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Investigation into the effect of the slipper compensating orifice size on the mechanical and volumetric efficiency of a fixed displacement axial piston hydrostatic unit

by

John Patrick Fleming

A Thesis Submitted to the
Graduate Faculty in Partial Fulfillment of the
Requirements for the Degree of
MASTER OF SCIENCE

Department: Mechanical Engineering
Major: Mechanical Engineering

Signatures have been redacted for privacy

Iowa State University
Ames, Iowa

1995

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ABSTRACT

The size of the compensating orifice in the slipper of hydrostatic units has an important influence on the volumetric, mechanical, and overall efficiencies. A simple model was derived to predict the changes in efficiencies caused by first reducing and then blanking the orifice. Testing in a hydrostatic unit was performed to measure the effects of the slipper orifice design on efficiency. Comparison of the test results to the model's predictions was then made.

Testing validated the predicted result that volumetric efficiencies should increase as the orifice was first reduced and then eliminated. The results from the mechanical efficiency testing did not agree with the predicted results. The best overall efficiencies were exhibited by the reduced orifice design.

INTRODUCTION

Hydrostatic pumps and motors have found uses in numerous diverse applications. The applications may be as simple as transmitting rotational energy from a prime mover into translational motion of a vehicle or as complex as steering large agricultural vehicles. The characteristics of infinite displacement setting within the limits of the swash plate motion offer smooth and continuous transmission of power.

Hydrostatic units rely primarily on the proper design of three main bearings -- the block face/valve plate interface, the piston/bore interface, and the slipper/thrust surface interface. The proper design of each of these is essential for the successful operation of a hydrostatic unit. Of these three bearings, the slipper interface offers probably the greatest challenge to the designer in regard to achieving optimum performance. Figure 1 shows a cross section through a typical hydrostatic unit. The interfaces described are indicated.

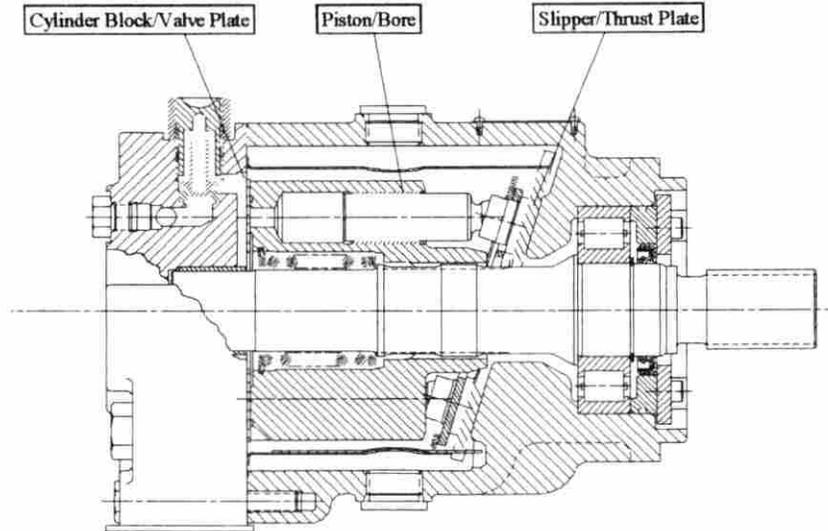


Figure 1: Cross section through a typical fixed-displacement, axial piston hydrostatic unit.

The block face/valve plate interface represents the truest form of hydrostatic bearing within the unit. In this instance, the block rotates about its axis with no translational motion and is supported from making contact with the valve plate by a pressure force acting on a closely toleranced area known as the balance area. Typical designs allow this bearing to be a very effective form of hydrostatic bearing with characteristically high efficiencies and load carrying capacity.

The piston/bore interface is a unique type of bearing in that actual surface-to-surface contact, otherwise known as boundary lubrication, occurs through much of the translational motion of the piston as it either moves into or out of the bore. Successful design of such a bearing is dependent primarily on limiting the magnitude of the reaction forces acting on the piston by the block wall, insuring that adequate lubrication is present at the point of contact, and applying material with excellent frictional properties.

Lastly, the slipper/thrust plate interface is unique in that this bearing is actually a type of hybrid hydrostatic bearing. The complete system of forces acting on the slipper is balanced primarily by hydrostatic forces acting on the face of the slipper. Also, at higher rotational speeds, a portion of the total load may also be resisted by hydrodynamic forces acting between the slipper and the thrust surface. The result is a complicated balance between the hydrostatic/hydrodynamic forces and the piston loading.

Many design parameters affect the successful operation of the slipper. These parameters must be properly chosen to insure efficient and trouble-free operation. High efficiencies need to be balanced with low wear in order to achieve satisfactory operation. Of all the various performance determining features of the slipper, one of the most critical is the size of the compensating orifice found in the slipper.

Several researchers have investigated the performance and associated parameters of axial piston slippers. Many investigators have looked into the effect of slipper surface curvature as being essential for proper operation. Hooke and Kakoullis (1978) showed that a small amount of running surface non-flatness is essential to assure proper operation. In later work Hooke and Kakoullis (1981) determined that tilting couples caused by centrifugal forces

acting on the slipper cause the slipper/piston assembly to rotate about its center axis. Böinghoff (1977) thoroughly investigated the frictional and operating behavior of slippers and developed a rough method for calculating losses at the start of and during operation. Research performed at the University of Aachen in Germany showed that, above some minimal rotational speed, slippers with no hydrostatic compensating orifice could operate satisfactorily while exhibiting levels of friction typical of running on a full oil film.

Nevertheless, little information is available on the effect of the compensating orifice size on performance. Koc, Hooke, and Li (1992) did explore the effect of orifice size on film thickness for underclamped and overclamped slippers. Additionally, they conducted film thickness measurements using blanked slippers, that is without orifice, whereby all loads were balanced purely by hydrostatic forces. They found that as the orifice size was decreased, the operating film thickness decreased as well and that slippers generally ran best when the orifice was kept small or blanked. However, their experiments were conducted using a single slipper test rig (as were the great majority of research experiments) and, as such, the results do not account for all factors which affect performance nor were efficiency effects measured.

This work attempts to explore in a straightforward manner the effects on operating performance by initially reducing and subsequently eliminating the slipper compensating orifice. The effects will all be measured in an actual axial piston unit which is currently available on the market.

OVERVIEW

The piston (#1) and slipper (#2) assembly from a hydrostatic unit is shown in Figure 2. Nine of these assemblies are commonly used within a unit. The slipper is usually made from brass and is crimped on to the piston. The majority of pistons produced today are made from a through-hardened alloy steel, for instance 4140 or 4340 steel.

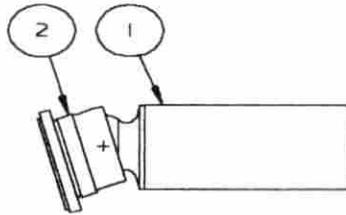


Figure 2: Piston and slipper of hydrostatic unit

Typical hydrostatic units have nine such assemblies in what is called a cylinder block as is shown in Figure 1. The assemblies are equally spaced in bores in the block at a pitch diameter D_p . The cylinder block is connected by means of a shaft to a prime mover, usually an internal combustion engine. As the shaft turns, the cylinder block rotates carrying the piston/slipper assembly with it. Due to forces acting on the piston and slipper, translational motion of the assembly takes place as the pistons move into and out of the block bores. As a result, the slipper moves in rotational sliding motion relative to the face of the thrustplate. The combination of the forces acting on the assembly as well as the sliding motion of the slipper leads to very demanding operating conditions for the slipper. Any number of books dealing with the subject of oil hydraulic power can be referred to for a more complete description of the functioning of hydrostatic units, for instance “Oil Hydraulic Power and Its Industrial Applications” by Ernst (1949).

Several forces act on the piston and slipper during normal operation. These forces are shown in Figure 3 together with a cutaway through the piston and slipper assembly.

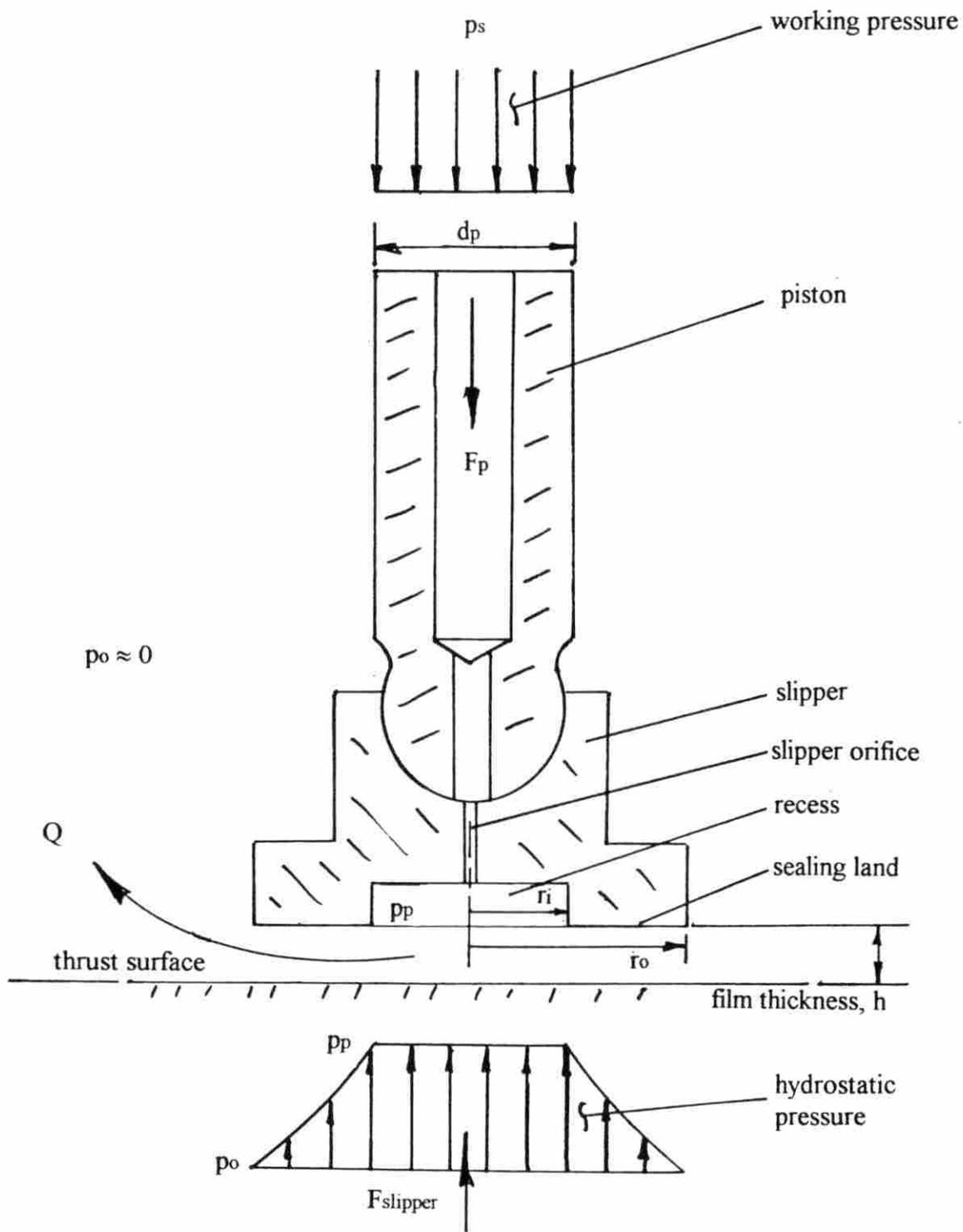


Figure 3: Forces acting on the piston and slipper.

The principal load acting on the piston is due to the working pressure, p_s . A hollow piston design allows for the working pressure to be connected to the slipper recess by means of an orifice in the slipper, also known as a compensating orifice. As a result, the recess pressure p_p develops. This recess pressure in turn gives rise to a hydrostatic pressure which acts across not only the recess but also across the sealing land of the slipper. The sealing land extends from the recess to the outside of the slipper, or, in terms of the nomenclature from Figure 3, from r_i to r_o .

The general characteristics of the hydrostatic pressure field are shown. Across the recess, the hydrostatic pressure is equal to the pressure, p_p . Between the recess and the outside diameter of the slipper (across the sealing land), the magnitude of the pressure decreases logarithmically to the reference pressure p_o .

During operation of the hydrostatic unit, a lubricating film of oil with thickness h separates the slipper from the thrust plate surface. This lubricating film represents a physical separation of the slipper from the thrust plate. Because of this separation, the recess is connected to the case of the hydrostatic unit and the reference pressure p_o .

