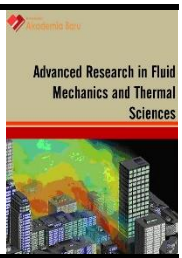
Penerbit Akademia Baru

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences

Journal homepage: www.akademiabaru.com/arfmts.html
ISSN: 2289-7879



Study the Effect of Unstable Air Flow in the Suction and Discharge System on the Performance of Reciprocating Air Compressor

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ARTICLE INFO

Article history:
Received 14 July 2019
Received in revised form 13 October 2019
Accepted 15 October 2019
Available online 26 January 2020

ABSTRACT

This research aims at studying the effect of unstable air flow in the suction and discharge system on the performance of reciprocating air compressor. The study was conducted on a reciprocating air compressor with two cylinders diagonally, which operates in three stages to generate the required pressure and cooling between the two cylinders by the air. To clarify the flow, three different pipes in length and diameter were used. A mathematical model was prepared to calculate the one-dimensional unstable flow in the pipes by using the characteristics method and the flow relationships that include effect of friction losses, change of entropy and heat transfer; as well as, the compressive waves formed in the pipes with their effect on the flowing mass inside the cylinder and the dynamic behavior of the compressor valves were predicted. In addition, the pressure, temperature, volume, mass and efficiency were calculated, and an optimal method was adopted to solve the differential equations system through numerical analysis and by using Rung-Kutta method. The results manifested that the use of multi-stage reciprocating air compressor gives a high efficiency due to the air cooling, and the length and diameter of the suction pipe and the speed of compressor have an effect on the capacity of compressive waves which affect the final pressure generated by the compressor that causes an increase in volumetric efficiency.

Keywords:
Reciprocating air compressor; Volumetric efficiency; Suction pipe

1. Introduction

The importance of reciprocating air compressor, which has many uses in various fields, includes the industrial and commercial applications. Therefore, it has increased the interest of researchers to put many studies and researches to predict the dynamic behavior for the reciprocating air compressor, and then improve the design and performance of the reciprocating air compressor to

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serve the industry. It helped the existing competition between the different international companies in developing the methods used to improve the design of the suction and discharge system.

The effect of the compressive waves formed in the entry and discharge pipes should be studied as well as the interferences for the compressive waves formed at the junctions when the air charge flows across the suction and discharge systems, where these waves can help or impede the flow process and thus affect the performance of reciprocating air compressor.

The compressive waves are generated in each suction and discharge pipes when the suction and discharge valves are opened, respectively, and as a result for the pressure difference between the cylinder and pipe, the amount of air flows in an unstable method, then these waves transfer inside the pipe and bounced when there are certain delimiters such as narrowing which is represented by the valve orifice. There is a series of studies in the research and development field.

Castagliola [1] studied the theory of basic design for the dynamics of self-active compressor valves, which contain the springs (loaded valves) and operate at the opening and closing on the pressure difference across the valve. Through the study, it was shown that the valve performance depends on the piston speed and effective flow area through the valve. In addition, the dynamic valves depend on the valve weight and spring stiffness, where they decrease each of the valve weight and spring stiffness which helps to increase the effective flow area, and thus the volumetric efficiency increases. Benson *et al.,* [2] investigated a numerical solution of the one-dimensional unstable flow, where it has introduced the effects of heat transfer, friction and entropy in the pipes. A comprehensive mathematical program was prepared in FORTRAN language for the compression ignition engine which includes a study of the effect of wave in the entry and exhaust systems by using the characteristics method. Giacomell *et al.,* [3] presented a mathematical model that includes the study of operational conditions to the compressor cylinder valves with the study of compressive waves formed in the pipes attached to the compressor. Mahmood Attallah [4] conducted a study on pressure changes and analysis of compressive waves in the suction pipe by using the characteristics method for flow to the multi-cylinder engine system. Schemes were put to show the effect of diameter and length of the entry pipe and timing of the entry valve on the volumetric efficiency. Enrico and Davide [5] conducted an experimental investigation on the performance of a semi-hermetic reciprocating compressor that operates by propane as a refrigerant. Tests were carried out with and without an internal heat exchanger between the vapor suction line and the liquid line at the different modes for extension valve. It was found that the superheating of suction vapor improves the volumetric efficiency; also, the high solubility of propane in mineral oils used causes an excessive reduction in the viscosity of the refrigerant oil mixture at low suction superheating. Kim *et al.,* [6] studied the use of suction muffler to control the noise generated by fluctuations of the impulsive pressure resulting from piston/valve in the reciprocating compressor by using the numerical scheme for the simulation of acoustic waves of the suction muffler. Through the study, it was found that the suction muffler has an adverse effect on the performance of compressor because it causes an additional pressure drop and heat transfer. Yusof *et al.,* [7] performed a numerical study on the adiabatic piston-cylinder to investigate the irreversible process; model of laminar flow was used to obtain the equations of momentum, continuity, and energy. The Arbitrary-Lagrangian-Eulerian and Continuous-fluid Eulerian methods were utilized. It was found that the irreversible process is affected by the cylinder diameter size; the piston surface average pressure is increased by increasing the diameter of the cylinder. Jawad *et al.,* [8] performed a numerical study intended to predict the performance of a modified centrifugal compressor used in turbo charger. In order to increase the performance of the conventional turbocharger compressor's to boost the pressure in the engine, by the impeller trimming. According to the results of the simulation, it was observed that the fluid flow into a compressor has been indicated a better understanding. In addition, the compressor performance was heavily affected by the impeller modification. Conclusively, it was observed the performance it was greatly affected by the double splitter design and the pressure ratio and air mass flow rate were increased. Ozsipahi *et al.,* [9] numerically and experimentally investigated the lubrication system of a compact inverter compressor (CIC). In the numerical modeling, a finite volume-based algorithm was used to model two-phase (air–oil) flow inside the compressor using Volume of Fluid Method (VoF) method. It was shown that with increasing crankshaft speeds, the average oil mass flow rate released from the upper part of the crankshaft was increasing almost linearly. It was also shown numerically that with increasing oil viscosity, the mass flow rate decreases.

The aim of this research is to study the effect of unstable air flow in the suction and discharge system on the performance of reciprocating air compressor using three different pipes in length and diameter. A Rung-Kutta method was used as a numerical analysis to calculate the one-dimensional unstable flow in the suction and discharge pipes. All tests were conducted, and the results obtained for the performance of reciprocating air compressor were discussed.

2. Theoretical Analysis

The reciprocating air compressor used in this research is of type (V), which contains two cylinders and the angle between them is (90°), where it works in three stages to generate the final pressure amount (150 bar) at different speeds (1000, 2000 and 3000 rpm). The first and second stages to generate the pressure were done in the first cylinder, and the third stage to generate the pressure was done in the second cylinder. The mathematical model was solved using a Rung-Kutta method of the fourth limit and FORTRAN program.

2.1 Calculation the Unstable Flow in Pipe

The basic equations to calculate the fluid flow in the suction and discharge pipes for the reciprocating compressor have to be the partial differential equations; therefore, the formulas of the equations must be changed so that they become applicable in point.

The characteristics method was adopted and by which the differential equations of type (hyperbolic) are reduced to a set of the ordinary equations, which are considered the basic form of numerical solution in the unstable flow. In the current study, the theoretical analysis depends on four basic principles of the one-dimensional unstable flow: continuity equation, momentum equation, energy equation and equation of entropy change for the liquid particles. Where, the cross-sectional area of the pipe is constant and the gas is ideal.

