accumulator required and generate performance graphs for pressure and volume to demonstrate the work the accumulator does.

The gas pre-charge pressure is established as part of accumulator sizing. This is the pressure of the nitrogen gas in the accumulator when the unit is not active. Pre-charge pressure influences the strength of the spring that provides the hydraulic storage force during operation. In general, the rule of thumb is that the pre-charge (P0) should be set at 90% of the minimum pressure (P1). There are exceptions to this rule, as with most rules, but this is always a good starting point.

Figure 18 shows effects of pressure and volume inside a bladder type accumulator. As seen in this image, the accumulator is pre-charged with nitrogen. The separation element (bladder) shuts off the fluid. The middle image infers that the minimum operating pressure should be higher than the gas pre-charge pressure. This prevents the separation element from striking the fluid port every time fluid is discharged. The image on the right shows that once the maximum operating pressure is reached, the effective volume ΔV is available in the accumulator.

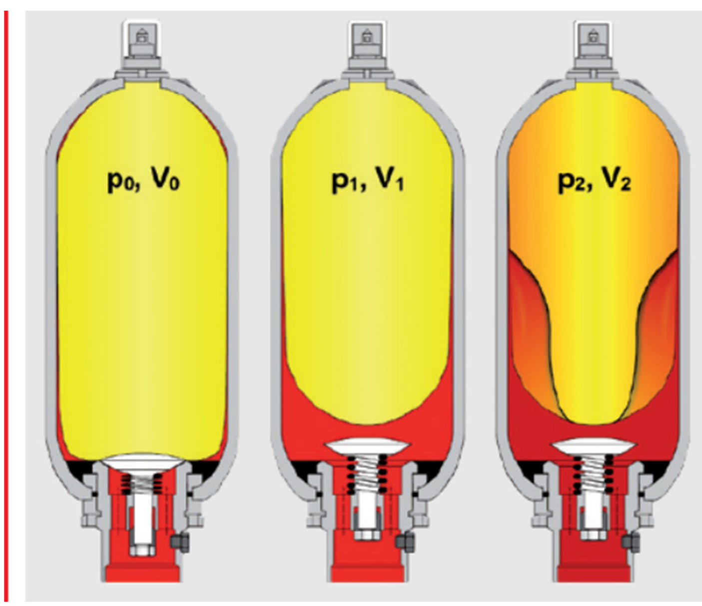

Figure 18: Action of Bladder Accumulator (Image courtesy of HYDAC)

The standard bladder accumulator consists of a “closed” elastomeric bladder inside a forged or fabricated stainless steel shell. An anti-extrusion (mechanically actuated) valve closes when the oil has been expelled, blocking off the fluid port, thereby enclosing the bladder within the shell preventing bladder failure by extrusion through the discharge port.

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Applications with corrosive environments can require shells furnished with the internal and/or external coatings, in addition to being manufactured from stainless steel as required by API 614. Figure 19 shows a cutaway view of bladder type accumulator.

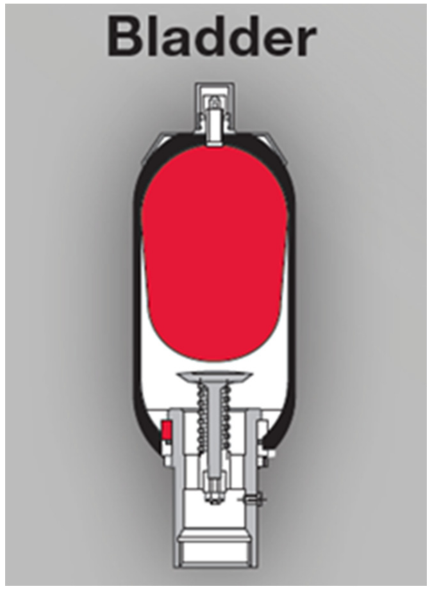

Figure 19: Cross-section of Bladder Type Accumulator (Image courtesy of HYDAC)

Generally, mounting of the accumulator bladder is vertical with the gas port on the top. However, some bladder accumulators incorporate pleated type bladders, which enable a less preferred horizontal configuration, if required. Where high discharge rates or stored volumes are required, exceeding available accumulator sizes, whereby multiple accumulators can be connected in parallel. Figures 20 and 21 show parallel accumulator in horizontal and vertical configurations, respectively.

Figure 20: Multiple Parallel Horizontal Accumulators (Image courtesy of G.J. Oliver, Inc.)

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Figure 21: Multiple Parallel Vertical Accumulators (Image courtesy of G.J. Oliver, Inc.)

Table 2 provides some typical values for bladder type accumulators.

Table 2: Bladder Type Accumulator (Courtesy of HYDAC)

Nominal Volume	Maximum Allowable Working Pressure MPa (psi)	Pressure Ratio	Flowrate lpm (gal/min)	Preferred Installation
0.95 lt. – 454 lt. (1 qt. - 120 gal)	1.03 – 34.47 (150 – 5,000) up to 69 (10,000)	4:1	up to 1,817 (480)	Vertical

The most common bladder material used for mineral oil service is Acrylonitrile Butadiene Rubber (NBR) or Nitrile rubber also called Buna-N. Low temperature Buna-N is also used where very cold temperatures are expected and Ethylene Oxide epichlorohydrin rubber is used if a large temperature range must be dealt with. Fluorine rubber (FKM) is another suitable bladder material for mineral and synthetic oil services.

