AL-17A
(REV. I)

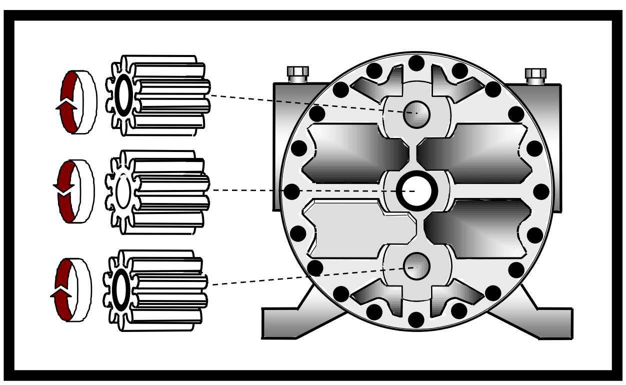
Do's and Don'ts

READ THIS BULLETIN BEFORE INSTALLING AND OPERATING A SMITH PUMP

The purpose of this bulletin is to describe systematically the causes of pump failure in systems using our units, with the underlying understanding that damage, started or aggravated by the human element, becomes a factor of the way in which the pumping system is used, the piping design, and the particular characteristics of the liquid in question, since the internal parts are actually cooled and lubricated by the fluid handled.

Smith Precision Products Company has made a very conscientious effort to keep its products as simple and as maintenance-free as possible, without sacrificing quality and service life. To this end, the working parts are all manufactured in house, and simply consist of axially free-floating balanced impellers in conjunction with our drive shaft/outboard ball bearing/mechanical seals assembly ("shaft-seal assembly"). To further ease the replacement of pumps, the Smith pump "Exchange Plan" was implemented.[1]

This exchange plan, which began many years ago, affords our company with a constant influx of used equipment for analysis, from virtually every possible type of field condition within our market. The data derived is invaluable in design improvement, and has been most helpful in developing special gear materials.



Smith pump gearing, as opposed to power transmission gearing, when applied to the transfer of liquids with low viscosity as liquefied gases, requires special attention to assume uniformity of surface finish, concentricity, parallelism, backlash, lubrication clearances, and bearing tolerances. Continually balanced internal loading afforded within the Smith pump can only be assured through application of superior wearing materials and even load distribution over relatively large areas .

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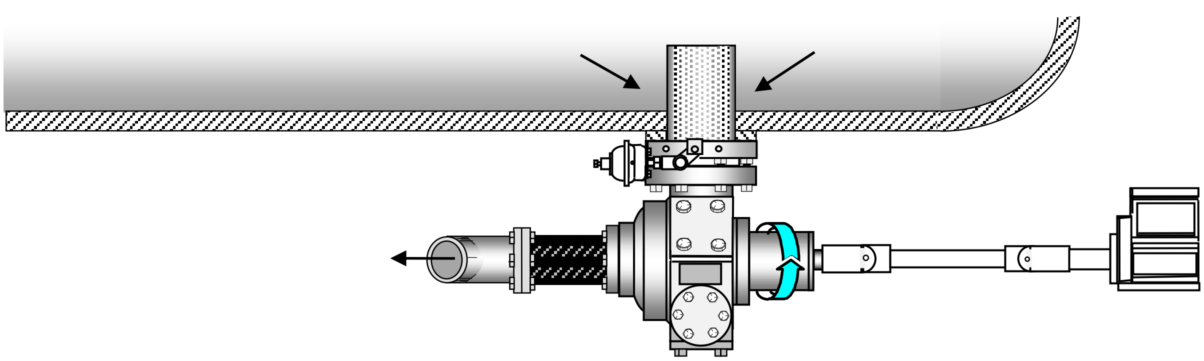
[1] See Bulletin "AL-1" for a more detailed explanation of the Smith pump "Exchange Plan".

The free-floating gears are held in close proximity, and remain equidistant from the ends of their respective bores ("pockets"), due to balanced hydraulic pressures, which keep them from physically contacting the casings.[2] For the sake of this discussion about the Smith pump, although some liquefied gases are said to have more "lubricity" than others, *the lubrication qualities are not as important as their continual refrigerative effects while being handled through the working areas of the pump.* This is especially true with certain highly polarized *non-lubricating* fluids, such as low-pressure liquid Carbon Dioxide. If the liquefied gas in question has at least a limited ability to wet the parts it contacts, the inherent "lubricity" is directly related to fluid viscosity, as appropriately indicated by field experience.

The closer the liquid is to the temperature at which it begins to boil, the greater the chances are that some of the load-balancing and cooling effects of flow inside the pump will be adversely affected by vapor. Ultimately, it has been shown that most failures relate to vapor displacement of flow around the internal working parts. This phenomenon is referred to as "cavitation" or "vapor-locking", and is caused by one or more of the following general categories to be discussed in detail: (1) vibrational stress, (2) inlet liquid condition, (3) differential pressure, (4) drive speed, (5) bypass return system, (6) ambient temperature differences, (7) liquid vapor pressure, (8) liquid temperature, (9) duty cycle, (10) storage of pump prior to its installation, (11) improper repair procedures, (12) vapor traps, and (13) "wind milling". Other failure causes only somewhat related to these categories are also discussed.

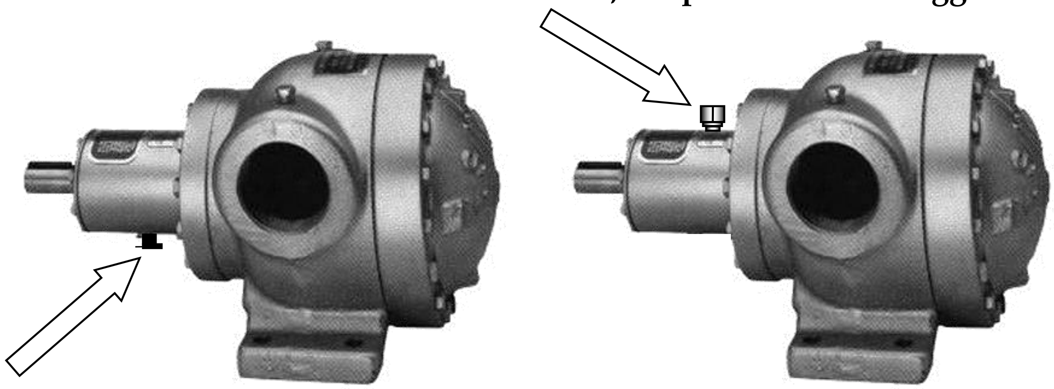
VIBRATIONAL STRESS. Coupling alignment, type of coupling used, universal joints, shafting to PTO, improperly adjusted belt drives, chain drives, bad motor bearings, etc., can all damage a pump when subjecting it to undue stress beyond design limitations. The lower the drive speed, the less chance there is that rapid wear will occur during adverse circumstances, but in any case, it is good practice to attempt to eliminate all misalignment and vibration through the pump drive mechanism.



[2] See Catalog "CP-3" for additional information.

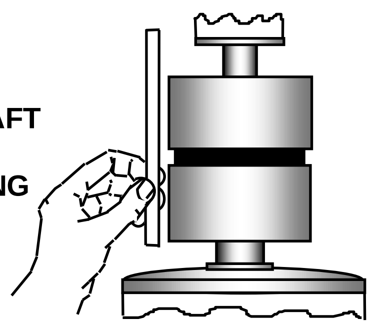
Excessive torsional loads produced through the drive shaft under improper conditions take up the pilot bearing clearances in one area, pulling the drive gear one way, causing imbalanced flow characteristics around all the gears, allowing physical contact between the housings and the gear faces, resultant galling, increased side clearances, and cavitation. A situation like this usually results in pump lock-up, or shaft seal failure.

Excessive vibration through the drive end of a Smith pump most commonly results in premature mechanical seal and pilot bushing failure, with periodic visible bleed through the port in the shaft end cover.[3] Each time this occurs, the problem can be aggravated by



highly abrasive ice, or frozen product, building up around the seals, accelerating their failure. During this time, material will also be removed from the gear teeth, and the drive gear(s) will be dragged into the bottom of their respective bores, enlarging critical clearances beyond limitations, allowing for cavitation or vapor-lock and highly decreased pumping efficiency.

NOTE: MOTOR AND PUMP SHAFT ALIGNMENT IS CRITICAL, TO ELIMINATE EXCESSIVE BEARING LOADS.



Fast coupling wear indicates misalignment of pump and motor shafts. Misalignment or the continued use of a worn coupling insert, can be responsible for rapid pump wear, continual seal leaks, and noise. The proper way to align the Smith drive coupling is shown in the sketch. (See Bulletin "AL-3").