Continuity Equation

$$\frac{\partial \rho}{\partial t} + \rho \frac{\partial u}{\partial x} + u \frac{\partial \rho}{\partial x} = 0 \tag{1}$$

Momentum Equation

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + \frac{1}{\rho} \frac{\partial P}{\partial x} + G = 0 \tag{2}$$

where,

$$G = f \frac{u^2}{2} * \frac{u}{|u|} * \frac{4}{D} \tag{3}$$

where,

$\frac{u}{|u|}$: represents the friction effects on the opposite direction to the direction of flow.

$$f = \frac{\tau_w}{\frac{1}{2}\rho\, u^2} = constant \quad (4)$$

Pipe Section Area

$$A = \frac{\pi}{4} * D^2 \quad (5)$$

Energy Equation

$$\left(\frac{\partial P}{\partial t} + u\frac{\partial P}{\partial x}\right) - a^2\left(\frac{\partial \rho}{\partial t} + \frac{\partial \rho}{\partial x}\right) - (k-1)\rho\,(q + u\,G) = 0 \quad (6)$$

Equation of Entropy Change

$$\frac{ds}{dt} = \frac{q+u\,G}{T} \quad (7)$$

Sound Speed of the Ideal Gas

$$a^2 = \frac{k*P}{\rho} = kRT \quad (8)$$

Three different types of pipes were selected in length and diameter, as shown in Table 1.

Table 1
Dimensions of the pipes

Pipe No.	Pipe Length L (mm)	Stroke S (mm)	L/S	Pipe Diameter D (mm)	Cylinder Bore B (mm)	D/B
1	800	131	6.1	40	46	0.869
2	600	131	4.5	30	46	0.652
3	400	131	3	20	46	0.434

2.2 Valve Calculation

The valves of a reciprocating air compressor are considered as check valves and through them the inlet and outlet flow for the compressor or cylinder can be controlled. In each compression chamber, there must be at least one suction valve and one discharge valve [10].

2.2.1 Mass flow through valves

For the purpose of calculating the entering and leaving mass from the cylinder across the suction and discharge valves, it was assumed the followings:

i. The one-dimensional flow is isotropic.
ii. Stable flow equations can be applied to calculate the instantaneous values through the unstable flow.

iii. For the purpose of calculating the instantaneous flow area of the open valve, the valve orifice can be considered as a simple orifice with a certain effective section area and a perfect discharge through the orifice regardless of the valve shape through the first law of thermodynamics, Figure 1 shows the mass flow through the valve.

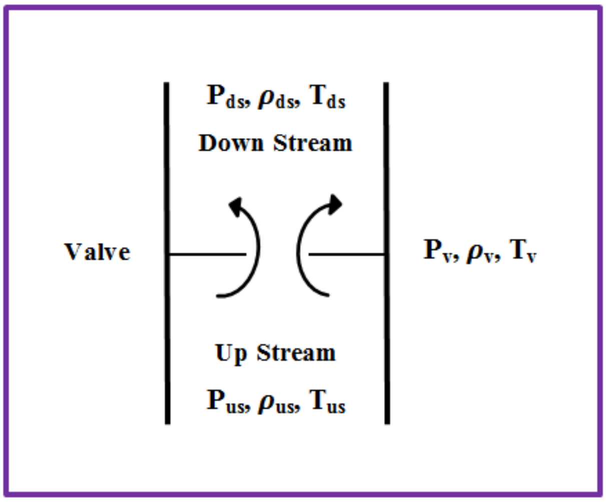


Fig. 1. Mass flow through the valve

$$h_u = h + \frac{u^2}{2} \tag{9}$$

$$h_u - h = C_P\,(T_u - T)$$

As for the ideal gas

$$k = \frac{C_P}{C_V} \quad ; \quad R = C_P - C_V \quad ; \quad C_P = \frac{k}{k-1} * R$$

$$\therefore \frac{kR}{k-1}\,(T_u - T) = \frac{u^2}{2} \tag{10}$$

Sound speed

$$a = \sqrt{kRT} \tag{11}$$

The Mach number can be expressed in the following relationship [11],

$$M = \frac{u}{a} = \sqrt{\frac{2}{k-1}\left(\frac{T_u}{T} - 1\right)} \tag{12}$$

$$\frac{T_u}{T} = \left(\frac{P_u}{P}\right)^{\frac{k-1}{k}}$$

$$\therefore M = \sqrt{\frac{2}{k-1}\left[\left(\frac{P_u}{P}\right)^{\frac{k-1}{k}} - 1\right]} \tag{13}$$

The mass flow rate through the valve is given by the following expression [12],

$$\dot{m}_v = \rho_v \, A_v \, u_v \tag{14}$$

From Eq. (12),

$$u_v = a_v * M_v$$

$$\therefore \ \dot{m}_v = \rho_v \, A_v \sqrt{kRT} * \sqrt{\frac{2}{k-1}\left[\left(\frac{P_u}{P_v}\right)^{\frac{k-1}{k}} - 1\right]} \tag{15}$$

But,

$$\rho_v = \frac{P_v}{R\,T_v} \qquad ; \qquad \rho_u = \frac{P_u}{R\,T_u}$$

Also, $\left(\frac{P_v}{P_u}\right)^{\frac{1}{k}} = \frac{\rho_v}{\rho_u} \qquad ; \qquad \left(\frac{T_v}{T_u}\right)^{\frac{k}{k-1}} = \frac{P_v}{P_u}$

$$\therefore \ \rho_v = \left(\frac{P_u}{R\,T_u}\right)\left(\frac{P_v}{P_u}\right)^{\frac{1}{k}}$$

One gets

$$\dot{m}_v = A_v \, P_u \sqrt{\frac{2k}{(k-1)\,R\,T_u}} * \sqrt{\left(\frac{P_v}{P_u}\right)^{\frac{2}{k}} - \left(\frac{P_v}{P_u}\right)^{\frac{k+1}{k}}} \tag{16}$$

Mass flow rate through the discharge valve can be calculated [12];

$$\dot{m}_{vd} = A_{vd} \, P_{ud} \sqrt{\frac{2k}{(k-1)\,R\,T_{ud}}} * \sqrt{r_d^{\frac{2}{k}} - r_d^{\frac{k+1}{k}}} \tag{17}$$

Mass flow rate through the suction valve can be calculated [12];

$$\dot{m}_{vs} = A_{vs} \, P_{us} \sqrt{\frac{2k}{(k-1)\,R\,T_{us}}} * \sqrt{r_s^{\frac{2}{k}} - r_s^{\frac{k+1}{k}}} \tag{18}$$

2.3 Cylinder Calculation

When the suction valve is opened, the air direction will be from the pipe to the compression space, and because of the difference between the compression space and the pipe, the compressive waves will be generated in the pipe. These waves transferred through the pipe and change at every moment. To analyze these waves, the pressure change and temperature in the compression space must be calculated. Table 2 shows the geometric dimensions of the reciprocating air compressor used in this research with different speeds (1000, 2000 and 3000 rpm).

