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OPTIMIZATION OF INTER–COOLER IN RECIPROCATING COMPRESSOR

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ABSTRACT

Based on the energy balance law and the mass balance law in thermodynamics, a mathematical model on the transient heat exchange process of the inter–cooler is established. Then a model on optimization of the cooler is proposed in which the objective function is the total entropy generation rate (TEGR), giving consideration to both the heat transfer and the flow resistance in the cooler. Finally, the results of the optimization on the structure size and the operating parameters are searched for the cooler. The feasibility of the simulating model and the reliability of the optimizating results have been proved by the experiments in a compressor.

INTRODUCTION

In order to increase the economy and reliability of multi–stage compressor during operation, studies on the inter–cooler of compressor should be paid attention to in addition to studing the compressor itself. From the standpoint of the whole compressor system including the compressor, inter–cooler, and later–cooler, the authors have conducted the simulation of the transient heat transfer and the optimization of the structure for the inter–cooler. In combination with our previous publications[1,2], the computer aided design for compressor system can be realized.

The shell–and–tube cooler with baffles is widely used in engineering. In the present paper, a mathematical model on the transient heat exchange of a counterflow shell–and–tube cooler has been established on the basis of the energy balance law and the mass balance law in thermodynamics. By using the flow path analysis method, the heat exchange coefficient for the shell side in the reference[2] is revised, thus the accuracy of the calculation is improved. At the same time, by means of the finite difference method, the numerical solutions for the gas inside the tube, the cooling water in the shell side, and the temperature in the tube wall are obtained, and the change rule of temperature with the time and the spacial position is determined in the heat exchange period. On this basis, a model on optimization of the cooler is proposed in which the objective function is the total entropy generation rating (TEGR), giving consideration to the two important factors, that is the heat transfer and the flow resistance in the cooler, and especially emphasising its research on energy quality. Finally, the results of optimization on the structure size and the operating parameters are obtained for the cooler by using the flexiable tolerance optimization method. Moreover, the experiments have proved the feasibility of the simulating model, and the reliability of the optimizating results.

MATHEMATICAL SIMULATION OF HEAT EXCHANGE PROCESS

A shell –and–tube cooler with baffles is shown schematically in Fig.1. Fig.2 shows the heat exchange model of the cooler, where high–temperature fluid is the air inside the tube, and the low–temperature fluid is the cooling water in the shell side. The following assumptions are made for the model.

(1) Thermal conductance is ignored;

(2) Thermal resistance of tube wall is ignored;

(3) The shall is adiabatic;

(4) The thermal capacity rates $(cm)_h$, and $(cm)_c$ is the two fluids are constant as are the thermal conductances $(hA)_h$, and $(hA)_c$, and the thermal capacity rate $(Cm)_w$ in the tube wall.

Based on the first law of thermodynamics, the heat exchange element of the cooler should meet the following equations[3]

For the tube wall,

$$(cm)_w \frac{\partial T_w}{\partial t} = (hA)_h (T_h - T_w) + (hA)_c (T_c - T_w) \tag{1a}$$

For the fluid inside the tube,

$$(hA)_h (T_w - T_h) = (A_x \rho c)_h U_h \left(T_h + \frac{\partial T_h}{\partial x} L \right) - (A_x \rho c)_h U_h T_h + (A_x \rho c)_h L \frac{\partial T_h}{\partial t} \tag{1b}$$

For the fluid in the shell side,

$$(hA)_c (T_w - T_c) = (A_x \rho c)_c U_c \left(T_c - \frac{\partial T_c}{\partial x} L \right) - (A_x \rho c)_c U_c T_c + (A_x \rho c)_c \frac{\partial T_c}{\partial t} \tag{1c}$$

The heat exchange coefficients, h_t, for the fluid inside the tube are separately calculated on such two cases, as the turbulent zone and the transitional zone. That is

when $R_{et} > 10^4$

$$h_t = 0.023 \frac{\lambda}{D_{ti}} R_{et}^{0.8} P_{rt}^{1/3} \left(\frac{\mu}{\mu_W} \right)^{0.14} \tag{2a}$$

when $2100 < R_{et} < 10^4$

$$h_t = h' + \frac{R_{et} - 2100}{10000 - R_{et}} \left[0.023 \frac{\lambda}{D_{ti}} R_{et}^{0.8} P_{rt}^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} - h' \right] \tag{2b}$$

$$h' = \left[3.66 + \frac{0.085 G_{zt}}{1 + 0.047 G_{zt}^{2/3}} \left(\frac{\mu}{\mu_w} \right)^{0.14} \right] \frac{\lambda}{D_{ti}}$$

