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Optimisation of Compressor Valve Designs Using the 'Nelder-Mead' Simplex Method

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ABSTRACT

A combination of a compressor simulation model and an optimisation technique was used to predict enhanced compressor performance through an improvement in valve design. The Nelder-Mead Simplex optimisation technique was used to determine optimum combinations of various compressor valve design parameters in a multi-variable, multi-constraint problem. In this investigation 10 independent variables, 15 explicit and 2 implicit constraints, were manipulated to achieve a minimum value of the chosen optimisation function - namely energy consumption per unit of flow delivered at specified operating conditions. An external barrier function comprising the sum of the objective function and a penalty term was used to prevent constraint violation. The paper illustrates the progress to the attainment of optimum values of the independent variables as well as the corresponding performance parameters (indicated power, delivery capacity, valve power losses, objective function). The paper illustrates a technique which is of great potential value in research and development work associated with compressors.

INTRODUCTION

The process of achieving the best combination of a set of independent variables under certain constraints in order to obtain an optimum value of an objective function may be called design optimisation. According to one's definition, the optimum value of the objective function may represent a maximum or a minimum value of that function. Design optimisation may utilise many methods and techniques but the choice of a particular method is influenced by the nature of the variables and the constraints as well as the relationships between variables, constraints and objective function.

Reciprocating compressor valves may be described by the following variables :- port diameter (or area), covering plate diameter (or area), covering plate thickness, permitted lift, valve spring stiffness, valve spring preload. i.e. 6 variables per valve. For a compressor having a single suction valve and a single discharge valve per cylinder some 12 independent variables have to be considered in a valve design optimisation procedure. In turn each variable may have values lying within certain ranges - the so called explicit constraints - according to considerations of structure, strength and other practical requirements. The present investigation placed explicit constraints upon port diameters, cover plate thicknesses, permitted valve lifts, valve spring stiffnesses, valve spring preloads. Implicit constraints were put on the variables by the specification of maximum values for the velocity with which the moving cover plate struck a motion limiting stop. (An impact velocity constraint).

The present investigation was centred on the need to obtain an optimum valve design from a thermodynamic viewpoint and thus the objective function was selected to be energy consumption per unit of flow delivered. It was the task of the optimisation procedure to minimise the value of the objective function for specific compressor design conditions (compressor speed, suction pressure, suction temperature, discharge pressure). Values of the objective function were calculated using a digital computer and a mathematical model of the compressor and its valves. An optimum valve design was achieved by combining the mathematical model and the "Nelder-Mead" Simplex optimisation method.

Previous publications [3], [4], [5] concerning work relating to the optimisation of compressor valves performed at the University of Strathclyde have reported the use of various optimisation methods and discuss the advantages and disadvantages of these methods vis-a-vis search speed, constraint handling and the identification of local optima as opposed to a global optimum condition. The work reported in this paper utilised a Simplex Method [2], a method generally considered to be both powerful and effective for many optimisation problems.

VARIABLES

If a compressor valving system is deemed to comprise a simple spring loaded disc covering a port then for a single cylinder having one suction valve and one discharge valve the 12 independent variables at the disposal of the designer may be represented in matrix notation as :-

$$\bar{X} = (x_1, x_2, x_3, - - - - -, x_n) \quad n = 1, 12$$

where in the present study

x_1, x_6 = suction, discharge valve disc thickness

x_2, x_7 = suction, discharge valve port diameter

x_3, x_8 = suction, discharge valve permitted lift

x_4, x_9 = suction, discharge valve spring stiffness

x_5, x_{10} = suction, discharge valve spring preload

x_{11}, x_{12} = suction, discharge valve disc diameter

The variables x_{11}, x_{12} can be eliminated from the study if fixed values are introduced for the ratio $\frac{\text{valve disc diameter}}{\text{valve port diameter}}$, i.e. x_{11}/x_2 or x_{12}/x_7.
By utilising this result the total number of independent variables used in the present study was reduced to 10.

EXPLICIT CONSTRAINTS

A number of independent variables were constrained directly by specifying upper and lower bounds to their values.