2.3.1 Stroke calculation

The stroke is a displacement through which the piston moves from the top dead center (TDC) to the bottom dead center (BDC), and the instantaneous displacement is calculated from the following equation [13]. The boundaries used in this study are cylinder, suction valve and entry pipe.

$$Z(t) = R_1 \left[1 - \cos\theta(t) + n - \sqrt{n^2 - sin^2\theta(t)}\right] \tag{19}$$

Table 2
The geometric dimensions of the reciprocating air compressor used

First Cylinder	Dimension (mm)	Second Cylinder	Dimension (mm)
Radius of Crank	18	Radius of Crank	18
Radius of Piston	23 20	Radius of Piston	17 19
Cylinder Bore	46 40	Cylinder Bore	35 38
Cylinder Length	131	Cylinder Length	109
Length of Connecting rod	68.615	Length of Connecting rod	68.095
The Distance Between the Surface Piston and Cover	1.2 – 1.7		

2.3.2 Volume calculation

The cylinder volume (compression space) is calculated from the following equation [11].

$$V(t) = V_c + V_d$$

$$V(t) = V_c + \frac{\pi}{4} D^2 * Z(t)$$

$$V(t) = V_c + \frac{\pi}{4} D^2 * R_1 \left[1 - \cos\theta(t) + n - \sqrt{n^2 - sin^2\theta(t)}\right] \tag{20}$$

The rate of change in volume is calculated from the following equation.

$$\frac{dV_t}{dt} = \frac{\pi}{4} D^2 * R_1 \left[\sin\theta(t) + \frac{1}{2}\left\{\frac{\sin 2\theta(t)}{\sqrt{n^2 - sin^2\theta(t)}}\right\}\right] * 2\pi V \tag{21}$$

2.3.3 Pressure calculation

The cylinder pressure is calculated from the following equation [14].

$$\frac{dP_c}{dt} = \frac{1}{V_c}\left\{a_{05}^2 \left(\frac{dm}{dt}\right) - a_c^2 \left(\frac{dm}{dt}\right)_d - kP_c \frac{dV_c}{dt} + (k-1)\frac{dQ}{dt}\right\} \tag{22}$$

2.3.4 Temperature calculation

The temperature in the compression space is calculated from the following equation [15].

$$T_C = \frac{P_C * V_C}{R * m_C}$$

$$\therefore \frac{dT_c}{dt} = \frac{1}{R*m_C}\left\{a_{05}^2 \left(\frac{dm}{dt}\right)_s - a_c^2 \left(\frac{dm}{dt}\right)_d + (k-1)\frac{dQ}{dt} - (k-1)P_c\frac{dV_c}{dt}\right\} - \frac{T_c}{m_c}\left\{\left(\frac{dm}{dt}\right)_s - \left(\frac{dm}{dt}\right)_d\right\} \quad (23)$$

2.4 Volumetric Efficiency Calculation

The Volumetric efficiency is calculated from the following equation.

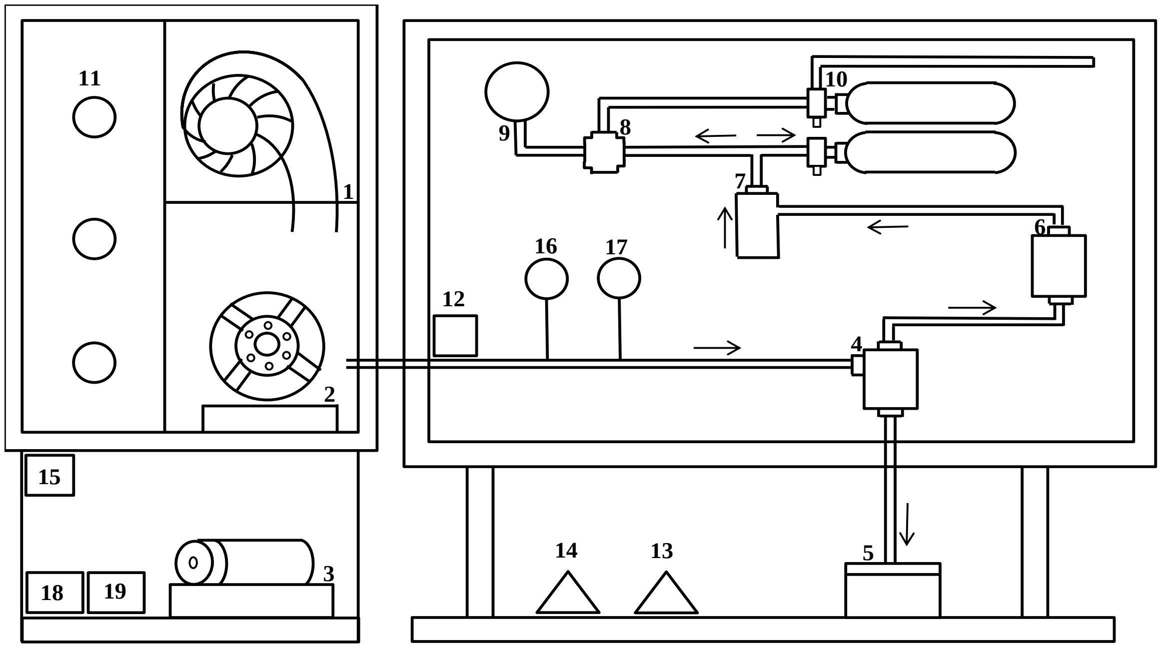
$$\eta_v = \frac{mass\ of\ air\ to\ cylinder\ per\ cycle}{\rho_o * V_d} \quad (24)$$

3. Experimental Work

The experimental work in this study was conducted on a multi-stage reciprocating air compressor consisting of two cylinders in the form of (v) and by using three different pipes in length and diameter for air flow. The test device used consists of a reciprocating air compressor installed on the base inside a testing platform. A testing platform contains a panel with five gauges, three of them for reading the pressures and two gauges for measuring the temperature, a fan was installed on the upper side of the testing platform for cooling the compressor during the work, the testing platform also includes an electrical control unit. In addition, the device used consists of three copper pipes to transfer the compressed air from the second cylinder of the compressor to the air storage cylinders, and to discharge the excess compressed air, a control valve was used. The test device comprises an electrical motor to operate the compressor, and Figure 2 depicts a schematic diagram of the experimental test device.

The tests were conducted by running the reciprocating air compressor, where the final pressures in the first, second and third stages were recorded at different speeds (1000, 2000 and 3000 rpm). The test was repeated for three cases on the reciprocating air compressor by using three different pipes in length and diameter, where in each case, one of these pipes was used.

The pressure gauge of type (ITEC EN 837-1) was used to measure the final pressure required for the first, second and third stages. An electric tachometer of type (DRAG Specialties 2.4) was also used to measure the speed of the compressor.



1. Cooling Fan.
2. Installation Base.
3. Electrical Motor.
4. Humidity and Oil Insulation.
5. Oil Container.
6. Pressure Valve.
7. Humidity and Oil Insulation.
8. To Reduce Valve.
9. Pressure Gauge.
10. Compression Air Cylinder.
11. Electrical Control Unit.
12. Control Valve.
13. Tachometer.
14. Electrical Dynamometer.
15. Rato meter.
16. Oil Temperature Gauge.
17. Oil Pressure Gauge.
18. Spring Balance.
19. Resistance Loading Unit.