As can now be seen, a direct connection exists between the working pressure side of the piston and the case of the hydrostatic unit. Since the reference pressure is taken to be equal to zero, hydraulic oil, designated Q in Figure 3, flows across the slipper orifice and sealing land as soon as working pressure is generated. Working pressures normally range from 1000 psi to over 5000 psi. This flow of oil through the slipper to the case is both beneficial and detrimental to the performance of the hydrostatic unit, as will be discussed.

The forces acting on the piston and slipper are found as follows. Working pressure, p_s , acts directly on the piston with diameter d_p giving rise to the force F_p . This force acts to clamp the slipper to the thrust plate. The force F_p can be found with the equation

$$F_p = \frac{\pi}{4} * d_p^2 * p_s$$

where F_p is in lb (force), d_p is in inches, and p_s is in psi.

Depending on the operating mode of the hydrostatic unit, additional loads may add or subtract from the piston load F_p . These loads arise due to the piston inertia forces, friction between piston and bore, and any spring hold down forces. These loads will be neglected in this analysis since they represent only a minor effect on the overall loading acting at the slipper face.

The hydrostatic reaction load at the slipper face is found by integrating the area across the face of the slipper and multiplying this area times the magnitude of the hydrostatic pressure. The resulting form of the equation is

$$F_{\text{slipper}} = \frac{\pi}{8} * p_p * \frac{(d_o^2 - d_i^2)}{\ln\left(\frac{d_o}{d_i}\right)}$$

where $d_o = r_o * 2$ and $d_i = r_i * 2$, both in inches. The recess pressure p_p has the units psi. The hydrostatic reaction load is a function of the recess pressure and slipper geometry only.

For some slipper designs, a hydrodynamic lifting force, caused by the slipper moving relative to the thrust plate, may also partially assist in reacting against the piston load. Whether or not the hydrodynamic lift component is significant when compared to the hydrostatic force can be found by calculating the balance of the slipper.

The balance of a slipper is a useful tool for evaluating slipper designs. Slipper balance is the ratio between the hydrostatic force acting on the slipper face and the force on the piston. Assuming that the pocket pressure equals the working pressure ($p_p \approx p_p$), the “balance equation” can be written as

$$E_{\text{th}} = \text{balance} = \frac{F_{\text{slipper}}}{F_{\text{piston}}} = \left\{ \frac{0.5 * (d_o^2 - d_i^2)}{d_p^2 * \ln\left(\frac{d_o}{d_i}\right)} \right\}$$

For $E_{\text{th}} \leq 1$, the slipper is said to be “underbalanced” (overclamped). In other words, the piston load is larger than the hydrostatic load and the slipper is held clamped to the mating

thrust surface. Such a slipper design must rely partially on hydrodynamic forces to assist in counteracting the piston load and lifting the slipper off the thrust surface.

For $E_{th} > 1$, the slipper is described as “overbalanced” (underclamped) and the hydrostatic loading at the face of the slipper exceeds that of the piston. This type of slipper is considered to be held separated from the thrust surface solely by hydrostatic pressures. Hydrodynamic effects are minimal with overbalanced slippers. Values for E_{th} commonly range from about 0.8 to 1.2.

As has been mentioned, the slipper operates on a thin film of oil which separates the slipper face from the thrust surface. The presence of an oil film separating the surfaces is a functional necessity for the slipper since any direct contact between the mating surfaces results in boundary lubrication conditions and excessive frictional losses. The oil film is generated primarily from the previously mentioned effect of hydrostatic lift and also, depending on design, by hydrodynamic effects.

The magnitude of the operating film thickness is critical for a number of reasons. First, the oil film must be thick enough during operation to maintain full surface asperity separation between the slipper and the mating surface. Otherwise, frictional losses will become excessive and wear could possibly occur. A lower limit of the required film thickness can be found by relating the minimum film thickness required to the composite surface roughness of the slipper and the thrust plate. This relationship is termed the lambda ratio.

$$\lambda = \frac{h}{\sqrt{\sigma_{slipper}^2 + \sigma_{t'plate}^2}}$$

Here, $\sigma_{slipper}^2$ is the RMS surface roughness of the slipper while $\sigma_{t'plate}^2$ is the RMS surface roughness of the thrustplate. For values of λ greater than three, a full film of oil exists between the two surfaces and no asperity contact occurs. Based on this, then, a lower limit of the film thickness can be specified as follows.

$$h \geq 3 * \sqrt{(\sigma_{slipper}^2 + \sigma_{t'plate}^2)}$$

In actuality, the film thickness will be quite a bit greater than this lower limit, yet this relationship helps to visualize what minimum thickness is required.

On the other hand, the oil film cannot become too thick or else flow out of the system will become excessive. The oil which flows through the slipper and across the lands results in a direct volumetric efficiency loss of the hydrostatic unit. Designs which allow too thick of an oil film will result in high efficiency losses.

In summary, proper sizing of the slipper orifice is key to an optimum film thickness. With too thick of an oil film, the unit volumetric efficiencies become excessive and the power required to pump the oil through the slipper becomes unacceptable. A thinner oil film offers a greater resistance to leakage flow resulting in better efficiencies at the possible expense of increased surface heating, frictional power losses, and wear.

The orifice in the slipper has the important function of regulating the flow of oil into the slipper pocket and, consequently, across the lands of the slipper. A cross section through a slipper is shown in Figure 4.

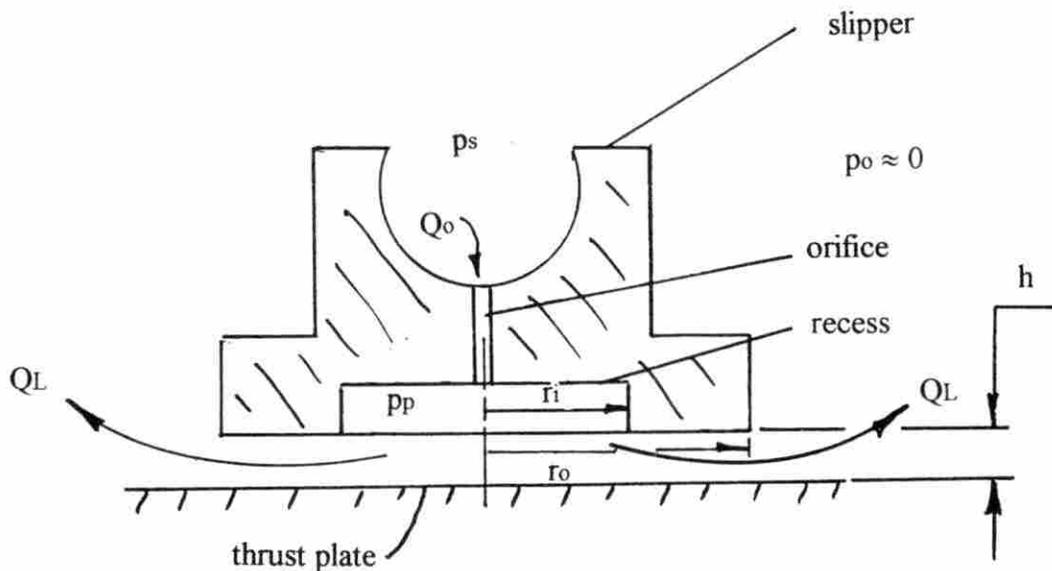


Figure 4: Cross section through an orificed slipper.

As a result, the pressure in the pocket, p_p , will be somewhat less than the working pressure, p_s . The flow through the slipper orifice can be described by the orifice equation as

$$Q_o = CA\sqrt{\frac{2\Delta P}{\rho}}$$

where Q_o is the orifice flow, C is defined as the orifice coefficient, A the cross sectional area of the orifice, ΔP the pressure differential acting across the orifice, and ρ the density of the oil.

The orifice coefficient C is required in the equation to account for the contraction of the fluid jet after it leaves the orifice as well as the fluid friction and turbulence that develops as the oil flows through the orifice. In addition, the coefficient varies with the ratio of the upstream passage diameter to the orifice diameter. The flow through most orifices is turbulent (high Reynolds numbers) and, according to Merritt (1967), the value of the orifice coefficient can be taken as, based on experience, $C=0.611$. This value applies to a sharp-edged round orifice.

Based on considerations by Merritt, if the flow through the orifice is turbulent and the upstream passage diameter is much greater than the orifice diameter, C can be taken to be approximately equal to 0.6. Furthermore, with the assumption that

$$C\sqrt{\frac{2}{\rho}} \approx 100 \text{ in}^2 / \sqrt{\text{lb} - \text{sec}},$$

the orifice equation can be simplified to the form

$$Q_o = 100A\sqrt{p_s - p_p}$$

where the flow has the units in^3/sec , the orifice area has the units in^2 , and the pressures are in psi. This form of the equation was used in this work for finding the slipper orifice flow, Q_o .

The reader is encouraged to refer to Merritt's book for further considerations into the modifications that must be made for laminar flow through the orifice. Additional references on the potential errors which may arise due to application of the simplified form of the orifice equation can be found in the work by Johnson (1963) as well as the paper by Scharrer and Hibbs (1990).

Flow from the slipper recess to the case is a function of the film thickness between the slipper land and the mating thrust surface. The flow of oil across the land with separation h is given by the equation

$$Q_L = \frac{\pi}{6} p_p h^3 \left(\frac{1}{\eta \ln \frac{d_o}{d_i}} \right)$$

where Q_L is in in^3/sec , η is the oil viscosity in $\text{lb}/\text{sec}\text{-in}^2$, p_p is in psi , h is in inches, and d_p and d_i are also in inches. The equation is valid for flat, circular hydrostatic bearings.

Flow balance considerations necessitate that the flow into the slipper pocket equals the flow out across the lands. In equation form, this becomes

$$Q_O = Q_L.$$

Substitution of the appropriate relations gives

$$100A\sqrt{p_s - p_p} = \frac{\pi}{6} p_p h^3 \left(\frac{1}{\eta \ln \frac{d_o}{d_i}} \right).$$

The film thickness can now be estimated. First, based on considerations from Shute and Turnbull (1962) as well as experimental results from Koc, Hooke, and Li (1992), the assumption can be made that the piston load is balanced completely by the hydrostatic load at the slipper face, or,

$$F_{\text{slipper}} \approx F_p$$

As a result, an expression for the slipper pocket pressure, p_p , can be found from the equation for the hydrostatic slipper reaction load, F_{slipper} , in terms of the piston force, F_p . The expression for p_p then becomes

$$p_p = 8F_p \frac{\ln\left(\frac{d_o}{d_i}\right)}{\left[\pi(d_o^2 - d_i^2)\right]}$$

With the value of p_p known, the slipper orifice flow, Q_o can be found since the working pressure is a known quantity. Substitution of this expression for p_p in to the equation for Q_L , which is known from finding Q_o , and solving for the film thickness h gives

$$h = \left\{ \frac{\left[6Q_L \eta \ln \left(\frac{d_o}{d_i} \right) \right]}{\pi p_p} \right\}^{1/3}$$

When typical values of slipper orifice size, slipper geometry, and operating conditions are substituted into the above equation, film thicknesses of the order of 0.0005 inches are calculated. Based on experience, this result for slipper film thickness is reasonable.

A comparison can now be made to a blanked slipper design. If the orifice and recess are removed completely, or blanked, no hydrostatic reaction load can develop and the slipper will be supported totally by a hydrodynamic film as long as the slipper is in relative motion to the thrust surface. The load of the piston acting on the slipper is countered by the hydrodynamically generated pressure within the film. Figure 5 shows a cross section through a blanked slipper.