In the present studies, the heat exchange coefficient in the shell side is calculated by Bell–Delaware method[4], in which the effect of the gap and pass–by caused by the baffles is in consideration.

$$h_s = h_i (J_e J_l J_b J_r J_s)$$

where h_i is the heat exchange coefficient of the tube core. J_c, J_l, J_b, J_r, and J_s are the revised factors

The initial condition of the equation is that

$$T_{wi}^{j} = T_{wi}^{j+z}$$

where z is the number divided equally in on period. For the boundary condition, the change of air temperature at the inlet with time is

$$T_{h\,in}^{j} = f(t)$$

and the cooling water temperature at the inlet is assumed to be a constant

$$T_{cin}^{j} = \text{const}$$

Having dispersed the equation group (1), the transient temperature change of the gas inside the tube with the time and spacial position is solved by using Jacobi iterative method, as shown in Fig.3. Thus , it is proved that the gas temperature of the inter–cooler in the reciprocating compressor is definitely not a constant, but is a pulsating value. The two peaks occurred in the figure is caused by the two cylinders in the compressor. Fig.4 shows the curved surface of transient temperature of the cooling water in the shell side. It is indicated that the temperature of cooling water slightly increases with the direction of the cooler length, but is independent on the time.

MATHEMATICAL MODEL OF COOLER OPTIMIZATION

Objective Function

At present there are many indexes to evaluate the performance of the heat exchanger. Among them the temperature efficiency, heat exchange efficiency, and the number of transfer units, et al., are often used. However, in these indexes the heat transfer is separated from the flow resistance, and the quality of the energy is not concerned. What index can be used to consider both the quantity and quality of the heat transfer and the flow resistance, and the optimum value can be searched? The authors consider that the total entropy generation rating is available. Thus, the TEGR is considered as the objective function in this optimizating model.

In order to evaluate the economy of the cooler, the TEGR as an index is a benefit, but with a certain expense. The benefit of the cooler is the heat exchange quantity, Q. The expense is the useful energy loss caused by the compressor and water pump and the heat transfer due to the temperature difference. Thus, the TEGR can be expressed by

$$Y_s = T_o[\Delta\dot{S}_{\Delta t} + n_t\Delta\dot{S}_{\Delta Pt} + n_s\Delta\dot{S}_{\Delta Ps}] / \dot{Q} \tag{3}$$

where

$\Delta\dot{S}_{\Delta t}$ — entropy generation rate caused by the temperature difference;

$$\Delta\dot{S}_{\Delta t} = C_{ps}\dot{M}_s ln(T_{so} / T_{si}) - C_{pt}\dot{M}_t ln(T_{ti} / T_{to})$$

$\Delta\dot{S}_{\Delta Pt}$ — entropy generation rate caused by the pressure loss inside tube;

$$\Delta\dot{S}_{\Delta Pt} = 2\dot{v}_t\Delta P_t / (T_{ti} + T_{to})$$

$\Delta\dot{S}_{\Delta Ps}$ — entropy generation rate caused by the pressure loss in the shell side;

$$\Delta\dot{S}_{\Delta Ps} = 2\dot{v}_s\Delta Ps / (T_{si} + T_{so})$$

n — conversion factor in which the power expense caused by the pressure loss is converted to the useful energy loss. The sub t, and s express the tube side and the shell side separately.

$$n_t = 1 / \eta_{ce} \cdot \eta_m \cdot \eta_c$$

$$n_S = 1/\eta_{ce} \cdot \eta_m \cdot \eta_p$$

where

η_{cc} — relative efficiency from Carnot circulation to effective circulation;

η_m — efficiency of electrical power system;

η_c — efficiency of compressor;

η_p — efficiency of pump;

$\dot{Q}$ — heat transfer quantity per unit time

Eq.(3) expresses the useful energy expended per unit heat transfer quantity in the cooler. The smaller the Y_s value, the better the economy of the cooler.

