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# Hydrostatic Bearing Oil Replacement

Shane 120" Telescope  
Lick Observatory, Mt. Hamilton

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## Reference:

1. Theory and Practice of Lubrication for Engineers, **Fuller**  
John Wiley & Sons, 1984
2. Design of Hydrostatic Bearings, **Rippel**  
Machine Design, Aug-Dec 1963  
(See Jack Osborn's folder "Hydrostatic Bearings")

## Overview

Mobil 95 Flying Horse Telescope Oil has been used in the hydrostatic bearings that support the Shane 120" Telescope, probably for over 30 years. This oil was specially formulated with a very high viscosity index (VI) so as to have low change in viscosity with the temperature extremes experienced on Mt. Hamilton. A large minimum purchase was required, and was probably expected to last forever. At the time of this report, a single 55 gallon barrel remains. In the search for a replacement, an off-the-shelf product would be desired. Mobil has suggested one of their synthetic oils as a replacement – SHC 629. Included herein are recommendations for replacement, and the associated evaluation.

## Summary

The Mobil synthetic oil should work fine as a replacement however the **SHC 627** (rather than 629) better matches the viscosity of the current oil at the operating temperature range of the system. **Amsoil Synthetic Antiwear Series AWK** hydraulic oil is also a possible replacement. Amsoil reports similar viscosity values to the Mobil SHC 627 in the expected temperature range, but with a slightly higher VI. The evaluation that follows was done for the SHC 627. For the reported viscosities of the Amsoil AWK, this oil will perform slightly better than the Mobil Synthetic. It is also less expensive. The Mobil SHC 627 is about \$1100 for 55gal. The Amsoil is \$978 for 55gal. The Amsoil therefore, is probably a better choice. Amsoil ordering information can be found on their website. Mobil SHC oil can be bought from their San Jose supplier (408 292-1041).

The following should also be done:

1. Drained (and retain) the current oil, replace with new oil and replace the filters.
2. Quantify and document the current bearing pad upper and lower gap limit settings, and compared to the values estimated in this report.
3. If possible, the viscosities of samples of the current oil, and the synthetic replacement, should be evaluated at 40F and 75F to check manufacturer's viscosity data.
4. Replace the seals on the Vickers Pumps to fluorocarbon type. This should be done for at least one pump, prior to fluid replacement. (See [Compatibility with Seals.](#))

## Evaluation

### System Information:

#### Pump:

2 Vickers V10 vane # P2P IA20 (in parallel)

(Only one operating at any given time.)

Inlet – 15psi

Exit – 900psi

#### Motor:

Dayton 3N732

3hp

3ph 60hz

Speed – 1735rpm

### Oil Operating Temperature Range

40F to 75F

### Existing Hydraulic Oil:

Mobil 95 Flying Horse Telescope Oil

Viscosity:

@40C=130cSt

@100C=26.9cSt

VI = 240

Pour Pt = -20F

### Replacement Oil Options:

Mobil SHC 627 Synthetic Bearing Oil

Viscosity:

@40C=99.1cSt

@100C=13.9cSt

VI = 148

Pour Pt = -49F

Product Data Sheet:

<https://dalind6.dal.mobil.com/gis/MobilPDS.nsf/82cc319cc4c4b7bb862566660070dae6/ec51fc2f59a079c1852567770071d980?OpenDocument>

Amsoil AWK Synthetic Hydraulic Oil

Viscosity:

@40C=98.8cSt

@100C=16.0 cSt

VI = 174

Pour Pt = -48F

Product Data Sheet:

<http://www.pecuniary.com/synthetics/technical/industrial/awh.pdf>

## Bearing Operation

Assume the Vickers V10 vane pump is constant volume. The pressure increases in the internal pad recess of the bearing, until the telescope lifts off, and fluid begins to flow out through the gap. When the fluid exits the gap the pressure is zero (gauge). Once flow begins, the pressure distribution within the pad remains constant unless the load born by the pad (telescope weight) changes. The fluid power is **pressure-drop x flow**. It is dissipated in the shear force in the fluid, as it moves through the bearing gap mostly, but there is some loss in the plumbing, prior to the pads. If the load increases, the gap will narrow, and both the mean velocity and velocity gradient for a given flow will increase; requiring more power - or more pressure from the pump, since rpm is constant. Similarly, assuming a constant telescope load, the gap will change in response to change in flow. This again would affect both the mean velocity and shear force (velocity gradient). A relatively small change in gap will account for a large change in flow. (The gap will have the same response to change in viscosity of the fluid as it does to flow.)

