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GEOMETRICAL OPTIMIZATION OF RADIAL AND NON-RADIAL SLIDING VANE AIR COMPRESSORS.

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ABSTRACT

This paper presents results of analytical studies carried out to assist in the production of sliding vane air compressors. By linking a mathematical model with an optimization algorithm it was possible to predict a combination of 13 geometrical variables that would produce for a given machine capacity the maximum air throughput per unit power input. A constrained multi-variable direct search technique was used in which the 13 geometric variables and a mixture of 27 explicit and implicit constraints were accommodated. Theoretical studies on a 6 radially and non-radially disposed vane compressor indicated that a 5.5% improvement in specific capacity could be achieved.

INTRODUCTION

Owing to their compactness and geometric simplicity, rotary sliding vane compressors are widely used in air compression, especially in refrigeration industries. In their simplest form, see figures 1 and 2, they comprise a cylindrical rotor mounted between end-plates and positioned eccentrically within a circular stator. The rotor has a number of radial or non-radial slots machined along its length into which closely fitting vanes are located.

When the rotor is spun about its longitudinal axis centrifugal forces cause the vanes to move radially outwards in the slots until the vane tips touch the stator surface. The combined rotary and sliding motions result in the formation of a number of variable volume cavities (cells) whose size depends on the angular positions of a pair of adjacent vanes and the dimensions of the mechanism. Provision of appropriately placed suction and discharge ports allows the device to function as a positive displacement compressor with a built-in volume compression ratio. The overall performance of a given machine depends on a balance of frictional effects (vane tip/stator surface, vane ends/stator end-plates, vane flanks/vane slots, rotor sides/stator end-plates and finally rotor shaft/bearings), compression and cell to cell leakage effects (at vane tips, vane flanks and between vane sides and end-plates), suction and discharge port effects.

Over the years since the basic form of the machine was conceived many experimental and theoretical considerations have aided in its design and development. Significant performance improvements were conceived by the advent of oil or liquid injection [1] during the compression process. This feature not only reduced frictional effects, but also promoted better sealing of the cells formed by the rotor, the stator and the sliding vanes. Oil injection did however, necessitate the provision of oil injection and oil separation facilities.

This paper illustrates how a theoretical study could be used to predict performance characteristics by pointing the way to the selection of the most effective combination of design variables and thereby avoiding the need for extensive ad hoc testing. Besides, the analysis may also be used to assess the essential design modifications that may be made in the interest of more cost effective manufacture without undue sacrificc of hard won performance improvements.

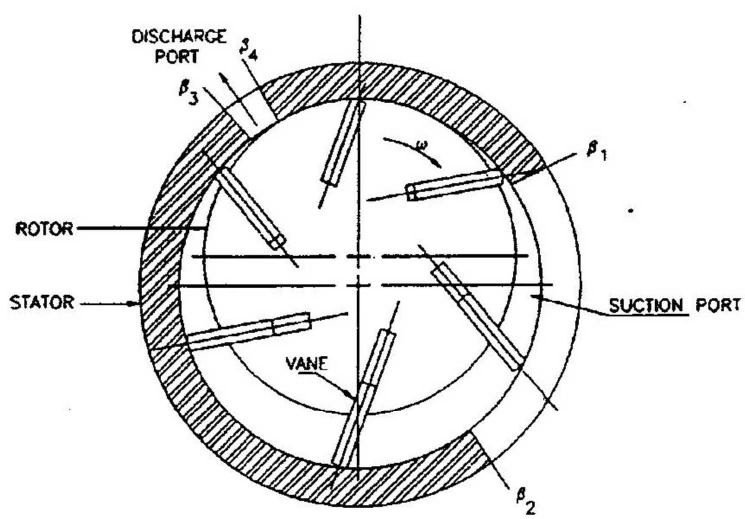

Fig. 1(a): Rotary Sliding Vane Compressor

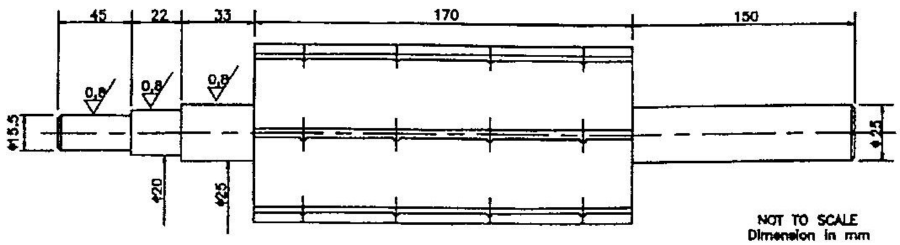

Fig. 1(b): A Vane Compressor Rotor.