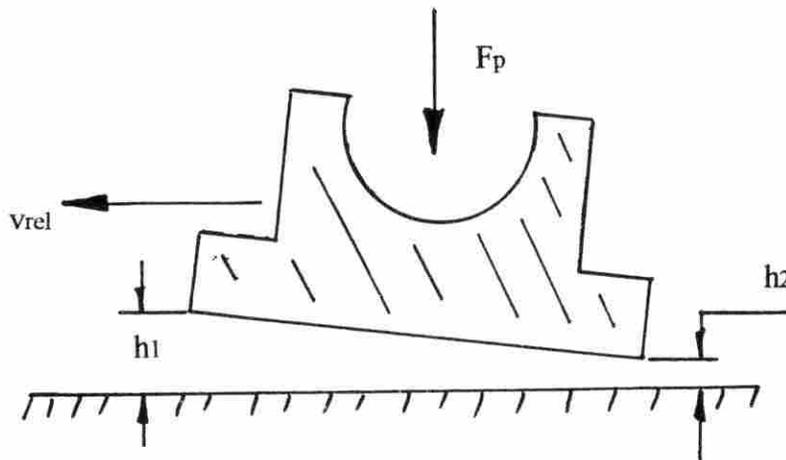


Figure 5: Cross section through a blanked slipper

The opposing surfaces of the slipper and the thrust plate can be considered to be two conformal planes in relative motion to one another. A simplified form of the Reynolds equation, which has been modified for a slipper type bearing, may be used to estimate the film thickness present between the slipper face and the mating thrust surface. In using the simplified Reynolds equation, it is assumed that the flow of oil between the two surfaces is laminar, the fluid film is thin compared with the size of the bearing, and that the dominant forces are due to viscosity. Already simplified, the relationship becomes

$$h = \left(\frac{\pi \eta k N D_p d_o^3}{10 F_p} \right)^{1/2}$$

Here k is a numerical factor which is a function only of the ratio between the inlet and outlet film thicknesses of the hydrodynamic film and is actually quite insensitive to this ratio according to Hutchings (1992). The maximum load carrying capacity of the hydrodynamic film occurs when $k=0.027$ and this value will be used in this work to estimate the slipper film thickness. Additionally, N is the rotational speed in rpm of the piston/slipper assembly due to the block's rotation, D_p is the pitch diameter of the block in inches, and d_o is the outside diameter of the slipper land in inches.

If the same values from the hydrostatic film thickness calculation used previously are substituted into the above equation for hydrodynamic film thickness, a result of $h=0.0002$ inches is obtained. Once again, this is a reasonable answer based on experience.

The efficiency losses can be calculated based on the above relationships. Efficiency losses to be considered are, first, that due to the pumping of oil across the orifice and slipper lands into the unit's case. Furthermore, the mechanical losses generated at the slipper/thrust surface interface due to friction lead to a loss of power. Both mechanical and volumetric losses are directly related to the fluid film thickness, yet the dependencies are diametrically opposite as will be shown shortly.

Pumping losses are commonly examined by evaluating the volumetric efficiency. Volumetric efficiency is calculated by relating the volume of oil actually being pumped to the theoretical displacement of the pump being tested. The relationship is expressed as

$$\text{Volumetric Efficiency, } \eta_v = \frac{\text{Output Flow, gpm}}{\text{Theoretical Output Flow, gpm}}$$

The theoretical output flow of a hydrostatic unit is found by the equation

$$\text{Theoretical Output Flow, gpm} = \frac{\text{Displ}_{\text{Theo}} * N}{231 \text{ in}^3 / \text{gal}}$$

where $\text{Displ}_{\text{Theo}}$ is the theoretical displacement of the unit in in^3/rev and N is the unit speed in rev/min .

Sources of volumetric losses within a hydrostatic unit are the slipper/thrust plate interface, the piston/bore clearance, as well as the cylinder block/valve plate interface. Of these, the slipper and thrust surface interface usually is the most significant. Any reduction in the leakage at this position will result in an increase in the volumetric efficiency of the unit. The amount of leakage occurring at the slipper interface is found from the equation for Q_L which is a function of the film thickness. Clearly, as the film thickness increases, so does the leakage. Therefore, a reduction in the film thickness (and, thus, Q_L) leads to increased volumetric efficiencies. Furthermore, with the orifice and recess removed, no leakage losses whatsoever can exist due to the slipper.

Mechanical efficiencies are found in the same manner as the volumetric counterpart. The ratio of the input torque provided by the prime mover to the torque delivered by the pump is known as the mechanical efficiency.

$$\text{Mechanical Efficiency, } \eta_m = \frac{\text{Torque}_{\text{out}}}{\text{Torque}_{\text{in}}}$$

Here, the hydraulic torque out is found from

$$\text{Torque}_{\text{out}} = \frac{\text{Displ}_{\text{Theo}} * (p_s - p_c)}{2\pi}$$

where p_c is the charging pressure of the hydrostatic unit in psi.

Mechanical losses arising at the slipper surface, assuming a full oil film exists, are due to the shearing of the oil present in the clearance, h , between slipper and thrust surface. An

expression for the generated shearing torque, designated T_{shear} , is derived as follows. The velocity of the slipper relative to the thrust surface is expressed as

$$v_{\text{rel}} = \frac{2\pi ND_p}{120}$$

where v_{rel} is in inches/sec, N is the unit rotational speed in rev/min, D_p is the cylinder block pitch diameter in inches. Applying Newtonian fluid theory, an equation for the shear stress, τ , arising from the shearing of the oil at velocity v_{rel} , can be stated as

$$\tau = \frac{N\pi\eta D_p}{60h}$$

Here, τ is expressed in lb/in².

This action of shearing the oil results in a torque loss for the hydrostatic unit. The force associated with the torque can be found by multiplying the shear stress, τ , by the total face area of the slipper, A_{slipper} . The moment arm is $D_p/2$. The torque loss T_{shear} then becomes

$$T_{\text{shear}} = \frac{D_p}{2} (\tau * A_{\text{slipper}}) = \frac{\pi\eta ND_p^2 A_{\text{slipper}}}{120h}$$

This equation shows that, as the film thickness decreases, the torque loss, T_{shear} , increases.

To summarize, the film thickness plays a key role in the overall efficiency of a hydrostatic piston unit. On the one hand, a reduction in film thickness leads to an increase in the volumetric efficiency of a unit by reducing the leakage losses of the slipper. Conversely, the same reduction in film thickness leads to increased oil shear torque losses with the result that the mechanical efficiency is reduced. The subject of this work is to investigate the effect of reducing the compensating orifice size and, as a result, the film thickness on the overall efficiency of an actual hydrostatic unit.

In order to visualize how the volumetric losses and torque losses are affected by the reduction or elimination of the orifice, a set of assumed conditions can be used in conjunction with the above derived equations to predict efficiency changes due to modifications in the orifice. Figure 6 shows the calculated increase in unit volumetric efficiency along with the

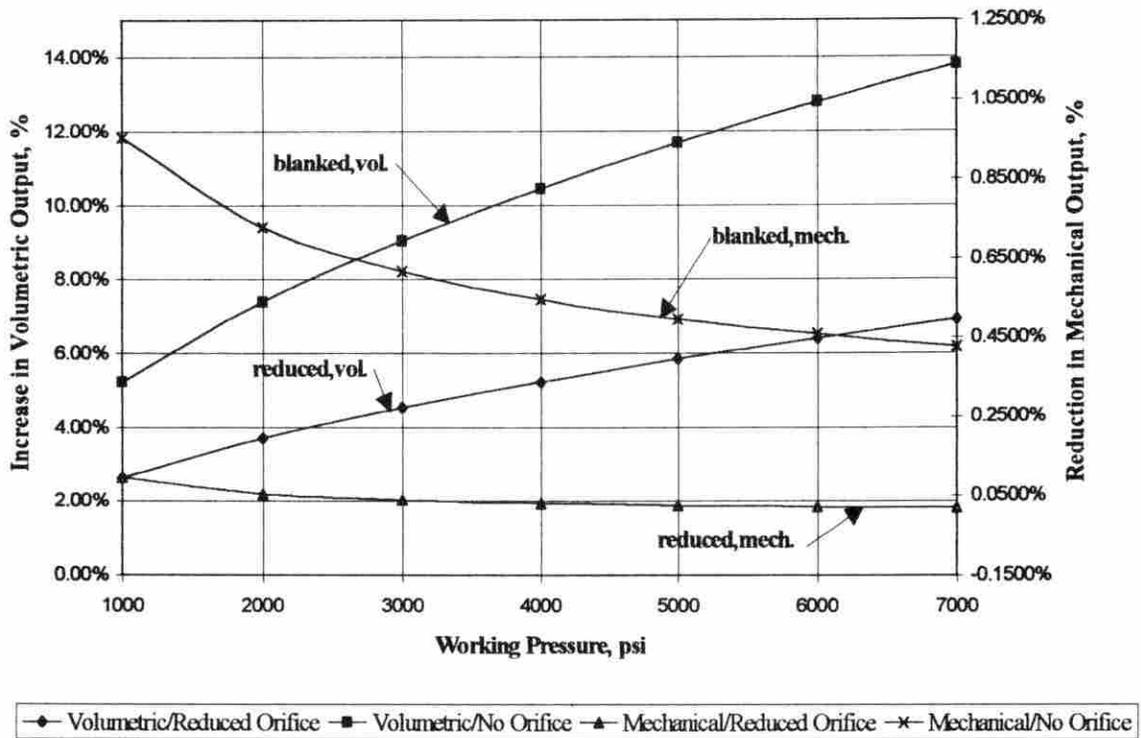


Figure 6: Predicted effect of orifice size on hydrostatic unit efficiency compared to standard configuration.

corresponding decrease in mechanical efficiency versus working pressure for slippers with reduced compensating orifice and blanked running surface. In this calculation, the reduced orifice area is one-half the area of the standard design.

The increase in volumetric efficiency is calculated on the basis of the reduction in Q_L per slipper as the orifice size is initially reduced and then blanked. To equate the per slipper flow to a hydrostatic unit basis, 4.5 pistons (slippers) are assumed to be loaded by working pressure, p_s . The total number of pistons in most hydrostatic units is nine and, on average, only half are exposed to working pressure at one time. The increase in unit efficiency can then be found from

$$\text{Unit Volumetric Efficiency Increase} = \frac{4.5(Q_{L,\text{standard}} - Q_{L,\text{reduced}})}{\text{Theoretical Output of Unit}}$$

where $Q_{L,\text{standard}}$ and $Q_{L,\text{reduced}}$ are the per slipper leakage flows and must be converted to gallons per minute for this calculation. The term in parenthesis is a positive number and results in an increase in volumetric efficiency.

For the mechanical losses, a similar approach is used.

$$\text{Unit Mechanical Efficiency Reduction} = \frac{4.5(T_{\text{shear,standard}} - T_{\text{shear,reduced}})}{\text{Theoretical Mechanical Efficiency of Unit}}$$

$T_{\text{shear,standard}}$ and $T_{\text{shear,reduced}}$ represent the torque to shear the film of oil. The term in parenthesis results in a negative number and, thus, a reduction in efficiency.

A rotational speed of 1500 rpm was used in the calculations for Figure 6. This is a common operating speed for many applications.

The curves in Figure 6 indicate that, as the orifice size is initially reduced and finally eliminated, the increase in volumetric efficiency is roughly an order of magnitude greater than the reduction in mechanical losses. Hence, based on this simple model, noticeable gains in overall unit efficiency are expected with first a reduction in size and finally the elimination of the compensating orifice.

Even so, the majority of hydrostatic units on the market today perform to high levels of efficiency and are not as inefficient as the curves in Figure 6 might indicate. The calculated

improvements in volumetric efficiency are more than likely greater than that which can actually be attained. Nevertheless, the curves in Figure 6 do offer insight in to the relative improvements which are possible in efficiency when the orifice is either reduced or eliminated.

To evaluate the predictions, testing was performed with a nine-piston, fixed-displacement hydrostatic unit. A range of speeds and pressures was chosen so that a better understanding of the effects that these two operating parameters have on the performance of the various slipper designs could be obtained. The speed and pressure ranges encompass the majority of actual field application conditions.

A matrix of speed versus pressure conditions was used as shown in Table 1.

Table 1: Matrix of testing conditions

Pressure Speed	1000 psi	3000 psi	5000 psi
500rpm	Kit 1	Kit 4	Kit 7
1500rpm	Kit 2	Kit 5	Kit 8
2500rpm	Kit 3	Kit 6	Kit 9

There are several sources of mechanical and volumetric losses within a hydrostatic unit. Mechanical losses arise not only from mechanical friction and oil shear at the slipper face but also from friction within the bearings at each end of the unit's shaft (a journal and roller bearing), oil shear at the cylinder block interface, mechanical friction between piston and bore, and mechanical friction between the slipper retaining ring and its associated holddown bearing. Sources of additional volumetric losses are found at the cylinder block interface as well as between piston and bore. A testing arrangement had to be chosen such that these additional losses were either negligible when compared with the changes associated with the slipper configuration or were held constant from run to run.

In order to evaluate the impact of compensating orifice size changes on efficiency, a hydrostatic unit was chosen which had been tested for long hours on a high pressure cycle test. The unit was "well broken-in", so to speak.

