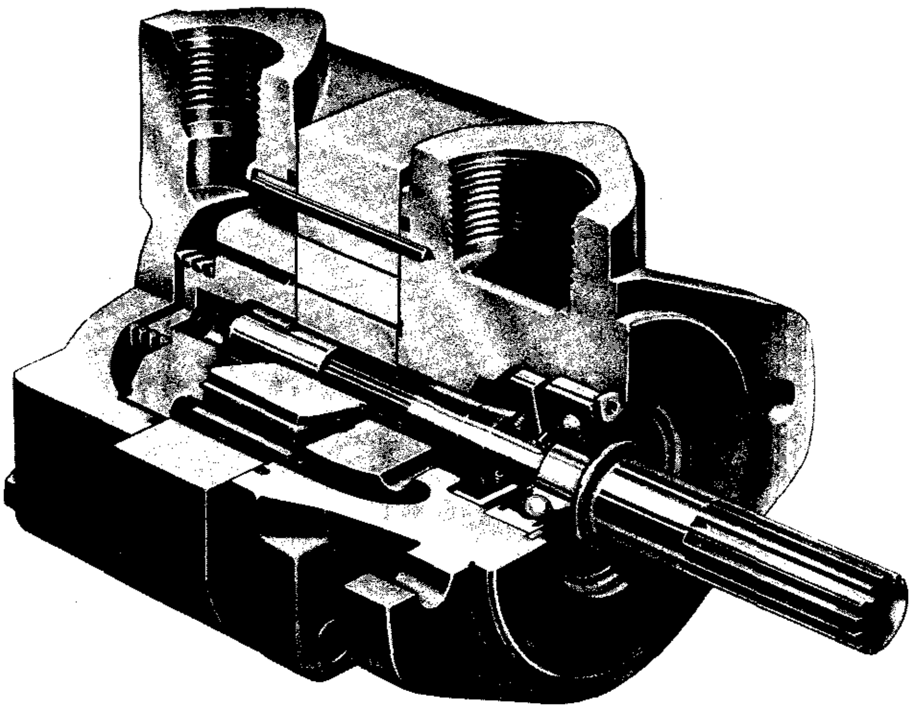
VICKERS®

A TRINOVA COMPANY

Overhaul Manual

Vane Pumps

V10, V20, V10F, V10P,
V20F, V20P Series



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I-3143-S

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Section I - INTRODUCTION

A. PURPOSE OF MANUAL

This manual has been prepared to assist the users of Vickers balanced vane type hydraulic single pumps in properly installing, maintaining and repairing their units. In the sections which follow, the single pumps are described in detail, their theory of operation is discussed and instructions are given for their proper installation, maintenance and overhaul.

The general series of models covered are V10, V20, V10F, V10P, V20F, and V20P. The information given applies to the latest design configurations listed in Table 1. Earlier designs are covered only insofar as they are similar to the present equipment.

B. GENERAL INFORMATION

1. Related Publications - Service parts information and installation dimensions are not contained in this manual. The parts catalogs and installation drawings listed in Table 1 are available from any Vickers Application Engineering office, or from:

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2. Model Codes - There are many variations within each basic model series, which are covered by variables in the model code. Table 2 is a complete breakdown of the code covering these units. Service inquiries should always include the complete unit model number, which is stamped on the pump cover.

TABLE 1.
PARTS CATALOGS AND INSTALLATION DRAWINGS

<table>
<tr><th>MODEL SERIES</th><th>PARTS DRAWING</th><th>INSTALLATION DRAWING</th></tr>
<tr><td>V10
V10F
V10P</td><td>M-2005-S</td><td>MB-53</td></tr>
<tr><td>V20
V20F
V20P</td><td>M-2004-S</td><td>MB-53</td></tr>
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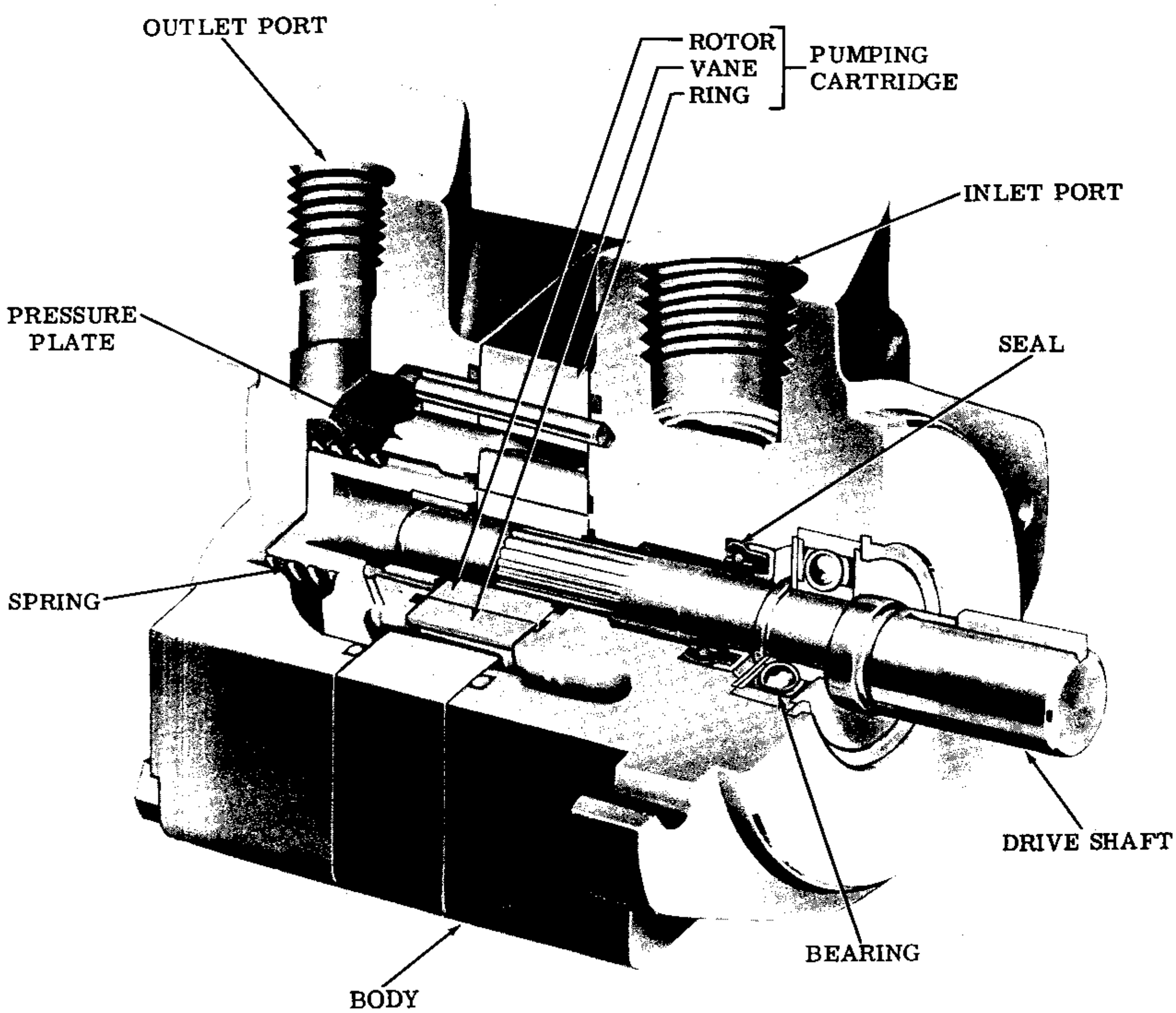


Figure 1

MODEL CODE BREAKDOWN

V10-(P) * * * * - * * (*)-(*) 20- * * * L

VANE PUMP

SERIES
10 OR 20

F - FLOW CONTROL COVER
P - PRIORITY VALVE COVER
OMITTED - STD. COVER

MOUNTING
1-2 BOLT FLANGE (SAE "A" SIZE)
2-FOOT BRACKET AT 12 O'CLOCK (VIEWED FROM THE SHAFT END)
23-FOOT BRACKET AT 3 O'CLOCK
26-FOOT BRACKET AT 6 O'CLOCK
29-FOOT BRACKET AT 9 O'CLOCK

INLET PORT		
CODE	MODEL	SIZE
E	V20	1. 5"Dia. -2 BOLT FLG.
F	V20	1. 156"Dia. -2 BOLT FLG.
H	V10	1" O. D. TUBE CONN.
K	V10	1. 3125"-12 UN2B THD.
P	V10	1" N. P. T. F.
P	V20	1. 25" N. P. T. F.
R	V20	1" N. P. T. F.
S	V10	1. 3125"-12 STR. THD.
S	V20	1. 625"-12 UN2B THD.
T	V10	1. 1875"-12 STR. THD.

RING CAPACITY-1200 RPM-100PSI

(V10)	(V20)
1 - 1 USGPM	6 - 6 USGPM
2 - 2 USGPM	7 - 7 USGPM
3 - 3 USGPM	8 - 8 USGPM
4 - 4 USGPM	9 - 9 USGPM
5 - 5 USGPM	11 - 11 USGPM
6 - 6 USGPM	12 - 12 USGPM
7 - 7 USGPM	13 - 13 USGPM

				OUTLET PORT CONNECTIONS		
		FLOW CONTROL COVER		PRIORITY VALVE COVER		
CODE	STANDARD COVER	PRESSURE	TANK	PRIMARY OUTLET	SECONDARY OUTLET	TANK
K	——	——	——	9/16-18 ST. THD. (V10P)	3/4-16 ST. THD. (V10P)	9/16-18 ST. THD. (V10P)
P	1/2 IN. NPT THD. (V10 ONLY)	3/4-16 ST. THD.	1/2 IN. NPT THD.	——	——	——
	3/4 IN. NPT THD. (V20 ONLY)	V10F AND V20F	V10F AND V20F	——	——	——
S	3/4-16 ST. THD.	——	——	——	——	——
	1 1/16-12 ST. THD. (V20 ONLY)	3/4-16 ST. THD. (V20F)	1 1/16-12 ST. THD. (V20F)	——	——	——
T	——	3/4-16 ST. THD. (V10F)	3/4-16 ST. THD. (V10F)	3/4-16 ST. THD. (V20P)	7/8-14 ST. THD. (V20P)	3/4-16 ST. THD. (V20P)

FOR LEFT HAND ROTATION VIEWED FROM SHAFT END

SPECIAL FEATURE SUFFIX

DESIGN & MODIFICATION
V10-10 V20(F)-11
V10(F)-20 V20P-11/12

PRESSURE SETTING
A-250 PSI
B-500 PSI
C-750 PSI
D-1000 PSI
E-1250 PSI
F-1500 PSI
G-1750 PSI
H-2000 PSI
J-2250 PSI
K-2500 PSI

FLOW RATE THROUGH ORIFICE IN COVER
2-2 GPM
3-3 GPM (V10F ONLY)
4-4 GPM
5-5 GPM (V10F ONLY)
6-6 GPM
7-7 GPM (V10F ONLY)
8-8 GPM (V20F ONLY)

PRESSURE PORT POSITIONS
VIEW FROM COVER END
A-OPPOSITE INLET CONNECTION.
B- 90° COUNTERCLOCKWISE FROM INLET CONNECTION
C-INLINE WITH INLET
D-90° CLOCKWISE FROM INLET CONNECTION

SHAFTS
1-STR. KEYED
3-THD.
4-THD.
6-STR. STUB-KEYED
11-SPLINE-9 TOOTH
12-SPLINE-13 TOOTH
15-SPLINE-13 TOOTH
27-TANG
34-THD.
38-SPLINE-11 TOOTH

Section II - DESCRIPTION

A. GENERAL

Pumps in this series are used to develop hydraulic fluid flow for the operation of Mobile and Industrial equipment. The positive displacement pumping cartridges are the rotary vane type with shaft side loads hydraulically balanced. The flow rate depends on the pump size and the speed at which it is driven.

All units are designed so that the direction of rotation, pumping capacity and port positions can be readily changed to suit particular applications.

B. ASSEMBLY AND CONSTRUCTION

The V10 series pump illustrated in the cut-a-way of Figure 1 is representative of all single pumps in this series. The unit consists principally of a ported body, a ported cover and a pumping cartridge. Components of the pumping cartridge are an elliptical cam ring, a slotted rotor splined to fit the drive shaft and twelve vanes fitted to the rotor slots.

The pumping cartridge cam ring is sandwiched between the body and cover. A ball bearing and bushing located in the body and pressure plate respectively support each end of the drive shaft and center the rotor within the cam ring. As the drive shaft is driven by the prime mover, the rotor and vanes generate flow by carrying fluid around the elliptical cam ring contour. Fluid enters the cartridge through the inlet port in the body and is discharged through the pressure plate into the outlet port of the cover.

C. FLOW CONTROL AND RELIEF VALVE

V10 and V20 pumps are available with an integral Flow Control and Relief Valve in the pump cover. This limits the fluid flow in the system to a maximum prescribed rate and prevents excessive pressure build-up. Fluid not required in the system is recirculated to tank.

D. PRIORITY VALVE

V10 and V20 pumps are also available with a priority valve located in the pump cover. The priority valve maintains nearly a constant flow to a primary circuit and diverts the remaining flow to a secondary circuit. Flow going to the secondary circuit is determined by pump delivery. The primary circuit is protected by an integral relief valve but an external relief valve must be provided for the secondary circuit.

E. APPLICATION

Pump ratings in GPM as shown in the model coding are at 1200 RPM and 100 PSI. For ratings at other speeds, methods of installation and other application information, Vickers Application Engineering personnel should be consulted.

Section III - PRINCIPLES of OPERATION

A. PUMPING CARTRIDGE

As mentioned in Section II, fluid flow is developed by the pumping cartridge. The action of the cartridge is illustrated in Figure 2. The rotor is driven within the cam ring by the driveshaft, which is coupled to a power source. As the rotor turns, centrifugal force causes the vanes to follow the elliptical inner surface of the cam ring.

Radial movement of the vanes and turning of the rotor cause the chamber volume between the vanes to increase as the vanes pass the inlet sections of the cam ring. This results in a low pressure condition which allows atmospheric pressure to force fluid into the chambers. (Fluid outside the inlet is at atmospheric pressure or higher.)

This fluid is trapped between the vanes and carried

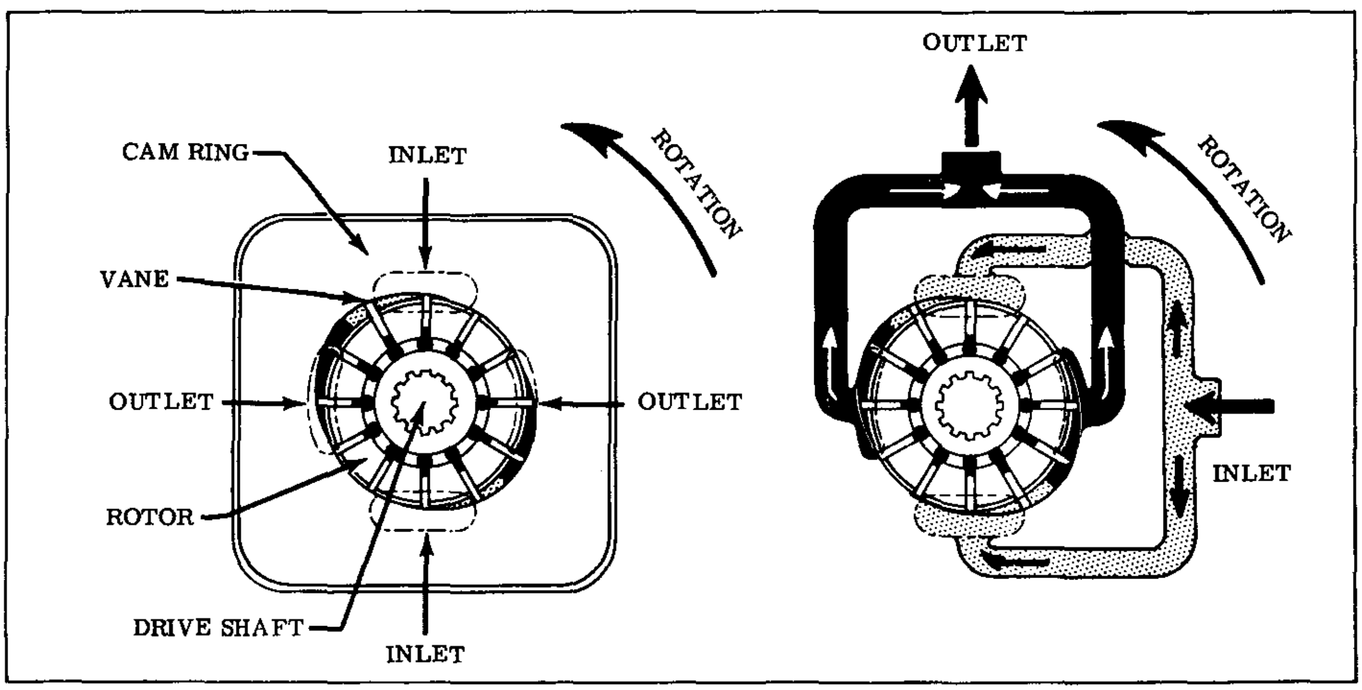


Figure 2

past the large diameter or dwell section of the cam ring. As the outlet section is approached, the cam ring diameter decreases and the fluid is forced out into the system. System pressure is fed under the vanes, assuring their sealing contact against the cam ring during normal operation.

B. HYDRAULIC BALANCE

The pump cam ring is shaped so that the two pumping chambers are formed diametrically opposed. Thus, hydraulic forces which would impose side loads on the shaft are cancelled.

C. PRESSURE PLATE

The pressure plate seals the pumping chamber as shown in Figure 3. A light spring holds the plate against the cartridge until pressure builds up in the system. System pressure is effective against the area at the back of the plate, which is larger than the area exposed to the pumping cartridge. Thus, an unbalanced force holds the plate against the cartridge, sealing the cartridge and providing the proper running clearance for the rotor and vanes.

D. FLOW CONTROL AND RELIEF VALVE

1. Maximum flow to the operating circuit and maximum system pressure are determined by the integral flow control and relief valve in a special outlet cover used on some V10 and V20 pumps. This feature is illustrated pictorially in Figure 4. An orifice in the cover limits maximum flow. A pilot-operated type relief valve shifts to divert excess fluid delivery to tank, thus limiting the system pressure to a predetermined maximum.

2. Figure 4A shows the condition when the total pump delivery can be passed through the orifice.

This condition usually occurs only at low drive speeds. The large spring chamber is connected to the pressure port through an orifice. Pressure plus spring load in this chamber slightly exceeds pressure at the other end of the relief valve spool and the spool remains closed. Pump delivery is blocked from the tank port by the spool land.

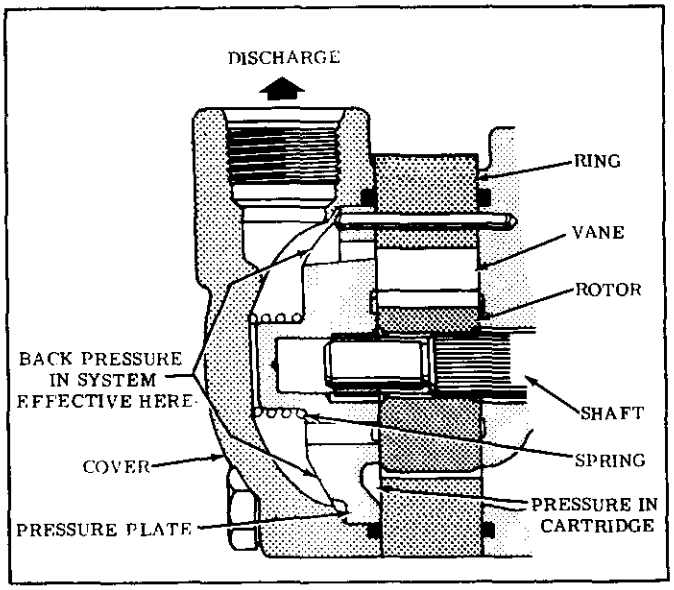


Figure 3

3. When pump delivery is more than the flow rate determined by the orifice plug, pressure builds up across the orifice and forces the spool open against the light spring. Excess fluid is throttled past the spool to the tank port as shown in Figure 4B.

4. If pressure in the system builds up to the relief valve setting (Figure 4C), the pilot poppet is forced off its seat. Fluid in the large spring chamber flows through the spool and out to tank. This flow through the small sensing orifice, causes a pressure drop and prevents pressure in the large spring area from increasing beyond the relief valve setting. As pressure against the right end of the spool starts to exceed the relief valve setting, the pressure differential forces the spool to the left, against the light spring, porting the full pump flow to tank.

