

OPTIMIZING RECIPROCATING AIR COMPRESSORS DESIGN PARAMETERS BASED ON FIRST LAW ANALYSIS

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Reciprocating compressors are used widely in industry due to their ability to deliver high pressure compressed gas. Understanding the behavior of the reciprocating compressors and studying effects of various parameters are interesting subjects. The mathematical modeling is proved to be an effective tool to study performance of the reciprocating compressors. Here a mathematical model has been developed based on the conservation of mass and energy. The suction and discharge valves have been also modeled. The model could predict in-cylinder pressure, in-cylinder temperature and valves motions at various crank angles. The indicator work and deliver air are also calculated. The results from the developed model have been validated against the previous measured values. The effects of angular speeds, clearance, entrance gas temperature and discharge and suction value areas on the performance of the compressors have been investigated. An optimum value for suction to discharge valve ratio has been found.

Keywords: Reciprocating compressor, Thermodynamic, First law, Optimization.

1. Introduction

Reciprocating compressors are used widely in industry. They could deliver high pressure compressed gas. The modifications of design parameters of such compressors lead to more efficient use of the machines. By modeling of these compressors, it is possible to study effects of various parameters on their performance and to identify the optimum design parameters. The modeling and simulation could also enable us to diagnosis possible fault which degrade compressor performance.

The reciprocating compressors have been modeled with various methods. These methods could be generally classified into global models and differential models where the variable depends on crank angle [1]. Stouffs et al. [1] presented a global model for the thermodynamic analysis of reciprocating compressors. Their model is based on five mains and four secondary dimensionless physically

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meaningful parameters and expressions for the volumetric effectiveness, the work per unit mass and the indicated efficiency were derived. Casting et al. [2] simulated compressor behavior using efficiency definitions such as volumetric, isentropic and effective. They supposed that these efficiencies depend essentially on two parameters, the dead volumetric ratio, having particular influence on volumetric efficiency, and a friction factor mainly influencing both isentropic and effective efficiencies. Elhaj et al. [3] presented a numerical simulation of a two-stage reciprocating compressor for the development of diagnostic features for predictive condition monitoring. Eric et al. [4] presented a simplified model of an open-type reciprocating compressor. Their analysis revealed the main processes affected the refrigerant mass flow rate and the compressor power and the discharge temperature. Ndiaye et al. [5] presented a dynamic model of a hermetic reciprocating compressor in on-off cycling operation.

The mentioned researches have utilized the first law of thermodynamics as main tool. The second law of thermodynamic is also employed to analysis performance of the reciprocating compressors. McGovern et al. [6] analyzed the compressor performance using the second law. The non-idealities are characterized as exergy destruction rates as losses to friction, irreversible heat transfer, fluid throttling and irreversible fluid mixing. Defects in the use of a compressor's shaft power identified and quantified. Apera et al. [7] presented a study that identified for variable speed compressors the current frequency that optimizes the exergy, energy and economy aspects. In another research, Tang Bin et al. [8] studied thermal performance of reciprocating compressor with stepless capacity control system. In this research an experimental setup was working and the compressor with designed stepless capacity control system operate all right. Morriesen et al. [9] study an experimental investigation of transient fluid and superheating in the suction chamber of a refrigeration reciprocating compressor. Also Bin Yang et al. [10] presented a comprehensive model for a semi-hermetic CO₂ reciprocating compressor.

The current study has been extended considerably from previous studies by presenting a detailed numerical method for simulating reciprocating air compressor. The effects of a few more designing parameters on the performance of the compressor have also studied. Perez-Segarra et al. [11] performed the detailed analysis of different well-known thermodynamic efficiencies such as the volumetric efficiency, the isentropic efficiency, and the combined mechanical-electrical efficiency, which these efficiencies usually used to characterize hermetic reciprocating compressors. They detach these efficiencies into their main components (physical sub-processes). Enrico Da Riva et al. [12] reported the experimental performance of a semi-hermetic reciprocating compressor. The compressor has been installed in a 100kW heating capacity heat pump using propane. The influence of the use of an internal heat exchanger between liquid