Strategic (Design) Variables

In the present work, the optimization on the cooler consists of the structure size optimization, and operating parameter optimization. For the former, the vector of the strategic variable can be expressed by

$$X_1 = [x_1, x_2, x_3, \cdots, x_8]^T = [D_{to}, L_{te}, L_g, D_S, L_{bc}, B_c N_b, N_{tt}]^T$$

where $B_c = L_{hcb} / D_S$,

$N_b = (L_{to} - 2L_{ts} - L_{bi} - L_{bo}) / L_{bc} - 1$

N_{tt} — number of tube

Other symbols are shown in Fig.1.

For the latter, the vector of the strategic variable is expressed by

$$X_2 = [x'_1, x'_2, x'_3]^T = [n, p, \dot{M}_s]^T$$

where n, p, are $\dot{M}_S$ are the rotatory speed of compressor, discharge pressure, and the flow quantity of cooling water, separately.

Restrained Condition

(1) The restrained condition for optimization structure size of the cooler:

for the inequality restraint

$g_i(X_1) \geqslant 0 \qquad i = 1,2,\cdots,18$

$$g_1 = \frac{20}{D_{to}} - 1.0 \qquad g_2 = \frac{D_{to}}{5} - 1.0 \qquad g_3 = \frac{L_{to}}{D_S} - 2.5$$

$$g_4 = 1.5 - \frac{L_{tp}}{D_{to}} \qquad g_5 = \frac{L_{tp}}{D_{to}} - 1.15 \qquad g_6 = \frac{Z_{oo}}{D_s} - 1.0$$

$$g_7 = \frac{D_s}{40} - 1.0 \qquad g_8 = 15 - \frac{D_s}{D_{to}} \qquad g_9 = \frac{D_S}{D_{to}} - 5.0$$

$$g_{10} = \frac{D_s}{L_{bc}} - 1.0 \qquad g_{11} = \frac{L_{bc}}{D_s} - 0.20 \qquad g_{12} = \frac{0.5}{B_c} - 1.0$$

$$g_{13} = \frac{B_c}{0.10} - 1.0 \qquad g_{14} = \frac{0.78 D_{otl}^2}{c_1 L_{tp}^2} - N_{tt} \qquad g_{15} = \frac{N_{tt}}{7} - 1.0$$

$$g_{16} = \frac{\Delta P_{tmax}}{\Delta P_t} - 1.0 \qquad g_{17} = \frac{\Delta P_{smax}}{\Delta P_S} - 1.0 \qquad g_{18} = \frac{Q}{Q_{min}} - 1.0$$

for the equality restraint

$h_j(X_1) = 0 \qquad j = 1$

$$h_1 = N_b + 1 - \frac{L_{to} - 2L_{ts} - L_{bi} - L_{bo}}{L_{bc}}$$

(2) The restrained condition for optimizating operating parametr of the cooler:

for the inequality restraint

$g_i(X_2) \geq 0 \qquad i = 1,2,\cdots,8$

$$g_1 = \frac{600}{n} - 1.0 \qquad g_2 = \frac{n}{100} - 1.0 \qquad g_3 = \frac{8}{p} - 1.0 \qquad g_4 = p - 1.0$$

$$g_5 = \frac{25}{\dot{M}_s} - 1.0 \qquad g_6 = \frac{\dot{M}_s}{25} \qquad g_7 = \frac{\Delta P_{tmax}}{\Delta P_t} - 1.0 \qquad g_8 = \frac{\Delta P_{smax}}{\Delta P_s} - 1.0$$

Pressure Loss

In the tube side, the pressure loss is the sum of the pressure loss during the gas going through the straight tube part, and the pressure loss at the inlet and outlet. The expression is that

$$\Delta P_t = \frac{4f_t \dot{M}_t L_{to} N_{tp}}{2\rho_t D_{ti}} \left(\frac{\mu}{\mu_w}\right)^{-0.14} + 1.5\frac{\dot{M}_N^2}{2\rho_t}$$

In the shell side the pressure loss includes the pressure loss (ΔP_c) of the pure transverse flow in the zone between the tops of the baffles, the pressure loss (ΔP_w) in the window of the baffles, and the pressure loss (ΔP_l) at the end of the cooler, as shown in Fig.5. The expression is that[4]

$$\Delta P_s = \Delta P_{bi}(N_b - 1)R_b R_l + N_b\left[(2 + 0.6N_{tcw})\frac{\dot{m}_w}{2\rho_s} \times 10^{-3}\right]R_l$$
$$+ \Delta P_{bi}\left(1 + \frac{N_{tcw}}{N_{tcc}}\right)R_b R_s$$

where ΔP_{bi} is the ideal pressure loss, R_b is the revised factor, and R_e is the leakage factor.