The relationship of volume flow Q, viscosity  $\mu$ , gap h, recess pressure P, and pad geometry is as follows (ref. 1),

$$Q = (h^3 \mu P / 6 \ln(R/R_o)) \quad \text{equ'n 1}$$

In this case, the telescope load is assumed steady, therefore P constant. Similarly,  $\ln(R/R_o)$  describes the pad geometry, which is fixed. If, as mentioned earlier, the volume flow is constant, then  $(h^3/\mu)$  is constant also and,

$$(h_1^3/\mu_1) = (h_2^3/\mu_2) \quad \text{or,}$$

$$(h_1/h_2) = (\mu_1/\mu_2)^{1/3} \quad \text{equ'n 2}$$

The SHC 629 recommended by Mobil, is the closest match of this family of oils at 40C and 100C, which are the temperatures used to characterizing oil on an ASTM chart. The viscosity values in the following Table 1, were taken from an ASTM viscosity/temperature chart provided by Mobil (Figure 1) at a lower (assumed) temperature range of 40F to 75F. SHC 627 is a better match at the low temperatures as shown below in Table 1. (Note that with the SHC 627 at 75F, the bearing gap will fall below that of the current oil. Minimum gap is addressed later)

deg F	kinematic viscosity cSt			Change in pad gap	
	Mobil 95	SHC 629	SHC 627	SHC 629	SHC 627
40	600	1250	800	+28%	+10%
75	240	310	210	+9%	-4%

