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THEORETICAL STUDY OF DESIGN AND OPERATING PARAMETERS ON THE
RECIPROCATING COMPRESSOR PERFORMANCE

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ABSTRACT
Mathematical simulation for the reciprocating compressor cycle incorporating the effects of common faults has been developed. The governing equations for different processes are derived by using first law of thermodynamics, continuity equation, equation of mass flow through valves, kinematic equation for piston movement and the cylinder heat transfer correlations. This paper gives the results of theoretical study of effects of design and operating parameters on compressor performance.

NOMENCLATURE

\begin{itemize}
  \item \( \rho \) density of air inside the cylinder
  \item \( \beta \) square root of the ratio of orifice and flow area at inlet
  \item \( \gamma \) ratio of specific heats at constant pressure and constant volume
  \item \( \omega \) angular velocity of crank shaft
  \item \( \xi \) viscous damping factor for valve springs
  \item \( A \) area
  \item \( A_k \) spring stiffness of valve
  \item \( C_d \) valve discharge coefficient
  \item \( C_{de} \) drag coefficient of valve plate
  \item \( C_v \) specific heat of air at constant volume
  \item \( D_e \) equivalent diameter
  \item \( F_{d} \) valve plate area on which the pressure acts
  \item \( F_{init} \) initial force on valve plate
  \item \( h \) instantaneous heat transfer coefficient
  \item \( Re \) Reynolds number
  \item \( m \) mass of air inside cylinder
  \item \( P \) cylinder air pressure
  \item \( k \) thermal conductivity
  \item \( \rho_{Pr} \) square root of the R
  \item \( R \) ratio of orifice and flow area at inlet
  \item \( \gamma \) ratio of specific heats at constant pressure and constant volume
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INTRODUCTION

Most of the simulation studies on the reciprocating compressors carried out to date do not incorporate the effect of common faults like leaking valves, weak valve springs, leaking piston rings, slipping belt drives and clogged air filters [1, 2]. A mathematical model has been developed [3] to simulate the reciprocating compressor cycle incorporating the effects of common faults for a single stage reciprocating compressor with disc type of valves. The compressor performance is dependent on the values of the design parameters like initial force on the valve disc, stiffness of the valve springs and some other constants like drag coefficient of valve disc and valve discharge coefficient. In addition, the performance is also dependent on operating parameter like suction air temperature. These parameters are used in the input data for simulating the compressor performance and the computed results depend on the correctness of the chosen values of these parameters. This paper gives the results of the theoretical study of their effect on the compressor performance.

MODELLING OF COMPRESSOR THERMODYNAMIC CYCLE

Physical model of a single stage reciprocating compressor and its thermodynamic cycle are given in Figures 1 and 2. The equations governing the different processes of expansion, suction, compression and discharge are derived by applying the continuity equation, equation of mass flow through valves [4], first law of thermodynamics, the cylinder heat transfer correlation, valve dynamics equation and kinematic equation of piston movement to the control volume shown in Figure 1. The governing equations used are as follows-

\[ \frac{dm}{dt} = \left( \frac{dm_s}{dt} + \frac{dm_p}{dt} \right) - \frac{dm_d}{dt} \]  \hspace{1cm} (1)

\[ \frac{dm_o}{dt} = C_d \cdot A_o \sqrt{\frac{2 \rho \Delta P}{1 - \beta^4}} \]  \hspace{1cm} (2)

\[ \frac{dP}{dt} = -P \left[ \frac{dV}{dt} \left( \frac{1}{V} + \frac{R}{VC_v} \right) + \frac{h \cdot A R}{mC_v} \right] + \frac{hA T_v R}{VC_v} + \gamma \frac{R}{V} \left[ T_s \left( \frac{dm_s}{dt} + \frac{dm_p}{dt} \right) - T_d \frac{dm_d}{dt} \right] \]  \hspace{1cm} (3)

\[ \frac{h(t) D_e(t)}{k(t)} = 0.053 \left[ Re(t) \right]^{0.8} \left[ Pr(t) \right]^{0.6} \]  \hspace{1cm} (4)
All these equations are solved using Runge-Kutta method. This simulation model can give cylinder air pressure, temperature, mass and valve displacement for the complete compressor cycle and also the harmonic components of pressure. It can predict the compressor performance giving the values of volumetric efficiency, volume flow rate and the performance ratio.

**STUDY OF PARAMETERS AFFECTING COMPRESSOR PERFORMANCE**

The compressor performance is dependent on the values of the design parameters like initial force on valve disc \( F_{\text{init}} \), stiffness of valve spring \( A_k \) and some other constants like drag coefficient of the valve disc \( C_{df} \) and valve discharge coefficient \( C_d \). In addition, the performance is also dependent on operating parameters like suction air temperature \( T_s \). Some of these design and operating parameters like \( F_{\text{init}} \), \( A_k \) and \( T_s \) may vary in practice in due course of time. The study of the effect of the variation of each of the above parameters has been carried out by assigning different values to each of these parameters individually in the known range while keeping the others as constant for a fixed discharge pressure.