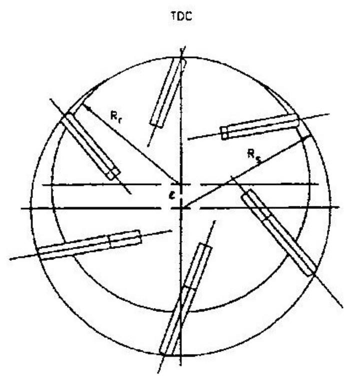

Fig. 2(a). Rotor-stator

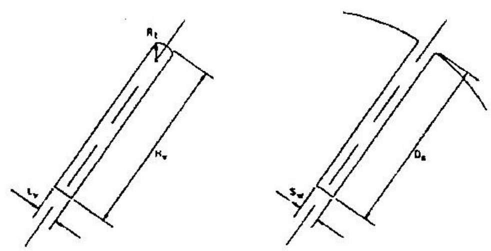

Fig. 2(b). Vane and Vane Slot

THEORETICAL APPROACH

In order to implement this study, it is necessary to consider the following;

(a) A mathematical model of the compressor to be developed.

(b) Validation of the model by predicting operational behaviour and performance characteristics of existing machines.

(c) A test on the role of size, performance, reliability, economic, manufacturing was done so as to define optimum parameters which conform to an optimum machine.

(d) To establish rational relationships for obtaining optimum or 'best' design.

(e) A suitable optimization algorithm chosen to identify the optimum design.

(f) Developing an optimum prototype and validate experimentally the predicted performance characteristics.

Successful results of steps (a) through (f) could help establish a systematic approach to an optimum design. This could be succeeded by an implementation of mass production through market oriented new product development.

MATHEMATICAL MODEL DEVELOPMENT

The modelling of sliding vane compressors forms part of an overall compressor model development. All forms of positive displacement compression machinery were a main concern of mathematical modellers. Maclaren [2], especially reciprocating, screw, vane [3,4,5,6,7,8], rolling piston, scroll and other novel geometrical forms have been studied.

Manufacturers used mathematical modelling to assist them in the production of machines which operate efficiently, reliably and at higher rotational speeds than may have initially been contemplated. Modelling has benefitted for more than three decades of computer and computational development so that today geometrical, dynamic, stress and both heat transfer and fluid flow effects could be studied with comparative ease.

However, such models should be as simple as possible, while assuring a good capability to identify significant aspects of any design. This is particularly true as demonstrated in this work. In the case of the sliding vane compressor, the model must establish geometric, dynamic, fluid mechanics and heat transfer effects.

Geometric Considerations

A representation of the cell volume V has been suggested by Chang [9] which makes use of 12 variables as seen from equation (1),

$$V = f(R_s,R_r,\varepsilon,L_r,H_v,D_s,t_v,S_w,R_t,N_v,C_s,\theta) \tag{1}$$

The meaning of some of these terms is illustrated in figure 2. In order to optimise the vanes, they are assumed to have semi-circular profile tips where,

$$2R_t = t_v \approx S_w \tag{2}$$

and by implication the vane thickness and slot width are equal.

Vane Dynamics

In a vane dynamic analysis assumption made is that centrifugal forces cause the vane tip surface to remain in contact with the stator surface. It can be seen from figure 3 that by postulating the existence of a number of vane position modes, it is possible to determine the normal contact and friction forces acting upon the vanes, the rotor and the stator and hence determine the friction energy component of the compressor driving power [10]. Optimization analysis is simplified further if a Coulomb frictional model is employed while using a consistent coefficient of friction with an equivalent value deduced from hydrodynamic lubrication theory. Since vane dynamic analysis depends on the pressure distributions along the vane surfaces, it must therefore be coupled to the thermodynamic and fluid flow analyses so as to calculate pressure, temperature and phase distributions of the working fluids which occupy a cell volume.

Thermodynamic Processes and Analysis

In the case of an oil injected air compressor the cells will contain air, oil liquid and oil vapour. The air and oil vapour should be assumed to be a homogeneous mixture occupying part of the cell volume not occupied by oil liquid. That is;

$$V_a = V_{ov} = V_c - V_{ol} \tag{3}$$

The cell pressure Pc is the sum of the partial pressure of the air and the saturation vapour pressure of the oil at a given temperature.

$$P_c = P_{ol} = P_a + P_{ov} \tag{4}$$

Assuming that thermodynamic equilibrium exists between the air and oil vapour then;

$$T_a = T_{ov} \tag{5}$$

Applying the law of the conservation of energy, the thermodynamic properties of each phase may be defined by the following differential equations;

For the air phase:

$$\frac{dQ_{ac}}{d\theta} + F_1.\frac{dQ_{fr}}{d\theta} - \frac{dQ_{evp}}{d\theta} - P_a\left(\frac{dV_c}{d\theta} - \frac{dm_{ol}}{d\theta}.v_{ol} + \frac{dm_{evp}}{d\theta}.v_{ol}\right) =$$

$$- \frac{dm_{a.e}}{d\theta}.(h)_{a.e} + \frac{dm_{a.l}}{d\theta}.(h)_{a.l} + \frac{d(mu)_a}{d\theta} \qquad (6)$$

For the liquid-oil phase:

$$\frac{dQ_{ol.c}}{d\theta} + F_2.\frac{dQ_{fr}}{d\theta} - (P_a + P_{ov})\left(\frac{dm_{ol}}{d\theta}.v_{ol} - \frac{dm_{evp}}{d\theta}.v_{ol}\right) =$$

$$- \frac{dm_{ol.e}}{d\theta}.(h)_{ol.e} + \frac{dm_{ol.l}}{d\theta}.(h)_{ol.l} + \frac{d(mu)_{ol}}{d\theta} \qquad (7)$$

