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Computational Parametric Study of Scroll Compressor Efficiency, Design, and Manufacturing Issues

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COMPUTATIONAL PARAMETRIC STUDY OF SCROLL COMPRESSOR EFFICIENCY, DESIGN, AND MANUFACTURING ISSUES

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ABSTRACT

The present article briefly discusses the basic theory of the scroll compressor concept from a design point of view. The effect of different physical parameters on energy losses, design limitations, and manufacturing issues will be demonstrated. Each category is examined using realistic physical terms to indicate in detail the contribution and significance of each parameter. By means of using real physical parameters, a simple and easily understood optimization approach is demonstrated as a guide tool towards scroll compressor design.

INTRODUCTION

After the oil embargo of the 1970's, more attention has been paid towards the concept of energy conservation. For HVAC applications, the majority of the energy consumption occurs within the compressor. This puts emphasis on the requirement for high efficiency compressors, which are of primary importance for HVAC applications within the American market. Overseas, in particular within European and Japanese markets, the need for low noise/vibration is more predominant than the efficiency. An alternate compression mechanism which potentially offers these advantages for both markets is the scroll compressor.

The scroll concept was invented by a French engineer in the early 1900's [1]*. Prior to the 1970's, for lack of numerical control (NC) machines, no attempt was made to develop the concept. Since then, an extensive amount of work has been conducted, nevertheless, the concept still has not been fully developed and applied to employ all its potential.

From a technical point of view, previous investigators [2-6] have solely demonstrated the effect of certain parameters on overall lumped efficiency, but no detailed design optimization procedure has been presented. It is the purpose of the present study to look at the concept from the viewpoint of energy losses, manufacturing, and design limitations by means of using realistic parameters. This should provide a simply understood design approach to evaluate the effect of relevant parameters for optimum design of the scroll compressor.

PRINCIPLE OF SCROLL COMPRESSOR

Basic structure

The basic structure of the scroll compressor, as shown in Figure 1, consists of five major components: fixed scroll, orbiting scroll, anti-rotation coupling, crankshaft, and crankcase. The two scrolls are generally defined by involutes of circles and assembled with a 180° phase difference. The fixed scroll is attached to the crankcase while the moving scroll orbits by means of a simple crankshaft. The anti-rotation coupling permits the moving scroll only to orbit and prevents any rotation.

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* Numbers in [brackets] designate references at the end of the article.

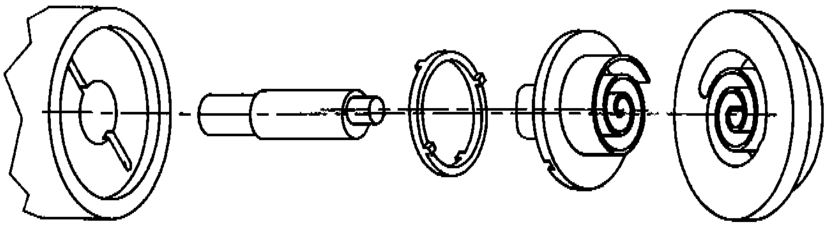

Figure 1. Basic Structure of the Scroll Compressor

Basic concept

Figure 2 shows the principle of scroll operation. The suction gas is brought in simultaneously at two locations from the periphery of the scrolls. Thereafter, the two symmetric crescent shaped pockets are moved towards the center, with a resulting reduction in pockets volume. At the center, the pair of pressurized pockets are merged together and discharged through a single port. Generally, it takes 2-3 shaft rotations to bring the fluid from the suction to discharge stage.

