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Performance Optimization of a 1

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PERFORMANCE OPTIMIZATION OF
A 1.5 TON HERMETICALLY SEALED COMPRESSOR THROUGH PARAMETER DESIGN

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ABSTRACT

Optimizing the performance of a product by selecting proper design variants is an essential part of any product design exercise. This paper describes the compressor simulation model and complete parameter design procedure for optimizing the performance of a 1.5 ton single cylinder reciprocating compressor. A computer model incorporating the thermodynamic process, gas flow dynamics, heat transfer and valve dynamics was developed to simulate the performance for the given design variants.

Eleven design variants were considered for optimizing the cooling capacity, power, EER (**Energy Efficiency Ratio**) and discharge superheat. The L 36 Orthogonal Array table was used for making combinations of the design variants having three levels each. Analysis of variance and level-wise Signal to Noise ratios were used for determining parameters which give best compressor performance. Actual prototype compressors as per the optimized design were built and tested. The test results showed 10% improvement in the overall performance.

INTRODUCTION

Computer simulation models to predict the hermetic compressor performance have been developed over the last two decades. However, it is difficult to obtain the best design by simply using a simulation model, due to the complex nature of the many parameters involved. Attempts have been made to evolve optimization methods to get the best design. Analytical methods and gradient techniques cannot be used directly because the transfer function is so complicated that an exact expression cannot be found for it. The performance optimization through parameter design method is very convenient when the transfer function is not in the form of a simple equation. This method utilizes the orthogonal array technique, analysis of variance, signal to noise ratios and performance criteria.

PERFORMANCE SIMULATION MODEL

The computer model used in the present study was developed based on in-house technology and the work reported in references (1), (2) and (3). The assumptions made for analysis are that the flow is one dimensional, gas follows perfect gas law relationship and uniform cylinder properties at any instant of time. The control volume is considered to be an open system, shown in Fig.1, with suction valve as one flow boundary and discharge valve as another boundary with both work and heat transfer across the boundary. First law of thermodynamics in its rate form is -

$$m\,Cv\,\frac{dT}{dt} + \frac{mRT}{V}\,\frac{dV}{dt} + K\,Cv\,Td\,\frac{dmd}{dt}$$

$$- K\,Cv\,Ts\,\frac{dms}{dt} + Cv\,T\,\frac{dm}{dt} - \frac{dQ}{dt} = \text{————————————} \quad (1)$$

Where m-mass of gas, Cv-specific heat at constant volume, T-temperature, t-time, R-gas constant, V-volume of gas, K-ratio of specific heats, Td-disch.gas temp., md-mass of disch.gas, Ts-suction gas temp., ms-mass of suction gas and Q-heat transfer. In this equation the unknown quantities are m(t), T(t), V(t), $\frac{dmd}{dt}$, $\frac{dms}{dt}$ and $\frac{dQ}{dt}$. From valve flow model $\frac{dms}{dt}$, $\frac{dmd}{dt}$ and m(t) are determined, from Kinematics model V(t) is calculated and from heat transfer model $\frac{dQ}{dt}$, is evaluated. Since it is assumed that suction and disch. processes are constant both Ts and Td are known. T(t) can be obtained by solving equation (1). The pressure inside the cylinder p(t) is obtained by using the perfect gas equation -

$$P(t) = \frac{m(t)}{V(t)} R\, T(t) \text{ ----------------- (2)}$$

The temperature of the refrigerant in the shell is determined from heat balance equation (Fig. 2) -

$$Tg = Ta + \left[\frac{(Wi + Wl)}{\eta m} - \dot{m}\,(cpd\ Td - cps\ Ts)\right] /(A\lambda) \text{ ---------- (3)}$$

Where Tg-temperature of the gas in the shell, Ta-ambient temperature, Wi-indicated work, Wl-mechanical work, ηm-motor efficiency, $\dot{m}$-rate of gas flow, cps-specific heat of suction gas, cpd-specific heat of discharge gas, A-area of compressor shell, λ-heat transfer co-efficient.

To start with, the temperature of the refrigerant inside the shell is assumed and then temperature and pressure inside the cylinder are calculated at each crank angle. This process is repeated until convergence of temperature and pressure are achieved. The predicted P-V diagram for the present compressor is shown compared with the measured P-V diagram, reference (4), in Fig. 3. The agreement is very close. This simulation model can be used for analysing the performance for the given dimensions of the compressor.

PARAMETER DESIGN PROCEDURE

Selection of Design Variants and Levels

Based on technological considerations the eleven parameters shown in Table-1 were selected as design variants and each design variant is having three levels. The levels of these variants are as wide as possible in the feasibility zone such that the non-linearity zones are explored.