and vapour line on the performance of the compressor is discussed. Damle et al. [13] presented the model that applies an object-oriented unstructured modular methodology for the numerical simulation of the elements forming the hermetic reciprocating compressor domain to predict the thermal and fluid dynamic behavior (temperature, pressure, mass flow rates, power consumption, etc.) of the compressor. Rodrigo Link et al. [14] presented a simulation methodology, experimental validated, to analyze the compressor in transients time. Furthermore, their model is employed to estimate the minimum voltage required for the compressor startup as a function of the equalized pressure and the auxiliary coil actuation time. Negrao et al. [15] presented a semi-empirical mathematical model to simulate the unsteady behavior of mass flow rate and power of reciprocating compressor. The model is based on thermodynamic equations fitted to manufacturer data by using linear correlations. Yuan Ma et al. [16] was developed a semi-hermetic reciprocating compressor for application in CO₂ refrigeration. The movement of the valve was discussed in detail for the trans-critical CO₂ compressor with the experimental results. This work has been extended dramatically from above mentioned researches by presenting a detailed mathematical method for modeling reciprocating compressors. The results from the developed model have been validated against the previous measured values. The effects of various parameters on the performance of the compressors have been investigated. An optimum value for suction to discharge valve area ratio has been found.

2. The thermodynamic modeling

The schematic diagram of a reciprocating compressor with spring type suction and discharge valves is shown in Fig. 1. The rotary motion of crankshaft is converted to the reciprocating motion of piston by connecting rod. Gas in cylinder is assumed as lump open system. It is assumed that no leakage take place in the compressor. The governing equation for simulating the compressor is presented in this section.

2.1 Energy Equation

The cylinder wall, cylinder head and piston end face are considered as boundaries for control volume. The first thermodynamic law is written as follow:

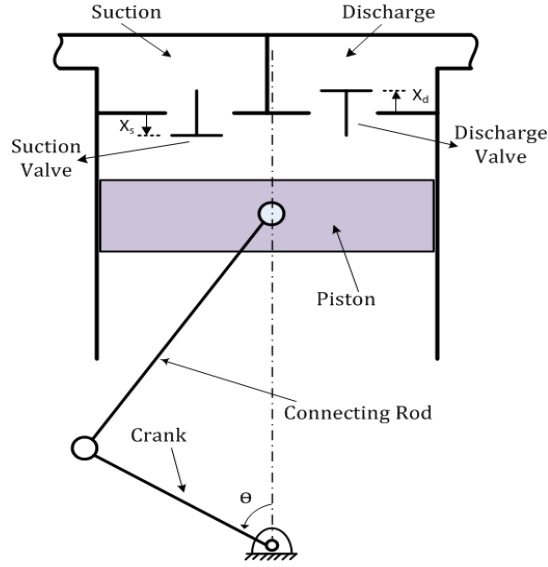


Fig.1. Schematic of reciprocating compressor

$$\dot{Q} + \sum \dot{m}_i h_i = \dot{W} + \sum \dot{m}_e h_e + \frac{d(mu)}{dt} \quad (1(a))$$

If variation potential energy is neglected, then the first law of thermodynamic could be written as follow:

$$\frac{dQ}{dt} + \frac{dm_i}{dt} h_i = \frac{dm_e}{dt} h_e + \frac{d}{dt} (m_c u) + \frac{dW}{dt} \quad (1(b))$$

Where

$$\frac{dW}{dt} = P_c \frac{dV_c}{dt} \quad (2)$$

If the in-cylinder gas could be treated as ideal, then

$$u = c_v T \quad h = c_p T \quad c_p - c_v = R \quad (3)$$

Consequently,

$$\frac{d}{dt} (m_c u) = m_c C_v \frac{dT}{dt} + C_v T \frac{dm_c}{dt} \quad (4)$$

Then, the first law of thermodynamic could be modified as follow:

$$\frac{dT_c}{dt} = \frac{1}{m_c (C_p - R)} \left\{ \frac{dQ}{dt} + C_p \frac{dm_s}{dt} T_s - P_c \frac{dV}{dt} - C_p \frac{dm_d}{dt} T_c - (C_p - R) \left(\frac{dm_s}{dt} - \frac{dm_d}{dt} \right) T_c \right\} \quad (5)$$

Differentiating respect to time could be converted to crank angle by considering the following equation

$$\frac{d}{dt} = \frac{d}{d\theta} \times \frac{d\theta}{dt} = \omega \frac{d}{d\theta} \quad (6)$$

In which ω is the rotational speed of the crank shaft.