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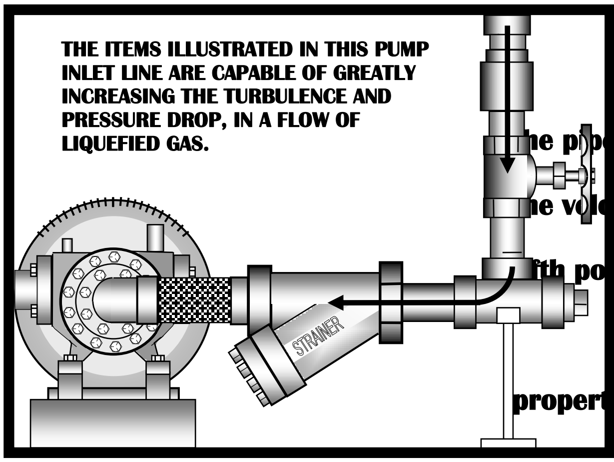
[3] All Smith pumps are provided with a "seal bleed port" for the purpose of channeling seal leakage away from the ball bearings, and also for facilitating leak detection. In all but the "SQ-Series", the discharge point of this channel is located on the bottom, or top, of the shaft end cover, and is protected by what appears to be an oil fill cover, or a small adapter fitting. This cover, or fitting, prevents the entrance of moisture and debris. It is not for lubricating the ball bearings. The ball bearings are permanently sealed, and require no periodic lubrication. *Do not lubricate or obstruct the opening, which is for venting and observing leakage*. **Observed mechanical shaft seal leakage from this leak detection port, or "seal bleed port" may be a symptom of excessive internal wear. Obviously, if leakage has been detected from the seal bleed port, the mechanical shaft-seal assembly, in its entirety, must be immediately replaced. However, the cause of seal failure must still be investigated.** *Before the pump is put back into service, the seal failure cause must be determined*. **Inspect the internal parts and corresponding potential wear areas by following the steps as outlined in the appropriate service manual.**

INLET LIQUID CONDITION. This subject concerns any substances being simultaneously supplied to the pump intake other than pure homogeneous liquid. Aside from the relatively common contaminants, vapor bubbles are a particular problem, especially in liquefied gas service, because these liquids are handled at their boiling points. For this reason, "Net Positive Suction Head Available (NPSHA)" is so very critical to proper system design.[4] Other substances which can cause severe damage include dirt, water or ice, oil, welding slag, rust flakes, pieces of hose, small metal parts, sand, Teflon® tape shreds, excessive sealant, soldering residues, and metallic chips. As with most adverse pumping conditions, handling excessive amounts of contaminants usually leads to overly-enlarged gear clearances, cavitation, gear face galling, and resultant seal failure from vibration and/or unit seize-up. When a typical liquefied gas pump inlet line is in service, the calculated vapor displacement occurring in the liquid on its way toward the pump is always *inversely proportional to its natural vapor pressure*. It is important to know that "Head Loss" computations are contingent upon six key factors as shown in the following illustration:



Factors Considered in Computing the Head Loss of Liquid Flow in Pipes

(1) The Head Loss varies directly as the length of the pipe.
(2) The Head Loss varies almost as the square of the velocity.
(3) The Head Loss varies almost inversely as the fifth power of the inside pipe diameter.
(4) The Head Loss depends upon the interior surface roughness of the pipe wall.
(5) The Head Loss depends upon the fluid properties of density and viscosity.
(6) The Head Loss is *independent* of the pressure.

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[4] "Net Positive Suction Head (NPSH)" is one of several similar terminologies (such as "Discharge Head", "Friction Head", "Pressure Head", "Static Head", "Suction Lift", "Total Head", "Velocity Head"), which equate pressure to elevation, in *equivalent feet of vertical liquid column*. In this particular case with the Smith pump, we are referring to the resultant pressure difference between the natural vapor pressure in the supply tank, and the measured liquid pressure at the inlet connection *while the pump is running*. By factoring in the height of the stored liquid above the pump centerline, plus any other factors which increase pump suction pressure independent of the liquid's natural vapor pressure, *less* the value of the resistance to flow in the pump inlet line, the figured pressure "overage" must be at least high enough *above* the tank vapor pressure to continually fill the internal liquid transfer cavities completely with "solid liquid". In other words, the acceleration of the liquid column when the pump first starts, and the continuing liquid flow into the pump, should be "force-fed" or "flooded", essentially by gravity, *to prevent boiling, or "cavitation"*. The "Net Positive Suction Head Available (NPSHA)", provided by the installation at the pump inlet, should be higher than the "Net Positive Suction Head Required (NPSHR)" by the pump. This figure generally determines how high the tank should be mounted above ground level.

The same restrictions will produce a higher volume of vapor as the pressure decreases. Therefore, the lower the pressure, the more important it is to prevent excessive cavitation during use. This is illustrated by considering the effects of calculated resistance to flow within the inlet line components, as the liquid passes through them toward the pump inlet connection. *(Inlet line flow resistance may be expressed in "equivalent feet of pipe" to more readily visualize the required height).* **For example, in an average LPG transfer system, with an external pump supply line, there are obvious abrupt directional changes through the internal valve (or excess flow valve and external shut-off valve), elbow, and strainer.** *"Static Head", or "gravity pressure gain", provided by the liquid level in the tank is the primary energy source which compensates for inlet line "Head Loss", determined by the flow rate through the valves, fittings, and pump liquid inlet.* **This is especially important during the initial start-up phase, where the transferred fluid at rest, is accelerated up to the pump demand velocity.**

At this time, the pump should never have to initiate "Suction Lift", or *negative* **pressure, in order to supply itself. Such a situation would instantly cause at least a small percentage of the supplied liquid, which is already at its boiling point, to vaporize, or "cavitate".[5] That could vapor-lock the pump, and expose it to excessive cavitational stress. The solution to this potential problem relates to the weight of the fluid handled. The idea is to force-feed the pump, relying primarily on** *the pressure increase per increment of height, related to its density.* **Taking full advantage of** *gravity pressure gain* **compensates for restrictions upon the liquid flowing through the line. It is a natural way to prevent excessive inlet flow vaporization. Standard technical references show that average quality Propane at ambient temperatures weighs approximately 4-1/2 lbs. per gallon, and that the amount of "Head" gained per unit of elevation is about 0.22 PSI/FT. Depending upon the given relative density figures, the approximate gravity pressure gain per foot of height, and approximate vapor pressure values of other common liquefied gases most frequently handled by Smith pumps, can be compared in the following table (unless specified otherwise, they are at 70° F.)[6] :**

LIQUID	PSI/FT OF "HEAD"	PSIG
Propane	0.22	109
Carbon Dioxide, 0°F.	0.43	294
Halon 1301	0.71	200
HCFC-22, 77° F.	0.52	136
HFC-134a, 77° F.	0.22	83
Anhydrous Ammonia	0.26	114
Butane	0.25	17
Sulfur Dioxide	0.60	35
Nitrous Oxide, 0°F.	0.43	312

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5 See "Table 1" on page 17; also, see "Booklet A (AL-36)" for additional information.

6 These values are only intended for general comparisons of vapor pressure and density characteristics with liquefied gases commonly handled. Frequently, inertia losses determined by fluid density, are more responsible for pressure drop than viscosity. For exact information, consult the appropriate engineering texts.

With our small and medium-capacity pumps in liquefied gas service, the extra head required to overcome resistance within the pump, itself, may be safely neglected, as proven by both factory and field tests, carried out over a period of many years. However, with high-capacity in-line Smith pumps run at *rated drive speeds*, a small NPSHR allowance should be made for best results, in accordance with the following table:

NOMINAL CAPACITY **	NPSHR
100 USGPM	1 FT
150 USGPM	1.5 FT
200 USGPM	2 FT
250 USGPM	3 FT

** "Nominal Capacity" ratings are used for comparative purposes (see Catalogs "CP-1", "CP-3", and "CP-9"), based on classifying the pump outputs at maximum design speed and "0" differential pressure. Actual outputs will vary, as per the pump performance formulae detailed in these references.

The resistance to flow of all components, including flow-limiting protective valves in the tank liquid outlet such as the excess flow check valve used in LPG and Anhydrous Ammonia services, *must be computed*.[7] Enough Static Head must be allowed to overcome this resistance, so that the liquid will not boil, or "lose its prime", on its way to the pump. Sometimes, heating the liquid supply accomplishes this end. Also, injection of an inert gas above the liquid in the tank to artificially raise the tank pressure has the same effect as elevating the tank.[8]

Due to the fact that some components used in liquefied gas pump inlet lines have high resistance to flow values, it is not uncommon to find that Static Head allowances of *six feet or more should be made*. Most generally, this is done by elevating the tank, maintaining a certain minimum level of liquid within the tank, or lowering the pump with relation to the tank.

