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BREAK-IN BEHAVIOR OF SCROLL COMPRESSORS

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ABSTRACT

This paper presents an analytical model that captures the essential physics of the break-in phenomenon in scroll compressors. The model describes the time dependence of the wear-in of mating surface asperities in the presence of a lubricant, and relates the resulting time-dependent reduction in friction coefficient and leakage area to compressor EER. A discussion of the factors affecting break-in period is presented, along with a comparison of break-in behavior of lower- and higher-capacity compressors of the same family. It is shown that higher-capacity compressors will approach their mature EER more rapidly than their lower-capacity counterparts.

INTRODUCTION

Manufacturing asperities and irregularities in the rubbing surfaces of the orbiting and fixed scrolls lead to performance degradation because of two loss mechanisms:

1) Irregular surfaces are not likely to be mutually conforming over their entire operating contact patch. Gaps are likely to exist between the surfaces with leakage of gas from high to low pressure pockets.

2) Asperities with boundary or no lubrication result in increased mechanical friction between the orbiting and fixed scrolls and between the orbiting scroll and crankcase/seals.

As the mating surfaces rub against each other in the presence of a lubricant, the mechanical friction and the heat generated by it result in wear, which, if controlled, could lead to a beneficial reduction in surface irregularities (roughness) with time. This is the wear-in or break-in process, which can be described as:

Asperities → Friction → Wear-in → Smaller Asperities → Less Friction & Leakage.

Figure 1 shows the qualitative effect of break-in on surface asperity height distribution. The end result of this process is a gradual improvement with time in the compressor rated EER, tending asymptotically toward the rated EER of a mature, fully broken-in compressor. This paper presents a simple model that describes scroll compressor break-in behavior. It is assumed that all break-in is carried out at the same operating load, e.g., maximum load, and that all EER values are estimated at a standard rating condition, e.g., at ARI conditions.

CONTACT SURFACE CHARACTERISTICS

Surface asperities are usually randomly distributed protrusions of variable shapes and heights. Several models exist for describing surface asperities height and area distribution for the purpose of analyzing the tribological behavior of mating surfaces [1, 2, 3]. For the present simple analysis, we will assume that surface asperities can be characterized by a single global parameter, \( \sigma \), given by

\[
\sigma = \left( \frac{\int \zeta^2 \, dA}{A} \right)^{1/2}.
\]  

In this definition, \( \zeta \) is the surface deviation in the direction of interest over the area patch \( dA \) and \( A \) is the total surface area. The term roughness will be employed here for \( \sigma \), although it does not represent any of the standard measures of surface roughness (\( R_a, R_q \), etc.).
For mating surfaces $o$ (orbiting) and $f$ (fixed), a composite measure of their "roughness" is

$$\delta = (\sigma_o^2 + \sigma_f^2)^{1/2}. \quad (2)$$

The true contact area (bearing area) of the two surfaces will increase during break-in from a small fraction of the total contact patch in virgin surfaces with high asperities to almost the entire contact patch for well worn-in surfaces. The variation of the true contact area fraction $\chi$ with $\delta$ has two important implications for compressor performance:

1) a low $\chi$ implies that much of the rubbing surfaces is not in true contact and that relatively large gaps exist between the surfaces with higher leakage (tip and flank leakage in scroll compressors), and

2) a low $\chi$, coupled with high $\delta$ implies that a significant portion of the surface is outside any hydrodynamic or elastohydrodynamic (EHD) lubricant film that may develop under the prevailing loads and rubbing speeds, which leads to higher friction and wear.

**LEAKAGE AND FRICTIONAL LOSSES**

The tip leakage power loss in the compressor pump varies with $\delta$ according to,

$$w_l = w_l^{\infty}(\delta/\delta^{\infty})^m, \quad (3)$$

in which the superscript $\infty$ refers to conditions after full break-in and $m$ is an exponent that ranges from 3.0 for laminar flow to ~1.7 for turbulent flow. In most cases of interest, the flow in the small tip clearance of a scroll compressor is more likely to be laminar. With $W_g^{\infty}$ representing the gas pumping power in the absence of tip leakage, this power loss can be expressed in terms of an internal efficiency,

$$\eta_i = \eta_i^{\infty}(W_g^{\infty} + w_l^{\infty})(W_g^{\infty} + w_l),$$

or

$$\eta_i \equiv \eta_i^{\infty}(1 + \alpha_i \eta_i^{\infty}[(\delta/\delta^{\infty})^m - 1])^{-1}, \quad (4)$$

in which

$$\eta_i = \eta_i^{\infty}(1 + \alpha_i \eta_i^{\infty}[(\delta/\delta^{\infty})^m - 1])^{-1}, \quad (5)$$

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in which \( \alpha_t = \frac{w_t}{\omega^0/W_g^0} = (1 - \eta_t^e/\eta_t^e) \). The presence of \( W_g^0 \) in the denominator of the subtractive term in Eq (5) indicates that the internal efficiency of a low capacity compressor (low \( W_g^0 \)) will be more adversely affected by higher tip leakage (higher \( \delta \)) than a larger capacity compressor of the same family.