For the oil-vapour phase:

$$\frac{dQ_{ov.e}}{d\theta} + (1 - F_1 - F_2)\frac{dQ_{fr}}{d\theta} + \frac{dQ_{evp}}{d\theta} - P_{ov}\left(\frac{dV_c}{d\theta}\right.$$

$$\left.- \frac{dm_{ol}}{d\theta}.v_{ol} + \frac{dm_{evp}}{d\theta}.v_{ol}\right) = - \frac{dm_{ov.e}}{d\theta}.(h)_{ov.e} + \frac{dm_{ov.l}}{d\theta}.(h)_{ov.l} + \frac{d(mu)_{ov}}{d\theta} \qquad (8)$$

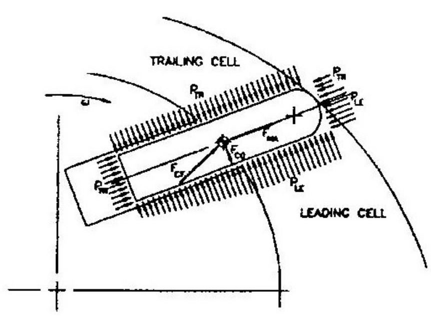

Fig. 3(a) Floating/Bottoming

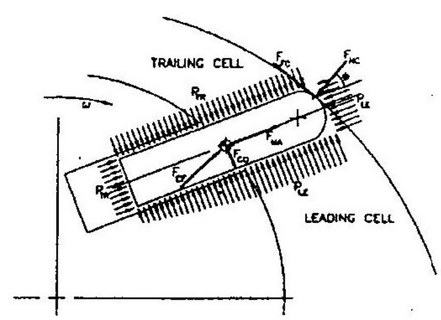

Fig. 3(b) Topping

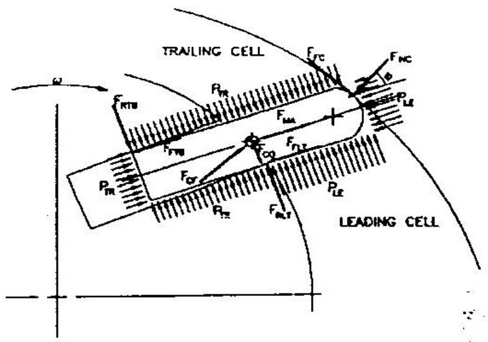

Fig. 3(c) (Forward Tilt) Reaction at the tip, middle and bottom of the vane.

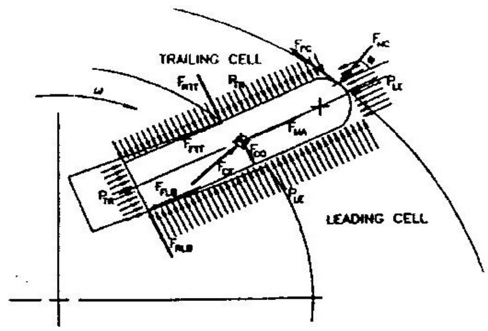

Fig.3(d) (Backward Tilt) Reaction at the tip, middle and bottom of the vane.

Mkumbwa [10], Peterson and McGahan [11] and Smith [12], verified that the presence of oil in this type of machine did not contribute significant internal heat transfer effects between air and oil during compression. Furthermore, Smith clarified the difference between the discharge air temperature and the predicted adiabatic air temperature as being caused by after-cooling effects which occurred at the outlet of the discharge port where a highly turbulent air-oil mixture existed.

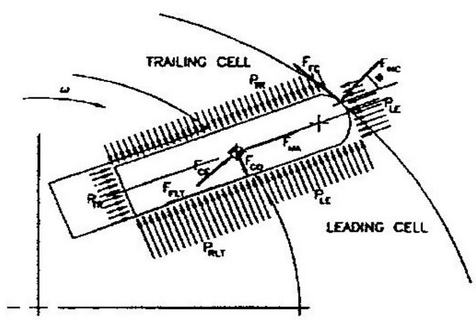

Fig. 3(e) (Hard Forward) Reactions at all contact points

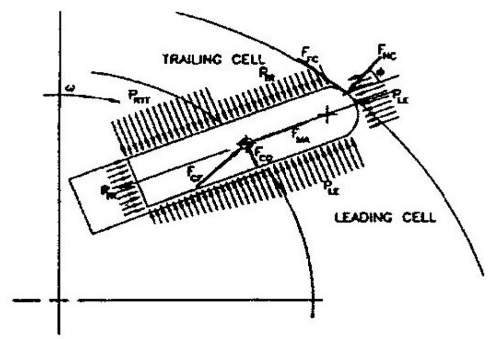

Fig. 3(f) (Hard Backwards) Reaction at all contact points.

Comprehensive simulation studies must account for the effects caused by the presence of lubrication oil in the machine. However, in any optimization study, compromise between the necessity for a realistic model and minimal computing time is needed. Some loss of the completeness of the model in representing the real physical system is unavoidable but may be acceptable provided that the adequacy of the model to represent the real system within practical tolerance bounds is maintained.