Table 1

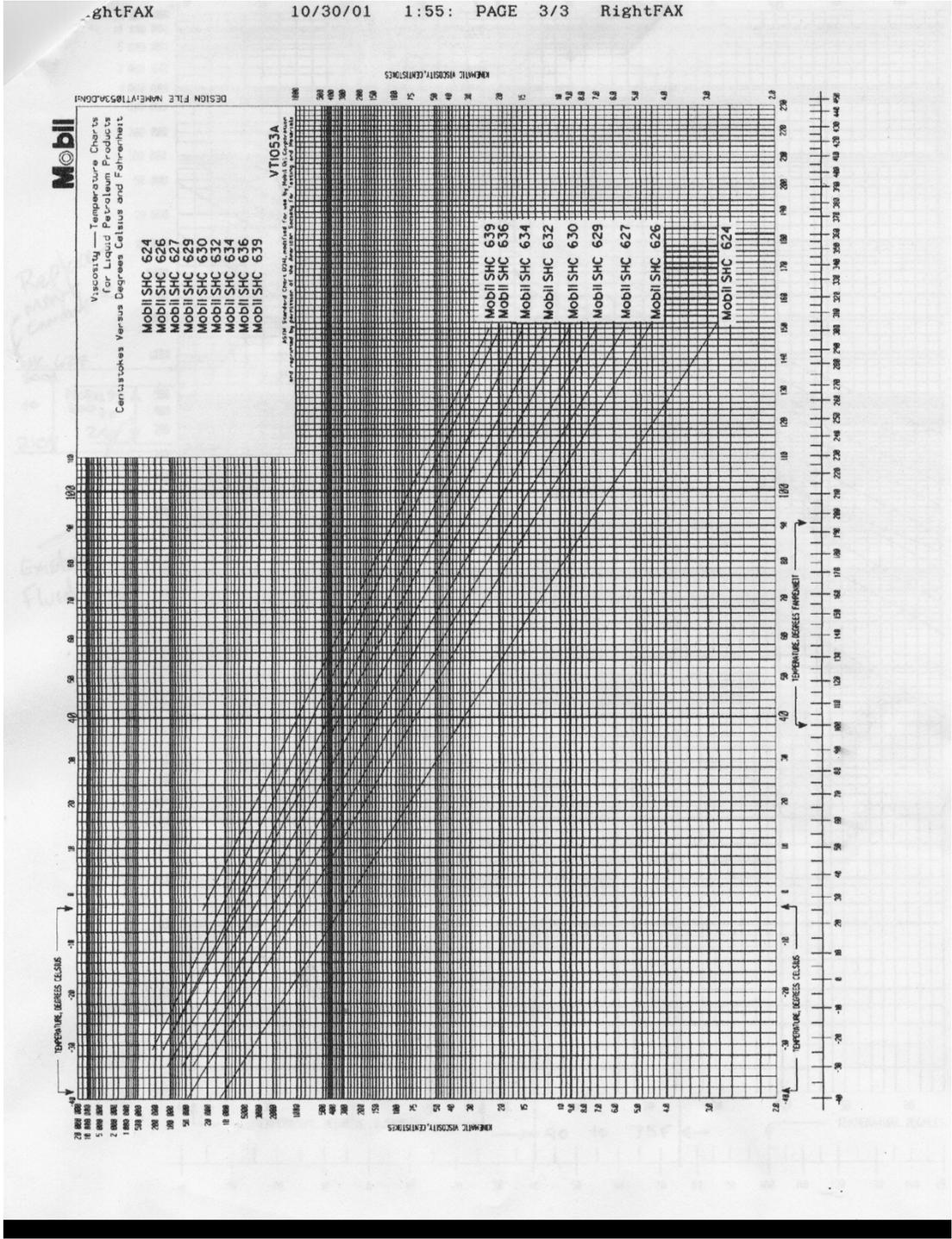


Figure 1

Note: density would also be require with kinematic viscosity to calculate total viscosity. The specific gravities of these fluids are roughly 0.85, and equivalent for purposes of this evaluation and use in equation 2.

The actual balance point for flow and gap, as viscosity changes, will vary slightly because of the response of other system elements. The higher viscosity at the lower temperature will require slightly higher pressure at the pump to generate the necessary pad recess pressure. This will require more input power to the pump, and will create competing effects on the pad gap from flow and viscosity. That is, higher viscosity tends to increase the bearing gap. However, the flow will be reduced slightly by increased pressure at the pump and reducing pump efficiency. This tends to decrease the gap. Finally, the increase in power is delivered directly to the fluid (except for any motor slip). This will raise the temperature and reduce viscosity, which decreases gap. At the same time, pump pressure required goes down, efficiency increases, and therefore flow rises. This serves to increase the gap.

The effects just described, are assumed to be negligible for the following reason. Roughly 95% of the fluid power is dissipated in the pad, which changes for the most part, only with bearing load. Therefore changes in power (i.e. pressure) requirements come primarily from changes in the remaining 5% of the viscous power loss in delivery to the pad, which will be small relative to the total power dissipation.

### Sluicing

Assume a rate of 3deg/sec. The axial and pad surfaces have a relative tangential velocity of,

$$\frac{3}{360} * 3.1415 * 84 = 2.2 \text{ in/sec}$$

The slowest (worst case) mean fluid velocity can be estimated by evaluating the widest gap which occurs at the highest expected viscosity, for the replacement oil.

$$\text{Viscosity @ SG} = .85, \mu = 800 \text{ cSt is } 95\text{E-6 reyns } (\# \text{sec/in}^2) \quad (\text{from ref. 1})$$

The recess pressure of the pads is 850 psi. The slowest radial velocity of the fluid will occur on the bearing pad with the largest diameter, 12in OD, R; and 6.4in recess diameter, R<sub>0</sub>. Equ'n 1 can be used to evaluate the flow for a given gap. Sample calculation for .007 inches is shown below.

$$\begin{aligned} Q &= (0.007\text{in})^3 * 850\#/\text{in}^2 * 3.1415 / (6 * 95\text{E-6}(\# \text{sec/in}^2)) * \ln(12\text{in}/6.4\text{in}) \\ &= 1.01 \text{ in}^3/\text{sec} \text{ (0.264 gpm)} \end{aligned}$$

Table 2 below shows the flow estimates for several gap heights.

Table 2

For $9.5 \times 10^{-5}$ reyns (45F), 850 psi				
Gap	Flow			Velocity
	in <sup>3</sup> /sec	gpm	5 X gpm	Mean u in/sec
0.0070	1.010	0.26	1.32	1.9
0.0080	1.508	0.39	1.97	2.5
0.0090	2.147	0.56	2.80	3.2
0.0091	2.219	0.58	2.90	3.2
0.0100	2.945	0.77	3.84	3.9

The data from Table 2 is plotted in Chart 1 below.

