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Minimum Viscosity for Bearing Reliability in Rotary Compressors

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ABSTRACT

To ensure bearing reliability, it is traditional for compressor manufacturers to specify a minimum oil sump temperature difference above the saturated discharge temperature in a high-side compressor and suction temperature in a low-side compressor. This minimum oil sump temperature specification is, in essence, a surrogate measure of the minimum viscosity the compressor manufacturer requires to ensure adequate lubricant film thickness under system flooding conditions. The minimum viscosity required to maintain an adequate lubricant film thickness in the bearings depends on the bearing geometry, the load and the speed. Because the load, and to a much lesser extent, the speed vary from point to point over the compressor operating envelope, so will the minimum viscosity required to maintain adequate film thickness. This paper describes a method for specifying the minimum safe viscosity over the entire operating envelope of high-side rotary compressors based on film thickness calculations.

NOMENCLATURE

c	*Radial clearance*	P_θ	*Pressure at compression angle* θ
D	*Bearing diameter*	R_c	*Cylinder radius*
F_b	*Bearing force*	R_r	*Roller radius*
h	*Cylinder height*	S	*Sommerfeld number*
k	*Polytropic compression exponent*	T	*Temperature*
N	*Shaft frequency, Hz*	V_θ	*Compression volume at angle* θ
μ	*Absolute viscosity*	$V_{\theta 0}$	*Compression volume at* $\theta = 0$
P_b	*Bearing pressure*	ε	*See Figure 3*
P_{lmix}	*Pressure of lubricant mixture*	θ	*Crank position after TDC*
P_{gref}	*Pressure of refrigerant gas*	ψ	*See Figures 2 and 3*
P_d	*Discharge pressure*	χ	*Mass fraction*
P_s	*Suction Pressure*		

INTRODUCTION

Compressor development involves reliability testing at various points within the operating envelope. It is typical that maximum load flood-back condition is part of the qualification to represent the maximum bearing load and minimum oil viscosity that will result in the minimum bearing film thickness of any continuous qualification tests. This condition is typical of systems with fixed expansion devices and are subjected to high charge levels, low air flow and high ambient condition. This becomes an experienced based point in the operating envelope, where the bearing life is known based on worst case film thickness condition from qualification. From this standpoint, where the bearing load and oil viscosity can be determined, the bearing load and lubricant mixture properties will be described in a form to specify the minimum viscosity and minimum oil temperature at other conditions though the operating envelope. For a bearing of a given length-to diameter ratio (L/D), the minimum film thickness may be computed as a function of the Sommerfeld number,

$$S = (D/c)^2.N.\mu/P. \qquad (1)$$

This includes all the variables from the designer's standpoint and is a dimensionless number. c, D, and N are constant in a given fixed-speed design and the active system variables are μ and P. The lubricant viscosity, μ, is that inside the bearing gap. For the purposes of this paper, however, we will employ the lubricant viscosity in the oil sump as a surrogate. This in turn is determined based on the temperature and pressure of the refrigerant over the oil-refrigerant mixture in the compressor sump, assuming mixture equilibrium state conditions. This paper describes s method for establishing minimum acceptable sump viscosity and temperature at all points within the operating envelope of a rotary compressor that successfully meets bearing reliability criteria at maximum load.

ROTARY COMPRESSOR BEARING LOADS

Dynamic Bearing Load

The cross section of the compressor is shown in Figure 1. The rotary compressor is a high side compressor with the lubricant subjected to discharge conditions. The bearings are hydrodynamic journal bearings on the upper and lower sides and eccentric bearings. The magnitude and direction of the dynamic load P_θ are shown in Figure 2. Figure 3 shows the pertinent geometric details. P_θ may be determined by compressor instrumentation or by computing P_θ based on gas properties and compressor geometry [1] as given by the following equations:

$$P_\theta = P_s(V_{\theta 0} / V_\theta)^k \text{ and} \tag{2}$$

$$F_b = (P_\theta - P_s).h.(2R_c \sin(\psi/2)) \text{ until } P_\theta = P_d, \tag{3a}$$

$$F_b = (P_d - P_s).h.(2R_c \sin(\psi/2)) \text{ until } TDC. \tag{3b}$$

V_θ is purely geometry dependent and is given by:

$$V_\theta = h/2\{R_c^2(2\pi-\theta) - R_r^2(2\pi-\psi) + \varepsilon\cos\theta(R_r^2 - \varepsilon^2\sin^2\theta + \theta)^{1/2}. \tag{4}$$

The bearing film thickness requires numerical solution to the Reynolds equation where P_θ is the dynamic load input [2]. The minimum film thickness can be computed as a function of S to allow other similar bearing conditions to be determined. This was done for a typical rotary compressor and is shown in Figure 4.