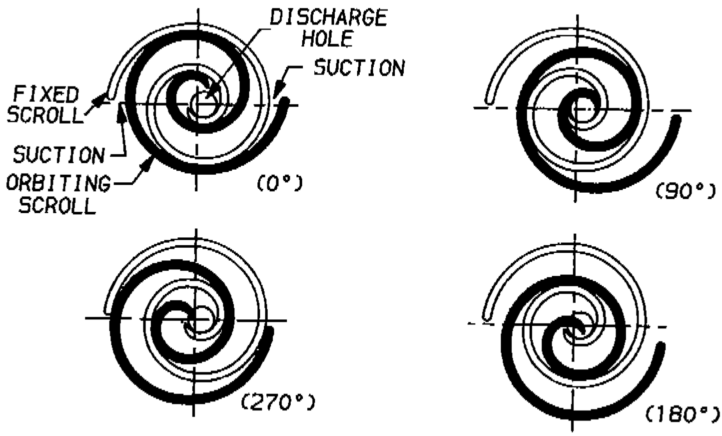

Figure 2. Principle of Scroll Compressor Operation

Governing Equations

The basic geometric variables which determine scroll profile are radius of generating circle (R_g), involute starting angle (A_s), and involute thickness angle (A_t); these are shown in Figure 3. From profile geometry, the pitch is defined as

$$P = 2\pi R_g \quad (1)$$

the wrap thickness is

$$t = R_g . A_t \quad (2)$$

and, the radius of orbit, or eccentricity, is

$$R_o = (P-2t)/2 \quad (3)$$

The displacement, or suction, volume is

$$V_s = P.R_o.H\ (2A_w - A_t - 3\pi) \quad (4)$$

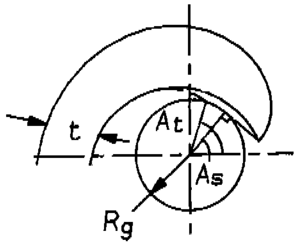

Figure 3. Basic Geometry of Scroll Profile

where H is the wrap height and A_w is the final involute wrap angle corresponding to the first sealing point which contains the outer pair of crescent shaped pockets at the end of the suction stage.

Using equation (4) and applying it to the innermost pocket, gives the final discharge volume,

$$V_d = P.R_o.H\ [2(A_s+3\pi)-A_t-3\pi] \quad (5)$$

The design built-in volume reduction ratio is obtained by dividing V_s by V_d:

$$V_s/V_d = [2A_w-A_t-3\pi]/[2(A_s+3\pi)-A_t-3\pi] \quad (6)$$

PARAMETRIC STUDY

For a scroll compressor, the science or art of optimum parameter selection is more complicated than for reciprocating compressors. In the latter case, pumping geometry is selected based on an optimum bore/stroke ratio for a given displacement volume. In the case of the scroll compressor, due to the larger number of variables, the optimization study becomes more complex. In order to generate a scroll pumping geometry, it is required to satisfy equations (1) through (6). From a design point of view, the number of unknowns are more than the governing equations. Thus, it is very important to recognize the influence of each parameter to tailor the concept according to specific needs.

In the present study, by knowing the capacity, volume reduction ratio, and wrap thickness, a range of wrap height and starting angles are used to generate the data. The output data is plotted versus height for different starting angles and categorized based under the three distinct areas of manufacturing, design limitations, and energy losses. The flow chart indicated in Figure 4 provides the approach taken to define the scroll geometry, and thereafter, to evaluate the parameters for the study and selection of an optimum scroll geometry.

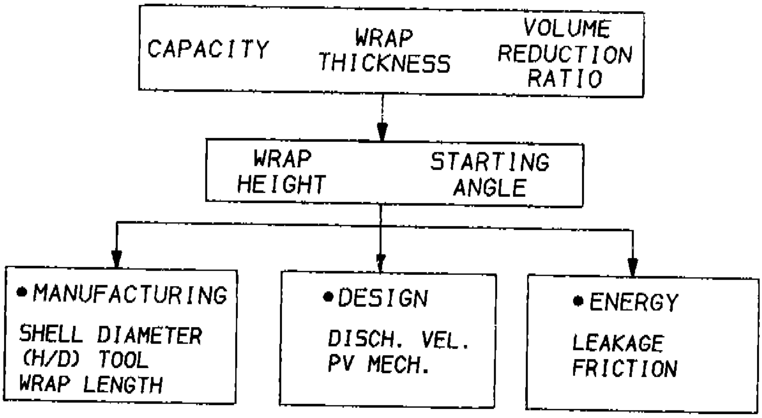

Figure 4. Optimization Flow Chart

Capacity

The capacity is chosen and considered as an input. The relation between the capacity and displacement volume is,

