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AN INTEGRATED APPROACH TO RECIPROCATING COMPRESSOR DESIGN

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A B S T R A C T

A typical reciprocating compressor consists of a motor, a pump, suction - discharge system and a lubrication system. An integrated computer simulation model incorporating all these components simultaneously was developed to predict, evaluate and optimize the overall performance and reliability.

For the given design variants of a compressor, the computer program first calculates the mass flow rate, indicated work, gas loads, thermodynamic parameters using pump and suction - discharge models. Next, the program evaluates the bearing performance and calculates also the total frictional and lubrication loads on the shaft. Using the indicated work, frictional loads, the program analyses the motor performance and calculates heat generation. Then the program invokes the overall heat balance model of the compressor to calculate winding temperature, gas temperature in the shell and the suction gas entry temperature. The whole process is performed iteratively until converged results are obtained.

A low back pressure compressor was redesigned using this integrated computer model approach to enhance the performance and to improve the reliability. Actual prototype compressors as per this design were built and tested. The test results showed considerable improvements in performance and reliability.

I N T R O D U C T I O N

The ever growing demand at market place for higher efficiency, greater reliability and lower cost compressors and the inherent opportunity for a competitive edge have activated the manufacturers to develop state-of-the-art analytical tools to predict, evaluate and optimize the existing as well as new designs. In the last two decades considerable number of such tools have been published to simulate the overall performance of reciprocating, rotary, scroll and screw type compressors, for example, references [1], [2], [3] and [4].

The present integrated model of reciprocating compressor design includes -

1. Thermodynamic process, heat transfer and valve dynamics of the pump.
2. Gas flow dynamics of the suction-discharge systems
3. 3-dimensional hydrodynamic model for evaluating bearing performance and a semi-empirical model for evaluating lubrication performance.
4. A comprehensive model to analyse the hermetic motor performance.
5. And, finally an overall energy balance model incorporating heat transfer effects.

The general organization of the integrated simulation model is shown in Fig. 1.

Analytical methods and gradient techniques to optimize the design 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. The present model incorporates this parameter design method which utilises orthogonal array technique, analysis of variance, signal to noise ratios and performance criteria.

PUMP SIMULATION MODEL

The pump simulation model used in the present study is reported in reference [5]. The control volume is considered to be an open system 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

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

$$- K\,Cv\,Ts\,\frac{dms}{dt} + Cv\,T\,\frac{dm}{dt} - \frac{dQ}{dt} = 0 \qquad \text{----- (1)}$$

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 temperature, 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), dmd/dt, dms/dt and dQ/dt. From valve flow model dms/dt, dmd/dt and m(t) are determined, from kinematics model V(t) is calculated and from heat transfer model dQ/dt is evaluated.

SUCTION - DISCHARGE MODEL

The present model incorporates the design and analysis of reflective, concentric tube, and extended tube types of resonators and cross flow expansion, and cross flow contraction types of mufflers both on suction and discharge side. The model gives both acoustic performance and pressure loss in the suction - discharge mufflers. The acoustic model is based on the work reported in reference [6] and the pressure losses are calculated using a semi-empirical model developed by the company.

BEARING AND LUBRICATION ANALYSIS

Steady flow of an incompressible fluid with constant physical properties in a three-dimensional journal bearing is considered. The cartesian scheme of coordinates is illustrated in Fig. 2. The Reynolds equations of mean motion for this case are [7] (considering only laminar flow which prevails in this case)

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{\mu}{\rho}\frac{\partial^2 u}{\partial y^2} \qquad \text{-----} \ (2)$$

$$u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z} = -\frac{1}{\rho}\frac{\partial p}{\partial z} + \frac{\mu}{\rho}\frac{\partial^2 w}{\partial y^2} \qquad \text{-----} \ (3)$$

