Geometrical Optimisation of the Revolving Compressor

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GEOMETRICAL OPTIMISATION OF THE REVOLVING VANE COMPRESSOR

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ABSTRACT

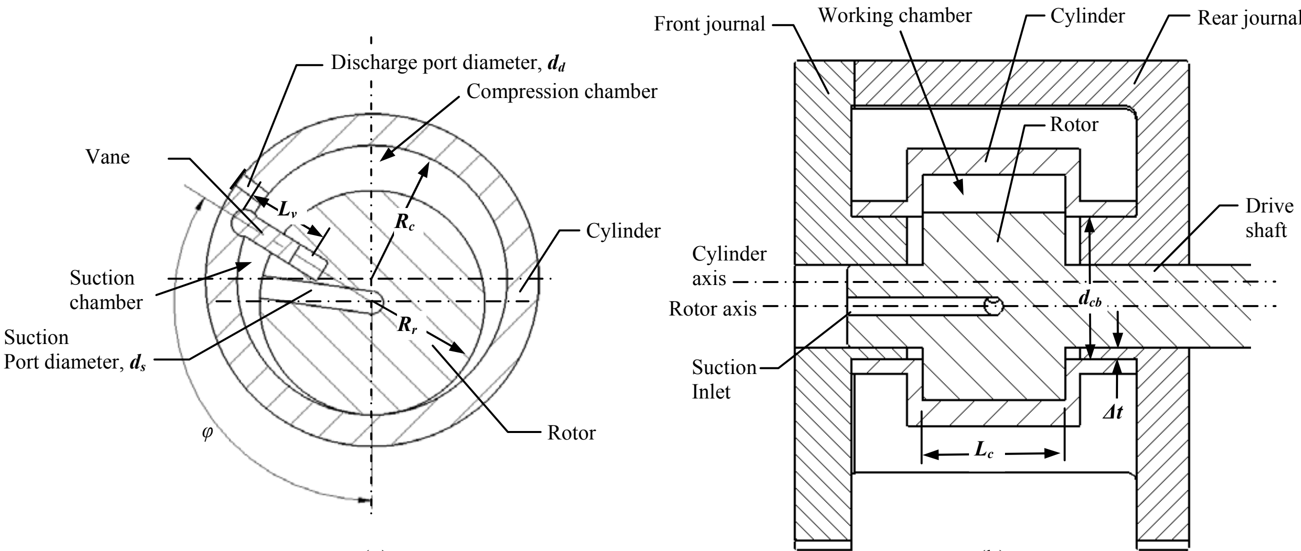
The newly designed compressor, namely Revolving Vane Compressor has been introduced [1]. As compared with conventional rolling and sliding vane compressors, such a design exhibits significant mechanical benefits. This paper shows how the geometry of this compressor can be further optimized to enhance its mechanical performance. A constrained multivariable direct search optimization technique has been linked with the mathematical model to search for a set of combination of design dimensions that produces the highest coefficient of performance (COP), under preset design constraints and operational conditions. Theoretical results suggested that an increase of 4 % in COP is possible. This is achieved mainly by having a thinner and longer compressor as opposed to a shorter and fatter design.

1. INTRODUCTION

Rotary compressors are widely used in air and refrigeration industries. They possess many good inherent characteristics such as fewer parts, low vibration and cheaper to produce over the reciprocating counterpart. However, they possess high mechanical losses due mainly to high friction of the moving parts. Thise high friction loss is the consequence of high relative speeds among the moving parts, in particular those rub against the stationary components. For example, the sliding of the vane tip on the stator and between the rotor and the stator end faces of a sliding vane compressor. Similar situations also occur in the rolling piston compressor. In order to minimize the friction loss by minimizing the relative speed between these rubbing surfaces, the casing of the compressor in the revolving vane compressor (RV) is set in motion. This has resulted in a significant reduction in frictional loss, the details of which are shown in paper C046 [1] presented in this conference. This paper presents the theoretical analysis in an attempt to further enhance the compressor performance through geometrical optimization. By linking the mathematical model with a constrained multivariable direct search optimization algorithm, it is possible to obtain a set of combination of geometrical dimensions that produces optimum compressor performance. The details of which are shown in this paper.