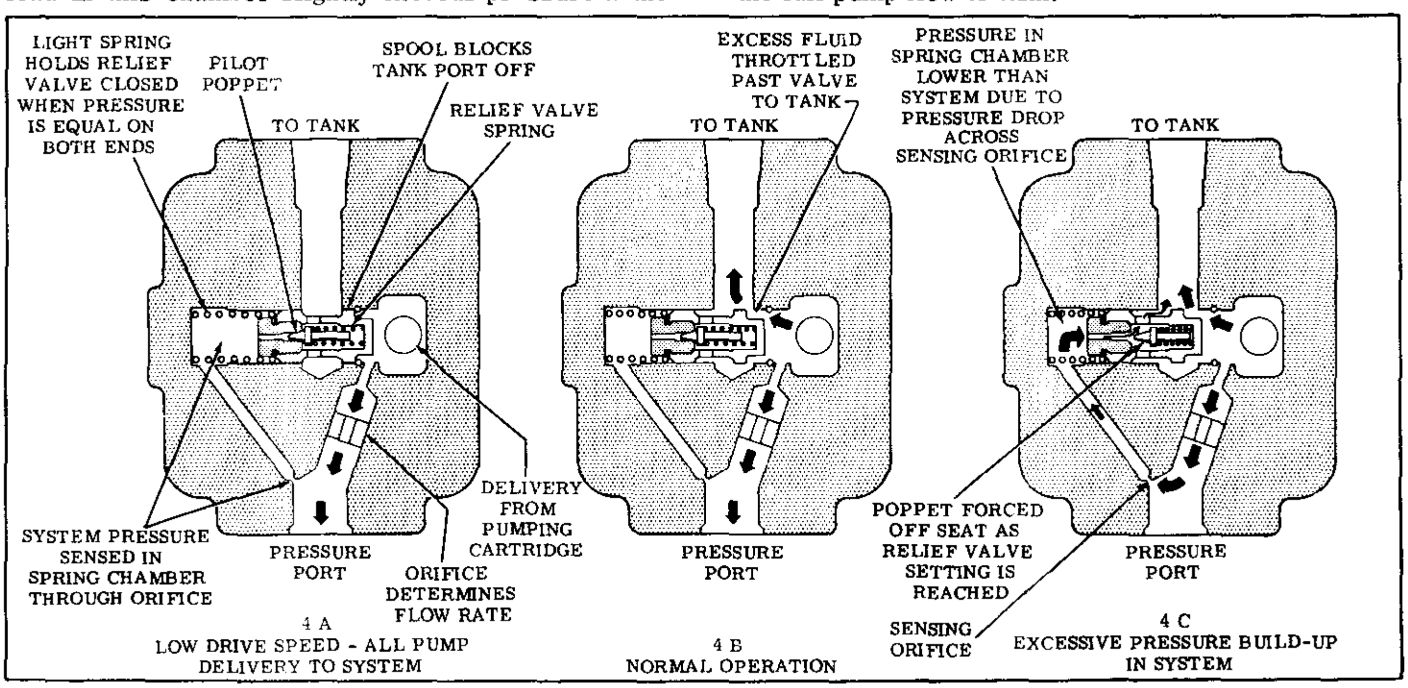


Figure 4

E. PRIORITY VALVE OPERATION

Referring to the V20P Cover Schematic, Figure 5, pressure is sensed in cavities "A", "B" and "C". Primary flow into cavity "A" is restricted by the controlled flow orifice "O". Secondary flow will be zero until the pump flow rate through orifice "O" develops a pressure differential across the control spool.

When pump delivery is increased, pressure builds up in cavities "B" and "C" because of the resistance to flow through orifice "O". This causes the spool to shift toward cavity "A" against the spring. The amount of spool shift is proportional to the pressure differential between cavities "A" and "C".

Flow from the primary port is held to an almost constant volume, as determined by orifice "O", and the metering action of the control spool at area "D". Flow to the secondary port varies with pump delivery. Metering area "E" diverts excess flow to the secondary port.

This single spool design cannot give precisely controlled flow to the primary circuit because of the effects of varying conditions of flows and pressures. For example: If the primary circuit is operating at 1000 PSI and the secondary at 100 PSI, the spool must be metering at "E". However, if primary pressure is 100 PSI and secondary is 1000 PSI, the spool must be metering at "D". As the two systems approach the same pressure, the probability of flow fluctuation increases because the spool may shift between these two metering points.

CAUTION

The pump has a built-in relief valve in the primary circuit. However, an external relief valve must be provided for the secondary circuit to protect the pump.

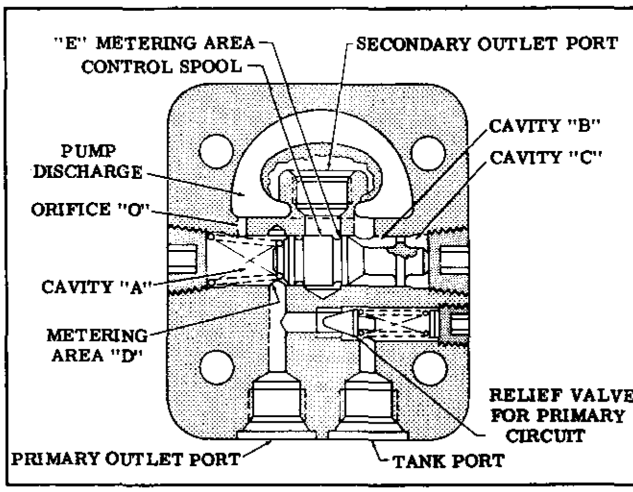


Figure 5

Section IV
INSTALLATION and OPERATING INSTRUCTIONS

A. INSTALLATION DRAWINGS

The installation drawings listed in Table 1 show the correct installation dimensions and port locations.

B. DRIVE CONNECTIONS

CAUTION

Pump shafts are designed to be installed in couplings, pulleys, etc., with a slip fit or very light tap. Pounding can injure the bearings. Shaft tolerances are shown on the pump installation drawings. (See Table 1.)

1. Direct Mounting - A pilot on the pump mounting flange (Figure 6) assures correct mounting and shaft alignment. Make sure the pilot is firmly seated in the accessory pad of the power source. Care should be exercised in tightening the mounting screws to prevent misalignment.

If gaskets are used, they should be installed carefully and should lay flat. Shaft keys and couplings must be properly seated to avoid slipping and possible shearing.

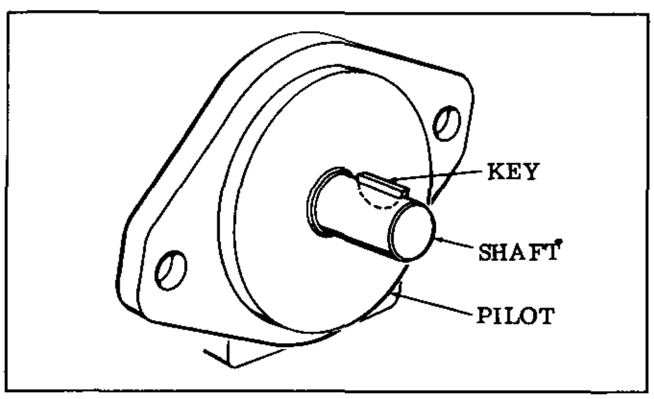


Figure 6

2. Indirect Drive - Chain, spur gear or v-belt pulley drives may also be used with these pumps. Flat belt drives are not recommended because of the possibility of slipping.

To prevent excessive side loads on pump bearings, it is important to check for correct alignment and guard against excessive belt or chain tension.

For best results on indirect drive applications, use the largest permissible pully diameter at the pump and position it close as possible to the pump mounting face. For specific indirect drive application data, contact your Vickers Application Engineer.

C. SHAFT ROTATION

Pumps are normally assembled for right-hand (clockwise) rotation as viewed from the shaft end. A pump made for left-hand rotation is identified by an "L" in the model code (See Table 2).

NOTE

These pumps must be driven in the direction of the arrows cast on the pump ring. If it is desired to change the direction of drive rotation, it is necessary to reverse the ring. (See Section VI-B-D and Figure 10.)

CAUTION

Never drive a pump in the wrong direction of rotation. Seizure may result, necessitating expensive repairs.

D. PIPING AND TUBING

1. All pipes and tubing must be thoroughly cleaned before installation. Recommended methods of cleaning are sand blasting, wire brushing and pickling.

NOTE

For instructions on pickling refer to instruction sheet 1221-S.

2. To minimize flow resistance and the possibility of leakage, only as many fittings and connections as are necessary for proper installation should be used.

3. The number of bends in tubing should be kept to a minimum to prevent excessive turbulence and friction of oil flow. Tubing must not be bent too sharply. The recommended radius for bends is three times the inside diameter of the tube.

E. HYDRAULIC FLUID RECOMMENDATIONS

GENERAL DATA

Oil in a hydraulic system performs the dual function of lubrication and transmission of power. It constitutes a vital factor in a hydraulic system, and careful selection of it should be made with the assistance of a reputable supplier. Proper selection of oil assures satisfactory life and operation of system components with particular emphasis on hydraulic pumps. Any oil selected for use with pumps is acceptable for use with valves or motors.

Data sheets for oil selection are available from Vickers,Inc. Technical Publications, Troy, MI. 48084.

For Industrial Applications order data sheet I-286-S. For Mobile Applications order M-2950-S.

The oil recommendations noted in the data sheets are based on our experience in industry as a hydraulic component manufacturer.

Where special considerations indicate a need to depart from the recommended oils or operating conditions, see your Vickers representative.

CLEANLINESS

Thorough precautions should always be observed to insure the hydraulic system is clean:

A. Clean (flush) entire new system to remove paint, metal chips, welding shot, etc.

B. Filter each change of oil to prevent introduction of contaminants into the system.

C. Provide continuous oil filtration to remove sludge and products of wear and corrosion generated during the life of the system.

D. Provide continuous protection of system from entry of airborne contamination, by sealing the system and/or by proper filtration of the air.

E. During usage, proper oil filling and servicing of filters, breathers, reservoirs, etc., cannot be over emphasized.

F. Thorough precautions should be taken, by proper system and reservoir design, to insure that aeration of the oil will be kept to a minimum.

SOUND LEVEL

Noise is only indirectly affected by the fluid selection, but the condition of the fluid is of paramount importance in obtaining optimum reduction of system sound levels.

Some of the major factors affecting the fluid conditions that cause the loudest noises in a hydraulic system are:

1. Very high viscosities at start-up temperatures can cause pump noises due to cavitation.

2. Running with a moderately high viscosity fluid will impede the release of entrained air. The fluid will not be completely purged of such air in the time it remains in the reservoir before recycling through the system.

3. Aerated fluid can be caused by ingestion of air through the pipe joints of inlet lines, high velocity discharge lines, cylinder rod packings, or by fluid discharging above the fluid level in the reservoir. Air in the fluid causes a noise similar to cavitation.

4. Contaminated fluids can cause excessive wear of internal pump parts which may result in increased sound levels.

F. OVERLOAD PROTECTION

A relief valve must be installed in the system, unless it is an integral part of the pump. The relief valve limits pressure in the system to a prescribed maximum and protects the components from excessive pressure. The setting of the relief valve depends on the work requirements of the system components.

G. PORT POSITIONS

The pump cover can be assembled in four positions with respect to the body. A letter in the model

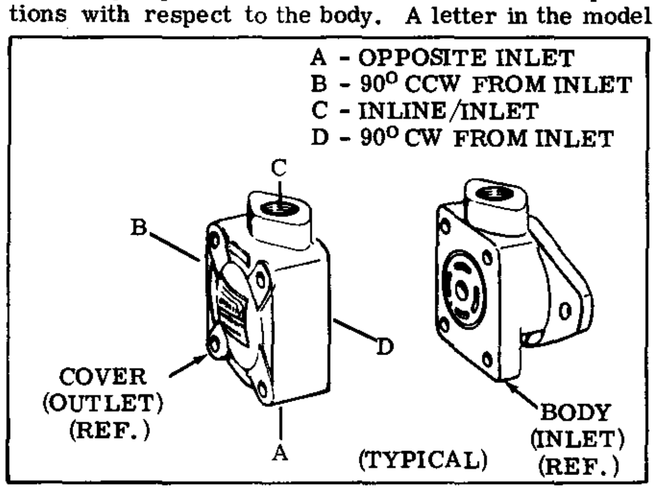


Figure 7

code (Table 2) identifies the cover position as shown in Figure 7.

Disassembly and assembly procedures are in Section VI-B and D.

H. START-UP

With a minimum drive speed of 600 RPM, a pump should prime almost immediately, if provision is made to initially purge the air from the system. Failure to prime within a reasonable length of time may result in damage due to lack of lubrication. Inlet lines must be tight and free from air leaks. However, it may be necessary to crack a fitting on the outlet side of the pump to purge entrapped air.

Section V
SERVICE, INSPECTION AND MAINTENANCE

A. SERVICE TOOLS

No special tools are required to service these pumps.

B. INSPECTION

Periodic inspection of the fluid condition and tube or piping connections can save time-consuming breakdowns and unnecessary parts replacement. The following should be checked regularly.

1. All hydraulic connections must be kept tight. A loose connection in a pressure line will permit the fluid to leak out. If the fluid level becomes so low as to uncover the inlet pipe opening in the reservoir, extensive damage to the pump can result. In suction or return lines, loose connections permit air to be drawn into the system, resulting in noisy and/or erratic operation.

2. Clean fluid is the best insurance for long service life. Therefore, the reservoir should be checked periodically for dirt or other contaminants.

If the fluid becomes contaminated the system should be thoroughly drained and the reservoir cleaned before new fluid is added.

3. Filter elements also should be checked and replaced periodically. A clogged filter element results in a higher pressure drop. This can force particles through the filter which would ordinarily be trapped, or can cause the by-pass to open, resulting in a partial or complete loss of filtration.

4. A pump which is running excessively hot or noisy is a potential failure. Should a pump become noisy or overheated, the machine should be shut down immediately and the cause of improper operation corrected.

C. ADDING FLUID TO THE SYSTEM

When hydraulic fluid is added to replenish the system, it should always be poured through a fine wire screen (200 mesh or finer).

It is important that the fluid be clean and free of any substance which could cause improper operation or wear of the pump or other hydraulic units. Therefore, the use of cloth to strain the fluid should be avoided to prevent lint getting into the system.

D. ADJUSTMENTS

No periodic adjustments are required, other than to maintain proper shaft alignment with the driving medium.

E. LUBRICATION

Internal lubrication is provided by the fluid in the system. Lubrication of the shaft couplings should be as specified by their manufacturers.

F. REPLACEMENT PARTS

Reliable operation throughout the specified operating range is assured only if genuine Vickers parts are used. Sophisticated design processes and material are used in the manufacture of our parts. Substitutions may result in early failure. Part numbers are shown in the parts catalogs listed in Table 1.

G. TROUBLE-SHOOTING

Table 6 lists the common difficulties experienced with vane pumps and hydraulic systems. It also indicates the probable causes and remedies for each of the troubles listed.

It should always be remembered that many apparent pump failures are actually the failures of other parts of the system. The cause of improper operation is best diagnosed with adequate testing equipment and a thorough understanding of the complete hydraulic system.

Section VI - OVERHAUL

WARNING

Before breaking a circuit connection, make certain that power is off and system pressure has been released. Lower all vertical cylinders, discharge accumulators and block any load whose movement could generate pressure.

A. GENERAL

Plug all removed units and cap all lines to prevent the entry of dirt into the system. During disassembly, pay particular attention to identification of the parts, especially the cartridges, for correct assembly. Pump bearings are pressed in the bodies or on the shafts and should not be removed unless defective. Figure 8 is an exploded view which shows the proper relationship of the parts for disassembly and assembly. Refer to Figure 1 and Figure 8 for the correct assembled relationship of the parts.

B. DISASSEMBLY

1. Disassembly of Basic Pump-See Figure 8. If a foot bracket is used, remove before dismantling the pump. Clamp the pump body in a vise (not too

TABLE 6 - TROUBLE SHOOTING CHART

TROUBLE	PROBABLE CAUSE	REMEDY
PUMP NOT DELIVERING FLUID	DRIVEN IN THE WRONG DIRECTION OF ROTATION	The drive direction must be changed immediately to prevent seizure. Figure 10 shows the correct ring position for each direction of rotation.
	COUPLING OR SHAFT SHEARED OR DISENGAGED	Disassemble the pump and check the shaft and cartridge for damage. (See Section VI.) Replace the necessary parts.
	FLUID INTAKE PIPE IN RESERVOIR RESTRICTED	Check all strainers and filters for dirt and sludge. Clean if necessary.
	FLUID VISCOSITY TOO HEAVY TO PICK UP PRIME	Completely drain the system. Add new filtered fluid of the proper viscosity.
	AIR LEAKS AT THE INTAKE. PUMP NOT PRIMING	Check the inlet connections to determine where air is being drawn in. Tighten any loose connections. See that the fluid in the reservoir is above the intake pipe opening. Check the minimum drive speed which may be too slow to prime the pump.
	RELIEF VALVE STUCK OPEN. (MODELS WITH INTEGRAL RELIEF VALVE ONLY)	Disassemble the pump and wash the valve in clean solvent. Return the valve to its bore and check for any stickiness. A gritty feeling on the valve periphery can be polished with crocus cloth. Do not remove excess material, round off the edges of the lands or attempt to polish the bore. Wash all parts and reassemble the pump.
	VANE(S) STUCK IN THE ROTOR SLOT(S)	Disassemble the pump. Check for dirt or metal chips. Clean the parts thoroughly and replace any damaged pieces. If necessary flush the system and refill it with clean fluid.
INSUFFICIENT PRESSURE BUILD-UP	SYSTEM RELIEF VALVE SET TOO LOW	Use a pressure gage to correctly adjust the relief valve.
	COMPLETE LOSS OF FLOW FROM PUMP.	-A valve is stuck open permitting free flow to tank. -Broken inlet or pressure line. -Actuator bypassing the full flow. (Motor valve plate lift)
PUMP MAKING NOISE	PUMP INTAKE PARTIALLY BLOCKED	Service the intake strainers. Check the fluid condition and, if necessary, drain and flush the system. Refill with clean fluid.
	AIR LEAKS AT THE INTAKE OR SHAFT SEAL. (OIL IN RESERVOIR WOULD PROBABLY BE FOAMY)	Check the inlet connections and seal to determine where air is being drawn in. Tighten any loose connections and replace the seal if necessary. See that the fluid in the reservoir is above the intake pipe opening.
	PUMP DRIVE SPEED TOO SLOW OR TOO FAST	Operate the pump at the recommended speed.
	COUPLING MISALIGNMENT	Check if the shaft seal bearing or other parts have been damaged. Replace any damanged parts. Realign the coupled shafts.

tightly), cover end up, and remove the four cover screws. Note the position of the cover port with respect to the body port before lifting off the cover and "O" ring. (See paragraph 2 for disassembly of flow control covers and paragraph 3 for disassembly of the priority valve covers).

Remove the pressure plate and spring. Note the position of the ring for correct reassembly. Lift off the ring and remove the locating pins. Separate the vanes from the rotor and remove the rotor from the shaft.

Turn the pump body over then remove the shaft key and the snap ring which retains the bearing. Tap with a soft hammer on the splined end of the shaft to force the shaft out of the body. Remove the small snap ring, located on the shaft, behind the bearing. Support the bearing inner race and press the shaft out of the bearing. Pull the shaft seal out of the body with a suitable hooked tool.

CAUTION

Do not disassemble the relief valve S/A removed in the following step. The unit is factory set and could malfunction if disassembled.

2. Disassembly of Flow Control and Relief Valve Covers - See Figure 8. Remove the plug (and "O"ring latest design V10 models) from the snap ring side of the cover. Then remove the plug (and "O"ring latest design V10 models), that releases the spring and relief valve S/A. Insert a suitable tool from the snap ring end of the bore. Slide the relief valve S/A from the cover. Remove the snap ring with care - DO NOT scratch the bore.

3. Disassembly of the Priority Valve Cover - See Figure 8. Remove the plug (and "O"ring latest design V10/V20 models) and spring from one end of the priority valve bore, and the plug (and "O"ring latest design V10/V20 models) from the other end of the bore. Insert a suitable tool into the snap ring end of the bore (Snap ring used with pipe thread plugs only) and slide the priority valve spool from the cover. If the snap ring exists, remove it from the cover. Disassemble the relief valve by removing the plug spring poppet and shims. On later designs, a spring guide is used. See Figure 8. DO NOT remove the seat unless inspection of the poppet contact area reveals a problem in the seat area. If removal of the seat is required, thread the seat with a suitable tap approximately 3/8 inch into the seat. Thread a long bolt into the seat and pull the bolt and seat from the bore with a small gear puller.

C. INSPECTION AND REPAIR

CLEANING. All parts must be throughly cleaned and kept clean during inspection and assembly. The close tolerance of the parts makes this requirement more stringent than usual. Clean all removed parts, using a commercial solvent that is compatible with the system fluid. Compressed air may be used in cleaning, but it must be filtered to remove water and contamination. Clean compressed air is particularly useful in cleaning spools, orifices, and cover passages.

1. Discard the used shaft seal and all "O" rings. Wash the metal parts in a solvent, blow them dry with filtered compressed air and place them on a clean surface for inspection.

2. Check the wearing surfaces of the body, pressure plate, ring and rotor for scoring and excessive wear. Remove light score marks by lapping. Replace any heavily scored or badly worn parts.

3. Inspect the vanes for burrs, wear and excessive play in the rotor slots. Replace the vanes and rotor if the slots are worn.

4. Check the bearings for wear and looseness. Rotate the bearings while applying pressure to check for pitted or cracked races.

5. Inspect the oil seal mating surface on the shaft for scoring or wear. If marks on the shaft cannot be removed by light polishing, replace the shaft.

6. Flow Control Cover: Check the relief valve sub-assembly for free movement in the cover bore. Remove burrs from the valve by polishing, but DO NOT round off the corners of the lands. Do not attempt to rework the valve bore. If the bore is damaged, replace the cover.

7. Priority Valve Cover: Inspect the priority valve spool and bore for burrs. Remove burrs from the spool by light polishing with crocus or # 500 grit paper. DO NOT round off sharp corners of the lands. Inspect the cover bore for scratches, wear and/or a pitted surface. DO NOT attempt to rework the bore. If the bore is damaged, replace the cover. The priority valve spool must fit and move within the bore without evidence of bind. Rotate the spool through 360° while inspecting for bind. Inspect the snap ring for damage. (V20 units only). If worn or bent, replace with a new snap ring. If the snap ring is bent, inspect the snap ring groove in the cover for sufficient depth and rounded edges of the snap ring groove. If the groove is defective, replace the cover.