Fig. 2. Schematic diagram of the experimental setup

4. Results and Discussion

The tests were conducted to evaluate the performance of a reciprocating air compressor consisting of two cylinders in the form of (v) which operates in three stages to generate the required pressure. All tests were carried out at different speeds (1000, 2000 and 3000 rpm), and by using three different pipes in length and diameter for air flow. The pressure, temperature changes and speed of air inside the cylinder of a reciprocating air compressor and in the suction and discharge pipes were calculated theoretically using a Rung-Kutta method. Through the experimental results obtained, it is found that there is a good agreement with the theoretical results, and the results obtained were discussed as shown in below:

Figure 3-5 reveal the variation between the pressure changes inside the cylinder for the first, second and third stages, respectively and at different speeds (1000, 2000 and 3000 rpm) versus the crank angle. The change in pressure inside the cylinder increases when the discharge valve is opened, i.e. when the piston starts moving up. Through these figures, it is found that the pressure reaches its highest value when the piston reaches to the top dead center (compression space), this is due to the exit of a larger amount of compressed air inside the cylinder to the receiving cylinders of compressed air through the discharge pipe, and then the pressure begins gradually to decrease due to the pressure drop inside the cylinder. Also, through these figures, it is found that the change in pressure inside the cylinder increases with the increase of compressor speed, this is because the increasing of compressor speed is a function of the increase of the piston speed inside the cylinder, and this increase in piston speed leads to increase the pressure during the compression stroke.

It was obtained that the highest value of the pressure inside the first cylinder (in the first and second stages) reached to 100 bars and in the second cylinder (third stage) 150 bars.

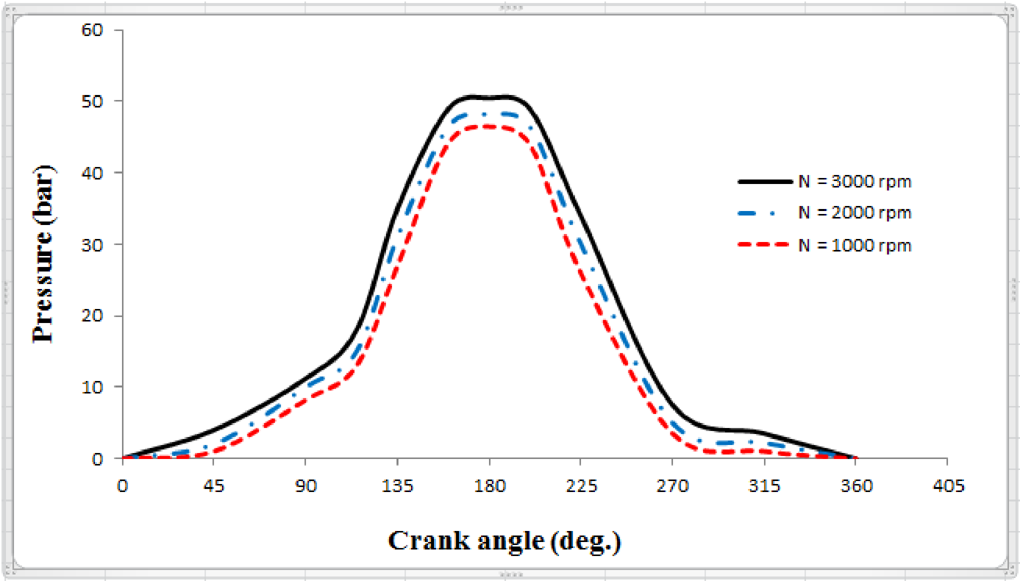


Fig. 3. The pressure changes in the compression space for the first stage at different speeds versus crank angle

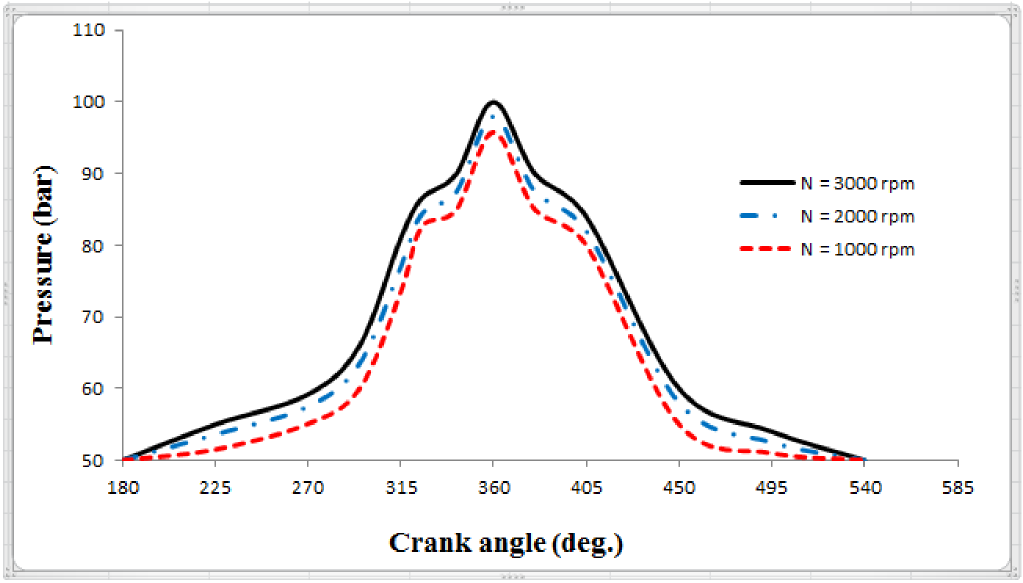


Fig. 4. The pressure changes in the compression space for the second stage at different speeds versus crank angle

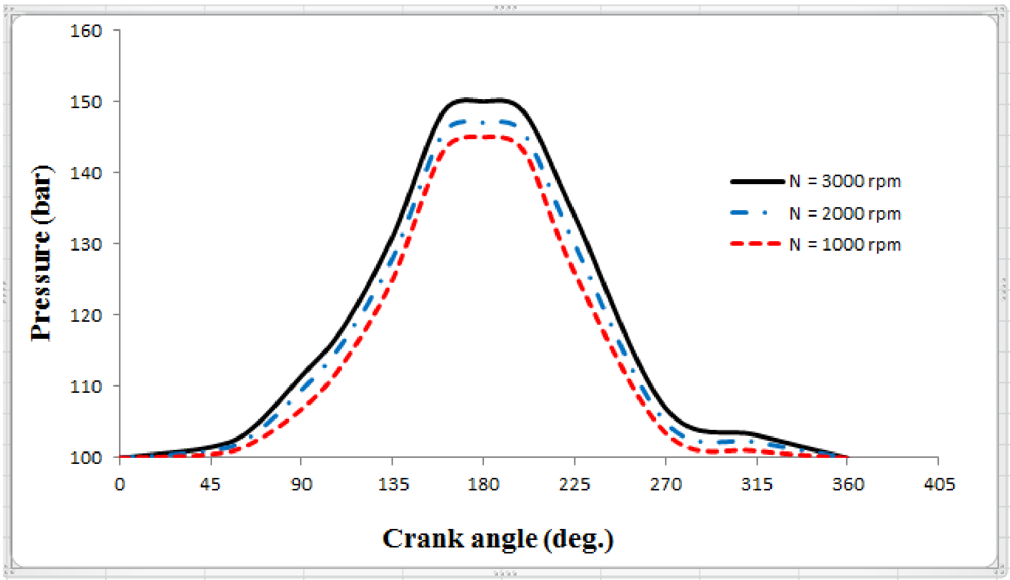


Fig. 5. The pressure changes in the compression space for the third stage at different speeds versus crank angle

Figure 6-8 illustrate the temperature change inside the cylinder of the first, second and third stages, respectively for the different speeds (1000, 2000 and 3000 rpm) versus the crank angle. When the air charge enters inside the cylinder and the piston starts moving up, the compression process will lead to an increase in the temperature inside the cylinder. Through these figures, it was observed that the temperature reaches to its highest value when the piston reaches to the top dead center (compression space), this is due to the pressure rise inside the cylinder, and then the temperature gradually decreases and almost reaches to the initial temperature, this is owing to the presence of air cooling between the three stages to reduce the friction and oil viscosity between piston rings. It was also found that increasing the compressor speed leads to increase the pressure inside the cylinder and thus the temperature increases.