BEARING LOAD AND MIXTURE VISCOSITY

Viscosity Requirement

In order to prevent bearing wear, a minimum hydrodynamic film thickness larger than the RMS value of the composite surface roughness ($\Lambda>1$)[1] must be maintained under all sustained operating conditions [3]. The minimum bearing hydrodynamic film thickness can be estimated for any given sump temperature, suction pressure and discharge pressure from the bearing load (a function of suction and discharge pressure) and the viscosity of the lubricant/refrigerant mixture in the sump (a function of discharge pressure and sump temperature.) If P_b is a function of P_d and P_s, then the Sommerfeld number will yield μ_{min} as a function of P_d and P_s.

Given the mixture properties, the viscosity μ is expressed as a function of P_d, T in a high side compressor where $P_{lmix} = P_d$. The dynamically loaded bearing film thickness can be represented in terms of the integrated average load on the bearing as shown through the operating envelope in Figure 5. A more rigorous analysis has also been carried out with the dynamic film squeezing effect taken into account, but the use of the average load was deemed adequate for the purposes of the present paper Assuming dynamic similarity throughout the operating map of a given rotary compressor, the Sommerfeld number could be employed to provide a reasonable estimate of $\mu_{min}(P_d, P_s)$.

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[1] Adsorbed anti-wear films generated by the reaction of the lubricant (and its additives, if any) and the sliding surfaces could offer wear protection even when $\Lambda<1$.

Mixture Properties and Temperature Effect

Lubricant mixture viscosity is available in terms of any two state properties of the mixture from lubricant manufacturers and other organizations [4]. For example, the temperature and pressure of the compressor sump can be measured to determine the viscosity. The discharge pressure is very close to the sump pressure and the oil temperature can be inferred from measurement of the compressor shell base temperature. The mixture properties may be obtained from a Daniel diagram as shown in Figure 7. It can be seen that as the lubricant/refrigerant mixture temperature approaches the saturation temperature for any given isobar, the viscosity drops precipitously to values approaching those of liquid refrigerant, which is typically several orders of magnitude smaller than that of the pure lubricant.

To bring this diagram into perspective for a vapor compressor, by specifying a minimum temperature at a particular discharge pressure, a minimum viscosity will be maintained. The discharge pressure and lubricant viscosity are the common variables that couple the load and lubricant mixture in the compressor. The bearing load is translated into a viscosity requirements through the Sommerfeld number,

$$P_b(P_d, P_s) \Rightarrow S \Rightarrow \mu_{min}(P_d, P_s). \qquad (6)$$

The required viscosity at the prevailing refrigerant conditions can only depend on the temperature of the sump. For a given compressor, the minimum allowable sump temperature, T_{min} is, therefore, a function of P_d *and* P_s only. The minimum viscosity that must be maintained at any condition through the operating envelope is shown for a typical rotary compressor in Figure 8.

CONCLUDING REMARKS

From well defined and understood high load conditions, the safe viscosity can be mapped as a function of discharge and suction conditions. From this required viscosity, the mixture properties will allow a minimum sump temperature to be specified. This allows the compressor supplier to provide the potential customer with the floodback guidelines to assure compressor reliability within the complete operating envelope. The complete chart should be taken in its entirety, not only at a single point. That is, under no condition within the complete operating envelope of the system should the compressor be continuously operated with sump viscosities lower than those indicated in the chart.

Special attention must be paid to overcharged systems with orifice- or capillary-type expansion devices, because these systems may satisfy the minimum viscosity requirements if tested at light loads but fail them at higher loads. Therefore, a continuous floodback test performed on such systems at light loads with clean coils and unobstructed air flow may yield satisfactory results, leading to false security, when the same system tested at higher loads may fail the guidelines (high condensing pressures resulting from high outdoor temperatures, dirty coils or obstructed air flow).