Integral Relief Valve: Inspect the spring. The spring ends must be paralled to prevent cocking of the poppet. The poppet requires a close inspection in the seat contact area. A slight wear pattern should exist around the poppet at the area of seat contact. If the wear pattern is broken, a possible leakage path exists between the poppet and seat. Inspect the seat for possible errosion or other defects. Refer to the seat removal procedure if the seat is defective. (Paragraph VI. B. 3)

D. ASSEMBLY

Coat all parts with hydraulic fluid to facilitate assembly and provide initial lubrication. Use small amounts of petroleum jelly to hold "O" rings in place during assembly.

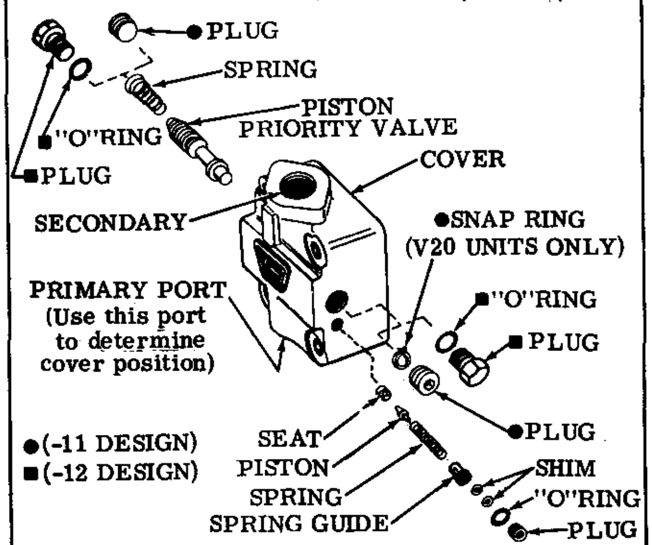
IMPORTANT

During handling and shipping of the precision machined cartridge parts, it is possible to raise burrs on the sharp edges. All sharp edges on the parts of a new cartridge kit should be stoned prior to installation.

1. Assembly of Flow Control Cover - See Figure 8. Assemble the snap ring in place within the bore, (early design only) seat firmly in the groove. Insert the valve in the bore, small land first. Then install the spring and both plugs. Use new "O"rings if straight thread plugs are used.

2. Assembly of priority valve cover - See Figure 8. If the relief valve seat was removed, a new seat must be pressed into the body. Lubricate and insert the new seat chamfered end first into the cover open-

PRIORITY VALVE COVER V10-10/V20-11/-12



NOTE: Priority Cover Information:

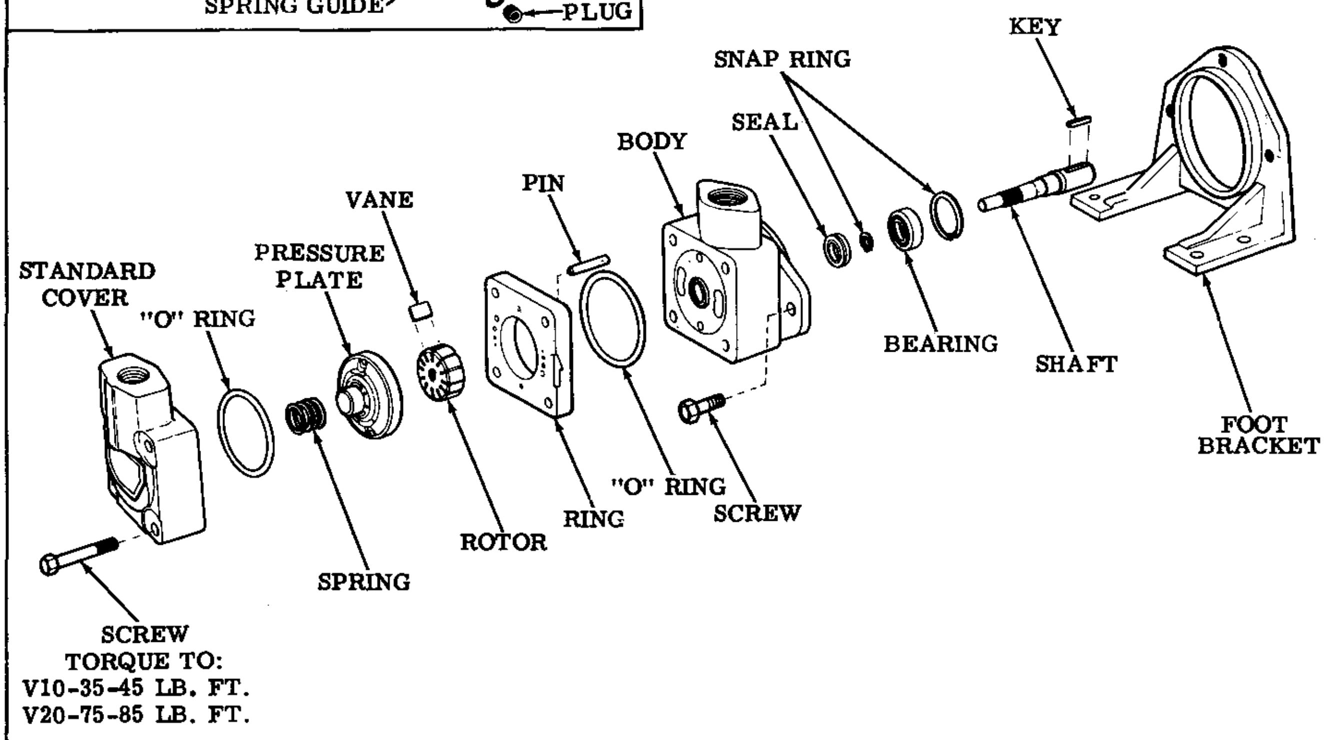
V10-10 early models and V20-11 models use pipe threaded plugs.

V10-10 current design and V20-12 models use straight threaded plugs and "O"ring seals.

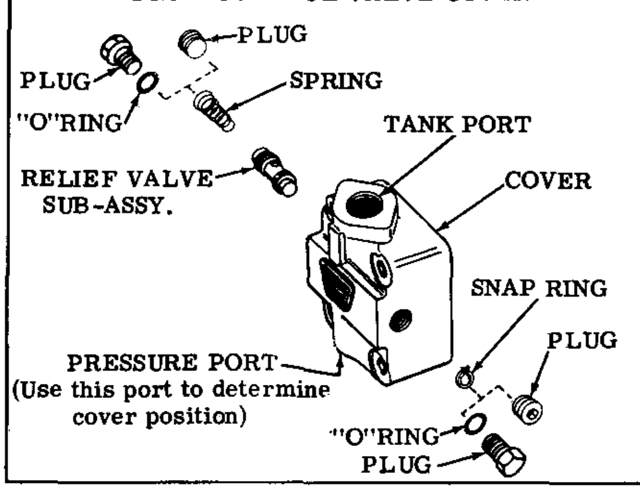
Flow Control Cover Information:

V10-20 early models and V20-11 models use pipe threaded plugs.

V10-20 current design uses straight threaded plugs with "O"rings. The V20-11 is the current design and still uses pipe threaded plugs.



FLOW CONTROL VALVE COVER



SECTIONAL VIEW

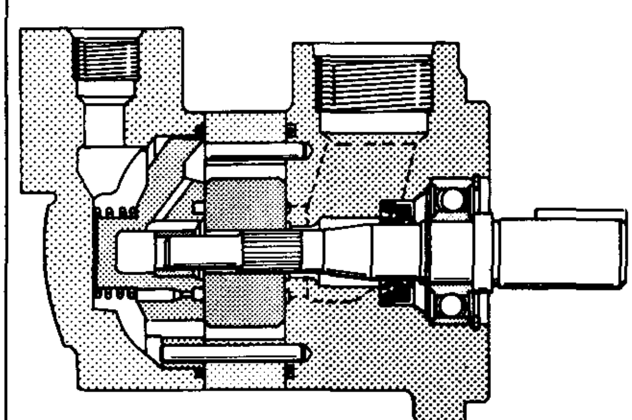


Figure 8

ing. Align square and press into place. Use a short length of brass rod as a pressing tool, to prevent seat damage. Clean the relief valve bore to remove chips and filings. Insert the poppet into the bore, align square and lightly tap the stem of the poppet to mate the poppet and seat. Install the spring, shims, and plug into the cover. (later design uses a spring guide and straight thread plug with "O"rings) Be sure to check the pressure setting of the relief valve against the model code. If the setting is out of tolerance, readjust by removing or adding shims. (Removing shims reduces pressure while adding shims increases pressure.)

Priority Valve - Install the snap ring within the priority valve cover bore, (early V20 series only); make sure the snap ring is seated within its groove. Insert the priority valve spool, small land first, into the bore. Install plugs at each end of the bore and secure. Refer to Figure 8 for spool orientation.

3. Assembly of Pump - See Figure 8. Begin assembly by pressing the shaft into the front bearing while supporting the bearing inner race. Install the small snap ring on the shaft.

NOTE

Before assembling the shaft seal, determine the correct position of the sealing lip. (See Figure 9.) Seals are assembled with the garter spring toward the pumping cartridge. Press the seal firmly in place and lubricate the lip with petroleum jelly or other grease compatible with the system fluid. Slide the drive shaft into the body until the bearing is seated. Tap lightly on the end of the shaft if necessary. Install the snap ring.

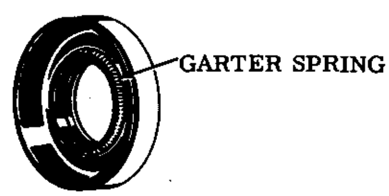
Install new "O" rings in the body and cover. Insert the ring locating pins in the body and assemble the ring so that the arrow on the perimeter points in the direction of rotation. Check the assembly against Figure 10. Install the rotor on the shaft and insert the vanes in the rotor slots. Be certain the radius edges of the vanes are toward the cam ring.

Place the pressure plate on the locating pins and flat against the ring. Place the spring over the pressure plate, and then install the cover with the outlet port in the correct position. Tighten the cover screws to the torque shown in Figure 8. Rotate the shaft by hand to insure that there is no internal binding. Install the shaft key.

If a foot mounting is used, assemble the pump to its foot mounting. If a gasket is used, be certain it is flat to avoid misalignment of the shaft.

Section VII - TESTING

If a test stand is available, the pump should be tested at the recommended speeds and pressures shown on the installation drawing. (See Table 1).



SHAFT SEAL
(Assemble with spring toward
cover end of pump)

Figure 9

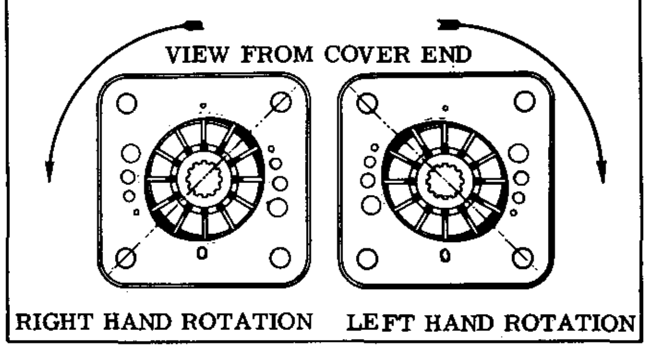
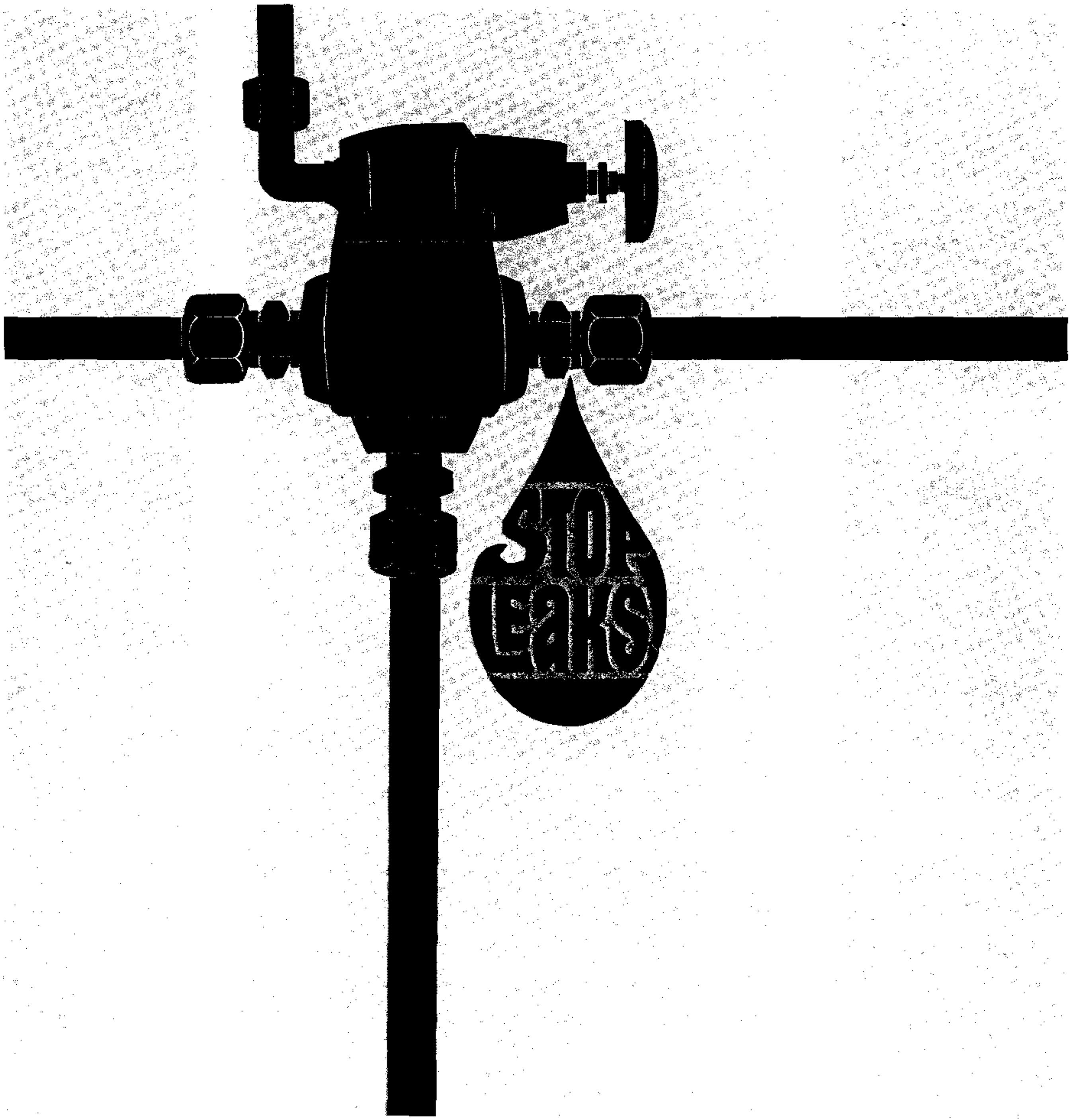


Figure 10

Stop Leaks

Stop Leaks



Introduction

Hydraulics And Industry

Vickers is committed to drying up leaks because leakage has been a stumbling block to complete acceptance of hydraulics by industry. We have launched a three–pronged attack on leakage.

First, we are busy improving the leakage resistance of our products.

Second, we are expanding our knowledge of sealing technology through research.

Third, we are working to assist you in reducing leakage from your plant hydraulic system.

Hydraulics And You

This booklet, a part of our third approach, is for people who build, install, or maintain hydraulic machinery. Right now, you can do more to stop leakage than anyone else in the industry. No new technology is needed. Leakage comes from many individual points, usually where some small detail has been overlooked. Unless the sealing technology at each point is understood, it is hard to correct details. So the details of how sealing elements work, what makes them fail, and how to correct failures are collected in this booklet for your study, reference, and application.

Contents

Must A Hydraulic System Leak?

Absolutely not! The aircraft industry has virtually eliminated leaks in their hydraulic systems.

The mobile equipment industry has significantly reduced leakage in their machines and systems.

How has this been accomplished? One of the factors contributing to aircraft industry success was a switch to better fittings. The mobile equipment industry improved some of their designs by shifting from pipe thread to either straight thread or flange fittings. The most important factor, however, was DETERMINATION. Once committed, these industries followed through.

The Need To Stop Leaks

It has been estimated over a hundred million gallons of oil could be saved each year if external leaks from hydraulic machines and systems were eliminated. Add to this the cost of the labor for spill clean-up, re-filling of reservoirs, etc., and the total savings potential of no-leak systems could be tens of millions of dollars per year.

The HFI Yardstick

Is there a yardstick for checking a plant's hydraulic oil use? Mobile Oil Corporation has developed the Hydraulic Fluid Index (HFI), comparing oil consumption to machine capacity. They discovered that each year the average plant used four times more hydraulic oil (HFI=4) than its machines actually hold. Ideally, the fluid should last almost indefinitely, and the HFI should be less than one. In practice, fluid is lost through leakage, line breaks and fluid changes. Any plant can cut its HFI (oil consumption) in half by developing skilled hydraulic maintenance people who can stop leaks before they occur, and control fluid contamination.

Preventive Maintenance

Before getting down to the specifics of finding and fixing leaks (corrective maintenance), let's consider another important part of leakage control – preventive maintenance.

Handling Repairs

Hydraulic components are precision devices. Careless handling of them or other parts of the system can result in malfunction or failure. To avoid creating problems when installing or repairing hydraulic components, follow these tips:

1. Clean away the dirt in and around equipment before taking apart lines and removing parts.
2. Cap off all disconnected lines and open ports.
3. Protect the overhaul area from grinding dust, machining chips, and wind driven dirt.
4. Work only on metal or hard finished bench tops, easy to keep clean.
5. Handle parts carefully to avoid nicks and burrs.
6. Use lint-free cloths to wipe parts.
7. Use smooth burr-less tools, especially when working with O-rings.
8. Lubricate all sliding parts during assembly.
9. Cover sharp grooves and threads with thimble or shim stock when installing O-rings and other seals.
10. Discard all used O-rings to avoid re-use.
11. Make certain that seals are of the right size and material.
12. Use only recommended replacement parts.
13. Examine all prematurely worn or malfunctioned parts for clues as to the cause of failure.
14. Test the overhauled device before reinstalling it, if possible. It's embarrassing as well as costly to remount a repaired component in the system, only to find that it leaks or doesn't work properly.

Fluid Temperature

Petroleum oils are used in most hydraulic applications to lubricate parts as well as transmit power. As oil temperature increases, however, the lubricating film thins out. The result is rubbing parts supported by the oil film move closer together; friction and wear increases; seal materials age more quickly, become stiff and hard, and may readily permit leakage.

To avoid trouble, industrial bulk oil temperature measured at the reservoir should be kept in the 100° to 130° F. range. Why? Localized hot spots – such as the one at the shaft seal – may be 100° hotter than bulk oil temperature. As bulk oil temperature rises, hot spot temperatures rise an equal amount.

A thermometer on the end of a wire and placed in the reservoir fill pipe will measure fluid temperature. Temperature stickers are available for attachment to the side of a reservoir (or suspected hot spot). Spots on the sticker permanently change color to indicate the maximum temperature.

If oil temperature in your operation exceeds 150° F., determine from your oil supplier whether you are using the correct grade and quality of oil. A higher viscosity oil with anti-wear and anti-oxidation additives may be recommended.

A quick test! **WARNING: APPROACH THIS TEST CAREFULLY, BECAUSE A SEVERE BURN IS POSSIBLE.** If you can't hold the palm of your hand on the pump inlet side of the reservoir when the system is running, the oil temperature is probably too hot.

The general rule for maximum reservoir temperature in mobile applications is ambient temperature plus 80° F (27° C). On a hot day, for example, of 90° F (32° C.), bulk oil temperatures could go to 170° F (90 + 80) (76° C). Only premium oil loaded with additives is satisfactory here.

Nitrile (Buna N) seals can last almost indefinitely in 200°F (93° C) oil, but life is cut in half for every 25°F temperature rise.

Fluid Temperature		Life – Hours
°F: 225	°C: 107	2000
250	121	1000
275	135	500

Fire resistant fluids with a water base require low temperature (130° F or 54° C maximum) to prevent excessive evaporation of water.

Fluid Contamination

"Keep it cool and keep it clean" is the secret of long fluid life. Contamination may be in the form of gas, liquid or solid; it may be chemically active or inert.

Common contaminants in hydraulic oil are:
- Gas – air
- Liquid – water and cutting oils
- Solid – rust, chips and grit

It is usually easier to keep contaminants **out** of a system rather than remove them after they are **in** the system.

Bulk handling and the re–use of oil containers almost guarantee "new" oil will be dirty. Unless it comes from sealed quart containers, filter all "new" oil before adding it to your system.

Control solid particle contamination through filtration. Make is a practice to change all filters on a regular basis **before** they become clogged.

Fire resistant hydraulic fluids, whether water-based or non water-based, require a high standard of filtration within the circuit to maintain system reliability.

Seal Wear

In long life applications, a major cause of dynamic seal leakage is wear. Wear can also be the cause of static seal leakage. Shock, pressure surge and vibration cause static sealing surfaces to slide an separate in small amounts, where movement is measures in thousandths of an inch. Seal wear occurs and the seal eventually leaks. The rougher the sealing surface, the heavier the wear, and the sooner the leak occurs.