Results of the Optimization

For the cooler size, as listed in Table 1, the operating parameters are optimized by the flexiable tolerance optimization method. The results of the optimization is shown in Fig.6 and Table 2. The optimum of the TEGR is 6.5417. Compared with the initial scheme, the value of the TEGR is reduced by a factor of 45.21%, and the pressure loss is decreased in the tube side. In addition, the perfectness K_{tt} of cooling is increased from 79.24% to 89.15% by using the optimization. It is clear that optimizing the operating parameters can lead to saving energy for the compressor. In order to verify the reliability of the calculation; the experiment has been conducted in a later–cooler of an air compressor type Ⅱ ZA–1.5 / 8. The influence of the rotatory speed (n), cooling water quantity (M_s), and pressure ratio (E) on the TEGR and the heat exchange quantity are shown in Fig.7–9 separately. The calculation values are in agreement with the experimental results, thus it is proved that the optimizing procedure is correct.

Table 1 cooler size

Size parameter	D_o (mm)	D_S (mm)	L_{tp} (mm)	L_{bc} (mm)	L_{to} (mm)	N_{tt} (mm)	N_b	B_C	L_{bi} (mm)	L_{bo} (mm)
value	11	65	13	48	678	14	13	0.26	74	68

Table 2 The results of optimization for the operating parameters

Parameter	M_s (kg / s)	n (r / min)	E	u (W / m² · k)	Q (W)	Y_s	ΔP_t (KPa)
Initial Scheme	0.1667	330	5.0	163.6	1159.7	11.937	4.217
Optimum results	0.1737	250	5.99	188.9	1194.3	6.540	3.033

Parameter	ΔP_s (KPa)	T_{ti} (℃)	T_{to} (℃)	T_{si} (℃)	T_{so} (℃)	K_{tt}	/
Initial Scheme	0.199	118	25.6	19.2	19.5	79.24	/
Optimum results	0.208	113	23.5	18.2	19.6	89.15	/

The results of the optimization on the structural size of the cooler are shown in Fig.10, and Table3. The value of TEGR is reduced by a gactor of 83.39%, the useful energy loss is reduced by a factor of 63.47%, and the perfectness K_{tt} of cooling is increased by 19.54%. By using the optimization, the purpose of enhancing heat exchange and decreasing heat resistance is reached. Due to the limitation of the experimental condition, the optimùm results on the structural size have not been verified.

Table 3 The results of optimization for the structural size

Parameter	D_{to} (mm)	D_s (mm)	L_{tp} (mm)	L_{bc} (mm)	L_{to} (mm)	N_{tt}	N_b	B_c	u (W / m² · k)	Q (w)
Initial Scheme	11	65	13	48	678	14	13	0.26	157.0	1161.7
Optimum results	13	90	15	36	605	16	13	0.24	201.4	2547.4

Parameter	Y_s	ΔP_t (KPa)	ΔP_s (KPa)	T_{ti} (℃)	T_{to} (℃)	T_{si} (℃)	T_{so} (℃)	K_{tt}
Initial Scheml	7.453	3.239	0.215	139	26.0	18.2	19.9	73.8
Optimum results	1.242	0.794	0.038	139	21.5	18.2	21.9	93.4

CONCLUSION

By means of the experiments carried out in a cooler of a laboratory compressor, the transient temperature at the inlet and outlet of the cooler and the performance parameters of the compressor have been measured. It is proved that the model on the transient heat transfer is viable, and the optimizing results in which the objective function is the total entropy generation rating is correct.

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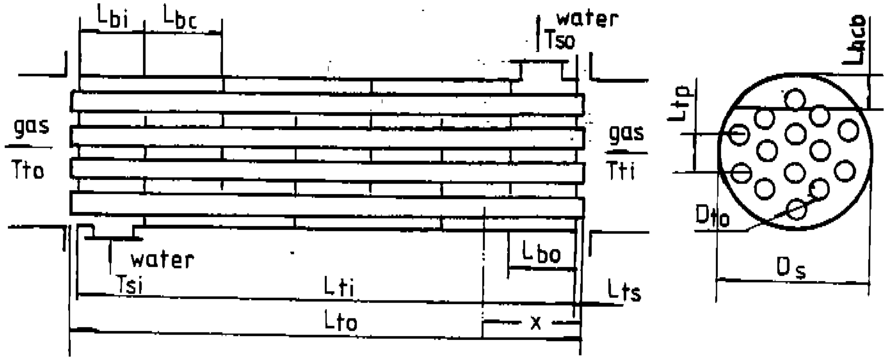