2.2 Piston motion equation

The exact expression for the instantaneous position of the piston displacement from top dead center in terms of the crank angle may be given by:

$$x(\theta) = \frac{S}{2} \left[1 - \cos\theta + \frac{L}{a} \left(1 - \sqrt{1 - \left(\frac{a}{L} \sin\theta \right)^2} \right) \right] \quad (7)$$

And the instantaneous volume of cylinder is given by:

$$V_c = A_c \cdot s(\theta) + V_0 \quad (8)$$

2.3 Continuity equation

Applying the mass conservation law for the in-cylinder gas as control volume:

$$\frac{dm_c}{d\theta} = \frac{dm_s}{d\theta} - \frac{dm_d}{d\theta} \quad (9)$$

Where $\frac{dm_s}{d\theta}$ and $\frac{dm_d}{d\theta}$ are the mass flow rate through suction and discharge valves respectively, which are calculating from following equations [17]

$$\dot{m}_s = \begin{cases} C_{ds} \rho_s A_{sv} \sqrt{\frac{2(P_{suc} - P_c)}{\rho_s}} & \text{for } P_s > P_c \text{ and } x_s > 0 \\ -C_{ds} \rho_c A_{sv} \sqrt{\frac{2(P_c - P_{suc})}{\rho_c}} & \text{for } P_c > P_{suc} \text{ and } x_s > 0 \end{cases} \quad (10)$$

$$\dot{m}_d = \begin{cases} C_{dd} \rho_c A_{dv} \sqrt{\frac{2(P_c - P_{dis})}{\rho_c}} & \text{for } P_c > P_{dis} \text{ and } x_d > 0 \\ -C_{dd} \rho_d A_{dv} \sqrt{\frac{2(P_{dis} - P_c)}{\rho_d}} & \text{for } P_d > P_c \text{ and } x_d > 0 \end{cases} \quad (11)$$

Where A_{sv} and A_{dv} are the flow areas through the suction and discharge valves which take place from cylinder respectively. They are given by $2\pi x_s r_{vs}$ and $2\pi x_d r_{vd}$, where x_s and x_d are the suction and discharge displacement from the closed position, and, r_{vs} and r_{vd} are radius of suction and discharge valves respectively.

Due to non-ideality of the valve, it does not shut down instantaneously as soon as a negative pressure difference is created from its reference motion, turn the direction and shut the opening. Coefficient of C_{ds} and C_{dd} account these non-ideality of valves.

2.4 Valve movement equation

The valve dynamic equations are derived based on the following assumption; (i) the valve is considered as a single degree of freedom system, (ii) the valve plate is rigid, (iii) the valve displacement is restricted by a suspension device. Reference point of motion is the closed position of the valve and the valves do not have any negative displacement. Considering the forces acting to the valve plates, the general equation of motion for a valve plate is then given by [18]:

$$\frac{d^2 x_s}{d\theta^2} = \frac{1}{m_s \omega^2} \{-k_s x_s + C_{fs} A_s (P_s - P_c) + F_{ps}\}, \text{ for } x_s > 0 \text{ and } x_s < x_s^{max} \quad (12)$$

$$\frac{d^2 x_d}{d\theta^2} = \frac{1}{m_d \omega^2} \{-k_d x_d + C_{fd} A_d (P_c - P_d) + F_{pd}\}, \text{ for } x_d > 0 \text{ and } x_d < x_d^{max} \quad (13)$$

Where F_{ps} and F_{pd} are pre-load forces, that these forces are neglected respect to another forces. Also m_s and m_d are mass of suction and discharge, respectively. Coefficient of C_{fs} and C_{fd} account loss of the energy due to the orifice flow and these coefficients can be obtained from [19].

2.5 Heat transfer equation

Heat transfer due to convection in compression chamber can be calculated for each degree of crank angle from Eq (14) as:

$$\frac{dQ}{d\theta} = \frac{hA}{\omega} (T_c - T_w) \quad (14)$$

Where h is the heat transfer coefficient, A is surface area in contact with the gas, T_c is the in cylinder gas temperature and T_w is the wall temperature. Adair et al. [20] observed that the cylinder wall temperature varies less than $\pm 1^\circ\text{F}$ as a result the wall temperature is assumed constant.

To calculate convective heat transfer coefficient, the Woschni correlation has been employed [21]. This correlation is originally derived for internal combustion engine. The correlation could also predict the heat transfer rate during compression stage of engine motion. Consequently, it could be used to model heat transfer in a reciprocating compressor. According to the correlation, the heat transfer coefficient is given by:

$$h = 3.26 D^{-0.2} P^{0.8} T^{-0.55} v^{0.8} \quad (15)$$

Where, P is instantaneous in-cylinder pressure, T is instantaneous gas temperature, v is the characteristic velocity of gas and D is diameter of the cylinder. According to Woschni correlation, the correlation characteristic velocity for a compressor without swirl is given as [21]:

$$v = 2.28 S_p \quad (16)$$

Where, S_p is average velocity of piston.