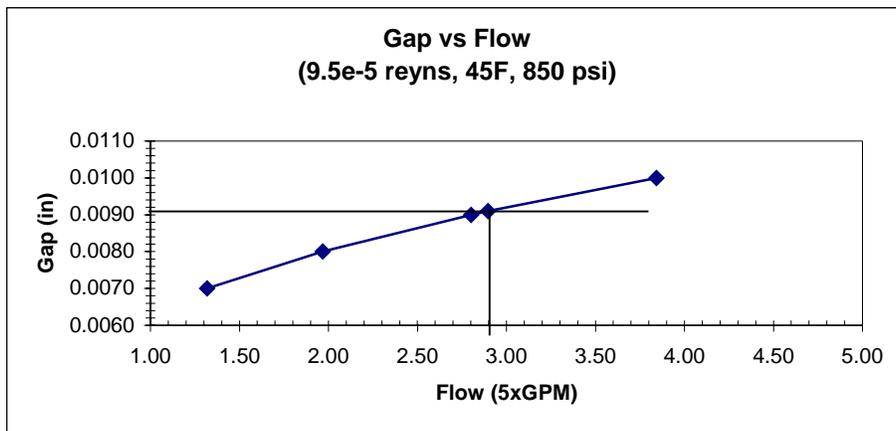


Chart 1

The volume flow from the Vickers pump literature is 2.9 gpm at this pressure. Assuming this is distributed equally between 5 pads, the gap for this viscosity is 0.0091 inches, and the mean fluid velocity is 3.2 in/sec.

This estimated mean velocity is 1.45 times the relative velocity of the bearing surfaces. Maximum velocity at the center of the gap will be even greater, and mean velocity will also increase as viscosity drops when the oil warms up. Therefore there should be sufficient oil available during sluing.

### Minimum Bearing Gap

The maximum gap is a qualitative problem. It affects alignment of the telescope. If too big, it indicates a poorly tuned system and excessive heat will be added to the telescope. The minimum gap on the other hand, prevents metal on metal hardware damage. The minimum gap can be estimated using the calculations in the previous section on Sluing, and the minimum expected absolute viscosity. The pressure will remain at 850 psi. The minimum value for absolute viscosity drops to  $2.8 \times 10^{-5}$  reyns (@75F).

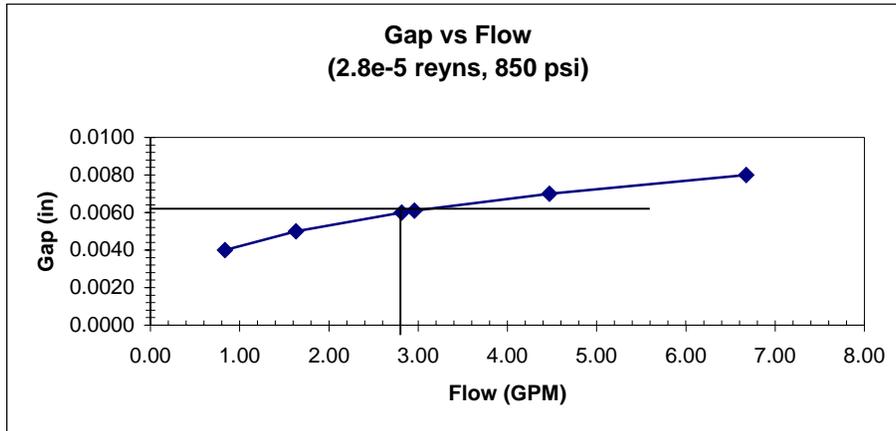


Chart 2

The minimum estimated gap from Chart 2 above is 0.0061 inches. The bearings should operate fine with this gap. (Chart 2 also displays the stability of the gap relative to flow.)

Another way to view the operational envelope of the gap is to rearrange Equation 1, to solve for gap as a function of flow. The recess pressure will again remain constant. The gap can then be plotted against viscosity for several flow rates. In Chart 3 below, this is shown for 20% of the 2.9 gpm total flow rate (1/5, for 5 bearings), and +/- 5% on either side. Viscosity is determined by temperature, which would be the actual variable measured. Therefore the horizontal axis in Chart 3 is oil temperature. Note that a bearing receiving only 15% of the total flow, at a temperature of 100F will still have a gap of 0.004 inches.