1. Maximum and minimum values for disc thickness

$$H_{s1} \leqslant x_1 \leqslant H_{s2}$$

$$H_{d1} \leqslant x_6 \leqslant H_{d2}$$

2. Finite positive values for valve port diameters

$$x_2 \geqslant 0$$

$$x_7 \geqslant 0$$

3. Finite positive values for permitted valve lifts

$$0 \leqslant x_3 \leqslant Y_s$$

$$0 \leqslant x_8 \leqslant Y_d$$

4. Finite positive values for valve spring stiffness.

$$x_4 \geqslant 0$$

$$x_9 \geqslant 0$$

5. Finite positive valve spring preloadings

$$x_5 \geqslant 0$$

$$x_{10} \geqslant 0$$

6. Non overlapping suction and discharge valves lying wholly within the compressor cylinder. This condition was recognised by placing a limit on the sum of the valve port diameters

$$x_2 + x_7 \leqslant \frac{2}{3} D$$

where D is the compressor cylinder diameter

IMPLICIT CONSTRAINTS

7. Impact velocity constraints

Limits were placed upon the maximum values for the velocities with which the moving element in a valve strikes a displacement limiting stop. Practical experience has determined that the stresses associated with valve breakages are related to the impact velocity and the types of material being brought into contact. The values used are empirical and the constraints are implicit since the impact velocity depends in a complicated way upon the valve geometry and the compressor operating conditions. The impact velocities are calculated by the mathematical model and are clearly dependent variables which cannot be constrained explicitly.

$$v_s \leqslant \mathbf{V}_s$$

$$v_d \leqslant \mathbf{V}_d$$

Consideration of the foregoing statements reveals that some 17 limiting conditions (constraints) are applicable in the present study. In any given study the number of constraints will vary according to the number of independent variables employed and the restrictions placed upon them by the designer in the light of his previous practical experience.

GENERAL TREATMENT OF CONSTRAINTS

It is convenient to represent the constraint conditions in a unified manner which may then be used when penalty functions are introduced.

Thus in general $G_j(\bar{X}) \geqslant 0 \qquad j = 1,2, \ldots 17$
or for the 17 constraints applied in the present study

$$G_1 = C_1 (x_1 - H_{s1})$$

$$G_2 = C_2 (H_{s2} - x_1)$$

$$G_3 = C_3 (x_6 - H_{d1})$$

$$G_4 = C_4 (H_{d2} - x_6)$$

$$G_5 = C_5 x_2$$

$$G_6 = C_6 x_7$$

$$G_7 = C_7 x_3$$

$$G_8 = C_8 (Y_s - x_3)$$

$$G_9 = C_9 x_8$$

$$G_{10} = C_{10} (Y_d - x_8)$$

$$G_{11} = C_{11} x_4$$

$$G_{12} = C_{12} x_9$$

$$G_{13} = C_{13} x_5$$

$$G_{14} = C_{14} x_{10}$$

$$G_{15} = C_{15} \left(\frac{2}{3} D - x_2 - x_7 \right)$$

$$G_{16} = C_{16} (\mathbf{V}_s - v_s)$$

$$G_{17} = C_{17} (\mathbf{V}_d - v_d)$$

The factors C_j are used to ensure that the same order of magnitude prevails for all the constraint functions G_j thereby making it possible to introduce penalty functions which will prevent any variable from exceeding prescribed bounds.

In the present case

$$C_1, C_2 -- C_9,\ C_{10},\ C_{15},\ C_{16},\ C_{17} = 1.0$$

$$C_{11},\ C_{12} = 0.01$$

$$C_{13},\ C_{14} = 10.0$$

OBJECTIVE FUNCTION

For a given compressor operating condition a compressor design might be deemed to be an optimum if it requires a minimum power input per unit volume of fluid delivered. Thus considering a single compressor cycle

$$F(\bar{X}) = \frac{W_i/W_o}{Q/Q_o} = \frac{1}{(Q/Q_o)\ (W_o/W_i)}$$

where Q = volume of fluid induced at suction conditions

Q_o = cylinder swept volume

W_i = indicated cycle work

W_o = theoretical adiabatic cycle work

Thus $$F(\bar{X}) = \frac{1}{\eta_v(\bar{X})\ \ \eta_i(\bar{X})}$$

where η_v = volumetric efficiency

η_i = indicated adiabatic efficiency

The relationship between the variables $\bar{X}$ and the objective function $\mathbf{F}(\bar{X})$ is a highly non linear function and is evaluated numerically using the compressor simulation model programmed for a digital computer.

PENALTY FUNCTIONS

An external penalty function is introduced into the optimisation routine to prevent any variable taking on values which cause a constraint to be violated. The penalty function is obtained by adding an extra penalty term to the objective function :-

$$\mathbf{P}(\bar{X}) = \mathbf{F}(\bar{X}) + \gamma_k \sum_{j=1}^{j=17} \text{Min}\ [G_j\ (\bar{X}),\ 0]$$

The meaning of the term Min $[G_j\ (\bar{X}),\ 0]$ is that the penalty term is zero and the penalty function is equal to the objective function when all the constraint conditions are satisfactory $[G_j\ (\bar{X}) \geqslant 0]$.