3. Numerical Solution

Differential equations that should be solved simultaneously are energy equation, continuity equation and valve motion equations. Because the equations of valve motion are second order equations, they are solved by introducing intermediate variables $y_s = \frac{dx_s}{d\theta}$ and $y_d = \frac{dx_d}{d\theta}$. The number of ordinary differential equations that to be solved with these two additional equations, reaches to six. In this research a standard 4th order Runge-Kutta method is used and the simulation start from top center point. As the state of gas thermodynamic in start of simulation is unknown, the shooting method is employed. After the

thermodynamic specifics are determined in the cylinder at every crank angle, indicator work is obtained as:

$$W = -\oint P_c dV \quad (17)$$

4. Result and discussion

To validate the numerical method discussed above, a single stage, single cylinder reciprocating compressor is modeled and the numerical results have been compared with available measured value of Venkatesan et al. [22]. The variation of cylinder pressure against crank angle is shown in Fig. 2. Also volume flow rate and peak pressure are compared in Table 1. In the numerical study, the suction and discharge pressure assumed constant. In the actual case, the volume displaced by the piston is greater than the amount of air entering into the cylinder during suction period. As a result for the actual case, a decrease in cylinder pressure is expected. This is could be seen in the figure. Similarly, the volume displaced by the piston is greater than the volume of air discharged through discharge port for the actual case during discharge period. Consequently, an increase in cylinder pressure is expected. But generally, there are a good agreement between the measured and numerical values.

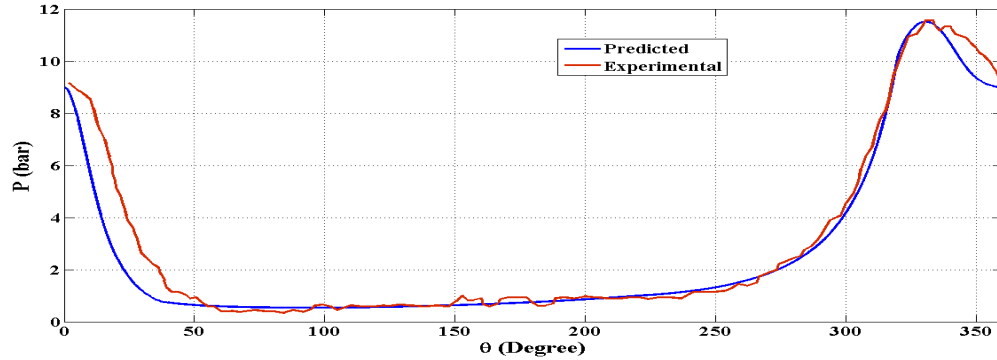


Fig.2. Comparison between numerical and measured values of cylinder pressure at $\omega=1800$ rpm

Table1. Comparison between measured and numerical value of two points pressure at $\omega=1800$ rpm

	Peak pressure (bar)	Free Air delivered(liter/min)
Experimental	11.58	277.4
Predicted	11.51	280.2
Error (%)	0.6	1.01

The rest of the results presented in this study are for a compressor with following specifications: $B=12.39$ cm, $S=10$ cm, $r_s=1.99$ cm, $r_d=2.93$ cm, $M_s=M_d=0.02$ kg, $k_s=k_d=5$ N/mm, $T_s=296$ K. The pressure of discharge plenum and also pressure ratio are considered to be 7MPa and 4.67, respectively. The effects of various parameters are also studied in separated sections.

4.1 Effect of angular speed on compressor performance:

The variation of cylinder pressure against crank angle is shown in Fig. 3(a). The effects of angular speed are also studied in the figure. The considered angular speeds are 50, 100, 150, 200, 250, 300 and 350 rad/s. As stated previously, the simulation starts from top dead center (TDC) where cylinder volume is the same as clearance volume. With motion of piston from top dead center (TDC) towards bottom, cylinder volume is increased and subsequently pressure and temperature (Fig. 3(b)) are decreased.