2. COMPRESSOR DESIGN

In its basic form, the Revolving Vane (RV) compressor comprises of a vane, a cylinder and a rotor, as shown in Figure 1. During operation, the vane, with one end pivoted in the inner wall of the cylinder and with the other end sliding radially inward and outward in the vane slot. The latter is machined in the rotor. The rotor rotates about the driving axis of the prime mover and drives the cylinder through the pivoted vane. As the axes of the rotation for the rotor and the cylinder are offset, there exists crescent shape chamber volume which varies continuously in size and hence forms the complete compressor cycle. The performance of the compressor depends largely on the balance of the thermodynamic and dynamic effects, which is mainly governed by the selection of the geometrical dimensions of a particular design.



(a) (b)

Figure 1: (a) Front sectional view of RV compressor; (b) Side sectional view of RV compressor with journal supports shown

During the design process, once the capacity of the compressor has been specified, some geometrical dimensions could be identified as design variables and their relative values can be varied to improve the performance of the compressor. In this study, some 6 geometrical dimensions have been selected as design variables and these are: cylinder radius R_c, rotor radius R_r, cylinder length L_c, minimum thickness of the cylinder bearing Δt, suction port diameter d_s and discharge diameter d_d. Hence the performance of a compressor can be written as:

$$COP = f(R_c, R_r, L_c, \Delta t, d_s, d_d) \quad (1)$$

The notations used in eqn (1) are shown in Figure 1. Refer to ref [1] for the details of the mathematical model. In this paper, a geometrical optimization study for a compressor is carried out. The compressor in question is designed for specifications shown in table 1.

Table 1: Compressor specifications

Volumetric Displacement	32.5 cm^3	Suction pressure	0.63 MPa
Cooling capacity	5.4 kW	Discharge pressure	2.17 MPa
Motor speed	2875 rev/min	Working fluid	R22

3. OPTIMIZATION STUDY

The computer has been used extensively in assisting the design of compressors. It is now relatively easy to implement computerized optimization procedures to carry out an optimization study for a compressor design. This is done by linking the simulation model with the optimization algorithm which allows for the selection of design parameters that produces optimum compressor performance under prescribed operational conditions within preset design constraints. In present study, a multi-variable direct search constrained optimization technique developed by M.J. Box [2] was employed.

In general, an optimization study may be mathematically represented by:

Optimise: $$F(x) = f(x_1, x_2, ..., x_N) \quad (2)$$

Subject to:

$$L_E(x_i) \le E(x_i) \le H_E(x_i), i = 1,2,......N \quad (3)$$

$$L_G(x_j) \le G(x_j) \le H_G(x_j)\ j = 1,2,......,M \quad (4)$$

$$L_I(x_k) \le I(x_k) \le H_I(x_k)\ k = 1,2,......,L \quad (5)$$

where subscripts *E*, *G* and *I* represent the explicit, geometrical and implicit constraints respectively while *L* and *H* represent lower and upper limits, and *i*, *j* and *k* are the number of explicit, geometrical and implicit constraints respectively.

The Complex optimization method is implemented with the following steps:

1. A feasible starting point is picked that satisfies all geometrical, explicit and implicit constraints. K-1 additional points are generated from pseudo-random numbers and constraints for each of the independent variables:
2.

$$x_{i,j} = L_E(x_i) + r_{i,j} \times (H_E(x_i) - L_E(x_i)) \qquad i = 1,2\cdots\cdots, N;\ j = 1,2\cdots\cdots, K-1 \quad (6)$$

These additional points and the starting point together are called the original Complexes. In the above, N is the number of the explicit free variable constraints and K=N+1; $L_E(x_i)$ and $H_E(x_i)$ are the lower and upper constraints of the free variables respectively; $r_{i,j}$ is the pseudo-random number between 0 and 1. The selected points satisfy the explicit constraints of the free variables but may violate other constraints.

