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Software support for screw rotor design, manufacture and quality control

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ABSTRACT

As machine elements the helical rotors of a screw compressor are unusual because of their complicated shape. This poses serious challenges in all phases of design, manufacture and quality control, which can be satisfactorily met only by the aid of the computer. A number of software packages addressing various aspects of these problems have been described in the literature (*e.g.*, [2], [3]), but they all seem limited in scope.

A fully satisfactory approach must be very *general*, in that no restrictions should be placed on the rotor profile. It should also be *comprehensive*, so that it covers all aspects of the process from design of the profile, through application of appropriate clearances and creation of tool coordinates to evaluation of the quality of the finished rotors.

INTRODUCTION

The early practioners of the art of making screw rotors had to meet the challenges of designing and making the rotors with a good deal of ingenuity and inspired guesswork. The reason for this is the unusual combination of complicated shape and small tolerances which has few parallels in mechanical engineering. As a result of these difficulties the rotor profiles used could seldom be optimized with regard only to function and ease of manufacture; one also had to use profiles whose shapes could be easily described and calculated. Because of the effort required in designing a profile, only a few profiles were available in the early days; the same profile had to be used in quite different circumstances. Eventually the difficulties were somewhat eased by the introduction of computers for calculating tool shapes *etc.*, but available profile systems were still quite inflexible.

With the introduction of powerful desktop computers there are no longer any valid reasons for such restrictions. A specific profile can be designed for each application, if desired. This assumes, of course, that software of sufficient generality is available to support all aspects of design, manufacturing and quality control. For this to work satisfactorily one needs a carefully thought out system for describing the rotors and tools and for transferring needed data between the

different parts of a software package. We have, during a period of several years developed such a software system which is now nearing completion. The paper [1] *A Modern Way to Good Screw Rotors* by Sören Edström describes the philosophy behind. In the present paper the software package is described. It must be said at the outset, however, that no software will by itself be able to design a good profile; this must be left to the expert of long experience. What the software can do, is speed up the design work to a great extent. This will allow the design of a more closely optimized profile for each application. Furthermore, the software can substantially ease all the design work needed *after* the basic profile has been settled upon. Many aspects of manufacture and quality control can also benefit from the package.

Before we go on to the details of different parts of the package, first a few words on the user interface of the programs. This could be laid out in a number of different ways, independent of the algorithms used. However, software such as this will be most useful if it is so easy to use, that any engineer familiar with the field can use it productively after only a brief introduction. Therefore the programs are basically menu-driven, so that all the user has to do is answer questions, or in some cases choose from a few different alternatives. The input of the programs will in most cases be both a file containing tool or profile coordinates and some information given by the user via the keyboard, *e.g.*, tool setting data. The output is mostly a file and, optionally, a printout or plot. The files are ASCII text files, and start with some general information about the profile or tool in question, such as the number of lobes and the diameter. In the case of a tool all data about the setup, like cutting angles and angle and distance between tool and rotor axes, are in the file. Then follows a list of coordinate points that define one thread of the rotor or, in the case of a tool, a blade for cutting one thread. The number of coordinate points will generally be many hundreds (up to 2000) on each thread of the rotor. The coordinate points are numbered, and some will be marked as special identification points for the purpose of checking the rotor. The same numbering will be kept consistently in all files, including those for tool coordinates. For the details of this we refer to the paper [1] in these proceedings. The exact format of the files will not be described here. We only point out that it is essential that each file is complete, *i.e.*, contains all information needed by other parts of the software. This means that in addition to coordinates the file will need to contain data like center distances, diameters, setting angles, *etc.* Otherwise the risk of confusion would be great, for example between tools for cutting the same profile in different setups.

The present versions of the programs require an IBM-compatible PC of AT type and, for hardcopy output, a printer and plotter. A mathematics coprocessor is useful to speed the computations. Most standard printers and plotters can be used, but we have found that a standard laser printer produces monochrome printouts and plots of excellent quality. In spite of being limited to letter size, monochrome plots, a laser printer should be sufficient for the needs of most users. An important factor in the utility of the programs is, that the time required for calculations by any of the programs is measured in seconds rather than minutes. It is therefore quite easy to do extensive experimentation with various settings before settling on a final design.