For the optimization study reported in this paper, a simplified dry model embodying perfect sealing and adiabatic processes was employed and the above equations were reduced to the equation of the air phase alone. Rearranging the temperature, the pressure and the mass terms and assuming ideal gas behaviour gives;

$$\frac{dT_c}{dt} = T_c\left(\frac{1}{V_c}\frac{dV_c}{dt} + \frac{1}{P_c}\frac{dP_c}{dt} - \frac{1}{m_c}\frac{dm_c}{dt}\right) \tag{9}$$

$$\frac{dP_c}{dt} = \frac{1}{V_c}\left((\gamma - 1)\frac{dQ}{dt} - \gamma P_c\frac{dV_c}{dt} + \gamma R\left(\sum T_i\frac{dm_i}{dt} - \sum T_o\frac{dm_o}{dt}\right)\right) \tag{10}$$

$$\frac{dm_c}{dt} = \sum\frac{dm_i}{dt} - \sum\frac{dm_o}{dt} \tag{11}$$

where P_c, T_c and V_c are cell pressure, temperature and volume respectively. The air mass flow during suction and discharge processes was obtained by applying the one dimensional compressible flow equation across the flow area assuming that this area behaves like an ideal orifice, i.e.

$$\dot{m}_{or} = \frac{P_u}{\sqrt{RT_u}}A_{or}\sqrt{\frac{2\gamma}{\gamma - 1}\left(r^{\frac{2}{\gamma}} - r^{\frac{\gamma+1}{\gamma}}\right)} \tag{12}$$

where r = Pd/Pu.....(13)

Equations (1) and (12) were solved simultaneously and equations (9) and (10) were integrated using Merson's version of a Runge-Kutta integration technique [13]. The simulation study allows the performance of a machine to be assessed and the effects of changing design parameters on the performance observed, that is the free air delivery and the amount of energy consumed.

OPTIMIZATION TECHNIQUE

A general optimization study is expressed as;

$$FOPT = f(x_i) \; , \qquad i = 1, 2, \ldots N \tag{14}$$

N is the number of free variables. In this optimization, the choice of independent variables was made with care and parameters with little influence on the outcome of the optimization discarded. Reduction in the number of independent variables was of benefit by way of a reduction in the number of functional evaluations and a reduction in computing time. Optimization constraints are;

$$(i) \quad L_{Ei} \le E_i \le H_{Ei} \; , \qquad i = 1, 2, \ldots N \tag{15}$$

$$(ii) \quad L_{Gj} \le G(x)_j \le H_{Gj} \; , \qquad j = 1, 2, \ldots Z \tag{16}$$

$$(iii) \quad L_{ik} \le I(x)_k \le H_{ik} \; , \qquad k = 1, 2, 3 \tag{17}$$

E_i represents the explicit constraints imposed on the free variables where L_{Ei} and H_{Ei} are the lower and upper limits of the free variables. $G(x)_j$ are called geometrical constraints in which each constraint is usually a function of one or more of the free variables. They describe geometrical relationships peculiar to the type of design under consideration and ensure that the combination of a particular set of variables yields a practical consideration. $I(x)_k$ are implicit constraints and represent the outcome which a particular calculation may have to satisfy.

The Complex Method used in the present study is a modification of the Simplex optimization technique developed by Spendley et al. [14], and later modified by Nelder and Mead [15] whereby an irregular Simplex which can adapt itself onto the geometry of the search region is allowed. However, in their basic forms, neither the original version nor the modified version could handle constraints. In principle any unconstrained optimization technique can be modified to handle a constrained optimization problem by the introduction of penalty functions or barrier functions into the objective function. The constrained optimization thus becomes an unconstrained optimization problem but with a modified objective function. This feature introduces complications especially when the constraints themselves are complicated inequalities and are possibly non-linear. In 1965 M.J. Box [16] introduced the version of the Simplex technique which is called Constrained Simplex or Complex technique which could handle constraints in a much better and simpler way and avoided complicated transformation of the original objective function. The method has been used extensively since its development by Box because of its basic simplicity and adaptability to suit individual optimization problems [10,17,18,19].

Basically the technique begins by specifying a first complex which satisfies all the imposed constraints and follows by setting up additional (K-1) initial complexes in a pseudo-random manner according to,

$$x_{i,j} = v_j + r_{i,j}(h_j - v_j) \tag{18}$$

where $x_{i,j}$ is any free variable
$r_{i,j}$ is the pseudo-random number in the interval 0,1.
h_j is the upper limit of the independent variable
v_j is the lower limit of the independent variable
K = (N+1), where N is the number of free variables involved in the study.

The search strategy begins by successive comparisons of the objective function values of the complexes. The worst complex is rejected by reflecting it through the centroid of the remaining Complexes. The process is repeated until any of the constraints is violated. If an explicit constraint is violated the trial point is moved a small distance inside the boundary of the constraint. If any of the geometrical or implicit constraints has been violated the trial point is moved halfway towards the centroid of the remaining complexes.

Convergence is assumed to have occurred when for a specific number of consecutive successful iterations, which was chosen as 5 in the present study, the objective function at a new point which satisfies all the above conditions lies within a preset value 0.001 (about 4% of the function value) of the value of optimizing function of the best Complex.

