

## Valve fault detection for single-stage reciprocating compressors



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### ABSTRACT

In the present study, a zero-dimensional numerical method is developed based upon the crank angle to investigate reciprocating natural gas compressor with valve faults. This model aims to take piston movement, valve dynamic, and mass flow rate through valve and orifice equations into account. To this end, three control volumes including compressor cylinder, suction, and discharge chambers with equivalent mass and energy equations are investigated. Valve faults, which include valve plate failure, wearing on the plate and its seat, and springs deterioration, lead to valve leakage. For valve leakage simulation, a hole in the valve plate is considered. Simulated results validate the previous experimental results for normal operating compressor with healthy valves. The predicted results show that the suction and discharge valve failures decrease compressor mass flow rate and increase discharge gas temperature. Furthermore, effect of the suction valve fault is more serious than that of the discharge valve fault. In addition, valve fault can be detected by monitoring the gas temperature of the suction and discharge chambers.

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### 1. Introduction

Reciprocating compressors are the most important and oldest machines that have been widely used over the decades for industrial purposes such as gas refineries, CNG (Compressed Natural Gas) stations and refrigeration systems. These compressors use a huge amount of energy, thus their inefficient working conditions lead to wasting energy. Fault in suction and discharge valves is a very common defect that affects compressor operation and reduces compressor performance and effectiveness (Hanlon, 2001). Early diagnosis of these faults is very crucial for not only the reduction of maintenance costs, but also the sustainable production and reliability of the plant.

The large amount and variety of researches and models have been presented over recent decades show how detecting and diagnosing valve faults in reciprocating machines and compressors are of paramount importance. Valve problems affect the speed fluctuation of the compressor rotating shaft (Zhi et al., 2005), motor electrical current consumption (Schantz et al., 2010), cylinder pressure (McCarthy, 1994), vibration signals emitted from the compressor (Ahmed et al., 2012), and discharge gas temperature. To

find these faults, different methods, including instantaneous angular speed (IAS) (Al-Qattan et al., 2009), in-cylinder pressure (Elhaj et al., 2010), motor current signals (Gu et al., 2011), and vibration analysis (Liang et al., 1996), have been developed and investigated experimentally and numerically.

Computer simulation has been developed in the recent decades as a powerful, rapid, and economical method for compressor design, performance, and behaviour prediction. In the simulation, physical events should be identified and mathematical model for them should be developed with algebraic equations. Multi-, one-, and zero-dimensional approaches can be used for the mathematical analysis. In three-dimensional modelling, conservation laws along with turbulent model are considered for the simulation (Yasar and Kocas, 2007). Two-dimensional simulation with turbulent flow is used for the prediction of flow feature in the discharge valve (Rovaris & Deschamps, 2006) and heat transfer in the cylinder chamber (Disconzi et al., 2012). Due to the complication and high computational effort of these methods, one-dimensional method with simplified structure has been used by researchers (Escanese et al., 1996). This method can be applied throughout the entire compressor domain (Rigola et al., 2005). Furthermore, complex and turbulent regions with big volumes including the cylinder, suction, and discharge chambers are considered volumes and other parts such as pipes in which the flow has a specific direction are assumed as ducts (Bassi et al., 2000). In this method, each volume is shown by one control volume, while ducts are discretized into a number of

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## Nomenclature

$A$	Valve plate surface, $m^2$
$A_{sl}, A_{dl}$	Valve Leakage flow area, $m^2$
$A_{vs}, A_{vd}$	Valve flow area, $m^2$
$C_f$	Force coefficient
$c_p$	Specific heat at constant pressure, $J kg^{-1} K^{-1}$
$c_v$	Specific heat at constant volume, $J kg^{-1} K^{-1}$
$D$	Piston diameter, $m$
$E$	Energy of system, $J$
$h$	Enthalpy, $J kg^{-1}$
$k$	Valve spring constant, $N mm^{-1}$
$l$	Connecting rod length, $m$
$M_{vs}, M_{vd}$	Equivalent valve plate mass for suction and discharge valve, $kg$
$m$	Mass of gas, $kg$
$p$	Pressure, $Pa$
$Q$	Heat transfer, $J$
$\dot{Q}$	Heat transfer rate, $J s^{-1}$
$R$	Universal gas constant, $J kg^{-1} K^{-1}$
$r$	Crank radius, $m$
$T$	Temperature, $K$
$t$	Time, $s$
$u$	Internal energy, $J kg^{-1}$
$V_{cyl}$	Compressor cylinder volume, $m^3$
$V_0$	Clearance volume, $m$

$W$	Work, $J$
$\dot{W}$	Power, $J s^{-1}$
$x_s, x_d$	Displacement of suction and discharge Valves, $m$
$x_{pls}, x_{pld}$	Spring pre-load displacement of suction and discharge valves, $m$
$x_{max}$	Maximum valve displacement, $m$

### Greek symbols

$\alpha$	Coefficient to account for non-ideality
$\theta$	Instant angle of connecting rod (Degree)
$\rho$	Gas density, $kg m^{-3}$
$\omega$	Angular velocity, $rad s^{-1}$

### Subscript

<i>chamber</i>	Suction and discharge chambers
<i>cyl</i>	Compressor cylinder
<i>cv</i>	Control volume
<i>d</i>	Discharge
<i>down</i>	Downstream
<i>e</i>	Output stream
<i>i</i>	Input stream
<i>le</i>	Output leakage stream
<i>li</i>	Input leakage stream
<i>s</i>	Suction
<i>up</i>	Upstream

control volumes in one dimension. These approaches are not suitable for specific goals such as process control design, optimization, and evaluation (Castaing-Lasvignottes & Gibout, 2010).

For eliminating complexity in one-dimensional simulation and increasing computational efficiency, main and basic parts that have big volumes are considered for simulation and ducts are neglected. Thus, single pressure and temperature can identify each system for the main elements in the compressor domain. In this method, called zero-dimensional or thermodynamic simulation, energy and mass equations are used to calculate the temperature and pressure for each control volume (Longo, 2003). Thus, mathematical governing equations are ordinary differential equations (ODEs), which are developed and used by researchers for compressor analysis. Farzaneh-Gord et al. (2014, 2015) considered just compressor cylinder chamber in their simulation and investigated the effect of various parameters, such as clearance volume, pressure ratio, and angular speed on the reciprocating compressor. Elhaj et al. (2008) developed numerical simulation in the compressor cylinder chamber domain for a two-stage reciprocating air compressor. They simulated in-cylinder pressure and IAS for detecting and diagnosing valve faults.

In-cylinder pressure simulation in cyclic steady-state is frequently employed for valve fault detection by researchers (McCarthy, 1994; Farzaneh-Gord et al., 2015). In this technique they used cylinder control volume for compressor simulation. As a fact, after valve failure incident, discharge temperature will increase owing to gas re-circulating. Also, in case of suction valve failure, gas leaks to the suction chamber and affects its temperature. As a result, the cylinder chamber is not enough for accurate modelling of the in-cylinder pressure after valve failure and the suction and discharge chambers should be implemented in the simulation. Furthermore, it is plausible to utilize suction and discharge gas temperatures monitoring in transient and cyclic steady state for detecting valve problems. One of the merits of temperature

monitoring (in the suction and discharge chambers) is that it is easier to measure, compared with the pressure in the cylinder. In addition, in industry process, it is possible to monitor the gas temperature in the suction and discharge chambers continually compared to in-cylinder pressure.