The external penalty function is the sum of the objective function and the penalty term which is made much larger than the objective function when one of the constraints is violated. $[G_j\ (\bar{X}) < 0]$. γ_k is the so called penalty factor which ensures that the penalty term is made much greater than the objective function. The Simplex method has to discard a design point which would entail a constraint violation and the penalty function technique provides a convenient method of ensuring that all variables stay within their permitted bounds.

THE SIMPLEX METHOD

In the SIMPLEX method an N variable optimisation problem requires the formation of an N + 1 sided polygon with one of the vertices corresponding to an initial choice of variables (possibly an existing design configuration). Each vertex represents a given combination of the independent variables and is associated with a corresponding value of the objective function $\mathbf{F}(\bar{X})$. In the present study the SIMPLEX method requires that the vertex associated with the maximum value of $\mathbf{F}(\bar{X})$ be discarded and a new vertex (having a lower value of $\mathbf{F}(\bar{X})$) is introduced to create a new polygon. The process is repeated and the shape and position of the polygon change as the technique seeks to establish a combination of variables which yields a minimum value of $\mathbf{F}(\bar{X})$. Unsatisfactory vertices are replaced by procedures involving reflection, expansion, contraction, reduction.

Figure 1 illustrates in a 2 dimensional manner the basic steps of the SIMPLEX method. The figure represents an N + 1 sided polygon for an N dimensional optimisation problem. $\bar{X}_h$ represents the vertex having the highest value of $F(\bar{X})$. $\bar{X}_l$ represents the vertex having the lowest value of $F(\bar{X})$ and $\bar{X}_b$ represents the centroid of the polygon formed when the point $\bar{X}_h$ is excluded. In figures 1 (a), (b), (c) $\bar{X}_r$ is the point obtained by reflecting the point $\bar{X}_h$ through the centroid $\bar{X}_b$. The distance $(\bar{X}_h - \bar{X}_b)$ is equal to the distance $(\bar{X}_b - \bar{X}_r)$. Point $\bar{X}_e$ corresponds to the situation where the point $\bar{X}_h$ is reflected to a point twice as far from the centroid as $\bar{X}_h$ was from the centroid.

i.e. $\bar{X}_b - \bar{X}_e = 2(\bar{X}_b - \bar{X}_r) = 2(\bar{X}_h - \bar{X}_b)$

In figure 1 (c) the point $\bar{X}_c$ represents a contraction of the reflected point $\bar{X}_r$ towards the centroid $\bar{X}_b$. As illustrated point $\bar{X}_c$ is midway between points $\bar{X}_r$, $\bar{X}_b$.

In figure 1 (d) the point $\bar{X}_c$ represents a contraction of the point $\bar{X}_h$ towards the centroid. In figure 1 (e) the point $\bar{X}_{re}$ represents a reduction in the size of the polygon so that the polygon has been reduced to half its initial size. It must be remembered that whilst the illustration in figure 1 is a two dimensional representation of the SIMPLEX method, the terms $\bar{X}$, $\bar{X}_h$, $\bar{X}_l$ are N dimensional arrays representing combinations of N variables.

OPTIMISATION PROCEDURE

1) Set up the initial N + 1 sided polygon in which one of the vertices corresponds to a feasible combination of the variables $\bar{X}_o$.

2) Compare the values of the objective function $F(\bar{X})$ for each of the vertices of the current polygon and establish the vertices having the minimum and maximum values of $F(\bar{X})$, i.e. $F(\bar{X})_l$, $F(\bar{X})_h$.

3) Calculate the values of the variables corresponding to the centroid of the polygon that excludes values corresponding to the point $\bar{X}_h$.
i.e. $\bar{X}_b = \frac{1}{N} \sum_{i=1}^{i=N} \bar{X}_i \qquad i \neq h$

4) Test whether the current polygon is small enough to establish that an optimum condition has been achieved.
i.e. is $\frac{1}{N+1} \sqrt{\sum_{i=1}^{i=N+1} (F(\bar{X}_i) - F(\bar{X}_b))^2} \leqslant \epsilon$
If this condition is satisfied then the optimisation has been completed and the optimum values of the variables correspond to those of the point $\bar{X}_l$, the point having the lowest value of $F(\bar{X})$. In the present problem ϵ was set as 0.001.