COMPRESSOR OPTIMIZATION

Compressor designs were optimized for their thermodynamic and mechanical performance under a prescribed operating condition.Thus whilst meeting a required free air delivery the minimum shaft power input was sought and the objective function was chosen to be the volume throughput divided by the power input to the compressor (l/kWs).

To assist the optimization algorithm at the decision making stage by causing rejection of any design which exhibited unwanted behaviour, a penalty function was introduced into the objective function. The objective function was then modified to be the following expression:-

$$FOPT = \frac{FAD}{P_{in}} - ABS(FLOWL) \qquad (19)$$

In the design of a sliding vane compressor for a specified operational condition, some 19 geometric parameters were identified. These are R_s, R_r, ε, L_r, N_v, L_v, H_v, t_v, R_t, D_s, S_w, C_s, β_1, β_2, β_3, β_4, L_s, L_d, and S_d. A given design must achieve a specified free air delivery. The 11 variables which were allowed to vary in the present optimization study are :- R_s, R_r, ε, H_v, L_r, β_1, β_2, β_3, β_4, L_s and L_d. The number of vanes (N_v) can clearly be varied but the optimization was only performed for specified number of vanes to avoid difficulties encountered with the optimization algorithm in respect of integer variables. Detail of the explicit, geometric and implicit constraints applied during the present study are given in tables 1, 2 and 3 respectively.

Table 1. Direct explicit constraints

CONSTRAINT No.	LOWER LIMIT	FREE VARIABLES	UPPER LIMIT	UNITS
1	15	$\le R_s \le$	70	(mm)
2	15	$\le R_r \le$	70	(mm)
3	1×10^{-4}	$\le \varepsilon \le$	30	(mm)
4	$0.5(360/N_v)+5$	$\le \beta_1 \le$	150	(Deg)
5	100	$\le \beta_2 \le$	200	(Deg)
6	300	$\le \beta_3 \le$	$360-0.5(360/N_v+1)$	(Deg)
7	320	$\le \beta_4 \le$	355	(Deg)
8	10	$\le H_v \le$	70	(mm)
9	100	$\le L_r \le$	500	(mm)
10	30	$\le L_d \le$	L_r	(mm)
11	1.0	$\le L_s \le$	L_r	(mm)

Table 2 Geometrical Constraints

CONSTRAINT No.	LOWER LIMIT		GEOMETRICAL CONSTRAINTS		UPPER LIMIT	UNITS
1	1×10^{-6}	≤	$(R_r+\epsilon-R_s)$	≤	50	(mm)
2	1×10^{-6}	≤	(R_s-R_r)	≤	50	(mm)
3	1×10^{-6}	≤	$(\beta_2-\beta_1)$	≤	360	(Deg)
4	1×10^{-6}	≤	$(\beta_4-\beta_3)$	≤	360	(Deg)
5	$360/N_v+5$	≤	$(360-\beta_4+\beta_1)$	≤	360	(Deg)
6	$360/N_v+5$	≤	$(\beta_3-\beta_2)$	≤	360	(Deg)
7	1×10^{-6}	≤	(L_r-L_d)	≤	1×10^{20}	(mm)
8	1×10^{-6}	≤	(L_r-L_s)	≤	1×10^{20}	(mm)
9	$360/N_v+1$	≤	(SEALANG)	≤	180	(Deg)
10	1×10^{-6}	≤	(R_p)	≤	1×10^{20}	(mm)
11	-1.0	≤	$(\cos\lambda)$	≤	0.0	(rad)
12	1×10^{-6}	≤	$(R_b-0.15R_r)$	≤	60	(mm)
13	1×10^{-6}	≤	$(R_b\cos\lambda+\sqrt{R_p}-D_t)$	≤	$0.85(H_v)$	(mm)

Table 3 Indirect implicit constraints.

CONSTRAINT No.	LOWER LIMIT		INDIRECT IMPLICIT CONSTRAINTS		UPPER LIMIT	UNITS
1	1×10^{-5}	≤	P_i	≤	30.0	(kW)
2	14.5	≤	FAD	≤	50.0	(l/s)
3	-0.3	≤	FLOWL	≤	+0.3	(l/s)

RESULTS AND DISCUSSION

The techniques outlined in the previous sections were applied in the optimization study of a rotary sliding vane compressor having circular arc rotor-stator geometry and 6 non-radially disposed vanes under a fixed operating condition. The non-radiality value was varied from -30° to +30°. This paper presents the results for a compressor fitted with vanes inclined at 5° in the direction of rotation.

Variations of the parameters of a +5° non-radial vane machine fitted with 6 vanes are plotted against the successful search number. The latter item being the number of times that the simulation model was used with a particular set of free variables and producing results within all the imposed constraints. The whole process required more than 400 simulation model executions but only 180 lay within the feasible region. Calculations required about 45 minutes of CPU-time, on a VAX-780 computer. The results shown are compared with an existing design which acts as a reference. Figures

4(a) to 4(e) and figures 5(a) to 5(f) show that dimensions may increase or decrease. The optimum values for the stator and rotor radii were smaller than those of the first feasible design, see figures 4(a,b). The variation of the rotor-stator centre offset is out of phase with the variation of the length of the rotor-stator unit. This implies that a machine can produce the same output by having a combination of large rotor-stator centre offset and a short machine length or vice-versa.