Therefore, this paper aims to develop a zero-dimensional numerical model of a single-stage reciprocating natural gas compressor for the simulation of gas temperature monitoring in the suction and discharge chambers for detecting valve failure. In addition to the compressor cylinder chamber, the suction and discharge chambers are implemented in this analysis to calculate suction and discharge chamber temperatures. A hole in the valve plate is considered for simulating the valve leakage. For the simulation, first thermodynamic law and mass conservation for each control volume along with equations for valves dynamic, mass flow through valves, ports and hole on the valve plate, and mechanical motion of crank mechanism are solved simultaneously. Thereafter, the effect of suction and discharge valve faults on suction/discharge temperature and mass flow rate are studied.

## 2. Governing equations

In zero-dimensional modelling method, continuity and first law thermodynamic equations are used to find thermodynamic properties. In this approach, temperature and pressure distribution in control volumes are neglected due to momentum equation elimination and the open system will undergo a quasi-state process (Follen, 2010).

As shown in Fig. 1, the configuration is a single-acting compressor with suction and discharge chambers. For the simulation, the model consists of a compressor cylinder, piston-driven mechanism, suction/discharge valve, and suction/discharge chamber.

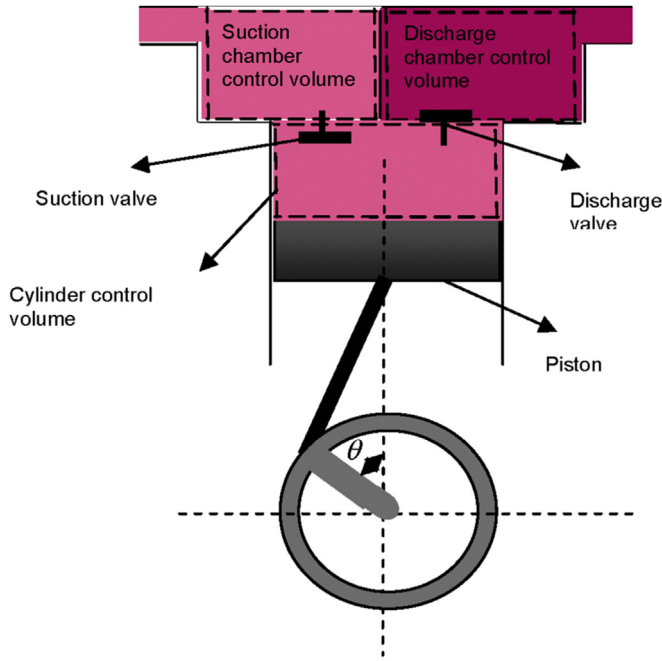


Fig. 1. Schematic of reciprocating compressor.

### 2.1. Mechanical motion

Rotational motion is turned to linear reciprocating motion by crankshaft and connecting rod in reciprocating machines. The instantaneous cylinder volume from top dead centre, as the reference point for motion, is given by (Mabie, 1987):

$$V_{cyl}(\theta) = V_0 + \frac{\pi D^2}{4} \left( r(1 - \cos \theta) + l \left[ 1 - \sqrt{1 - (r/l)^2 \sin^2 \theta} \right] \right) \quad (1)$$

where  $V_0$  is the clearance volume.

### 2.2. Thermodynamic properties of control volumes

The mass conservation equation for control volume is written as follows:

$$\frac{dm_{cv}}{dt} = \frac{dm_i}{dt} + \frac{dm_{li}}{dt} - \frac{dm_e}{dt} - \frac{dm_{le}}{dt} \quad (2)$$

in which  $dm_i/dt$  and  $dm_e/dt$  are mass flow rates through inlet and outlet ports (valves in compressor and orifice in suction and discharge lines), while  $dm_{li}/dt$  and  $dm_{le}/dt$  are considered for leaking to and from the control volume, respectively.

Differentiating with respect to time can be converted into crank angle by:

$$\frac{d}{dt} = \frac{d}{d\theta} \times \frac{d\theta}{dt} = \omega \frac{d}{d\theta} \quad (3)$$

where  $\omega$  is angular velocity of the crank shaft, which is considered constant in this simulation. Finally, the continuity equation is re-written as shown below:

$$\frac{dm_{cv}}{d\theta} = \frac{dm_i}{d\theta} + \frac{dm_{li}}{d\theta} - \frac{dm_e}{d\theta} - \frac{dm_{le}}{d\theta} \quad (4)$$

The first thermodynamic law for a control volume in the general

form, neglecting both potential and kinetic energy, can be given as follows (Moran et al., 2007):

$$\dot{Q}_{cv} - \dot{W} + \sum \dot{m}_i h_i - \sum \dot{m}_e h_e = \frac{dE_{cv}}{dt} \quad (5)$$

The above equation, considering Equation (3), is re-written as follows:

$$\frac{dQ_{cv}}{d\theta} + \frac{dm_i}{d\theta} h_i + \frac{dm_{li}}{d\theta} h_{li} = \frac{dW}{d\theta} + \frac{dm_e}{d\theta} h_e + \frac{dm_{le}}{d\theta} h_{le} + \frac{d(mu)_{cv}}{d\theta} \quad (6)$$

For the ideal gas ( $u=c_v T$ ,  $h=c_p T$ ,  $m=pV/RT$ ), considering  $dW/d\theta = pdV/d\theta$  and  $d(mu)/d\theta = mdu/d\theta + udm/d\theta$ , the first thermodynamic law equation for ideal gas in compressor cylinder control volume can be expressed as follows:

$$\begin{aligned} \frac{dT_{cyl}}{d\theta} = \frac{1}{m_{cyl} c_{v,cyl}} \left\{ \frac{dQ_{cyl}}{d\theta} + c_{p_s} \frac{dm_i}{d\theta} T_s + c_{p_{li}} \frac{dm_{li}}{d\theta} T_{li} - p_{cyl} \frac{dV_{cyl}}{d\theta} \right. \\ \left. - \left( \frac{dm_e}{d\theta} + \frac{dm_{le}}{d\theta} \right) c_{p_{cyl}} T_{cyl} - c_{v,cyl} \frac{dm_{cyl}}{d\theta} T_{cyl} \right\} \quad (7) \end{aligned}$$

In the above equation,  $T_{li}$  for suction and discharge valve faults are the temperature of the suction and discharge chambers, respectively.  $Q_{cyl}$ , is heat transfer in the compressor cylinder chamber that is calculated according to the method described by Farzaneh-Gord et al. (2014).