The first duty of the software is to convert a 'recipe' for a rotor profile to coordinates for a rotor of a particular size. These will be so called zero clearance coordinates, *i.e.*, the calculated rotor shapes will fit perfectly with no intermesh clearances. These coordinates are used for further calculations involved in the design of a compressor. A profile 'recipe' consists of a sequence of segments of various mathematical curves (*e.g.* a circular or involute arc). Each one can be located either on the male or the female rotor. A matching segment on the other rotor is assumed and its coordinates are calculated by the program. The various segments are joined together smoothly, although a corner may also be included.
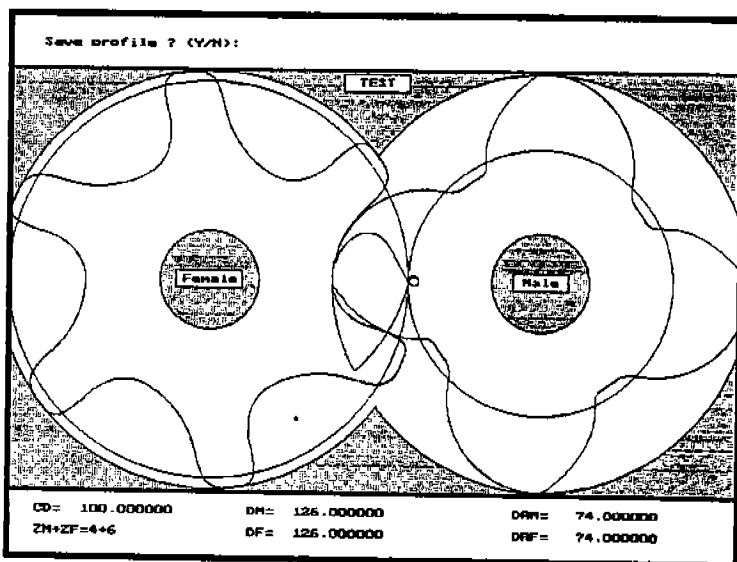
Screen of the FORM *program*

As described in [1], the sealing line is of crucial importance for the qualities of the compressor. There is therefore a facility to plot (on screen or paper) the three dimensional sealing line from an arbitrary point of view and compute its coordinates and arclength.

The compressed gas between the rotors will exert a torque on each rotor. This will always be much larger on the male than on the female rotor. It is important to check the magnitude of the torque acting on the female rotor, since insufficient torque could cause 'rattling', leading to rapid wear and early failure of the rotors. There is a program to evaluate the rotor torques. It produces a screen plot of 'specific torque' on the two rotors in a plane perpendicular to the axes of the rotors as a function of their angle of rotation, and calculates the percentage of the average total specific torque exerted on the female rotor.

To determine rotor forces, port shapes, *etc.*, for a given pair of rotors the so called volume curve is needed. This is a plot of the volume enclosed between two cooperating threads of the rotors, as a function of the angle of rotation. The plot is normalised so that the maximum value is at 100%. The volume curve is calculated making no specific assumptions about rotor geometry so that any conceivable rotor profile can be used. It is plotted on screen and, if desired, on the plotter. The volume data is also saved in a file to serve as input for other programs.

From the volume curve one can determine at which angles the inlet and outlet ports should open and close so as to achieve a given built-in volume ratio; this is of course determined by the desired output pressure. The theoretical shape of the ports is calculated and can be plotted. The data is also saved to a file, *e.g.*, so as to form the basis for data for milling the ports on numerically controlled machine tools.

From the zero clearance coordinates and the volume curve one can determine the forces exerted in different directions on the rotors under given running conditions. This can be used for example to correctly dimension the rotor bearings for the intended compressor duties. Again, the program makes no assumptions as to rotor geometry so that it can be used with any desired rotor profile.

The next step will be to apply nominal working clearances in all the coordinate points. Clearances have to be chosen so as to insure that interference and consequent seizure of the rotors can not occur under any reasonable running conditions. They are also used to create a well defined contact band on the rotor so as to have torque transfer take place under controlled conditions. Although the profile designer will normally be expected to suggest a clearance distribution this may well have to be modified by experience. The optimal clearance distribution will obviously depend on achievable machining accuracy, operating temperature range, *etc.* Thus a certain fine tuning of clearances would normally be needed. The program allows this to be done with minimum effort. In practice, the total clearances need to be distributed between the two rotors. The clearances specified in the program are therefore actually deviations from the zero clearance profile; the actual intermesh clearance is the sum of the 'clearances' specified on each rotor. For each rotor the user specifies a clearance in a number of coordinate points, typically around 20. The program interpolates smoothly to determine appropriate clearances in all other coordinate points, in such a way that interference can not occur anywhere. Common interpolation techniques (*i.e.*, cubic splines) can not be used here since non-interference can not then be guaranteed.

As an aid in deciding what clearances to apply there is a program which will calculate the change in clearances caused by temperature changes of the rotors. Finally, we have under development a program to calculate the size of the 'blow hole' on the high pressure side of the mesh. This is the opening between different threads caused by the fact that modern profiles have sealing lines that do not extend all the way to the 'cusp' where the bores for male and female rotors meet.

$$F_{rx} = -\int_{\theta_1}^{\theta_2} P \, \mathcal{F}(\theta) \, \cos\theta \, d\theta \tag{22}$$

$$F_{ry} = -\int_{\theta_1}^{\theta_2} P \, \mathcal{F}(\theta) \, \sin\theta \, d\theta \tag{23}$$

where the form of $\mathcal{F}(\theta)$ depends upon the current value of θ_M. This loading is represented in Figure 3.

DETERMINATION OF BEARING LOADS

The methods presented for computing the loads due to compression are applied to both rotors of a screw compressor and the free body diagrams, presented in Figures 4 and 5, are obtained. The forces, F_{mx}, F_{my}, F_{fx} and F_{fy}, include the components from both the tangential and the radial loads. The rotors are modeled with radial bearings at each end of the shafts and a thrust bearing at one end of the shaft of each rotor. The compression and contact force resultants are shown to act in separate planes.