5) If convergence has not been achieved then the point with the highest value of $F(\bar{X})$, $\bar{X}_h$ is reflected through $\bar{X}_b$ to $\bar{X}_r$ using the relationship $\bar{X}_r = 2\bar{X}_b - \bar{X}_h$.

6) Recalculate $F(\bar{X}_r)$. If $F(\bar{X}_r) < F(\bar{X}_l)$ then moving in the direction of the reflection has yielded an improvement and this process may be taken a stage further by reflecting $\bar{X}_b$ through $\bar{X}_r$ to $\bar{X}_e$ using the relationship
$\bar{X}_e = 2\bar{X}_r - \bar{X}_b$.
If $F(\bar{X}_e) < F(\bar{X}_l)$ then $\bar{X}_e$ is considered to be an improved point and the point $\bar{X}_e$ is substituted in the simplex in place of $\bar{X}_h$ (expansion as in Figure 1 a). The procedure then returns to step 2.

If $F(\bar{X}_e) \nless F(\bar{X}_l)$ replace point $\bar{X}_h$ by point $\bar{X}_r$ and return to step 2.

7) If $F(\bar{X}_r) > F(\bar{X}_l)$, judge whether $F(\bar{X}_r) > F(\bar{X}_i)$ where $(\bar{X}_i)$ covers all points in the simplex excluding the point $(\bar{X}_h)$.

If this condition is not fulfilled then point $\bar{X}_r$ is a better point than $\bar{X}_h$. Point $\bar{X}_r$ is substituted for $\bar{X}_h$, reflection as in Figure 1b and the routine returns to step 2.

8) If however $F(\bar{X}_r) < F(\bar{X}_h)$ some improvement has been achieved by a move in the reflection direction but the reflection has been excessive. Contract $\bar{X}_r$ to $\bar{X}_c$ when using
$\bar{X}_c \doteq (\bar{X}_r + \bar{X}_b)/2$. as in Figure 1c

Substitute point $\bar{X}_c$ for point $\bar{X}_h$ and return to step 2.

If $F(\bar{X}_r) \nless F(\bar{X}_h)$ the current polygon is contracted using $\bar{X}_c = (\bar{X}_b + \bar{X}_h)/2$.

9) If $F(\bar{X}_c) < F(\bar{X}_h)$, replace $\bar{X}_h$ by $\bar{X}_c$ as in Figure 1d and return to 2. If this condition is not satisfied the current polygon is too large and includes a point where $F(\bar{X}) > F(\bar{X}_h)$. The polygon is reduced in size by moving each vertex towards the vertex having the lowest value of $F(\bar{X})$ i.e. $F(\bar{X}_1)$. This move is achieved by writing

$$\bar{X}_i = (\bar{X}_i + \bar{X}_1)/2. \quad \text{as in Figure 1e}$$

and returning to step 2.

The computer program embodying the optimisation algorithms is relatively simple and is linked directly to a number of routines which evaluate the objective function, check the constraint functions and set up the penalty functions. Figure 2 illustrates the way in which a control function subroutine links the various parts of the necessary computer program.

When the optimisation process has converged values of the independent variables for the centroid of the simplex are printed out together with the corresponding values of the objective function, valve opening and closing angles, compressor capacity indicated power and volumetric efficiency obtained from the compressor simulation model.

RESULTS OF OPTIMISATION PROCEDURE

The previously described optimisation procedure was applied to a reciprocating air compressor having a bore of 38.1 mm, a stroke of 25.4 mm run at a speed of 1000 rev/min. Optimisation was performed for a fixed pressure ratio of 3:1 when the nominal suction conditions were, suction pressure 1 bar absolute, suction temperature 290 K.

Calculations were performed for a fixed value of the ratio (valve plate area/valve port area) of 1.3 . Impact velocity constraints were set at $V_s = 2.5$ m/s, $V_d = 5.0$ m/s for the suction and discharge valves respectively. Table 2 shows the values of the major design parameters for an existing compressor together with the parameter values used to commence the optimisation. Table 2 also contains values for these parameters at the mid point (after 100 iterations) and the endpoint (after 240 iterations) of the optimisation process.

Figures 3 and 4 show the progress of the optimisation process as it seeks to establish an optimum combination of the design variables. Figure 3 illustrates the variation of the independent variables whilst Figure 4 shows the effect upon the derived parameters. Both figures are drawn to a base of iteration number (the number of times the compressor simulation routine was called following the formation of the initial simplex). Figures 3 and 4 show that the optimised values of both the independent variables and the objective function may differ significantly from their initial values. For the present study some 200+ iterations were needed to attain an optimum combination and significant improvements are predicted for the compressor performance parameters as revealed by the marked reduction in the value of the objective function. The delivered volume remained approximately constant at 26 litre/min but the indicated power reduced from 136W to 65W. This reduction in indicated power represents a reduction in the energy required to drive the compressor and reveals the potential of design optimisation techniques.