Table 1 : Design variants and their levels

S No	Variant	Level-1	Level-2	Level-3
1	Crankshaft throw, Cm, CR	1.15	1.30	1.45
2	Connecting rod length, Cm, CL	5.0	5.5	6.0
3	Cylinder bore diameter, Cm, DC	3.8	4.5	5.2
4	Clearance volume, Cm^3, VC	1.0	2.0	3.0
5	Suction port diameter, Cm, SPD	0.6	0.7	0.8
6	Disch. port diameter, Cm, DPD	0.6	0.7	0.8
7	Suction valve thickness, Cm, SVT	0.033	0.038	0.043
8	Disch. valve thickness, Cm, DVT	0.033	0.038	0.043
9	Suction valve lift, Cm, SMT	0.15	0.20	0.25
10	Discharge valve lift, Cm, DMT	0.15	0.20	0.25
11	Valve plate thickness, Cm, VPT	0.4	0.6	0.8

Primary Orthogonal Array Table

The eleven design variants are allocated to eleven columns in L36 OA table as shown in Table 2. This OA table where design variants are assigned is called PRIMARY OA table and in this there are 36 different combinations of parameters. L36 table was chosen because all the factors are at 3 levels and the interaction between any two columns is equally distributed.

Table 2 : Primary Orthogonal Array table

Variant	1	2	3	4	5	6	7	8	9	10	11	12	13
Expt No.	CR	CL	DC	VC		SPD	DPD	SVT	DVT	SMT	DMT	VPT	
1	1	1	1	1		1	1	1	1	1	1	1	
4	1	1	1	1		2	2	2	3	3	[illegible]	3	
10	1	1	3	2		3	2	3	2	1	3	2	
15	3	1	2	3		1	2	3	3	1	3	1	
20	2	3	2	1		1	2	3	3	2	3	1	
25	1	3	2	1		3	3	1	3	1	2	3	
30	3	2	1	1		3	3	2	1	2	3	2	
36	3	2	3	1		1	2	3	1	1	2	3	

Responses and level selection criteria

The following four responses and corresponding performance criteria were chosen for selecting the level of a design variant. The response data for each level is expressed in a single measure known as Signal to Noise Ratio (SN ratio). [Reference (5) and (6)]. This measure meets the performance criterion and also reduces its variation as shown in Table 3.

Table 3 : Responses and SN Ratios

Sl. No.	Response	Performance criteria	SN Ratio formula
1	Cooling capacity	Nominal the better	$SNN = 10 \log_{10}(\frac{\bar{Y}^2}{S^2})$ Where, $\bar{Y} = \frac{1}{n} \sum_{i=1}^{n} yi$ $S = \left[\frac{1}{n-1} \sum_{i=1}^{n} (yi - y)^2\right]^{\frac{1}{2}}$ yi = Response of ith experiment n = No. of experiments
2	Input power	Lower the better	$SNL = -10 \log_{10}\left[\frac{1}{n} \sum_{i=1}^{n} yi^2\right]$
3	E E R	Higher the better	$SNH = -10 \log_{10}\left[\sum_{i=1}^{n} (\frac{1}{yi})^2\right]$
4	Disch. gas temp.	Lower the better	SNL

ANOVA table and Significance table

Significance of each design variant in the Primary OA with respect to each response was determined by using analysis of variance (ANOVA) technique shown for cooling capacity in Table 4.

Table 4 : ANOVA table for Cooling Capacity
F - Ratio at 5 percent = 3.39
F - Ratio at 1 percent = 5.57

Sl. No.	Variant	DF	SS	MS	F.CAL.	SIGNIFICANCE
1	CR	2	0.377E08	0.188E08	12.732	XX
2	DC	2	0.131E09	0.657E08	44.357	XX
3	SPD	2	0.483E08	0.241E08	16.300	XX
4	DVT	2	0.591E07	0.295E07	1.993	
5	SMT	2	0.117E08	0.587E07	3.960	X
6	ERROR	25	0.370E08	0.148E07		
	TOTAL	35	0.272E09			

Where DF-degrees of freedom, SS-sum of square, MS-mean square, F.CAL-'F' calculated, X-significant at 5 percent, XX-significant at 1 percent.

Using ANOVA table and level-wise SN ratios calculated from Table 3, a significance table was prepared which is shown in Table 5.

Table 5 : Significance Table

Sl. No.	Variant	Cooling capacity	Input power	EER	Disch.gas temp.	Feasible combinations
1	CR	3	1	1	1	1,3
2	CL	NS	NS	NS	NS	2
3	DC	3	1	1	1	1,3
4	VC	2	2	NS	NS	2
5	SPD	3	3	3	NS	3
6	DPD	NS	NS	3	NS	3
7	SVT	NS	NS	NS	NS	2
8	DVT	NS	NS	1	NS	1
9	SMT	3	NS	3	NS	3
10	DMT	NS	NS	NS	NS	2
11	VPT	1	1	1	1	1

Where NS - Not significant

Selection of Feasible and Optimum combinations

From the significance table, four feasible combinations were selected and performance parameters for these combinations were predicted using the computer simulation program as shown in Table 6.

