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SIMPLIFIED COMPRESSOR PERFORMANCE SCALING

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ABSTRACT

This paper presents an analytical method adopted to scale performance data from one machine to another within the same general family type using simplified assumptions. This is used for a fast and accurate performance prediction before committing to costly test procedures.

For example, the capacity variation between scaled machines is first broken up into leakage effects, pressure drop effects, and so on. Similarly, variations in power consumption and other effects are also broken down into separate components. Relationships between the various loss mechanisms and a linear scaling factor are assigned. Constants in the scaling relationships are determined using performance data from existing machines.

As a more general exercise, it is also possible to break the scaling factor into a number of independent variables which describe the particular compression device being considered. In this manner more general design and optimization studies may be conducted. This paper uses a small-capacity oil-flooded twin screw compressor as an example.

INTRODUCTION

Compressor designers must often predict performance of different sized machines of similar types or within the same family. These different machines essentially have the same suction, compression and discharge processes. However, they may differ in displacement or size. Obviously, testing to evaluate performance of various compressors of a given family is a time consuming and expensive process. This paper presents an approach which minimizes testing needs to develop performance trends across a family of compressors. Only one particular compressor of the family of compressors is tested and the data from this standard machine is used to predict performance of other machines of the same family with similar scaling. In this study, machine scaling is performed for a single rating condition.

When properly developed, this method can be used to predict and optimize compressor performance prior to embarking on costly test programs. Due to space restrictions, the scope of this paper is limited to a study sufficient to illustrate the method itself and to present some performance scaling trends for an example based on a commercially available twin screw compressor.

ANALYTICAL APPROACH

This method of scaling performance data from one machine to other within a given family of compressor centers on breaking down various performance parameters into a combination of geometric variables, mechanical variables and constant coefficients. For example, in the case of a screw compressor, seal line length, blow hole area, and end clearance are some geometric parameters used to predict performance changes if the screw compressor rotors were made bigger or smaller. Tip speed or rotational frequency of the driven rotor are some of the mechanical variables. Once these parameters, their relationship to compressor size, and their influence on performance are established, they may be computed for any size machine. The influence on performance may be established analytically, empirically, or using a combination of methods. This approach can be applied to any given parameter, such as capacity, leakage losses, parasitic
losses, and others as long as one can write an initial relation linking the parameter with compressor geometry and operating speed. We will look at a screw compressor as an example and assume that base line machine test data is available. We are interested in scaling this machine either up or down in size and are interested in both capacity and power.

**Machine Scale**

We use a parameter $L$, called machine scale, to identify various products within a family of compressors. As we go to larger and smaller machines, all components, at least initially, scale linearly with $L$. When $L = 1$ we are working with the original or so-called baseline machine. When $L = 0.5$, we have a screw compressor whose dimensions are half those of the baseline machine. A more sophisticated approach will break the compressor geometry into a number of parameters, such as rotor diameter, rotor length, centerline spacing, and so on, each of which scale separately, and which can be used for more detailed optimum design studies. This simplified example mainly shows us the basics of the technique and reveals general performance trends of this compressor type.

**Capacity Scaling**

We consider leakage and suction pressure drop due to flow losses as the parameters most influencing capacity, even though there may be other factors at work. We separate these parameters into the influence of seal line length, blow hole area, end clearance and tip clearance for leakage losses and flow velocity for pressure losses.

The seal line area $A_s$ may be given by

$$A_s = L_s \delta_s$$

(1)

where $L_s = \text{seal line length}$ and $\delta_s = \text{seal line clearance}$. The seal line length is directly proportional to $L$ in this case and the seal line clearance will scale with $L^i$ where $i$ is between zero and one. A value of $i$ greater than zero recognizes the fact that tolerances increase with part size, although a value of $i$ less than 1 also recognizes that tolerances are a function of process capability as well as part size and may scale less than directly with size. A specific value of $i$ must be chosen based on the designer's experience or demonstrated process capability.

The seal line area becomes

$$A_s = L_{s0} \delta_{s0} L^{(1+i)}$$

(2)

where $L_{s0}$ and $\delta_{s0}$ are baseline values of seal line length and clearance.

Similarly, where the end clearance leakage area $A_e$ is given by the product of a characteristic lobe size $S$ and an end clearance $\delta_e$, this area becomes

$$A_e = S_{e0} \delta_{e0} L^{(1+j)}$$

(3)

where $S_{e0}$ and $\delta_{e0}$ are baseline values of the characteristic lobe size and end clearance and $j$ is a constant with a range and meaning similar to $i$ above.