The contact between the rotors is modeled using involute helical gear theory [2], as stated in the assumptions in the section. Therefore the equations relating the components of the contact forces are

$$C_x = C_y \tan\psi_n \sec\sigma \tag{24}$$

$$C_z = C_y \tan\sigma \tag{25}$$

where ψ_n defines the surface normal at the contact point.

The equations of motion for each rotor, along with the equations relating the contact forces, are used to obtain the bearing loads as functions of the rotor compression loads. The contact force C_y is given by

$$C_y = M_{fz}/R_f \tag{26}$$

The bearing loads are as follows

$$B_{1x} = -\frac{1}{L_f}[M_{fy} + C_x L_2 + C_z R_f + F_{fx}L_2'] \tag{27}$$

$$B_{1y} = \frac{1}{L_f}[M_{fz} - C_y L_2 - F_{fy}L_2'] \tag{28}$$

$$B_{1z} = -[C_z + F_f] \tag{29}$$

$$B_{2x} = \frac{1}{L_f}[M_{fy} - C_x L_1 + C_z R_f - F_{fx}L_1'] \tag{30}$$

$$B_{2y} = -\frac{1}{L_f}[M_{fx} + C_y L_1 + F_{fy} L_1']$$ (31)

$$B_{3x} = \frac{1}{L_m}[-M_{my} + C_x L_4 - F_{mx} L_4' - C_z R_m]$$ (32)

$$B_{3y} = \frac{1}{L_m}[M_{mx} + C_y L_4 - F_{my} L_4']$$ (33)

$$B_{3z} = C_z - F_{mz}'$$ (34)

$$B_{4x} = \frac{1}{L_m}[M_{my} + C_x L_3 + C_z R_m - F_{mx} L_3']$$ (35)

$$B_{4y} = -\frac{1}{L_m}[M_{mx} - (C_y L_3 + F_{my} L_3')]$$ (36)

CONCLUSIONS

Typical moment loadings, due to the gas compression process, on the male and female rotors are shown in Figure 6. It is evident that the moment loading is periodic, with a frequency equivalent to the fundamental male rotor speed multiplied by the number of male rotor lobes. The remaining loadings have similar characteristics. The summation of these periodic loads therefore causes oscillatory reaction loads on the rotor bearings in the absence of rotor chatter. In addition, the distances L, and L'_i, locating the planes of the contact resultants and pressure resultants, are functions of the rotor rotation angle, θ_M. The variance of the locations of these planes adds another component to the oscillatory nature of the bearing reaction forces. The variation in the location of the contact resultant plane is due to the fact that it is virtually impossible to assure equal contact with all lobe contact points. A worse case scenario of rotor contact is represented by a new compressor, with little oil circulation, non-compliant lobes, and low manufacturing accuracy. Such a compressor would experience a potentially wide oscillation of the contact resultant plane location, causing large bearing load oscillations. An older, run-in compressor, with large oil circulation, compliant lobes and high initial manufacturing accuracy would represent the other extreme. It would be less prone to vibration excitation through varying bearing loads and therefore likely to generate less noise.

References

1. Andrews, R.W. and Jones, J.D., Noise Source Identification In Semi-Hermetic Twin-Screw Compressors. In *Proceedings of 1990 International Compressor Engineering Conference at Purdue*. Purdue University, West Lafayette, IN, 1990.

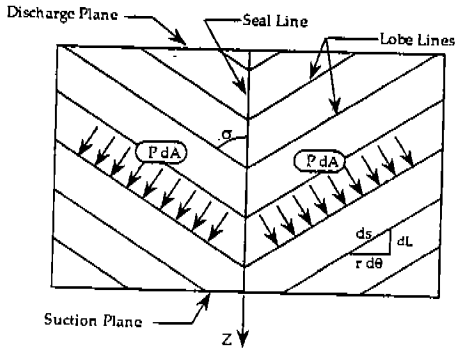2. H. E. Merritt. GEARS. Sir Isaac Pitman & Sons. London, 1962.

Figure 1: 'Unwrapped' rotor plan

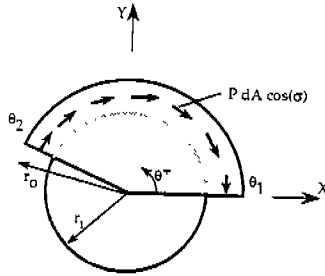
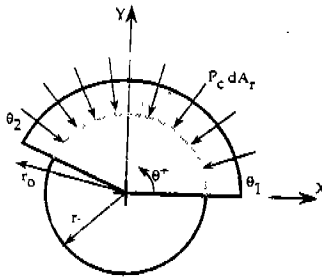Figure 2: Ilustration of tangential load due to gas compression