Table 6 : Performance of Feasible Combinations

Combination	Cooling capacity-KCal/hr	Input power Watts	E E R KCals/Whr.	Disch.gas temp. Deg. C.
1	3090	1252	2.47	112
2	4500	1956	2.30	120
3	4020	1827	2.20	123
4	4230	1967	2.15	127

From this table, it is observed that the combinations 1 and 2 give the highest EER and lowest disch. gas temperature. However, in both cases the cooling capacity is lower than the required value of 4735 KCals/hr. To bring the cooling capacity to the nominal value, the design variant which is contributing most is identified by calculating contribution ratio and such variant is varied accordingly to achieve the target value. The contribution ratio (CR) is calculated from the equation.

$$CR = \frac{SS - DF \times MSE}{ST} \times 100$$

where ST - Total sum of squares, MSE - Error mean square.

EXPERIMENTAL VERIFICATION OF PARAMETER DESIGN

Actual prototype compressors as per the optimised design were built and tested at normal load conditions of ASHRAE standard in a calorimeter test rig. Table 7 shows a comparison of the performances of existing design and optimized design. As seen from the table, the tested results compare well with the predicted values. Compared to existing design, there is a reduction in input power and discharge gas temperature and also a ten percent improvement in EER.

Table 7 : Performance Comparison

Design	Cooling capacity KCal/hr	Input power Watts	E E R KCal/Whr	Disch.gas temp.-Deg.C
Existing design (tested)	4700	2200	2.14	130
Optimized design (tested)	4710	2020	2.35	125
Optimized design (predicted)	4735	2000	2.37	123

CONCLUSIONS

The transfer function in the form of computer program is predicting the performance characteristics satisfactorily. The concepts of parameter design are successfully applied for optimizing the hermetic compressor performance. Reduction in energy levels and improvement in EER were achieved through re-design of the compressor. The program can be further used to build a **ROBUST COMPRESSOR.**

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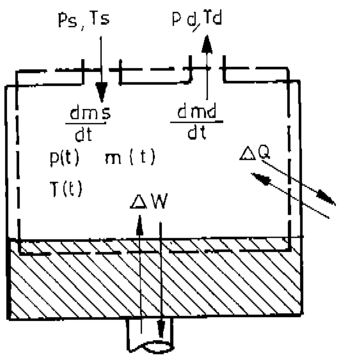

Figure 1 : Control volume

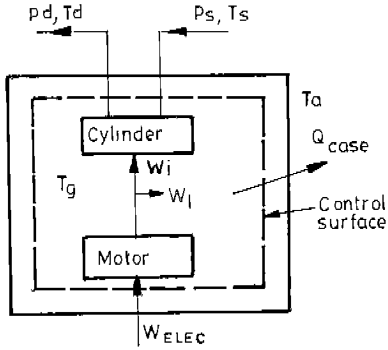

Figure 2: Heat balance

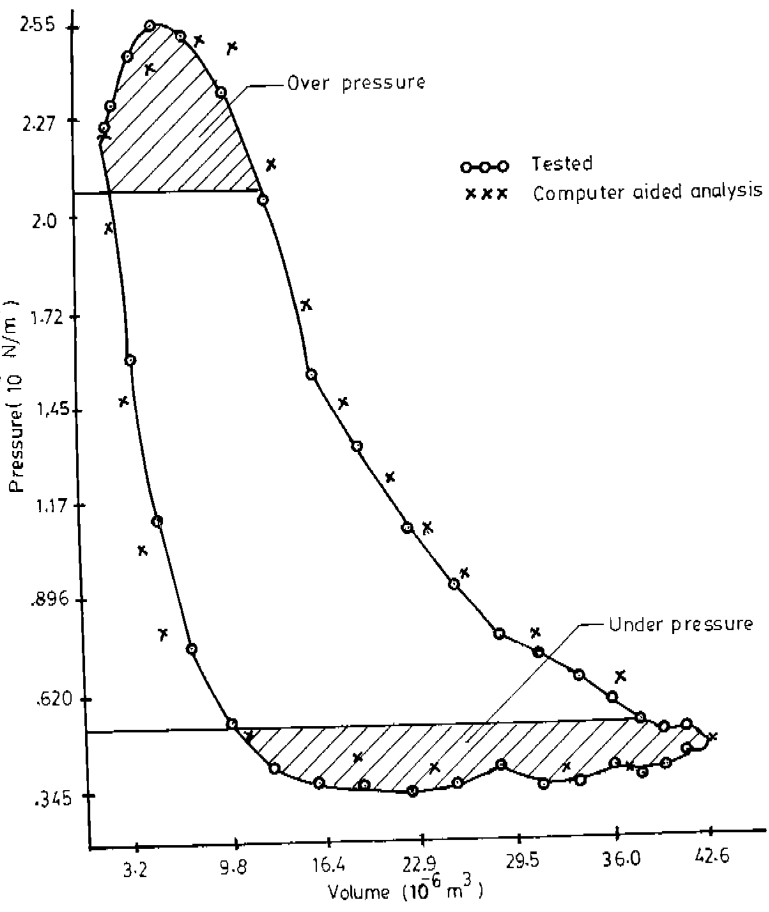

Figure 3: P-V Diagram