Measuring Surface Roughness

Inspectors use an electronic instrument called a profilometer to measure surface roughness and report the roughness height value in microinches (millionth of an inch). The smoother the surface, the smaller the reported value.

For static sealing, where surfaces to be seals do not move relative to each other, commercial practice is to specify a 32 to 63 microinch (abbreviated μ in.). As long as surface separation or seal movement does not occur, satisfactory sealing is possible on surfaces with light circular tool marks up to 100. But no matter how smooth the overall surface, a nick, gouge, longitudinal scratch or spiral tool mark provides a leak path.

For dynamic sealing, the surface moving relative to the seal requires a nominal 16 microinch finish.

As a guide, the surface roughness produced by common production methods is:

Production	Roughness Height
Milling	32 – 250
Die Casting	32 – 63
Boring, Turning	8 – 250
Grinding	4 – 63
Polishing	4 – 16

Cut Through The Jungle Of Fluid Lines

If your system is a jungle of fluid lines and connections with components almost inaccessible, the odds of controlling leakage are against you.

Simplify the hydraulic system whenever possible. The fewer the connections, the fewer the possible sources of leaks. Consider back-mounting your valves, with all plumbing connections made permanently to a mounting plate. Even better is the use of manifolds, which provide interconnections between valves and eliminate a lot of external plumbing.

Tubing Clamps

Shock, pressure surge, and vibration will flex metal tubing. This flexing will fatigue the tubing, particularly around line fittings. The consequences are cracked fluid lines. A well designed fluid power system will include tubing clamps to minimize shock and vibration. All tubes should be clamped on both sides of a bend, as close to the bend radius as possible.

Manufacturers generally recommend spacing between clamps as shown:

Tubing Diameter OD in. (MM)	Distance Btw. Clamps Feet (m)
$^3/_{16}$–$^3/_8$ (4.8 – 9.5)	3 (.9)
$^1/_2$–$^7/_8$ (12.7 – 22.2)	5 (1.5)
1 – $1^1/_4$ (25.4 – 31.8)	7 (2.1)

Corrective Maintenance

Finding The Leak

At first this discussion might seem absurd. After all, isn't a leak obvious? Certainly the effects of the leak are obvious...oil on machine, floors and product.

Remember that we must know the exact location of a leak before we can eliminate it. Anyone would be hard pressed to point immediately to the source of a leak on a machine which has oil over the top, sides and dropping onto the floor. In fact, there might not even be a leak. The real cause might be an over–filled reservoir or a spill.

The firsts step then in locating leakage sources is to clean the area and watch. Focus on the four general areas where leaks can occur:

1. Fittings
2. Hoses
3. Dynamic seals
4. Static seals

Most leaks occur at fittings, but, too often, finding the fittings that is leaking is difficult because the fluid runs along the line and drips off at some other point. Always suspect the inaccessible connection; they are often installed carelessly because they are so hard to get to. Leaks in high pressure lines sometimes are difficult to pin–point because the fluid comes out as a mist.

(Use Caution: High pressure leaks can cause personal injury).

Determine Cause Of Leak

Once you find the location of a leak, the specific cause has to be determined before it can be corrected. A scratch in a fitting seat or a cut in a seal lip that is big enough to leak excessively can still be too small to find with the naked eye. A four power magnifying glass, used frequently to search for defects, will cut down on repeat repairs. What do you look for? The detailed information in the following sections will help you decide that. It will also help to correct leakage and prevent its reoccurrence.

Threaded Fittings

Types Of Threads

There are two basic types of threaded joints. One has tapered thread that produces a metal-to-metal seal by wedging surfaces together as the pipe is screwed in. The other has straight threads and no wedging action, but has a rubber like element that does the sealing.

Tapered threads have the advantage that an additional fraction of a turn usually will cure a slight leak. Their sealing ability, however, depends o how perfectly they are formed. In practice, threads are often carelessly machined and won't seal regardless of how much they are tightened. In such cases excessive torques are applied, and yielding or cracking results. Because of the frequency of such leaks there is a trend to quit using tapered pipe threads. Some companies limit their use to low pressure (below 500 psi) lines.

USA Standard (Symbol NPT)

The USA Standard pipe thread is tapered and shaped to engage mating threads on their flanks as shown in Figure 1. The leaves a small groove along their tips that must be closed with a sealer (dope). The sealer also lubricates the threads and prevents galling.

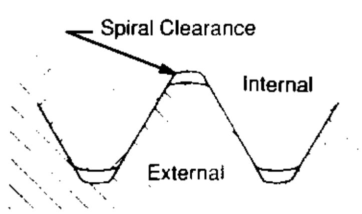


Figure 1. Wrench Tight NPT Thread

USA Standard Dryseal (Symbol NPTF) (SAE J476)

These are very similar to NPT pipe threads, except they are shaped to make first contact at their roots and crests as shown in Figure 2. When the joint is pulled up tight with a wrench the thread crests are crushed until the thread flanks make full contact as shown in Figure 3. They don't have the built–in leakage path of the National thread but they can still leak due to machining imperfections. Dope should still be used on these threads for a lubricant, but dope can't be expected to seal a poorly cut thread.

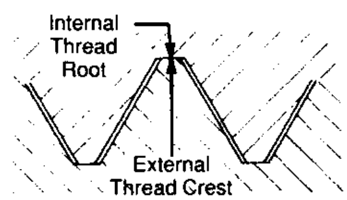


Figure 2. Hand Tight Dryseal NPTF

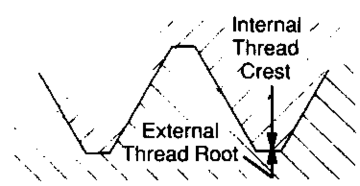


Figure 3. Wrench Tight Dryseal NPTF

Sealing Tapered Thread

A large number of sealants are available for hydraulic fittings. Choice may depend on a particular application. We have found Loctite 92, for example, an effective sealer on pressure plugs which require removal after extensive service.

The sealant should always be applied to the male thread, never the female. Avoid the first two male threads from the end to keep sealer out of the system. TFE–fluorocarbon tape is not recommended for use on hydrauic fittings because pieces may get into the system and jam close-fitting parts.

Sealants generally are lubricants, so don't depend on a **mechanic's feel** for tightening pipe fittings. There is a general rule of thumb to get maximum sealing without distorting or cracking steel parts.

Engage tapered pipe thread hand tighten, then 3 turns with a wrench

NOTE: This "rule" may vary with materials, temperature or wall thickness.

Some tips for troubleshooting tapered threads leakage are given in Table 1.

Possible Sources of Trouble	Suggested Remedy
1. Fitting is under-torqued	Tighten using "General rule of thumb" in text.
2. Female part expanded from heat	Retighten while hot.
3. Vibration has loosened fitting	Retighten if fitting is not cracked. Use clamps with vibration dampeners for support.
4. Hydraulic shock	Retighten fitting if not cracked. Recharge accu-mulators. Check use of low shock control valves, deceleration and decompression components.
5. Female threads in part are over-size	Inspect. Replace if oversize.
6. Male threads are undersize on pipe	Inspect. Replace if undersize.
7. Straight male thread put into tapered thread port	Inspect. Replace with tapered Dryseal thread fitting.
8. Threads galled, dirty or damaged before assembly	Rework, if pos-sible, with sharp taps and dies or replace faulty parts.
9. Check for cracks. Replace damage parts	Check for cracks. Replace damage parts.

Table I. Tapered Thread Leakage Analysis

Straight Thread Fitting (SAE J514)

The SAE straight thread fitting does not depend on thread surfaces for sealing because an O-ring is the seal. Straight thread fittings may be classified as fixed-hex, Figure 4, or adjustable nut, Figure 5.

Installation torque values recommended are listed in Table II for both fixed nut and adjustable nut fittings.

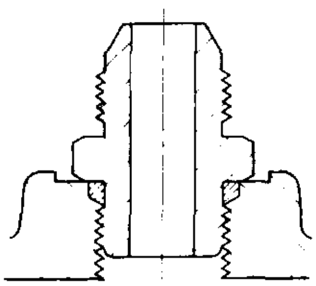


Figure 4. Adapter (Fixed Nut)

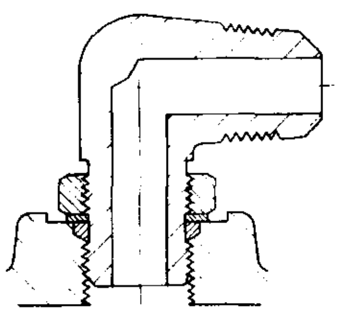


Figure 5. Elbow Connector (Adjustable Nut)

If the fitting leaks, either the seal was damaged during assembly, the seal lost its flexibly from use or the machined boss may be defective. Make certain the size and material are correct for the system fluid and operating temperature.

A fixed procedure should be used to avoid cutting or trapping the O-ring during assembly of the adjustable straight thread fittings, as follows:

1. Lubricate the O-ring with the fluid to be sealed or a light grease such as petroleum. Use a plastic or metal thimble over the threads when replacing the O-ring to avoid a nick or cut from the threads. Slide the O-ring over thimble and onto undercut section of fitting and against the back-up washer, Figure 6.

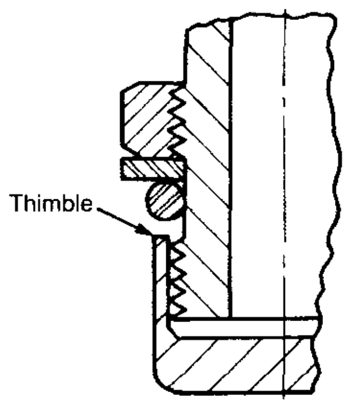


Figure 6. Installed O-Ring

2. Screw adjustable nut fittings by hand into the SAE straight thread boss until the back–up washer bottoms on the boss face, with the O–ring squeezed into the boss cavity. Figure 7.

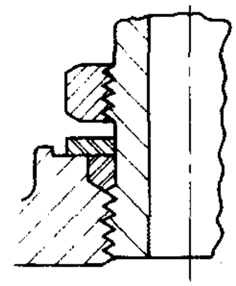


Figure 7. Bottomed Washer

3. Next, position the fitting to meet the joining tube by unscrewing out as far as necessary, up to a maximum of one full turn. Tighten the locknut with a wrench so the back–up washer contacts the boss face (Figure 8) using torque values in Table II.

Table III gives some straight thread leakage troubleshooting hints.

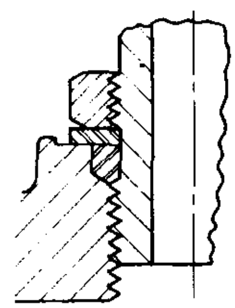


Figure 8. Positioned Elbow

Nominal Tube Size		4	6	8	10	12	16	20	24
Fixed or Adjustable	Min.	160 (18)	215 (25)	430 (49)	630 (72)	820 (94)	1340 (153)	1750 (200)	1850 (211)
Hex/Lock Nut	Max.	175 (20)	245 (28)	470 (54)	700 (80)	90 (103)	1470 (168)	1930 (220)	2040 (233)

Table II. Straight Thread Fitting Boss Installation Torque, in.lb. (Nm)

Possible Sources of Trouble	Suggested Remedy
1. O-ring cut	Replace O-ring lubricating with petrolatum or fluid to be sealed. Use thimble, Figure 6, to protect ring. Inspect for burr.
2. O-ring pinched on assembly into port	Inspect roughness of port sealing surface. Repair or replace part if finish exceeds 100 micro. Lubricate O-ring before assembly.
3. Sealing surfaces of port or fitting are scratched or gouged	Repair if possible, otherwise replace.
4. Sealing surfaces of port or fitting are dirty	Clean and lubricate before reassembly of parts.
5. Port spotface is too small. Nut or washer hangs up on spotface shoulder	Enlarge spotface so fitting can seat properly, or replace faulty part.
6. O-ring edges nibbled because pressure is lifting fitting	Check pressure relief valve setting. Increase seating torque on fitting.

Table III. Straight Thread Leakage Analysis

Tube Fittings

Three Broad Groups

A wide variety of tube fittings is found in hydraulic installations. They all fall into three broad groups: flared, flareless, and welded or brazed. None is a universal fitting. Each is designed to do a specific job. In making a choice of fittings, effectiveness for the job is measured against cost.

Flared fittings are generally less expensive and more readily available in a wider variety of sizes and materials than any other type. Some bite-type flareless are recommended over flared fittings where vibration fatigue failures have been a problem. But flareless are very sensitive to proper assembly torque. They are prone to leak if over or under torqued. In normal service, both flared and flareless are equally effective. Welded fittings meet the most severe requirements of high pressures, temperature and severe mechanical load, and cost the most.

Flared Fittings

A typical three–piece flared fitting – body, nut and sleeve – is shown in Figure 9. The metal–to–metal seal occurs when the softer tubing is pressed against the hardened, conical seat area of the connector body. As the nut is tightened, the sleeve is pushed against the tube. The sleeve absorbs the twisting friction of the nut, transferring only axial forces against the flared tube. This eliminates tube twisting found with two-piece fittings.

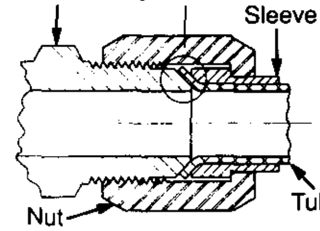


Figure 9. Three-Piece Flared Fitting

The 37° Flare (SAE J514)

The standard flange angle for hydraulic tubing is 37° from the center line, Figure 10. Most tube end flares are made by hand or power tools which swage the tube end over a split die. Flares made in this manner tend to have a ridge of metal extruded from the outer surface at the parting line of the die. This ridge will provide a leak path unless cleaned up. Common fabrication errors are to make the flare too broad to too narrow. See Figures 10 and 11.

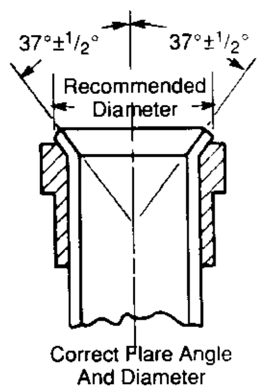


Figure 10.

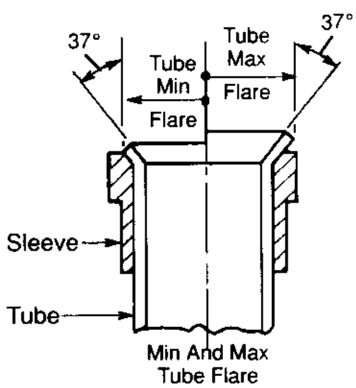


Figure 11.

Refer to SAE J533 for recommended flare diameter.

A tube flared too narrow may be squeezed this because only a small area of the tube is clamped. It may also leak, break at the flare, or pull out under strain.

A tube fared too broad may jam on threads of the nut when assembling the joint.

The 45° Flare

Another common flare angle, used mainly in low pressure automotive and refrigeration applications, is 45° from the center line. Don't mix 45° with 37° fittings. A simple method to tell a 37° from a 45° flare fitting is to use the edge of a salesperson's calling card. If placed inside and along one edge of the 37° flared tube, the card will bottom out on the outside lip, Figure 12. The card placed inside a 45° flare will fit perfectly, Figure 13.

Calling Card Test

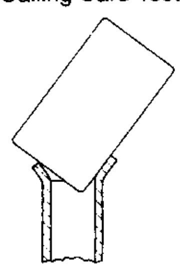


Figure 12. 37° Flared Tube

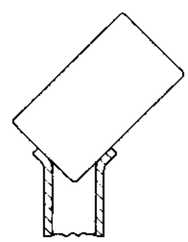


Figure 13. 45° Flared Tube

When The Joint Leaks

Most leaks on the flared tubing type of fitting are caused by poor tubing flare, irregularity in connector body seat surface, or lack of joint tightening. The only cure for a poorly flared tube is a well flared tube. Several companies make small truncated cones of ductile metal for a quick fix when irregularities in conical seats provide a leak path. Placed between seat and flare, the thin ductile washers tend to fill in non-uniformities in body seat and tubing flare when the joint is tightened up.

Proper Joint Torque

You can't tell by looks alone whether the joint has been tightened. If the nut is more than finger tight, you can't tell by observation how much. If many joints are made up at one time, the user must depend on his memory to know if he has tightened all the joints.

An excellent method by which anyone can tell whether a flare joint has been tightened and how much it has been tightened is the alignment-mark method.

Flare Joint Torque By Alignment-Mark Method

1. Tighten nut finger tight until it bottoms on the seat.
2. Mark a line lengthwise on the nut and extend it onto the connector body. Use a felt pen or marker, Figure 14.

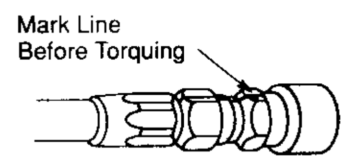


Figure 14.

3. Using a wrench, rotate the nut to tighten, Figure 15.

Figure 15.

4. The recommendations of Aeroquip Corporation are listed in Table IV for nut rotation and torque for different sizes of SAE 37° flare fittings for best sealing. Aeroquip recommends the nut rotation method. If torque values are used, Aeroquip suggests that the lower value apply to machined seals, and the higher value to be used in conjunction with flare tubing.

Nominal Tube Size Inch (MM)	Fitting Size	Rotate No. of Hex Flats	Torque	
			In. Lb.	Nm
$^{1}/_{4}$ (6.4)	4	$2^{1}/_{2}$	135-145	15-17
$^{3}/_{8}$ (9.5)	6	2	260-290	28-33
$^{1}/_{2}$ (12.7)	8	2	430-470	49-54
$^{5}/_{8}$ (15.9)	10	$1^{1}/_{2}$ - 2	680-750	78-86
$^{3}/_{4}$ (19.1)	12	1	950-1050	109-120
1 (25.4)	16	$^{3}/_{4}$ - 1	1300-1360	149-155
$1^{1}/_{4}$ (31.8)	20	$^{3}/_{4}$ - 1	1520-1600	174-183
$1^{1}/_{2}$ (38.1)	24	$^{1}/_{2}$ - $^{3}/_{4}$	1900-2000	217-229

Table IV. Nut Rotation and Torque Recommendations

Flareless Fittings

Flareless fittings were developed for heavy wall tubing not suitable for flaring.

How They Seal

A typical bite style flareless tube fitting is shown in Figure 16.

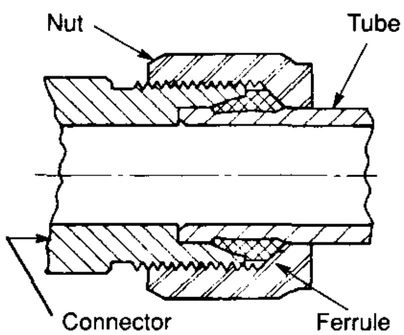


Figure 16. Bite-Type Three-Piece Fitting

The wedging action of the ferrule when drawn down by the nut forms a seal between ferrule and connector body. At the same time, the cutting edge of the ferrule bites into the tube wall forming another positive seal completely around the circumference of the tube. The keys to reliable operation of the bite style flareless fitting are presetting of the ferrule and inspecting for good preset.

Where it is desirable to use heavy wall tubing along with the 37° connectors, a flareless tube fitting is available from Aeroquip Corporation under the tradename Versil-Flare™ Flareless. No special preparation or presetting tools are needed with these fittings. (Figure 17)

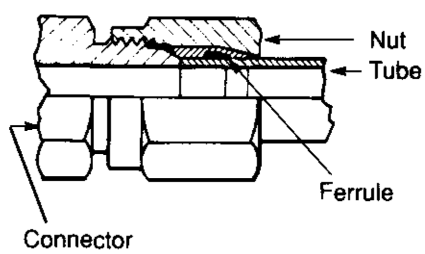


Figure 17. Versil-Flare™ Flareless Tube Fitting

Avoid Thin Wall Tubing

Bite-type fittings are not recommended for thin wall tubing because the ferrule can collapse the tube wall before an adequate bite is achieved.

Caution On Assembly

While the flareless fitting is less prone to poor workmanship than the 37° flare, it is more sensitive to torque or tightening. It will leak if under-tightened, nothing can be done. It must be replaced with new tube and ferrule. The nut and body can be salvaged. As with the flared fitting, follow the manufacturer's recommended assembly procedures very closely. Tightening after preset is usually 1/6 to 1/3 of a turn from finger right (one to two flats).

Welded Fittings

High pressure and/or high temperature systems that do NOT require frequent disassembly may use welded or brazed fittings. A typical welded fitting is shown in Figure 18 where tubing is inserted into a fitting body, positioned and welded.

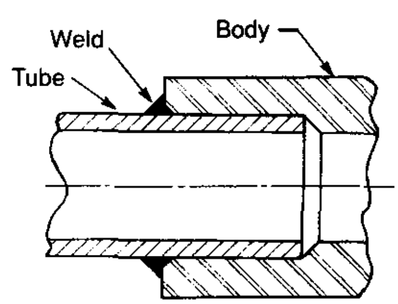


Figure 18. Welded Fitting

Stress Relieve

Low carbon steel welds are sometimes porous or cracked. In each case, the porous or cracked section must be ground out and the joint again welded. Then, as with all welded joints, STRESS RELIEVE the welded area. Use a torch and heat the weld area up to a dull red, then air cool.

Caution

Rigid pipe and tubing require proper alignment before final attachment to avoid high stress and ultimate line failure. Never attempt to pull a line into place using clamps, bolts, or tube nuts to hold it in position. If a particular connection is a difficult one, use the proper strength flexible hose.