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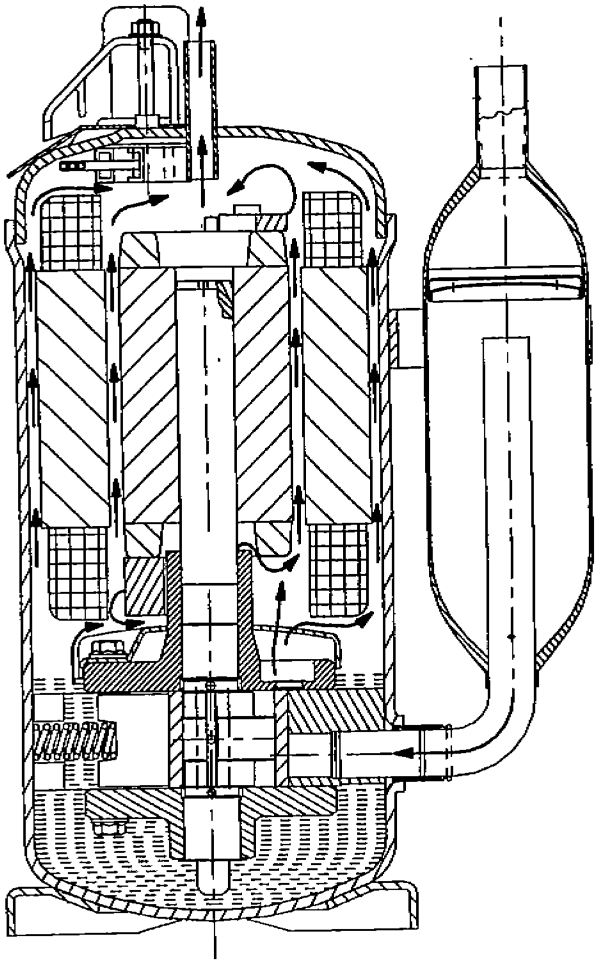