Figure 3: Ilustration of radial load due to gas compression
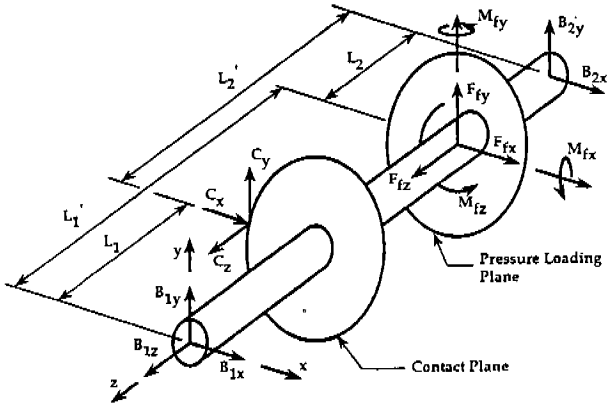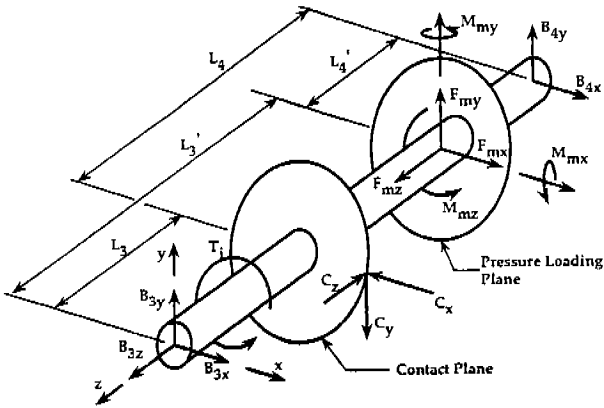
Figure 4: Free body diagram of the female rotor
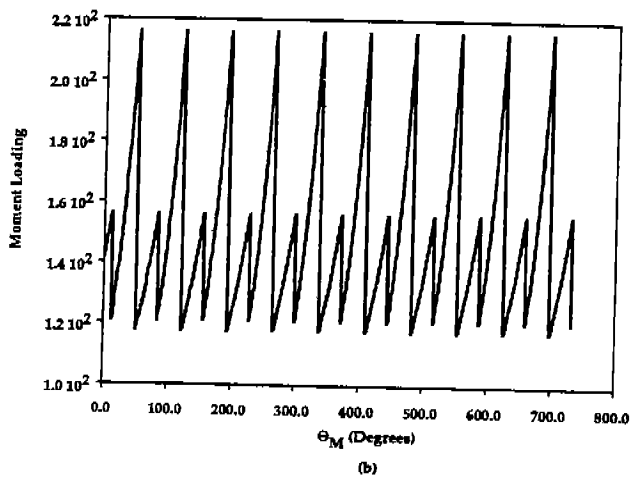

Figure 5: Free body diagram of the male rotor
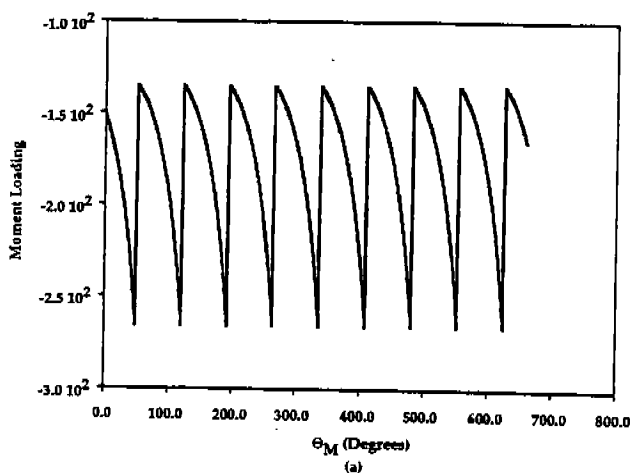
Figure 6: Typical moment loading due to gas compression on (a) the male rotor, M_{mz}; (b) the female rotor. M_{fz}.

MAIN GEOMETRIC CHARACTERISTICS OF THE TWIN SCREW COMPRESSOR

by

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ABSTRACT

The geometric characteristics of twin screw compressors greatly effect their performance and efficiency. In the use of computer modeling and simulation methods for the prediction of twin screw compressor performance, a basic problem is the analysis and calculation of the geometry characteristics. This paper presents the calculation of the main geometry characteristics, such as the compression volume curve, the sealing line length, the flute area, and the wrap angle factors, and the blow hole area.

NOMENCLATURE

xyz, XYZ :	body fixed and inertial reference frames for the male and female rotors
r :	pitch radius for the male and female rotors
R :	radius of the lobe for the male and female rotors
i :	transmission ratio between the rotors
m :	teeth number of the rotors
ϕ_1 :	turning angle for the male rotor
v :	volume of the rotor working space
h :	screw characteristics for the male and female rotors
T :	rotor lead for the male and female rotors
β_{01} :	angle at the point where the housing cross
τ_{1z} :	wrap angle of the male rotor
p :	pressure
l_r :	rotor length
v_d :	discharge volume
η_v :	volumetric efficiency
f_0 :	flute area of the rotor
c_n :	flute area factor
c_ϕ :	wrap angle factor
n :	rotation speed
D_1 :	diameter of the male rotor
Δv_{01} :	occupied volume of the male rotor by the female rotor
v_0 :	flute volume of the rotor
λ :	parameter variable
ψ :	position variable
k :	$= T/2\pi$
$1, 2$:	scripts referring to the male and female respectively
s, d :	scripts referring to the suction and discharge

1. INTRODUCTION

The twin screw compressor, as a rotary, positive displacement compressor, has gained an important position in the gas compressor industry. This position has led to the consideration of different screw profiles and the development of modern machine processes for their production [1,2]. The use of computer simulation and modeling to predict the performance and efficiency of these different profiles has been used to speed the development of improved profiles.