For the applications of providing auxiliary oil to cover pump transfer (switching) and/or to provide oil for hydraulic servo transient flow conditions, accumulator sizing is performed utilizing adiabatic gas compression and expansion due to the short cycle times. Equation 5 can be used for determining the required accumulator vessel volume or the amount of available stored oil for a given accumulator size based on the operating conditions and pre-charge pressures.

$$V_1 = \frac{V_x\left(\frac{P_1}{P_0}\right)^{\left(\frac{1}{n}\right)}}{\left(1-\left(\frac{P_1}{P_2}\right)^{\left(\frac{1}{n}\right)}\right)} \quad or \quad V_x = \frac{V1\left(1-\frac{P_1}{P_2}\right)^{\left(\frac{1}{n}\right)}}{\left(\frac{P_1}{P_0}\right)^{\left(\frac{1}{n}\right)}} \tag{5}$$

where:

V1 = Required Accumulator Vessel Volume, *liter (In3)*

Vx = Store Volume Required, *liter (In3)*

P0 = Gas Pre-charge Pressure, *kPa absolute (psia)*

P1 = Minimum Discharge Pressure @ End of Discharge Cycle, *kPa absolute (psia)*

P2 = Maximum (Normal) Operating Pressure, *kPa absolute (psia)*

n = Ratio of Gas Specific Heat [For Nitrogen in Adiabatic Service with Less than 6 second discharge cycle time and operating pressures less than 2,758 *kPa* (400 *psig*), n= 1.4]

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The example below illustrates how an accumulator is sized based on the provided inputs. Actual steps and methods can vary depending on applications and designs. In this example, the oil system accumulator is sized for four-second flows of a 379 l/min (100 gal/min) system, which requires a stored volume of 25 liters [96.7 Gallons (1,547.7 In3)]. Atmospheric pressure is assumed as 101.3 kPa absolute (14.7 psia).

Example service conditions and requirements:

P2 =	963 kPa absolute (139.7 psia)	Maximum (normal) system pressure
P1 =	722 kPa absolute (104.7 psia)	Minimum system pressure
P0 =	653 kPa absolute (94.7 psia)	Gas pre-charge pressure
Vx =	25.4 liters [6.7 Gallons (1,547.7 In3)]	Required oil volume
T = 15 °C to 35 °C (59 ºF to 95 ºF)		Ambient temperature range
D = 4 Seconds		Duty cycle

These conditions result in a minimum of 146.5 liter (38.7 gallon or 8,939.7 In3) capacity accumulator to provide the required 25.4 liters (6.7 gallons) of oil between the given pressures and temperatures. If the capacity falls between a manufacturer's two standard sizes, the next larger size is selected for the application. In this application, a standard 150 liter (40-Gallon) accumulator would be utilized.

When determining pre-charge pressure, it is also important to consider changes in ambient temperature, which impacts the accumulator's stored oil volume due as a result of the gas volume change with temperature based on Charles's Law (V1/T1 = V2/T2) where temperatures are minimum, and maximum expressed in absolute temperature units. Keep in mind that during normal system steady state operation, the accumulator pressure is maintained constant by the oil system and hence the gas volume changes with temperature.

For the example given above, consider the accumulator as pre-charged at the 15ºC (59ºF) ambient temperature. At an elevated ambient temperature of 35ºC (95ºF) there would be a 6.7% expansion of the gas volume, reducing the stored oil volume from 25.3 to 23.6 liters (25.3 liters – 25.3 x 0.067) [6.7 to 6.25 gallons (6.7 Gal – 6.7 x 0.067)]. As a result, at elevated temperature the accumulator provides less than the original four seconds of oil flow. Therefore, accumulator sizing and pre-charge determination should take into consideration this temperature influence.

In the performance graph shown in Figure 22, the colored lines represent the accumulator performance curves when in operation at the minimum and maximum ambient temperatures based on the pre-charging at the minimum temperature. This difference shows the effect of temperature change on the stored oil volume supply duration.

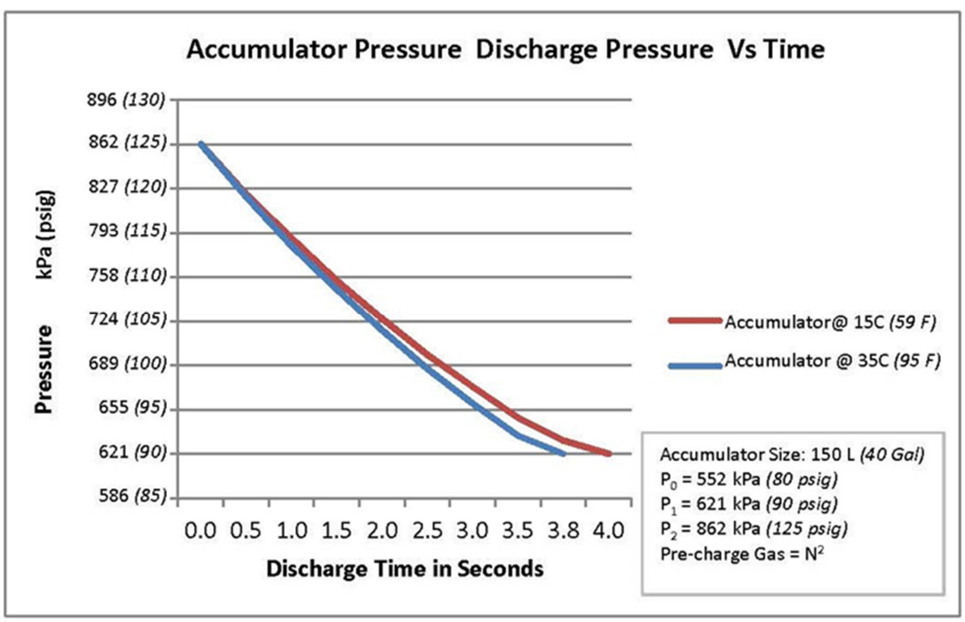

Figure 22: Accumulator System Pressure – Time Curve (Image courtesy of G.J. Oliver)

If the accumulator is to be installed in North America, it is required to be designed to the ASME Boiler and Pressure Vessel Code and be National Board registered or Canadian Registration Number (CRN) stamped.