It was obtained that the highest value of the temperature inside the first cylinder (in the first and second stages) reached to 370 K and in the second cylinder (third stage) 410 K.

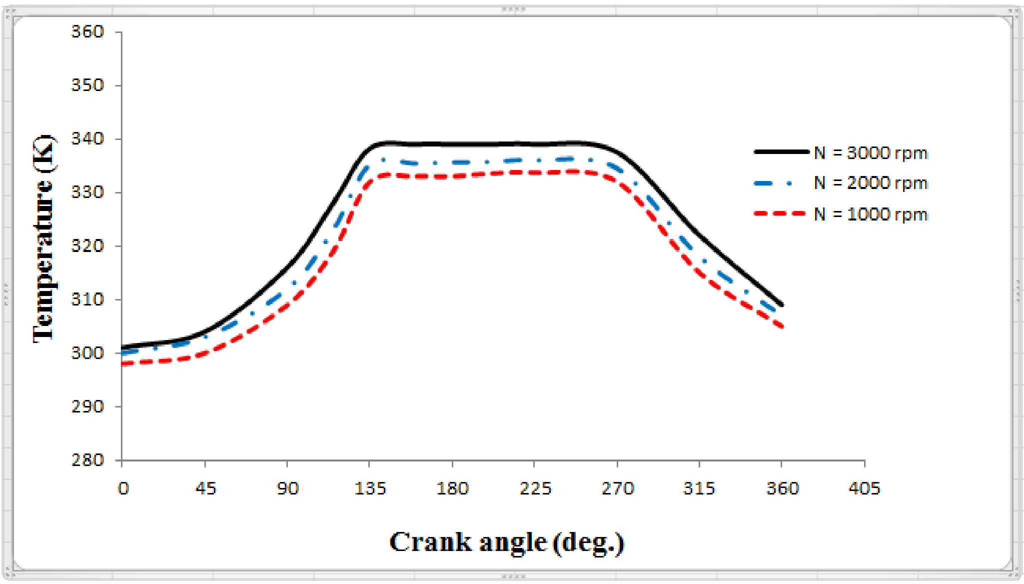


Fig. 6. The temperature change in the compression space for the first stage at different speeds versus crank angle

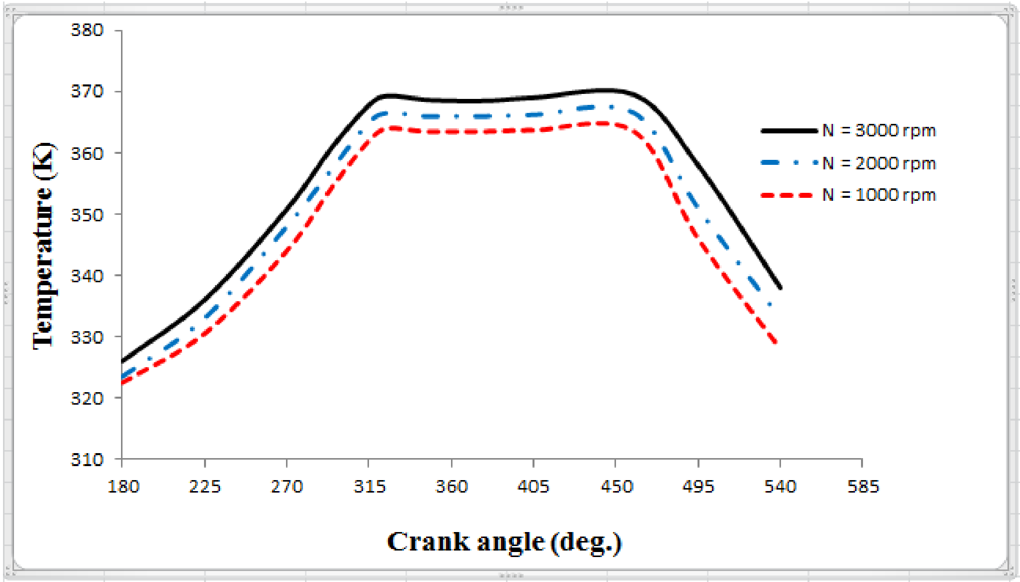


Fig. 7. The temperature change in the compression space for the second stage at different speeds versus crank angle

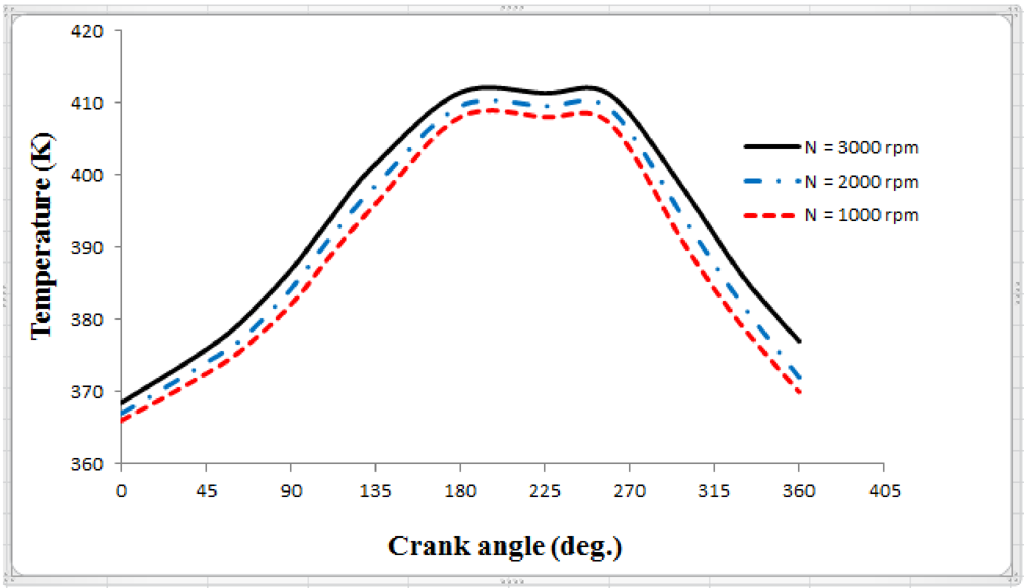


Fig. 8. The temperature change in the compression space for the third stage at different speeds versus crank angle

Figure 9(a)-(c) demonstrates the pressure change at the end of suction pipe at different speeds (1000, 2000 and 3000 rpm), respectively versus the crank angle. Through this figure, it was observed that the presence of changes in the compressive waves for the end of suction pipe near the cylinder is because of the presence of a difference in pressure between the suction pipe and cylinder, this difference causes the easy entry of air charge inside the cylinder. In addition, the figure manifests that the pressure change at the end of suction pipe increases with the increase of the speed of reciprocating air compressor.

It was found that the highest value of pressure at the end of suction pipe for speeds (1000, 2000 and 3000 rpm) is (0.00435, 0.025 and 0.05 bar), respectively.

