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OPTIMISING THE BACK PRESSURE PORT
OF A SCROLL COMPRESSOR

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ABSTRACT
This study attempts to search for the combination of the back pressure port size and its location which results in minimum mechanical losses and yet providing sufficient axial sealing. Theoretical study employing a mathematical model coupled with numerical optimisation algorithm predicted that a reduction of 9% in mechanical losses can be achieved. The main cause of gain that resulted in such a significant mechanical loss was found to come from the consequence of optimising the location, and not the dimensions of the back pressure port.

INTRODUCTION
Scroll compressors are geometrically simple and rugged machines with few moving parts. In its simplest form, it uses two identical spiral plates, the fixed and the orbiting scrolls to achieve compression effects. The self adjusting back pressure mechanism was introduced into the design of the scroll compressor for the purpose of providing axial sealing and hence minimising leakage of working medium. A good design should accomplish this at a minimum mechanical losses. The basic components of this mechanism are back pressure chamber and the back pressure port, see Fig. 1. The back pressure chamber is situated behind the base plate of the orbiting scroll while the back pressure port is located on the base plate of the orbiting scroll. During the operation, the back pressure port serves as a communicating channel between the compression chamber and the back pressure chamber. This allows the back pressure chamber to be charged with gas pressure and hence causes the orbiting scroll to axially seal against the base plate of the fixed scroll plate.

This paper illustrates how the location and the size of the back pressure can be optimised using mathematical modelling and optimisation algorithm. The study may also be used to assess what design modifications may be made in the interest of more cost effective manufacture without undue sacrifice of hard won performance improvements.

MATHEMATICAL MODELLING
With the advent in the computer technology, comprehensive mathematical models of real physical systems may be formulated with relative ease. But for the case of the optimisation studies, which may involve many repeated model evaluations [1,2] during the search for an optimum, it is advisable to employ a simple model which operates within the required practical tolerance bounds.

Geometrical Model
The geometrical model describes the variations of the volume of the working chamber of the machine in relation with the rotational angle \( \theta \), i.e.

\[
V(\theta) = f(r, H, P, t, \alpha)
\]

(1)

Where \( r \) is the radius of generating circle, \( H \) is the height of the scroll, \( P \) is pitch, \( t \) is thickness of the scroll wraps and \( \alpha \) is the generating angle. For the purposes of optimisation study, the orbiting and the fixed scroll are assumed to be geometrical identical. In the present model, the number of compression chamber is taken as 3.


**Thermodynamics Model**

In the thermodynamics description of the compressor, the processes occurred in the compression and back pressure chambers are assumed adiabatic. Leakage flows through these working chambers and the neighbouring chambers are considered. The pressure, $p$, and the temperature, $T$, in the compression chamber assuming real gas simulation may be given by:

$$
\frac{dp}{d\theta} = \left( \frac{\partial p}{\partial T} \right)_T \frac{dT}{d\theta} + \left( \frac{\partial p}{\partial v} \right)_T \frac{dv}{d\theta}
$$

(2)

$$
\frac{dT}{d\theta} = \frac{1}{m} \left[ \frac{d(h_i - h)}{d\theta} - \frac{d(h_o - h)}{d\theta} \right] - \left[ \left( \frac{\partial h}{\partial T} \right)_T - \nu \left( \frac{\partial p}{\partial T} \right)_T \right] \frac{dv}{d\theta}
$$

(3)

$$
\frac{dm}{d\theta} = \frac{dm_i}{d\theta} - \frac{dm_o}{d\theta} - \frac{dm^*}{d\theta}
$$

(4)

Where $h$ is the refrigerant enthalpy, $v$ is the specific volume, $m$ is the refrigerant mass in the working chamber, suffices $i$ and $o$ represent inlet and outlet quantities respectively while $^*$ denotes in back pressure chamber.