For each matrix condition, a new set of nine pistons and cylinder block -- described as a "kit" -- together with an associated mating thrust plate, the hardened steel plate which acts as the thrust surface for the slippers, were used within this same hydrostatic unit. This was done in order to minimize the effect of the other described losses. For example, the two shaft bearings were well broken in due to the long hours of endurance testing, and, as such, would not exhibit any significant variability in losses from run to run. The cylinder block, once rotating, runs on a full hydrostatic film of oil, the thickness of which is a function of speed and pressure primarily. Because the matrix parameters of speed and pressure were closely controlled during testing, the variability due to the cylinder block interface is minimal. Similar statements can be said for all other sources of losses. In other words, by using the same hydrostatic unit throughout testing while only changing the kit out for each matrix run, efficiency results were influenced only by the change in slipper design.

A new thrust plate was exchanged each time with the kits as well. As some run-in occurs between the slippers and the thrust plate, using only one thrust plate for all tests would have distorted the mechanical losses for slippers as testing progressed. Using a single thrust plate with each separate kit minimized the break-in effects between different hardware.

To determine the actual variability associated with the additional losses, supplementary testing was conducted at one matrix condition to evaluate the variability associated with this approach from run-to-run and between kits. This data allowed an estimate of the distribution and confidence intervals for all the results.

Since the same basic unit was used for each test, the volumetric displacement per revolution remained the same for each test run. This fact is important because the exact displacement of the unit was not known and, indeed, is very difficult in reality to determine precisely. Yet, to evaluate the volumetric and torque impacts accurately, the displacement of the unit must at least be constant to be able to make a comparison of slipper designs. A normalization of results can then be made to allow accurate comparisons of the various designs.

TESTING

A nine-piston, fixed displacement axial piston unit was chosen which is currently available on the market today. The unit had a theoretical volumetric displacement of 55cc/rev and was rated to a maximum pressure of 7000 psi and speed of 4000 rpm. The components used in the unit, except for the kits and thrust plates, had been run in high pressure cycle testing for several hundred hours prior to efficiency testing. All hardware used in the hydrostatic unit was produced with high volume production tooling to tightly controlled and monitored manufacturing processes.

In order to better visualize the test circuit, a hydraulic circuit diagram has been made. The diagram is shown in Figure 7.

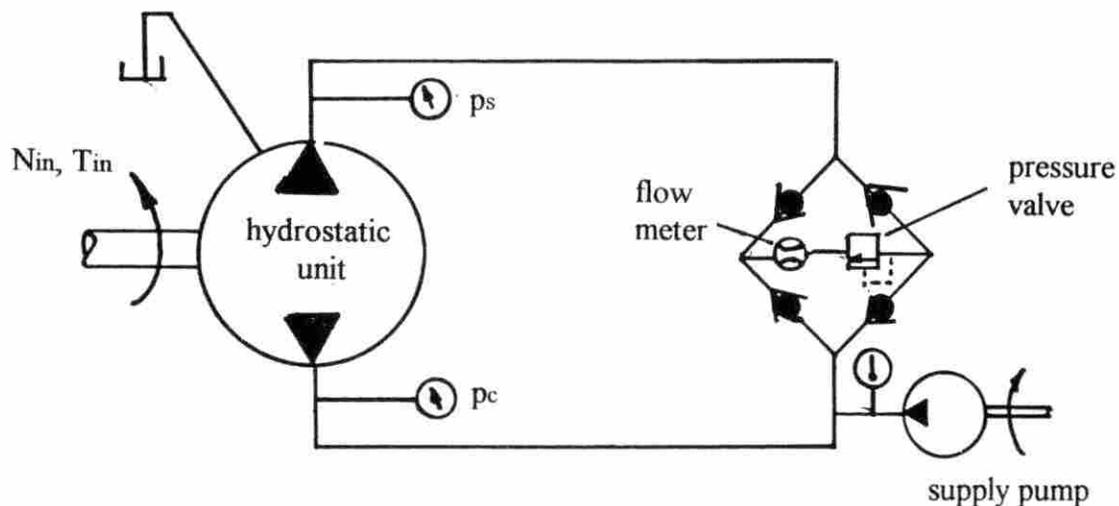


Figure 7: Hydraulic circuit diagram of the test stand.

Testing was performed on a test stand equipped with a 600 Hp variable speed electric prime mover. The prime mover was produced by General Electric, Model 5GE-769C1, Type CD.

Transducers were used to measure input torque, working pressure, and output flow of the hydrostatic unit. Torque was measured with a torque shaft produced by Himmelstein &

Co., Model 9-0295 (6-3) which had a measurement range of 0-6000 in-lbs. Working pressures were measured using a pressure transducer made by Sensotec, Model Z-996-01. The pressure transducer was rated from 0-10,000 psi. Lastly, output flows from the units were measured using a Fisher and Porter turbine flow meter which was rated from 1 to 75 gpm. Transducers were calibrated on the test stand at the start of testing sessions.

The full scale percent error associated with each transducer was determined before testing. The pressure transducer demonstrated a maximum full scale percent error of 0.29%. The Himmelstein & Co. torque transducer was found to have an excellent maximum full scale percent error of only 0.14%. Finally, the flow meter exhibited a maximum full scale error of 0.18% at the low flows associated with the 500 rpm test condition while the error improved to 0.13% or better at higher test speeds and flows.

A data acquisition system was used to collect the transducer data. The board used was a DT-28 with a sampling rate of one data point per second. Since steady-state testing conditions applied, this sampling rate was acceptable.

The hydraulic oil used throughout testing was Mobil Type F. This is a general purpose hydraulic oil with no anti-wear additives.

Oil inlet temperatures were controlled to within a temperature range of $125^{\circ} \pm 5^{\circ}\text{F}$. Temperature had to be controlled due to the effect of oil viscosity, which varies with temperature, on efficiencies.

A small constant displacement pump was used to supercharge the hydrostatic unit with oil. The flow from the supercharging pump was maintained at 5 gpm. This supercharging flow resulted in a charge pressure for the unit of 380 ± 10 psi.

The slippers were produced from a high alloy brass material with the following chemical composition:

61.5% Copper

34.5% Zinc

4% Silicon, Manganese, Lead

This particular type of brass material is well qualified for highly mechanically stressed bearing applications where excellent friction and wear properties are required.

The running surfaces of all slippers tested were machined to a profile flatness of 0.00008 inches and a surface finish of 8 μ inches R_a . For the slippers equipped with a compensating orifice, the orifice extended centrally through the slipper connecting the slipper's recess to the piston and, as a result, working pressure. For the blanked slipper configuration, neither an orifice nor a recess was present. The running surface was machined to smooth surface per the above specified parameters.

The orifice size for the standard slipper design is 0.032 inches. The orifice size for the reduced orifice configuration, 0.018 in., was chosen to result in exactly half the effective opening area of the standard size.

For the orificed slippers, the following geometry was used:

$$r_o=0.465\text{in.}$$

$$r_i=0.300\text{in.}$$

$$E_{th}=105\%$$

Both of the orificed slipper designs had a balance, E_{th} , of 105% -- both underclamped designs. The blanked slipper had an outside diameter of 0.930 inches ($r_o=0.465$ inches). Of course, neither r_i nor E_{th} applied for the blanked configuration since no recess was present.

A through hardened 1070 steel material was used in the mating thrust plate. The running surface of the thrust plate was lapped to a 12 μ inches R_a surface finish and a flatness of 0.0005 inches/inches.

Testing proceeded as follows. For each matrix combination of speed and pressure, a separate kit and thrust plate were assembled into the hydrostatic unit. To maintain test run time consistency between kits, every kit was run approximately 10 minutes at 1500 psi and 1500 rpm until the inlet temperature reached the targeted range. Once the inlet temperature was reached, the matrix speeds and pressures were set and a computer data acquisition system was used to collect the transducer data during a 100 second test time duration. Following a test run, the prime mover was shut down and a new kit and thrust plate were inserted into the

unit. The next kit was then tested per the above steps. This process was repeated for each of the nine matrix positions and associated kits of the three slipper designs.

To evaluate the variability associated with this testing procedure, a single matrix condition was chosen and three separate, untested kits from each of the three slipper designs were run per the above procedure. These results were used to estimate the confidence limits and standard deviations associated with run-to-run and kit-to-kit changes.

It should be noted that through the course of testing the torque transducer became defective. Due to this, a new transducer was assembled into the test stand. Since all the slipper designs had previously been tested with the bad torque transducer, the complete testing matrix was repeated with the new transducer. This simply resulted in all kits being tested approximately 20 minutes rather than 10 minutes.

RESULTS

The effect of slipper design on the volumetric efficiency of the hydrostatic unit as a function of working pressure for the test input speeds are presented. Figures 8,9, and 10 present the volumetric efficiencies at 500, 1500 and 2500 rpm, respectively.

In presenting the data for efficiency, a curve-fitting routine (Microsoft Excel) was applied. Some results may be implied that are not accurate.

The volumetric efficiency data has been normalized, as already mentioned. The normalizing parameter used was the maximum overall volumetric efficiency value obtained from the test conditions.

Several observations can be made from the volumetric efficiency results. All curves exhibit similar trends in that the volumetric efficiency declines with increasing working pressure. This is to be expected since an increase in pressure acting on the slippers and other various bearings within the unit generally results in an increase in leakage flow.

The effect of speed on the efficiency performance is most significant at 500 rpm. Above 500 rpm speed appears to have little effect on the leakage as the volumetric efficiencies remain relatively constant with changes in speed. The 500 rpm results show a noticeable reduction in efficiency for all slipper designs, especially for the standard design.

A comparison of the reduced orifice and blanked orifice designs shows little difference in efficiencies at the speeds of 1500 rpm and 2500 rpm. At 500 rpm, the blanked design is approximately 1% better than the reduced design for the full pressure range.

The standard design shows the poorest efficiencies under all conditions. At the speeds of 1500 rpm and 2500rpm, the standard design is consistently 1% to 2% less efficient than the blanked orifice design. This is significant.

The performance of the standard design at 500 rpm is even poorer. Here differences in efficiencies of up to 3.5% are seen when compared to the blanked.

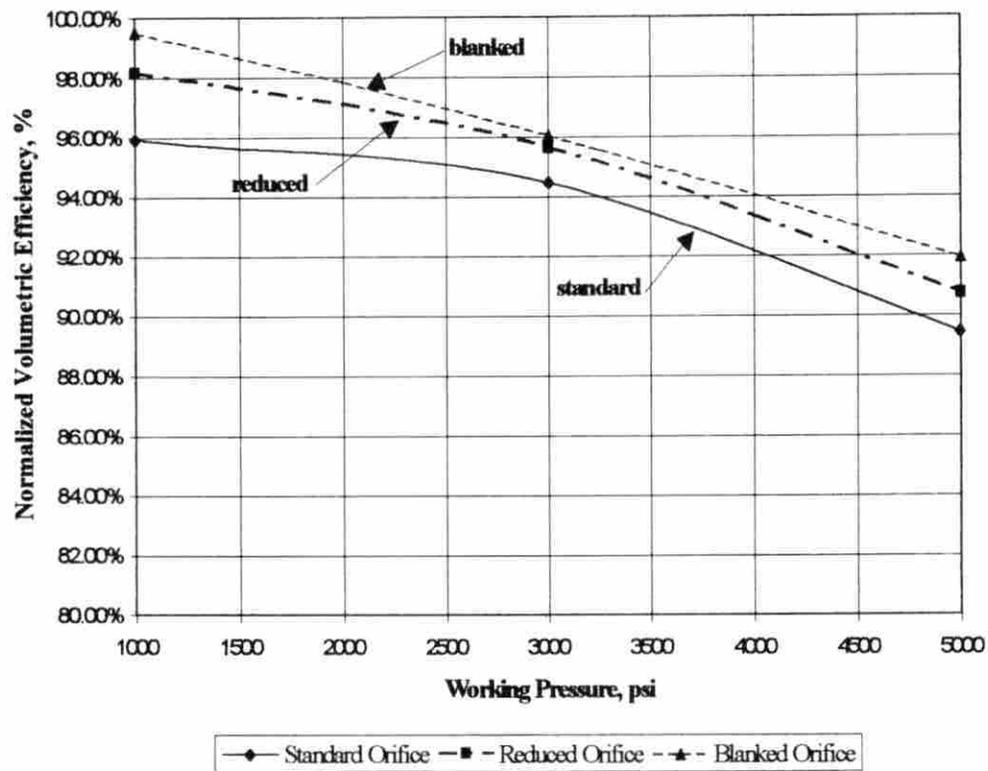


Figure 8: Normalized volumetric efficiencies versus system pressure at 500 rpm.