Hose and Hose Fittings

Hose Construction

Whenever a connection between moving parts is necessary, a flexible hose is usually the answer. They are also used in some hydraulic systems to reduce the effects of vibration and hydraulic shock. The typical industrial hose consists of (1) an inner tube for conducting the fluid, (2) a reinforcement for the inner tube, and (3) a cover layer to protect the reinforcement. All three are bonded together, see Figure 19.

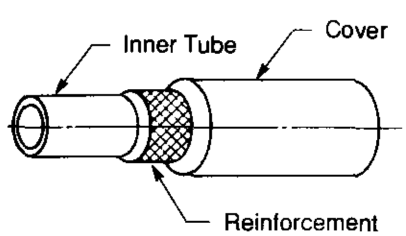


Figure 19. Hose Section

Both elastomeric (rubber-like) and plastic hoses are available to cover a variety of fluids and operating conditions.

Hose Fittings Types

The hose fittings may be either formed or reusable. The formed type, Figure 20, may be crimped or swaged onto the hose at the factory or in the field. Special equipment is required for the crimping or swaging. The entire assembly is thrown away when the hose is replaced.

The reusable type hose fitting, Figure 21, has a definite advantage today with many items in short supply. Only the hose is discarded, while the fittings are saved.

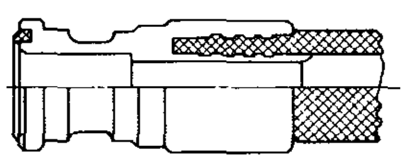


Figure 20. Non-Reusable Swaged or Crimped Hose Fitting

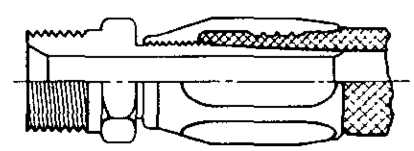


Figure 21. Reusable, Screw Together Hose Fitting

Maximum Hose Life

Hose leakage or failure many times occurs where the end fitting grips the hose. It may be that the hose manufacturer's instructions were not followed on proper crimp or swage of the hose fitting. Check the system for pressure spikes or surge. Make sure operating pressures do not exceed 25 percent of hose rated minimum burst pressure. If bulges or bubbles occur on a flexible hose, a leak is taking place within the inner layers. The hose should be replaced.

High oil temperatures (over 200°F, 93°C) quickly harden or stiffen a ribber inner tube. When pressure pulses flex a hardened hose, it fails by cracking. Every increase of 25°F (14°C) cuts hose life in half. Use a hose rated for actual fluid temperatures. Keep a log of hose use so replacement can be made before failure occurs.

Sometimes on negative/low pressure lines, the hose is clamped with fittings like those found on auto radiator hoses. While there may be no apparent fluid leakage out past the clamp, there may be considerable air leakage into the system, especially on pump inlet lines. As a result, the system gets spongy. Actuator response may be slow and the pump may cavitate with the sealing surfaces being eroded. Tighten clamps! Double clamp if necessary or revise the system to eliminate hose clamps.

Hose Installation

Under pressure, a hose may change in length. The range is generally from –4% to +2%. Always provide some slack in the hose to allow for this contraction or elongation. The following drawings illustrate wrong and right hose installation.

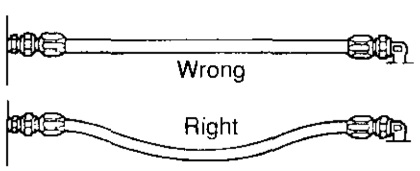


Illustration courtesy of Aeroquip Corp.

At bends, provide enough hose for a wide radius curve. Too tight a bend pinches the hose and restricts the flow. The line could even kink and close entirely. In many cases, use of the right fittings or adapters can eliminate bends or kinks.

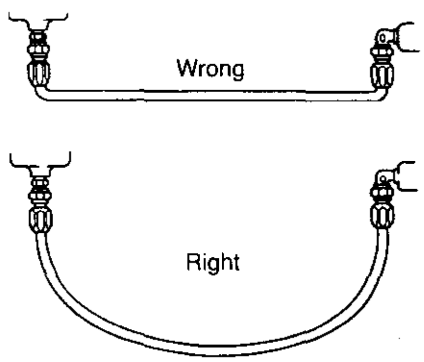
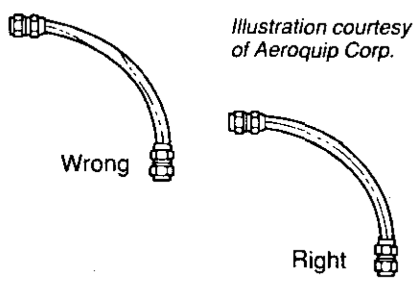


Illustration courtesy of Aeroquip Corp.

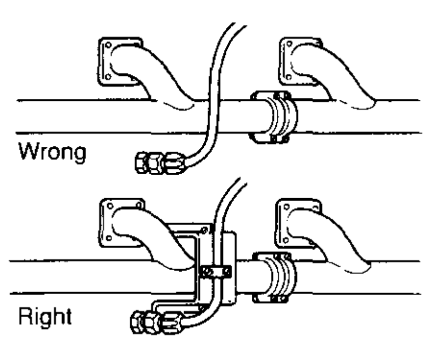
If a hose is installed with a twist in it, high operating pressures tend to force it straight. This can loosen the fitting nut or even burst the hose at the point of strain.

Illustration courtesy of Aeroquip Corp.



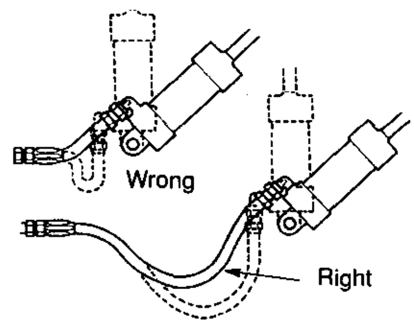
When hose lines pass near a hot surface, they should be insulated by a heat-resistant boot or a metal baffle. In any application, brackets and clamps keep hoses in place and reduce abrasion.

Illustration courtesy of Aeroquip Corp.



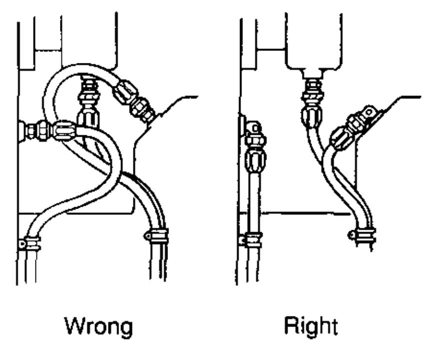
In applications where there is considerable vibration or flexing, allow additional hose length. The metal hose fittings, of course, are not flexible, and proper installation protects metal parts from undue stress and avoids kinks in the hose.

Illustration courtesy of Aeroquip Corp.



When 90° adapters were used, this assembly became neater looking and easier to inspect and maintain. It uses less hose, too!

Illustration courtesy of Aeroquip Corp.



Dynamic Seals

Seal Life

Dynamic seals prevent or control leakage between surfaces that move past each other. Since these seals contact moving surfaces, they will eventually wear out or fail. Periodic replacement of the seal is required. With proper installation and maintenance, however, dynamic seals may last several hundred to several thousand hours. High pressure, temperature, speed and surface roughness work to reduce seal life.

Where The Action Is

Typical locations for dynamic seals are pump and hydraulic motor drive shafts, pintles on variable displacement pumps and motors, directional valve push pins and actuator rods. In brief, dynamic seals prevent leaks where the action is taking place.

Three types of dynamic seals are lip seals, face seals, and packings.

Radial Lip Seals

Radial lip seals, commonly called oil seals or shaft seals, are used to retain fluids in or keep dirt out of equipment with reciprocating or rotating shafts. The simplest type, the single lip seal, Figure 22, is used only for low speeds and low pressures.

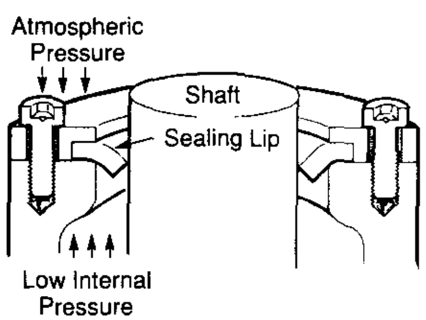


Figure 22. Simple Lip Seal

Seal Dynamics

Sealing is normally a result of an interference fit between the flexible sealing lip and a shaft. However, as seals age and temperatures change, the interference fit or lip pressure falls off. To maintain a more constant load on the shaft, a garter or finger spring is used, Figure 23. This permits operation at higher speeds and moderate pressures. It should be noted, the seal lip does NOT act like a squeegee to wipe the shaft dry. The lip must ride on a thin film of lubricant to be successful: If the film gets too thick, the seal leaks. If it gets too thin, the seal lip wears and gets hard. The harder the seal, the more difficulty the lip has in following the shaft movement.

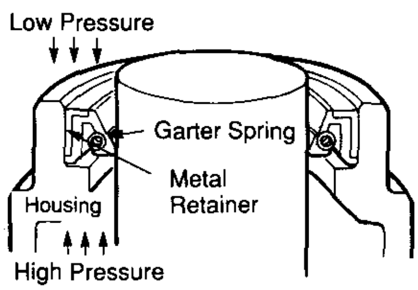


Figure 23. Lip Seal with Garter Spring

Surface Finish

Tests have shown maximum seal life is obtained when the shaft sealing surface is 8 to 20 microinches. If the shaft is too smooth, it won't support a film. If too rough, it wears the seal lip. In either case, premature seal failure may occur. Finish marks should be circumferential rather than axial to retain the fluid. A spiral tool mark will pump oil out or air in past a seal, depending on shaft rotation.

Lip Seal Installation

1. Lip seals must be installed correctly to operate successfully. But you must start with a good product. Examine the seal to be sure it is the correct part. has not been damaged, nor lost its spring.
2. A press should be used for installing the oil seal into the bore. The press ram or driving tool should not be more than .010" smaller in O.D. than the bore diameter and should have a flat face to contact the back of the metal case on the seal. If installing the seal in a reverse position, be sure that ram pressure is applied only to the rollover bead around the outer diameter of the seal face and not to the inside face or filler ring inside the metal outer case.

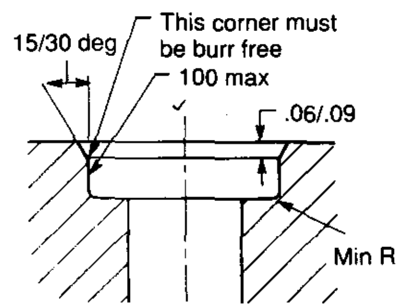


Figure 24. Optimum Seal Bore Dimensioning

3. Polish shaft to remove burrs, sharp or rough edges that touch seal lip during assembly. Use mounting thimble or a sheet of shim stock as in Figure 25, to protect the seal. The thimble wall should be as thin as possible (.012 in. 3mm max.) to avoid seal lip distortion during assembly. We recommend lubricating the shaft and oil seal lip before mounting the seal over shaft.

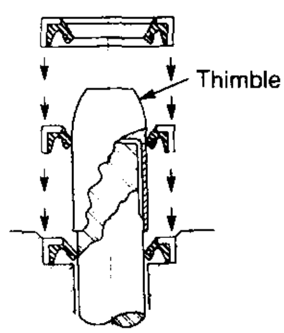


Figure 25. Shaft In Place

4. If a press cannot be used, the seal may be seated with a driving plug or tool, Figure 26. This tool is placed into position and tapped with a mallet. When large seals are being seated, or in an emergency, a block of wood resting squarely on the seal may be used instead of a driving tool. **Never hit the seal directly!**

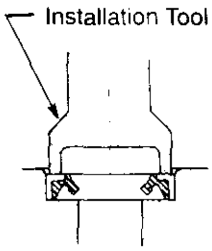


Figure 26. No Shaft

5. Check shaft-to-bore misalignment and dynamic runout. **Misalignment** is the distance that the shaft is off center with respect to the bore, Figure 27. **Runout** is the amount by which the shaft, at the sealing surface, does not rotate around its true center, Figure 28.

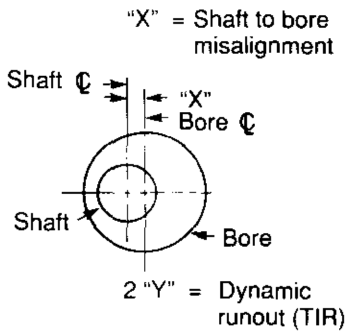


Figure 27. Shaft-To-Bore Misalignment

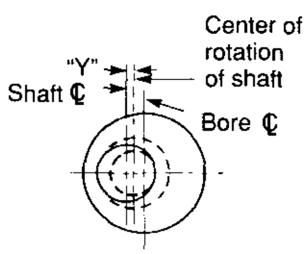


Figure 28. Dynamic Runout

Misalignment plus runout is called eccentricity. For a given eccentricity, the probability of shaft seal leakage increases as shaft speed increases. The recommended maximum eccentricity which should exist in the standard lip seal application is shown in Figure 29.

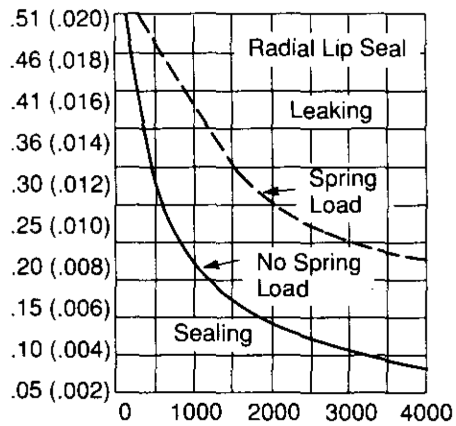


Figure 29. Shaft Speed (RPM)

To reduce misalignment, it is necessary to remove the cause: shaft bearings, housing, or shaft side load. To lower runout, replace the shaft. Table V offers some suggestions in reducing lip seal leakage.

Possible Source Of Trouble	Suggested Remedy
1. Worn shaft	Check shaft sealing surface hardness–Rockwell C 30 minimum necessary. Replace seal. Use wear sleeve on shaft if available. Otherwise replace shaft. Lubricate parts.
2. Rough finish on shaft	Finish shaft surface to 8-20 μ in.
3. Damaged shaft	Replace shaft. Protect sealing surfaces during handling and assembly.
4. Adhesive or paint on sealing surface	Clean with crocus cloth. Mask shaft during seal assembly into bore or painting of unit.
5. Seal cocked in bore	Use proper driving tool. Install seal at right angle to shaft surface.
6. Seal lip reversed	Check stock seal whether it has double lip before replacement. Some seals have double lips. One lip faces inward to retain fluid, one faces outward to exclude dirt.
7. Seal lip cut or torn	Replace seal. Lubricate seal/shaft. Use thimble to carry lip over keyways, splines and sharp edges. Be sure lip I.D. is not stretched over .035 in. (.9 mm).
8. Seal lip worn glazed or hardened, shaft OK	Check for hot oil, high case pressure, and correct seal size. Is seal lubricant good?
9. Seal spring damaged	Replace seal avoiding excessive spreading of sealing lip and spring. Check for proper storage and handling of seals.
10. Excessive eccentricity or misalignment–seal lip can't follow shaft movement	Align shaft, eliminate shaft side load, or use a better flexible coupling.
11. "Built-in" seal flaw such as contamination, poor rubber bond to metal or flash on seal lip	Replace seal.

Table V. Lip Seal Leakage Analysis

Face Seals

The mechanical face seal is one of the most effective devices in preventing leakage along a rotating shaft which passes in or out of an area of pressurized oil. Two ultra-flat sealing faces are mounted perpendicular to the shaft. The seal seat is attached to and rotates with the shaft, while the spring loaded seal head is stationary, Figure 30.

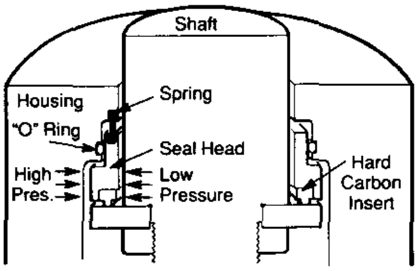


Illustration courtesy of Battelle Research

Figure 30. Face Seal With Stationary Head And Rotating Seat

The usual seal face materials in hydraulic applications are bronze or hard carbon for the seal head, and steel or cast iron for the seal seat. The two are separated by an oil film. With an excellent matching of sealing forces and seal flatness, oil surface tension can complete the seal and there is no leakage. Elevated pressures can induce seal wear, but with proper balancing, pressure induced sealing forces can be kept low.

Repair Of Worn Parts

Only a properly trained person should attempt to repair the sealing surfaces of face seals. The condition of the seal surfaces is so critical that one company provides 40 hours of training to its personnel on face seal operation, repair and installation.

Handle With Care

With the new, correct replacement parts, don't touch the sealing surfaces with fingers or an old wiping rag. Make sure the seal seat is perpendicular to the shaft within .001 inch (.025 mm) TIR, Figure 31. Lubricate the sealing surfaces well with the fluid to be sealed before installation.

TIR is the total change in Indicator Reading during one complete rotation of the shaft.

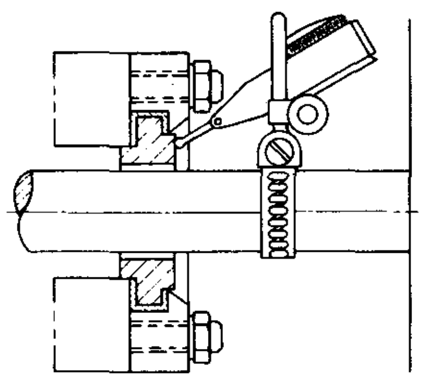


Figure 31. Dial Indicator Mounted On The Shaft Makes Alignment Of The Stationary Seal Seat To 0.001 Inch TIR Easy

Troubleshoot

Examine the old parts for telltale signs. Abrasive wear of the sealing faces means contaminated oil. Burned faces indicate dry running of the seal. Heavy wear may mean either excessive operating pressure or a hung-up spring. A cracked carbon ring leaks badly. Worn bearings should be replaced if end play exceeds .002 inch (.05 mm) or radial looseness is greater than .004 inch (.10 mm). Replace the shaft with a new one if runout exceeds .002 inch (.05 mm) TIR. Polish the new shaft to remove burrs or scratches that might damage static seals.

Test Assembly

To insure against goofs, test the mechanical seal assembly with low pressure filtered air (5 to 20 psi, .3-1.4 bar) before installing the component on a machine. For example, an externally drained piston pump housing is easily pressurized through the drain port connection.

Packings

A packing is a material, deformed so as to throttle leakage between a moving or rotating part and a stationary one. With rapid motion, there must be enough leakage to lubricate and cool the packing. On some large applications using compressed packings, the desired leakage rate may be as high as 10 drops per minute. On some small O-ring applications with rapid motion, the leakage rate may be as low as one drop per every other hour. Where there is relatively little motion, packings can seal without fluid leakage. Three basic types are: compression, lip and squeeze packings.

Compression Packing

Compression packing used in chemical processing is rarely found in industrial hydraulic service. The packing is made of twisted, woven, or braided cotton, flax or asbestos fiber. Metal foil or wire is sometimes added for reinforcement along with solid lubricants such as graphite, mica or PTFE.

Characteristics

The packing is sufficiently pliable when axially compressed to provide radial sealing for a moving shaft or rod.

It will not scratch or corrode the moving shaft or rod.

It requires frequent adjustment to compensate for packing wear.

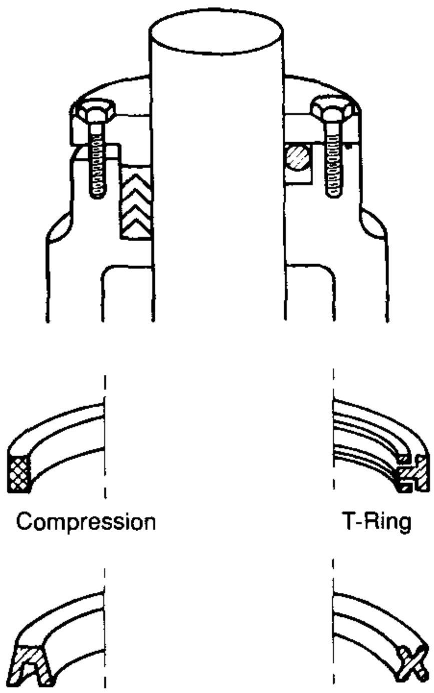


Figure 32. Rod Application With V-Ring And O-Ring

Molded Lip Type Packing

Lip type packing is molded from rubberized fabric, Nitrile, polyurethane, or PTFE. Two common shapes are U-ring and V-ring.

Characteristics

Distortion of the packing lips from an interference fit on assembly creates a counter sealing force adequate for low pressure.

With packing lips facing fluid pressure, any rise in pressure flattens the lips against wall surfaces and raises the sealing force.

Lip type packing has lower leakage, less friction, and longer life than compressed packing.

The U-ring is an effective seal when used singly. A ring support or pedestal with cross drilled holes assures equal pressure loading on both seal lips.

The V-ring needs a stack of three rings along with male and female adapters to contain low pressure fluid. Five or more rings are needed in the stack for high pressure.

Molded Squeeze Packing

Squeeze packing is molded from synthetic rubber, polyurethane or PTFE. Common shapes are the O-ring, T-ring and X-ring.

Characteristics

Distortion of the squeeze packing from an interference fit on assembly generates an internal sealing force within the packing to contain low pressure.

High pressure puts an additional squeeze on the packing, raising the internal sealing force to counter the pressure force. This action tends to extrude the packing through any clearance gap. Anti-extrusion rings are shown with the T-ring in Figure 32.

Squeeze packings have less friction than either compression or lip type packing and also seal in both directions.