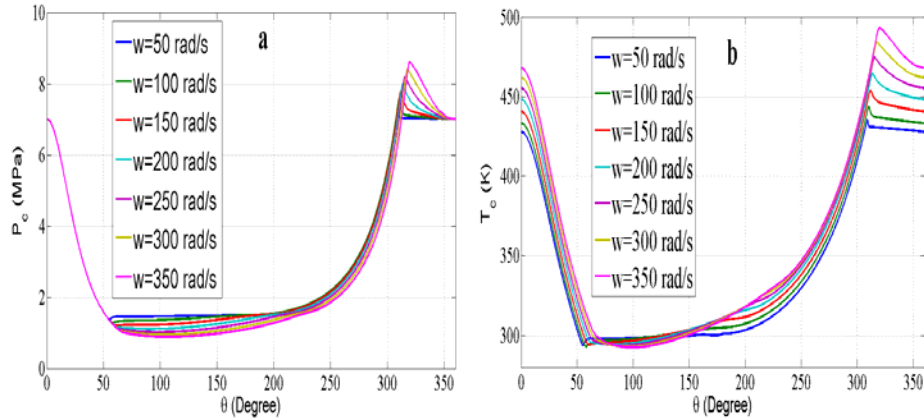


Fig 3: Variation of in-cylinder (a) pressure vs. crank angle (b) temperature vs. crank angle for various angular speeds

Once in-cylinder pressure is lesser than suction pressure as such that the force created by the pressure difference is more than force of suction valve spring, suction valve will be opened (Fig. 4(a)). This causes air flows through suction valve and enters into cylinder (Fig. 5(a)). Suction process continues until force due to pressure difference is balanced by the force of suction valve spring. Then around bottom dead center (BDC), suction valve is closed. In BDC, piston motion is become backward and cylinder pressure and subsequently temperature is increased. When force due to pressure difference (in-cylinder pressure and discharge pressure) is more than force due to discharge spring valve, discharge valve is opened (Fig. 4(b)). This causes that air flows out through discharge valve (Fig. 5(b)). When piston reaches to the TDC, the compressor completes a whole cycle. Figure 4 shows variation of pressure for difference angular speeds. For all of angular speeds, graphs are coincidence until suction valve is opened. Once suction valve is opened suction process starts (Fig. 5(a)). With increase in angular speeds cylinder pressure decreases more sharply. Due to lower time for heat lost in high angular speeds, the in-cylinder temperature reaches to higher values (see Fig. 3(b)). Consequently, in-cylinder pressure has been reached to higher value due for high angular speeds. Fig. 3(b) shows effect of angular speeds on cylinder temperature. There is a direct relation between increase in angular speed and rise

in discharge temperature. For example, as angular speed increases from 50 to 350 rad/s, the discharge temperature rises about 44 K.

The effects of angular speeds on suction and discharge valves motion are shown in Figure 4(a) and 4(b) respectively. The figures show that valves vibration occurs at low crank angle. As the angular speed increases, the valves opening time increases.

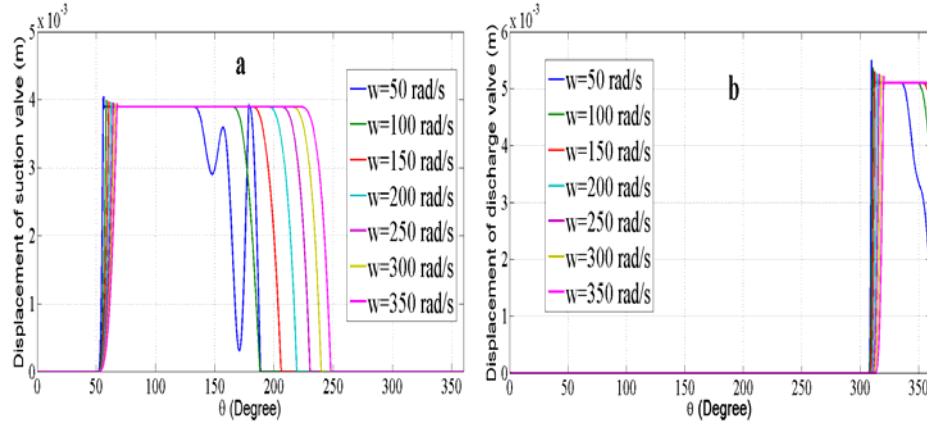


Fig 4: Displacement of (a) suction valve vs. crank angle (b) discharge valve vs. crank angle for various angular speeds

The effects of angular speeds on instantaneous suction and discharge valves mass flow rate are shown in figure 5(a) and 5(b) respectively. It could be realized that there is backward flow for suction valve.