For the suction and discharge chambers, work is considered zero due to fixed control volume and these chambers are assumed to be adiabatic. Therefore, the first law thermodynamic equation for the suction and discharge chambers can be evaluated using the following equation:

$$\begin{aligned} \frac{dT_{chamber}}{d\theta} = \frac{1}{m_{chamber} c_{v,chamber}} \left\{ c_{p_i} \frac{dm_i}{d\theta} T_i + c_{p_{cyl}} \frac{dm_{li}}{d\theta} T_{cyl} - \left( \frac{dm_e}{d\theta} \right. \right. \\ \left. \left. + \frac{dm_{le}}{d\theta} \right) c_{p_{chamber}} T_{chamber} - c_{v,chamber} \frac{dm_{chamber}}{d\theta} T_{chamber} \right\} \quad (8) \end{aligned}$$

For the suction chamber, subscript  $i$  is considered for suction line and  $e$  is considered for the compressor suction valve, while for the discharge chamber, subscripts  $i$  and  $e$  are considered for the discharge valve and line, respectively.

### 2.3. Mass flow through valves and ports

Orifice shows resistance to gas flow that is a function of geometry, operating condition (upstream and downstream pressures), and gas properties. Consequently, mass flow rate through orifice can be written as (Habing, 2005):

$$\frac{dm}{d\theta} = \frac{1}{\omega} \alpha \rho_{up} A \left( \frac{2(p_{up} - p_{down})}{\rho_{up}} \right)^{1/2} \quad (9)$$

where  $\alpha$  is the semi-empirical flow coefficient that is considered for viscosity and expansion after flow separation.

When the valves of reciprocating compressors are in an open position, mass flow can be simulated as a mass flow through orifice (Farzaneh-Gord et al., 2013).

$$\frac{dm_s}{d\theta} = \begin{cases} \frac{1}{\omega} \rho_s \alpha_s A_{sv} \left( \frac{2(p_s - p_{cyl})}{\rho_s} \right)^{1/2} & \text{for } (x_s > 0 \ \& \ p_s > p_{cyl}) \\ -\frac{1}{\omega} \rho_{cyl} \alpha_s A_{sv} \left( \frac{2(p_{cyl} - p_s)}{\rho_{cyl}} \right)^{1/2} & \text{for } (x_s > 0 \ \& \ p_{cyl} > p_s) \end{cases} \quad (10)$$

$$\frac{dm_d}{d\theta} = \begin{cases} \frac{1}{\omega} \rho_{cyl} \alpha_d A_{dv} \left( \frac{2(p_{cyl} - p_d)}{\rho_{cyl}} \right)^{1/2} & \text{for } (x_d > 0 \ \& \ p_{cyl} > p_d) \\ -\frac{1}{\omega} \rho_d \alpha_d A_{dv} \left( \frac{2(p_d - p_{cyl})}{\rho_d} \right)^{1/2} & \text{for } (x_d > 0 \ \& \ p_d > p_{cyl}) \end{cases} \quad (11)$$

where  $A_{sv}$  and  $A_{dv}$  are the flow area of suction and discharge valve plates, respectively. Mass flows into the suction chamber (via suction line) and out of the discharge chamber (via discharge line) are considered through orifice as well.

#### 2.4. Valve dynamic

For the valve dynamic simulation, valve plate and supporting spring with equivalent mass  $M$  are considered single degree freedom mass-spring system (Soedel, 2006). Valve displacement,  $x$ , is restricted between seat ( $x=0$ ), and limiter ( $x=x_{max}$ ). For the valve with effective spring constant  $k$ , valve dynamic equations are (Srinivas and Padmanabhan, 2002):

$$M_{vs} \omega^2 \frac{d^2 x_s}{d\theta^2} + k_s (x_s + x_{pls}) = C_{fs} A_s (p_s - p_{cyl}) \quad \text{for } x_s > 0 \ \text{and } x_s < x_{s,max} \quad (12)$$

$$M_{vd} \omega^2 \frac{d^2 x_d}{d\theta^2} + k_d (x_d + x_{pld}) = C_{fd} A_d (p_{cyl} - p_d) \quad \text{for } x_d > 0 \ \text{and } x_d < x_{d,max} \quad (13)$$

where  $A_s$  and  $A_d$  are the surface area of suction and discharge valve plates, respectively and,  $C_{fs}$  and  $C_{fd}$  are force coefficients for the suction and discharge valves that are changing with valve displacement and can be obtained from Farzaneh-Gord et al. (2013). Pre-load displacements,  $x_{pls}$  and  $x_{pld}$ , are not used in this modelling, because their effect on the simulation can be neglected (Srinivas and Padmanabhan, 2002).

#### 2.5. Valve leakage modelling

Valves are the key parts of the reciprocating compressors and valve failure is the most recurrent fault in the reciprocating compressors. Valve fault causes 36% cases of the compressor shutdown (Deffenbaugh, 2005). Valves are vulnerable to Liquid droplets and impurities in the gas. These harmful components can break valve

plate. High temperature gas and vibration impact also wear the sealing surface between valve plate and its seat. Furthermore, these tough working conditions deteriorate valve springs. As a result, valve leakage is a frequent problem in the reciprocating compressors. For the valve leakage modelling, a hole can be considered in the suction or discharge valve plates. Therefore, mass flow through this hole can be calculated by considering the expansion through an orifice and calculated with Equation (9).

The mass flow rate of suction leakage for the compressor cylinder can be calculated based on Equation (9) by

$$\frac{dm_{li}}{d\theta} = \frac{1}{\omega} \rho_s \alpha_s A_{sl} \left( \frac{2(p_s - p_{cyl})}{\rho_s} \right)^{1/2} \quad \text{for } p_s > p_{cyl} \quad (14)$$

$$\frac{dm_{le}}{d\theta} = \frac{1}{\omega} \rho_{cyl} \alpha_s A_{sl} \left( \frac{2(p_{cyl} - p_s)}{\rho_{cyl}} \right)^{1/2} \quad \text{for } p_{cyl} > p_s \quad (15)$$

Where  $A_{sl}$  is the leakage flow area of the suction valve,  $dm_{li}/d\theta$  is the leakage from the suction chamber to the compressor cylinder, and  $dm_{le}/d\theta$  is the leakage from the compressor cylinder to suction chamber. The mass flow rate for suction leakage to and from the suction chamber,  $dm_{li}/d\theta$  and  $dm_{le}/d\theta$ , are equal to  $dm_{le}/d\theta$  and  $dm_{li}/d\theta$  for the cylinder chamber, respectively.

The mass flow rate through discharge leakage for the compressor cylinder can be written based on Equation (9) as follows,

$$\frac{dm_{di}}{d\theta} = \frac{1}{\omega} \rho_d \alpha_d A_{dl} \left( \frac{2(p_d - p_{cyl})}{\rho_d} \right)^{1/2} \quad \text{for } p_d > p_{cyl} \quad (16)$$

$$\frac{dm_{de}}{d\theta} = \frac{1}{\omega} \rho_{cyl} \alpha_d A_{dl} \left( \frac{2(p_{cyl} - p_d)}{\rho_{cyl}} \right)^{1/2} \quad \text{for } p_{cyl} > p_d \quad (17)$$

Where  $A_{dl}$  is the leakage flow area of the discharge valve,  $dm_{di}/d\theta$  is discharge leakage from the discharge chamber to the compressor cylinder and  $dm_{de}/d\theta$  is the leakage from the compressor cylinder to discharge chamber. The mass flow rate for discharge leakage to and from the discharge chamber,  $dm_{di}/d\theta$  and  $dm_{de}/d\theta$ , are equal to  $dm_{de}/d\theta$  and  $dm_{di}/d\theta$  for the cylinder chamber, respectively.