The initial force \( F_{\text{init}} \) depends upon the stiffness and the initial compression of the spring. To study the effect of \( F_{\text{init}} \) on suction valve plate, the values in the range of 7 to 47 N were assigned and the compressor performance was computed. The cylinder air pressure, temperature and mass time diagrams are shown in Figures 3, 4 and 5 respectively. With the increase in \( F_{\text{init}} \), the suction and discharge processes are delayed. The compression process begins at a little lower pressure and the maximum cylinder air pressure increases. The mass of air inside the cylinder decreases with increase in \( F_{\text{init}} \). The cylinder air temperature is not much affected. The instantaneous valve displacement for different values of \( F_{\text{init}} \) are given in Figure 6. It has been observed that the suction and discharge valves open for shorter duration as \( F_{\text{init}} \) increases. The performance parameters are given in Table 1. The mean cylinder pressure, volume flow rate, volumetric efficiency and performance ratio decrease with the increasing values of \( F_{\text{init}} \). The cylinder air pressure harmonics for different values of \( F_{\text{init}} \) are given in Table 2, which indicate a prominent increase in 4th and 12th harmonic.

Similar study on \( F_{\text{init}} \), range 25-100 N, for discharge valve plate indicates that its increase leads to the shifting of
compression and discharge curves of the pressure-time diagram towards the right hand side. The performance parameters are not affected significantly. The 1st, 2nd, 3rd, 4th and 10th pressure harmonics show an increasing trend and 6th harmonic shows a decreasing trend with the increase in $F_{\text{init}}$.

The study on spring stiffness $[Ak]$ of suction valve, range 0.45-18 kN/m, indicates no significant variation in the cylinder air pressure and temperature-time diagrams. The duration of suction valve opening is minimum for an optimum value of Ak and increases for the other values. The performance parameters have optimum values corresponding to a particular value of Ak causing suction valve closure at BDC. For other values the performance parameters show a decreasing trend. The 3rd, 5th, 9th pressure harmonics show an increasing trend and 7th, 8th harmonics show a decreasing trend for increasing values of Ak. In case of discharge valve, (range 10-45 kN/m), weak spring shows lower pressure during the discharge process and complete closure of discharge valve at the end of the cycle. The variations in the magnitude of valve lift are also reduced. With the increase in Ak a slight decrease in the performance ratio has been observed. The 1st, 2nd, 3rd, 4th, 5th, 9th and 10th pressure harmonics show an increasing trend whereas 6th, 11th and 12th harmonics show a decreasing trend with the increasing Ak.

The study on suction air temperature $[T_s]$ in the range 20-45°C shows that cylinder air temperature decreases and mass of cylinder air increases with decrease in its value. The performance parameters decrease with the increase in $T_s$.

The study on drag coefficient of valve disc $[C_{dv}]$, in the range 0.25-0.40, shows that increase in its value shifts the compression curve of the cylinder air pressure-time diagram towards left hand side. The maximum cylinder pressure and the fluctuations of pressure during the discharge process are reduced with the increased value of $C_{dv}$. The durations of opening of suction and discharge valves also increase with the increasing $C_{dv}$. The performance parameters increase with the increasing values of $C_{dv}$. The 1st, 2nd, 7th, 10th, 11th and 12th pressure harmonics show an increasing trend while, 3rd, 4th, 5th, 6th, 8th and 9th harmonics show a decreasing trend with the increasing values of $C_{dv}$.

The study on the discharge coefficient $[C_d]$ in the range 0.5-0.9 shows that increase in its value indicates increased fluctuations in the cylinder air pressure during the discharge process. The duration of suction valve opening is minimum for a particular value of the $C_d$ and increases for all other values of $C_d$. Other performance parameters have maximum values for the particular value of $C_d$ and decrease for all other values. The 5th pressure harmonic increases whereas the 2nd and 3rd harmonics decrease with the increase in the value of $C_d$. 

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CONCLUSIONS

The computed results in the simulation program depend on the design parameters like $F_{\text{init}}$ and $A_k$. Some other constants like $C_d$ and $C_{df}$ affect the compressor performance significantly and hence it is important to choose these parameters carefully in practice for getting meaningful results. The performance is also dependent on operating parameter like $T_s$.

REFERENCES


Table 1. Performance parameters for different values of initial force on suction valve plate.

<table>
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<tr>
<th>Performance parameters</th>
<th>Initial force on suction valve plate, (N)</th>
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<tr>
<td>Mean cylinder pressure (Pa)</td>
<td>$1.89 \times 10^5$</td>
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<tr>
<td>Volume flow rate (scfm)</td>
<td>25</td>
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<tr>
<td>Volumetric efficiency (%)</td>
<td>81.7</td>
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<td>Performance ratio (kg/J)</td>
<td>$4.66 \times 10^{-6}$</td>
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</table>

Table 2. Amplitude of cylinder air pressure harmonics for different values of initial force on suction valve plate (Pa).

<table>
<thead>
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Fig. 1 Physical model of the single stage reciprocating air compressor.

Fig. 2. Thermodynamic cycle of the reciprocating air compressor.

Fig. 3 Cylinder air pressure time diagram for different values of Finit on suction valve.

Fig. 4 Cylinder air temperature diagram for different values of Finit on suction valve.

Fig. 5 Valve displacement-time diagram for different values of Finit on suction valve.

Fig. 6 Cylinder air mass-time diagram for different values of Finit on suction valve.