Bar charts of mass flow rate and indicator work are shown in Fig. 6(a) and Fig. 6(b) respectively. As angular speed increases, the number of cycles is done by compressor in specific time increases too, therefore the mass flow rate and subsequently indicator work will be increased.

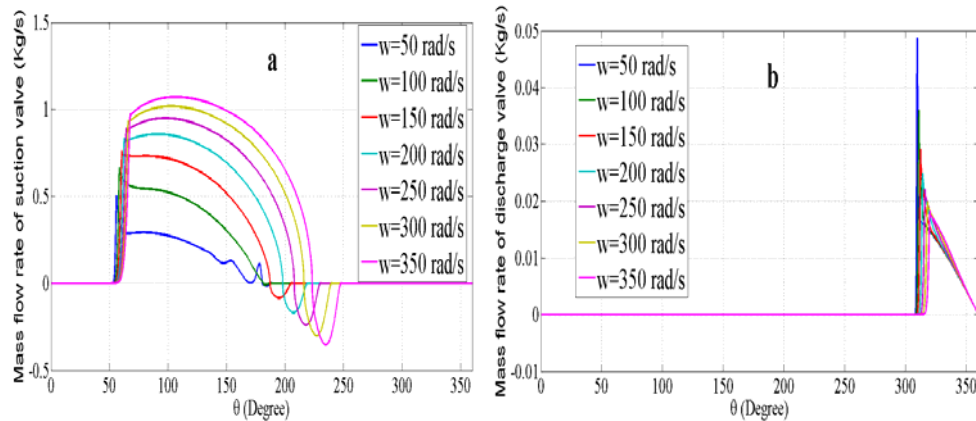


Fig 5: Mass flow rate through (a) suction valve vs. crank angle (b) discharge valve vs. crank angle for various angular speeds

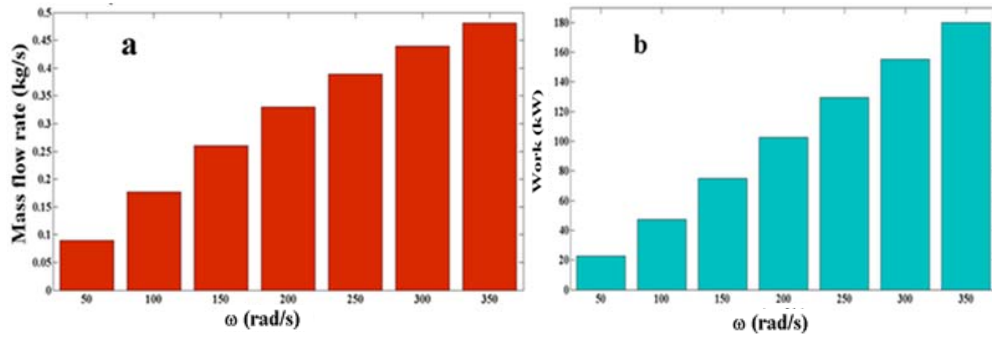


Fig 6: (a) Mass flow rate for various angular speeds (b) Indicator work for various angular speeds

4.2 Effect of clearance value on compressor performance

Because of various reasons such as the heat expansion of compressor pieces, exist of clearance in reciprocating compressor is invertible. In order to evaluate clearance value on performance compressor, percent of clearances 3, 5, 7, 9, 11, 13 and 15 have been considered and studied. Fig. 7(a) shows variation of pressure verse to difference clearance percentage. As it is shown, the lines of pressure in suction and discharge processes are almost coincidence. With decrease in clearance, expansion process ended faster and suction valve is opened sooner (Fig. 8(a)). Value of clearance affected on compression process as decrease in clearance value cause to compression process ended faster and discharge valve is opened sooner (Fig. 8(b)). Fig. 7(b) reveals that the effect of clearance value on discharge temperature isn't much. For example, temperature difference of discharge gas between clearances 3% and 15% is 5 K. As clearance decreases, the temperature of expanded gas is reduced. As an example, temperature difference of expanded gas between clearances 3% and 15% just before suction process is 8 K.