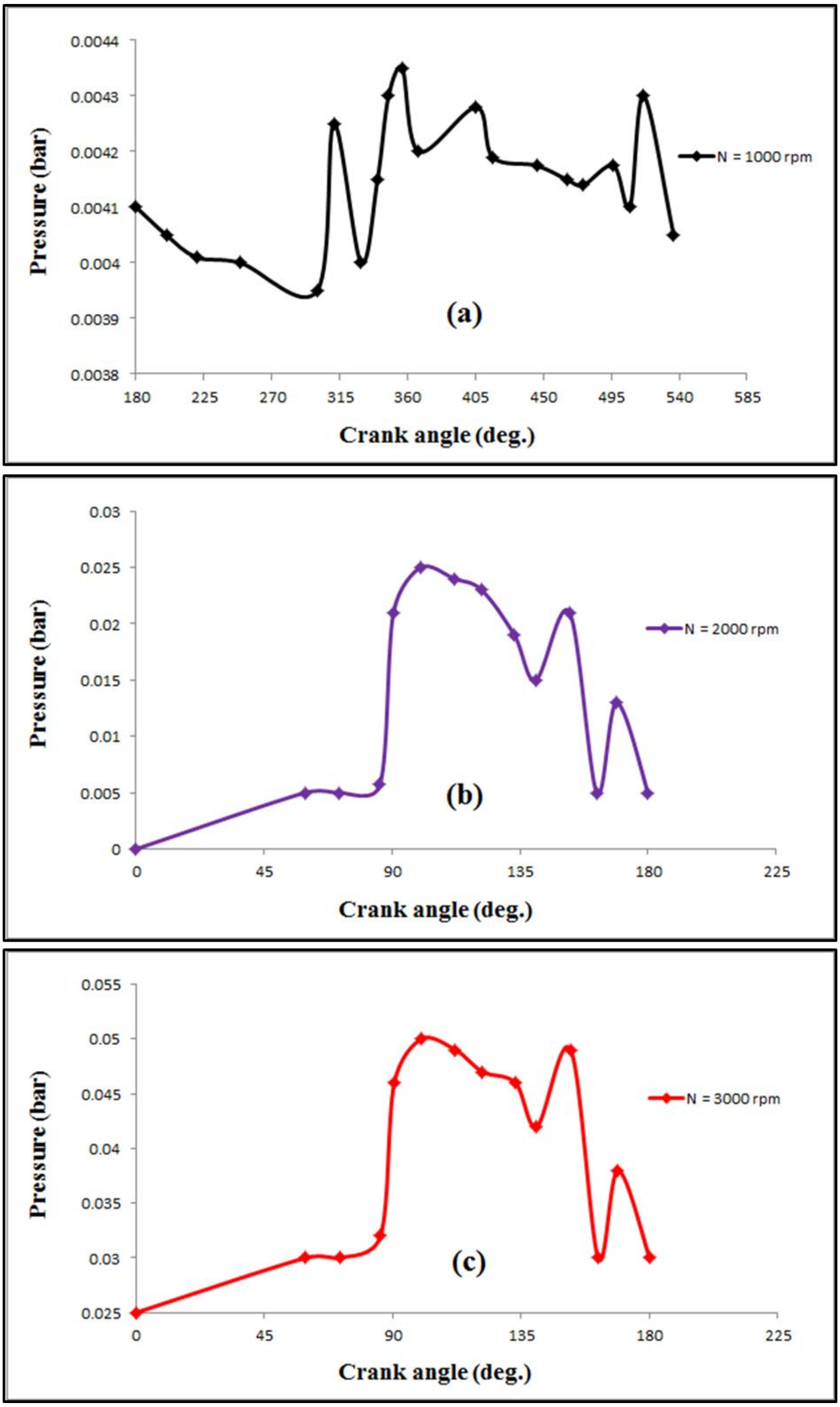


Fig. 9. The pressure changes at the end of suction pipe at speeds (a) 1000, (b) 2000, and (c) 3000, respectively versus crank angle

Figure 10 evinces the mass of air charge flowing into the first compression space at different speeds versus crank angle. Through this figure, it was found that there is a slight oscillation of the mass of air charge flowing, this oscillation is due to the movement of the suction valve and ease of compressor piston speed. Also, through this figure, it was observed that when the compressor speed increases, there is a slight increase in the mass of air charge flowing.

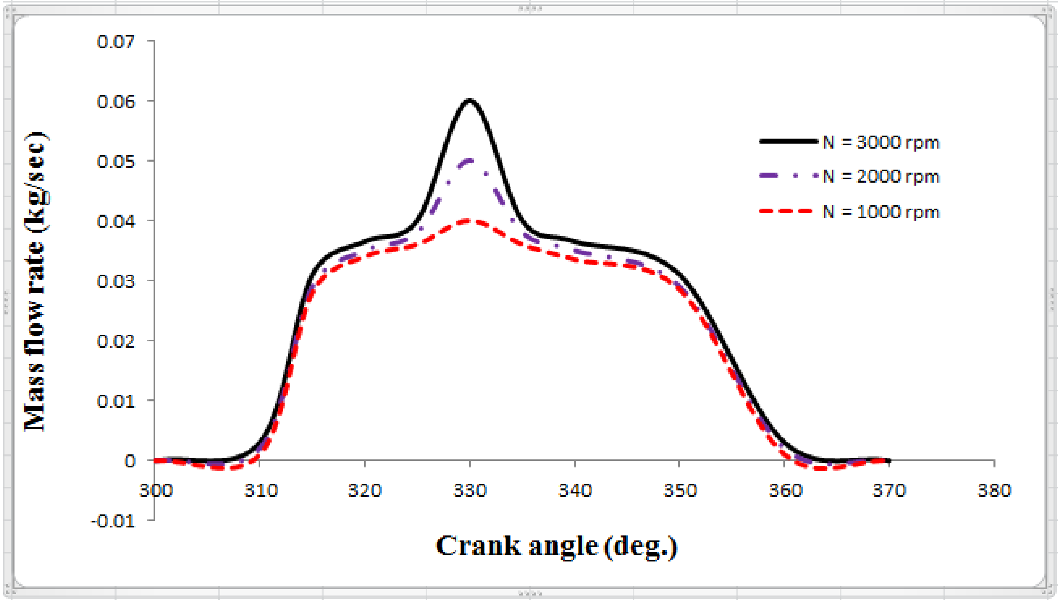


Fig. 10. The mass of air charge flowing into the first compression space at different speeds versus crank angle

Figure 11-13 elucidate the effect of the valve displacement (Lift = 1.4 mm), length and diameter of the suction pipe and cylinder on the volumetric efficiency at L/S = 6.1, 4.5 and 3, respectively versus the compressor speed. From these figures, it was observed that at the slow speed of the compressor, the volumetric efficiency increases, this is due to that when the speed is slow, the period of the air entering to the compression space is long, leading to an increase in the amount of air charge flowing into the compression space and thus increasing the volumetric efficiency.