On assembly the O-ring should be squeezed about 10 percent, the T-ring about 5 percent and the X-ring as low as 1 percent.

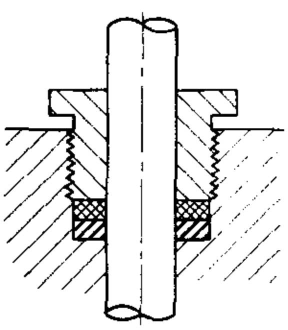
Probable Source of Failure	Suggested Remedy
1. Shaft or rod worn	Replace shaft. For contaminated fluid, change filter. For dirty atmosphere, install protective shield or boot. Check shaft hardness: Rockwell C 30 min.
2. Sealing surfaces are scratched. Seals damaged	Use proper assembly and dis-assembly tools on overhaul. Replace damaged parts.
3. Dynamic runout of shaft or eccentric motion is excessive	Inspect bearings, replace if too loose. Check side loads on shaft or rod. Seals are not to be used as bearings.
4. Rapid wear out of seal	Seal compressed too much. Loosen if adjustment is available Other-wise check to be certain of correct seal size.
5. Glazed or hardened	Check for high oil temperature. Correct if seal lubrication is inadequate.
6. Seal edges are extruded	Check parts for too much clearance. Replace faulty parts. Use anti-extrusion rings on low pressure side of seal.

Table VI. Packing Failures

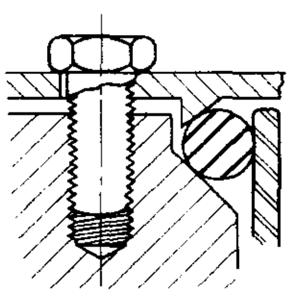
Static Seals

Installation Activated Seals

Static seals prevent leakage between stationary surfaces. To contain pressure, the seal and its mating parts must be in contact at a pressure level higher than the pressure to be sealed. This pressure level may be obtained by parts installation alone, as with the crush washer, or jam packing, Figure 33.



Crush Washer



Jam Packing

Figure 33.

Pressure Activated Seals

In other applications, such as with the O-ring, V-ring, and X-ring, Figure 34, the initial sealing pressure from installation alone is sufficient to contain only low pressure oil. High pressure deforms or changes the seal shape and increases the sealing pressure level to complete the seal.

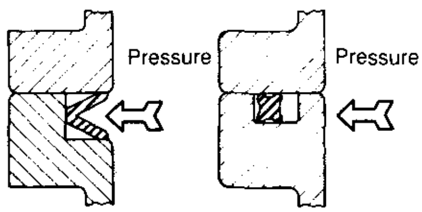


Figure 34. V-Ring and X-Ring

Gaskets

A gasket is an installation activated seal made of relatively soft material. It must be deformed or compressed to fill surface irregularities and close the gasket structure to fluid leakage.

The O-ring is replacing the gasket in many of the new hydraulic designs because of its greater reliability and ease of application.

In general, the more compressible (softer) materials are used for low pressure gasket applications. Common materials are asbestos, cork, paper, plastic, rubber or a combination of these materials.

The majority of gasket applications can be represented by two basic flange designs: the flat face, Figure 35, and the grooved face, Figure 36.

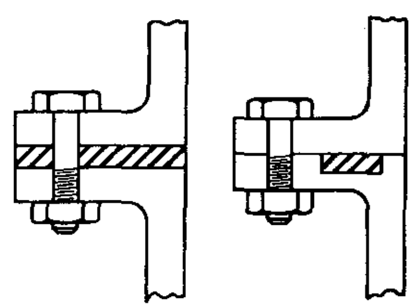


Figure 34. V-Ring and X-Ring

For a given gasket cross section, each material has a minimum clamping load necessary to close its structure to fluid leakage and fill surface irregularities to a maximum of 250μin. Higher internal pressures require higher clamping loads. The forces acting on a gasketed joint are shown in Figure 37.

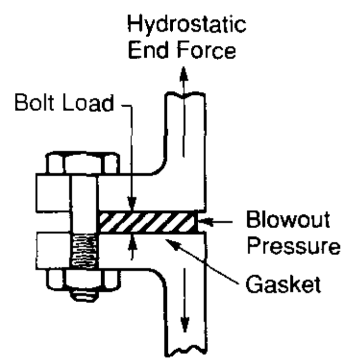


Figure 35. How Internal Pressure Acts On A Gasket Joint

Most gasket materials relax and creep after being clamped. A loss in loading occurs, most of it within 18 hours. For critical applications, wait a day after initial assembly and retighten to the original loading, preferably at system operating temperature but with no internal pressure.

As pressures rise and operating conditions become more severe, metal or metal-with-soft-core gaskets are necessary. The required clamping force and surface finish to obtain a good seal vary widely, depending on the type of gasket. Follow the manufacturer's recommendations.

O-Ring Static Seals

One of the most common static seals in the O-ring. It has found increasing use as a high pressure seal. Recommended surface roughness is 32 to 63 microinch.

With high pressure, the sealing surfaces may either slide or separate. Sliding causes seal wear. The rougher the surface, the greater the rate of seal wear. If sliding can't be prevented, surface roughness of 16 microinch may be necessary to obtain satisfactory seal life.

Separation of the sealing surfaces due to manufacturing tolerances or pressure loading permits the O-ring seal to be extruded into the clearance gap, Figure 38a. With pressure pulsations, the extruded edges will be nibbled away. Eventually, the seal will leak.

Seal Extrusion

It has been found that as O-ring hardness (called durometer) increases, its resistance to extrusion damage also increases. For example, in lab tests of a rod seal without back-up rings (Figure 38a) with 160°F (71°C) oil, 100,000 pressure pulses at 1500 psi (103 bar) caused significant O-ring extrusion damage when the diametral extrusion gap was greater than:

(a) .004 inches (.10 mm) with 70 durometer seals

(b) .008 inches (.20 mm) with 80 durometer seals

(c) .014 inches (.36 mm) with 90 durometer seals

Caution: Don't jump at the first chance to put in high durometer O-rings! As pressure levels rise, they do resist extrusion better than softer seals, but they leak more readily against rougher surfaces. Back-up rings (Figure 38b) can minimize extrusion on rod seals and are required where pressures exceed 1500 psi (103 bar).

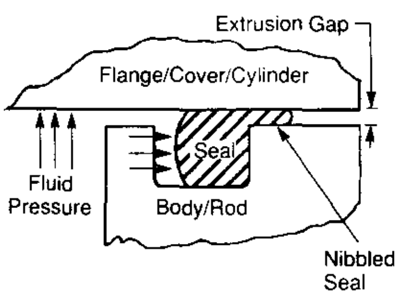


Figure 38a.

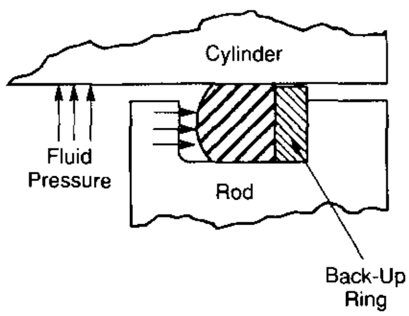


Figure 38b.

Remember: If two surfaces slide or separate, seal wear will always be a problem. Troy holding the surfaces together. Use a higher class of bolt so bolt torque may be increased. Coordinate this action with the parts supplier to avoid product failure, such as stripped threads.

In all work with O-rings, they should be protected:

(a) Lubricate with light grease or fluid to be sealed.

(b) Avoid use of sharp tools during assembly or parts removal.

(c) Use brass, paper or plastic cone to pass O-ring over threads.

SAE Split Flange Fitting (SAE J518)

The SAE split flange adapter fitting may be joined to either pipe, tube, or hose. It makes an excellent, easily removed, high pressure static seal. There are two pressure series: code 61, sometimes referred to as "Standard Series " (500 to 5000 psi) and code 62, the "High Pressure Series" (6000 psi). Special note is made of this design because of difficulties that may be encountered during assembly of the joint.

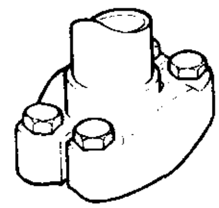


Figure 39. SAE 4-Bolt Split Flange

The split flange fitting seals on its face. The shoulder which contains the seal must be squarely against the mating surface and held there with even tension on all bolts. The shoulder sticks out past the flange halves by .010 to.030 inches to insure contact with the mating accessory surface before the flange halves do, Figure 40.

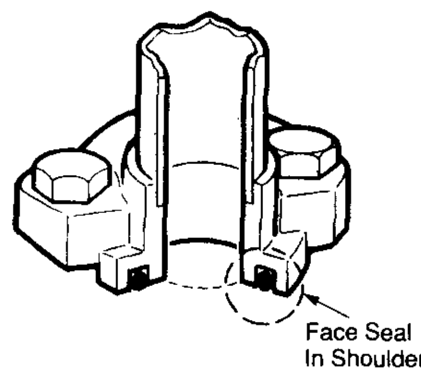


Illustration courtesy of Aeroquip Corp.

Figure 40. Cutaway View Split Flange Connection

Surface Finish

Before bolting parts together, examine the sealing surfaces. The seal will leak with gouged or scratched surfaces. They should be smooth. The seal will wear with rough surfaces–32 microinch is recommended but 64 is acceptable. The seal will extrude if the surfaces are not flat (Figure 38a)–all points over the surface should be within .0005 inches (.013 mm) of flatness.

The Problem

This connection is sensitive to human error. When bolts are tightened on one end, the flanges tend to tip up a seesaw fashion. The O-ring may get pinched, Figure 41. Use a new O-ring of correct size to match the flange and of correct material to match the fluid. Apply a light grease to the O-ring before assembly to hold it in place. Be sure all surfaces are clean. Finger tightening will help to get the flanges and shoulder started squarely.

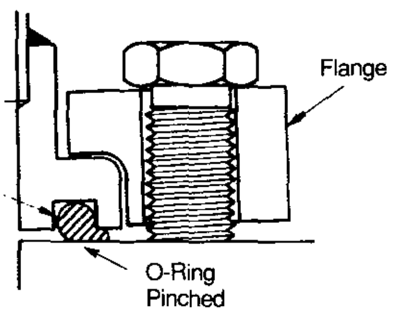


Illustration courtesy of Aeroquip Corp.

Figure 41. Tipped Flange and Shoulder

When excessive torque is applied to the bolts, the flanges often bend down until they bottom on the port face and bolts bend outward, Figure 42. Bending of flanges and bolts tends to lift the flange off the shoulder in the center area between the long spacing of the bolts. Much of the high torque on all bolts, which must be Grade 5 or better, is lost in overcoming the bending of the flanges and the bolts.

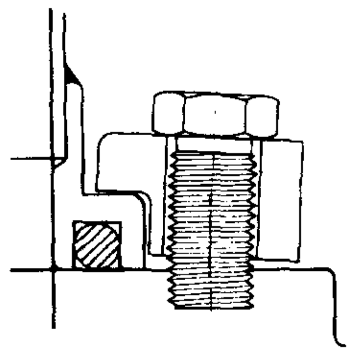


Figure 42. Bent Flanges Cause Bent Bolts

The Solution

As a solution, torque all bolts evenly. Do not tighten one bolt fully before going to the next bolt. Don't use air wrenches because they tend to cause flange tipping. All bolts are not alike. The higher the bolt grade, the stronger the bolt. Always use graded bolts, with the head identified as in Figure 42, and torque to the values recommended in Table VII. Socket head bolts may be used in place of hex head since they are grade 8 or better.

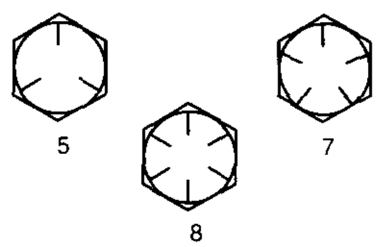


Figure 43. Grade Identification

SAE O-Ring Face Seal Fitting (SAE J1453)

Another face seal fitting, similar to SAE split flanges, is also available. This fitting provides a resilient seal and is capable of high pressures, similar to split flanges. The SAE O-ring face seal fitting uses a swivel nut to hold the flange head in lieu of split clamps, and the O-ring in on the port adapter in lieu of the flange head, Figure 44.

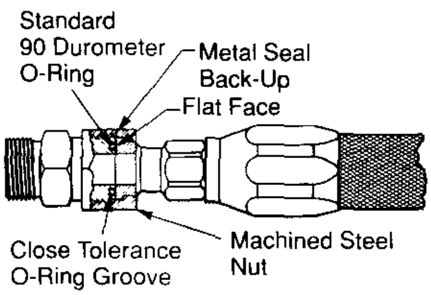


Illustration courtesy of Aeroquip Corp.

Figure 44.

The Aeroquip ORS connection has an o-ring groove machined into its flat male face. This flat face and o-ring mate with the connection's other close tolerance machined face to form a virtually leak-free seal.

As in all face type fittings, it is important that sealing surfaces be parallel as well as clean and free from burrs. Clamping forces may not be able to correct for nonparallel faces.

In a properly fabricated assembly, bottoming of the faces will have a solid feel, thus making this fitting less prone to error and less torque-sensitive than 37° flare or bite-type fittings. Table IV provides recommended torque values for Aeroquip Corporation's ORS™ brand SAE O-ring fittings.

Nominal Tube O.D. Inches (mm)	Connection Size	Torque In. Lb. (Nm)	
		Code 61	Code 62
$^1/_2$ (12.7)	-8	175-225 (20-25)	175-225 (20-25)
$^3/_4$ (19.1)	-12	250-350 (28-40)	300-400 (34-45)
1 (25.4)	-16	325-425 (37-48)	500-600 (56-68)
$1^1/_4$ (31.8)	-20	425-550 (48-62)	750-900 (85-102)
$1^1/_2$ (38.1)	-24	550-700 (62-79)	1400-1600 (158-181)
2 (50.8)	-32	650-800 (73-90)	2400-2600 (271-294)

Table VII. SAE 4-Bolt Flange Torque

Static seal leakage problems and solutions are in Table VIII.

Possible Source Of Trouble	Suggested Remedy
1. Seal has extruded or been nibbled to death	Replace seal and check the following: a. Sealing surfaces must be flat within .0005 inches (.013 mm), replace part if out of limits. b. Initial bolt torque may have been too low. Check manual for proper torque. c. Pressure pulses may be too high. Check for proper relief valve setting. d. If normal operating pressures exceed 1500 psi (103 bar), back-up rings are required.
2. Seal is badly worn	Replace seal and check the following: a. Sealing surface too rough, polish to 16 μ in. if possible or replace part. b. Undertorqued bolts permit movement. Check manual for correct setting. c. Seal material or durometer may be wrong. Check manual if in doubt.
3. Seal is hard or has taken excessive permanent set	Replace seal and check the following: a. Determine what normal operating temperature is. Check system temperature. b. Check manual to determine that correct seal material is being used.
4. Sealing surfaces are scratched, gouged or have spiral tool marks	Replace faulty parts if marks cannot be polished out.
5. Seal has been pinched or cut on assembly	Use petrolatum to hold seal in place with blind assembly. Use protective shim if seal must pass over sharp threads.
6. Seal leaks for no apparent reason	Check seal size and parts size. Get correct replacement parts.

Table VIII. Static Seal Leakage Analysis

VICKERS

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Troy, Michigan
48007-0302
USA

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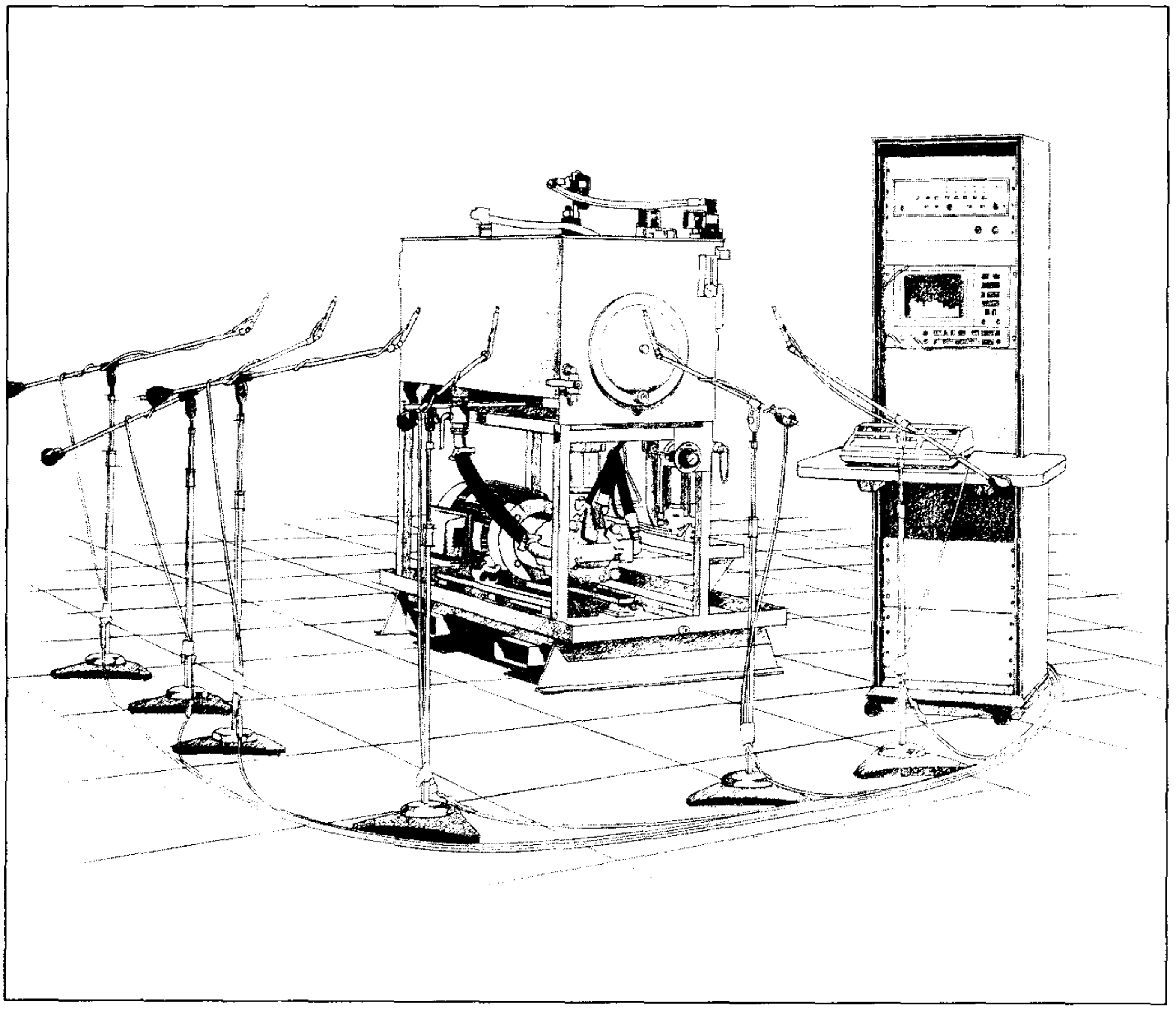
Noise Control in Hydraulic Systems



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INTRODUCTION

Although a certain amount of noise control is required in hydraulic systems just to conform to government regulations, a conscientious noise control program actually provides a competitive edge. The combination of a quiet pump, well-engineered vibration and pulsation controls, and good, economical installation practices will result in a product with a distinct advantage in the marketplace.

The Vickers publication "More Sound Advice," issued in the 1970's, illustrated a variety of machine noise control methods. Because of the nearly infinite variety of hydraulic applications, it's not possible here to discuss the individual features of particular systems. There are, however, a number of installation techniques that can be applied to almost *all* hydraulic systems. When used correctly, these techniques can yield significant reductions in noise.

This publication describes the following:

1. Noise generation and noise control techniques.
2. Noise terms, definition and the use of the decibel.
3. Noise measurement procedures.

SYSTEM NOISE CONTROL

Quiet Hydraulics – A Team Effort

A successful noise control program requires a team effort by individuals in several areas of expertise. A quiet hydraulic pump does not guarantee a quiet system. The choice of a quiet pump should be only one part of a multifaceted program that calls upon the talents of the system designer, fabricator, installer, and maintenance technicians. And if any member of the team fails to do their job, it can mean failure of the entire noise control program.

The pump and system designers play a key role in achieving successful noise control. They must evaluate every noise control technique available from the standpoints of both cost and practicality. Three of the basic approaches used in quieting hydraulic power systems are:

A) Internal and external pump pulsation control

B) Pump and structure isolation

C) Damping and/or stiffening

Noise Transmission and Generation

Noise is defined as the unwanted by-product of fluctuating forces in a component or system. In a hydraulic system, this noise can be transmitted in three ways: through the air, through the fluid, and through the system's physical structure. We generally think of noise as travelling only through the medium of the air, going directly from its source to some receiver (our ear). This is called *airborne noise*. That airborne noise, however, must have a source within some component of the hydraulic system. That component is normally the pump. Whether it's a piston, vane, or gear pump, the internal pumping and porting design can never be perfect. As a result, uneven flow characteristics and pressure waves are created and transmitted through the fluid. This is known as *fluidborne noise*. The pressure wave fluctuations of fluidborne noise in turn create corresponding force fluctuations. These result in vibration, also known as *structureborne noise*. This structureborne noise is transmitted not only through the pump body, but through attached structures as well. These structures then emit an audible sound.

The surrounding structures and surface areas in a hydraulic system tend to be much larger than the pump itself, and therefore radiate noise more efficiently. For this reason, while the pump design should minimize internal pulsations, it's also important to use proper isolation techniques to keep the remaining vibrations from reaching adjoining structures.