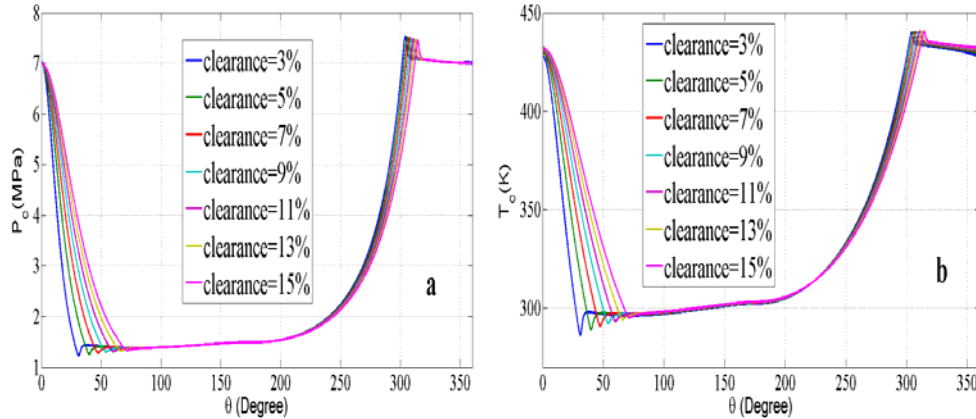


Fig 7: Variation of (a) pressure vs. crank angle (b) temperature vs. crank angle for various clearances

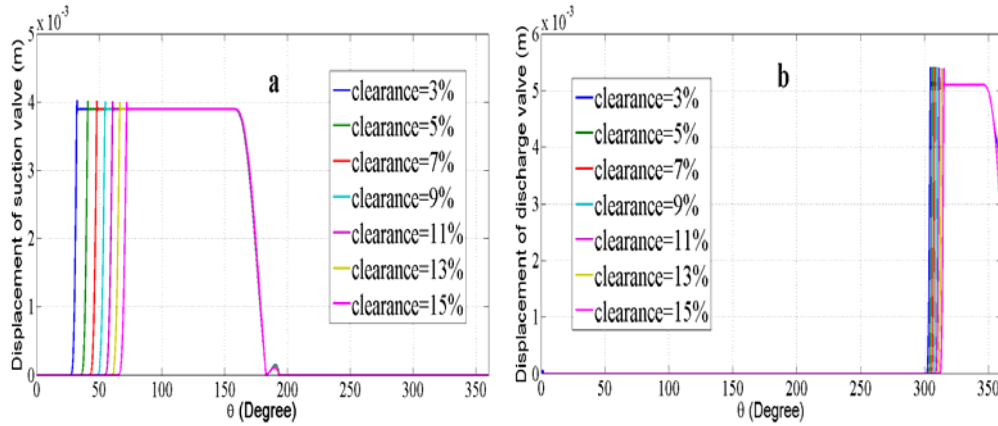


Fig 8: Displacement of (a) suction valve vs. crank angle (b) discharge valve vs. crank angle for various clearances

Bar charts of mass flow rate and indicator work for various clearances are shown in Fig. 9(a) and Fig. 9(b) respectively.

As clearance decreases, mass flow rate and subsequently indicator work are increased. Fig. 10 shows work in unit mass of delivered air. Clearance value doesn't affect much specific work. Therefore necessary work for compressing a specific value of gas in various clearances is almost equal.

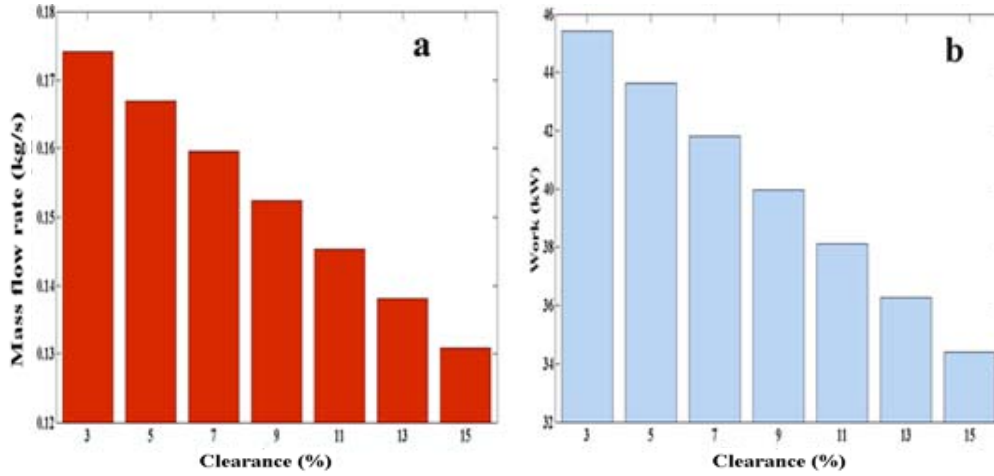


Fig 9: (a) Mass flow rate for various clearances (b) Indicator work for various clearances