It follows that attempting to increase system capacity by using an oversized pump in relation to the inlet pipe size will only lead to problems with the pump, as it begins to handle boiling liquid. As long as vapor makes up less than 10% of the liquid pumped, the unit will not become damaged, even though output may surge erratically. It is not uncommon to observe this problem, brought about by cold weather and improperly designed inlet lines, when the ambient temperature is also the liquid temperature. As the

[7] See "Booklet A (AL-36)" for more detailed information.

[8] Be sure to always follow all applicable safety codes and procedures when designing, building, and operating liquid transfer systems. Avoid potentially dangerous situations. If there are any questions, contact the appropriate local, state, or federal authorities. Always follow the equipment manufacturer's instructions, and be sure to read and understand their literature thoroughly.

weather gets colder, the liquid vapor pressure goes down, and the volume of liquid displacement by vapor bubbles, caused by the same restrictions, is greater than the maximum tolerable 10%. The resulting vapor-locking, or partial vapor-locking, is very damaging to the working parts.

Another specific problem that causes vapor-locking from lack of Static Head occurs when the pump is used primarily in one direction, but occasionally in the opposite direction, whereupon the outlet line becomes an overly-restrictive inlet line. For example, in the case of a transport-mounted 100 USGPM unit, designed specifically for *off-loading* low-pressure liquid Carbon Dioxide with a standard length 1-1/2" filler hose, if the pump is used to *fill* the transport at its maximum speed, the hose alone, can represent a head loss of 10 feet or more, even though the pressure drop is under 6 PSID. It can cause a 50% vapor displacement in the liquid supplied to the pump. It can easily be responsible for the equivalent of years of normal use wear, in a very short relative period of time.

Aside from the obvious reasons that a plugged strainer creates a lack of sufficient NPSHA at the pump inlet connection, one not so apparent fact is that dry ice can plug a strainer in a Carbon Dioxide transfer system. When the line full of liquid is *rapidly* discharged, the pressure drop through the strainer element causes some of the product to freeze. Factory tests show that dry ice can block the element, regardless of the perforation sizes or strainer design. If the pump is changed after rapid blow-down, and the dry ice not removed, it is easy to see how the pump could be run dry for a while, and exhibit all the telltale signs of starvation (such as gear face galling and rapid seal wear), even in a very well designed piping system. Atmospheric discharge should be done *slowly*.

Another not so obvious low NPSHA cause, can be attributed to whirlpools which form around the tank outlets and are at their worst when the liquid level is very low. Depending upon the height, size, shape of the tank, the size of and flow through the tank outlet, the density and viscosity of the liquid in question, they can vapor-lock the pump in spite of the fact that there is enough Static Head to overcome all resistance to flow in the liquid inlet line. Vapor passes down through the center of the whirlpool, possibly aggravated by the bypass return within the tank if it discharges conducive to circular motion, and by the shape of the tank if it facilitates Coriolis acceleration. Careful consideration should be given to the pump inlet line in order to eliminate this problem cause. The angular acceleration of the liquid itself allows the rotary movement to start, so lowering the outlet velocity will aid insuring that the whirlpool will not cause pump starvation. In an ideal piping system, the flow may have to be decreased, or the pipe size increased, if an additional Static Head allowance is impossible.

Improper multiple manifolding of two or more pumps together in parallel causes many problems, especially if the system design does not handle the multiple pumps as if they were just one larger unit. Ideally, if each pump supplies a different area, each pump used should have its own separate, dedicated, tank outlet. In this manner, for example, the onrush of accelerating liquid into a void outlet line will not cavitate the operating unit(s), especially in continuous-duty applications. Exceptions to this rule can be found with

highly intermittent services, where it is unlikely that more than one pump would be running at any one time. In loop systems, discrepancies in pressure drop between one loop and another can be corrected by proper use and installation of differential or bypass valving to Insure adequate liquid flow, to minimize heat gain, and to regulate pressures. Contact our Engineering Department if there are any questions.

Unusually lengthy pump inlet lines handling liquefied gases with low vapor pressures may vapor-lock the pump, in spite of the Static Head design allowance which is theoretically correct.[9] This happens because when the pump is started, the liquid must *accelerate*. In a properly designed, average installation, this initial pressure drop caused by acceleration is mostly negligible. However, the initial resistance to flow acceleration in an unusually long liquid column may still be significant enough to cause excessive volumetric vapor displacement within the pump. The rapid accumulation of vapor in the unit can stop it from pumping, and is quite capable of accelerating all wear factors.

A perfect pump inlet line would be inclined slightly throughout its total length, so that vapor bubbles would travel back to the supply tank where they could harmlessly dissipate. Since this is not practical in many installations, the top half of the pump and the highest areas in the suction line will fill with vapor when the system is not being used. Upon being activated, the pump becomes partially filled with vapor, and may run for several seconds before being filled completely with liquid. During this time, since vapor tends to rise, damage may be done due to frictional contact between the gear faces and the housings, especially in the top portion of the unit.

Vapor is eliminated from within the pump and its supply line by either *slowly* blowing-off product to atmosphere beyond the pump until liquid is observed, or by momentarily recirculating output flow back to the supply tank through the bypass line. Little damage will result if the pump purges itself against essentially no differential pressure for a very short interval. The proper method for piping is illustrated, below. The liquid flows out of the pump, and either through a hand valve and back to the supply tank, or through a bypass valve, set at a predetermined differential pressure, which avoids overloading the pump, and back to the tank.[10]

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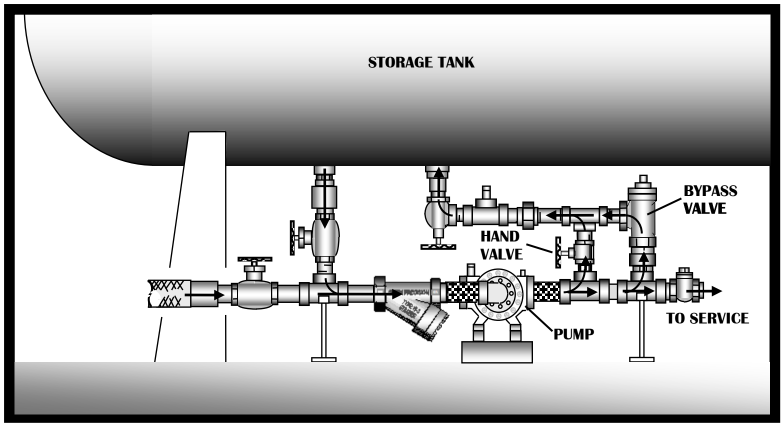
[9] Care should be taken to insure that regardless of use circumstances, there will be no design errors conducive to excessive vaporization of product *when the pump first accelerates the flow in its inlet line*. During this initial acceleration phase, until the flowing liquid column reaches the pump demand velocity, its speed can easily be more than double the calculated velocity based upon pump output. Even if the line is relatively short, a clogged strainer can readily produce enough resistance to cause excessive product vaporization during flow acceleration. This is why we <u>always</u> recommend that the pumps be mounted right under, or very close to, the supply tanks, that the pump inlet lines be of a comparatively *short* length, and that these lines be properly sized as per *our* recommendations in Bulletin "AL-3". See Bulletins "AL-3", "AL-40", "CP-6" , Booklet "A", and other available literature from Smith Precision, for additional information.

[10] See Bulletin "AL-3" for additional information.



When the demand for product within the system exceeds the pump capacity, or if improper pressure equalization procedures are at play, this occurrence would attempt to turn the pump, independently, of its motor. This kind of energy utilization, and interference with the motor, causes an internal pressure drop.

Since the liquid handled is already at, or very near, its boiling point, a substantial amount of vapor displacement could easily occur within the pump during this time, while it were being driven by a flow of liquid. In a highly aggravated situation, the pump could actually be run much faster than the rated speed of the motor. These conditions, "wind milling", can be extremely detrimental to pump longevity. Even during a very short interval, "wind milling" can overspeed the pump, run the unit with insufficient internal cooling and lubrication effects, create undue vibrational stress, or even produce rapid wear from crystals of frozen product within the close tolerances around the working parts. Excess flow check valves, and check-valved bypass lines around the pumps in question help to prevent "wind milling", by either shutting off the flow, or allowing it to go *around* the pump.