Design for Low Noise

An intelligent program of noise control should start at the source: the pump. A quiet pump is the responsibility of the pump manufacturer. The problem for the designer is that although a hydraulic pump is required to perform over a wide range of speeds and pressures, noise control can only be optimized for a relatively narrow portion of that range. The most common strategy is to use porting design to limit the pressure pulsations at the pump's rated speed and pressure. The pulsations are reduced as much as possible without creating a large amount of noise at lower speeds and pressures. Piston, vane, and gear pumps are similar in that their total output flow is the sum of the flows from the individual pumping elements or chambers. Fluid fills the chambers at the pump inlet, is compressed mechanically and/or hydraulically through orifices, and is then combined into a single discharge flow. Each pumping element in a piston pump delivers its fluid to the discharge port in a half-sine profile. The pump discharge is the total of the equally spaced half-sines added in phase. The result is an inherent flow ripple, as shown in Figure 1 for a nine-piston pump. This ripple is independent of any fluid compression, either through piston motion or any type of internal hydraulic metering. Vane pump flow ripple is more controllable. Cam contours can be designed to reduce mechanical compression effects. This is done by making pressure transitions in the dwell section, where there is controlled change in vane chamber volume. For this reason, vane pumps will normally generate less noise over a wider range of speeds and pressures than piston pumps.

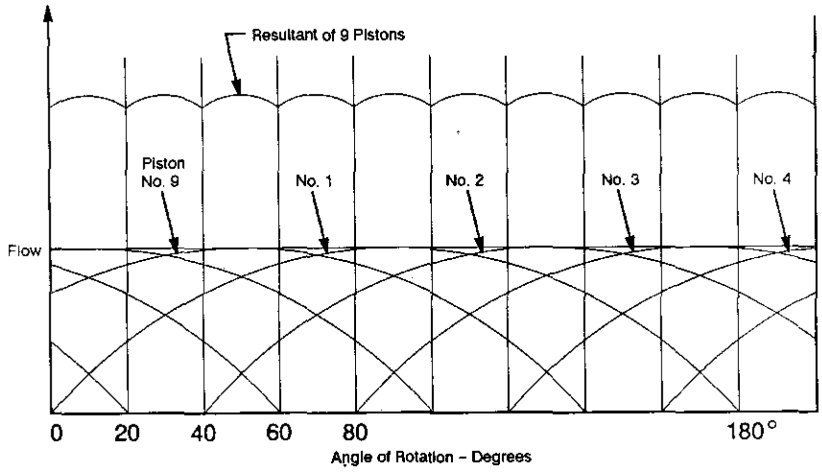


Figure 1. Inherent Piston Pump Discharge Ripple

Noise Frequencies

Pump noise energies are generated in several ways. Vibrational energy is created by an imbalance in the pump, drive motor, or couplings. It can also be produced by some undesired interaction in the assembly, but it's rare for any significant audible noise to be generated by these interactions. Nonetheless, care should be taken to minimize its effects on pump or motor life. Figure 2 shows the frequency spectrum of a ten-vane pump operating at 1800 rpm with shaft rotation frequency of 30 Hz. Any misalignments in the power train will produce noise components at twice and four times this frequency.

The strongest energy components occur at pumping frequency. This frequency equals the number of pumping chambers times the shaft frequency (300 Hz in Figure 2). Noise energy is also produced at multiples, or harmonics, of this frequency. 600 Hz and 900 Hz are the second and third harmonics seen in Figure 2. These harmonics have enough amplitude to produce significant noise. This noise comes not only from the pump itself, but from attached structures which are often more efficient at radiating the noise transmitted from the pump.

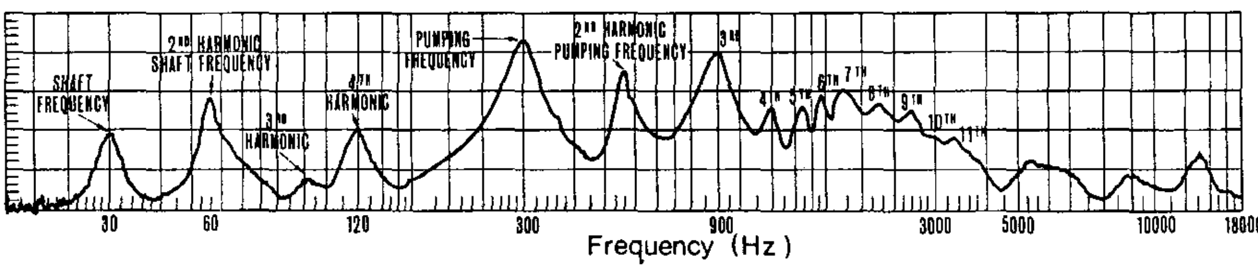


Figure 2. Structureborne or Fluidborne Spectrum Identifying Shaft and Pumping Frequency and Harmonics

Vibration Noise Control

Vibration control is used to prevent pulsation energy from the pump from being transmitted to machine structures. Most pump and drive motor assemblies are attached through a flexible coupling and mounted on a common base to maintain alignment. The common base is resiliently mounted to the support structure, as seen in Figure 3. An isolator should be selected that has a natural frequency approximately $^1/_2$ or less the pump's rotational frequency. For example, an isolator with a natural frequency of 10 Hz or less would be appropriate for a pump with a rotational or forcing frequency of 20 Hz at 1200 rpm, and would work even better for a pump with a frequency of 30 Hz at 1800 rpm. The higher the ratio between the forcing frequency and the natural frequency of the isolator system, the greater the amount of isolation (see Figure 4). A typical commercial isolator (which costs about $15 in moderate quantities) can reduce transmitted vibration energy by 10 dB at 1200 rpm and 15 dB at 1800 rpm (Figure 5).

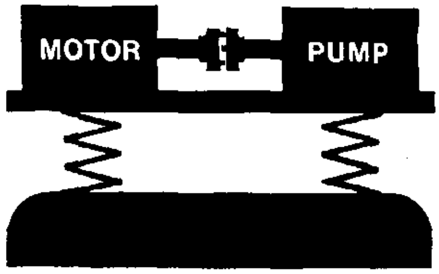


Figure 3. Pump and Motor on Subplate, Isolated from Stiff Foundation

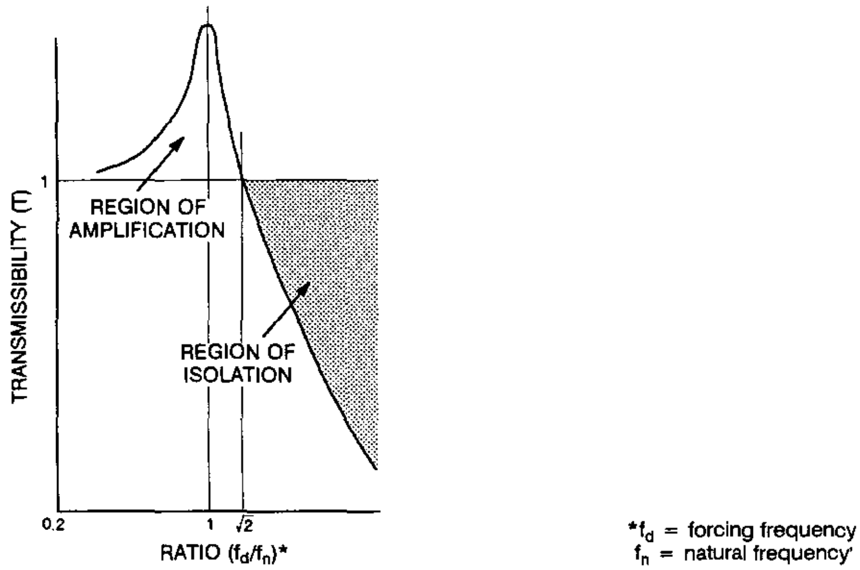


Figure 4. Typical Transmissibility Curve for an Isolated System

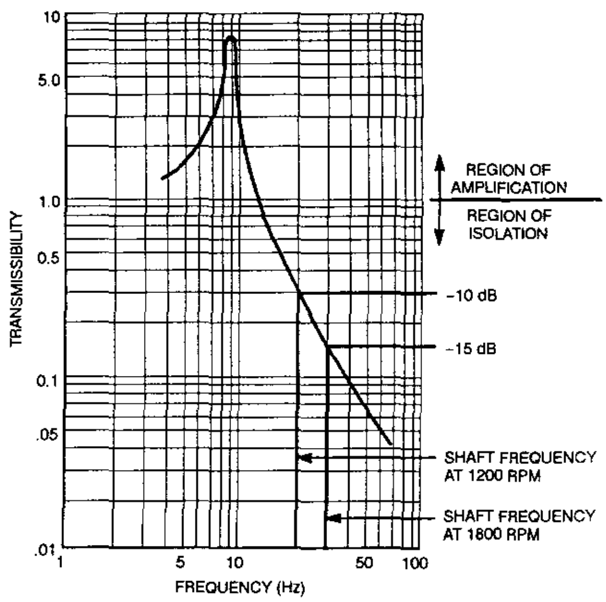


Figure 5. Transmitted Vibration Energy Reduction Using Typical Commercial Isolator

Isolators are classified by their load carrying capacity and related natural frequency. When pump, motor, and subplate assembly weights are known, the amount of evenly distributed weight on each of the isolators can be calculated. Isolators should be selected that will not be loaded above 60% to 70% of their capacity. This will allow a sufficient safety margin in the event of shock loading.

The same type of isolators can also be used on power units with overhead reservoirs. Eight isolators can be installed either under the reservoir feet (Figure 6), or under the upright leg structures supporting the reservoir. The isolators shown would be very effective because the ratio of forcing frequency to natural frequency is very high. For example, a nine-piston pump operating at 1200 rpm would have a pumping frequency of 9 x 20 rev/sec or 180 Hz. If a 10 Hz isolator system were used, there would be a very low level of vibration transmission, because the ratio between the two frequencies would be 18:1.

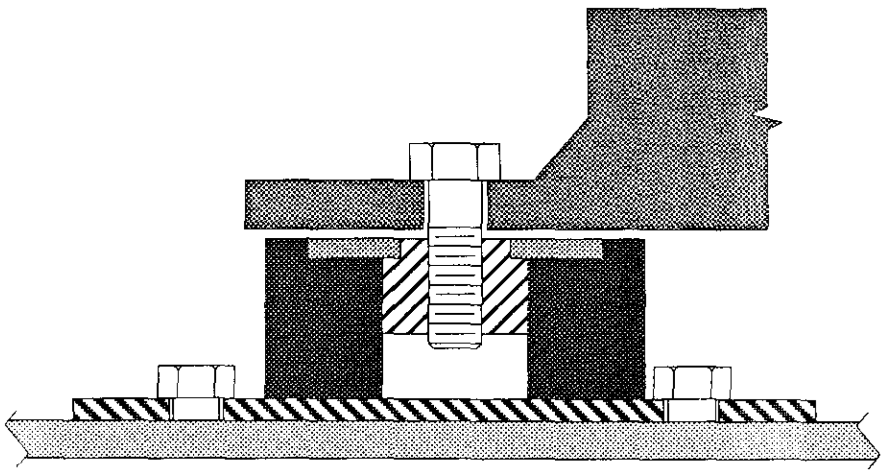


Figure 6. Reservoir Foot On Isolator

The chart on the right lists load ratings of typical isolators with carrying capacities of 60 to 4400 lbs. per isolator.

Load Series	Isolator Number	Max. Static Load Per Isolator (lbs.)
Light	L1 L2 L3 L4 L5	60 100 130 200 260
Medium	M1 M2 M3 M4	300 450 550 700
Heavy	H1 H2 H3	700 1000 1500
Extra Heavy	EH1 EH2 EH3 EH4	1500 2000 3000 4400

Table 1 on the following page provides two examples of proper isolator selection:

A)	Pump, motor, and subplate weight	=	800 lbs.
	Load on each of 6 isolators	=	133 lbs.
	Maximum static load when loaded to 60 to 70% of capacity	=	190 to 222 lbs.
	Selection: # L4 (from chart on page 4)		
B)	Reservoir weight	=	550 lbs.
	Attached accessories weight	=	150 lbs.
	100 gallons oil weight (7 lbs./gal.)	=	700 lbs.
	TOTAL	=	1400 lbs.
	Load on each of 4 isolators	=	350 lbs.
	Maximum static load when loaded to 60 to 70% of capacity	=	500 to 583 lbs.
	Selection: # M3 (from chart on page 4)		

Table 1. Isolator Selection

Isolation With Hose

Rubber hose must be used to maintain the vibration isolation afforded when the pump and motor assembly are mounted on isolators. The isolation capabilities of hose will reduce the amount of vibration energy entering the system. Unfortunately, improper use of hose is probably the main cause of noise in many systems.

Although structureborne noise can be reduced by using long lengths of hose, the pressure pulsations from the pump will cause the hose to undergo cyclic radial expansion. One of the drawbacks of hose is that it acts as an efficient radiator in the frequency range where most of the energy is generated: the first few harmonics. Because of these two factors, long lines of hose are less effective for noise reduction than the use hose at either end of a solid line (Figure 7).

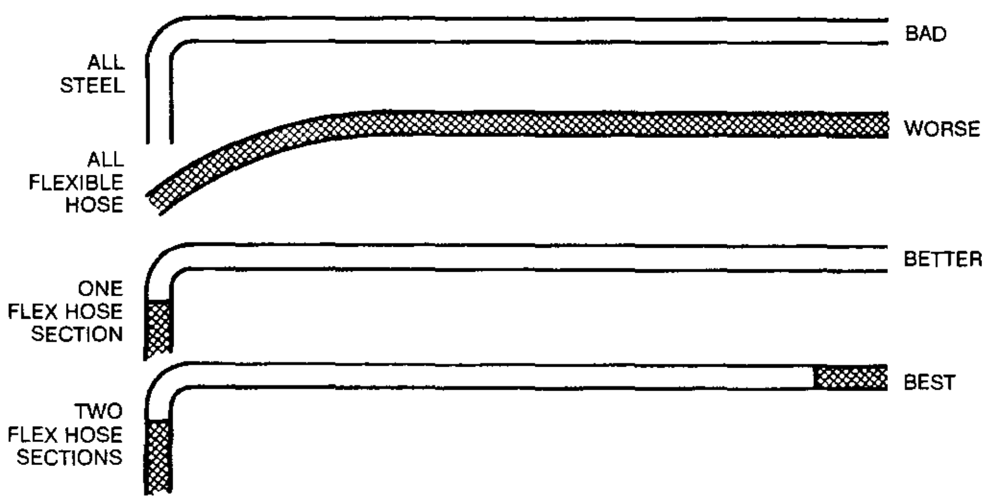


Figure 7. **Long Hydraulic Line Configurations**

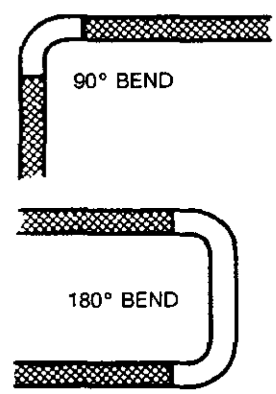
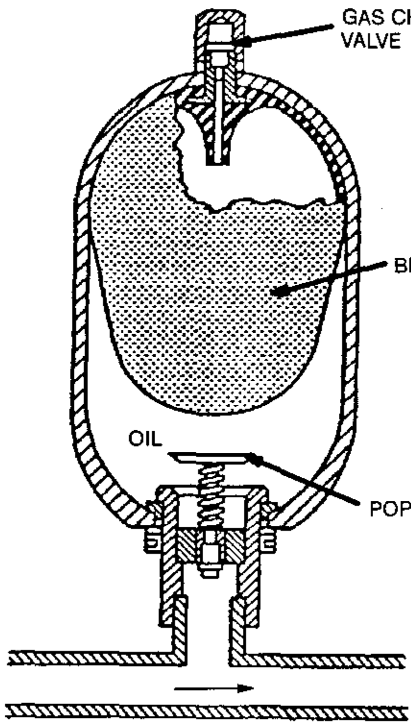


Figure 8. Preferred Short Hydraulic Line Configurations

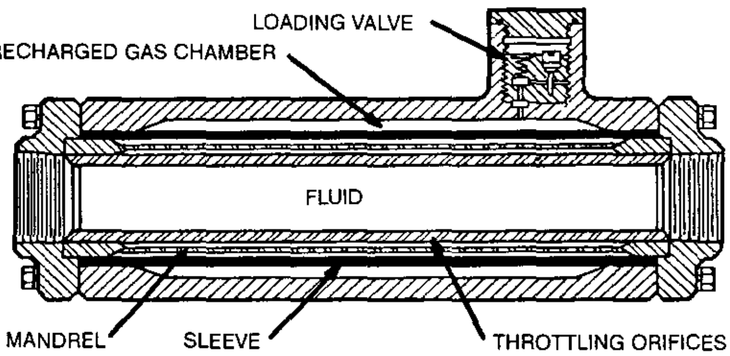
Two other shortcomings of hose are that its length changes with pressure, and that when it's bent through a radius it acts like a Bourdon tube, trying to straighten out with increasing pressure. Both produce forces that act on connecting structures. The best way to maintain noise control while making bends with hose is to use a solid elbow with a hose section on each end. This eliminates problems caused by the Bourdon tube effect. Any change in the length of one hose is accommodated by bending in the other hose. Figure 8 illustrates the preferred configurations for 90° and 180° bends.

Fluidborne Noise Control

Fluidborne noise control begins with a pump's internal design. The ports should be configured so that the lowest practical pressure pulsations are generated. Additional external controls can be added to prevent as much pulsation energy as possible from being communicated to the system. This is usually done by adding expansion volume at the outlet of the pump. These acoustic filters, as they are called, can take many forms. The two most common are gas charged side branch accumulators and flow-through type pulsation filters. Each has its advantages and disadvantages. The side branch type is cheaper, but limited to attenuating pulsations in only a narrow range of frequencies. As a result, it isn't totally effective throughout the first four pump harmonics (where most of the pulsation energy is generated). The flow-through device, although generally larger and more expensive, has a distinct performance advantage: pulsations throughout the spectrum are reduced, including those at the most significant harmonics. The filters (shown in Figure 9) have optimum effectiveness when gas charged to $^1/_3$ the maximum operating pressure of the hydraulic system.



SIDE BRANCH ACCUMULATOR



PASS-THROUGH GAS CHARGED FILTER

Figure 9. Acoustic Filters

Application of Techniques

Experimental evaluations were made on a typical automotive application: a power unit with a 150 gallon overhead reservoir supplying oil to a piston pump. The pump delivered 40 gpm at 1200 rpm at a pressure of 750 psi. The noise level, as received, was 88 dB(A). (Standard accepted noise measurement procedures are explained in the subsection entitled System Noise Evaluation.) Four different noise reduction techniques were applied to the system, starting with those that would have the greatest effect on noise levels. The results are shown in Table 2.

ITEM	DESCRIPTION	RESULTING NOISE LEVEL	CHANGE IN NOISE LEVEL
1	Changed radiused pressure hose to two pieces, separated by right angle fitting.	83 dB(A)	-5 dB(A)
2	Item 1, plus structure isolators under reservoir and upright supports (8 additional isolators).	79 dB(A)	-4 dB(A)
3	Items 1 and 2, plus valve plate designed for 1000 psi rather than 3600 psi. (Pulsations reduced from 200 psi to 140 psi.)	76 dB(A)	-3 dB(A)
4	Items 1, 2, and 3, plus flow-through pulsation filter. (Pulsations reduced from 140 psi to 35 psi.)	74 dB(A)	-2 dB(A)

Table 2. Noise Reduction Techniques

It's important to note that if the changes outlined above had been made in some other sequence, the amount of noise reduction at each step would have been different from that shown - particularly if items 3 and 4 had been evaluated before items 1 and 2. The final noise level of 74 dB(A) would be the same, but the first item tried wouldn't have yielded such a significant reduction. In the example shown, the areas of highest noise radiation were addressed first. This is essential because no appreciable noise reduction can be achieved unless the most significant noise source is identified and its level reduced first.

NOISE TERMINOLOGY

Noise Terms and Equations

Sound at a particular point in air is defined as the rapid variation in air pressure around a steady state value. Sound pressure is measured in the same units as atmospheric pressure. Since it's an alternating quantity, the term "sound pressure" is usually referred to by its root mean square (rms) value. At a frequency of 1000 Hz, a sound with an rms pressure of 2×10^{-4} microbars (ubar), or about 2×10^{-10} atmospheres, is just below the hearing threshold of someone with good ears. Expressed in more familiar terms, that level of sound pressure would be 2.9×10^{-9} psi. The fact that slightly greater pressures become audible shows the amazing sensitivity of the human ear. It can detect variations in atmospheric pressure as small as a few parts in 20,000,000,000.