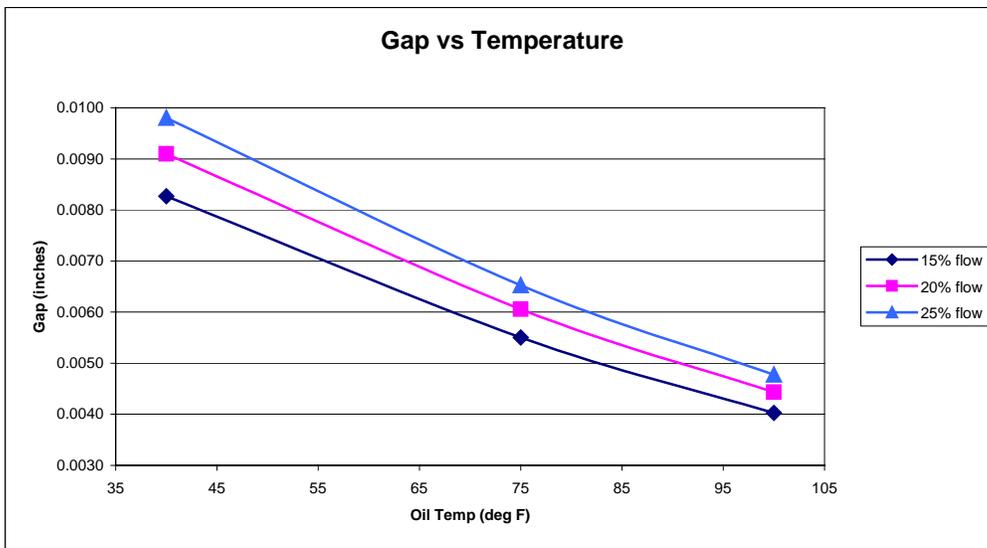
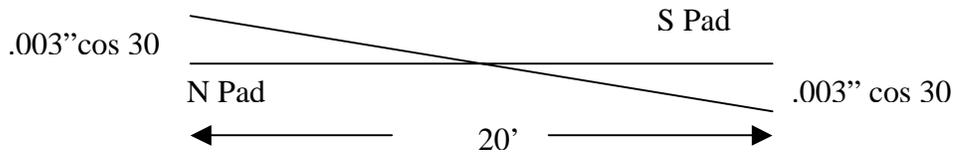


Chart 3

### Alignment Affects

Assume the pad gap varies .003" and the pads are located 30 degrees east or west from the bottom, around the polar axis. The pad separation is 20 ft.



Variability in alignment of the polar axis,

$$\begin{aligned} &= \text{atan} (2 \cdot .003 \cos 30 / 12 / 20) \\ &= 4.5 \text{ arcsec} \end{aligned}$$

The (anecdotal) image stability for the 120" telescope is 10 arc seconds. There is also a positioning system that tracks a guide star. Therefore this misalignment is considered negligible.

### Notes on Bearing Gap and System Tuning

The above evaluation of bearing operation is based on the fact that the current system functions adequately. Extensive measurements to characterize the current system – flow, bearing load, gap, temperature – haven't been done. This comparison hinges on the viscosities values of both the existing and replacement fluids within the anticipated temperature range. From the preceding analysis, the gap is estimated to be between .0061 and .0091 inches for 45F to 75F operating temperature. The recommended gap, for these systems from ref. 2, is not greater than 0.010 inches, and as narrow as practical. (The limit switch settings are not known at this time. However a gap below .004 would warrant more scrutiny.) The SHC 627 appears to be able to produce an acceptable gap, for the assumed temperature span.

The uncertainty of the viscosity values for both existing and replacement oils in absolute terms, is presumed to be high (values in Table 1 were determined by extrapolating manufacturer-supplied viscosity values for both fluids from higher temperatures on a highly non-linear ASTM chart, see Figure 1). If actual viscosity values differ greatly from manufacturer-supplied data, the bearing gap for the replacement oil may not be within existing limit-switch settings. It may therefore be desirable to verify viscosities of existing and replacement oil samples, in advance, thru an oil analysis laboratory.

What then is the contingency, if for some reason the replacement oil creates bearing gaps that are outside existing limits? The following steps would be taken in such a case. First, the changed operating range of the gaps would be evaluated for impact on the telescope operation (see **Alignment Sensitivity** and **Sluing** above). If truly insignificant, the limits can be opened. However this is probably invalid for gaps below the current lower limit. The next thing to do would be to determine if a slight change in viscosity is all that's needed. This should be fairly obvious. Finally some system tuning is possible if necessary. The operating ranges of the pad gaps can be shifted collectively by changing the flow from the high-pressure pump. The high-pressure pump has a bypass valve just down stream. However it is unknown whether there is surplus flow being shunted through this bypass that is available for **increasing** the collective gaps. It is also unknown whether this is an adequate device for metering flow. There are capillary flow restrictors for each pad. These restrictors are shunted with bypass valves. (However, gages on either side of all capillaries show no noticeable pressure drop indicating that the bypass valves are open, or the restrictors do very little.) The operating range of a single pad can be shifted by modifying a capillary restrictor. (Note that there is currently little headroom to **increase** flow to a single pad because of the negligible pressure drop across the capillaries.)