In liquefied gas recirculation systems which require periodic venting to atmosphere, in order to protect the pump from this "wind milling", during an over demand situation, there should be a supply line bypass around the pump, with a suitable check valve that prevents immediate recirculation to the tank, when the over demand subsides. Ideally, this supply line bypass would come out of a separately-dedicated tank outlet, as opposed to the pump inlet line, to prevent the possibility of pump cavitation during an unusually prolonged over demand episode. It would prolong the life of the mechanical seal assembly, but only if used properly. If continuous or excessive overdraw takes place, the system obviously would benefit from a larger pump.[11]

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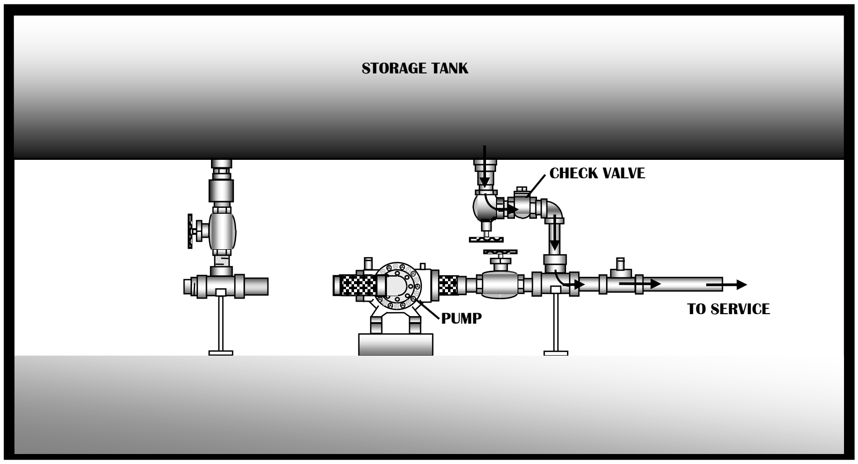
[11] See Bulletin "AL-93" for additional information on Liquid Carbon Dioxide recirculation systems.



As opposed to the flow over demand pump bypass system shown above, the return-to-tank bypass line as shown in the previous sketch, would only be used temporarily, either to prime the pump or to prevent the pump from "dead-heading" in an emergency situation. However, from a practical standpoint, many installations require frequent or continuous bypassing to maintain constant load on the pump. If this recirculated liquid is directed right back to the inlet of the pump, it will shortly absorb enough heat to boil, and cause vapor-lock, in the same manner as would happen with a built-in bypass valve discharging internally to the suction side. For this reason, the bypass discharge is directed *back to the supply source*, and the heat can safely dissipate without damaging the pump.[12]

Another item of concern, which causes problems, especially in recirculation systems, is moisture. We concede that it is virtually impossible to keep water from entering a supply tank every time it is filled with a hose, which is open to atmosphere before it is connected. However, if all hoses are kept pressurized, or tightly capped, most of the time, the accumulation of moisture will be minimized. Likewise, there will be a decrease in the probability of oxidation damage. With liquids kept at below 32° F., care should be taken to guard against water ice expansion and resultant fatigue of system components, especially after recharging a depressurized storage system which has not been used for some time.

In spite of the possibility of damage from moisture contamination, this is a rare problem in most applications with Smith pumps, which can handle practically any non-contaminated liquid at any viscosity, that is not corrosive to alloy irons and steels. If the average viscosity is above 500 SSU (or 110 Centistokes), one or both of the following must be done to insure proper internal lubrication: (1) utilize the pump within a lower-than-maximum speed range, and (2) increase the gear clearances. If this is not done, the pump will wear rapidly, and may seize.

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[12] See Booklet "A", and Bulletin "AL-93" for additional information on this and related subjects of concern.

For example, a medium capacity Smith pump originally set-up with standard clearances and higher drive speeds (up to 1800 RPM), for intermittent use in transferring a pure low viscosity fluid, if exposed to very thick oil as a contaminant in the liquid handled, would soon develop excessive internal friction, due to the drive velocity which would not be recommended under these circumstances. The gears could gall against the sides of the gear pockets, the drive gears could crack, and the pocket diameters could enlarge. These conditions in turn would cause the pump to vapor-lock and freeze-up.

Other contaminants in liquids pumped, such as excessive residues of Teflon® tape shreds left over from sealing the threaded piping connections, can work their way into critical clearances and subject the gears and bushings to abrupt irregular shock beyond design limitations. Some contaminants are damaging because they are corrosive. In cases where the liquid contains substances chemically incompatible with construction materials, the damage is always concentrated in the areas most subject to heat accumulation.

This is why in an extreme situation, the mechanical seals may be initially attacked, then the bearing surfaces of the shafting, and finally, the gear teeth. If the concentration is high enough, the gears will seize against the housings, as the corrosion takes-up the lubrication clearances. However, in most instances, internal corrosion shows external symptoms of seal chatter, similar to those caused by insufficient NPSHA. Particles of rust from the piping system, pump housings, and working parts, break free and circulate internally, resulting in excessive clearances. It is the resultant *cavitation* that leads to pump failures.

<u>DIFFERENTIAL PRESSURE</u>. Unit suitability in a particular system hinges heavily upon the differential pressure required. Since there is a predictable "slippage flow" within the Smith pump, the probable output given the pumping conditions is easily calculated per the performance formulae in catalogs "CP-1", "CP-3", and "CP-9". The "slippage factors" utilized, are in accordance with the viscosity of the liquid(s) to be handled. The higher the differential pressure, the lower the output, the faster the unit must run, and the greater the internal wear factors will be. Many times, compromises must be worked out between differential pressure drive speed requirements, and acceptable unit life.

If the differential pressure (the difference between inlet and outlet pump pressures) is too high, too much flow will be redirected back across the gears from the outlet side to the inlet side. This would have the same effect on pump life as recirculating the major part of the output right back to the pump inlet. In liquefied gas service, the resultant vapor-locking and vibrational stress would accelerate gear wear and would probably damage the mechanical seal assembly, causing it to leak within a short time.

Our catalogs always show the comparative, or "nominal" drive speeds and outputs of the most common Smith pumps as related to the majority of applications. Most Smith pumps are used in the intermittent bulk transfer low viscosity fluids, such as light petrochemicals, and liquefied gases. Therefore, the drive speeds shown relate to standard

differential pressures and standard viscosity values within these markets. When the pumps are required only for intermittent bulk transfer with low relative differential pressures, they can be driven up to their *maximum recommended drive speed*. The serial tags are usually stamped to reflect this, and other corresponding maximum limitations. However, *other requirements with liquid transfer may require different drive speeds*. With the exception of operations involving certain highly polarized non-lubricating fluids, or low capacity pumps used strictly in intermittent LPG service, the following table serves as a general guide for Smith pumps rated at 1800 RPM ("ATC-Series", "MC-Series", and "SQ-Series").[13]

TYPE OF DUTY	MAX. RPM	MAX. DIFFERENTIAL PRESSURE
CONTINUOUS	900	40 PSID (2.8 Kg/Cm²)
LIGHT CONTINUOUS TO INTERMEDIATE	1200	80 PSID (5.6 Kg/Cm²)
HEAVY INTERMEDIATE TO INTERMITTENT	1800	125 PSID (8.8 Kg/Cm²)
HIGHLY INTERMITTENT	1800	250 PSID (17.6 Kg/Cm²)

Units handling liquids having viscosities of 1 Centipoise or more, may tolerate higher pressures and slower speeds, due to lower slippage factors. With most liquids handled by Smith pumps, other than in highly intermittent duty, generally speaking, the pump can lose up to 50% of its nominal capacity due to differential pressure before excessive damage occurs. Continuous duty requires lower pressures due to cumulative effects of friction.

Smith pumps can be staged in series to limit pressure differentials, provided each one in succession is of lesser capacity and that the excess flow is continuously bypassed back to the liquid supply source. Contact our Engineering Department for more information. Sometimes, excessive differential pressure across the pump is caused by hidden "culprits", when the installation has not been provided with a bypass return system, or when the bypass return system is only designed to function when the vapor return system is working properly at the same time. If such an installation is used for filling small tanks without vapor return connections, should the operator mistakenly leave the pump running too long, or if he does not turn the pump off before shutting off the valve at the

[13] The table is only intended for *general comparisons* within the defined duty cycle classifications of direct motor-connected models. See the appropriate product line catalogs, technical bulletins, and service manuals, or contact the factory, for *specific recommendations* regarding proper system design PSID requirements, and corresponding pump RPM, for any of these, and all other Smith pump models not mentioned, above.

end of the hose, the pump will either be "dead-headed", which can be dangerous due to the positive-displacement nature of the pumping method employed in Smith pumps; or the internal relief valve will discharge the total pump output back to the inlet port, which will instantly vapor-lock the pump causing highly accelerated internal wear. In a "dead-heading" scenario, before the pump is vapor-locked completely by recirculated flow across the gear faces ("slippage"), it will momentarily build-up a *very high pressure*. This will overload the drive mechanism, and may cause enough galling to abruptly stop the unit from turning.