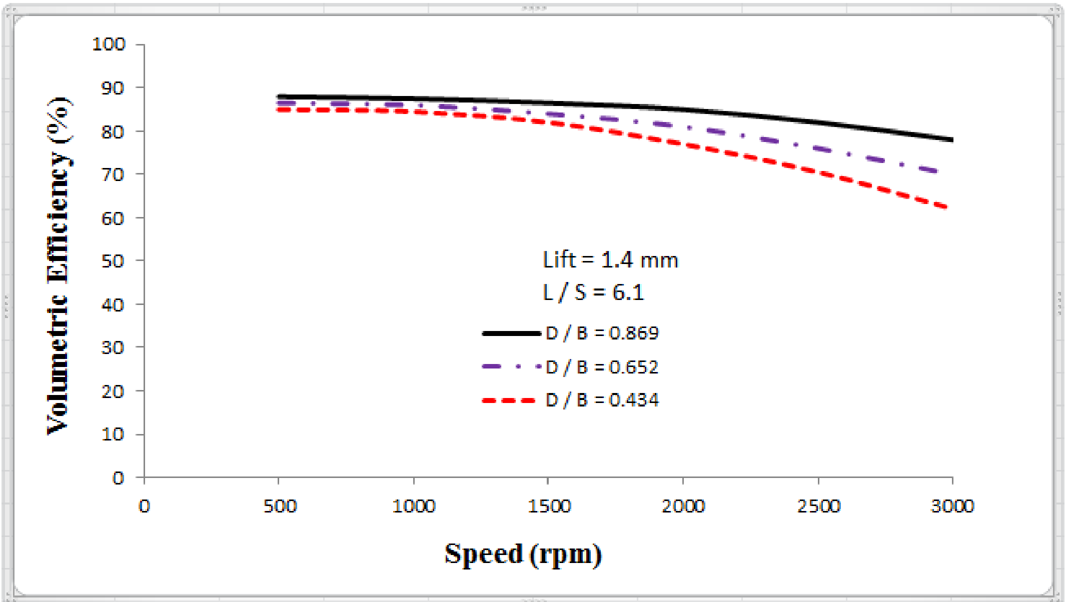


Fig. 11. The volumetric efficiency at a difference of the entry pipe diameter and length (Lift = 1.4 mm and L/S = 6.1) versus speed

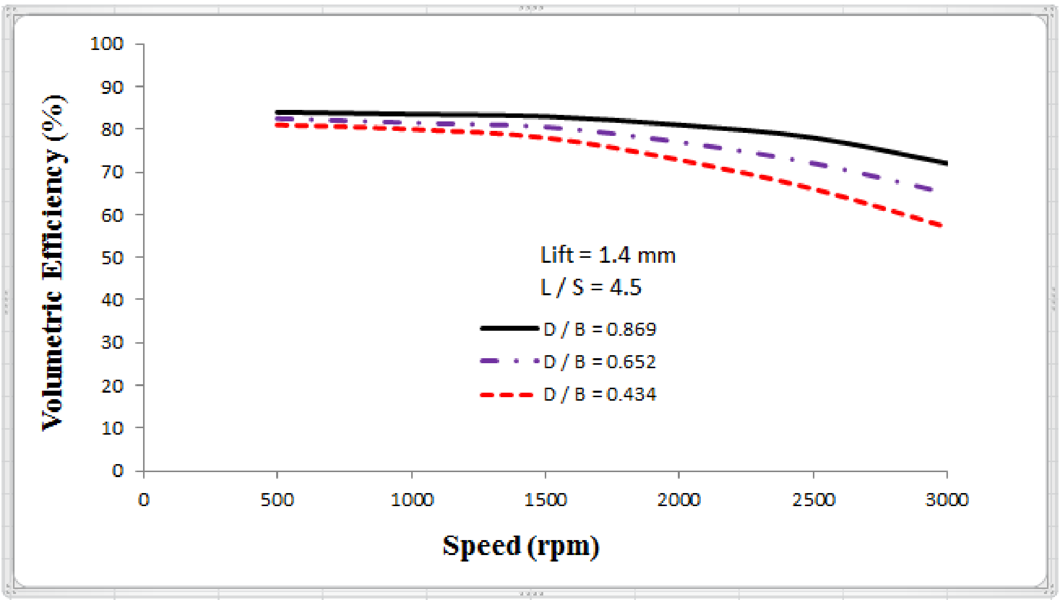


Fig. 12. The volumetric efficiency at a difference of the entry pipe diameter and length (Lift = 1.4 mm and L/S = 4.5) versus speed

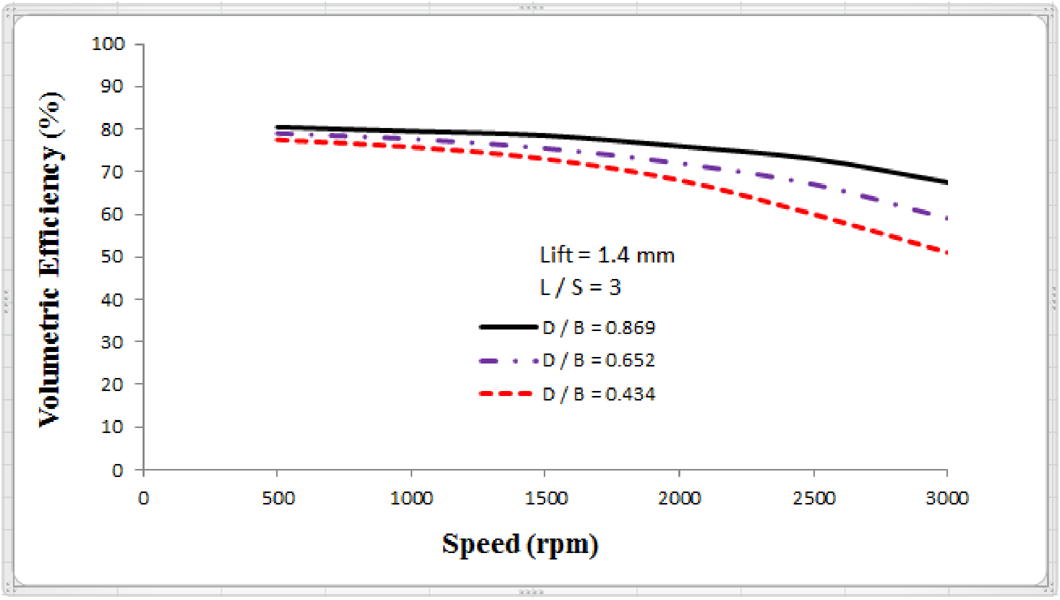


Fig. 13. The volumetric efficiency at a difference of the entry pipe diameter and length (Lift = 1.4 mm and L/S = 3) versus speed

Also, through these figures, it was found that when fixing both of the pipe diameter and valve displacement, the volumetric efficiency increases with the increase of entry pipe length. This is because that the long pipe gives an increase in the capacity of compressive waves generated at the valve orifice, which helps enter a larger amount from air charge and thus increases the volumetric efficiency. Also, when fixing both of the pipe length and valve displacement, it was obtained that the volumetric efficiency increases with the increase of pipe diameter, this is because when increasing the pipe diameter, the mass of the air charge flowing increases which leads to increase the volumetric efficiency.

It was found that the highest value of the volumetric efficiency at the Lift = 1.4 mm and (L/S = 6.1, 4.5 and 3) is (88, 83 and 80%), respectively.

Figure 14-16 exhibit the effect of the valve displacement (Lift = 1.6 mm), length and diameter of the suction pipe and cylinder on the volumetric efficiency at (L/S = 6.1, 4.5 and 3), respectively versus the compressor speed. Through these figures, when fixing both of the pipe diameter and valve displacement as well as when fixing both of the pipe length and valve displacement, it was found that the volumetric efficiency at (Lift = 1.6 mm and L/S = 6.1, 4.5 and 3) has the same characteristics and behavior as in the case of volumetric efficiency at (Lift = 1.4 mm and L/S = 6.1, 4.5 and 3) and for the same reasons that have been mentioned for Figure 11-13. But, increasing the valve displacement leads to increase the volumetric efficiency, this is because of the increasing in a flow area for the amount of air mass.