Fig.1 Schematic of the shell–and–tube cooler

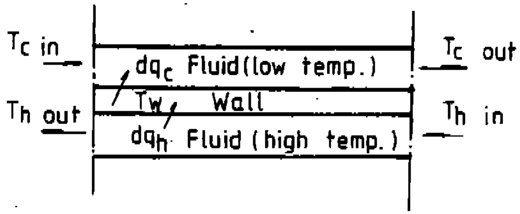

Fig.2 Heat exchange model of the cooler

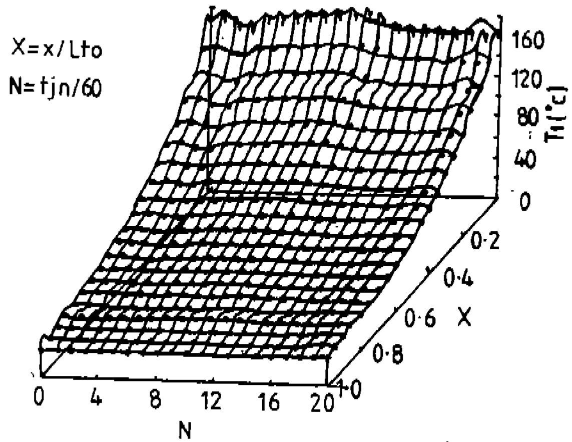

Fig.3 Curved surface of the transient temperature of the gas in the tube side with the change of time and spacial position

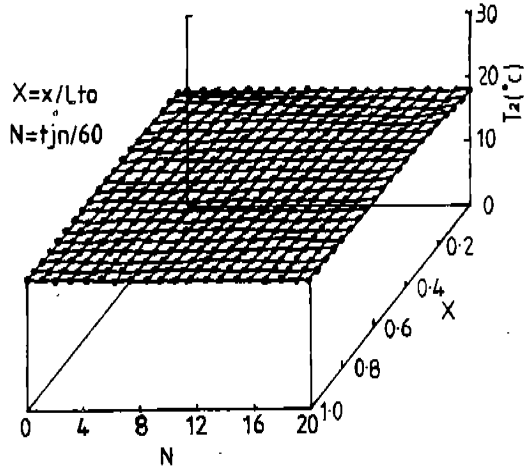

Fig.4 Curved surface of the transient temperature of the cooling water in the shell side with the change of time and spacial position