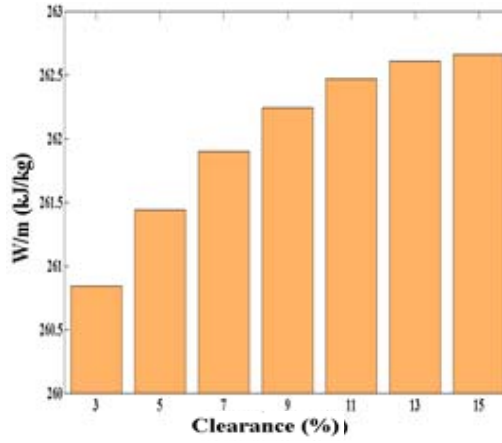


Fig 10: Work in mass unit for various clearances

4.2 Effect of discharge to suction valve area on compressor performance:

In this section, the effects of varying discharge to suction valve area on compressor performance are studied. The cylinder head area is assumed constant. Two auxiliary parameters $\alpha = \frac{A_d}{A_s}$ and $\beta = \frac{A_s + A_d}{A_c}$ are introduced for studying these effects. Variation of mass flow rate versus α for various β is shown in Fig. 11(a). It could be realized that as β increases mass flow rate increases too. In other hand as α decreases, mass flow rate decreases. For $\alpha < 0.4$, mass flow rate is nearly constant. Variation of indicator work versus α for various β is shown in Fig. 11(b).

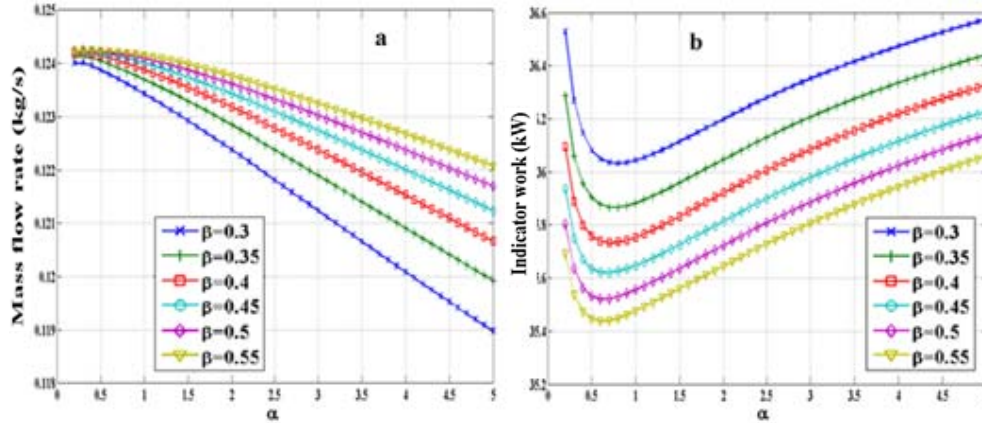


Fig. 11: (a) Mass flow rate vs. discharge (b) Indicator work vs. discharge to suction valve area

It is clear that as β decrease indicator work decreases too. There is a specific value of α (about 0.5) in which the indicator work is minimized. This

point could be treated as optimum design value for discharge to suction valve area.

5. Conclusion

Reciprocating compressors are used widely in industry due to their ability to deliver high pressure compressed gas. Understanding the behavior of the reciprocating compressors and studying effects of various parameters are interesting subjects. The mathematical modeling is proved to be an effective tool to study performance of the reciprocating compressors.

A mathematical model has been developed based on the conservation of mass and energy to study the performance of reciprocating compressors. The model could predict in-cylinder pressure, in-cylinder temperature and valves motions at various crank angles. The indicator work and deliver air are also calculated. The results from the developed model have been validated against the previous measured values and good agreement has been obtained. The effects of various parameters on the performance of the compressors have been investigated.

The results show that as angular speed increases, the in-cylinder pressure and temperature increases too. The valves vibration occurs at low crank angle and as the angular speed increases, the valves opening angle increases. There is backward flow for suction valve during each cycle. Clearance value doesn't affect specific work (indicator work to delivered air ratio) and so necessary work for compressing a specific value of gas in various clearances is almost equal.

Variation of indicator work verses ratio of discharge to suction valves shows that there is a specific value (about 0.5) in which the indicator work is minimized. This point could be treated as optimum design value for discharge to suction valve area.

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