Figures similar to figures 3 and 4 are valuable in illustrating the manner in which the independent variables and the dependent quantities change as the optimisation procedure progresses. Such figures are needed when determining whether the optimisation procedure should be stopped. Figure 4 clearly shows that this particular optimisation had to all intents and purposes converged after 200 iterations. Marginal gains were predicted from iterations 200 - 250 and virtually no changes emerge from iterations beyond 250.

An optimisation can show which parameters are truly significant and which constraints exercise the greatest influence on the value of the objective function. Alteration of the value of a constraint may render that constraint inactive as far as the optimisation procedure is concerned. For example, in the present case, setting valve impact velocity values of $V_s = 2.5$ m/s, $V_d = 5.0$m/s could be seen as establishing a boundary between an active and an inactive impact velocity criterion. If $V_s > 2.5$ m/s and $V_d > 5.0$ m/s the same ultimate optimum design was reached as was the case with $V_s = 2.5$ m/s, $V_d = 5.0$ m/s. cf Table 3, columns 1 and 3. By implication the calculated values of the impact velocities are less than the impact velocity constraint values and the constraints have effectively become inactive. The optimum design was being constrained more vigorously by one of the other explicit variable constraints.

However when $V_s < 2.5$ m/s and $V_d < 5.0$ m/s the impact velocity constraints become active and either V_s or V_d (or even both) determine the ultimate design point. A comparison of columns 2 and 4 of Table 3 shows that whilst the final design point is the same V_d had been raised from 2.5 m/s to 5.0 m/s. Clearly the suction valve impact velocity constraint is having a dominant effect upon the outcome of the optimisation whilst the discharge valve impact velocity constraint is inactive. The present work reveals the importance of having correct values for impact velocity constraints so that design procedures are not constrained unnecessarily.

Two variables were eliminated from the optimisation by use of fixed values for the ratio (disc diameter/port diameter). Figure 5 shows that this ratio has a profound effect upon the value of the objective function.

c.f. $F(\bar{X})_{opt}$ falling from 1.54 to 1.3 and then to 1.15 as the ratio (A_v/A_p) was reduced from 1.5 to 1.3 and then to 1.1. See Table 4.

This result shows a clear need to minimise valve seat widths, a feature which must be tempered in practice by the need for effective sealing and acceptable wear characteristics.

CONCLUSIONS

The combination of optimisation and mathematical modelling techniques equips the design engineer with a powerful tool with which to attack design improvement problems but does not absolve him from the exercise of his professional judgement in assessing the suitability of any model used by him or in the selection of the constraints used to circumscribe a particular set of possibilities. Work reported in this paper shows how the combination of a relatively simple mathematical model of a compressor [1] and the Simplex Optimisation Technique was able to indicate the existence of an optimum combination of some 10 valve design parameters subject to 15 explicit and 2 implicit constraints.

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TABLE 1

COMPRESSOR DATA

PARAMETER	UNIT	VALUE
Cylinder Diameter	mm	38.1
Stroke	mm	25.4
Connecting Rod Length	mm	88.9
Speed	rev/min	1000
Suction Pressure	Bar	1.0
Discharge Pressure	Bar	3.0
Suction Temperature	K^o	290
Clearance Volume	mm^3	1450