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Pressure vessel designs for other countries, such as the Pressure Equipment Directive (PED) can vary depending on the design code used. Accumulators outside of these ranges are also available but generally considered special designs.

After the accumulator is selected, the designer can decide on the accessories needed to complete the specified scope.

- The first consideration is how the unit is to be mounted. Here, there are as many different options as there are applications. The accumulator manufacturer's catalogue offers several options and provides a guide as to what is appropriate for a given accumulator.
- Next is to look at the fluid port connection. Does the application require a safety shut-off valve to isolate the unit from the rest of the system? Usually this is accomplished via a single block that incorporates a ball valve, relief valve, and drain valve.
- Finally, the method to charge the accumulator is selected. Back-up nitrogen bottles and different types of charging and gauging kits are available from the accumulator manufacturers for the plant maintenance staff to accomplish this task. An important consideration in this context is the location of the installation and the type of standard connections (SI or inch-series) used for gas source bottles. The right connection must be selected.

Gradual loss of pre-charge pressure during use is a common phenomenon in all types of accumulators. The rate of loss of pre-charge pressure depends on duty cycle, operating temperatures, and bladder's elastomer formulation. Pre-charge pressure should be initially checked at three-month intervals to establish a benchmark for gas loss after which the frequency can be adjusted to an interval to match the expected loss rate.

Accumulators are usually fitted with a permanent pressure gauge connection on the gas side for checking the nitrogen pre-charge pressure. Pre-charge pressure should be also verified prior to starting the lubrication and control-oil system for the served equipment train that was out of service for maintenance or due to plant outage. In bladder type accumulator, pre-charge pressure is checked when the bladder has fully expanded (oil is drained from the accumulator).

To perform the pre-charge pressure check, with the oil system running, the accumulator must be blocked in and fully drained down via the orificed return drain. Once all of the oil is drained, the drain valve is kept open and the pre-charge pressure checked and adjusted, as necessary. If there is no (zero) pre-charge pressure and adding nitrogen does not pressurize, the bladder is leaking and must be replaced.

After the pre-charge pressure is checked and adjusted, the pre-charge gas valve is locked in, the drain valve closed, and the accumulator block valve is cracked open.

NOTE: Opening the accumulator block valve too far can divert excessive flow to the accumulator causing a low pressure condition to occur at the unit. Once the pre-charge pressure reads the same as the system oil pressure, the accumulator is fully charged, and the block valve can be fully opened.

OIL CONSOLE ASSEMBLY

Generally, special purpose lube oil systems are mounted on a single baseplate, which is separate from the main equipment skid and is referred to as an oil console. In some instances, where space constraints exist, such as offshore applications, the complete oil system may be incorporated into the main equipment skid assembly. Further, utilizing reservoir top mounted vertical submersed pumps, system equipment and piping significantly reduces the required oil system space but does result in reduced maintenance accessibility. This is especially true for the pumps which must be pulled out of the reservoir for any maintenance activity.

Normally, the relative locations of the oil console and the served equipment skid will determine the preferred locations for the oil supply and return connections to facilitate the best interconnecting piping layouts.

Location of steam and cooling water utilities will often have significant influence on the oil system layout configuration. Additionally, consideration must be given to available maintenance access areas for servicing the oil console systems, ensuring that the adequate areas for shell and tube cooler bundle pull and reservoir heater removal exists.

The equipment layout within the oil console must be designed with operability and maintenance in mind. Adequate space around pumps and drivers must be incorporated. Block and bypass valves and adequate flange breaks are required to enable servicing of equipment while the system is on-line.

Instrument gauge board panels must be located in an area easily accessible during operator rounds.

The following images (Figures 24 through 26) show a system consisting of motor driven main and standby horizontal positive displacement pumps, duplex shell, and tube heat exchangers with cooling water manifold piping, duplex oil filters, instrument gauge board, and blow assisted reservoir vent mist eliminator.

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Figure 24: Typical Special Purpose Lube Oil Console (Image courtesy of G.J. Oliver, Inc.)

Figure 25: Typical Special Purpose Lube Oil Console (Image courtesy of G.J. Oliver, Inc.)

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Figure 26: Typical Special Purpose Lube Oil Console (Image courtesy of G.J. Oliver, Inc.)

Figure 27 is an image of a replacement upgrade lube and control oil system that incorporates a steam turbine main pump driver with a motor driven standby pump. Another noteworthy feature is this system incorporates a “low profile” reservoir, due to a small elevation difference between the oil console system and the main equipment. To ensure a continuous slope of the oil drain return line from the main equipment, which enters the oil reservoir above the maximum oil level, the reservoir had to be designed with a low height and larger footprint surface area.

This oil console also includes an accumulator, which is visible in the upper left hand part of the image.

Figure 27: Steam Turbine Driven Main Pump Lube/Control Oil Console (Image courtesy of G.J. Oliver, Inc.)

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Structural Steel Welding and Certification
Components such as ladders, handrails and walkways that are attached to oil system baseplate are considered as structural elements by regulatory and certification authorities in certain jurisdictions. It is prudent that prior to award of the purchase order, the purchaser reviews the material of these structural steel components to confirm they meet the governing specification requirements. Similarly, welding procedures, materials, quality standards, connection design, fabrication, and testing, welder qualifications, and Non-Destructive examination are required to comply with the local authentication and certification standards. These checks, if overlooked during bid evaluation or proposal conditioning, can often result in schedule delays and cost adders.