### 3. Calculation procedure

Modular structure of a single-stage reciprocating natural gas compressor (with fault in the suction or discharge valve) with the mathematical equations interactions is shown in Fig. 2.

There are various crank angle-dependent variables, including any variable in the mathematical model equations as well as its derivatives, which must be calculated in the simulation. Some of these crank angle-dependent variables, such as pressure and temperature of the cylinder, suction, and discharge chambers, can be used as the results for the simulation. There are other types of values which are called periodical results, such as average suction or discharge chamber temperature, and mass flow which must be considered in one cycle. These parameters vary over several cycles when there is a fault in the suction or discharge valve. These results may be more useful than their crank angle-dependent counterpart variables. In the reciprocating compressor with healthy valves, the governing equations need to be solved over one cycle iteratively to reach the steady-state variation of different variables. In the valve

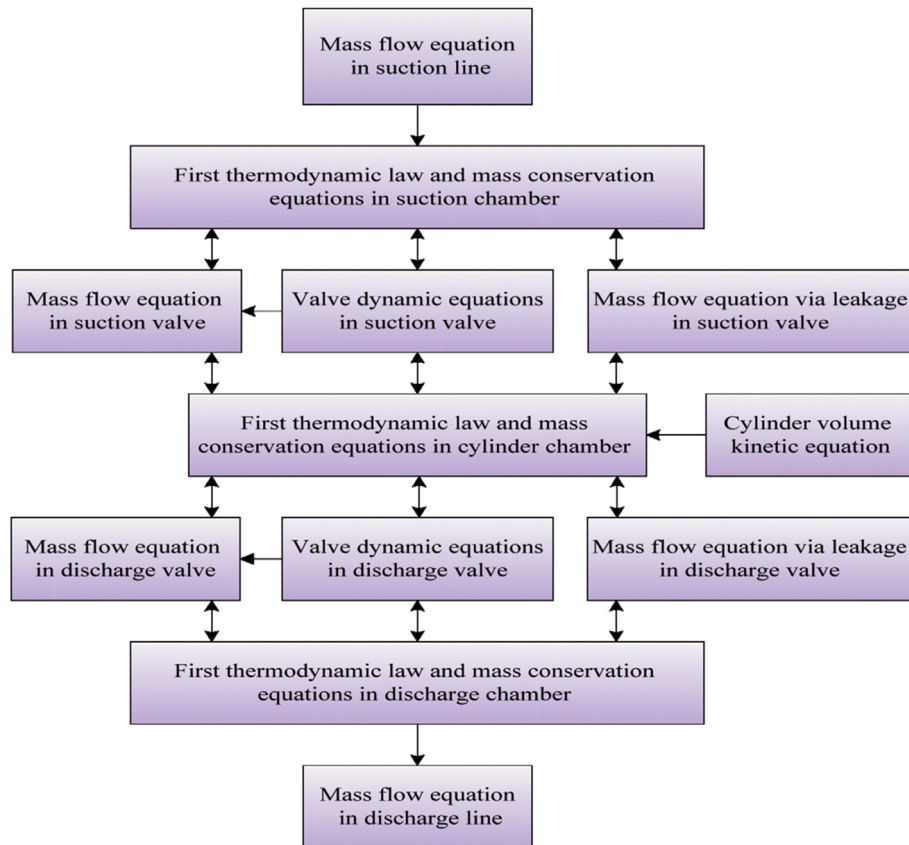


Fig. 2. Modular structure of the compressor model.

failure incident for both suction and discharge valve faults, these equations need to be solved over the cycles until periodical results, suction and discharge average temperatures per cycle remain constant. To this end, the flowchart that is shown in Fig. 3 is used in this simulation. The 4th order Runge-Kutta method is used to solve the differential equations in the mathematical modelling.

#### 4. Result and discussion

When compressor valves fail, gas re-compression takes place due to gas reverse flow (Hanlon, 2001). As a result, discharge gas temperature increases and reduces compressor performance parameters such as mass flow rate and efficiency. Furthermore, high discharge gas temperature has a detrimental effect on compressor valve system, including valve plate and its spring. So, discharge gas temperature must not exceed the limited temperature in the reciprocating compressor (Bloach, 1996).

The developed numerical method in this paper is implemented for a single-stage reciprocating air compressor and the simulated results are compared with the available measured values (Sun and Ren, 1994) in Fig. 4. According to Fig. 4, in the expansion process (A), when the piston moves from Top Dead End (TDE) to Bottom Dead End (BDE), the cylinder volume is increased and the gas that was trapped in the clearance volume expands and its pressure decreases. Then, in the suction process (B), when the cylinder pressure becomes smaller than the suction pipe pressure, the suction valve opens and the gas flows into the compressor cylinder while the piston keeps moving toward BDE. After that, as the piston reverses and moves from BDE to TDE, the cylinder pressure increases while the cylinder volume decreases. This process is called compression (C). Finally the discharge valve opens in the discharge

process (D), when the cylinder pressure overcomes valve force and pressure drop.

In this simulation, the considered reciprocating natural gas compressor has the specifications mentioned in Table 1. A small hole in the valve plate is assumed for the valve leakage modelling. Holes with 2, 4 and 6 mm diameters are considered for Leak 1, Leak 2, and Leak 3, respectively. The small diameter hole is considered for the modelling of small leakage and, in case of higher leakage, holes with larger diameters are implemented in the simulation.

##### 4.1. Suction valve failure

With the purpose to validate the reliability of the developed approach, the effect of the suction valve leakage on the dynamic cylinder pressure is shown in Fig. 5. During the expansion portion of the cycle, gas leaks out of the cylinder through the suction valve. There, therefore, is less gas in the compressor cylinder than the normal operation and the cylinder pressure is lower as well. As a result, the suction valve is opened sooner than that with normal angle which is consistent with the results reported by McCarthy, 1994 and Elhaj et al., 2008. In addition, during the compression and discharge portions of the cycle, gas leakage continues and gas exits through the hole in the suction valve from the cylinder chamber. So, the gas pressure in the cylinder chamber and clearance volume reduces further. As a result, the suction valve opening angle reduces much more owing to the lower clearance volume pressure. Cylinder chamber pressure during the suction process is higher than that under normal condition, which is due to the premature valve opening. In addition, mass flow due to leakage to the cylinder chamber in the suction portion of the cycle increases the cylinder chamber pressure too. McCarthy (1994) reported that