Similar equations may be derived for the back pressure chamber and these are given below:

$$
\frac{dp_b}{d\theta} = \left( \frac{\partial p_b}{\partial T_b} \right)_{v_b} \frac{dT_b}{d\theta} + \left( \frac{\partial p_b}{\partial v_b} \right)_{T_b} \frac{dv_b}{d\theta}
$$

(5)

$$
\frac{dT_b}{d\theta} = \frac{1}{m_b} \left[ \frac{d(h^*_b - h_b)}{d\theta} \right] - \left[ \left( \frac{\partial h_b}{\partial T_b} \right)_{v_b} - \nu_b \left( \frac{\partial p_b}{\partial T_b} \right)_{v_b} \right] \frac{dv_b}{d\theta}
$$

(6)

$$
\frac{dm_b}{d\theta} = \frac{dm^*}{d\theta}
$$

(7)

where $\frac{dm^*}{d\theta}$ is mass exchange between compression chamber and the back pressure chamber.

The working fluid is HFC134a, and it is simulated using a real gas equation of state. A detailed simulation of the working fluid is illustrated by Ooi et al [3].

**Mechanical Model**

The friction loss in a compressor is mainly caused by three rubbing parts: the shaft, the scroll plate and the Oldham ring. Only the friction due to scroll plate and Oldham ring are considered here. The force balance can be established once the pressures in the working chamber are known. These forces [4] may be given as:

$$
M_B(\theta) = f(F_0, F_t, F_1, F_2, F_m)
$$

(8)

$$
F_0(\theta) = (-M_m/R_{th} - F_t + F_{BP})/2
$$

(9)

$$
F_{cl}(\theta) = (M_m/R_{th} - F_t + F_{BP})/2
$$

(10)

Where $M_B(\theta)$ is the friction moment on the pin of the orbiting scroll, $F_0(\theta)$ and $F_{cl}(\theta)$ are forces acting on the thrust bearing to balance combination of back pressure force $F_{BP}$ the axial gas force $F_t$ and over-turning moment $M_m$. $F_0$ is the tangential gas force, $F_m$ is tangential inertia force. $F_1, F_2, F_3, F_4$ are the forces acting on the Oldham ring. $R_{th}$ is radius of thrust bearing.
Thus power loss at the orbiting scroll and Oldham ring are given as below:

\[
P_B = \frac{n}{60} \int_0^{2\pi} M_B(\theta) d\theta
\]

\[
P_{\text{plate}} = \frac{n}{60} \int_0^{2\pi} \mu_c(F_{c1}(\theta) + F_{c2}(\theta))r_o d\theta
\]

\[
P_{\text{oldham}} = \frac{n}{60} (4 \int_0^{\pi} \mu([|F_1| + |F_2|] \cos \theta + ([|F_3| + |F_4|] \sin \theta)) r_o d\theta
\]

Hence the total power loss due to friction is the sum of Eqs. (11) to (13):

\[
P_{\text{loss}} = P_B + P_{\text{plate}} + P_{\text{oldham}}
\]

Eqs. (2) to (7) were integrated simultaneously using 4th order Runge-Kutta numerical integration technique to establish the pressure, temperature and mass in the working chamber. The friction model is then evaluated to give the friction power loss of the machine as described by eq. (14).

The simulation study allows the performance of the compressor to be assessed and the effects of the changing of parameters on the performance may be observed. The use of a simulation model together with the optimisation algorithm allows the selection of design parameters which gives an overall optimum performance of a machine under a prescribed specification. In general, an optimisation study may be symbolically represented by:

\[
\text{minimise: } F(x) = f(x_1, x_2, \ldots, x_N)
\]

Where \( N \) is the number of free variables. In any optimisation study, a careful choice of independent variables is crucial. The parameters that have little influence on the outcome of the optimisation should be discarded. Any reduction in the numbers of independent variables brings benefits by ways of reduction in the computational time required as the number of function evaluation is reduced.

subjected to:

\[
(i) \quad L_E(x_i) \leq E_i \leq H_E(x_i), \quad i = 1, 2, \ldots, N
\]

\[
(ii) \quad L_G(x_j) \leq G(x_j) \leq U_G(x_j), \quad j = 1, 2, \ldots, M
\]

\[
(iii) \quad L_I(x_k) \leq I(x_k) \leq U_I(x_k), \quad k = 1, 2, \ldots, L
\]

Eqs. (16) to (18) represent constraints that applied to the problem. Eq. (16) represents limits imposed on the free variables. Eq. (17) represents limits imposed on the geometrical constraints, \( G(x_j) \). The geometrical constraints are usually functions of one or more of the free variables. They ensure that the outcome of the optimisation studies to be within the feasible geometrical design. Eq. (18) represents implicit constraints that the outcome of the particular calculation may have to satisfy.