LUBE OIL RUNDOWN TANK

API 614 specifies that lube oil rundown tanks shall be sized for no fewer than three minutes of normal operating lube oil flow. It is designed to ensure appropriate minimum oil static head pressure at the near empty condition to ensure adequate lubrication of the bearings during coast down during a low-low lube oil shutdown due to loss of the oil supply system. However, rundown tank elevation and orientation must be such so the lube oil static head pressure, when the tank is completely full, does not exceed the low lube oil pressure shutdown setting.

This requirement can be achieved by locating the tank above the compressor centerline, such that the top of the tank (full) will provide the static head that is just under the low-low lube oil pressure trip and in the near empty condition. This is when it has supplied most of the oil flow during the specified coast-down period. The remaining static head provided by oil will be enough to provide lubrication until the served equipment train comes to rest.

Rundown tanks can be oriented vertically or horizontally depending on their size and the space available around at the location of installation. The purchaser is required to provide information about available space and mounting locations at the necessary elevation above the unit. In cold climate installations, heat tracing and thermal insulation may be required on rundown tanks for maintaining oil temperature during cold ambient conditions.

Overhead lube oil rundown tanks can be an atmospheric type or a pressurized flow through type as depicted in the diagrams shown in Figure 28 and Figure 29, respectively.

The atmospheric type of rundown tanks are basically static oil tanks which have a continuous small flow [(6 – 15 l/min (1.5 – 4 gal/min)] diverted through them to help keep the oil warm. The pressurized tanks are connected in series with the lube oil supply piping requiring all of the lube oil supply flow through the tank.

Atmospheric lube oil tanks obtain oil from an orifice feed of a few gal/min. During startup, filling of the tanks can be accelerated by opening a quick fill bypass valve until full spill over oil flow is visible in the sight flow glass.

One design aspect, which should not be overlooked, is the atmospheric rundown tank’s overflow line. It should be sized for full-open quick fill valve scenario to prevent oil spillage from rundown tank vent should the quick fill valve be inadvertently kept open instead of closed after filling the rundown tank.

A level monitoring instrument is included with atmospheric tanks to assisting in monitoring the filling of the tank and to provide a unit start permissive and low level alarm.

Pressurized rundown tanks incorporate floating ball check valves, which allow all of the air to be expelled during filling. Once the tanks are full of oil, the ball float closes, and the tanks become pressurized to the lube oil supply pressure, less the head pressure associated with the tank elevation with respect to the served equipment shaft centerline.

Since normal unit lube oil supply header pressure cannot be achieved until the pressurized tank is completely full, a level monitoring instrument for starting permissive is not required.

The rundown capacity of the oil reservoir must include the volume of the lube oil rundown tank and associated interconnecting piping.

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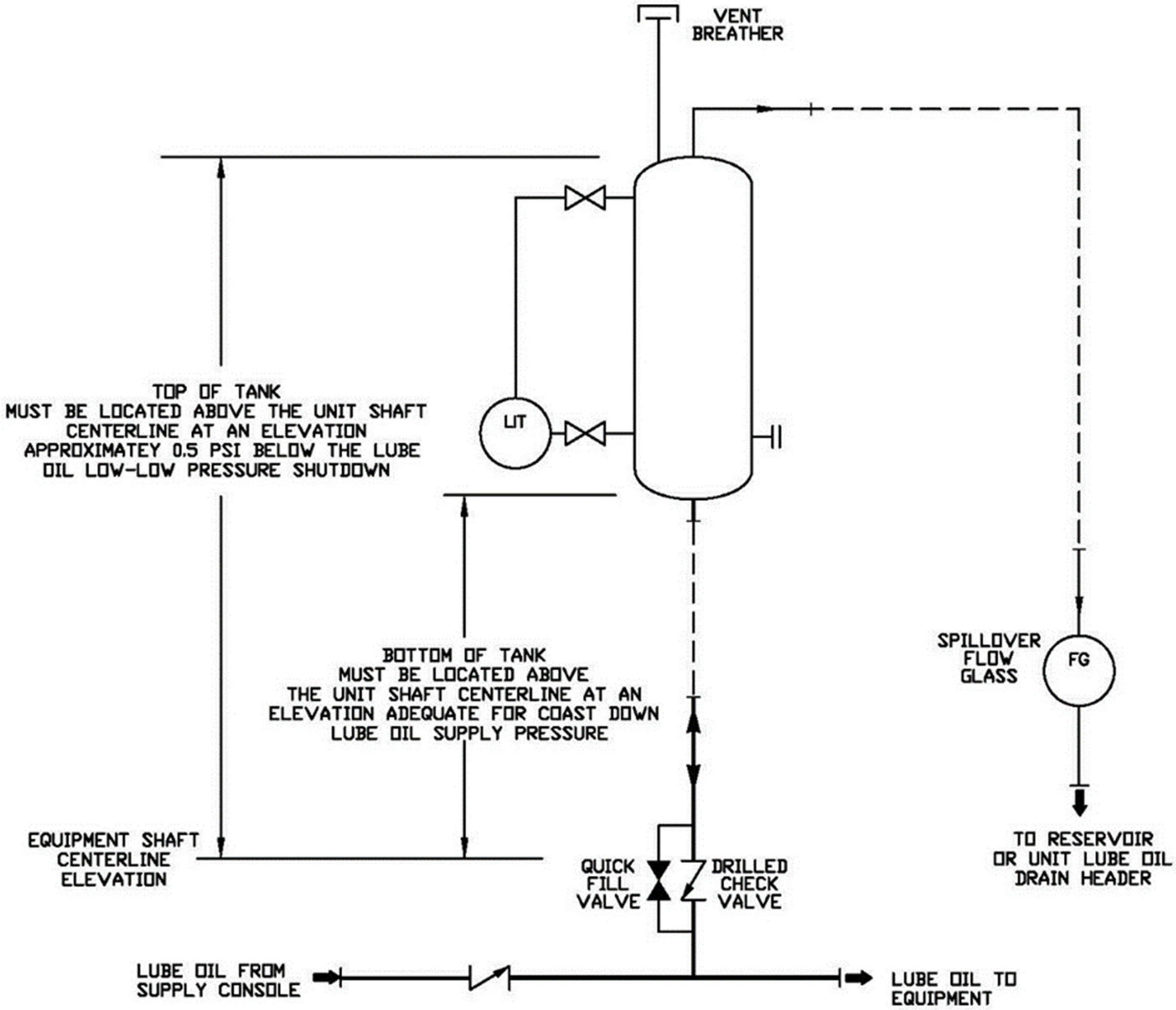

Figure 28: Schematic of Overhead Atmospheric Lube Oil Rundown Tank (Image courtesy of G.J. Oliver, Inc.)