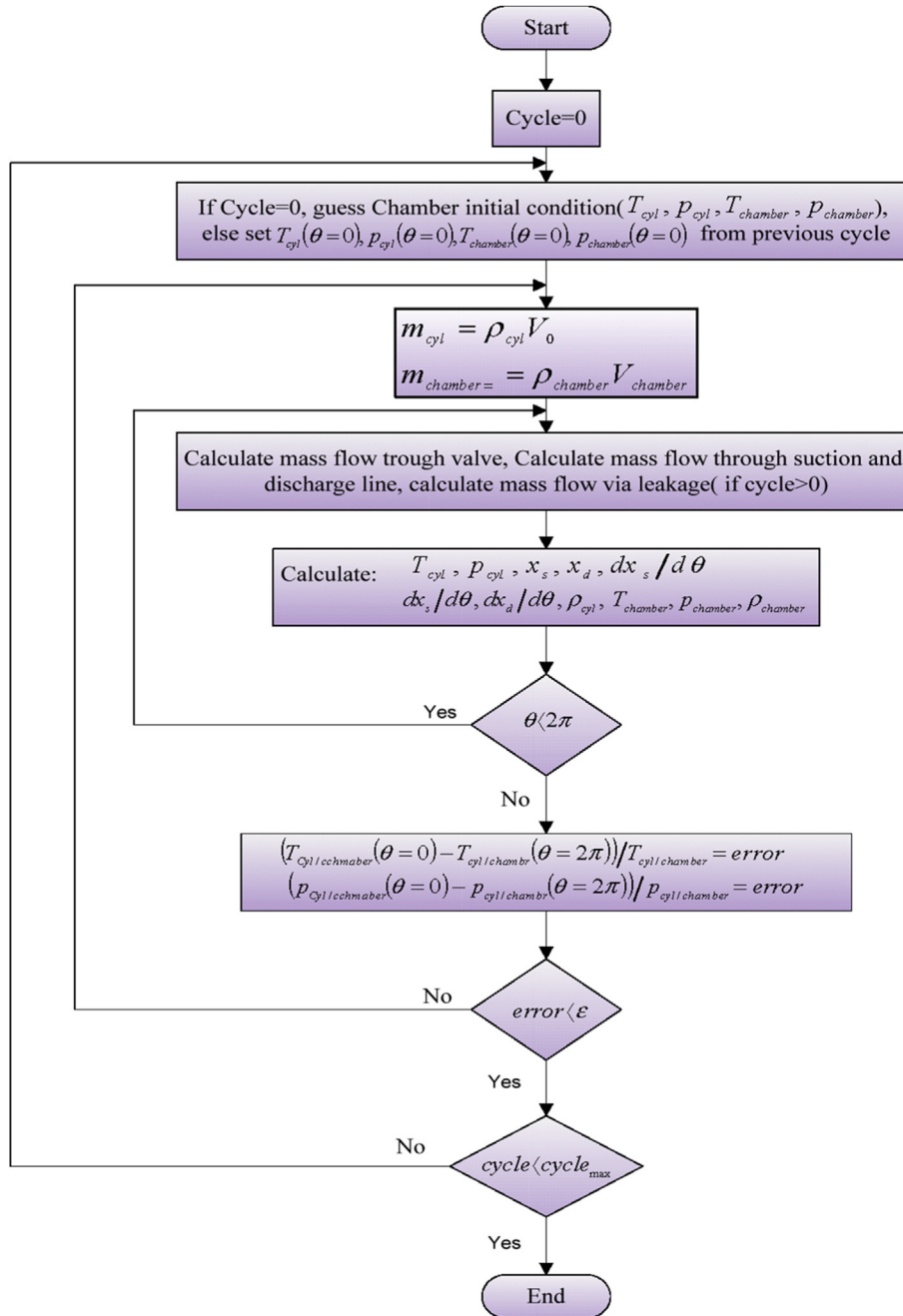


Fig. 3. Flowchart for the compressor simulation.

the discharge valve opens later and closes sooner than that normal angle because of lower dynamic cylinder pressure, which conforms to our prediction. This phenomenon is a consequence of continuous suction valve leakage during the expansion, compression, and discharge portions of the cycle. The opening angle of the suction valve decreases with increasing the suction valve leakage, illustrated Fig. 5. The suction valve closing angle also goes up when the related leakage increases. Furthermore, higher suction valve leakage leads to a delay in opening of the discharge valve and closing of the discharge valve ahead, observed in Fig. 5.

Fig. 6 shows the predicted average gas temperature per cycle of the suction chamber when the suction valve has leakage. In case of suction valve failure, as shown in Fig. 6, average gas temperature in the suction chamber increases rapidly and reaches a constant value.

As depicted in Fig. 7, during the expansion process when the suction valve fails, high temperature and pressure gas from the cylinder chamber are forced across the suction valve, and leaks into the suction chamber and increases its temperature. During the suction process, when the suction valve is opened, the suction chamber gas mixes with the low temperature incoming gas from the suction line and its temperature decreases. During the compression and discharge processes, when the suction valve is closed, gas temperature of the suction chamber increases again due to the reverse high temperature gas that leaks into the suction chamber from the compressor cylinder. This process keeps going through cycles and reaches to steady-state as shown in Fig. 7. As a result, average gas temperature of the suction chamber increases, as illustrated in Fig. 6. The effect of different valve leakages can be

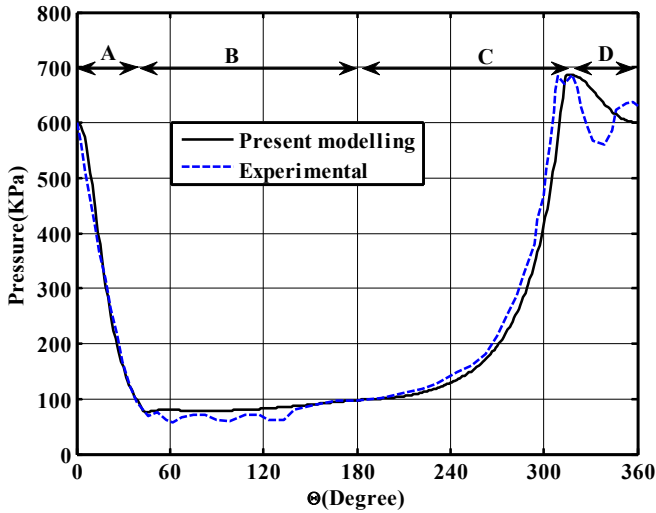


Fig. 4. Comparing the present modelling and experimental results (Sun and Ren, 1994).

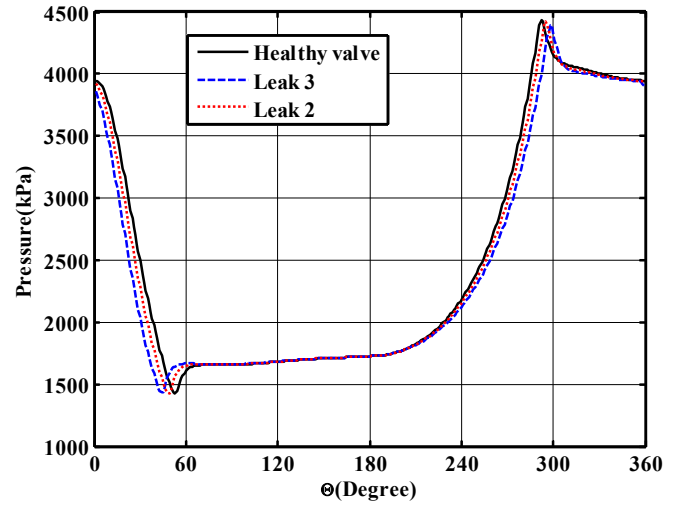


Fig. 5. Cylinder chamber pressure variation with crank angle for suction valve fault.

also observed in Fig. 6. The suction chamber average gas temperature increases as leakage goes up. Moreover, this figure shows that average gas temperature of the suction chamber needs more time to reach the constant value when leakage increases. Hereinafter, we simply name “average gas temperature per cycle” as “gas temperature”.