In present study, the Constrained Simplex or Complex optimisation technique was employed. The method has been used extensively since its development by Box because of its basic simplicity and adaptability to suit individual optimisation problems [1,2].

COMPRESSOR OPTIMISATION

Compressor designs were optimised for their mechanical performance under a prescribed operating condition, while meeting a required flow capacity. The objective function was chosen to be the total frictional dissipation of the machine as described by Eq. (14).

The independent variables that are chosen to vary with optimisation search are polar angle of back pressure port position, \( \beta \), and the back pressure port diameter, \( e \), as well as back pressure correction distance \( \Delta \). Their respective limits are:

\[
0^\circ \leq \beta \leq \theta^\circ - 40^\circ
\]
The thickness \( t \) of the scroll plate is made greater than the size of the back pressure port to prevent direct leakage between the adjacent working chambers through the back pressure chamber. This condition was reinforced by introducing \( e \leq t/2 \) as the upper limit of the back pressure port radius, \( e \) as given by eq. (20).

The implicit constraints which was introduced is the one that assures the circumference of the edge of the orbiting scroll plate to be always in contact with the bottom plate of the fixed scroll. This positive contact is seen if there exists a positive value of the contact force, i.e.

\[
F_{t1_{\text{min}}} > 0
\]  

The above optimisation was carried out at the constant machine capacity and at given operating conditions.

RESULTS AND DISCUSSION

Fig. 2 shows the variations of the normalised objective function (where the initial design is taken as 1), i.e. the total friction loss. Since the study is to minimise the friction loss, it can be seen that during the whole search process, the friction loss decreases as the search approaching the optimum. The computation converged at the optimum point which shows a 9% reduction in the total friction loss, as compared with the initial design.

Fig. 3 shows the variation of the minimum acting force of the thrust bearing, \( F_{t1_{\text{min}}} \) which was used as the implicit constraints during the search. It is obvious that to reduce the friction loss of the compressor, \( F_{t1_{\text{min}}} \) must be reduced to a minimum positive number. Figs. 4, 5 and 6 show the variations of the three free variables which are varied during the optimum search. The results show that the converged optimum \( \beta \) is 137.4°, as compared with the initial value of 208°. However the optimum value for radius of the back pressure port was not far away from its initial design. The initial design port radius was 1.5 mm and the optimum value is 1.46 mm. The variations shown in Fig. 5 suggested that the range of port radius \( e \) imposed in the study is not very sensitive to the mechanical loss of the machine. The correction distance \( \Delta \) varied from 0.1 mm to 0.33 mm from initial to final optimum values respectively. Again, the results suggested that the range of \( \Delta \) employed in the search is less sensitive to the mechanical loss. Fig. 7 and 8 show that the reaction forces decrease as would be expected.

CONCLUSIONS

From mechanical loss consideration, the optimum design suggested the position of the back pressure plays an important part in minimising the mechanical loss of the machine. The study showed that with properly selected position of back pressure port reduces the mechanical loss by minimising the thrust bearing forces. Experimental programme may be carried out to validate the theoretical prediction.

REFERENCES

Figure 1(a): Schematic of scroll compressor

Figure 1(b): Location of back pressure ports on orbiting scroll plate.

Figure 2: Variation of objective function
Figure 3: Variation of minimum thrust bearing force

Figure 4: Variation of port polar angle.

Figure 5: Variation of port diameter.

Figure 6: Variation of port position correction distance.

Figure 7: Variation of thrust bearing force $F_{tl}$

Figure 8: Variation of thrust bearing force $F_{t2}$