$$V_s = \text{Capacity}/60\ \rho_s\ N\ \eta_v \Delta H$$

where ρ_s is the suction gas density, N is the speed, η_v is volumetric efficiency, and ΔH is the enthalpy increase during evaporation. It should be noted that ρ_s and ΔH are determined by the operating condition (eg. for ASRE/T, $T_{sat.evap.}$ = 45^oF, $T_{sat.cond.}$ = 130^oF, Return gas superheat = 50^oF). Based on experimental data, the volumetric efficiency for the scroll type compressor family is generally greater than 90%, depending on the leakage and suction gas condition (c.f. 75% for reciprocating compressors).

Volume reduction ratio

Similar to a screw compressor, the scroll compressor is also a fixed compression ratio machine. Matching the proper volume ratio compressor to the application is important when optimizing for efficiency. In the present study, the volume ratio corresponding to the ASRE/T operating condition is used.

Wrap thickness

The magnitude of thickness, t, plays an important role on the following:

(a) rigidity of the scroll element structure during machining.
(b) sustaining gas forces and thermal distortion.
(c) minimizing the tip leakage.

Depending on the manufacturing process, the effect of H/t must be considered rather than thickness t by itself. One has to compromise between the machining conditions and the magnitude of H/t to avoid any undesirable warpage and surface finishes. The beginning and end of the wrap, where there is no side support, are the most critical regions in the manufacturing and machining process.

In terms of gas forces, the middle of the wrap length is exposed to the highest pressure differential during operation at design conditions, as well as the central portion when the machine is running at off-design conditions. The central portion of the wrap is also the weakest due to lack of side support.

Finally, the magnitude of t has a direct effect on the tip leakage. The dependency of thickness on tip leakage can be reduced by decreasing tip clearances or employment of a tip seal.

Height and starting angle

As will be demonstrated, the main reason for selecting the two variables, height and starting angle, as dependent variables are as follows:

- Most major parameters, except discharge velocity, are not very sensitive to the magnitude of starting angle within its practical range.

- Most parameters are a strong function of the height. This provides a simple way of demonstrating the data solely as a function of height only. In addition, the height is a real physical dimension which eases understanding of the data from a design point of view.

OUTPUT PARAMETERS

Having the five input parameters of capacity, volume reduction ratio, thickness, height, and starting angle, a computer simulation program is used for parameter evaluation. The computer simulation structure is divided into three sections:

(A) Geometry: This section generates the scroll geometry and evaluates parameters such as minimum required shell diameter, discharge port area, cutting tool parameters, etc. A graphics display of scroll geometry is also provided.

(B) Thermofluid: In this section, volumes and thus pocket pressures are calculated assuming a polytropic compression process. Real refrigerant gas properties are used and the gas flow through all ports are assumed steady and isentropic. Both flank and tip leakages are modeled assuming one-dimensional Fanno flow within the clearance space at the tips and flanks of the scroll wrap. A Fanno flow model is necessary to describe these leakages because frictional effects are significant due to the extremely narrow clearances relative to leakage path length. The flank leakage path length model has been discussed in more detail in Reference [7].

(C) Kinematic: By means of scroll compressor geometry, predicted pressures, and component masses the resultant force magnitudes and moments about the orbiting scroll axes are calculated. Bearing forces and phase angles on the crankshaft, as well as the forces on the anti-rotation coupling, are subsequently evaluated. Knowing the forces of interaction and sliding velocity on each moving component, the pressure-velocity (PV) and friction losses are evaluated. Further, the minimum oil film thickness is evaluated for each journal bearing.

Manufacturing

Shell diameter - The outside configuration of the compressor is a major contributing factor for HVAC unit design and marketing of compressors. Ideally, the intent is to provide the smallest overall pumping assembly diameter and height. A major factor affecting shell diameter is the diameter of the motor employed for that specific capacity.

From the geometry section of the simulation program, the minimum required pumping assembly diameter is calculated (D_{pump}). The data is normalized by dividing the calculated diameter by the motor diameter (D_{motor}) as shown in Figure 5. From Figure 5, the recommended range of operation corresponds to the areas where D_{pump} / D_{motor} is equal or below 1.0. This data is relatively insensitive to starting angle (below 2% variation).