The effect of the suction valve leakage on the dynamic cylinder temperature is shown in Fig. 8. As illustrated previously in Fig. 6, the gas temperature of the suction chamber increases with suction valve fault. This high temperature gas is compressed in the compressor cylinder. As a result, the cylinder chamber temperature increases as shown in Fig. 8. Moreover, the cylinder chamber temperature increases as leakage goes up, as depicted in Fig. 8.

The suction valve fault could affect discharge chamber gas temperature as represented in Fig. 9. As illustrated previously in Fig. 8, the gas temperature of the cylinder chamber increases with suction valve fault, resulting in an increase of the gas temperature of the discharge chamber. After no change of the gas temperature of the discharge chamber over first few cycles, a sharp increase in the gas temperature of the discharge chamber occurs before it evens out. As a fact, increasing the suction gas temperature causes the gas temperature of the discharge chamber to increase. However, the pressure of the compressor cylinder chamber decreases, when there is a fault in the suction valve, Fig. 5, compensating the rise of the discharge chamber gas temperature in the first three cycles, Fig. 9. In addition, the results in Fig. 9 show that gas temperature of the discharge chamber has similar features to the gas temperature profile in the suction chamber. Discharge chamber gas temperature is increased as leakage goes up. The gas temperature of the discharge chamber approaches a constant value in more cycles

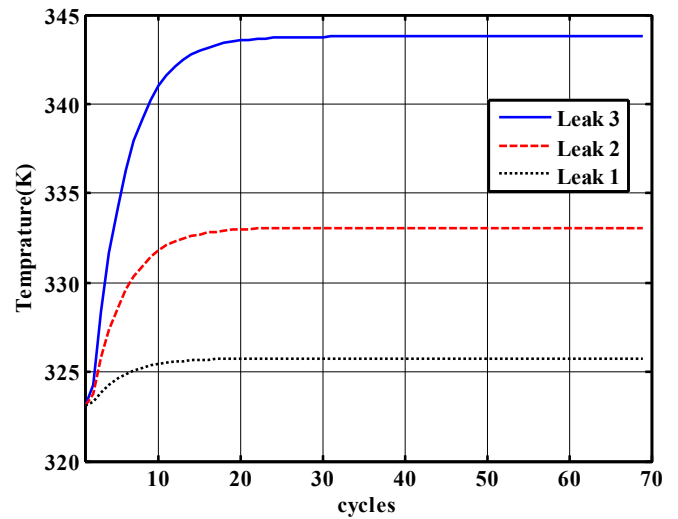


Fig. 6. Suction chamber average gas temperature variation with cycle for suction valve fault.

compared to that of the suction chamber.

The mass flow varies after the suction valve failure as depicted in Fig. 10. It can be concluded that the compressor mass flow reduces during first few cycles and then becomes constant. The physical reasons of mass flow reduction are as follow; 1) gas leaks out from the cylinder to the suction chamber; 2) inlet gas density decreases owing to the temperature increase in the suction chamber. Fig. 10 indicates that mass reduces too much as valve leakage increases, and the mass flow varies in much more cycles

Table 1

Operating condition and physical property of the compressor.

Cylinder	116 mm	Spring stiffness of valves	10000 N m <sup>-1</sup>
Connecting rod length	164.6 mm	Maximum valves lift	2.5 mm
Crank radius	41.15 mm	Flow coefficient (α)	0.5
Volume of suction chamber	0.004 m <sup>3</sup>	Rotational speed	1800 rpm
Flow area of suction pipe	0.0021 m <sup>2</sup>	Gas composition	Methane
Volume of discharge chamber	0.004 m <sup>3</sup>	Inlet pressure	1728 kPa
Flow area of discharge pipe	0.0021 m <sup>2</sup>	Inlet temperature	323.15 K
Radius of valves	30 mm	Outlet pressure	3945 kPa

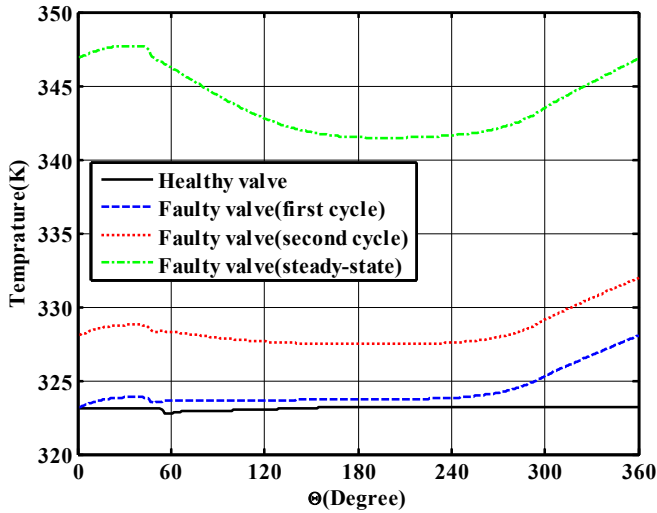


Fig. 7. Suction chamber gas temperature variation with crank angle (for Leak 3).

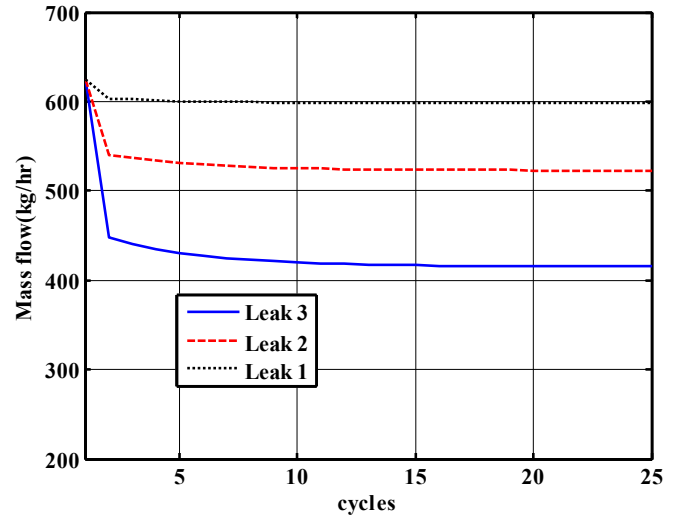


Fig. 10. Mass flow rate variation with cycle for suction valve fault.