Variations of the suction and the discharge port positions are shown in figures 5(a) to 5(d). These parameters have a very close relationship with the offset between the rotor-stator centres and the length of the sealing arc. Generally speaking the port positions have marked effects on the performance of the machine. The axial length of the discharge and the suction ports converged to smaller values than that of the initial design, see figures 5(e,f).

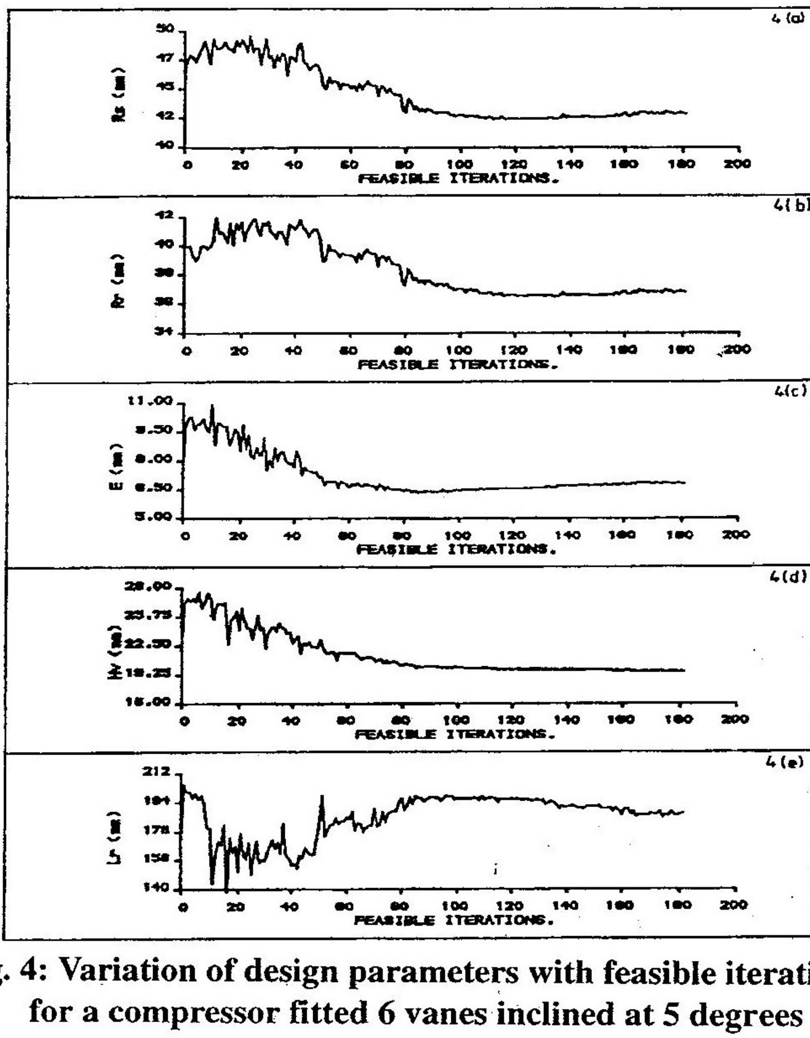

Fig. 4: Variation of design parameters with feasible iterations for a compressor fitted 6 vanes inclined at 5 degrees