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \qquad \text{-----} \ (4)$$

With the boundary conditions

$$u\ (x,o,z) = U,\ v\ (x,0,z) = O,\ W\ (x,o,z) = O$$

$$u\ (x,h,z) = O,\ v\ (x,h,z) = O,\ W\ (x,h,z) = O$$

$$P\ (x,o) = O,\ p\ (x,b) = O \qquad \text{-----} \ (5)$$

where u,v,w are mean velocities in x,y,z directions, p is the pressure, μ is the viscosity and ρ is the density. The solution of the governing equations will yield velocity and pressure fields and the solution is obtained by an iterative numerical method the details of which are given in reference [7]. The load capacity and frictional drag are obtained from

$$W^{+} = \left[Wx^{+2} + Wy^{+2} \right]^{\frac{1}{2}}$$

$$Wx^{+} = \int_{0}^{L} \int_{0}^{2\pi} p \cos\theta \, dx \, dz \qquad \text{------ (6)}$$

$$Wy^{+} = \int_{0}^{L} \int_{0}^{2\pi} p \sin\theta \, dx \, dz$$

$$D^{+} = \int_{0}^{L} \int_{0}^{2\pi} \mu \left. \frac{\partial u}{\partial y} \right|_{y = 0} dx \, dz \qquad \text{------ (7)}$$

where θ is the angle between the line of centres and any radial line.

For a given load (both gas load and accelerating forces) the program calculates the bearing eccentricity, quantity of oil flow required and frictional drag. The criterion for successful operation of the bearing is that the eccentricity ratio must be less than 0.8.

Semi-empirical relations are developed for calculating the lubrication oil pumped by the crankshaft from the oil sump and the oil flow through the oil grooves and through the bearing areas. These are checked against the oil flow required from the above bearing program. Semi-empirical relations are also developed to calculate the frictional loss between the piston and cylinder bore and in the wrist pin bearing.

HERMETIC MOTOR ANALYSIS

The motor model can analyse the hermetic motors designed for RSIR/CSIR/PSC/CSR electrical circuit operation. The simulation flow chart is shown in Fig. 3. For the given lamination design and winding distribution the program first calculates resistance, magnetic circuit, magnetic flux, reactance, including required constants and losses. Then the motor performance is calculated. The no load, torque speed characteristics, and locked rotor performance etc. are calculated using the method described in reference [8]. A semi-empirical method is developed to calculate the winding temperatures based on power loss in the motor, amount of gas cooling and gas temperature.

ENERGY BALANCE MODEL

The objective of the overall energy balance model is to determine the suction gas temperature iteratively so that the density, mass flow rate and compressor cooling capacity can be calculated. The energy balance model is similar to the models given in references [5] and [3].

To start with, the suction gas temperature is assumed and then using pump model, suction-discharge model and bearing model the gas load and mechanical frictional losses are calculated. The gas load and mechanical frictional losses are considered as the shaft output of the motor. The motor input to deliver this shaft output is calculatd using the motor simulation model. The heat generated by the motor loss, the mechanical friction, and the heat transferred by the discharge gas are combined into a internal heat source. This heat is partitioned into the heat gained by the suction gas and heat transferred by the shell to the ambient. A semi-empirical heat transfer model was developed to incorporate this. Using the energy balance model, a new suction gas temperature is calculated and the process is repeated until converged performance is obtained.

PARAMETER DESIGN PROCEDURE

The detailed parameter design procedure for optimizing pump design is described in reference [5]. Since procedure for optimizing the motor is similar to that of pump design details are not given here. Based on technological considerations the eleven parameters shown in Table 1, were selected as design variants for motor design. These design variants are having three levels each. An L36 orthogonal table is used in this case also.

Table 1. Design variants - Motor Design

S.No.	Variant
1	Stack height
2	Run capacitor
3	Main winding wire gauge
4	Auxiliary winding wire gauge
5	Main winding - No. of turns - coil 1
6	Main winding - No. of turns - coil 2
7	Main winding - No. of turns - coil 3
8	Main winding - No. of turns - coil 4
9	Aux. winding - No. of turns - coil 1
10	Aux. winding - No. of turns - coil 2
11	Aux. winding - No. of turns - coil 3

The following five responses, at each of the three loads (ASHRAE condition, less severe than ASHRAE and more severe than ASHRAE) were considered simultaneously. Totally, there will be 15 responses that are to be considered for selecting the level of a design variant.

Table 2: Responses and performance criteria

S.No.	Response	Performance criteria
1	Efficiency	Higher the better
2	Input power	Lower the better
3	Current	Lower the better
4	Winding temperature	Lower the better
5	Torque	Higher the better

Using performance criteria, Signal to Noise Ratio and analysis of variance, level of each design variant was chosen to give optimum performance (Reference [5]).