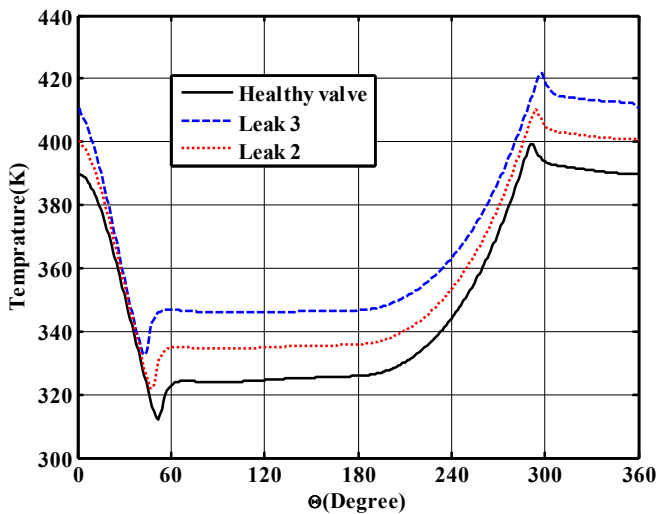


Fig. 8. Cylinder chamber temperature variation with crank angle for suction valve fault.

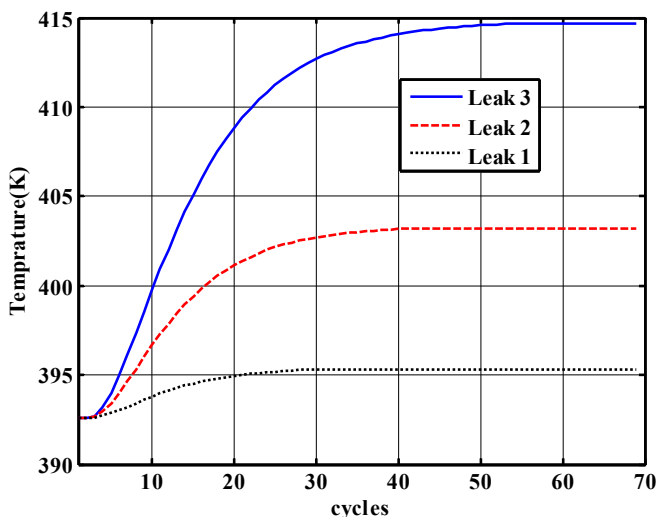


Fig. 9. Discharge chamber gas temperature variation with cycle for suction valve fault.

before reaching to a constant value.

#### 4.2. Discharge valve failure

The effect of the discharge valve leakage on the dynamic pressure of the compressor cylinder can be seen in Fig. 11. During the expansion process, gas leaks through the discharge valve to the compressor cylinder from the discharge chamber. Thus, in-cylinder pressure increases during the expansion process and causes the suction valve to open later compared to the normal valve which is in agreement with previous researcher's results (McCarthy, 1994). Furthermore, during the suction process, these continuous leakages from the discharge valve increases dynamic pressure and close the suction valve sooner than that the normal operation which is in line with McCarthy results (McCarthy, 1994). The opening angle of the suction valve increases with increasing the discharge valve leakage, as depicted in Fig. 11. It can be concluded that discharge valve opens sooner due to higher cylinder dynamic pressure which is in line with that available in the best-published literature (McCarthy, 1994). This phenomenon is a consequence of the continuous discharge valve leakage during the expansion, suction, and compression processes. In addition, the premature closure of the suction valve and the premature opening of the discharge valve increase as leakage goes up in the discharge valve, Fig. 11.

The dynamic cylinder temperature after discharge valve failure is shown in Fig. 12. With the discharge valve failure, high temperature gas from the discharge chamber leaks back to the compressor cylinder and mixes with the in-cylinder gas during the expansion, suction, and compression processes. This high temperature gas is compressed in the compressor cylinder and increases its temperature as depicted in Fig. 12. Furthermore, the cylinder chamber temperature increases as leakage goes up, as shown in Fig. 12.

Fig. 13 shows the discharge chamber gas temperature after the discharge valve failure. As presented in this figure, the gas temperature of the discharge chamber increases rapidly during first few cycles before becoming constant. With the discharge valve failure, as illustrated previously in Fig. 12, the gas temperature of the cylinder chamber increases. Consequently, the discharge chamber gas temperature increases, Fig. 13. As can be observed, with gas leakage increase in the discharge valve, discharge chamber gas temperature goes up as well. In addition, this figure shows that the gas temperature of the discharge chamber needs more cycles to



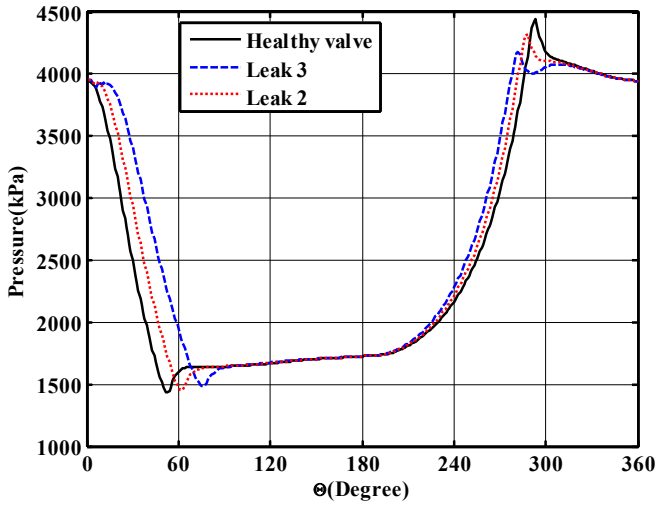


Fig. 11. Compressor cylinder pressure variation with crank angle for discharge valve fault.

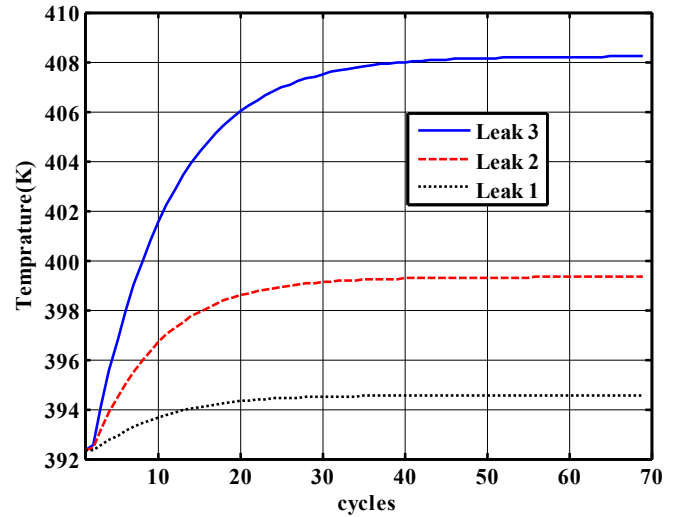


Fig. 13. Discharge chamber gas temperature variation with cycle for discharge valve fault.