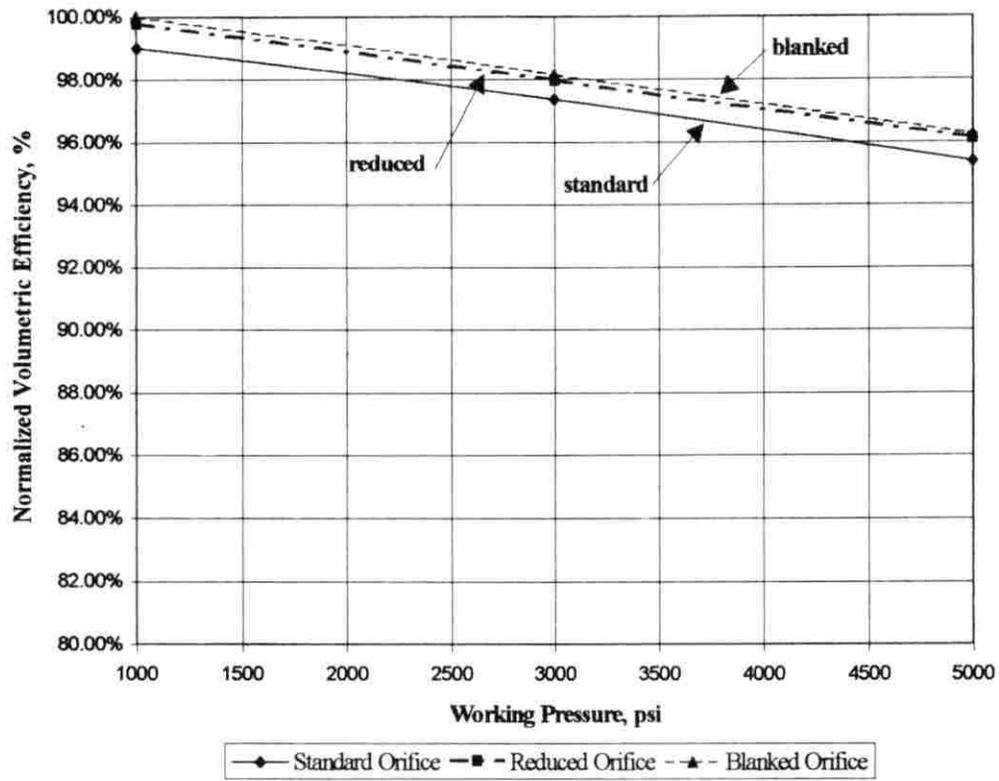


Figure 9: Normalized volumetric efficiency versus system pressure at 1500 rpm

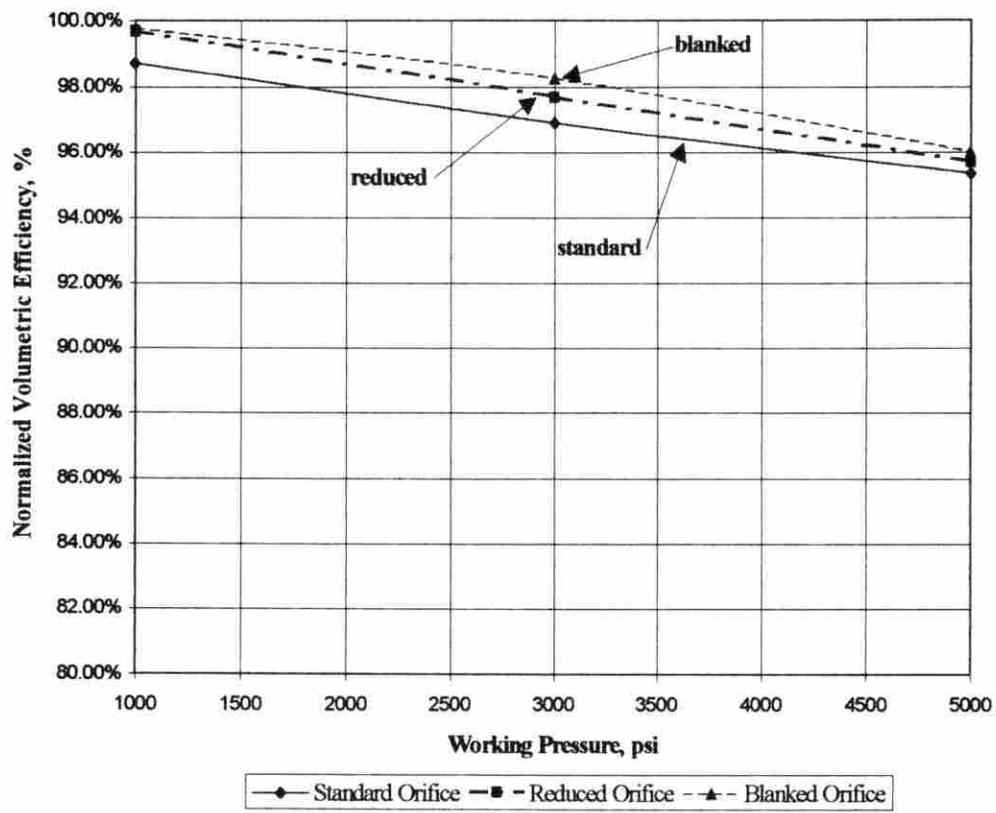


Figure 10: Normalized volumetric efficiency versus system pressure at 2500 rpm.

As has been seen, the strongest impact on volumetric efficiency from the blanked design appears to come in the reduced speed ranges. Here the blanked design is plainly better than the reduced orifice slipper and significantly better than the standard configuration.

In general, hydrostatic units today are designed to have high volumetric efficiencies under most operating conditions and, in actuality, losses at the slipper/thrust surface interface are not of great magnitude. The actual potential efficiency improvements with today's pump designs and hydraulic fluids are not great and, as such, the improvements predicted by the model are unrealistic. However, some improvements are possible and, depending on the demands of certain applications where volumetric efficiencies are crucial, changes in the size of the compensating orifice may offer a decided advantage.

The effects of slipper design on the mechanical efficiency of the hydrostatic unit as a function of working pressure for the test input speeds are presented. Mechanical efficiency results are presented in Figures 11, 12, and 13. Again, the normalizing parameter that was used was the maximum mechanical efficiency value measured throughout the course of the testing.

Considering the curves of mechanical efficiency, several conclusions can again be drawn. Except for the standard design at 500 rpm and 3000 psi, all the curves follow the same trends remarkably well. Mechanical efficiencies for the three designs, independent of speed, increase with increasing working pressure with the largest gain occurring between 1000 and 3000 psi. In this range of pressure, the mechanical efficiencies for all designs increase substantially by 3% to 7%. For the range of pressures between 3000 and 5000 psi, an increase in efficiency occurs as well, although not to the degree as that seen with the lower pressures. In this range, an increase of 1% to 2% is seen.

Mechanical efficiencies for the standard design at 500 rpm are clearly more varied than for the other two designs. Rather than continually increasing with pressure, the efficiency drops off after peaking at 3000 psi. The reduction in efficiency above 3000 psi does not follow the trend in results shown by the other designs nor of the standard design itself at

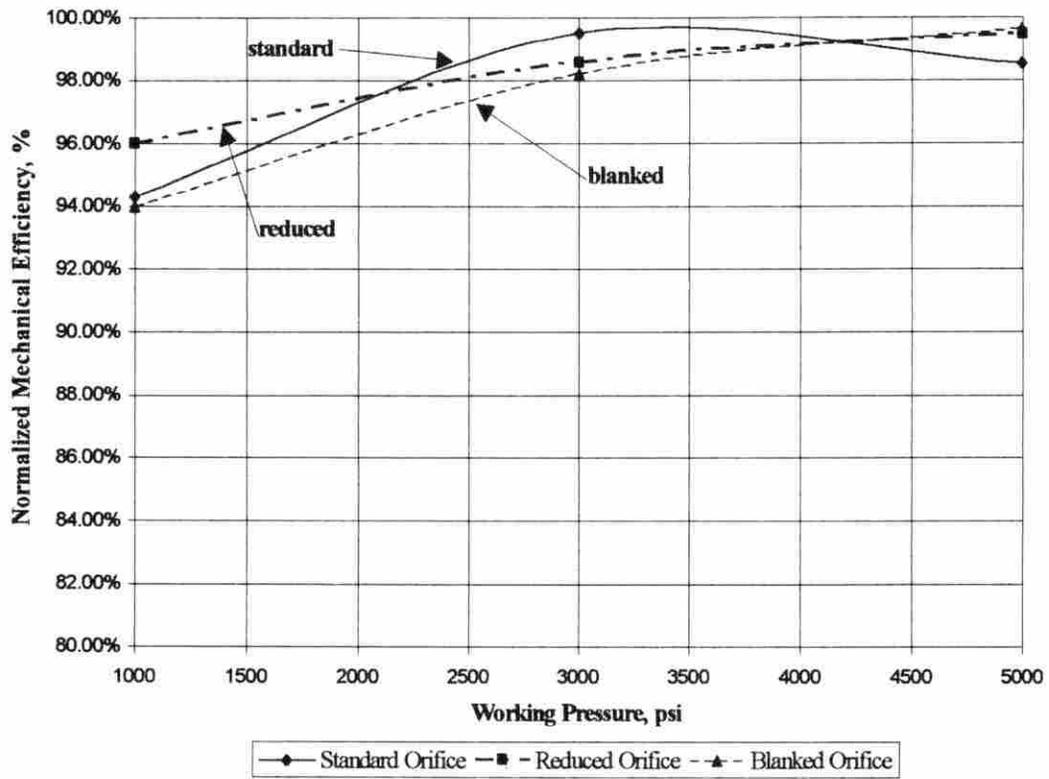


Figure 11: Normalized mechanical efficiency versus system pressure at 500 rpm.

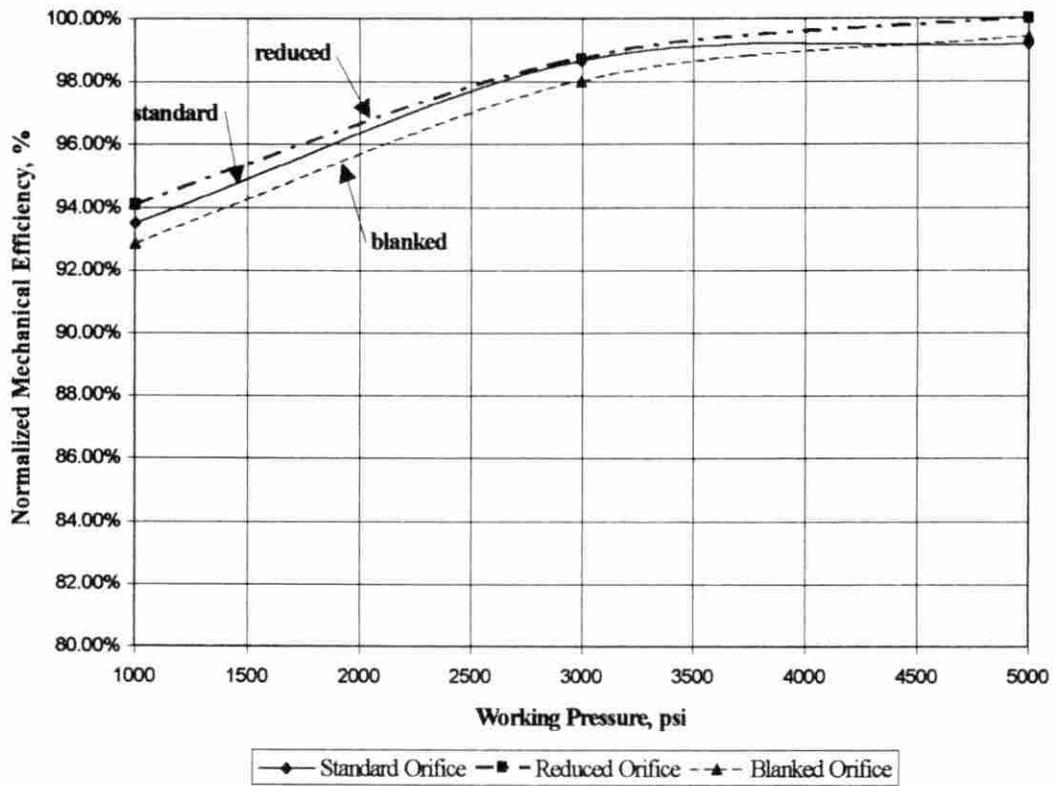


Figure 12: Normalized mechanical efficiency versus system pressure at 1500 rpm.