The geometric characteristics are complicated to model in the computer simulation of twin screw compressors. This paper presents the determination of certain important geometric characteristics which strongly affect the performance and efficiency of the compressor:

    a) compression volume curve
    b) sealing line length
    c) flute area factor
    d) wrap angle factor
    e) blow hole area

2. PROFILE CONSTRUCTION

A pair of rotor profiles are utilized as an example in the determination of the listed geometric characteristics. The following table presents the male and female profiles used. The profiles are also shown in Figure 1. Two separate rotor fixed coordinate frames x_1, y_1 and x_2, y_2 are used in describing the mating profiles.

Female	Profiles	Profiles mating with
a_2b_2	pitch circle of r_2	a_1b_1
b_2c_2	arc	b_1c_1
c_2d_2	straight line	c_1d_1
d_2e_2	arc	d_1e_1
e_2f_2	epicycloid curve	e_1f_1
f_2g_2	straight line	f_1g_1

Male	Profiles	Profiles mating with
a_1b_1	pitch circle of r_1	a_2b_2
b_1c_1	envelop curve	b_2c_2
c_1d_1	envelop curve	c_2d_2
d_1e_1	arc	d_2e_2
e_1f_1	epicycloid curve	e_2f_2
f_1g_1	epicycloid curve	f_2g_2

Table 1   Example Twin Screw Mating Profiles

3. COMPRESSION VOLUME CURVE

The compression volume curve can be obtained by integrating the swept area along the length of the rotor, or by employing the principle of virtual work [3]. A third method is presented here which computes the compression volume curve of twin screw compressors by using the profile functions directly.

3.1 Principle of Computation

When an area is enclosed by two lines (in parametric form),

$$L_1 = \begin{cases} x_1 = x_1(t) \\ y_1 = y_1(t) \end{cases} \qquad L_2 = \begin{cases} x_2 = x_2(t) \\ y_2 = y_2(t) \end{cases}$$

the value of the area can be calculated by integrating between the lines. The area shown in Figure 2 S_{abcd} will be

$$S_{abcd} = \frac{1}{2} \oint (x\, dy - y\, dx)$$

$$= \frac{1}{2} \int_{t_{1a}}^{t_{1c}} (\dot{y}_1 x_1 - \dot{x}_1 y_1)\, dt_1 + \frac{1}{2} \int_{t_{2c}}^{t_{2a}} (\dot{y}_2 x_2 - \dot{x}_2 y_2)\, dt_2$$

where t_{1a} and t_{1c} are the values of t_1 at point a and point c, and t_{2c} and t_{2a} are the values of t_2 at point c and point a. It is generally simple to calculate the compression area in the grooves of the rotors by employing this principle.

3.2 Profile Functions of the Rotors

Before computing the volume curve, it is necessary to establish the profile functions for the male and female rotors, which are based on the rotor fixed frames shown in Figure 3. For example, the functions of profile segment de are for the male rotor

$$\begin{cases} x_1 = r_1 + R \cos t \\ y_1 = -R \sin t \end{cases} \quad ; \quad t_1 \le t \le t_2$$

and for the female rotor

$$\begin{cases} x_2 = r_2 - R \cos t \\ y_2 = -R \sin t \end{cases} \quad ; \quad t_1 \le t \le t_2$$

where t_1 and t_2 are the values of the parameter at the points e and d. The integration process must be based on the same coordinate frame and therefore the profile functions for the female rotor are transfered into the coordinate frame of the male rotor. The transfer relation is as follows,

$$\begin{cases} x_1 = -x_2 \cos(K\varphi_1) + y_2 \sin(K\varphi_1) + a \cos\varphi_1 \\ y_1 = x_2 \sin(K\varphi_1) + y_2 \cos(K\varphi_1) - a \sin\varphi_1 \end{cases}$$

and the function of profile segment $d_2 e_2$ becomes

$$\begin{cases} x_1 = -r_2 \cos(K\varphi_1) + R \cos(t + K\varphi_1) + a \cos\varphi_1 \\ y_1 = r_2 \sin(K\varphi_1) - R \sin(t + K\varphi_1) - a \cos\varphi_1 \end{cases}$$

Note that there are two teeth taking part in the mating process for the rotor, the leading lobe and the trailing lobe, and that the transfer relations used for them are different.