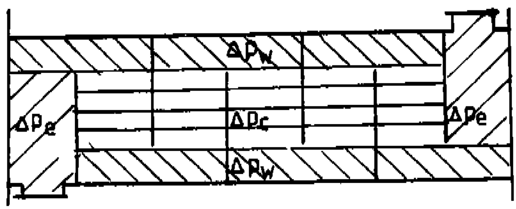

Fig.5 Distribution of pressure loss in the shell side

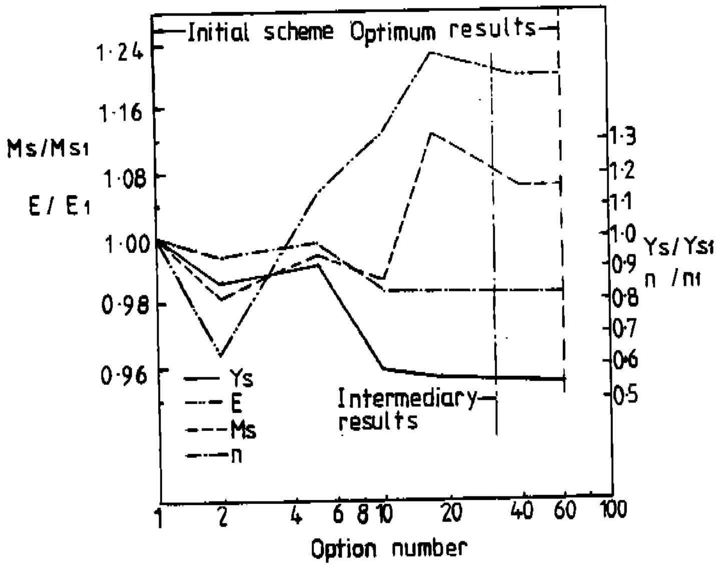

Fig.6 Optimizing results of operating parameters

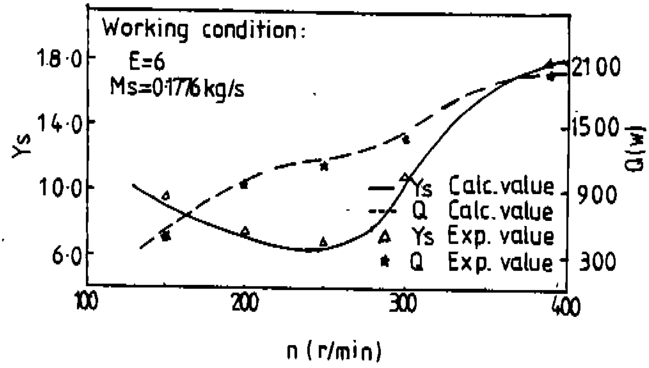

Fig.7 Relationship between Y_s and n

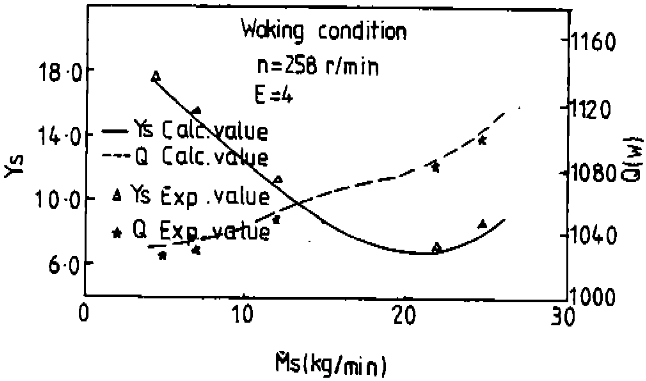

Fig.8 Relationship between Y_s and M_s

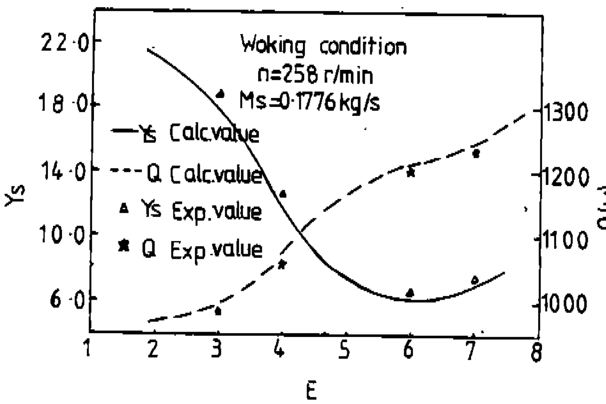

Fig.9 Relationshop between Y_s and E

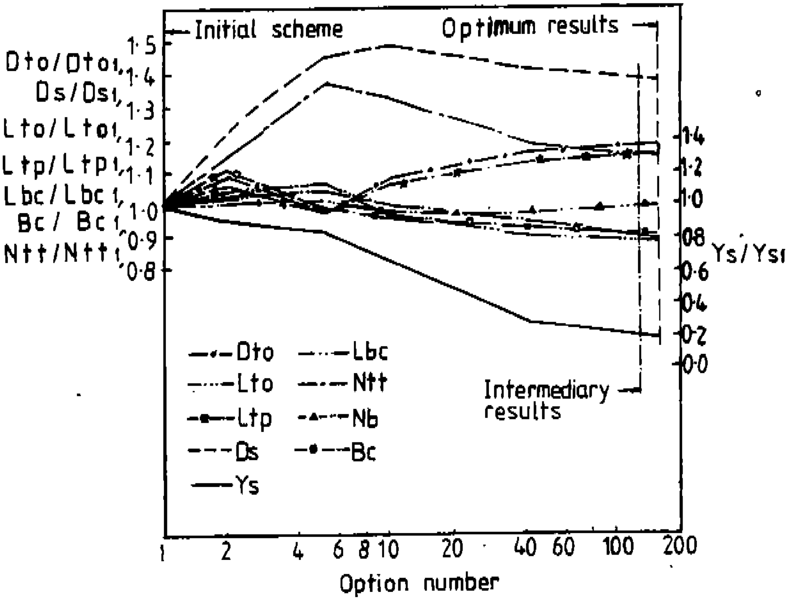

Fig.10 Optimizing results of structural size