Figure 1

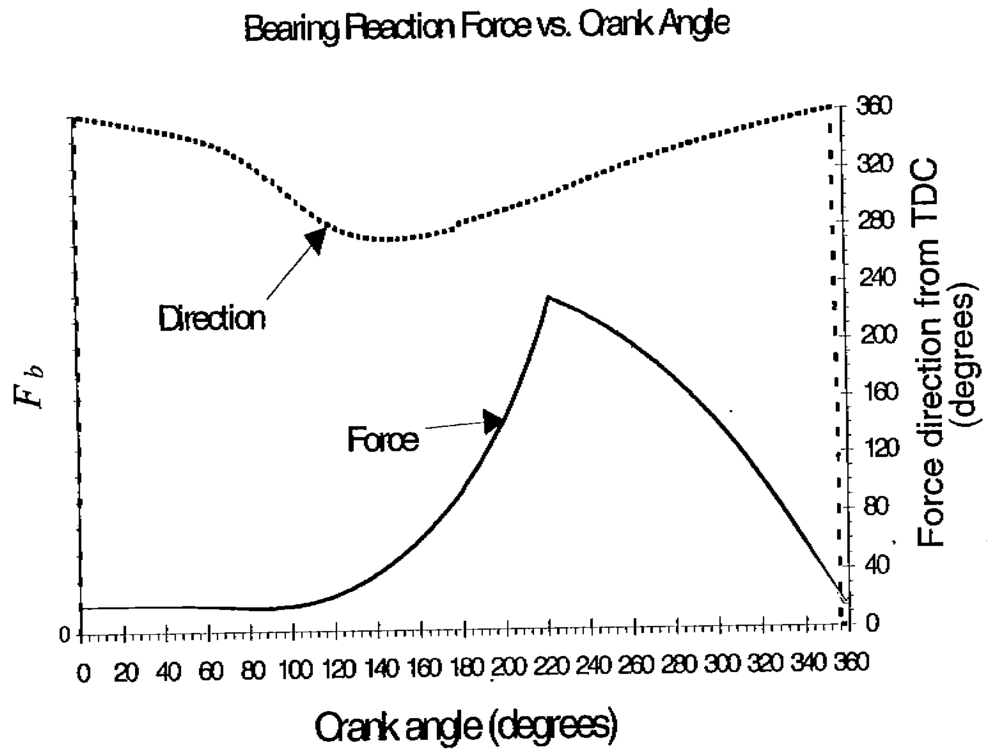

Figure 2

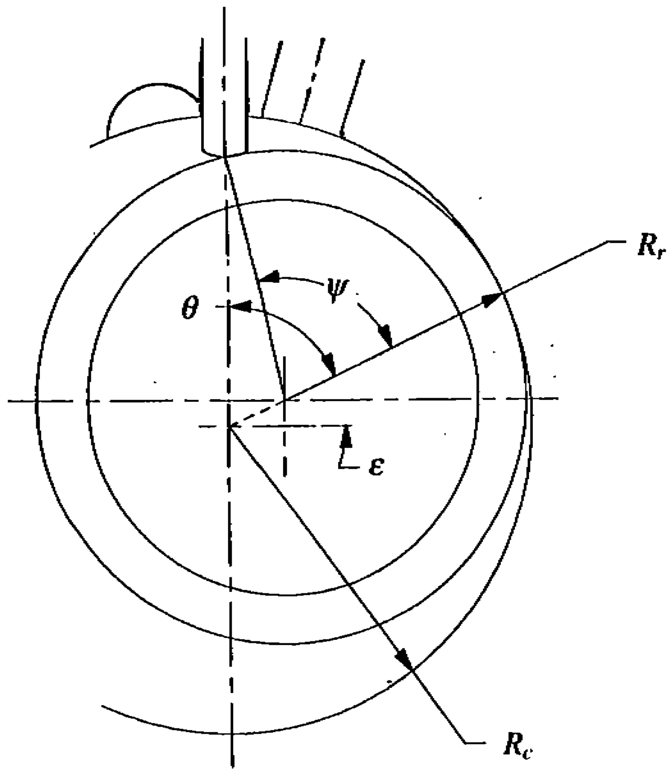

Figure 3

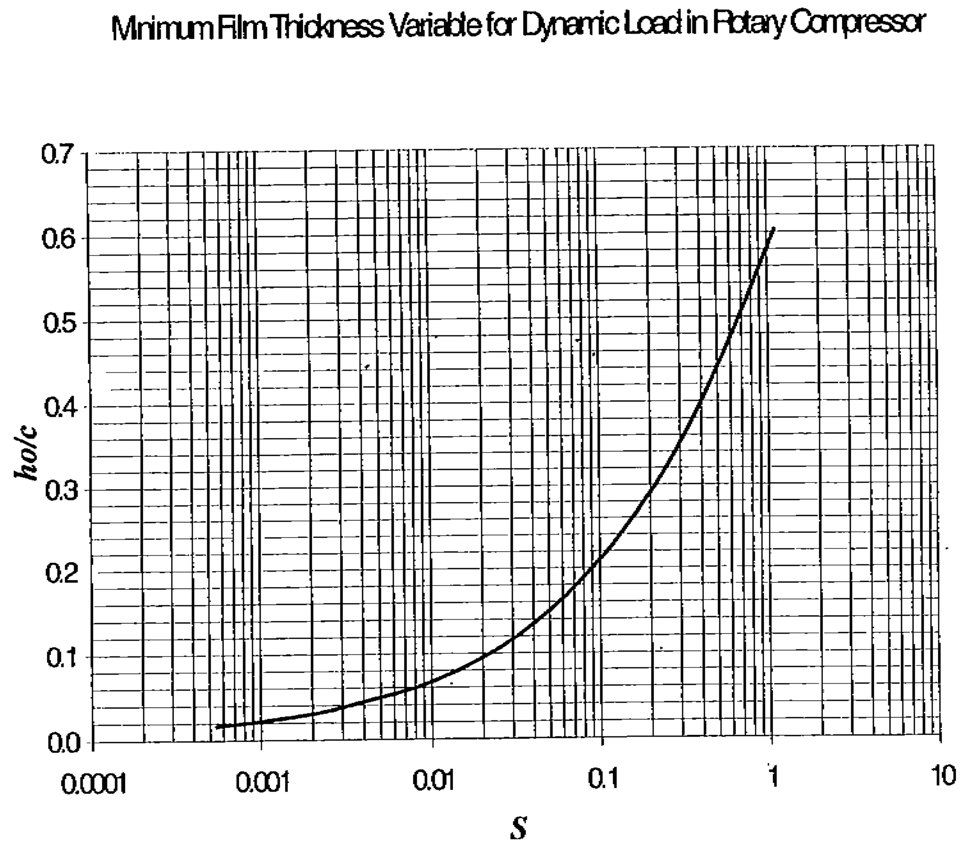

Figure 4