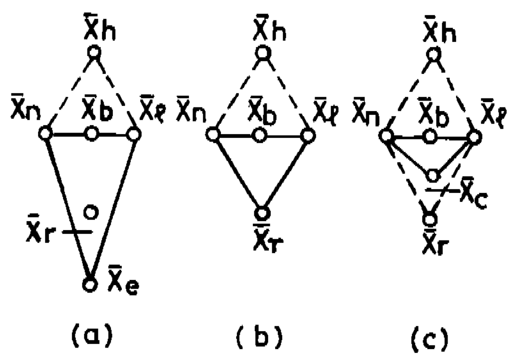

FIG.1. PROCEDURE OF SIMPLEX METHOD

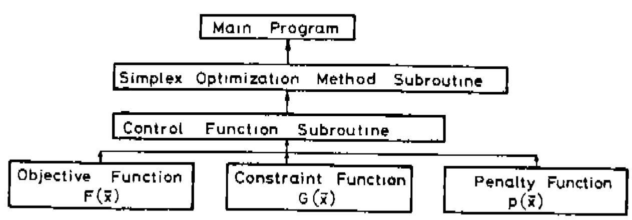

Fig. 2. THE COMPOSITION SCHEME OF COMPUTER PROGRAM

TABLE 2

PROGRESS TO OPTIMUM DESIGN VALUES

ITEM	UNITS	EXISTING COMPRESSOR	INITIAL VALUE I.N=1	VALUE AT I.N=100	VALUE AT I.N=240
Suction Valve					
Plate thickness	mm	3.3	1.0	1.57	1.23
Port diameter	mm	8.4	6.0	10.03	12.13
Valve plate lift	mm	0.6	0.4	1.07	1.00
Valve spring stiffness	N/m	200.0	50.0	89.9	69.8
Valve spring preload	N	0.2	0.05	0.14	0.18
Discharge Valve					
Plate thickness	mm	3.7	1.0	2.6	3.25
Port diameter	mm	8.4	3.0	8.1	12.45
Valve plate lift	mm	0.6	0.2	0.56	0.56
Valve spring stiffness	N/m	420.0	250.0	539.0	470.0
Valve spring preload	N	0.5	0.25	0.42	0.23
Performance Parameter					
Volume flowrate	litre/min	25.2	25.9	26.2	25.0
Suction power loss	W	6.07	6.64	2.56	2.32
Discharge power loss	W	16.73	70.5	17.89	12.0
Indicated power	W	78.72	136.83	76.89	66.8
Volumetric efficiency	%	87.2	89.6	90.5	86.3
Indicated efficiency	%	77.6	44.7	79.5	91.5
Objective function		1.478	2.498	1.391	1.267

Limiting conditions: Av/Ap = 1.3 ; V_s = 2.5 m/s ; V_d = 5.0 m/s

TABLE 3

EFFECT OF IMPACT VELOCITY CONSTRAINTS UPON OPTIMUM VALUES OF VARIOUS PARAMETERS

Impact velocity at valve stop	m/s	$V_s \geq 2.5$, $V_d \geq 5.0$	$V_s=1.25$, $V_d=5.0$	$V_s=2.5$, $V_d=2.5$	$V_s=1.25$, $V_d=2.5$
Suction Valve					
Plate thickness	mm	1.23	1.01	1.17	1.01
Port diameter	mm	12.13	16.32	14.09	16.32
Valve plate lift	mm	1.00	0.39	1.18	0.39
Valve spring stiffness	N/m	69.8	64.0	114.7	64.0
Valve spring preload	N	0.18	0.08	0.13	0.08
Discharge Valve					
Plate thickness	mm	3.25	3.61	3.31	3.61
Port diameter	mm	12.45	8.99	11.16	8.99
Valve plate lift	mm	0.56	0.59	0.59	0.59
Valve spring stiffness	N/m	470.0	1376.0	774.0	1377.0
Valve spring preload	N	0.23	0.67	0.78	0.67
Performance Parameter					
Volume flowrate	litre/min	25.0	26.4	27.0	26.4
Suction power loss	W	2.32	4.01	1.74	4.01
Discharge power loss	W	12.0	16.65	13.25	16.65
Indicated power	W	66.8	78.2	72.5	78.2
Volumetric efficiency	%	86.3	91.2	93.1	91.2
Indicated efficiency	%	91.5	78.1	84.3	78.1
Objective function		1.267	1.398	1.274	1.398