The coefficient of friction in boundary, partial- and full-film EHD lubrication is usually given as a function of the relative EHD lubricant film thickness \( \lambda = h/\delta \) \([1, 2]\). For the purpose of this analysis, the coefficient of friction is reasonably represented by,

\[
\mu = \mu_0 f(\lambda) \equiv \mu_0 [\alpha + (1 - \alpha) \exp(-\beta \lambda) + \gamma \lambda^2],
\]

where \( \mu_0 \) is the coefficient of dry sliding friction (\( \mu_0 \approx 0.15 \) for cast iron on cast iron) \([3]\), and \( \alpha, \beta \) and \( \gamma \) are curve fitting constants. It should be pointed out, however, that the increase in the coefficient of friction beyond \( \lambda = 3 \) in Ref \([1]\) is entirely due to higher \( h \) (not lower \( \delta \)), as would be expected from full-film lubrication theory \([1, 2, 3]\). It follows that \( \mu \) should remain constant or decline for \( \lambda > 3 \), if the increase in \( \lambda \) beyond 3 is due to a reduction in surface roughness by wear or polishing. The variation of the mechanical losses in the compressor with \( \lambda \) can now be expressed as,

\[
w_m = w_m^\infty + \omega R_o F \mu_0 (1 - \alpha) [\exp(-\beta \lambda) - \exp(-\beta \lambda^\infty)],
\]

where \( \omega \) is the angular speed, \( R_o \) is the orbit radius and \( F \) is the surface normal load. The mechanical efficiency of the compressor can be expressed in a manner similar to Eq (4), which, after algebraic manipulation, leads to

\[
\eta_m = \eta_m^\infty \eta_t^e \{1 + \eta_m^\infty \eta_t^e (\alpha_t [(\lambda^\infty/\lambda)^m - 1] + \alpha_t [f(\lambda) - f(\lambda^\infty)])\}^{-1},
\]

with \( \alpha_t = \frac{V F}{W_g^0} \) and the rubbing velocity, \( V = \omega R_o \). Again, the presence of \( W_g^0 \) in the denominator of the subtractive term in Eq (8) indicates that the mechanical efficiency of a low capacity compressor will be more adversely affected by higher friction than a larger capacity compressor of the same family.

**TIME DEPENDENCE OF SURFACE ROUGHNESS**

The attenuation of surface irregularities by friction and wear in the break-in phase is a time-dependent phenomenon that could be simulated as a process of material removal in which the wear rate (rate of reduction in \( \delta \)) decreases with an increase in the relative lubricant film thickness \( \lambda = h/\delta \). For values of \( \lambda < 1 \), the lubricant film is too thin to protect the surfaces from destructive wear (scuffing). It has been shown from lubricated wear theory \([1, 2]\) that full film EHD lubrication is established when the composite roughness of the rubbing surfaces is sufficiently smaller than the film thickness, typically \( \delta < h/3 \), at which point it is expected that the wear rate will be negligible. Once this point is reached, the height of asperities will remain at its minimum value, \( \delta_m \), and will change only if the lubricant film thickness falls below the specified minimum value as a result of changes in lubricant viscosity or load conditions. Introducing the modified relative film thickness, \( \xi = h/(\delta - \delta_m) \), and combining the above assumption with the wear model of Holm and Archard \([1]\), we obtain,

\[
\chi(\xi) \cdot d\delta/dt = -(KPV/H) \cdot g(\xi).
\]

The left-hand-side of Eq (9) is the volumetric wear rate \( \chi(\xi) \) is the true area fraction presented earlier). The constant \( K \) is the dimensionless wear coefficient \([1]\), \( P \) is the apparent surface unit load \( (P = F/A) \), \( V \) is the rubbing velocity, and \( H \) is the indentation hardness of the wearing material (same units as \( P \)). The term \( (KPV/H) \) on the right-hand-side of Eq (9) is the volumetric wear rate under dry conditions. Without loss of generality, \( \chi(\xi) \) could be incorporated into \( g(\xi) \) and \( (KPV/H) \) could be interpreted as the linear wear rate under dry conditions \([4]\). With these stipulations, \( g(\xi) \) should approach unity as \( \xi \) approaches \( 0 \). On the other hand, as \( \xi \) approaches \( \infty \), i.e., as \( \delta \) approaches \( \delta_m < h/3 \), aspery summit contacts will cease, full EHD film lubrication will be established and the wear rate will diminish to zero, that is, \( g(\xi \to \infty) = 0 \). A simple functional relationship is \( g(\xi) = \exp(-\xi^p) \), with \( p \) representing the ratio of the lubricant film thickness to a roughness scale factor \( (\delta^e - \delta^m) \). Therefore, Eq (9) can be modified to read
\[ \frac{d\xi}{dt} = \xi^2 \exp(-\xi/\Xi) \]

in which the nondimensional time \( \tau = t/t^* \), and \( t^* = hH/KPV \). This implies that the break-in time will be longer for well lubricated (higher \( h \)), harder (higher \( H \)) surfaces rubbing at lower \( PV \). Integrating Eq (10) by parts, we obtain an implicit equation for the normalized surface roughness,

\[ \tau = [\text{Ei}(\xi/\Xi) - \text{Ei}(\xi_0/\Xi)]/\Xi - \{\exp(\xi/\Xi) - \exp(\xi_0/\Xi)\} \].