The transfer relations for the male and female rotors are as follows:

For the male rotor

$$\left| \begin{array}{cc} \cos\left(\frac{2\pi}{m_2}\right) & -\sin\left(\frac{2\pi}{m_2}\right) \\ \sin\left(\frac{2\pi}{m_2}\right) & \cos\left(\frac{2\pi}{m_2}\right) \end{array} \right|$$

and for the female rotor

$$\begin{vmatrix} \cos\left(\dfrac{2\pi}{m_1}\right) & -\sin\left(\dfrac{2\pi}{m_1}\right) \\ \sin\left(\dfrac{2\pi}{m_1}\right) & \cos\left(\dfrac{2\pi}{m_1}\right) \end{vmatrix}$$

3.3 Calculation of the Compression Area

When the lobe tip point a_2 of the female profile reaches the point H, as in Figure 3, the compression process begins. At this moment the turning angle of the male rotor is φ_{10} and the groove areas f_{01} and f_{02} for the male and female rotor are at their maximum values.

At a different moment, shown in Figure 4, there is only one point mating between two rotors and the shaded area is the uncompressed area of the grooves. This area S can be computed using the principle stated in 3.1,

$$S = \varphi \left(L_1 + L_2 + L_3 + L_4 \right)$$

where

$$L_1 = f_2 f_1 a_1 b_1 c_1 d_1 e_1' \qquad L_3 = e_1' H$$

$$L_3 = H g_2 \qquad L_4 = g_2 f_2 e_2 d_2 c_2 b_2 a_2 f_2'$$

Lines L_1 and L_4 consist of several pieces of profiles for the male and female rotors. Profile segment a_1b_1 is an example to illustrate the use of this method. This segment has a circular shape with a profile given by

$$\begin{cases} x_1 = r_1 \cos t \\ y_1 = r_1 \sin t \end{cases} \quad ; \quad t_{a_1} \leq t \leq t_{b_1}$$

and t_{a_1} and t_{b_1} are parametric values. The time derivatives are

$$\begin{cases} \dot{x}_1 = -r_1 \sin t \\ \dot{y}_1 = r_1 \cos t \end{cases}$$

The integration function is

$$f(t) = \frac{1}{2} \left(\dot{y}_1 x_1 - \dot{x}_1 y_1 \right) = \frac{1}{2} r_1^2$$

It is clear that as the parameter t changes from t_{a_1} to t_{b_1}, the integration of the function will be

$$\int_{t_{a_1}}^{t_{b_1}} f(t)\, dt = f(t_{b_1}) - f(t_{a_1}) = \frac{1}{2} r_1^2 \cdot (t_{b_1} - t_{a_1})$$

that is, the area of a part of the circle with radius r_1 and angle $(t_{b_1} - t_{a_1})$.

Similarly, the integration for profiles $f_2 f_1$, $f_1 a_1$, $b_1 c_1$, $c_1 d_1$, and $d_1 e_1$ and lines L_2, L_3, and L_4 can be calculated giving the shaded part areas as the summation of all these integrations.

Since most of the profiles used in the twin screw compressors are circles, straight lines, epicycloids or the envelope curves, it is not difficult to get the solution of the integration functions for these curves. For instance, there are 14 functions in total in the example rotor profiles and most of them can be solved analytically.

The situation when two pairs of points are mating is shown in Figure 5. The shaded area is the uncompressed area. The computation of this area is the same as discussed, however, the two contacting points must be determined first.

After the interlobe contact lines have been completed, the uncompressed area reaches the minimum and the swept area of the grooves becomes maximum. If the rotor continues to rotate, the area between the lobes can not be further invaded by the teeth of the mating rotor and the reduction of the working space in the grooves is caused only by the movement of the interlobe contact lines towards the discharge end.

The subtraction of the uncompressed area from the maximum groove area $f_{01}+f_{02}$ determines the swept area during the compression process and is shown in Figure 6.

The compression volume curve is shown in Figure 7. It is obvious from Figure 7 that the swept area is increasing from

$$\varphi_{10} \longrightarrow \varphi_{1K} = \varphi_{10} + \beta_{01} + \frac{2\pi}{m_1}$$

during which the interlobe contact lines are forming; afterwards the swept area keeps constant from φ_{1k} to t_{1z} and then it becomes smaller while the mating teeth are losing contact from t_{1k} to t_{1z}.

The integration for the lines shown in Figure 2 is the working space volume $v(\varphi_1)$, which at any given turning angle φ is

$$v(\varphi_1) = h \int f_r(\varphi_1) d\varphi_1$$

3.4 Unreturn Area

Since most of the modern screw rotors have a large wrap angle there exists a unreturn area during processing, i.e., when the suction process has finished at the inlet end of the rotor. There is still a small part of the female rotor tooth occupied in the male rotor grove and the intake volume for the supply gas becomes smaller then normal. It is necessary to deduct this part of the area from the total. The principle of area computation is the same as before.

4. FLUTE AREA FACTOR AND WRAP ANGLE FACTOR

The discharge volume of the air compressor is calculated by

$$V_d = \eta_v C_n C_\varphi n_1 L_r D_1^2$$

where the C_n and C_φ are the flute area and wrap angle factors respectively. The flute area factor is

$$C_n = m_1 (f_{01} + f_{02}) / D_1^2$$

thus, when the factor of C_n is going to be calculated, the flute area for male and female profile f_{01} and f_{02} have to be computed using the principle of integrating within the closing lines (see Section 3.1).