In addition to this sensitivity, the human ear has an enormous dynamic range. Not only can it detect sounds as small as 2×10^{-4} ubar, it can accommodate sound pressures as high as 2000 ubar without being overloaded, i.e. causing pain. That's a dynamic range ratio from threshold to pain of 10,000,000:1 (Figure 10). Because this range is so large, it's more convenient to express ratios in powers of 10 (hence the use of the log scale). Sound pressure above the reference value of 2×10^{-4} ubar is referred to as sound pressure level (SPL) and expressed in decibels (dB).

$$\boxed{SPL = 20 \log \frac{P}{P_o}}$$

OR

$$\boxed{SPL = 10 \log \left(\frac{P}{P_o}\right)^2}$$

Where SPL = Sound pressure level (dB)
P = Sound pressure (bar)
P_o = Reference pressure (.0002 ubar)

From this equation, a pressure ratio of 10,000,000 (10^7) would result in the following noise level:

$$\begin{aligned} SPL &= 20 \log 10^7 \\ &= 7\,(20) \log 10 \\ &= 7\,(20)\,(1) \\ &= 140 \text{ dB (i.e., painful)} \end{aligned}$$

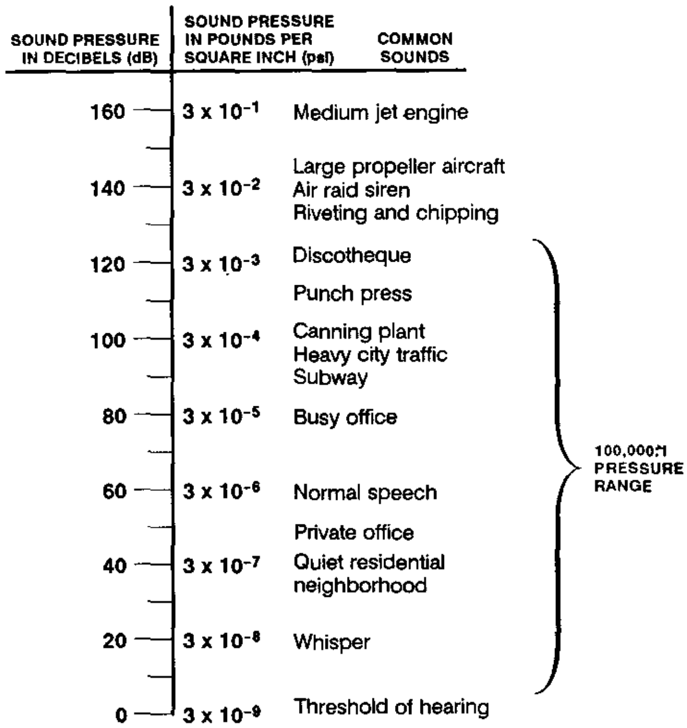


Figure 10. Dynamic Range of the Human Ear

Pressure ratios can also be calculated based on the changes in sound pressure levels (ΔSPL). For example, what is the pressure ratio (Rp) if the noise level changes by 3 dB?

$$Rp = 10^{\Delta SPL/20}$$

Where Rp = pressure ratio
$= 10^{3/20}$
$= 1.41$

There are two important conclusions to note here: If the noise level increases from 82 to 85 dB, there's actually a 41% increase in noise; if the noise level decreases from 85 to 82 dB, there's a 29% decrease in noise ($^1/_{1.41}$ or 71% of the original level).

Table 3 lists the pressure ratios for changes in SPL from +10 to -10 dB with the previous example in bold:

Change In SPL	Press. Ratio	Change In SPL	Press. Ratio
1	1.12	-1	.89
2	1.26	-2	.79
3	**1.41**	**-3**	**.71**
4	1.59	-4	.63
5	1.78	-5	.56
6	2	-6	.5
7	2.24	-7	.45
8	2.51	-8	.40
9	2.82	-9	.35
10	3.16	-10	.32

Table 3. Pressure Ratios

(A chart of typical noise levels is shown in Figure 10.)

Human Response to Noise – The "A" Scale

A microphone measures actual sound pressures emitted from a noise source, but the human ear doesn't treat equal levels with equal tolerance over the audible frequency range of up to 12,000 Hz. The ear is more sensitive to noise above 1000 Hz. This sensitivity is simulated by using the "A" scale filtering system in the signal processing of measured noise. This internationally standardized system gauges the ear's response to noise through the use of a passive frequency related filter placed between the microphone and output (Figure 11). The resulting energy, when summed in the respective weighted frequency bands, is then expressed in "A" scale, "A" weighted, or dB(A) levels.

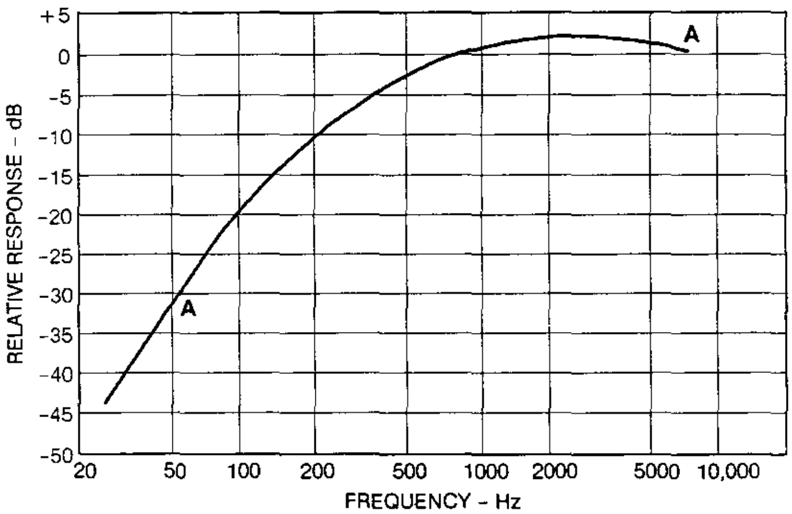


Figure 11. Response Characteristics of Standard A Filter

Sound Power

Sound pressure levels, in dB(A), are one measure of the noise of a source, but sound can also be expressed in terms of sound power level (PWL), also in dB(A). Sound power is the actual acoustic radiation (though not measurable), expressed in watts. This sound power remains constant, whereas sound *pressure* decays as the distance from the source increases. A good analogy would be light bulbs, which are classified in terms of watts (power) - not illumination level, which falls off with increasing distance.

Sound power levels can be calculated with the following formula:

$$PWL = 10 \log \frac{W}{W_o}$$

Where PWL = sound power level in dB or dB(A)
W = acoustic radiation of source in watts
W_o = reference radiation in watts (10^{-12})

As in the expression of pressure levels, the power level is a logarithmic expression of ratio. Applying the same 3 dB change in power level as was previously calculated for pressure level results in:

$$R_w = 10^{\Delta PWL/10}$$

Where R_W = power ratio
= $10^{3/10}$
= 2.0

In this example, if the power level increases from 82 to 85 dB, the power increases by a factor of 2.0; if the power level decreases from 85 to 82 dB, there's a 50% decrease in power ($^1/_2$ or 50% of the original level).

Table 4 lists the power ratios for changes in PWL from +10 to -10 dB with the above example in bold:

Change In PWL	Power Ratio	Change In PWL	Power Ratio
1	1.26	-1	.79
2	1.59	-2	.63
3	**2**	**-3**	**.5**
4	2.51	-4	.40
5	3.16	-5	.32
6	4	-6	.25
7	5	-7	.2
8	6.31	-8	.16
9	7.94	-9	.13
10	10	-10	.1

Table 4. Power Ratios

Relationship Between Sound Pressure and Sound Power

The correlation between sound pressure and sound power can be seen by comparing the two ratio tables shown above. For an equal change in decibel level the ratios are different. For example:

Change In Db	Pressure Ratio	Power Ratio
3	1.41	2.0

The relationship is such that ***power is proportional to the pressure squared.***

Effect of Distance on Noise Levels

In an environment where noise is radiated from a source into a reflection-free space, called a free field, the sound pressure level will vary according to the following formula:

$$\Delta\ SPL = 20 \log \frac{d_1}{d_2}$$

OR

$$\Delta\ SPL = 10 \log \left(\frac{d_1}{d_2}\right)^2$$

Where d_1 = initial distance from sound source (noise standards specify either 3 feet or 1 meter)
d_2 = distance of observer (greater than d_1)

This forms the basis for the *inverse square law*. If the distance of observer is doubled, the noise level is decreased by 6 dB.

For example:

$$d_1 = 3 \text{ feet}$$
$$d_2 = 6 \text{ feet}$$

then

$$\Delta SPL = 20 \log .5 = -6 \text{ dB}$$

OR

$$\Delta SPL = 10 \log (.5)^2 = -6 \text{ dB}$$

Noise Addition

Noises can only be added or subtracted on the basis of acoustic **power** - *not pressure*. Noise sources of equal sound pressure levels are combined as follows:

$$SPL_2 = SPL_1 + 10 \log X$$

SPL_2 = noise level of all sources
SPL_1 = noise level of 1 source
X = number of sources

For example:

$$SPL_1 = 80$$
$$X = 3 \text{ sources}$$
$$SPL_2 = 80 + 10 \log 3 = 80 + 4.8 = 84.8 \text{ dB}$$

A chart showing this relationship appears in Figure 12.

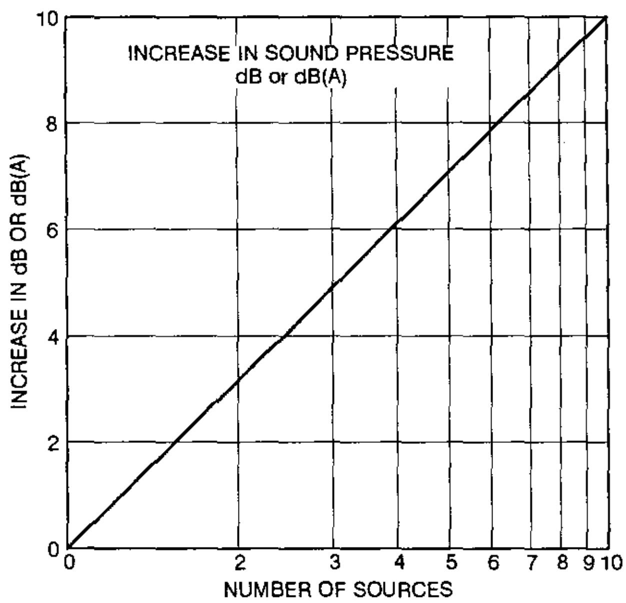


Figure 12. Addition of Equal Sounds

Unequal sound pressure levels can also be added. (See Figure 13.) The difference in the pressure levels of two sounds is used to determine how much their combined level will exceed the higher of the two. To add a third level, use the same process to combine it with the total from the first two levels.

The following example shows the two steps needed to find the total noise from sources of 70, 74, and 77 dB:

A) 74 - 70 = 4 dB (X Axis)
dB to add to larger = 1.5 (Y Axis)
TOTAL = 75.5 dB

B) 77 - 75.5 = 1.5 dB (X Axis)
dB to add to larger = 2.3 (Y Axis)
TOTAL = 77 + 2.3
= 79.3 dB

The total can also be calculated from the equation:

$$SPL = 10 \log \Sigma\, 10^{SPL/10} = 10 \log (10^{7.4} + 10^{7.0} + 10^{7.7}) = 79.3 \text{ dB}$$

NOISE MEASUREMENTS

Component Evaluation and Rating

To assist machine tool builders in their selection of components on the basis of noise, the National Fluid Power Association (NFPA) developed a standard that assures uniformity in the measurement and reporting of sound levels. This standard, T3.9.12, contains guidelines for obtaining standardized sound ratings. It is concerned only with the radiated noise of components, primarily pumps.

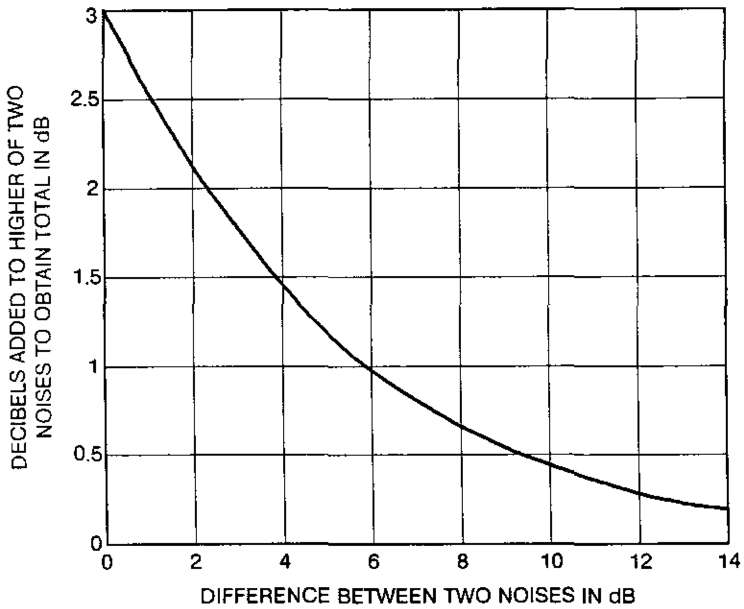


Figure 13. Addition of Unequal Sounds

The rating is usually expressed in dB(A) at a distance three feet in a free field above a reflecting plane (semi anechoic). This is a computed figure derived from a mathematical model. This model assumes that all the sound power from a pump is radiated from a single point located in the center of a hypothetical test hemisphere (Figure 14). The standard allows for masking all parts of the circuit that might contribute to noise. This includes wrapping hydraulic lines and enclosing any load valves. ***All radiated noise must be attributable to the pump, with no corrections for background.***

Microphones are positioned on spatial coordinates, each of which is located at the centroid of equal areas of the hemisphere surface. The rule of thumb is to use one microphone for each square meter of area. With the area of the hemisphere at $2\pi r^2$, and r equal to approximately 1 meter, six microphones are sufficient.

Power is defined as follows:

$$F = P \times A$$

where F = power
P = sound pressure
A = area acted on by P ($2\pi r^2$)

In terms of decibels:

$$PWL = \overline{SPL} + 10 \log 2\pi r^2$$

where PWL = sound power level
$\overline{SPL}$ = average sound pressure level of 6 microphones
r = radius in meters (3 feet = .914 meters)

Therefore:

$$PWL = \overline{SPL} + 10 \log 2\pi + 20 \log .914$$
$$= \overline{SPL} + 8 - .8$$
$$= \overline{SPL} + 7.2$$
(for 3 feet)

OR

$$PWL = \overline{SPL} + 8.0$$
(for 1 meter)

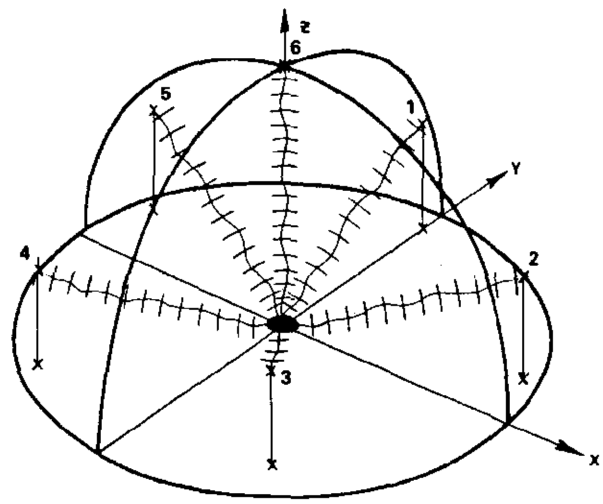


Figure 14. Microphone Positions for Measuring Pump Noise

System Noise Evaluation

The procedure for measuring system noise is different from the one used for components. Power unit systems are normally located in areas where background acoustics cannot be controlled. Guidelines for measurement in such environments are included in the National Machine Tool Builders Association's (NMTBA) "Noise Measurement Techniques" booklet.

Microphones are positioned ***1 meter from the perimeter of the machine and 1.5 meters above floor level***, as shown in Figure 15. It's extremely important that these distances be measured accurately. This insures uniformity of measurement and comparison, and compliance to customer noise level specifications. A 4-inch position error at a nominal 1 meter distance can result in a 1 dB error in measurement accuracy.

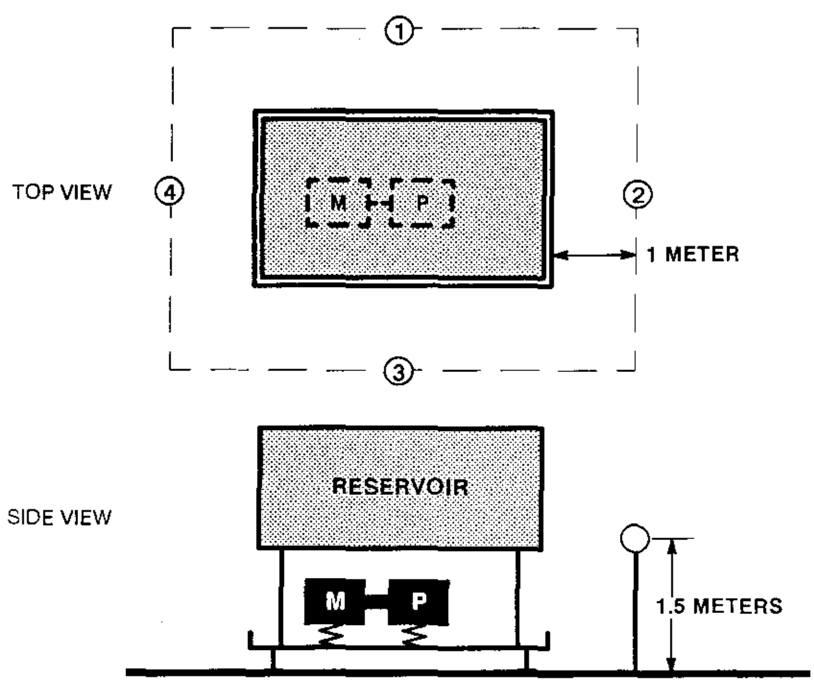


Figure 15. Microphone Positions for Measuring System Noise

At the very least, measurements should be taken on all four sides of the machine. It may also be necessary to measure at other locations around the envelope if highly directional noise levels are evident in other spots. Documentation of conditions is also needed to create some reference for comparison to other installations, acoustic environments, or pump types. The following conditions should be recorded:

- motor/pump speed
- pump type
- pump delivery
- operating pressure
- fluid type and temperature
- load valve location (if used)

Correction for Background Noise

Accurate system noise measurement may involve correcting for the noise of the surrounding area. When ambient sound levels are within 10 dB(A) of the levels when the machine is operating, correction factors may be applied. This is done in accordance with Table 5, derived from the NMTBA booklet.

Increase In Sound Level Due To Machine Operation (dB(A) above ambient)	**Correction Factor To Be Subtracted From Measured Sound Level** (dB(A))
3 or less	3
3 to 6	2
6 to 9	1
10 or more	0

Table 5. Background Correction Factors

One advantage of this chart is that it allows the use of whole numbers for dB(A). It's actually a "rounding off" of the curve shown in Figure 16 for the subtraction of sound levels. The example shown in the figure can also be expressed as:

$$\text{machine noise} = 10 \log [10^{6.0} - 10^{5.3}]\ \text{dB} = 59\ \text{dB}$$

Documentation of noise measurements made using four microphone positions might look like that shown in Table 6.

Noise level should be expressed as the maximum level measured (in this example, 78 dB in position 3) or possibly as the average of the four levels:

$$\overline{SPL} = 10 \log \left[\frac{10^{7.4} + 10^{7.0} + 10^{7.8} + 10^{7.5}}{4}\right] = 75\ \text{dB(A)}$$

Measuring Position	1	2	3	4
		dB(A)		
Total noise	76	73	79	75
Background noise	71	70	70	65
Background correction	-2	-3	-1	0
Machine noise	74	70	78	75

Table 6. Noise Measurement Documentation

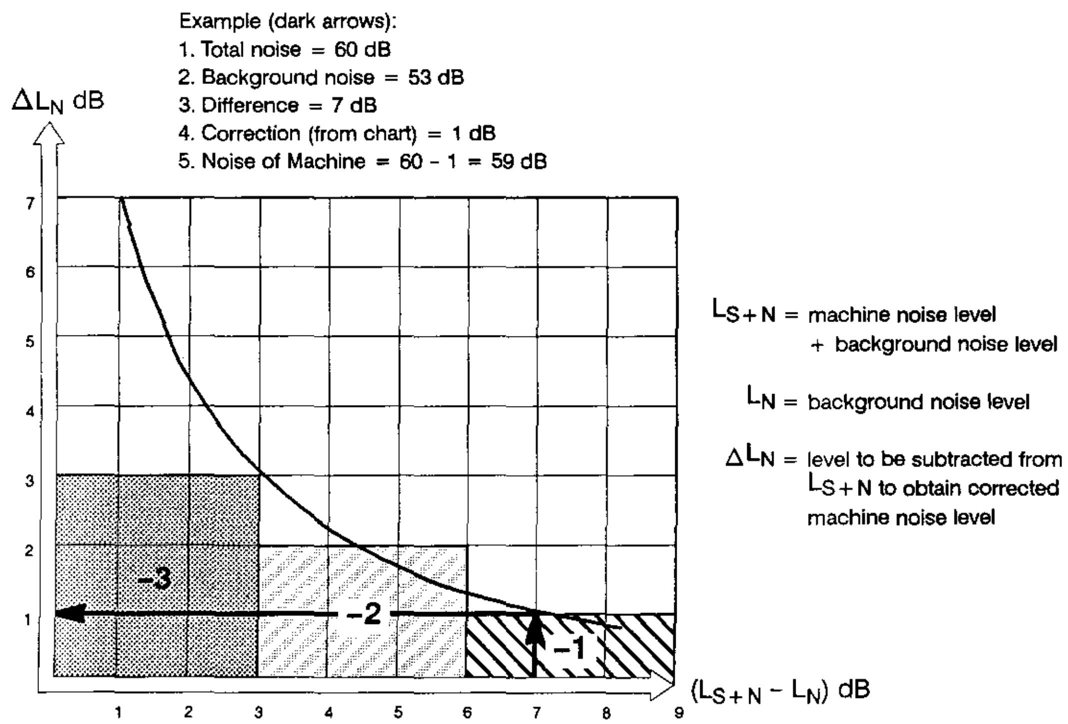


Figure 16. Subtracting Sound Levels

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