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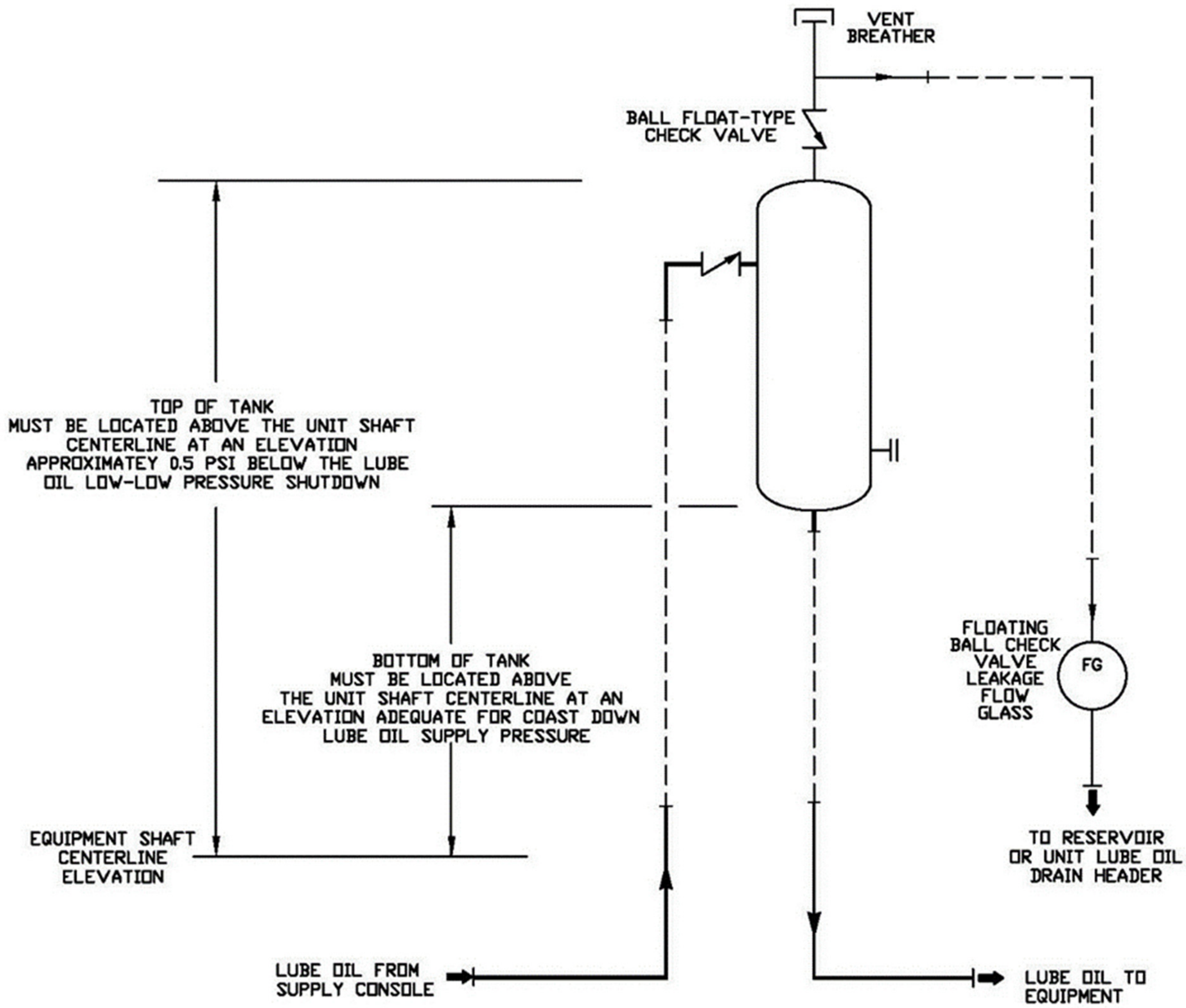

Figure 29: Schematic of Overhead Pressure Lube Oil Rundown Tank (Image courtesy of G.J.Oliver, Inc.)

The following image (Figure 30) is for a horizontal mounted oriented atmospheric type of lube oil rundown tank. The tank incorporates separate vent and overflow (spill over) connections along with a 300 mm (12 in.) blind flanged inspection cover. A differential pressure type level transmitters is included for level monitoring.

In addition, the tank will be field insulated with blanket type insulation after it is installed and hence is supplied with insulation studs for securing the blanket after it is installed.

The tank image shown in Figure 31, is a vertical pressurized lube oil rundown tank, which is horizontally secured to a wood skid for shipment.