The angle factor is

$$C_\varphi = 1 - \frac{(\Delta V_{01})_s + (\Delta V_{01})_d}{V_{01} + V_{02}}$$

where v_{01} and v_{02} are the flute volume of the male and female rotor, which is the product of the profile flute area f_{01} or f_{02} and the rotor length l_r. The volumes

$$(\Delta V_{01})_s \quad ; \quad (\Delta V_{01})_d$$

are the occupied volume by the female rotor tooth in one male rotor working space at the suction end and the discharge end respectively, which are calculated using the method described in Section 3.1 When the occupied volume Δv_{01} has the same value at both the inlet end and the outlet end, the most efficient wrap angle factor c_φ is obtained which depends upon the wrap angle of the rotor.

5. LENGTH OF THE SEALING LINES

Since they permit the leakage of the compressed gas, the length of the sealing lines and the blow hole area are important factors effecting the efficiency of twin screw compressors. It is important to attempt to shorten the length of the sealing lines and decrease the area of the blow holes when designing a screw rotor profile set.

When calculating the length of the sealing lines, it is necessary to first determine a set of functions for the male and female profiles in accordance with the coordinate systems, and then to find the rotor lobe surface functions. The general form of such functions is

$$X = X_0(\lambda)\cos\varphi \mp y_0(\lambda)\sin\varphi$$
$$Y = \mp X_0(\lambda)\sin\varphi + y_0(\lambda)\cos\varphi$$
$$Z = K\varphi$$

where plus and minus signs represent right and left twist directions. Adding the contact condition of the two local surfaces, i.e. at each contact point only sliding movement along the tangent direction is permitted, the function representation of the contact line or sealing line between two surfaces is then obtained,

$$X = X_0(\lambda)\cos\varphi \mp y_0(\lambda)\sin\varphi$$
$$Y = \mp X_0(\lambda)\sin\varphi + y_0(\lambda)\cos\varphi$$
$$Z = K\varphi$$
$$f(\lambda, \psi, \varphi) = 0$$

The length of the sealing line can be computed from integration along the profile. The differential form is

$$ds = \sqrt{(dx)^2 + (dy)^2 + (dz)^2}$$

and the sealing line length is

$$S = \int ds$$

The total length of the interlobe sealing lines is the summation of each section of the contact line of the respective profile. For example, the length of the sealing line for the profile section ef is obtained

$$\ell_{ef} = \frac{\pi}{180}\beta_{01}\sqrt{r_1^2 + \left(\frac{T_1}{2\pi}\right)^2}$$

A more detailed presentation of this analysis can be found in Reference [4].

6. THE BLOW HOLE AREA

The blow hole is generated by the non-symetrical profiles mating, which is widely employed in the modern screw compressors for the reason of the high efficiency. there are two blow holes existing at the ends of the interlobe sealing lines. The shape of the blow holes is an irregular three-d surface and the calculation of the blow hole area is quite difficult. This calculation can be simplified by assuming that the surface is approximated by a plane, which is generally good enough for engineering design. The approximate area of the blow hole as shown in Figure 8 is

$$S_{Hba} \cong \frac{1}{2}\overline{Hb}\cdot\overline{ab}'$$

At present the most widespread method of screw rotor manufacture is single index machining (milling or grinding), although there is a certain interest in hobbing, especially for smaller rotors. The current versions of the programs support the single index process, from production of the tool to control of the completed rotor. A similar series of programs for hobbing is in preparation.

In the case of single index milling a number of tool blades of identical shape are fixed to a disk shaped tool body, which is rotated to cut one thread of the rotor at a time. The TOOL program calculates the shape of these blades. The tool axis may be placed in different positions relative to the rotor axis, and the tool blades may be placed at different angles relative to the tool axis. The coordinates of the blades depend on the setup, and some setups will give better cutting conditions and more even wear on the tool than others. In the computation of the tool coordinates the positions of the tool axis and tool blade are given by the user via the keyboard. In addition, a file of 'clearance coordinates' for the rotor is needed as input. This means that the program computes a tool, which theoretically will cut the rotor with nominal clearances applied. Along the entire length of the thread a certain profile point will be cut by a certain point on the tool blade. Coordinate points on the tool will be the points that cut the corresponding coordinate points on the rotor, and they are numbered accordingly. The program reports the cutting angles (front and side rake) at each coordinate point. The user can thus experiment to find a setup with optimal cutting conditions. To improve cutting conditions, one can also make use of two (or more) sets of blades, set alternatingly in the tool body and at different angles, each set cutting part of the profile ('zig-zag tool'). The program is flexible enough to allow the tool blades to be set at any position relative to the tool axis and cut any prescribed part of the complete profile.