The exponential integral, \( \text{Ei} \), is a standard function whose values are tabulated in Ref [5]. The integration constant has been determined from the initial condition: \( \xi = \xi_0 = h/\delta_0 = \delta_0 - \delta_m \) when \( t = 0 \). Introducing the relative composite surface roughness, \( \Delta = \Xi/\xi = (\delta - \delta^*)/(\delta^* - \delta^c) \), we get,

\[ [\text{Ei}(1/\Delta) - \text{Ei}(1/\Delta_0)] - [\Delta \exp(1/\Delta) - \Delta_0 \exp(1/\Delta_0)] = \Xi \tau \].

The solution is presented graphically in Fig 2, which displays the relationship between \( \Xi \tau \) and the relative roughness, \( \Delta \) for \( \Xi = 0.37 \) and various values of the initial roughness, \( \langle \delta_0/h \rangle \). These equations allow us to compute the variation of compressor EER with time during the break-in period.

**VARIATION OF EER WITH BREAK-IN TIME**

Compressor EER is directly proportional to the efficiency product \( \eta_m \eta_i \). Therefore, the time dependence of the relative EER of a compressor can be obtained directly from the time dependence of the efficiency product expressed by Eq (8),

\[ \text{EER}(t) = \left(1 + \eta_m \eta_i \left[ \alpha_1 \left(\frac{\delta(t)}{\delta^*}\right)^m - 1 \right] + \alpha_2 \left(f(h/\delta(t)) - f(h/\delta^c)\right) \right)^{-1}. \]

In this expression, \( \text{EER}(t) = \text{EER}/\text{EER}^\infty \) and \( \text{EER}^\infty \) is the compressor EER after full break-in. A mature, well worn-in compressor is one in which the composite roughness of its mating surfaces is sufficiently below the EHD film thickness to prevent wear (\( \delta^c \leq h/3 \), or \( \lambda^c \geq 3 \), well into full film EHD lubrication regime).

**Fig 2**

Roughness Variation During Break-in

![Graph showing variation of roughness with time](image-url)
Figure 3 presents the relationship implied by Eq (13) for typical values of $a_i$ and $a_f$. It can be seen that compressors from the same population but with different initial values of their composite surface roughness would still reach the same asymptotic value of EER after complete break-in, with those having a higher initial level of roughness taking longer to break-in fully. It can also be seen that virgin compressors, with expected large variation of their initial roughness, will exhibit wide variability in their EER. This variability is expected to diminish as the compressors break-in with longer running time. This has been confirmed in numerous calorimeter tests of virgin scroll compressors.

![Variation of EER During Break-in](image)

Figure 4 compares the break-in curves for two compressor models from the same family; one having twice the capacity of the other. It can be seen that the smaller compressor starts at a lower EER and takes a longer time to break-in.

**CONCLUSIONS**

The break-in behavior of scroll compressors can be explained by the application of the physical principles of wear, friction and leakage, augmented by simple assumptions regarding the relationship of compressor EER to both leakage and friction. The break-in period of the compressor was shown to depend on a time constant that is directly proportional to the indentation hardness of the mating surfaces and the thickness of the lubricant film. The break-in time constant was also shown to be inversely proportional to the $PV$ loading of the rubbing surfaces and their wear coefficient. However, care must be taken in determining the extent to which $PV$ loads should be increased, or hardness and film thickness decreased to effect faster break-in. On the other hand, compressors with smaller orbit radius (low $V$) and larger contact surfaces, e.g., thicker wraps and larger thrust surfaces (lower $P$) are likely to have prolonged break-in periods.

It is shown that the smaller compressor in a family of similar compressors would take longer to approach its mature performance. It is also shown that the length of time it takes a particular compressor model to reach its mature EER, or to approach it within an acceptable practical tolerance, 1%, say, depends on the initial composite roughness of its rubbing surfaces, principally the tips and floors of the fixed and orbiting scrolls and the seal or thrust surfaces.

Because the initial roughness of a population of same-model compressors is expected to be randomly distributed with a relatively large dispersion (Fig 1), the EER statistics of such a population are also expected to exhibit larger variability. As the mean and dispersion of the roughness decreases during the break-in period, so will the variability of the EER data. It is for this reason that audit data of the compressors' after 1 or 2 hours of break-in are meaningless.
The model is only suitable for predicting relative behavior of compressors of the same family; its application to compare compressors of different families or different designs is not advisable.

REFERENCES