Tip clearance area $A_t$ is arrived at in the same manner

$$A_t = T_{t0} \delta_{t0} L^{(1+k)}$$

(4)
where $T_{10}$ and $\delta t_0$ are baseline values of tip length and clearance and $k$ is a constant with a range and meaning similar to $i$ and $j$ above.

Finally, the blowhole area $Ab$ is not generally a function of clearances but only of rotor size.

$$Ab = Ab_0 L^2$$  \hspace{1cm} (5)

where $Ab_0$ is the baseline blowhole area.

Now we let $A$, $B$, $C$, and $D$ respectively be coefficients which relate the individual seal line, end clearance, blow hole, and tip clearance leakage areas to their relative leak rates. The total leakage $L_t$ may be expressed as

$$L_t = ALS_0 \delta s_0 L^{(1+i)} + BS_{t0} \delta t_0 L^{(1+j)} + CAb_0 L^2 + DT_{t0} \delta t_0 L^{(1+k)}$$  \hspace{1cm} (6)

For suction pressure drop, we assume that the loss in volume flow rate is proportional to the loss in density and to the loss in vapor pressure.

$$\frac{\dot{V}_a}{\dot{V}_i} = \frac{\rho_a}{\rho_i} = \frac{p_s - \Delta P}{p_s}$$  \hspace{1cm} (7)

where $\dot{V}_a$ and $\dot{V}_i$ are the actual and ideal volumetric flow rates, respectively, $\rho_a$ and $\rho_i$ are ideal vapor densities, $p_s$ is the suction pressure, and $\Delta P$ is the flow pressure loss. The pressure loss and flow velocity $u$ may be related to the ideal displacement rate and the baseline flow area $A_o$.

$$\Delta P \propto u^2 = \left( \frac{\dot{V}_i}{A_o} \right)^2$$  \hspace{1cm} (8)

where $A_o$ is the baseline suction flow area. Scaling the components of $u$ to compressor geometry and rotor speed,

$$u \propto \frac{L^3 N}{L^2} = LN$$  \hspace{1cm} (9)

where $N$ is the compressor speed. Applying an appropriate coefficient $E$,

$$\frac{\dot{V}_a}{\dot{V}_i} = \frac{p_s - E(LN)^2}{p_s}$$  \hspace{1cm} (10)

Combining the results of these two analyses gives us an expression for overall volumetric efficiency

$$1 - \frac{ALS_0 \delta s_0 L^{(1+i)} + BS_{t0} \delta t_0 L^{(1+j)} + CAb_0 L^2 + DT_{t0} \delta t_0 L^{(1+k)}}{V_0 N_0 \frac{p_s - E(LN)^2}{p_s}}$$  \hspace{1cm} (11)

**Power Scaling**

Perhaps the greatest loss of compression power in screw compressors, other than motor losses, is through leakage of flow out of the compression chamber. As in the volume flow calculation, the four leakage paths are the seal line clearance,
end clearance, tip clearance, and the blow hole. These losses may be computed in a similar manner, with similar scaling relationships, except that now different coefficients are needed to properly describe the associated power losses. A modified form of the leakage equation becomes

\[ P_L = FL_0 z_0 L^{(1+n)} + GS_l z_0 L^{(1+j)} + HAA_0 L^2 + JT_0 z_l L^{(1+k)} \]  

(12)

where \( P_L \) is the leakage power loss and \( F, G, H, \) and \( J \) are conversion coefficients.

Another significant loss mechanism, mainly in high speed oil flooded machines, is the viscous drag of the lubricating oil between the rotor tips and the rotor bore. If we assume the drag to conform to Newton’s viscous shear equation

\[ P_v = u_t A_t \tau = u_t A_t \frac{u_t v}{\delta_t} \]  

(13)

where \( P_v \) is the viscous drag power, \( u_t \) is tip velocity, \( A_t \) is a characteristic tip area presented to the rotor bore, \( \tau \) is the film shear stress, \( v \) is oil viscosity, assumed constant, and \( \delta_t \) is tip clearance. Proportionality relations for these variables are

\[ u_t \propto LN \]  

(14a)

\[ A_t \propto L^2 \]  

(14b)

\[ \delta_t \propto L^k \]  

(14c)

Combining these, and introducing a conversion coefficient \( K \),

\[ P_v = KL^{(4-k)} N^2 \]  

(15)

Using flow relations similar to the suction flow pressure loss, the discharge flow power loss \( P_{df} \) is represented by

\[ P_{df} = M \frac{L^3 N}{L^2 N^2} = M \frac{L^5 N^3}{N} \]  

(16)

where \( M \) is another scaling coefficient and the first scaling expression in the intermediate term represents the discharge flow rate and the second expression represents the discharge pressure drop.