Limiting conditions: Av/Ap = 1.3 ; Iteration Number=240

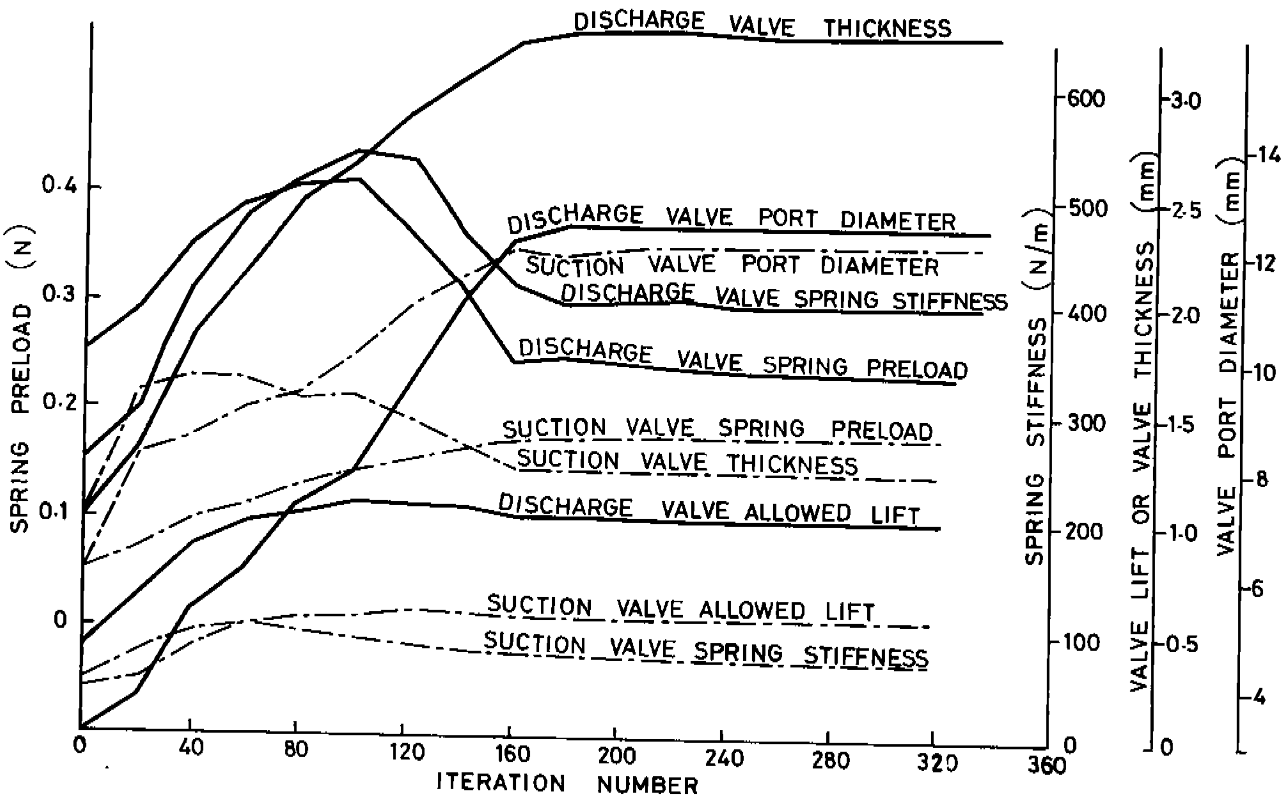

Fig. 3. VARIATION OF VARIABLES DURING OPTIMISATION

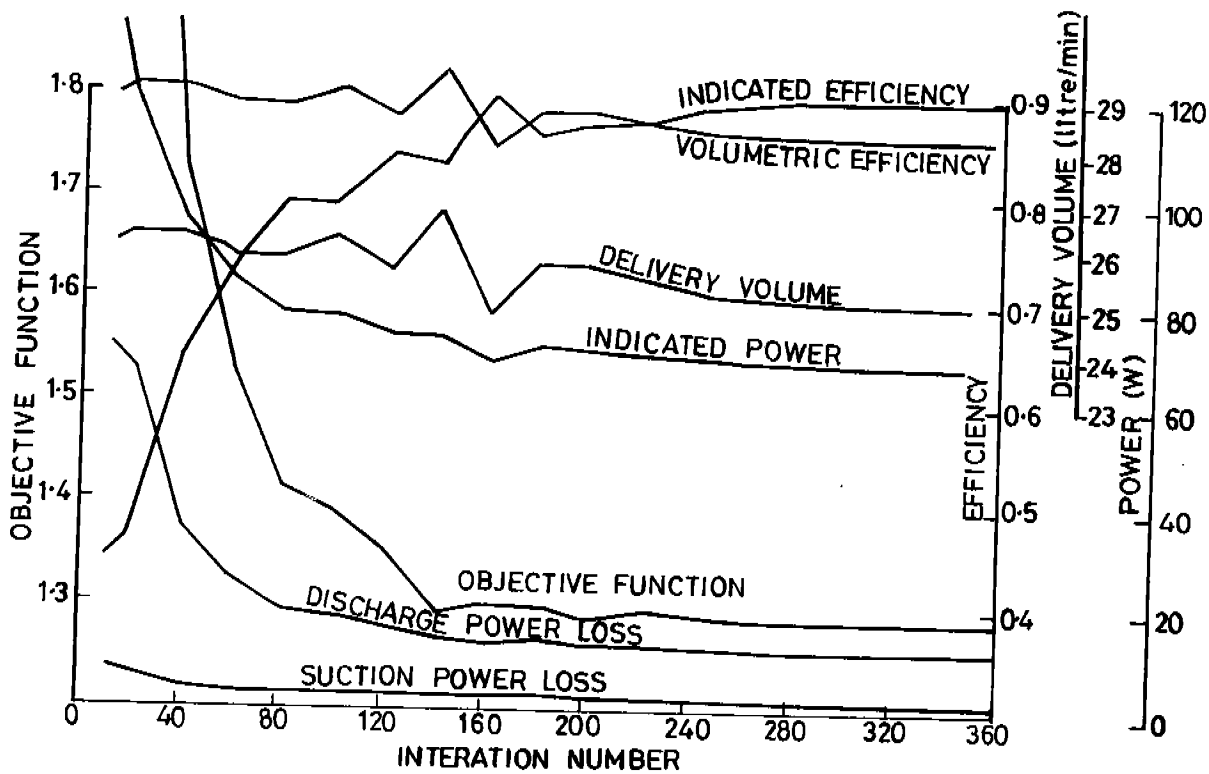

FIG. 4. VARIATION OF PERFORMANCE PARAMETERS

TABLE 4

EFFECT OF VALVE PLATE TO VALVE PORT AREA RATIO Av/Ap UPON OPTIMUM VALUES

Valve plate to valve port area ratio	Av/Ap	1.1	1.3	1.5
Suction Valve				
Plate thickness	mm	1.38	1.23	1.93
Port diameter	mm	13.12	12.13	13.38
Valve plate lift	mm	0.8	1.0	1.02
Valve spring stiffness	N/m	193.6	69.8	126.7
Valve spring preload	N	0.13	0.18	0.02
Discharge Valve				
Plate thickness	mm	3.94	3.25	3.92