The severity of damage depends upon the liquid viscosity and motor speed. Always follow recommended procedures and applicable safety codes for proper transfer system design and use. The properly designed transfer system has an adequately sized, and correctly adjusted, back-to-tank bypass installation.[14]

Another example can be found with operations which commonly use vapor return to equalize pressures, and where there is undue restriction either in the outlet line, the vapor return line, or the bypass return line. All of these items require proper sizing.[15] If any of these are compromised due to other considerations, there is a good chance that at some time, the pump will fail due to the outlet piping's inability to absorb total pump output on a continuous basis. Delivery trucks which actually count on simultaneous operation of both vapor return and bypass return to keep excessive differential pressure in check, are usually beset with pump breakdowns when unusual circumstances require higher than normal differential pressures and/or longer than normal pressure equalization intervals. For the future, there will be an increasing need for more careful attention to system design and use, particularly with mobile units, as transfer volume increases.

Highly variable or vacillating differential pressure across the pump, as opposed to constant pump loading on a continual basis, can be responsible for decreased service life. In theory it is better recirculation system design to have a continuous bypass from the pump outlet back to the supply tank, so that regardless of the demand on the system, the pump always sees a preset differential pressure. However, due to certain realities in these systems, it is not always possible to assure constant differential pressure. This is best illustrated by a continuous recirculation installation with periodic discharge of product to atmosphere in excess of the system pump capacity. Under this condition, pump PSID indications can be in an occasional state of fluctuation. The pump is protected from "wind milling" by another liquid supply line, which goes around it. This over demand bypass system contains a check valve to prevent the pump from discharging back to its inlet side when the line is not in use. The over demand bypass system allows a low-capacity pump to be used to maintain required liquid quality.[16] The pump in question would most likely handle a constant differential pressure *most* of the time, and would only see a substantial drop in pressure temporarily.

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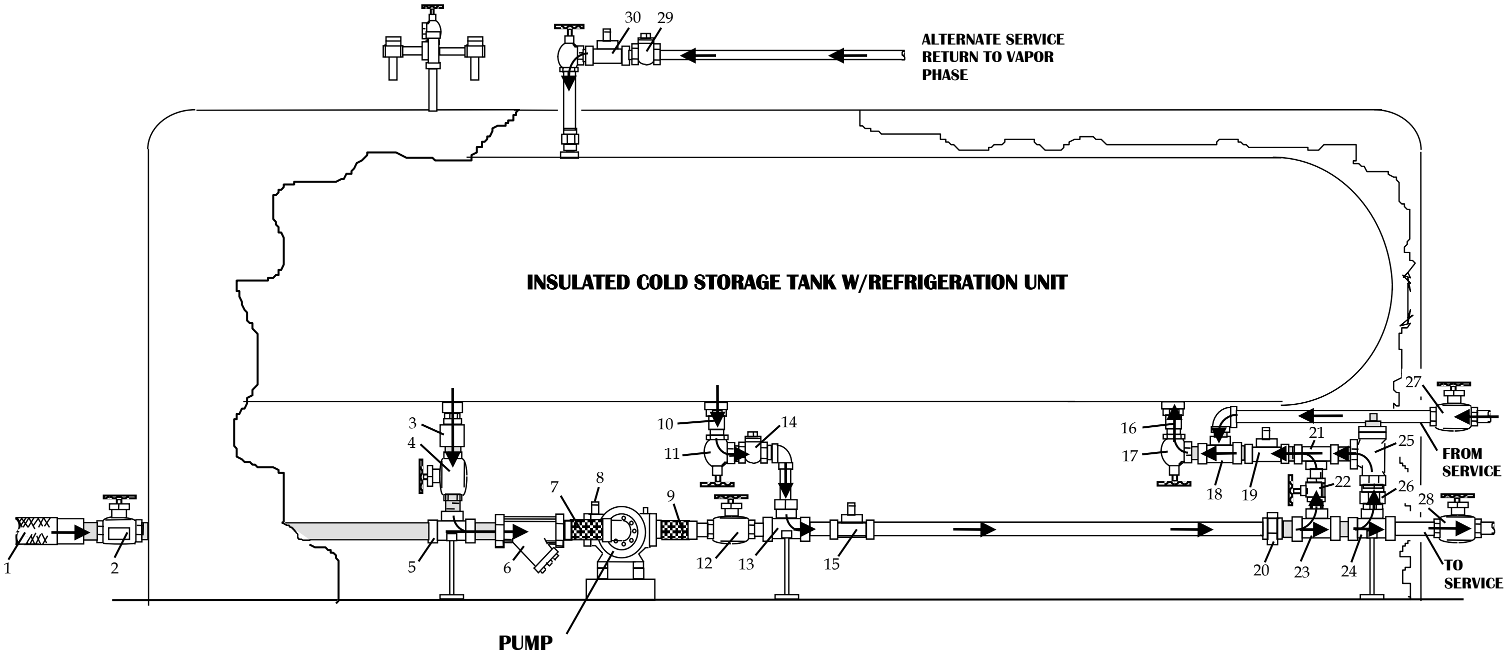
[14] See Bulletins "AL-3", "AL-93", "Booklet A (AL-36)", and Catalogs "CP-1", "CP-3", "CP-9" and "DBV-L" for additional information. Also, review the sketch on page 9 of this technical bulletin.
[15] See Bulletin "AL-3" and "Booklet A (AL-36)".
[16] Review pg. 9 in this technical bulletin. See Bulletin "AL-93" for additional information.

A typical example of a balanced recirculating system affording mostly constant loading, can be seen, below. This is a continuous-duty low-pressure liquid Carbon Dioxide recirculation system. Please see Bulletin "AL-93" for a detailed explanation of the different components and how they are utilized. This drawing should not be taken literally, as its purpose is merely to illustrate basics.

BASIC LOW-PRESSURE LIQUID CO_2 RECIRCULATION PUMP AND SUPPLY TANK SYSTEM



<u>DRIVE SPEEDS</u>. In accordance with the use parameters and design limitations, Smith pumps drive speed ranges must vary *per specific applications*. Handling the less viscous liquids, these drive speeds become more absolute, with probabilities of great damage being done to the pump components if the application differs from factory recommendations.

The maximum design limitation on speed with most models in most services is 1800 RPM. If the unit is run at speeds above this limitation, there is a good chance that insufficient internal lubrication and cooling will result, subjecting the working parts to excessive abrupt vibrational stress and friction, leading to accelerated internal wear. Exceptions to the 1800 RPM speed limitation are found with the D-Series, E-Series, and MC/GC-Series pumps, in LPG service (3600 RPM Max.), and with the slow speed models for PTO and belt-drives: MCAT-Series, 600 - 1200 RPM; TC-2/3 Series, 250 - 500 RPM; and TC-1044 Series, 450 - 900 RPM.

As mentioned previously, our catalogs and the serial tags on Smith pumps reflect the majority of low PSID transfer intermittent duty services, which comprise our major market. Specific applications may favor factory recommendations calling for *slower* speeds and *lower* differential pressures than those specified, due to wear considerations. The information on the pump tags usually reflects the maximum nominal capabilities under intermittent conditions, which are theoretically perfect.

The slower the drive speed is, relative to the maximum recommended speed, the lower the differential pressure must be to insure that slippage flow will not account for too much pumped fluid recirculation inside the pump, with resultant "flashing", or "cavitation". With liquefied gases in general, if the differential pressure is below 40 PSID, drive speeds of one half or less of the maximum are permitted. At differential pressures of up to 80 PSID in intermittent service, two thirds the maximum drive speed can be used. At higher PSID, the maximum drive speed must be used, and the duty cycle cannot be continuous (unless two or more units can be simultaneously staged so the pressure differential built up by each pump can be less, and the RPM of each unit can also be less). See specific application data.

Staging in series two or more units with continuous bypassing has been successfully done, but it does require a more complicated piping system, and more attention to proper operating procedures. Unlike a multi-staged pump, the individual units cannot be exactly "tuned" in succession. It is therefore necessary to insure that the first unit *always* supplies more fluid than the second one can use, and to insure that it does its "share" of the work. This requires that the excess flow, or that portion of the output of the first pump above the demand of the second pump, continuously be bypassed back to the supply source. The first pump, then, must be of larger capacity than the second one, meaning that different models must be employed or that the same models be run at different speeds. Failure to take this into consideration will cause the first unit to either "do all the work" and give the following one(s) a "free ride", or the pumps further down stream will cavitate and possibly vapor-lock.