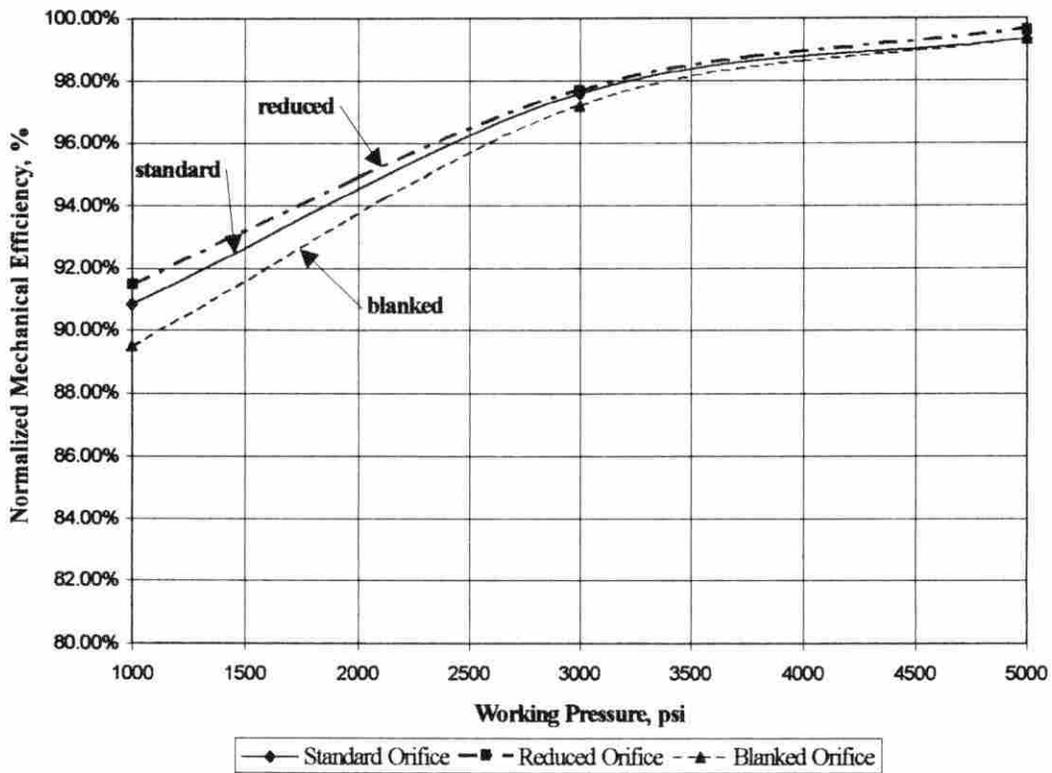


Figure 13: Normalized mechanical efficiency versus system pressure at 2500 rpm.

speeds above 500 rpm, yet this characteristic has been seen previously in other efficiency testing conducted on this particular product.

When the reduced and blanked designs are compared, a clear similarity is seen. The two designs appear to be more consistent with regard to efficiency changes due to changes in speed and pressure. The conclusion may be drawn that the two designs exhibit a greater degree of performance stability throughout the range of test parameters.

Throughout the speed range all designs show very little change in losses at higher pressures. At the middle pressure range, except for the standard design as has been said, losses are consistent but seem to be increasing slightly with increasing speed. The maximum pressure efficiencies do not appear to be affected by speed at all. However, at the low pressure range, speed effects are quite noticeable and losses are clearly increasing with increasing speed.

A general comparison between the three designs plainly shows that, except for the 500rpm/3000 psi data point, the reduced orifice design consistently exhibits the greatest mechanical efficiencies. Furthermore, the blanked design consistently demonstrates the worst mechanical efficiencies. The standard design falls in between.

Using the results from the volumetric and mechanical efficiency testing, the overall efficiency of the unit can be found for each kit tested. The overall efficiencies were found as follows.

$$\text{Overall Efficiency} = \text{Volumetric Efficiency} * \text{Mechanical Efficiency}$$

The volumetric and mechanical efficiency results at each matrix condition were used to find the associated overall efficiency. Figures 14, 15, and 16 show the overall efficiency results.

The overall efficiency results exhibit similar characteristics for all designs. The highest efficiencies occur at the mid-pressure range of 3000 psi while efficiencies drop at both higher and lower pressures. When input speed is considered, the highest efficiencies occur at the mid-speed range of 1500 rpm.

For nearly all the speed-pressure combinations, the highest overall efficiencies are associated with the reduced orifice slipper design. A similar statement can be made for the

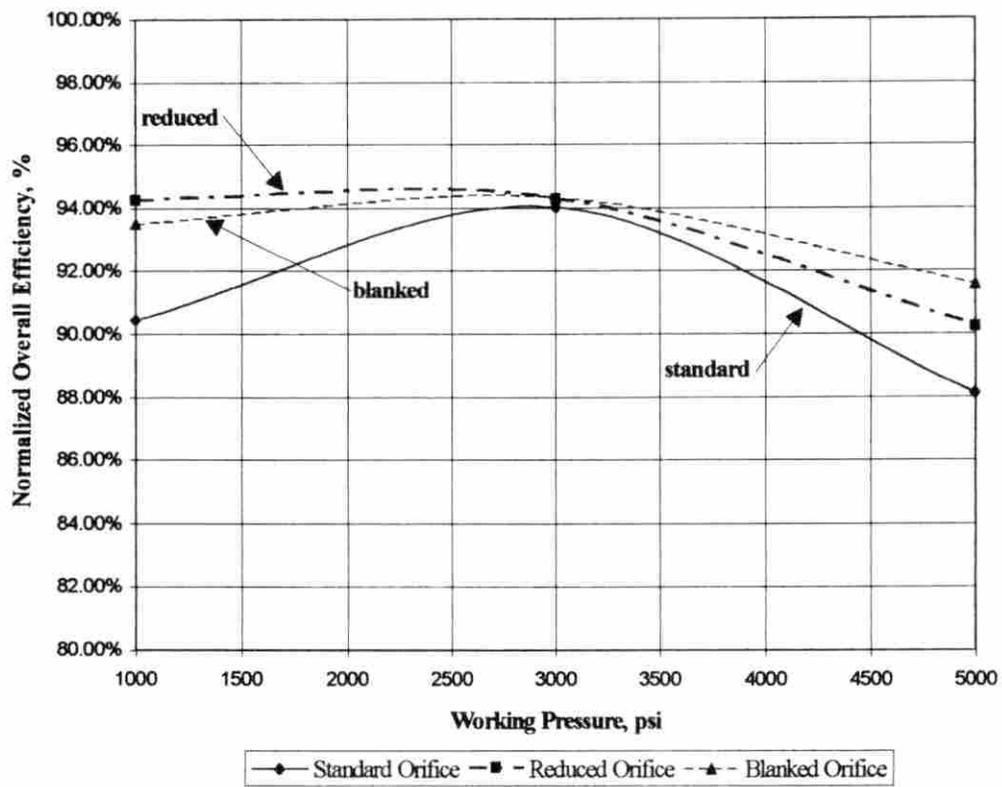


Figure 14: Normalized overall efficiency versus system pressure at 500 rpm.

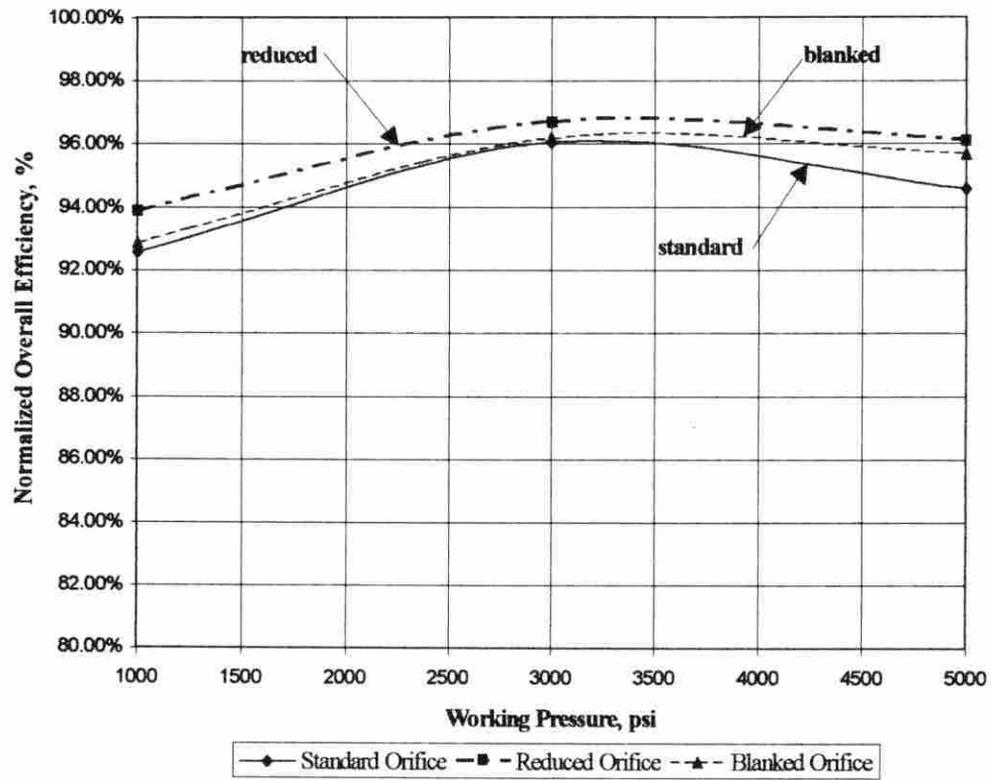


Figure 15: Normalized overall efficiency versus system pressure at 1500 rpm.

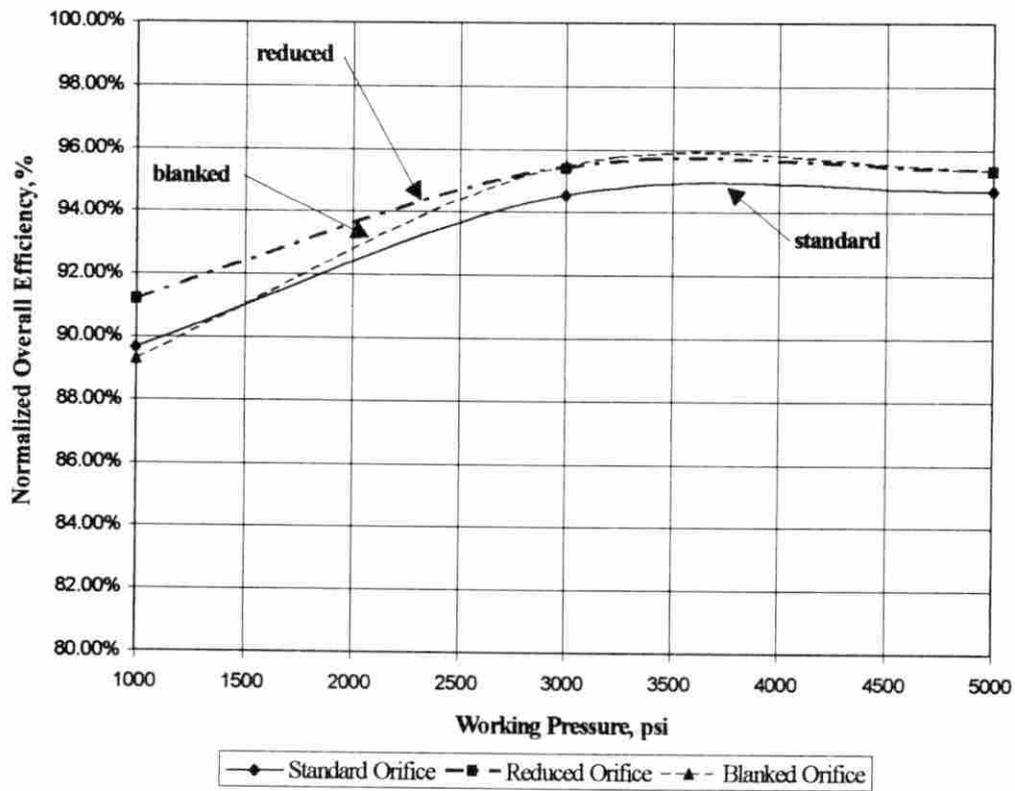


Figure 16: Normalized overall efficiency versus system pressure at 2500 rpm.

lowest overall efficiencies. In this case, except for one combination of speed and pressure (2500 rpm/5000 psi), the standard orifice design exhibited the worst overall efficiency of the three designs. This fact is somewhat amazing since the standard design is being used successfully in the market today. This testing shows that improvements are possible.