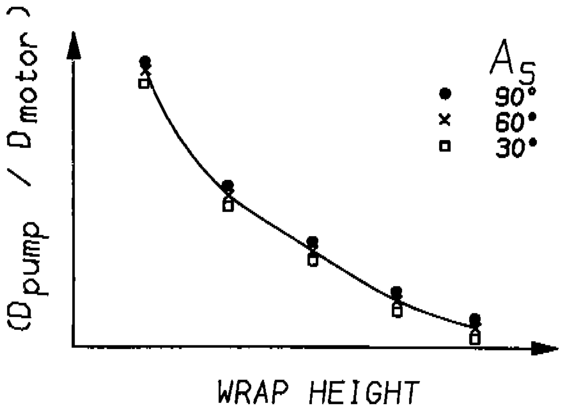

Figure 5. Effect of Scroll Geometry on Non-Dimensional Overall Diameter

Wrap length - One parameter which is of significance from a manufacturing point of view, is the overall scroll wrap length. The wrap length determines the manufacturing time required for machining each scroll wrap, which is one of the dominating cost factors. In general, for a given capacity, the wrap length decreases as the height increases. This effect is most significant at the smaller heights. In addition, the wrap length decreases by decreasing the starting angle.

Cutting tool parameter - The major manufacturing design issue influencing the optimum scroll parameter selection is the cutting tool dimension. Ideally, one requires a large diameter cutting tool and short flute length to avoid cutting tool deflection. For this purpose, $H/(P-t)$ is used to demonstrate the variation of cutting tool height-to-diameter ratio $(H/D)_{tool}$ as a function of the wrap height as shown in Figure 6. A rigid cutter (smaller deflection) corresponds to a smaller $(H/D)_{tool}$. The effect of starting angle is less than 5% on this parameter. The maximum allowable cutting tool parameter $(H/D)_{max}$ is determined experimentally for a given material, cutter, and cutting operation.

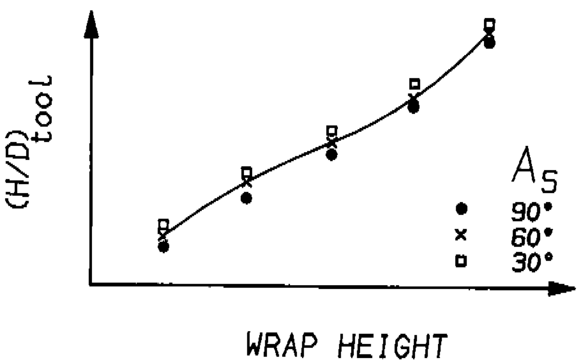

Figure 6. Effect of Scroll Geometry on Cutting Tool Parameter

A comparison of Figures 5 and 6 for $(D_{shell} / D_{motor}) \leqslant 1.0$ and $(H/D)_{tool} < (H/D)_{max}$ respectively, defines the boundaries for the wrap height geometry selection. The next step is to make certain the design limitation parameters are not exceeded for this range while choosing an optimum geometry for efficiency.

Design limitations

Discharge velocity - The size of discharge port is the major factor controlling the size of the central pocket and thus, the starting angle. The goal is to maximize the port area within the central oval shaped pocket formed between the orbiting and fixed scroll wraps just before the start of the discharge process. This requires a compromise between the discharge port area and manufacturing of a non-circular hole. Having the port area for a scroll geometry, the discharge gas Mach number, M_D is calculated as shown in Figure 7. Figure 7 indicates that for higher starting angles, the discharge gas parameter becomes insensitive to the height variation. The maximum limiting M_D (eg. $M = 0.3$) should be evaluated experimentally, or through related literature [8].