3. If any geometric or implicit constraint is violated, the trial point is moved half way towards the centroid of the remaining points:

$$x^{n+1}{}_{i,j} = (x^n{}_{i,j} + \bar{x}_{i,c})/2 \qquad i = 1,2,\cdots\cdots, N \quad (7)$$

where *n* is the iteration number and the coordinates of the centroid of the remaining points, $\bar{x}_{i,c}$ are defined by

$$\bar{x}_{i,c} = \frac{1}{K-1}\left[\sum_{j=1}^{K-1} x_{i,j} - x_{i,j}(old)\right] \qquad i = 1,2,\cdots\cdots, N \quad (8)$$

This process is repeated until all the constraints are satisfied. For a highly constrained optimization problem, especially when the constraints are non-linear, the setting up of feasible initial complexes may be very difficult.

4. Evaluate the objective function at each point. The point having the worst objective function value is reflected by a reflection factor of α , $\alpha = 1.3$ has been used, along the line linking the replaced point and the centroid of the remaining points. The resulting new point is:
5.

$$x^{n+1}{}_{i,j} = (1+\alpha)\bar{x}_{i,c} - \alpha x^n{}_{i,j} \quad (9)$$

Here, $x^n{}_{i,j}$ is the worst point and $\bar{x}_{i,c}$ is the calculated centroid of the remaining points. To take more advantage of information about the search area that has already been explored and to increase the effectiveness of the search, the centroid is calculated by weighting each point selected according to its objective function value. The following equation is used:

$$\bar{x}_{i,c} = \frac{1}{K-1}\left[\sum_{j=1}^{K} x_{i,j}\left(\frac{F_j - F_{worst}}{F_{best} - F_{worst}}\right)\right] \qquad i = 1,2,\cdots\cdots,N \tag{10}$$

6. If, after the reflection, an explicit constraint is violated, the point is moved inside the constraint boundary by a factor of δ.
7. The objective function is evaluated only if the point satisfies all the explicit and implicit constraints.
8. The newly obtained objective function value is then checked for any improvement. If the new point shows no improvement, it is moved halfway towards the centroid of the remaining points. However, if 5 successive moves towards the centroid do not yield any improvement in the objective function value, the point is moved halfway towards the best complex. This usually gives an improvement.
9. Convergence is assumed when after a specified number of consecutive successful iterations, the objective function at a new point which satisfies all the constraints lies within a pre-set tolerance of the value of the objective function of the best point.

The search could then be repeated with a different set of random numbers. The results obtained should be compared with those obtained in the previous search to confirm the existence of a global optimum.

4. COMPRESSOR OPTIMIZATION

In this paper, the objective is to search for the combination of geometrical dimensions of the rotary compressor with the maximum COP possible under a given compressor swept volume and at given operational conditions. Mathematically, this is defined as:

$$\text{Maximise } F_{obj} = f\,(R_c, R_r, L_c, \Delta t, d_{cb}, d_s, d_d) \tag{11}$$

In this study, the objective function is the COP, and hence the optimization routine will search for the dimensions in the direction that maximizes the cooling capacity and at the same time minimize the losses. The latter includes frictional, suction and discharge losses. Hence,

$$F_{obj} = COP \tag{12}$$

Table 2 tabulates the values of the explicit constraints. These are chosen to be the range of dimensions of interest where the final design is expected to be found. The geometrical constraints are also specified to ensure that the final geometries are feasible and it is shown in Table 3. In this case, it is to ensure that the length of the vane must not exceed the diameter of the cylinder, since part of the vane is reside in the rotor and part of it in the inner cylinder wall. The value of Δ can be chosen to be an extra allowance for the vane length. In this study, an implicit constraint is also specified and is shown in Table 4. It is to ensure that the capacity of the compressor remains within the required range. This is reinforced by specifying a fixed swept volume and determining the length of the cylinder using expression (13). Since the swept volume for the compressor is fixed at $32.5\text{x}10^{-6}$ m^3, the length of the cylinder is thus calculated by:

$$L_c = \frac{32.5\times10^{-6}}{\pi\left(R_c^{\,2} - R_r^{\,2}\right)} \tag{13}$$