### **Motor Power**

The power required from the motor is the **pressure-drop x flow/pump efficiency**. Plots provided by the pump manufacturer for oil with viscosity of 32cSt show, for 1000psi and 1700rpm, the input power required is 2.2hp and output is 2.8gpm. The efficiency in this case is,

$$2.2\text{hp}/(2.8\text{gpm}\cdot 1000\text{psi}/(.1337\text{ft}^3/\text{gal})\cdot(144\text{in}^2/\text{ft}^2)\cdot\text{min}/60\text{sec}\cdot\text{hp}/(550\text{ft}\#\text{/sec}))$$

$$=75\%$$

The current fluid power required (with no losses) is,

$$(2.8\text{gpm}\cdot 900\text{psi}/.1337\text{ft}^3/\text{g}\cdot(144\text{in}^2/\text{ft}^2)\cdot\text{min}/60\text{sec}\cdot\text{hp}/550(\text{ft}\#\text{/sec})) = 1.57\text{hp}$$

Eaton (Vickers) technical support (John Walesch, 952 937-7456) indicated the pump efficiency is affected most by pressure and is fairly insensitive to viscosity, until vane motion is impeded and/or cavitation occurs. Neither of which is expected for this system. Therefore, if the efficiency of the pump is around 75% for these conditions, then the motor power required will be,

$$1.57\text{hp}/75\% = 2.1\text{hp}$$

The existing motor is a 3hp, 1735rpm motor. This motor should be adequate.

Note: The time required for this motor to come up to speed should be short. If a noticeable lag occurs on startup, the motor is working harder than it's supposed to, and this analysis should be reevaluated. In this case the current to the motor can be measured and compared to the rated value on the motor placard.

### **Compatibility With Current Oil**

Mobil Oil Engineers say that this oil can be mixed with petroleum based oils with no adverse effects. Their recommendation was only to drain and replace. No flushing or cleaning of the system is recommended (for purposes of fluid replacement).

It was noted by Jack Osborn that there is cross contamination of the worm gear oil. The only concern the Mobil engineer had was that mixing significant volumes of different viscosity oils would change the viscosities.

### **Compatibility With Seals**

The serial number from the vane pumps, indicate they have standard nitrile seal material. The Mobil engineer I spoke with said these oils have "fair" compatibility with nitrile seals and a small amount of shrinkage may occur. He thought that after some time, they may need to be replaced. He didn't quantify "some time" nor could he provide any data on this issue. Cost for a Viton seal kit for the Vickers pumps, from Applied Industrial in Watsonville (831 722-0243) is \$50.

### **Heat input to the Telescope**

The intent of this effort was to find a replacement for the existing oil that will not change the current operation. The hydrostatic bearings do, and will deliver heat to the telescope though. This heat input, which can affect image quality or "seeing", can be reduced two ways. The first is to minimize the power input to the fluid. This will minimize the equilibrium temperature of the oil during operation. To accomplish this, both flow and pressure need to be minimized. This is an optimization problem since increasing the viscosity will increase the power necessary for delivery but reduce the required flow. The bulk of excess power, though, can be eliminated by minimizing the pressure drop in the plumbing upstream of the bearing. This is not usually recommended because a drop in pressure from pump to bearing, allows the bearing to respond with a rapid rise in recess pressure when the load increases (and vice-versa). Roughly 45% pressure drop is recommended to produce the "stiffest" bearing. In the case of the 120", the load change is minimal, so this pressure drop can be eliminated. This was either done in the original design, or at some point after. The current pump exit pressure is 900psi and the pad recess pressure is roughly 95% of that.

The other option to reduce the heat input from the bearing system would be to use an oil cooler, and exhaust the heat outside the dome. This is not addressed in this report. However, it may be included in a pending study, which will evaluate the feasibility of mirror cooling, to improve seeing, for the 120" Telescope.