Thus the TOOL program computes a tool that theoretically will cut the desired profile. For various reasons that may not be the optimal shape for the tool. For example, the tool will wear with use, and to increase the time between regrinds one may want to add metal to areas of the tool particularly subject to wear. One may also want to compensate for elastic deformations during the cutting process, or other imperfections in the machining process. This could be done by change in the clearance specification, but for several reasons this is inappropriate. The 'clearance coordinates' should describe the shape of the finished rotor, i.e., they are design data. Any changes to the theoretical tool shape needed to cut the rotor should therefore be applied directly to the tool coordinates. Consequently there is a program available to apply modifications to the tool shape which works similarly to the clearance application program.

For many reasons a program for geometric simulation of the cutting process is desirable. For example, it may be used to see the effects on the rotor profile of changing any of the parameters in the machining set up, such as the angle and distance between rotor and tool axes, the location of the tool blade relative to the tool body, etc. It can also be used to determine the theoretical shape of a rotor cut with a tool measured in a coordinate measuring machine (CMM)

using given settings. Such a program is available. In addition to what has already been mentioned the program produces 'sensitivity coefficients' for all the variables that influence the form of the rotor, *viz.* angle and distance between rotor and tool axes, rotor lead, tool blade rake angles and 'offset' as well as the actual tool coordinates. These coefficients are given for each point of the rotor, and have the following meaning: To see the effect on the rotor profile of *small* changes in all the variables, multiply each sensitivity coefficient by the change in the corresponding variable and add together. The result is the distance the rotor point moves in the direction of the profile normal at the point (presumably the point will also move parallel to the profile, but for small changes this should have very little effect). One use of the sensitivity coefficients is to determine which factors influence rotor form the most in each point; this could be used to determine which points it would be most effective to check, to find out if a certain setting is within acceptable limits.
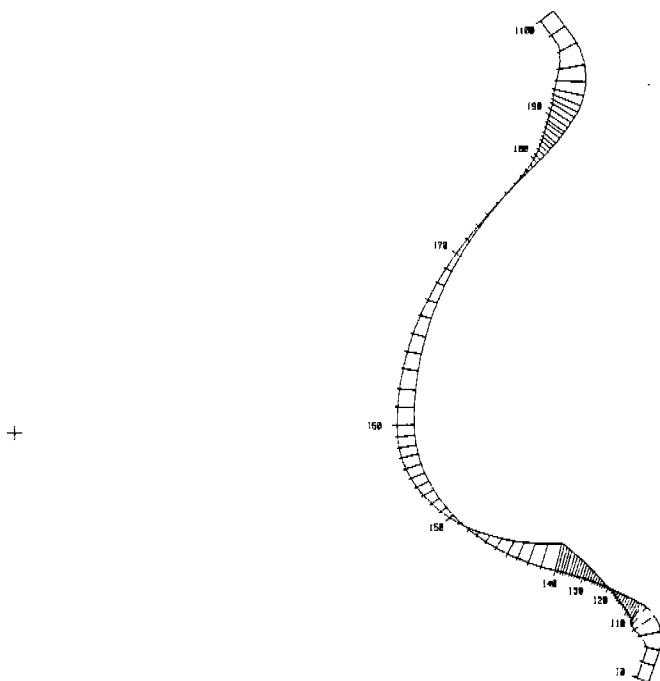
Plot of output from cutter simulation program,
showing effect of a slight increase of infeed and cutting angle

The program is also able to use the sensitivity coefficients to determine exactly how much to change each setting to correct a given, measured rotor shape. The measured coordinates are read from a file, and the program then produces a list of changes to be made in settings to get as close as possible to the desired rotor shape. The program then calculates the actual shape of rotor to be expected from the new settings, and the residual errors in the normal direction. The calculation can be done allowing only certain variables to be changed. For example, normally one would not want to change the pitch of the rotor! It might also be of interest to see how close one can get when changing only the machine tool settings without interfering with the tool shape.

In all these calculations only purely geometric factors are taken account of. This means that any elastic deformations in the setup will not be accounted for. This should not matter, however, assuming that profile errors caused by such factors are small. The reason is that the program calculates corrections from a measured profile, so that errors from such sources should be automatically compensated for, as far as is physically possible.

When a rotor is carefully checked in a coordinate measuring machine the output will be a massive amount of raw data. It would normally consist of deviations from the nominal coordinates of the identification points in all the threads and in several cross sections along the rotor. These need to be interpreted to indicate how well a particular rotor corresponds to the theoretical shape. The rotor checking program takes as input a file from the CMM and calculates lead and indexing errors. It draws a diagram to indicate how the rotor size varies along its length, to show errors like tapering or barrel shape, and other diagrams to show the deviation from the theoretical profile in each thread and each cross section.

Routine checking of rotors during manufacturing must of course be done in a less time consuming way than the coordinate measuring machine offers. A simple pairing stand and feeler gauges can be used to give a rough indication of the quality of a rotor pair. To make room for the feeler gauges one would like to set up the pairing stand with an increased center distance, and also always measure with contact between the rotors on the flanks opposite to those between which one is checking ('flip-flop measuring'). There is a program to calculate the permissible gaps at various points along the profile, given the center distance of the pairing stand and the tolerances in intermesh clearances when the rotors are mounted with nominal center distances.

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