For the case of the RV, the cylinder is also rotating. The diameter of the cylinder bearing, d_{cb} is given by:

$$d_{cb} = 2\times\left(R_c - R_r\right) + d_{sht} + \Delta t \tag{14}$$

where the Δt is the minimum thickness of the cylinder bearing diameter and d_{sht} the rotor shaft diameter. No other constraints are needed to be specified.

Table 2: Explicit constraints

i	L_{Ei}	E_i	H_{Ei}	Units
1	18 ≤	R_r ≤	35	mm
2	0.75 ≤	R_r / R_c ≤	0.95	
3	18 ≤	L_v ≤	30	mm
4	3 ≤	Δt ≤	10	mm
5	14 ≤	d_s ≤	16	mm
6	10 ≤	d_d ≤	20	

Table 3: Geometrical constraints

j	L_{Gj}	$G(x)_j$	H_{Gj}	Units
1	0	< L_v <	$2 \times R_c + \Delta$	mm

Table 4: Implicit constraints

k	L_{Lk}	$I(x)_k$	H_{Lk}	Units
1	4.5	≤ Q_{ref} ≤	5.5	kW

5. RESULTS AND DISCUSSION

Results of the optimisation study are shown in figures 2 to 5 and tables 5 to 6. Figure 2 shows the variation of the normalized objective function during the search for an optimum. The normalized value of the objective function is obtained by dividing the objective function value with the value of the initial design. The figure shows that the highest objective function value is about 4 % above the initial design. Figure 3 shows that it took more than 400 model executions of which only about 130 are feasible, which is equivalent to about 5 hours of computational time on a 3 GHz Pentium IV PC.

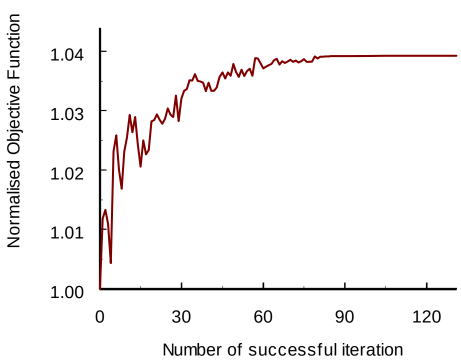


Figure 2: Variation of objective function

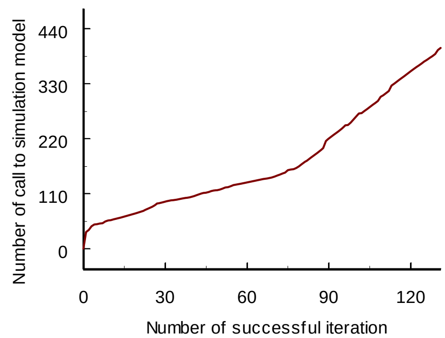


Figure 3: Variation of number of model execution

Figures 4 and 5 show the variation of the cylinder radius and length during the optimisation search. It can be seen that as the radius of the cylinder gets smaller, the cylinder lengthens. This is so as to keep the swept volume constant. This combination of dimensions gives better compressor performance. The variations of other dimensions are not shown here but their initial and final values are tabulated for ease of comparison. Table 5 show the comparison between the initial and the optimized dimensions. The ratios of the two are also shown. For a given swept volume, it can be seen that a thinner and longer compressor out performs the fatter and shorter design. The result also shows that the longer vane was used in the optimised compressor. This is because a longer vane helps to reduce the contact forces between the vane and the slot. The larger suction and discharge ports are also used in the optimised compressor as bigger port sizes help to improve the performance of the compressor by reducing the flow losses. Table 6 shows the corresponding comparison for the compressor performance parameters, where the various power required and frictional losses are shown. It can be seen that the power required and losses are all reduced and the most significant reduction comes from the lower frictional loss.