Port diameter	mm	12.02	12.45	11.07
Valve plate lift	mm	0.54	0.56	0.57
Valve spring stiffness	N/m	706.0	470.0	550.0
Valve spring preload	N	0.43	0.23	0.35
Performance parameter				
Volume flowrate	litre/min	26.8	25.0	26.1
Suction power loss	W	1.71	2.32	2.96
Discharge power loss	W	6.18	12.0	23.81
Indicated power	W	65.2	66.8	83.4
Volumetric efficiency	%	92.6	86.3	90.0
Indicated efficiency	%	93.7	91.5	73.3
Objective function		1.153	1.267	1.516
Limiting conditions	V_s = 2.5 m/s ;	V_d = 5.0 m/s ;	I.N = 240	

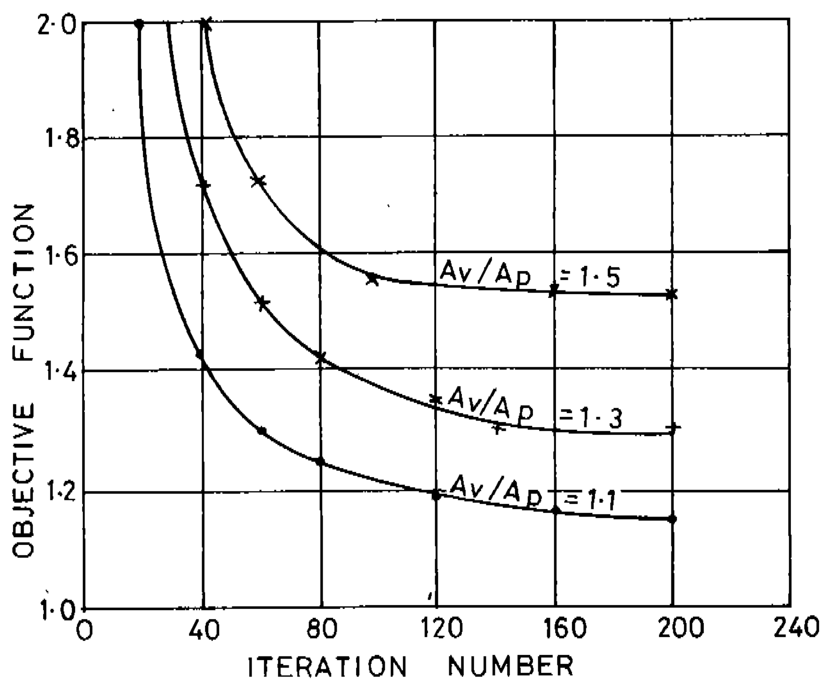

FIG. 5 VARIATION OF OBJECTIVE FUNCTION WITH RATIO A_v/A_p