EXPERIMENTAL VERIFICATION OF THE INTEGRATED MODEL

One of the compressors in our Low Back pressure range was stated to be giving slightly lower EER and slightly higher shell temperature compared to the competition in the field though it was meeting the buyer's specifications. Also, close examination of the inwarranty compressors revealed that in some cases connecting rod big end was wearing out, as shown in Fig. 4, though the customers were not complaining about it. Initial design review revealed that an integrated approach is required to improve the design and it was decided to press the integrated simulation model into service. Analysis of the existing design revealed that the valving and ports need to be optimized for improving EER and suction-discharge system is already at a optimum level and need not be considered for optimization. Bearing and lubrication analysis indicated that the conneting rod big end is wearing out under certain adverse working conditions due to excessive bearing clearance and also due to inadequate lubricant supply. Similarly, from the motor analysis, it was clear that motor optimization is required to improve EER and to reduce winding temperatures and inturn shell temperature. The pump and motor are optimized using the parameter design procedure described earlier. Table 3 shows the list of modifications carried out on the compressor. Actual prototype compressors as per the optimized design were built and tested at normal load conditions of ASHRAE standard in a calorimeter test rig. Table 4 shows a comparison of the performance of existing design and optimized design. As seen from the table, the tested results compare well with the predicted values.

Table 3. List of Design Modifications

1. Crank shaft throw
2. Suction port area
3. Suction valve thickness
4. Location of Suction Entry
5. Crank pin bearing clearance
6. Crank shaft length
7. Motor winding distribution
8. Motor circuit is changed from CSIR to CSR

Table 4. Performance comparison

Parameter	Existing design (Tested)	Optimized design	
		Predicted	Tested
Cooling capacity Kcal/hr	340	355	352
Input power Watts	400	338	325
Input current Amps	2.8	1.67	1.60
E E R Kcal/Whr	0.85	1.05	1.08
Max. winding temp. Deg C	125	107	111
Shell temp. Deg C	64	54	57

Compared to existing design, there is an improvement in EER by 25 percent and there is a reduction in winding and shell temperatures. Life tests and field tests indicated that the connecting rod big end wear out is totally eliminated thus improving the reliability.

CONCLUSIONS

The integrated simulation model is predicting the performance characteristics satisfactorily. The concepts of parameter design are successfully applied for optimizing both the pump and hermetic motor performance. The general validation of the integrated approach and its value to predict, evaluate and optimize the overall performance and reliability have been confirmed on reciprocating compressor design. This suggested approach could be the basis for building up "EXPERT SYSTEMS" for designing hermetic compressors - more work is needed to extend the design approach to other than reciprocating compressors like rotary, scroll and screw type of compressors.

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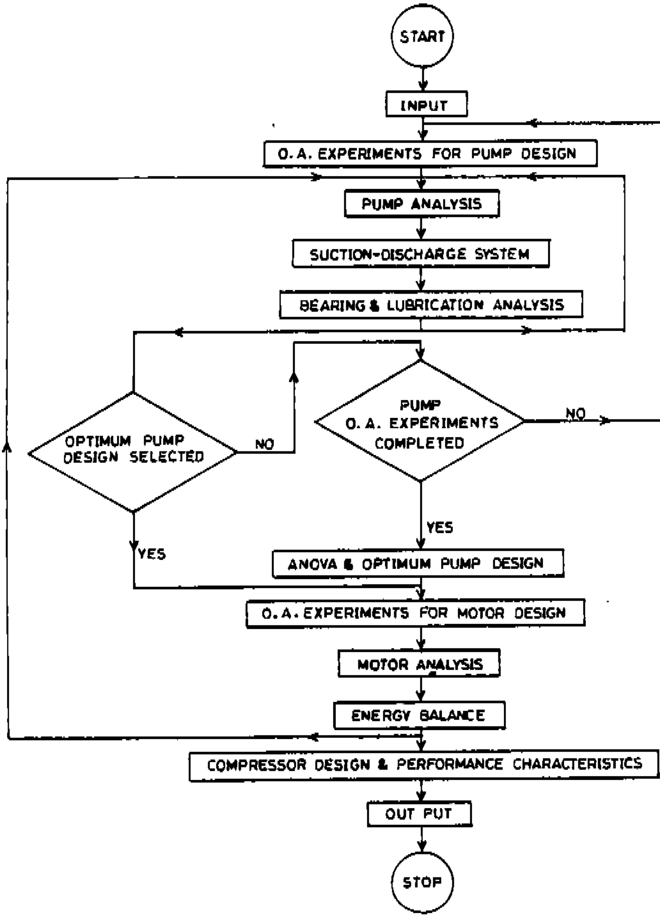