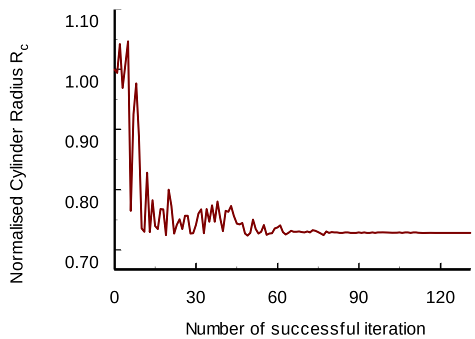


Figure 4: Variation of cylinder radius

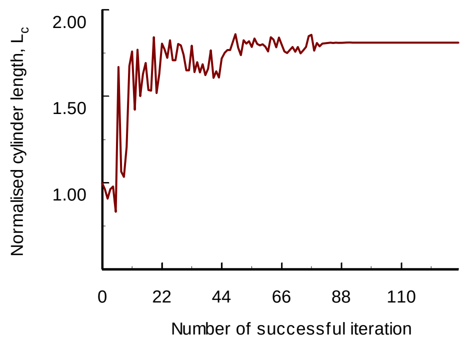


Figure 5: Variation of cylinder length

The latter reduces a significant 25% from the substantial initial amount of 80 watts. This has undoubtedly contributed significantly to the improvement of the compressor performance.

Table 5: Initial and optimized dimensions

&Parameter	R_r	R_c	L_c	L_v	Δt	d_s	d_d	d_{cb}	L_c/D_c	R_r/R_c
Initial value	25.00	30.00	38.00	21.00	6.00	15.00	10.00	38.00	0.633	0.83
Optimized value	18.58	22.35	67.74	29.98	3.03	17.96	19.98	32.57	1.516	0.83
*Ratio	0.74	0.745	1.783	1.43	0.51	1.20	1.998	0.857	2.393	0.998

&: all length dimensions are in units mm.

Table 6: Performance parameters

Parameter	COP	Q_{ref} (W)	Motor Power (W)	Indicated Power (W)	Mechanical Power (W)	Suction Loss (W)	Discharge Loss (W)	Friction loss(W)
Initial value	3.049	5491.99	1801.49	1325.11	1405.16	7.30	17.90	80.05
Optimized value	3.166	5499.25	1736.72	1293.99	1354.64	3.43	3.96	60.65
*Ratio	1.039	1.001	0.964	0.977	0.964	0.471	0.221	0.758

*: optimised value/initial value

5. CONCLUSION

In this paper, a geometrical optimization study has been carried out to optimize the performance of a revolving vane compressor. Some 6 design variables have been identified and used as free variables during the search for an optimum. Results of the optimization suggest that for a given compressor swept volume, a thinner and longer compressor out performs a fatter and shorter design. The results also suggest that through proper selection of design variables, 4% improvement in COP is possible. All this improvement comes from the reduction in suction, discharge and frictional losses, with the greatest contribution comes from the 25% reduction in frictional loss.

NOMENCLATURE

COP	coefficient of performance	
D_c	cylinder diameter	(m)
d_s	suction port diameter	(m)
d_d	discharge port diameter	(m)
d_{cb}	cylinder bearing diameter	(m)
L_c	cylinder length	(m)
L_v	vane length	(m)
R_r	rotor radius	(m)
R_c	cylinder radius	(m)
Q_{ref}	cooling capacity	(W)
Δt	minimum thickness of the cylinder bearing diameter	(m)
Δ	extra allowance for upper limits of the vane length	(m)

REFERENCES

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