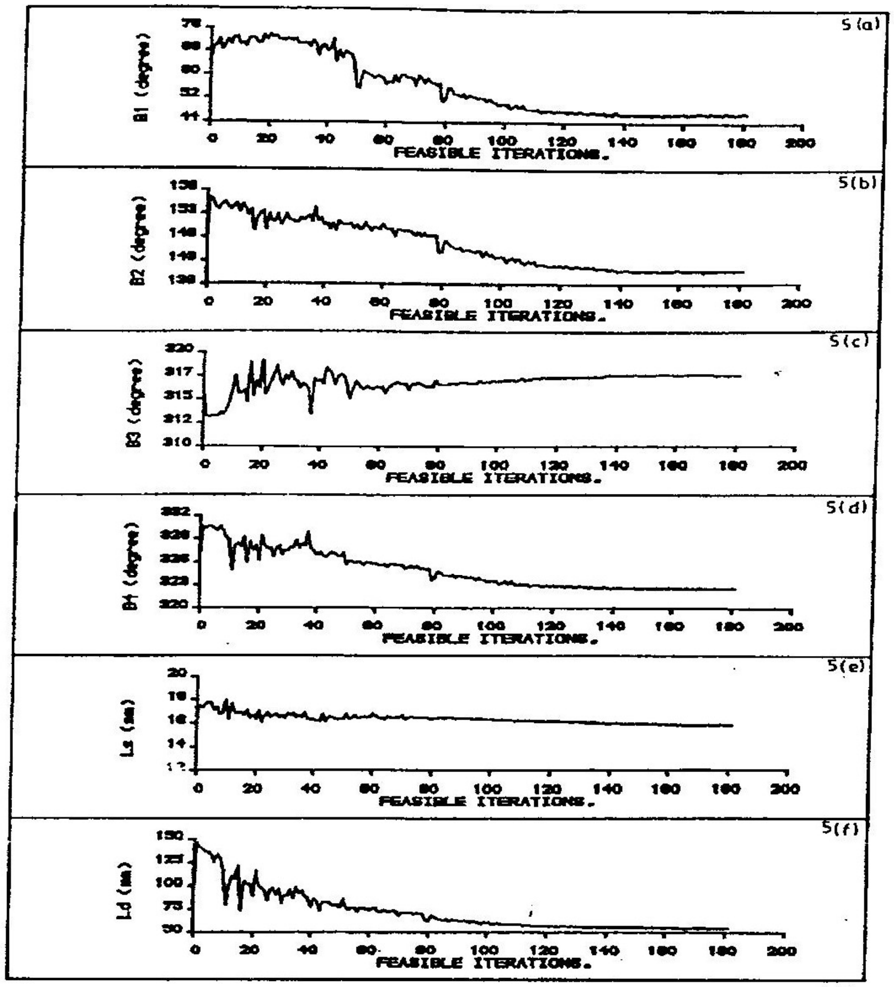

Fig. 5: variation of design parameters with feasible iterations for a compressor fitted with 6 vanes inclined at 5 degrees

It was noticed that although the frictional power dissipation increased at the optimum design, a decrease in the indicated powér, produced a net reduction in the power input. The variation of the frictional power dissipation was believed to have a close relationship to the variation of the height of the vane (Hv). For a given angular position the absolute location of the centre of gravity of the vane varies with the height of the vane, and this causes changes in the vane inertia forces.

The free air delivery figure 6(b) converges to a value close to the lower constraint value. It may be seen that the value of the objective function increases with the number of successful searches and finally converges after about 180 such searches. There was an increase in the predicted specific free air delivery as shown in figure 6(a) and when computed it was found to be a 5.5% increase when compared with that of the existing design. Table 4 shows a comparison of the dimensions of an existing feasible design with those dimensions of an optimised model while table 5 gives a comparison of the performance parameters of the existing design against the optimization parameters.

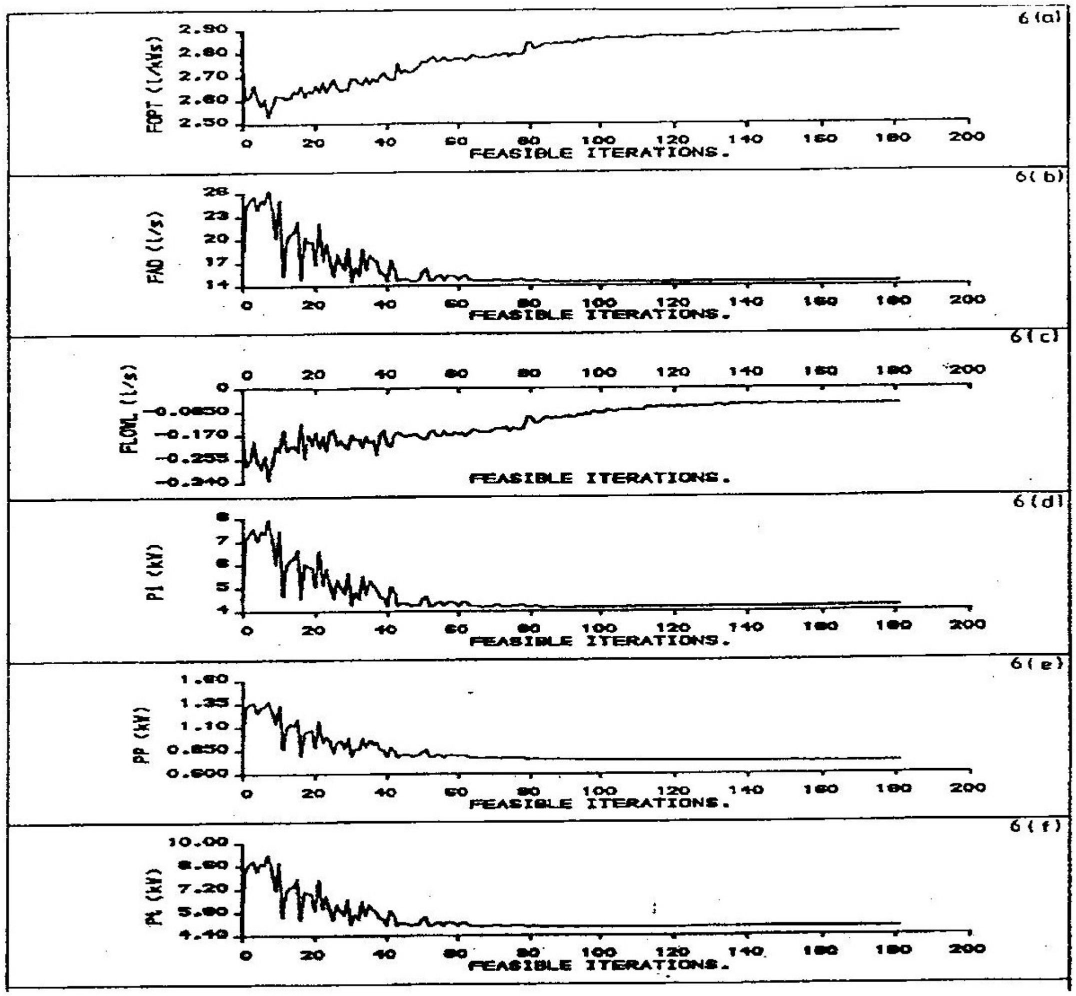

Fig. 6 Variation of optimization parameters with feasible iterations for a compressor fitted with 6 vanes inclined at 5 degrees.