FIG. 1. INTEGRATED SIMULATION MODEL

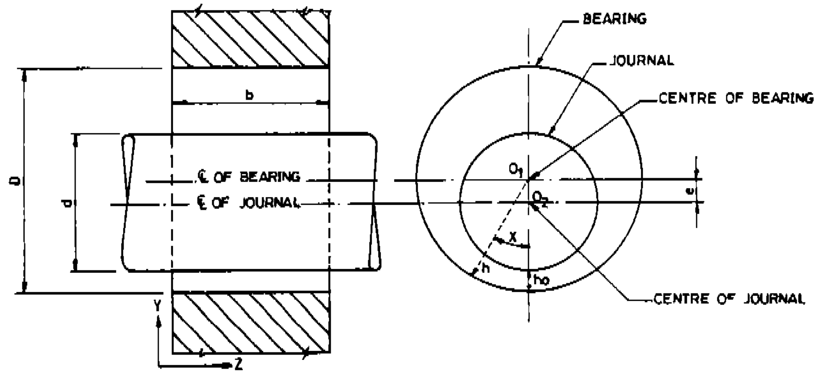

FIG. 2 SCHEME OF CO-ORDINATES - 3D BEARING

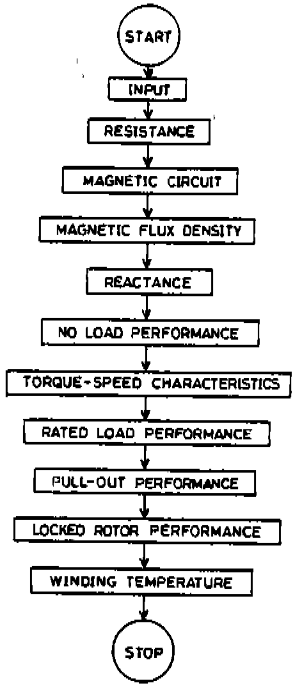

FIG. 3. HERMETIC MOTOR SIMULATION.

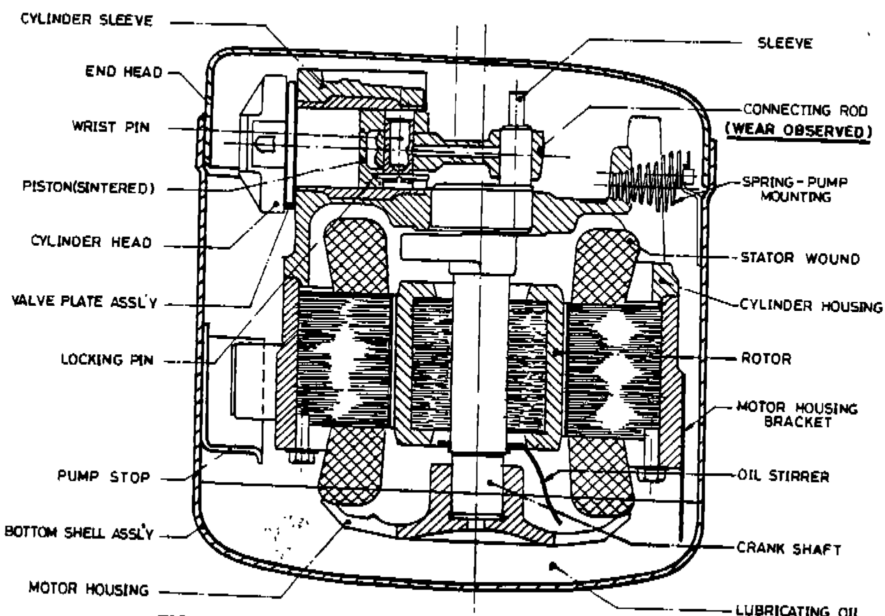

FIG. 4. LOW BACKPRESSURE COMPRESSOR