The total power loss \( P_t \) is the sum of these three components \( P_L, P_v, \) and \( P_{df} \) plus the motor efficiency loss and a fixed gear and bearing drag loss, assumed to be 0.5 percent for all rolling contact.

\[ P_t = \frac{P_L + P_v + P_{df}}{\eta_m - 0.05} \]  

(17)

where \( \eta_m \) is the motor efficiency.

**EXAMPLE**

We used this method to perform a design scaling study on a family of open drive (motor losses neglected) screw compressors for a series of commercial refrigeration and air conditioning applications. The existing product family consists of a constant rotor configuration which is gear-driven at a variety of speeds to generate a range of displacements. Figures 1 through 3 summarize the performance of this baseline family and of a series of compressor families scaled up and down in displacement. A typical refrigeration (high pressure ratio) operating condition forms the basis for comparison.
Figure 1 summarizes isentropic compressor efficiency. Figure 2 presents shaft input power normalized to unit displacement rate. Figure 3 presents compressor volumetric efficiency. The baseline (existing) compressor family is represented by the $L=1.0$ curve where $L$ is the linear scaling factor. Each individual speed range is limited at the top end by a constant predetermined rotor tip speed (same for all families) and at the bottom end by a constant ratio of the top speed, providing a uniform capacity range.

Referring first to Figure 3, we see steadily increasing volumetric efficiency with increasing speed for all compressor families. This reflects the fact that the constant leakage rate for a given geometry, which is not a function of rotor speed, becomes proportionally smaller relative to total capacity as the machine runs faster. The fact that none of the curves begins to dip down at the upper speed range reflects the fact that the baseline family was designed for low flow losses throughout the design speed range. Our constant maximum tip speed rule means that the product $LN$ is constant and scaling relation (10) tells us that the low flow losses should hold throughout our study. We see the volumetric efficiency steadily falling off for smaller machines as the leakage areas scale down more slowly than the overall displacement.

Figure 2, interestingly, shows the normalized power per unit displacement with a generally upward trend with increasing speed for the larger machines, shifting over to a generally downward trend for the smaller machines. Again, for the smaller machines, the effect of leakage predominates and becomes proportionally smaller as machine speed and thus displacement go up. However, as the speed reaches the maximum, the effects of viscous drag begin to show and the curves flatten out. On the other hand, the larger machines already have larger displacements at lower speeds relative to leakage areas, and the effects of viscous drag show up much earlier. Note that the baseline family, whose proportions are optimized for its application range, has a fairly flat curve, with the net effect of leakage and drag generally remaining constant.

In Figure 1 we see the total effect of power consumption and volumetric efficiency, translated into overall isentropic efficiency. Note that for the largest machine, we see efficiency at first increasing with increasing speed as the relative effect of leakage is reduced with increasing displacement. Toward the upper end of the speed range, we see the efficiency flatten and begin to fall off as the effects of viscous drag become pronounced. For each smaller machine, we see less and less effect of viscous drag on the performance. For the baseline family, we see the effect of drag just leveling the curve out at the highest speed. For all smaller machines, drag never becomes quite predominant.

Figure 1 also illustrates one of the inherent difficulties with trying to scale down screw-type or any compressor which relies on clearance control for sealing. The effect of leakage on both capacity and power are both amplified as the leakage areas scale down more slowly than compressor displacement. In general, we can relate the downward trend of screw compressor sizes in the market with improvements manufacturing tolerances which allow tighter clearance control.

Finally, all three figures might lead us to conclude that the smaller machines could be improved if they could be run even faster still. However, this scaling study did not take into account the effects of compressible flow at high speeds. The baseline machine was designed to have acceptable flow Mach numbers at the highest rated speed and the maximum tip speed limitation preserved this throughout the scaling study. If the machines were run much faster, these effects, not accounted for in our scaling relations, would begin to appear and cause both the volumetric efficiency curves to fall off and the power curves to rise much earlier than this study would predict.

CONCLUSIONS

This method is a powerful tool for preliminary evaluation of compressor sizing. It can project overall performance, reveal performance trends, and perform rough sizing for a desired displacement.

For the limited space in this paper, we only looked at the technique for machines which were scaled proportionately. This method is easily expanded, by adding separate scaling factors for each geometric variable, into a general model useful for design optimization. This gives much better resolution to the effects of size on performance and provides a vehicle for global optimization. However, practical operating limits (such as the illustrated limit on maximum speed) which are not reflected in the individual relationships, must be kept in mind when generating and interpreting results.