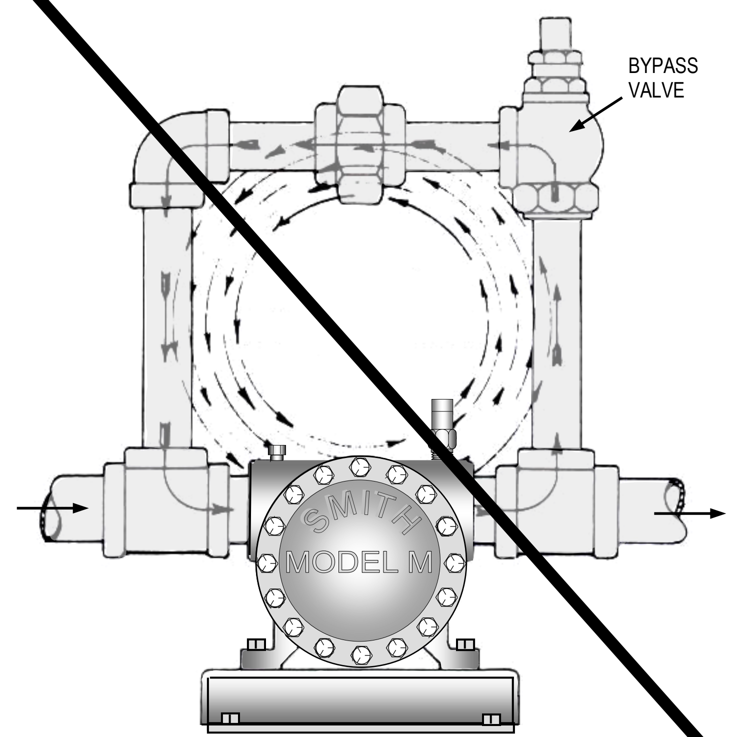
Liquids with viscosities higher than 500 SSU (110 Centistokes), present different problems. Slower drive speeds are required to insure proper internal lubrication, unless the differential pressure is several hundred PSID. Extra gear clearance may be necessary to insure adequate gear lubrication.

<u>BYPASS RETURN SYSTEMS</u>. When liquids being pumped are near their boiling point, any heat injected into the system creates a potential for cavitation, and resultant damage from vapor displacement of fluid passing through the pump interior. Normal functioning of a typical bypass valve coupled with that of a typical pump, adds small amounts of heat to the liquid handled. If the bypassed liquid were returned to the pump inlet before heat could be dispersed, the temperature would build through repeated recirculation in a short time. In the case of liquefied gases (such as LPG, Anhydrous Ammonia, and Carbon

Dioxide, under average conditions), it would be possible to almost instantly convert the liquid content to vapor at the pump intake.

A "built-in" bypass would even be worse than an externally piped bypass return to pump suction, because there would be less opportunity for dissipation of heat, since the total contents of the circuit would no doubt be less than that of an external system. The sketch, below, illustrates an improper bypass system for a liquefied gas pump. Attempting to refrigerate the liquid that is bypassed as it returns to the pump is at best a marginal approach to a problem better solved by proper initial design. Since bypassing is necessary due to safety and operating conditions, the installation should be so designed as to allow 100% recirculation over extended intervals without boiling fluid constantly returning to the pump.

THE "SHORT LOOP" BYPASS



Non-Recommended Primary Bypass Discharge System

In many cases, returning the bypass to the supply tank actually benefits the operation by increasing the pressure of the liquid fed to the pump. The higher this pressure is, the less liquid displacement by vapor there will be, and the lower the differential pressure will have to be to transfer at the rate desired.

Transfer from large to small cylinders benefits from this approach. Mobile installations can especially benefit. It will not only relieve excessive continual differential pressure across the pump, but it will also reduce noise, allow easier filling at a desirable rate, lower the amount of work the pump has to do, and provide a safety release mechanism should some restriction create excessive back pressure in the pump outlet line. In a *refrigerated* liquefied gas system, bypass to liquid phase helps to delay the vapor pressure increase, which may approach safety valve settings, in continuous operations.[17]

AMBIENT TEMPERATURE DIFFERENCES. A pressure drop in a supply line which normally would not cause any appreciable vapor formation in the fluid on its way to the pump, could, on a cold day, be responsible for a large amount of liquid displacement, proportionate pump output decrease, and resultant interior unit damage. In the case of a Propane tank mounted three feet lower than it should have been to provide the proper Static Head at the lowest liquid level, the percentage of output reduction is less than 10% even at +40° F.. However, at -20° F., the loss becomes approximately 37%, and the pump can be severely damaged in just a few hours.

TABLE 1. PERCENTAGE OF CAPACITY REDUCTION IN LPG PUMPS WHEN THERE IS INSUFFICIENT STATIC HEAD PRESSURE

Liquid and Temperature	The Difference Between "Required Elevation" and "Actual Elevation" in Equivalent Feet of Vertical Liquid Column											
	1	2	3	4	5	6	7	8	9	10	11	12
Propane 100° F.	0.7	1.4	2.0	2.7	3.3	4.0	4.6	5.2	5.8	6.5	7.1	7.6
Propane 70° F.	1.4	2.8	4.1	5.4	6.6	7.9	9.0	10.2	11.4	12.4	13.5	14.6
Propane 40° F.	3.0	5.9	8.9	11.1	13.6	15.8	18.0	20.1	22.0	23.9	25.7	27.3
Propane 10° F.	6.9	12.9	17.9	22.0	27.1	30.8	34.2	37.3	40.2	43.5	45.1	47.2
Propane -20° F.	16.3	28.0	36.8	43.7	49.3	53.9	57.6	60.8	63.7	66.1	68.1	70.0
Butane 100° F.	7.4	13.7	19.3	24.1	28.5	32.3	35.7	38.9	41.7	44.3	46.7	48.8
Butane 70° F.	15.3	26.5	35.1	41.8	47.3	51.9	55.8	59.0	61.8	64.2	66.3	68.3
Butane 40° F.	30.9	48.3	58.4	65.3	70.2	73.8	76.7	79.1	80.8	82.6	83.8	84.9

In systems, which are not heated or refrigerated by artificial means, the environmental temperature is a factor which should receive ample consideration. Also, in relation to this subject, a difference in temperature near the pump relative to the temperature of the liquid in the tank, may cause some problems if not given attention during the initial system design. Improper or poor supply tank insulation may give rise to problems caused by excessive differential pressure required of the pump, when the ambient temperature drops below 0° F., in a refrigerated system.

Any liquefied gas installation where the lack of sufficient Static Head will cause more than 10% vapor in the inlet liquid flow within a Smith pump, should be modified. Ambient temperature not taken into consideration initially can cause equipment break-

[17] See "Booklet A (AL-36)", and Bulletin "AL-93" for additional information.

down when the temperature drops far enough to allow higher than 10% liquid displacement by vapor in the pump supply line.
The wind chill factor has been known to cause pump problems in theoretically perfect designed installations, from excessive thickening of ball bearing greases. Smith pumps utilize synthetic greases with extended temperature range capabilities. If we are informed about these conditions ahead of time, the permanently sealed outboard ball bearing in a Smith pump can be modified to withstand temperatures as cold as -120° F..

<u>LIQUID VAPOR PRESSURE</u>. Liquefied gases when contained in pressure-tight receptacles, boil until equilibrium is reached between the liquid and vapor phase inside the containers. The pressure attained in this manner varies, depending upon each individual liquid and the temperature of the same. In this discussion, we primarily concern ourselves with those liquefied gases most commonly handled by Smith pumps, at moderate pressures and ordinary temperatures.

One difficulty that can be particularly exasperating to a customer, is when pump failure continually occurs in a system which apparently has been perfectly designed. Many times, this type of recurring problem is shown to be *improper consideration given to the natural vapor pressure of the liquid handled, and the effect it has on the pump*. This is especially true when the pressure of the liquid is less than atmospheric pressure.

If the standard vacuum option has not been specified for a Smith pump used in service with liquids at less than atmospheric pressure, when exposed to vacuum it will allow air into the system, which will very seriously reduce service life expectancy and pump efficiency. Also, at such low vapor pressures, unless additional allowances are made for NPSHA, a seemingly tolerable inlet line restriction can cause the pump to cavitate due to increased flow displaced by vapor formed from a normally insignificant pressure drop, aggravated by the inherently low tank pressure.[18]

<u>LIQUID TEMPERATURE</u>. It should not be assumed that all fluids are always affected by temperature and flow, in exactly the same way. At certain temperatures, or consistently throughout the transfer temperature range, some liquids can *drastically* change viscosity when subjected to agitation, or "shear". Agitation is caused within a pump as it handles the liquid. It is also caused by fluid passage through typical piping systems, especially on its way through or very near certain components, which produce resultant velocity changes with heavy turbulence. Consider how this phenomenon could affect the line loss calculations.