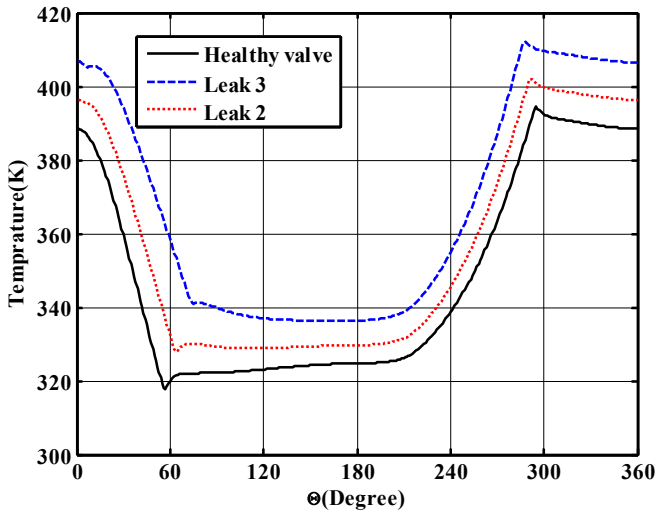


Fig. 12. Cylinder chamber temperature variation with crank angle for discharge valve fault.

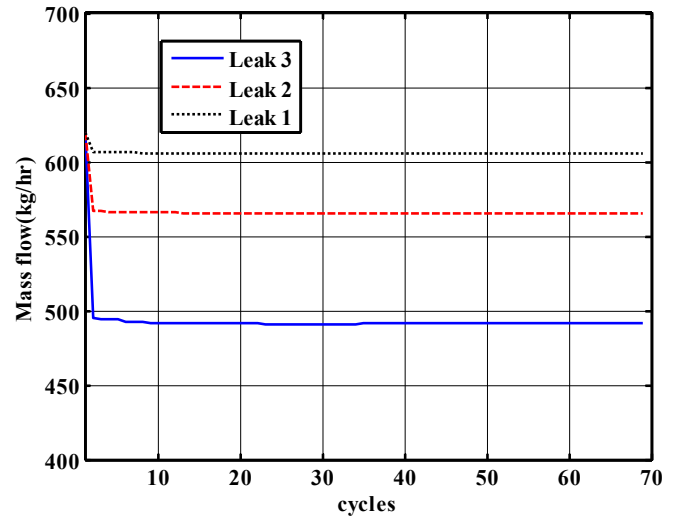


Fig. 14. Mass flow rate variation with cycle for discharge valve fault.

reach the constant value when leakage increases.

After discharge valve failure, the mass flow variation is evaluated as shown in Fig. 14. The simulation results show that, mass flow rate reduces rapidly during the first cycle and then remains constant. In other words, discharge valve leakage affects mass flow rate too much in the first cycle. Moreover, mass flow reduction goes up when leakage increases.

#### 4.3. Suction and discharge valve failure comparison

Fig. 15 shows the gas temperature of the discharge chamber comparison for two cases: i) the suction valve failure; ii) the discharge valve failure. The gas temperature for the former case is higher than the later one and reaches the constant value after more cycles. This finding is of paramount importance to acquire valve faults due to temperature limitation for discharge valve. If a very high temperature of the discharge valve is observed when the cooling system works properly, the first fault candidate could be a failure in the suction valve, Fig. 15.

As shown in Fig. 16, in the suction valve fault, mass flow reduces significantly compared to the discharge valve fault. For the suction valve fault, both back flow and decreasing suction chamber gas density affect mass flow rate and reduce it through cycles. In the discharge valve fault, back flow is the dominant reason for the mass flow reduction and reduces it in one cycle abruptly.

### 5. Conclusion

Reciprocating compressors are prone to valve failure, for which instant diagnosis is very crucial. A zero-dimensional numerical method was developed for a single-stage reciprocating natural gas compressor with suction and discharge valve faults. In addition to the compressor cylinder, suction and discharge chambers were implemented in the modelling to monitor gas temperature in the suction and discharge chambers. Crank-angle dependent variables such as temperature and pressure and periodical results including mass flow, and average gas temperature per cycle of the suction and discharge chambers were calculated. The validity of the results from the modelling was confirmed by previous work on healthy

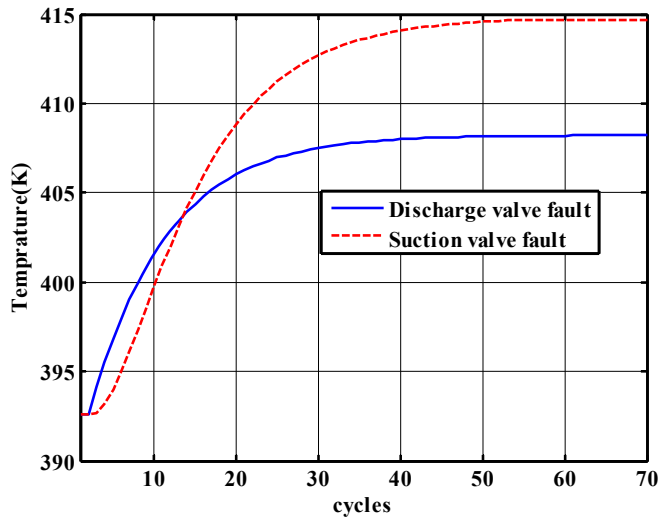


Fig. 15. Comparing discharge chamber gas temperature variation for suction and discharge valve fault (Leak 3).

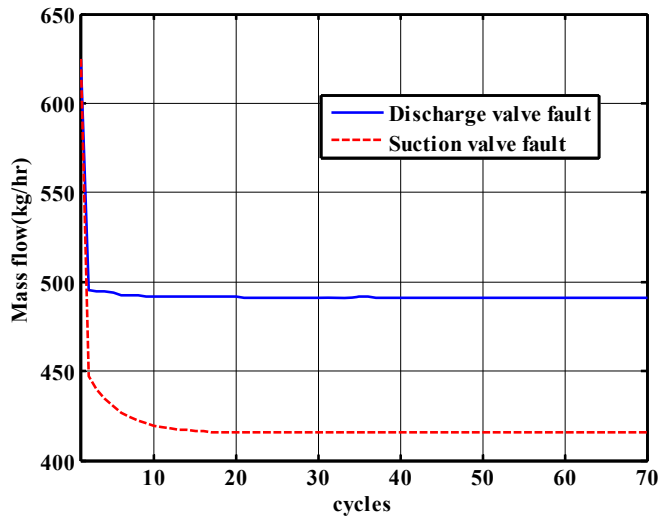


Fig. 16. Comparison of mass flow rate with cycle for suction and discharge valve fault (Leak 3).

valves. This simulation showed that suction and discharge valve leakages decreased mass flow rate and increased discharge gas temperature. Discharge temperature went up and mass flow rate reduced when leakage increased in the suction or discharge valve. Furthermore, the predicted results demonstrated that the effect of suction valve fault on reducing mass flow rate and increasing discharge gas temperature was higher than discharge valve fault. In addition, suction valve fault increased the gas temperature of the suction chamber. The results represented that suction and discharge gas temperatures monitoring can be used as a simple but effective method for finding valve faults in the reciprocating compressors.

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