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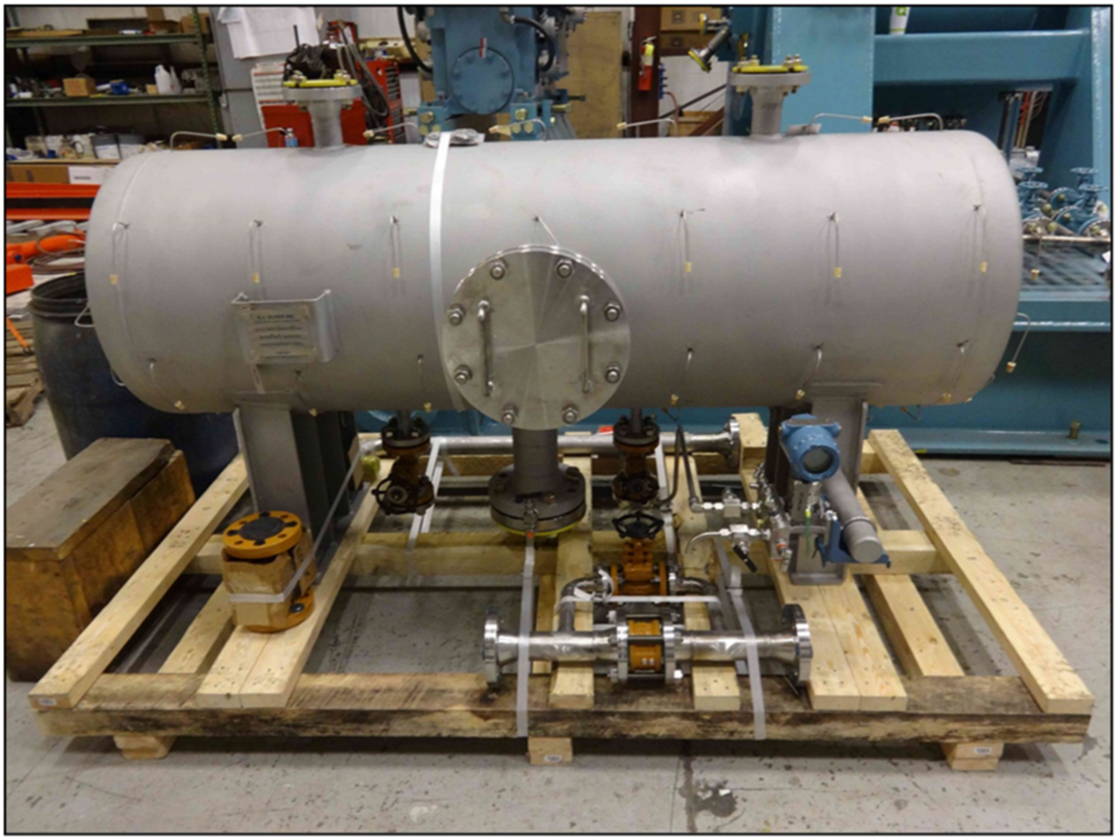

Figure 30: Horizontal Atmospheric Lube Oil Rundown Tank (Image courtesy of G.J. Oliver, Inc.)

Figure 31: Vertical Pressurized Lube Oil Rundown Tank (Image courtesy of G.J. Oliver, Inc.)

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OIL MIST ELIMINATORS

Mist eliminators are, in principle, coalescers that remove visible oil vapor from the air stream of vents in lubricating oil systems. It is a continuous process in which small aerosols come in contact with the fibers in the mist eliminator's filter media, combining with other collected aerosols and growing to emerge as a droplet on the downstream surface of the media, which is capable of being gravitationally drained away. The mist eliminator prevents oil mist from contaminating air or soil, helping facilities comply with environmental regulations. It also makes for a cleaner, safer work environment by eliminating oily residue build-up on the floors, enclosures, and stairwells, while improving indoor and outdoor air quality and reducing fire hazards. Oil mist eliminators also help to reduce make-up cost by returning oil to the reservoir that would normally be lost during the venting process.

Different configurations of mist eliminators are available in the market. They include static type, blower or fan assisted type and mist eliminator with heat exchanger. Blowers / fans can be single or duplex (2 x 100% capacity) and of constant-speed or variable-speed. Mist eliminators are designed to eliminate oil mist with an efficiency of 99.97% removal of all airborne contaminant regardless of particle size. Majority of oil mist particles are less than 1 micron with about 60% to 80% less than 0.5 micron.

Mist eliminators can maintain a vacuum in the oil reservoir in the range of 2" H2O to 4" H2O. Pressure monitoring can be provided on the oil reservoir for this purpose. The use of a mist eliminator should be reviewed if the environment around the installation is humid, dusty, corrosive, or contains particulates. The minimum data required to size a mist eliminator includes vent flow rate from the reservoir, allowable back pressure, or pressure drop, the size of mist eliminator connection on the reservoir, the oil temperature range, and the power rating of the served equipment train.

The vent flow of the reservoir can include combined flow streams from equipment outboard separation seal or bearing buffer air or nitrogen flows, ingress air flows from bearing, and gear box vent breathers, coupling windage, and nitrogen purge flows.

Static or coalescer-only type is suitable for use where back pressure on an oil system is not a concern. It can be sized large enough to meet pressure-drop requirements. It also can be used where an existing blower is available to assist removal of oil mist.

Blower assisted type eliminator is appropriate for oil systems that cannot withstand back pressure or where back pressure may not be desirable, or where a slight negative pressure is desired on the equipment drain system. Blower assisted types can be installed with an airflow-restriction device to pull vacuum or designed to maintain atmospheric pressure condition in the oil system reservoir.

Mist eliminator with heat exchanger is often used where the oil can reach very high temperatures 93 ºC (200 °F) and greater. By cooling the air stream, the oil mist becomes easier to coalesce and therefore more efficient. Figure 32 show pictures of these three types of mist eliminators.

Figure 32: Static or Passive Mist Eliminator, Mist Eliminator with Heat Exchanger and Blower assisted Mist Eliminator (Image courtesy of The Hilliard Corporation)

OIL CONDITIONERS

Water or moisture contamination in oil is due to free water homogenized in an either stable or unstable state (water in emulsion) or in a dissolved or absorbed state up to the saturation point (water in solution). Oil purifiers have been primarily used with lube oil systems that serve steam turbine driven machinery strings. Pleated cartridge elements can remove water to oil saturation level.