Possibly there should be another category in this bulletin relating to when the "effective" viscosity is not the same as the "apparent" viscosity as measured by standard instruments.

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[18] Never overlook potential problems with vacuum during the initial system design phase. See Bulletin "AL-43", "Booklet A (AL-36)", other literature from Smith Precision, and available general technical references. Contact the factory if there are any questions.

Most common liquids are "Newtonian" in nature, and behave predictably, following the predetermined laws of physics. One of these characteristics is maintaining a constant viscosity value regardless of changes in "agitation", or "Shear Rate" (as do sugar solutions, Gasoline, LPG, Carbon Dioxide, petroleum-based oils, and Anhydrous Ammonia). As such, their resistance to flow within the piping system is relatively easy to predict. However, "dilatant" and "thixotropic" fluids, which change their viscosities when subjected to turbulence, present a series of possible variations which are best determined by actual test. "Dilatant" fluids, *increase* their viscosity as the rate of "Shear" increases (as do slurries, and compounds used in the production of candy). "Thixotropic" fluids show a *decreasing* viscosity as the agitation increases (as do soap, resins, vegetable oils, and certain other plant extracts). These changes can have corresponding effects upon the pump differential pressure, and internal lubrication characteristics. If not taken into account, they can be responsible for pump failures.

Many storage facilities utilize refrigeration to lower the storage pressures and allow product transfer within more practical means. As with ambient temperature considerations, the full scale of the temperature range and its effect on the liquid density and vapor pressure is important. Also, of no less consequence is the effect of the temperature on the pump construction materials, as well as its related pressure. At the present time, the standard materials of construction permit Smith pumps to be modified as required, for efficient trouble-free service at temperatures from -150° F. to +400° F., at pressures with liquefied gases up to 600 PSIG, with an ample margin of safety. Check pump data plate for specifics.

Studies have shown that it is theoretically possible to produce units capable of pumping applications as cold as -320° F., or as hot as +600° F., using standard designs and special materials. Standard units can be constructed for thermal shock capability within a temperature range of -65° F. to +230° F., as positively shown by field experience.

The effect cold has upon impurities in the fluid handled is an important consideration, as this may change recommendations for gear clearances, drive speeds, and unit suitability. Whether or not these impurities are dissolved or suspended in the liquid, may also determine suitability, especially upon initial start-up in a cold temperature application.

The heat generated by the working parts of the pump in relation to the type of liquid and the liquid temperature is another subject of concern. For example, certain substances crystallize beyond a particular temperature limitation. Since the mechanical seal produces frictional heat during normal operation, it may play a role in causing the crystallization in its immediate area, which would be very detrimental due to the abrasive nature of the solids formed. Finally, the temperature affects liquid vapor pressure as well as liquid viscosity, and these factors are always important in calculating drive speeds.

Ice formation around the pump exterior is caused by differences between ambient temperature and liquid temperature. In continual service in cold conditions, this ice build-up can block the seal bleed port on all but the standard "SQ-Series" models. This bleed port is necessary to maintain atmospheric pressure on the back side of the mechanical seal assembly. If the unit is not provided with this small seepage passageway for trapped pressure release to the outside, in the very worst observed situations, this pressure eventually builds up to the point where the auxiliary bleed seal ruptures, whereupon it is relieved through the outboard ball bearing, slowly. Eventually, the bearing may be degreased in this manner. Also, during this time, especially in liquefied gas transfer, the lowering of temperature in the immediate area of pressure relief can cause entering moisture to freeze. The abrasive nature of the water ice crystals can wear out the mechanical seals. In a worst case scenario, if the liquid escapes fast enough, the temperature in that area can approach the freezing point of the liquid. When crystals of frozen product accumulate inside the pump, they can wear out the working parts. If this occurs at temperatures below -40° F., most o-rings become excessively hardened to function properly; some o-rings may actually shrink or crack, if they are close enough to the leak area.

Experience has shown that with the lower pressure liquefied gases commonly transferred by Smith pumps (such as LPG and Anhydrous Ammonia), the colder the liquid is, the more Static Head is needed to keep inlet line pressure drop from feeding boiling liquid into the pump inlet. The worst problems always occur when handling liquefied gases with very low vapor pressures, such as Butane on a cold day, or refrigerated Anhydrous Ammonia at -40° F.. In similar cases, it is not uncommon to recommend eight feet minimum Static Head for the inlet line system, as opposed to only four feet, at +70°F..

<u>DUTY CYCLE</u>. Unfortunately, one important piece of information all too frequently overlooked in typical pump inquiries from many companies, is the off-and-on time interval requirement. Many times, it is found that the duty required is highly intermittent, and therefore special construction materials are seldom necessary. This means that lead times are usually shorter and replacement parts are more readily obtainable. However, one's own interpretation of customer needs can be in error toward the opposite extreme, and the unit may be used in an even more difficult situation than that imagined by the supplier. The "duty cycle", therefore, is very important in determining pump options as well as drive speeds.

For example, if the duty requirement is continuous, with an average directly-connected electric motor driven Smith pump in liquefied gas service (see the appropriate catalogs), 900 RPM should be considered as the maximum drive speed, as otherwise, accumulations of frictional heat will create excessive internal wear conditions. A direct correlation can be made between pump drive speed and pump life expectancy. If a unit used twenty-four hours per day at 1800 RPM lasts six months, under exactly the same conditions at 1200 RPM the unit can last one year, and at 900 RPM eighteen months. On the other hand, in very intermittent duty where the unit is utilized for under two minutes at a time, pumping

Anhydrous Ammonia out of an accumulator, and then is turned off until the supply tank fills again, differential pressures of over 200 PSID are possible even with standard designs. As liquid viscosity increases, "duty cycle" becomes less important, however.

The majority of our applications are in liquefied gas transfer. These liquids have very low viscosity values. With "heavier" low viscosity liquids (such as light spirits of petroleum) in continuous duty service, a Smith pump could probably run at its *maximum* design speed and still give very long trouble-free service. In oil transfer, the "duty cycle" considerations give way to pressure, temperature, and viscosity value factors. In highly viscous liquid transfer, the most important consideration is whether or not the unit will need extra gear clearance for proper internal lubrication.

STORAGE OF PUMP PRIOR TO ITS INSTALLATION. Even though a Smith pump is brand new, if it has been kept out of service for more than one year from the date of shipment from the factory, and adequate steps have not been taken to safeguard the internal parts during this long interval, chances are that not only could the pump be potentially dangerous to use due to the probable shaft seal leaks from corrosion, but also it could gall up or exhibit vastly accelerated wear patterns similar to those caused by handling cavitating liquid. O-ring seals which are kept under mechanical compression for an excessive amount of time without being pressure activated, may be rendered non-functional from taking on "compression set". Ball bearing grease will revert to its original counterparts, oil and thickener, after prolonged storage of this sort, and will not perform satisfactorily under load.

In most cases, the average customer most certainly can maintain a Smith pump in safe operating condition even during long shutdown periods, if proper common sense procedures are followed. However, *there are certain specially constructed pumps, such as those used for Nitrous Oxide transfer, which do not lend themselves to prolonged storage, standard cleaning techniques, or average repair procedures*.[19] LPG and Anhydrous Ammonia pumps can be partially filled with Stoddards Solvent to withstand long storage periods. Automotive engine oil with "SA" or "SB" ratings can also be used as a protectant. Following this procedure will keep the pumps from suffering damage to the working parts other than o-ring seals and ball bearings. If the shaft-seal assembly is replaced in the field when the unit is put back into service later, chances are that it will perform satisfactorily even after being in storage for many years. Naturally the solvent or oil will have to be flushed out of the unit prior to system installation. If this is not done, the pumped liquid will be contaminated by the protectant residue. Also, if the

[19] Check the appropriate service manual and other technical bulletins from Smith Precision, to be sure if prolonged storage of the unit is possible. Certain very specialized services handling extremely reactive products require highly controlled procedures, which necessitate minimal time intervals between initial shipment from the factory, installation, and use in the system. Under these unique conditions, prolonged storage of pumps is *absolutely out of the question, due to safety factors.* Avoid potentially dangerous situations involving chemical incompatibilities with these pumps, and related equipment. Be sure to follow all of the company's, fabricator's, and manufacturer's safety procedures, including applicable local, State, and Federal regulations. Contact the factory for additional information.

pump is exposed to low temperatures when installed, any remaining protectant residues may congeal, and obstruct the internal lubrication flow channels. Contact the factory for additional information relating to the specific services.