The blanked design generally falls between the overall results of the standard and reduced slipper design. Yet, for almost all conditions, the blanked design is an improvement over the standard design. If not for the decidedly poorest mechanical efficiencies, the blanked design may have easily had the best overall results. As will be discussed later in this paper, practical solutions leading to increases in the mechanical efficiency of the blanked design would make this design attractive for actual application in hydrostatic units. The blanked design offers significant advantages not only with respect to efficiency but also with the costs of manufacturing and unit reliability.

The error associated with the testing results, caused by kit-to-kit and run-to-run as well as transducer variability, was evaluated by running three previously untested kits of each design to the one test condition of 1500 rpm and 3000 psi. In this manner, the statistical parameters of standard deviation and confidence limits could be estimated for the efficiency results at this condition. A single matrix condition was chosen simply due to restrictions on available hardware to test. The mid-range condition was chosen with the assumption that the associated error would tend towards the nominal of an overall "condition-to-condition" variability distribution, if indeed the error associated with the test data is not constant for all conditions. The results for the variability of the testing at one data point condition should at least offer an indication of the variability associated with testing at all the remaining conditions. Future investigations could be directed towards verifying this assumption.

Using the results from this phase of testing, the confidence intervals for the volumetric and mechanical efficiency population means can be calculated. In each case a sample size of three was used. The following equation was used for estimating the confidence interval for each efficiency's population mean.

$$\bar{x} - t_{\alpha/2} \left(\frac{s}{\sqrt{n}} \right) < \mu < \bar{x} + t_{\alpha/2} \left(\frac{s}{\sqrt{n}} \right)$$

The *t*-distribution was used since, with such a small sample size, the standard deviation, σ , was not known. In addition, the *t*-distribution is recommended for confidence interval evaluation when the sample size is again small, or when $n < 31$. The confidence level chosen for the confidence limit was chosen to be 90%, or $\alpha = 0.1$.

Table 2: Volumetric Efficiency

	Standard Deviation	Confidence Interval
Standard Orifice	0.10%	$\bar{x} \pm 0.17\%$
Reduced Orifice	0.10%	$\bar{x} \pm 0.17\%$
Blanked Orifice	0.11%	$\bar{x} \pm 0.18\%$

Table 3: Mechanical Efficiency

	Standard Deviation	Confidence Interval
Standard Orifice	0.40%	$\bar{x} \pm 0.67\%$
Reduced Orifice	0.36%	$\bar{x} \pm 0.61\%$
Blanked Orifice	0.19%	$\bar{x} \pm 0.32\%$

The magnitude of the standard deviations and confidence intervals for volumetric efficiency indicate good repeatability associated with the testing procedure applied. However, the mechanical efficiency results indicate an increased level of variability. The added variability is more than likely traceable primarily to the added losses caused by the pistons within the bores and also partially to the change in thrust plate with each kit. If the confidence intervals are compared to the results for the matrix condition of 1500 rpm and 3000 psi, the

difference in volumetric efficiencies would appear to be quite definite due to the separation in confidence intervals while such a statement can not be made for the mechanical efficiencies.

Hardware condition after each run for all designs was also investigated. For each of the standard and reduced orifice kits, the slippers showed only extremely slight indications of polishing probably due to the start-up. The same can be said for the mating thrust plates. No indications of surface-to-surface contact could be seen. This was expected, certainly, for the standard design yet was not totally expected for the reduced orifice design slippers. The reduced orifice design slippers should run with a reduced film thickness and, as such, can be expected to have a greater likelihood of making contact with the thrustplate. This did not appear to be the case, however.

Most surprisingly by far was the condition of the blanked orifice slippers. In all cases, the slippers showed only signs of polishing. Some slippers had no indications of contact at all. The degree of polishing present was decidedly greater than that seen with either the standard orifice or reduced orifice designs suggesting that the blanked slippers did experience more boundary lubrication contact with the mating thrust plates. The polished area in all cases extended in a roughly circular form from the center of the slipper outwards to a radius approximately midway to the outside diameter. The thrust plates also showed signs of light polishing but no evidence whatsoever of any brass transfer. This was not an expected result. At a minimum some brass transfer was expected, and a complete failure of a kit due to gross adhesive wear between slipper and thrust plate would not have been surprising. The implications of this result are quite encouraging.

DISCUSSION OF RESULTS

In a broad sense, the simple model presented to describe the effects on volumetric and mechanical efficiency due to changes in the compensating orifice was correct. The results of this testing bear this out.

Volumetric efficiencies did increase with first a reduction and then a blanking of the orifice in the slipper. As predicted, the greatest improvements were found with the blanked orifice although the reduced orifice configuration performed nearly as well as the blanked. This is a very encouraging result in that nearly all the improvements seen with the blanked orifice design appear to be possible with the less radical approach, compared with contemporary design practice, of simply reducing the compensating orifice size.

The actual increases in unit volumetric efficiency associated with reducing the orifice size were not as great, however, as that suggested by the model and Figure 6. This was a result presumably of the following reasons. First, typical volumetric efficiencies of a hydrostatic unit operating above, say 800-1000 rpm, will lie in the low to mid 90% range. A level of improvement such as that predicted by the model is just not realistic for the speed range chosen for this testing. Yet, for applications that may operate at extremely low input speeds, for example 200 to 500 rpm (transit mixers), the trend suggested by the results at 500 rpm appears to indicate that more significant improvements are possible at lower speeds. The standard design's volumetric efficiency is beginning to deviate from the other designs and shows signs of worsening at a quicker rate than the reduced or blanked designs. Future testing could be directed towards investigating this trend in more detail.

Secondly, the model is oversimplified by not considering additional forces, in particular the friction load between piston and bore. The friction between piston and bore acts to increase the loading on the slipper and lead consequently to a further reduction in the film thickness thus leading to less leakage.

Moreover, as several researchers have discovered, centrifugal forces acting on the slipper cause it to tilt relative to the thrust plate. This tilting leads to deviations between prediction and result since the model assumes a constant film thickness for all conditions.

Finally, under load, the slipper will tend to deflect into a slightly convex shape. An analysis by Hooke and Kakoullis (1978) showed that a small amount of non-flatness is essential for successful operation of a slipper. As a consequence of this deflection, the balance of the slipper will decrease slightly -- it will become less underclamped -- so that a lesser amount of the piston load will be carried hydrostatically. The effect would most likely be a reduction in the film thickness leading again to less leakage.

In summary, the improvements in volumetric efficiency for the hydrostatic unit were not as great as that predicted by the model. The reason for this is most likely due to factors that cause the slipper to run with less film thickness than that calculated in the model.

The repeatability in the volumetric efficiency data was good. The confidence intervals for each design were very comparable and little overlap existed between confidence intervals applied to the test condition.

The change in mechanical losses loosely followed the predictions of the model. Except for the 500 rpm/3000 psi test condition, the mechanical efficiency results for all designs were consistent from condition to condition. A relatively tight grouping of data exists between the designs with clear trends developing between conditions. Per the model, though, increases in torque losses were expected as the orifice was reduced and then blanked. This was not found to occur at all test conditions. On a positive note, the blanked design did exhibit the worst mechanical efficiency of the three designs at all conditions per the predictions.

The best mechanical efficiency results were seen with the reduced orifice design and not, as was expected, with the standard design. Koc, Hooke, and Li (1992) found that, as the orifice size was increased in an underclamped slipper, the slipper tended to run with more tilt and became destabilized. In light of this finding, it seems reasonable to assume that the tilted, destabilized slipper may operate more in a boundary lubrication condition with asperity contact taking place between the thrust plate and slipper surfaces. Mechanical losses increase.

The reduced orifice slipper, which experiences less tilt and is more stable, runs with a greater degree of fluid film lubrication. The losses will not be as great. The influence of the orifice size in underclamped slippers on stability would appear to have tangible implications with respect to mechanical efficiency in a hydrostatic unit.

The blanked design had the worst mechanical efficiencies. From the amount of polishing seen on the blanked slippers, boundary lubrication dominated the operation for at least some of the running time. However, as was also concluded by Koc, Hooke, and Li (1992), slippers generally ran best when the orifice is blanked. Blanked orifice slippers were most stable and had the most resistance to tilt when compared to orificed slippers. From the consistent pattern of polishing seen on the blanked slippers from this testing, stable running, with little or no slipper tilt, dominated operation.

Due to the heavier polishing seen on the blanked slippers, the mechanical losses were greater than for the other slippers. Yet, no brass transfer to the thrust plate took place. Adhesive wear was not occurring and the slippers appear to still have run with a low amount of friction at the interface to the thrustplate.

Despite the consistent mechanical results obtained throughout the matrix conditions, the repeatability was not as good as that seen with volumetric efficiency. As a result, the confidence intervals were larger. The increased variability is more than likely due to the fact that a total of 21 bearings were to be found in the unit tested -- nine slippers on the thrust plate, the cylinder block, nine pistons in bores, and two shaft bearings. Even though the testing scheme was devised to minimize the run-to-run and kit-to-kit effects of all these bearings on the outcome, the resulting effect was not expected to be zero. Indeed, the magnitude of the confidence intervals was reasonably small considering all the sources of losses and this is reflected in the relative consistency of the mechanical efficiency results.

The overall efficiency results clearly show the reduced orifice design to be better than the standard or the blanked designs. Superior mechanical and impressive volumetric efficiencies combined to result in the best performance overall. Surprisingly, the blanked orifice design had better overall efficiency results than the standard design at most test

conditions. If not for the poorer mechanical efficiencies, the blanked design could have easily been the best overall.

Efforts to improve the mechanical efficiency characteristics of the blanked slipper would be beneficial for many reasons. The performance aspects concerning volumetric efficiencies have been covered already. Manufacturing costs of the slipper would be favorably affected if the compensating orifice were not needed. The drills used are, naturally, small and the associated difficulties of drilling such a small hole without burrs or surface tears are considerable. Plus, the orifice is a significant stress riser in the slipper. The vast majority of slipper failures are due to fatigue cracks originating at the orifice. Reliability improvements would be realized. Efficiency performance is not the only benefit associated with elimination of the orifice.

CONCLUSIONS

The size of the compensating orifice in the slipper of hydrostatic units has an important influence on the volumetric, mechanical, and overall efficiencies. A simple model was derived to predict the changes in efficiencies caused by first reducing and then blanking the orifice. Testing in a hydrostatic unit was performed to measure the effects of the slipper orifice design on efficiency. Comparison of the test results to the model's predictions was then made.

The volumetric efficiency test results did follow the prediction made by the model with regards to design order.

Blanked Design Vol.% > Reduced Design Vol. % > Standard Design Vol.%

With a reduction and then blanking of the orifice, volumetric losses at the slipper were reduced and the efficiency improved.

The improvements in volumetric efficiency were not as great, however, as predicted by the model. The differences between predicted and actual results were due to one or more of the following reasons, all of which tend to reduce the film thickness.

- 1.) not all forces were considered in the loading of the slipper
- 2.) slipper tilt caused by centrifugal forces as well as orifice size
- 3.) deflection of the slipper under load affecting balance

The best improvements in volumetric efficiency were seen to occur at the low speed condition of 500 rpm. Improvements of up to 3.5% were seen at this speed. Such an improvement would be significant for applications which require good efficiency at these speeds.

The results for the mechanical efficiencies of the three designs did not match that predicted by the model. In this case the following the order was seen.

Reduced Orifice Mech.% > Standard Orifice Mech.% > Blanked Orifice Mech.%

The standard orifice efficiency was predicted to be better than the reduced. On a positive note, the blanked design indeed exhibited the poorest mechanical efficiencies as expected.

The standard design did not have the best mechanical efficiencies. The performance of overbalanced slippers is sensitive to the size of the compensating orifice. If the orifice becomes too large, excessive tilt of the slipper will occur and larger film thicknesses will develop. Such a slipper will become unstable. The greater degree of tilt could conceivably lead to contact between the tilted edge of the slipper and the thrust plate. This contact would lead to increased mechanical losses when compared to a slipper that is riding on a full oil film, as was very likely the case with the reduced orifice design slippers.

The blanked design slippers had the worst mechanical efficiencies. Poorer efficiencies were a result of the increased oil shear losses due to the reduced oil film and of operation in a boundary lubrication mode. Yet, the evidence of uniform polishing patterns indicated uniform film thicknesses dominated operation of this design.

The overall efficiency results showed the reduced orifice design to be the best of the three designs. Surprisingly, the standard design exhibited the worst overall efficiencies. Despite the poor mechanical efficiencies of the blanked design, the overall efficiency results were still encouraging. If the mechanical losses of the blanked design could be improved, for instance by using a material with better frictional characteristics, the blanked could feasibly have the best overall results.