It was obtained that the highest value of the volumetric efficiency at the (Lift = 1.6 mm and L/S = 6.1, 4.5 and 3) is (92, 87 and 83%), respectively.

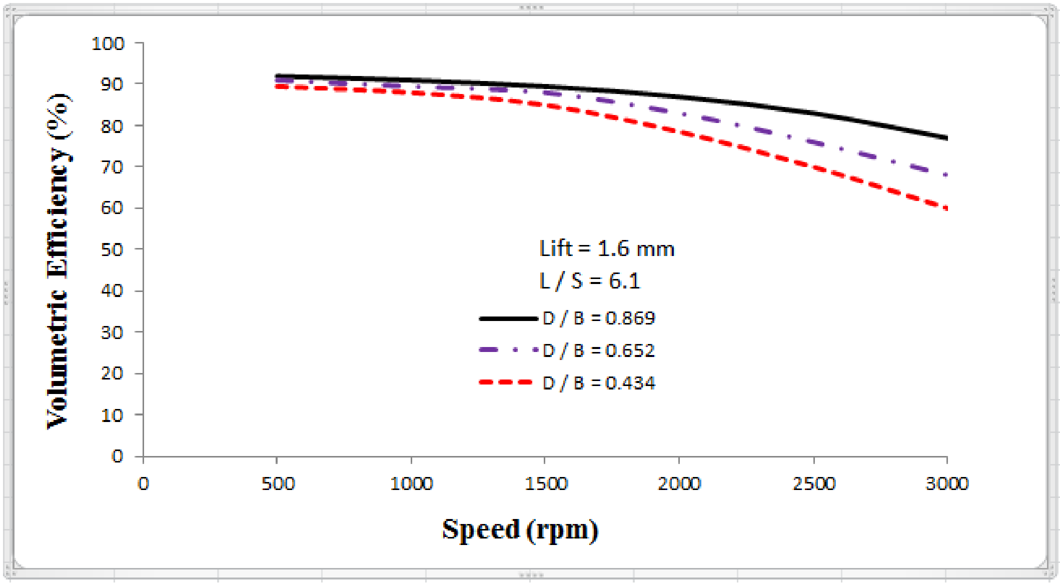


Fig. 14. The volumetric efficiency at a difference of the entry pipe diameter and length (Lift = 1.6 mm and L/S = 6.1) versus speed

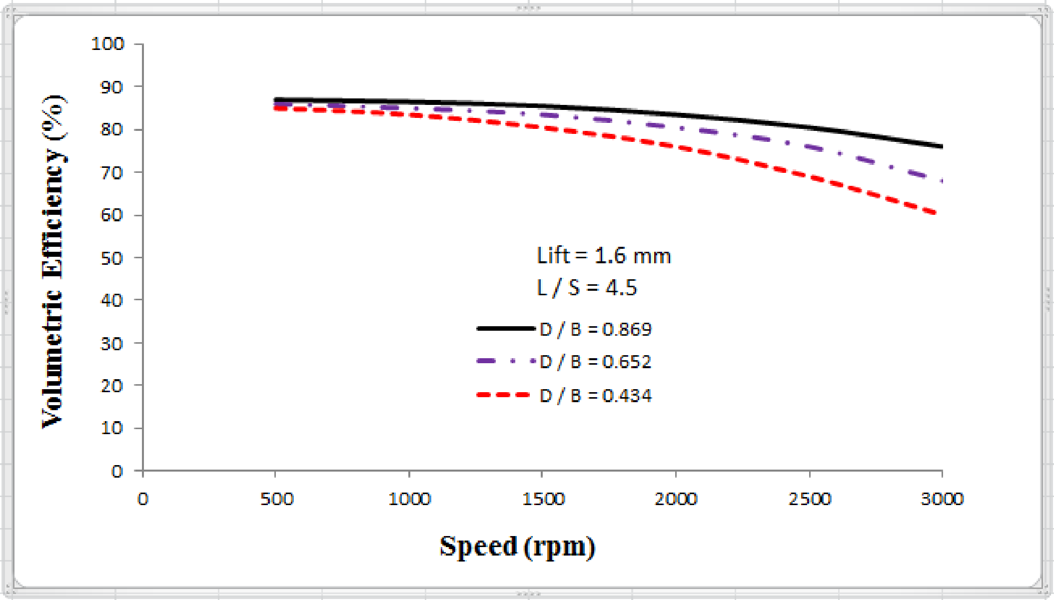


Fig. 15. The volumetric efficiency at a difference of the entry pipe diameter and length (Lift = 1.6 mm and L/S = 4.5) versus speed

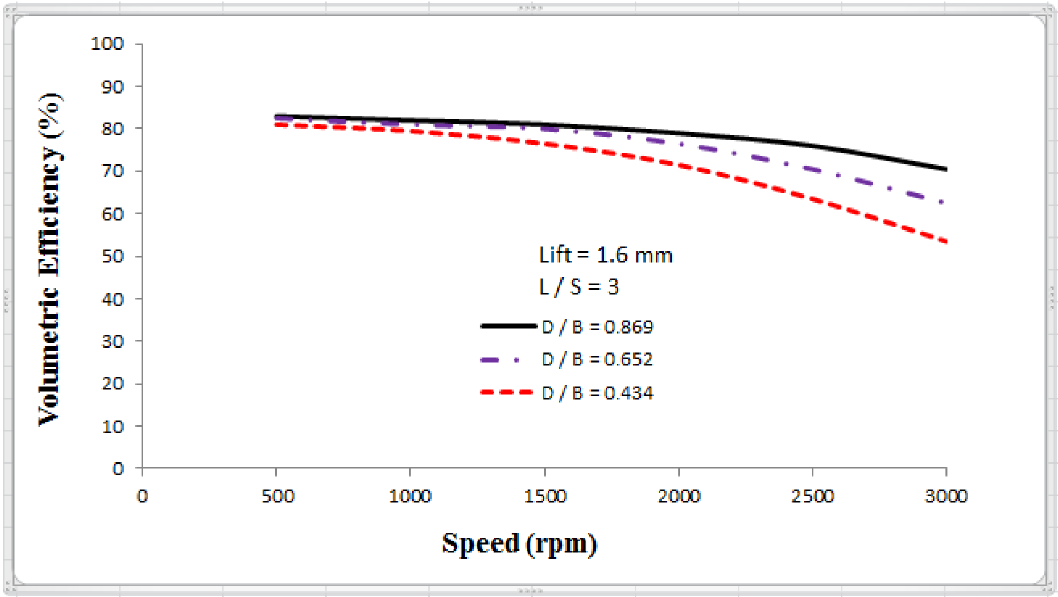


Fig. 16. The volumetric efficiency at a difference of the entry pipe diameter and length (Lift = 1.6 mm and L/S = 3) versus speed

5. Conclusions

Through the points obtained, the results can be extracted in the following points.

i. Increasing the speed of reciprocating air compressor leads to increase the piston speed inside the cylinder and thus the pressure and temperature increase.
ii. Increasing the mass of air charge flowing into the cylinder increases the pressure change at the end of a suction pipe.
iii. At the slow speed of the compressor, the period of the air entering to the compression space is long, which leads to increase the volumetric efficiency.
iv. Increasing the length and diameter of the suction pipe has an effect on the capacity of compressive waves and final pressure generated in the compression space, and this has a large effect on the increase of the volumetric efficiency.
v. Increasing the valve displacement leads to increase the flow area for the amount of air mass and thus increases the volumetric efficiency.

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