Coalescer/separators remove free and emulsified water to saturation level and offer a moderate capital cost but low operational cost

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solution. Figure 33 shows baseplate mounted coalescers / oil conditioners. Typical flow schematic of coalescer/separator is presented in Figure 34. Portable units complete with particulate filter and coalescing filter are also commonly used in the industry. The other types of oil separators are vacuum dehydrators and centrifugal separators. Vacuum dehydrators can remove free, emulsified, and dissolved water to below oil saturation level.

Figure 33: Coalescing Separators (Image courtesy of The Hilliard Corporation)

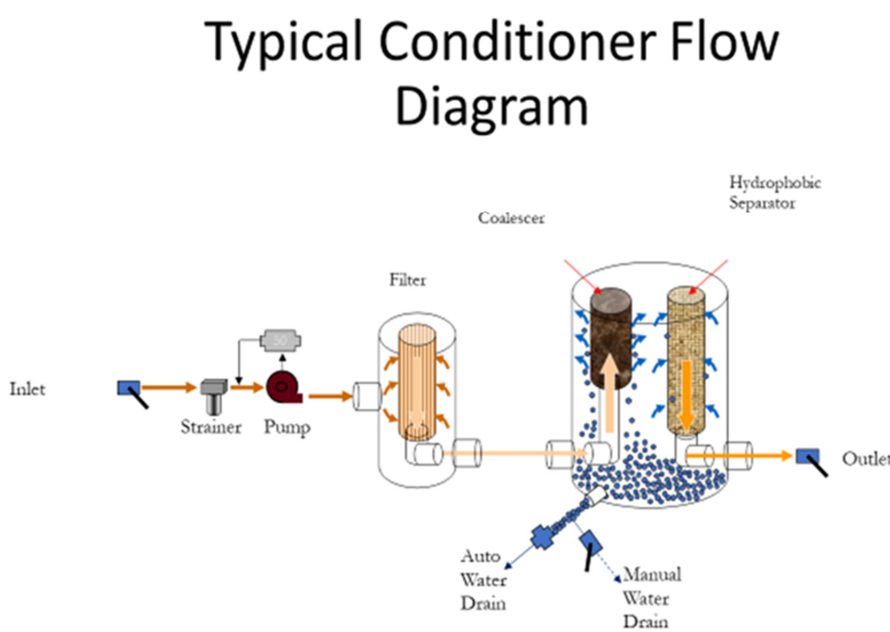

Figure 34: Flow Schematic of Oil Conditioner or Coalescing Separator (Image courtesy of The Hilliard Corporation)

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Oil conditioners are sized to deliver a small percentage, for example, 0.5% of total oil volume in circulation. In this case, an oil conditioner with 38 lpm (10 gal/min) flow rate will be selected for a 7,571 lt. (2,000 gallon) oil reservoir. Figure 35 presents typical performance of this system.

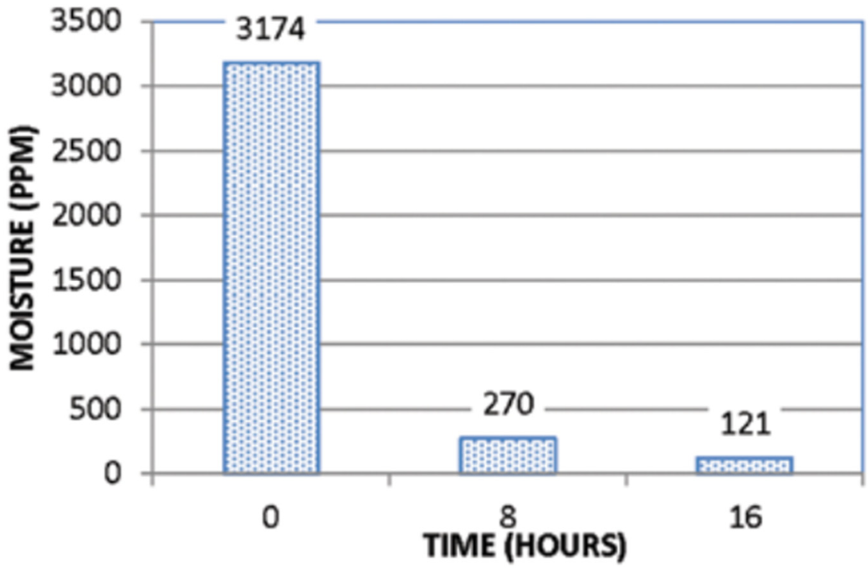

Figure 35: Moisture removal performance of a 10 gal/min oil conditioner
(Image courtesy of The Hilliard Corporation)

HAZARD AND OPERABILITY STUDY OF LUBE OIL SYSTEMS

Lube oil system is one of the nodes included and analysed in the Hazard and Operability Study (HAZOP) of centrifugal compressor trains. Engineering and design deficiencies that can create potential hazards and operability issues are identified and reviewed in a HAZOP session. The operational controls and methods are examined for each hazard to determine if they are adequate. A typical HAZOP worksheet includes deviations, causes for the deviations, their consequences considering no safeguards, severity, barriers considering the available safeguards, and recommended actions with responsible entity assigned to each action. Severity classification can be from one to five (or different) and rankings are allocated by HAZOP team to people, property, environment, and other impacts, if applicable.