The condition of the hardware follow testing was evaluated. None of the slipper designs showed any signs of brass transfer to the thrustplate. This result was good for the reduced design and very positive for the blanked design. Removal of the orifice from the slipper of a hydrostatic unit does not appear to be as extreme of a change as perhaps previously thought.

FUTURE WORK

The three parameters of outside sealing land diameter, mid-diameter, and orifice size control the performance of the slipper. Future work should concentrate on optimizing all three factors in regards to performance and life.

Mathematical modeling of the slipper is difficult, at best. The difficulty arises primarily due to the uncertainty in the forces acting on the slipper. For instance, the conditions giving rise to frictional forces acting between the piston and bore are complicated and difficult to evaluate. The same can be said for the frictional forces acting at the slipper and piston joint. Tilting couples also act on the slipper. Plus, loading of the slipper causes deflections which alter the hydrostatic and hydrodynamic pressures that develop at the slipper surface.

As an alternative, testing could be conducted which would systematically determine the effect of each of the design variables as they are changed while keeping the remaining two constant. In this manner, the interaction of the parameters could be determined.

Future testing with the blanked orifice design should concentrate on evaluating materials with improved frictional characteristics in a boundary lubrication mode. Operation in a boundary lubrication mode led to the mechanical losses being greater for the blanked slipper design than those for the reduced or standard designs.

One possible improvement might come from the use of ceramics in the design of the blanked slipper. Ceramics could be used either to make a complete slipper or possibly be used as a coating over a base material. Another option would be to use a ceramic insert within a blanked pocket similar to the recess which now exists in standard slippers. The insert would act as the load bearing member in a slipper which operates on a hydrodynamic film. The superior friction and wear characteristics of ceramic materials may lead to mechanical efficiencies equal to or even better than the hydrostatic version.

In choosing the proper ceramic material, many factors must be considered. Even though ceramics were originally touted as the next wonder material from a tribological stand

point, these expectations have not as yet fully materialized. Archard (1972) proposed the following model to predict the volume of wear that is generated due to adhesive wear.

$$V = K \frac{LW}{H}$$

In the above equation, V represents the volume of worn material, L is the sliding distance, W is the load, H represents the hardness of the material, and K is a coefficient of wear. As can be seen, the volume of wear is predicted to decrease as the hardness of the material increases. Since many ceramics have high hardnesses, these materials should be well suited for wear applications.

Ceramic materials are known for having high melting temperatures and maintaining their physical properties (hardness and strength) at elevated temperatures. In wear applications, this can be beneficial in consideration of frictional flash temperature effects. Flash temperature is a term used to describe the brief and rapid increase in temperature that occurs as asperities from two surfaces in relative motion come in contact, weld, and then separate again. As the asperities contact and then separate, thermal energy is released. The release in energy causes a rise in temperature of 1000°C or more which can lead to changes in the physical properties of the materials. Therefore, thermal stability is an important material property for wear applications

One material that has shown much potential in wear bench testing and actual field testing is silicon nitride, Si_3N_4 . Silicon nitride is very promising wear material due to its high fracture toughness, low density, high temperature resistance. Research has shown that silicon nitride has a lower friction coefficient than that for steel in lubricated sliding as well as showing better anti-wear performance.

Kano and Tanimoto (1991) have shown that silicon nitride exhibits superior friction and wear performance when used in rocker arm pads running against chilled cast iron cam shafts in automotive engines. Ferro-based sintered powdered metal has typically been used mated to chilled cast iron camshafts. This assembly in automobiles is a major cause of engine wear problems due to the environment of elevated temperatures, sliding and high contact

temperatures. When silicon nitride was substituted in the rocker arm pads, wear was virtually eliminated as can be seen in Figure 17.

Substitution of the powdered metal material with silicon nitride essentially eliminated wear of the rocker pads. From this, it is evident that ceramics, especially Si_3N_4 , hold much promise for use as substitute materials for brass in hydrostatic slippers. The combination of exceptional volumetric efficiencies and improved mechanical efficiencies through the use of ceramics offers an exciting potential improvement in axial piston hydrostatic performance. Future work should be directed towards investigating this possibility in more detail.

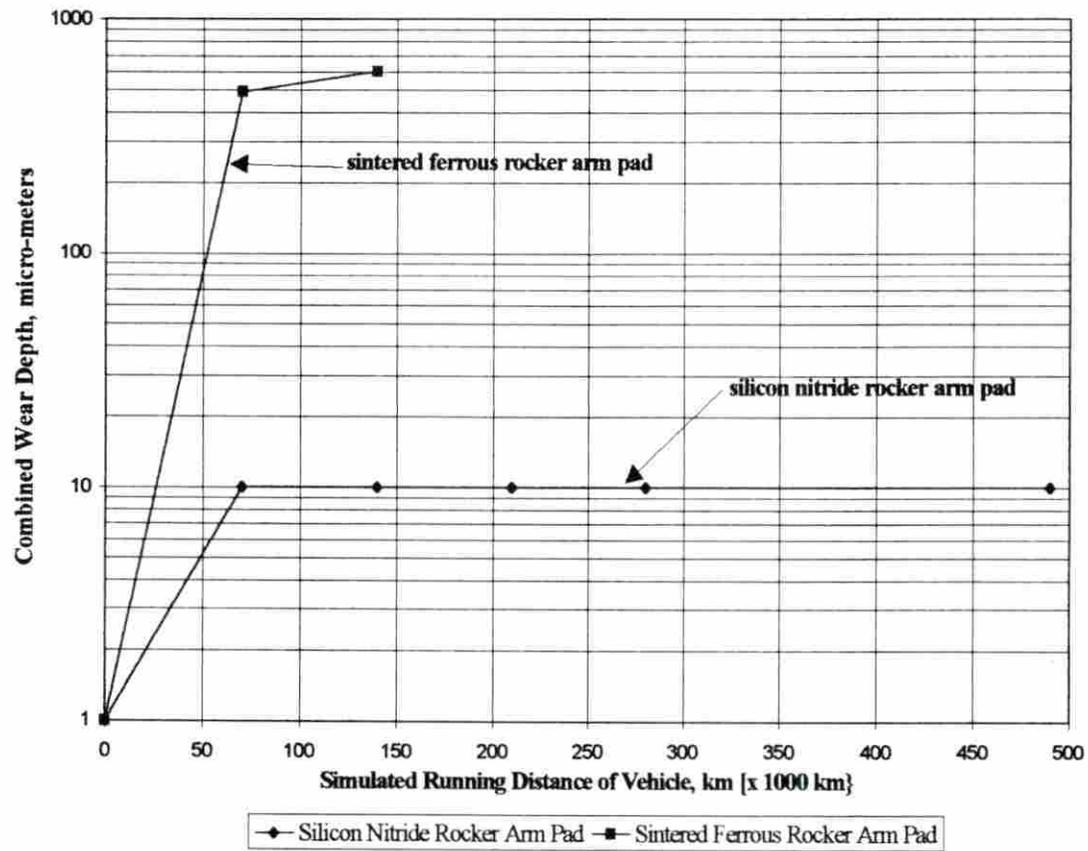


Figure 17: Comparison of wear between metallic and ceramic rocker pads.

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APPENDIX A: PICTURES OF TEST HARDWARE

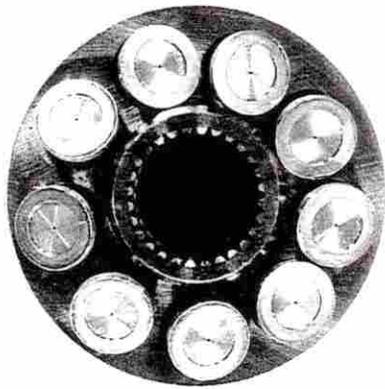


Figure 18: Standard kit hardware.

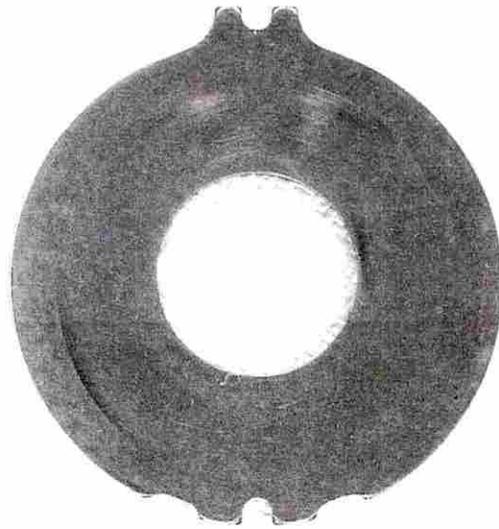


Figure 19: Standard thrustplate.

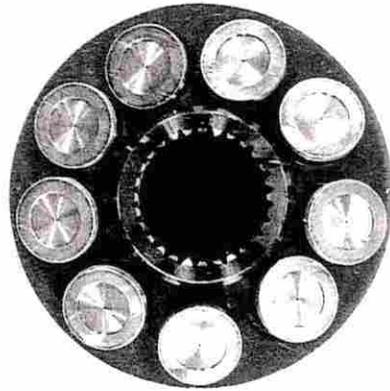


Figure 20: Reduced orifice kit hardware.



Figure 21: Reduced orifice thrustplate.

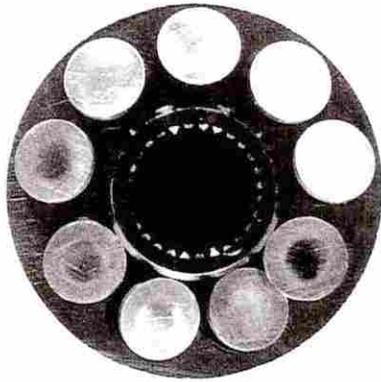


Figure 22: Blanked orifice kit hardware.



Figure 23: Blanked orifice thrustplate.

APPENDIX B: PICTURES OF TEST STAND

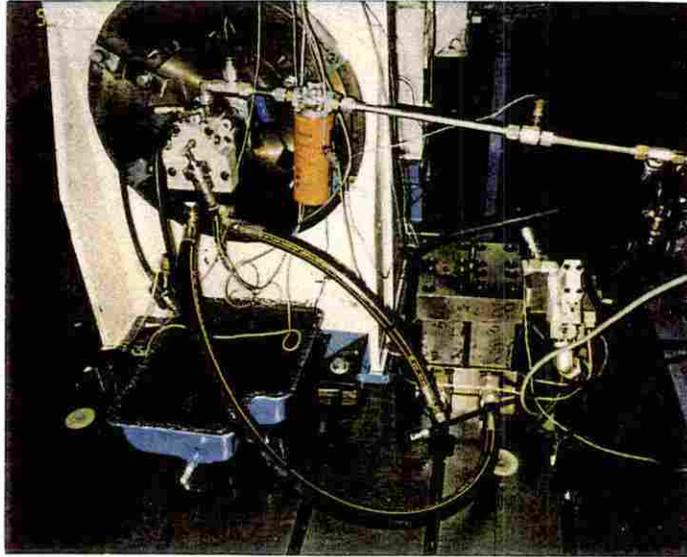


Figure 24: Picture of test stand with hydrostatic unit.

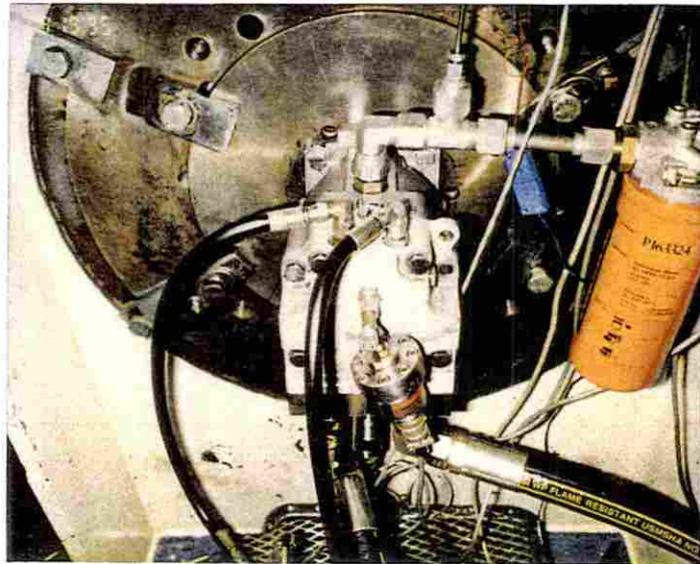


Figure 25: Picture of hydrostatic unit.