Figure 5

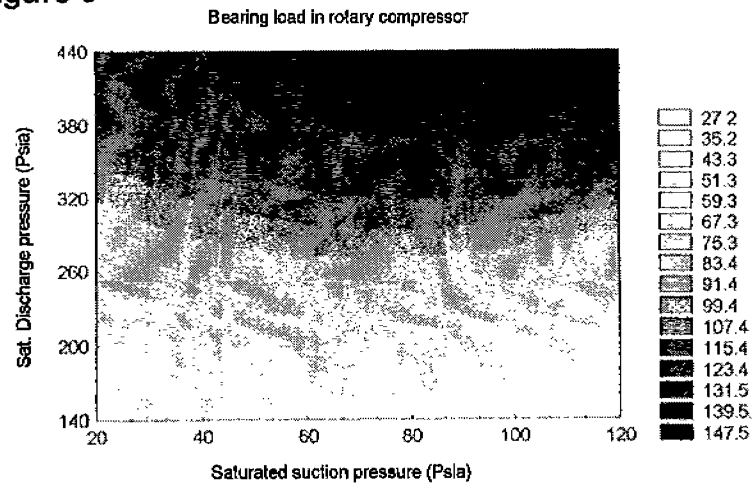

Figure 6

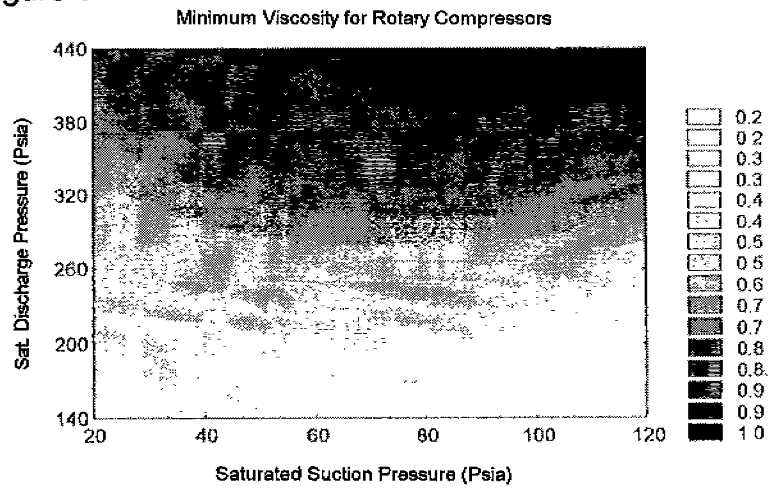

Figure 7

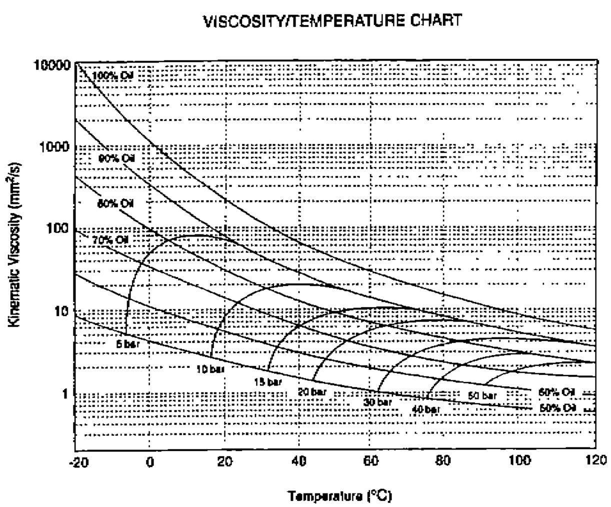

Figure 8