Table 4 Dimensions for an existing design and an optimised design.

PROPERTY	ABBREVIATION	EXISTING DESIGN	OPTIMISED DESIGN
Stator Radius	R_s	46.524 mm	42.972 mm
Rotor Radius	R_r	40.048 mm	36.873 mm
Eccentricity	ε	7.354 mm	6.831 mm
Suction Port Opening Angle	β_1	64.754 deg	47.240 deg
Suction Port Closing Angle	β_2	148.929 deg	141.110 deg
Discharge Port Opening Angle	β_3	316.565 deg	317.600 deg
Discharge Port Closing Angle	β_4	326.332 deg	322.620 deg
Vane Width	H_v	22.545 mm	19.564 mm
Rotor Length	L_r	170.000 mm	186.500 mm
Discharge Port Axial Length	L_d	70.061 mm	54.546 mm
Suction Port Axial Length	L_s	13.115 mm	15.812 mm
Rotor Slot Depth	D_s	22.830 mm	22.830 mm
Vane Thickness	t_v	3.945 mm	3.945 mm
Mass per Vane	m_v	0.064 kg	0.064 kg
Vane Length	L_v	85.000 mm	93.25 mm
Number of Vanes	n	6	6
Sealing Arc Clearance	C_s	0.0775 mm	0.0775 mm
Vane Tip Radius	R_v	5.000 mm	5.000 mm

Table 5 Performance Data for an existing model with that for an optimised model

PROPERTY	ABBREVIATION	EXISTING PERFORMANCE PARAMETERS	OPTIMISED PERFORMANCE PARAMETERS
Objective Function	FOPT	2.754 l/kWs	2.894 l/kWs
Free Air Delivery	FAD	15.651 l/s	14.514 l/s
Free Air Leakage	FLOWL	-0.156 kW	-0.053 l/s
Indicated Power	P_i	4.560 kW	4.199 kW
Frictional Power	P_f	0.820 kW	0.726 kW
Total Compressor Power	P_t	5.380 kW	4.925 kW

CONCLUSION

The work reported in this paper shows that modelling and computer aided design techniques are powerful tools for the compressor designer. Significant performance improvements have been predicted by the use of these techniques. These improvements include the reduction in frictional power and hence the total compressor power requirement.

These techniques can also in some cases reduce the overall dimensions of the compressor which will undoubtedly lead to a reduction in initial cost of these machines. The corroborative experimental programme [10,20] which followed was only required to validate the predictions.

NOMENCLATURE

C_s	Sealing arc clearance, mm
D_s	Slot depth, mm
F_1	Fraction of frictional energy transfer to air phase
F_2	Fraction of frictional energy transfer to oil liquid phase
FAD	Free air delivery, l/s
FLOWL	Difference between induced air and discharged air, l/s
FOPT	Objective function, l/kWs
h	Enthalpy, kJ/kg
H_v	Height of the vane, mm
L, H	Lower and Upper limit of constraints
L_s	Axial length of the suction port, mm
L_d	Axial length of the discharge port, mm
L_r	Length of the rotor-stator unit, mm
L_v	Length of the vanes, mm
m	mass flow, kg
N_v	Number of vanes
P	Pressure, N/m2
P_{fr}	Frictional dissipation of the sliding vane, kW
P_i	Indicated power, kW
P_t	Total power (sum of Pi and Pfr), kW
Q	Heat transfer, J
R_b bottom of	Radial distance between centre of the rotor and the the slot, mm
R_p instance, m	Distance from bottom of slot to the vane tip at any
R_r	Rotor radius, mm
R_s	Stator radius, mm
R_t	Radius of vane tip, mm
S_w	Slot width, mm
T	Temperature, K
TDC	Top Dead Centre
t_v	Vane thickness, mm
v	Specific volume, m3/kg

β_1	Suction port opening angle w.r.t. TDC, deg
β_2	Suction port closing angle w.r.t. TDC, deg
β_3	Discharge port opening angle w.r.t. TDC, deg
β_4	Discharge port closing angle w.r.t. TDC, deg
ε	Offset between the centre of rotor and stator, m
σ	Non-radiality angle, deg
θ	Angular position, deg

Subscripts

a	air
c	cell
d	downstream
i	in
o	out
E	explicit
G	geometrical
I	implicit
u	upstream
ov	oil vapour
ol	oil liquid
or	orifice
fr	friction
evp	evaporation

Second Subscripts

c	convective heat transfer
e	entering
l	leaving

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