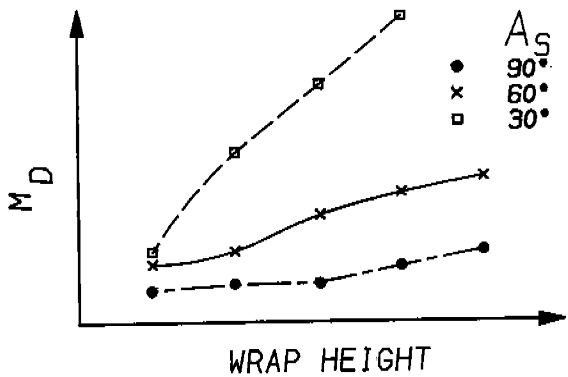

Figure 7. Effect of Scroll Geometry on Discharge Gas Mach Number

Bearing PV - Another major design limiting parameter is the PV on the thrust surfaces, anti-rotation pad sliding surfaces, and journal bearings. The maximum cooling load condition experienced by the compressor should be used for this computation.

In general, the PV parameter on the axial thrust surfaces and coupling pads decreases with increasing height, whereas on the journal bearings PV increases. The effect of starting angle is insignificant (less than 5%). By increasing the height, the radial and tangential gas forces on the scroll wrap increases, resulting in higher journal bearing forces. In contrast, by increasing the height, the orbiting scroll base area exposed to high pressure gas is reduced. This occurs while the thrust sliding velocity also reduces (smaller orbiting radius). A combination of the two produces a smaller PV on the thrust surface. The limiting case for each sliding surface is dependent on material combination, surface condition, velocity and lubricants.

Energy Losses

Leakages - In general, gas leakages between scroll pockets are from high pressure to intermediate pressure and from intermediate pressure to low pressure levels. Figure 8 shows the tip and flank leakage losses for two different clearances; the data are normalized using the indicated ideal work to compress the gas.

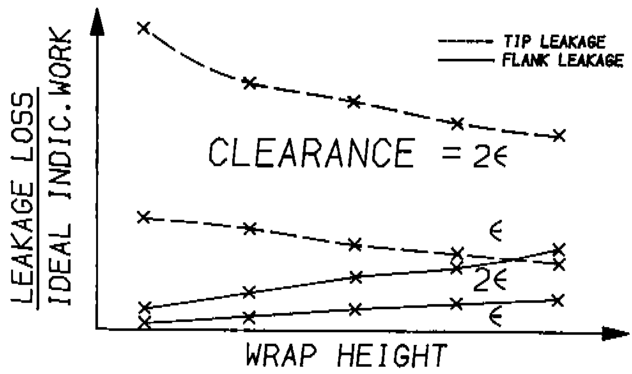

Figure 8. Effect of Flank and Tip Clearances on Non-Dimensional Leakage Loss Parameter

For a given clearance, the effect of tip leakage loss is significantly higher than the flank leakage. This is due to the difference in leakage path lengths, both along and through the clearances. The leakage path length along the clearance for tip leakage is related to the wrap length and in the case of flank leakage is in direct proportion with wrap height. In general, the former path length is longer than the latter, which results in a higher tip leakage than flank leakage loss along the clearances.

The leakage path length through the clearances corresponds to the wrap thickness for tip leakage loss, while in the case of flank leakage, it corresponds to the clearances created by the two mating wrap surfaces. The two mating curvatures have similar radii which acts as a longer restrictor for flank leakage than the tip leakage. This results in higher tip leakage than the flank leakage loss through the clearances.

Overall, the results indicate that more emphasis must be made on reducing tip leakage loss, in particular, for the configuration of low wrap height. Having selected the clearances between the scroll wraps, the total leakage loss determines the optimum scroll wrap height.

Friction - The frictional losses have been calculated by knowing the normal forces of interaction, the friction coefficients, and relative velocity between sliding surfaces.

The same trend as for bearing PV is also achieved for frictional losses. The coupling pad losses are at least an order of magnitude smaller than thrust and journal bearing losses. By increasing the wrap height, the thrust frictional losses reduce while the journal bearing losses increase. For final wrap height optimization, accurate values of the friction coefficient for thrust and journal bearing surfaces are required.

CONCLUSION

Having reviewed the basic theory of the scroll compressor, the two most important geometry parameters were identified as the height and starting angle. The significance of the starting angle was found to be small, except in the discharge velocity. The height was considered as a realistic physical geometry parameter which strongly controls the compressor optimization. This combination of weak and strong parameters simplified the optimization study.

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