A representative example of lube oil system HAZOP is shown in the table 3. This oil system serves a centrifugal compressor-gear-motor train. Note that the rankings increase with severity of the consequences and as required by this specific project, are based on the perceived effects of severity on people, assets (property, equipment), environment, and reputation. For example, a cause, or consequence that does not result in injury, has no impact on health, does not cause damage to asset and does not affect reputation is assigned zero severity. Consequences that can entail multiple fatalities, damage in millions of dollars, significant impact on environment (for example, release of process fluids), and major impact on the reputation due to attention given in media, and regulatory disciplinary actions are assigned severity ranking five.

The actual deviations analysed during HAZOP of the job lube oil system for this machinery train included the following situations:

- Low flow or no flow rate of lube oil: blockage in pump strainer; failure of oil pump; blockage in oil filter; inadvertent closure of pump discharge block valves; and isolation valves around pressure or temperature regulators.
- High flow rate of lube oil: malfunction of low oil pressure transmitter, incorrect valve operation.
- Reverse flow of lube oil: backflow.
- Lube oil flow in the wrong direction: drains left open; leaks from joints; rundown tank manual fill valve inadvertently kept open.
- High and low pressures: similar to high flow rate or low/no flow rate.
- High and low temperatures in oil circuit and cooling medium loop: high or low ambient temperatures; failure of heater on the oil; failure or malfunction of oil temperature control valve; obstruction in cooling medium flow to oil cooler.

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- Low level: similar to lube oil flow in the wrong direction.
- Start-up – incorrect valve operation during manual starting of the unit
- Emergency shutdown – process upset, fire, and gas related emergency

Table 3: Centrifugal Compressor Lube Oil System HAZOP

<table>
<tr><th>Deviation</th><th>Cause</th><th>Consequence</th><th colspan="4">Severity</th><th>Barriers</th></tr>
<tr><td rowspan="10">Low/No Flow</td><td rowspan="10">Partial blockage of lube oil Pump A suction strainer</td><td>Potential cavitation of the lube oil pump leading to its damage (this could be after a few hours).</td><td>0</td><td>1</td><td>0</td><td>0</td><td>No safeguards</td></tr>
<tr><td rowspan="3">Worst-case insufficient lube oil flow to the bearings of the electric motor, gear box, and compressor leading to mechanical damage of the bearings.</td><td rowspan="3">0</td><td rowspan="3">2</td><td rowspan="3">0</td><td rowspan="3">0</td><td>Pressure Transmitter on pump discharge will start the standby lube oil pump.</td></tr>
<tr><td>Pressure Transmitter on oil supply header will start the standby lube oil pump.</td></tr>
<tr><td>Pressure Transmitters A/B/C (for 2-o-o-3 voting logic) on the lube oil header will trip the compressor motor (breaks the power supply), closes the inlet and outlet valves on the compressor and oil would automatically flow from the rundown tank to provide coast-down lubrication and allow safe shutdown of the train.</td></tr>
<tr><td rowspan="3">Bearing damage could lead to other item damage and flying fragments of material (couplings), which could result in permanent injury to any operators who could be in the area.</td><td rowspan="3">4</td><td rowspan="3">3</td><td rowspan="3">0</td><td rowspan="3">3</td><td>Radial Vibration trip on the compressor will trip the compressor motor, close the inlet and outlet valves on the compressor, and oil would automatically flow from the rundown tank to provide coast-down lubrication and allow safe shutdown of the train.</td></tr>
<tr><td>Radial Vibration trip on the gearbox will trip the compressor motor (breaks the power supply), closes the inlet and outlet valves on the compressor and oil would automatically flow from the rundown tank to provide coast-down lubrication and allow safe shutdown of the train.</td></tr>
<tr><td>Radial Vibration trip on the gearbox will trip the compressor motor (breaks the power supply), closes the inlet, and outlet valves on the compressor, and oil would automatically flow from the rundown tank to provide coast-down lubrication and allow safe shutdown of the train.</td></tr>
<tr><td rowspan="3">With reference to the above consequence, if there was bearing damage it would lead to high vibrations of the motor, gear box and compressor leading to their physical damage. In addition, there could be a loss of containment in the compressor, resulting in release of process gas. Potential fire and/or</td><td rowspan="3">4</td><td rowspan="3">4</td><td rowspan="3">2</td><td rowspan="3">3</td><td>Pressure Transmitter on pump discharge will start the standby lube oil pump.</td></tr>
<tr><td>Pressure Transmitter on oil supply header will start the standby lube oil pump.</td></tr>
<tr><td>Pressure Transmitters A/B/C (for 2-o-o-3 voting logic) on the lube oil header will trip the compressor motor (breaks the power supply), closes the inlet and outlet valves on the compressor and oil would automatically flow from the rundown tank to provide coast-down</td></tr>
</table>

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Deviation	Cause	Consequence	Severity				Barriers
		explosion. Potential asset damage. Potential environmental impact and potential reputation impact.					lubrication and allow safe shutdown of the train.
							Radial Vibration trip on the compressor will trip the compressor motor (breaks the power supply), closes the inlet and outlet valves on the compressor, and oil would automatically flow from the rundown tank to provide coast-down lubrication and allow safe shutdown of the train.

CONCLUSION

Lubricating oil systems are vital to operation, long-term reliability, and availability of turbomachinery trains. This tutorial focusses on the selection and sizing considerations for components and elements that constitute the pressurized lubricating oil system for a modern turbomachinery string. The design principles explained in this text are commonly applied to oil systems that serve different types of driven and driving equipment. Often overlooked authentication, and certification requirements for oil console structural steel are presented for awareness of the purchasers and facility owners. The section on Hazard and Operability Study (HAZOP) of oil systems addresses common deviations, causes for the deviations, their consequences considering no safeguards, severity, barriers considering the available safeguards, and recommended actions.

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HYDAC Accumulators Catalogue PN#02068195 / 1.15 / ACU1102-1326

HYDAC Literature EN30000-5-06-18.

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