IMPROPER REPAIR PROCEDURES. In a highly adverse situation, where routine preventive maintenance is lacking, as long as the mechanical seals have not worn through, and the gears still have a portion of their teeth and have not gouged more than just a few thousandths into their respective bores, the Smith pump will continue to function, and may still build-up to its normal capacity as per the appropriate catalog, especially if the Static Head allowance is well above the minimum requirement. Unfortunately, there is a tendency to wait until the equipment does not function properly, before maintenance is done. By this time, irreparable damage may have occurred. *Untimely* repairs may only shortly prolong pump use, before a reoccurrence of failure. The fact that a pump keeps on working does not necessarily mean that it is in good condition.

QUICK VISUAL WEAR CHECK

Checking for wear on the pocket diameters.

Normal gear end cover face with no functional wear.

Checking for wear on the pocket ends.

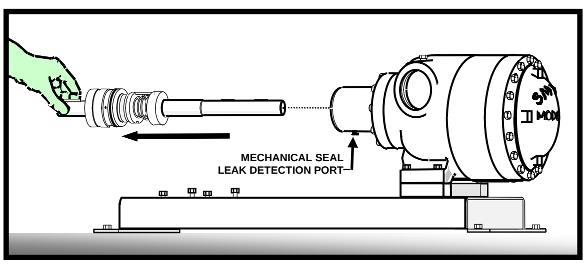
Frequently, a quick visual inspection can save time. For example, if there are visible signs of excessive material removal from the areas shown, the pump *cannot be repaired*. However, if there is no visible wear, the casings can be used again, and the pump can be repaired.

It may very well only mean that in spite of the wear and abuse, the design and the NPSHA are still compensating for wear, even under poor conditions. A mechanic may not notice excessive wear in the gear bores, when, in reality, there could have been overly-excessive material removal. When a unit has more than double the gear clearance of a new pump, merely replacing worn out gears and the shaft-seal assembly will not be a satisfactory repair job. Under these circumstances, chances are that the new parts will shortly wear out from lack of proper internal support, vibrational stress, and excessive slippage.[20] Our literature states that "No periodic maintenance, lubricating, or servicing is

[20] See Bulletins "AL-97", "AL-93", "AL-58", and other literature from Smith Precision. Repair procedures must be done in a safe manner, following all applicable local, State, and Federal regulations, as well as appropriate safety codes (such as "NFPA-58" for LPG service). Disassembly and reassembly must be done

required." However, there is a recommended routine inspection procedure, which, if followed as suggested, will virtually eliminate pump break-downs in a good system, and delay them substantially in a marginal system.[21] For units in improperly designed systems, routine replacement of critical parts *before* they wear out, is highly recommended, as determined by periodic inspections. These parts would probably include the shaft-seal assembly, as well as the gears and bushings.

Usually, the first part that fails in a bad system is the seal assembly, and its failure can subsequently damage other parts as well. When the seals wear out, the resultant leakage damages the drive gears and their respective pockets. Therefore, timely replacement of the shaft-seal assembly and the gear set, will definitely prolong pump life as well as serviceability. With pumps handling liquefied gases in continuous duty service more than six hours per day, one half of the maximum design RPM speed, and 40 PSID maximum differential pressure, are suggested to lower all wear factors by 3X. Even here, preventive inspections still prolong pump life greatly. The following drawing illustrates how the shaft-seal assembly can easily be replaced in the field, without removing the pump from the piping.[22] Note that under normal conditions, the shaft-seal assembly used for small capacity pumps up to 15 USGPM capacity should be replaced after 8,000 hours of use if not sooner, as determined by the recommended inspections. For pumps in the 20 – 250 USGPM range, the interval is 12,000 hours if not sooner, as determined by the inspections. Read Bulletins "AL-200" and "AL-97".



ONLY ONE EASY-TO-REPLACE MECHANICAL SHAFT SEAL ASSEMBLY IS REQUIRED IN SMITH PUMPS

There are some variables that affect the life of a mechanical seal: inadequate NPSHA, restrictions to flow on the pump inlet side, the type of fluid(s) being transferred, cavitation conditions, infrequent dry running, excessive differential pressure, dead-heading, excessive RPM, duty cycle, etc.

Since by time proven design the gears must be held in close proximity to the housings, it follows that too much clearance must be avoided. In the worst case scenario, housings damaged by metal-to-metal contact from the gears, must be remachined or

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in the proper sequence. Excessively worn parts should never be reused. The casings do not utilize gaskets, and must be resealed properly with approved sealing compound, after being disassembled and cleaned. The pump must be checked for leaks before being returned to service. The proper factory recommended medium should be used when effecting pressure tests on both new or repaired units. This test medium must always be compatible with the construction materials and the fluid(s) to be handled, must leave no adverse residues, and must be of a viscosity that matches that of the vapor from the liquid(s) the pump is designed to handle, as specified on the label plate. Avoid potentially dangerous situations. *Never use water.* Pressurizing with Nitrogen vapor may cause seal and casing leakage. Contact the factory for additional information.

[21] The procedures in question are explained in Bulletin "AL-200", which covers units in well-designed installations. Also see Bulletin "AL-97", for medium and high-capacity Smith pumps.

[22] Be sure to follow all proper safety guidelines, depressurize the pump, and wear gloves for this operation.

replaced. If this is not done, and only the gears are replaced, the unit will recirculate internally excessive amounts of pumped fluid. The resultant cavitation will supply the inlet side of the pump with too much vapor.[23]

<u>VAPOR TRAPS</u>. Vapor traps in pump inlet lines definitely affect inlet liquid quality. If there is enough trapped vapor in the pump inlet line, the pump may not be able to purge itself, and will vapor-lock. If the pump is running against a low head pressure, it may be able to force the vapor accumulations on downstream, at the price of some irreparable damage to the mechanical seals, or galling by the gear faces against the gear end cover and backs of the secondary housings. In systems where the pump is left pressurized, it is not necessarily free from the ravages of trapped vapor, since there are components in the inlet line which when contacted by warm air or sunlight, will vaporize small quantities of liquid and will trap vapor, which will feed into the pump when it first starts. Also, the pump, itself, is quite capable of trapping some vapor in its upper flow channels, exposing the top gears to partial dry-running, initially, on start up. If the NPSHA is insufficient at the pump inlet connection, these conditions will be aggravated, and will show up eventually as more distinctive galling by the gears in the upper portions of the unit. Generally speaking, vapor eliminating devices, if absolutely necessary, *should be used in the pump outlet line, not the pump inlet line*. Pump supply tanks that have liquid outlets through the ends, or the top, and not through the bottom, frequently have internal "dip tubes" attached to the pump outlet connection which are built-in vapor traps that can easily cause problems.

<u>WIND MILLING</u>. "Wind milling" can be defined as the tendency of the pump to act as a motor, being turned by a liquid or vapor flow through it, independent of its own motor. For the sake of this discussion, it is usually caused by operator error in pressure equalization, truck tank filling line valving, or gas purging from inlet lines (mistakenly called "cooling-the-pump-down"), aggravated of course by the tank pressure itself, and the physical size of the passage through which the gas is escaping to atmosphere, or transferring to another vessel. It is possible for the unit to actually overspeed under these circumstances, as well as run dry and build up deposits of abrasive frozen product in critical working areas. Under the worst case condition, pumps usually gall-up right away to such an extent that they will not turn, similar to what usually happens when the drive speed is excessive.[24]

<u>CONCLUSION</u>. In all of the aforementioned, it has been shown that many conditions exist, which can create problems for a Smith pump, but which can easily be avoided

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[23] See Bulletin "AL-1", for additional information on basic repairs and maintenance.

[24] If driven by liquefied gases or vapors, even within the recommended drive speed range, there is still not enough internal lubrication. However, Smith pumps actually have been used as hydraulic motors, running at the proper design RPM on an appropriate flow of compatible oil with an average viscosity of 500 SSU throughout the temperature range. Contact the factory for additional information.

through recommended system designs and pump applications. Proper understanding of how the units are run in their particular installations will explain why they last as long as they do, and what can be done to increase their durability. The life expectancy of an average Smith pump is actually increasing, in situations where the pumping conditions grow ever more demanding. This is a direct result of our ongoing analyses of used Smith pumps, constantly being returned to us under the Exchange Plan. We have substantial renewing physical evidence on hand which enables us to closely follow the evolution of pump market serviceability trends. As a matter of fact, this unique, invaluable up-dated information, readily supports our technological advancement in the design and materials of construction that affect the Smith pump's performance. In this manner we continue to work together with our customers, to insure a better future.

SMITH PUMPS

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