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Numerical Analysis of Flow Structure in Reciprocating Compressor

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ABSTRACT

In line with current rapidly developing technology, mechanization in industry is increasing. In this case, all components that form a machine are becoming important. Compressor plays an important role in operations of numerous machineries. They are used for increasing pressure by compressing gas. Compressors have various types and especially in industries such as automotive, textile, and manufacturing, reciprocating compressors are frequently used. Reciprocating compressors consist of cylinder, piston, and crank and connecting rod mechanism. These components work by narrowing the volume inside the cylinder and increasing gas pressure inside. In this study, flow structure inside a reciprocating compressor was numerically analysed. For this purpose, models were created for 2 different crank angles of a reciprocating compressor. Simulations were made for different pipe angles of inlet pipe and outlet pipe. Valve openings were kept constant at 6 mm. Input and output angles of inlet and outlet pipes to cylinder was changed as 0°, 30°, 45°, 60° and 90°. As a result, it was determined that as the angle of inlet and outlet pipes changed, general flow structure inside the cylinder changed as well. Especially, there were differences in velocity vectors plotted for input and output of the flow from valve to cylinder.

***Keywords*:** Reciprocating compressor, inlet input angle, computational fluid dynamics, CFD

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1. INTRODUCTION

Compressors are important in developing industrial environment and current technology. Compressors are often used in sectors such as energy plants, industry, and automotive. Compressors have various types. This variety depends on compressor power, flow rate, and working principle. In literature, there are numerous studies regarding compressors. However, there are few studies regarding reciprocating compressor in the literature. Some of those limited studies are presented below.

Gord and Khoshnazar [1] analysed zero-dimension numerical analyses method based on crank angle to evaluate faulty valve piston natural gas compressor. Authors determined that leakage in discharge valve decreased mass flow compared to healthy valve and caused increase in temperature. Yusha et al. [2] experimentally investigated cooling efficiency of theoretical slow velocity compressor by forecasting piston air. For this purpose, trigger rates with different diameters such as L3, RJ were used and working parameters were determined for temperature conditions. Xu and Hrnjak [3] evaluated a new method to measure oil retaining and circulation rate (OCR) inside a compressor outlet pipe. Authors observed that as oil mas increased, flow velocity increased as well. When oil drops travelled with higher velocity, authors showed that most of the oil was on film layer as a mass.

Pont et al. [4] evaluated reactions caused by piston crank movement in compression room of reciprocating compressor mechanism with different A13SRATC crank angles. Authors conducted complete numerical dynamic analysis and evaluated power losses. Authors determined that oscillations of angular velocity caused significant changes in flow behaviour of compressor. Zhao et al. [5] attempted to measure sudden temperature and pressure changes inside a compressor with thermistor under the scope of thermodynamic process. For this purpose, authors places two thin wired thermocouples with 18 and 35 millimetre diameter to piston working region. For each cases, maximum temperature difference was measured as 9.7^{o}C. Dutra and Deschamps [6] numerically simulated hermetic reciprocating compressor including electric motor which are necessary to design high-efficient compressor. With control volume formulation, fundamental energy balances were applied to create a thermal model. Authors conducted analysis for different working principles and parametrically determined dependence of engine temperature to input voltage.

Duprez et al. [7] created thermodynamic models for two type compressors used in home-type heat pumps. Authors calculated cooling fluid amount and power production and conducted analysis to determine optimum compressor parameters by using different cooling fluids. Busarov et al. [8] analysed working temperature, working pressure, mechanical, and thermal effects of a reciprocating compressor.

2. DEFINITION OF PROBLEM AND MATHEMATICAL MODEL

In this study, flow structure inside the cylinder of industrial type compressor was analysed for suction and discharge states. For this purpose, geometric dimensions and working principles of a real reciprocating compressor used commercially was used as reference. Catalogue data of this compressor was given in Table 1. Based on data presented in Table 1, general geometric model of reciprocating compressor was created. This compressor was directly coupling and had a crank shaft with 1500rpm. By directly connecting motor shaft to compressor shaft using special coupling connecting tool, motion is translated to compressor. Geometric dimensions and working data given below were obtained from compressor with direct measurements during operation.

In this compressor, air was used as fluid. This air was sent from A_c: $1.962 \times 10^{-3}\ m^2$ channel with constant cross-section, and with V_a: $6.369\ m/s$ velocity. Then, the air passed through 8 separate cylindrical opening A_{c8}: $3.14 \times 10^{-4}\ m^2$ area and h_8: 0,01m depth. Later, the air passed through clap with A_v: $1.256\ \times 10^{-3}\ m^2$ area, and lastly, the air arrived in a cylinder with cylinder diameter D_c: 0.09 m , and stroke h_c: 0.08 m.

Table 1. Data of used compressor.

F.A.D [m^3/min - CFM]	Pressure [bar]	Engine [Kw/hp]	Air Tank [L]	Dimensions [mm]
1.5 - 52.5	7.5	7.5 - 10	500	740 × 1970 × 1440

For different geometries of compressor air input and output pipes, simulations were made with Computational Fluid Dynamics (CFD) and flow characteristics were evaluated by obtaining velocity and pressure distribution of the problem. For this purpose, angle of input and output pipes where fluid entered the piston and cylinder axis were changed as 0°, 30°, 45°, 60°, 90° and 5 different geometric models were determined. During analysis, valve opening was kept constant at 6 mm and repeated for 5 different models.

2. 1. Differential Equation of Problem

At steady-state, for three dimensional and turbulence flow, continuity, momentum, and energy equations were given blow. Since the flow had turbulence, k-ε turbulence model was preferred for turbulence modelling.

Continuity Equation

$$\frac{d}{dx_i}(u_i) = 0 \tag{1}$$

Momentum Equations

$$\rho\frac{d}{dx_j}(u_i u_j) = -\frac{dP}{dx_i} + \frac{d}{dx_j}\left[\mu\left(\frac{du_i}{dx_j} + \frac{du_j}{dx_i} - \frac{2}{3}\delta_{ij}\frac{du_i}{dx_i}\right)\right] + \rho\frac{d}{dx_j} \tag{2}$$

Energy Equation

$$\frac{d}{dx_i}(\rho u_i T) = \frac{d}{dx_j}\left(\frac{k_{eff}}{C_p}\frac{dT}{dx_j}\right) \tag{3}$$

Turbulence Kinetic Energy Equation

$$\frac{d(\rho k u_j)}{dx_j} = \frac{d}{dx_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{dk}{dx_j}\right] + G_k - \rho\varepsilon \tag{4}$$

Turbulence Energy Dissipation Equation

$$\frac{d(\rho \varepsilon u_j)}{dx_j} = \frac{d}{dx_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{d\varepsilon}{dx_j}\right] + \rho C_1 S_\varepsilon - \rho C_2 \frac{\varepsilon^2}{k+\sqrt{\nu\varepsilon}} \tag{5}$$

Reynolds stress term and equations representing turbulence viscosity in above equations are as follows. Here, "μ_t" represents turbulence viscosity, "k" represents turbulence kinetic energy, and δij represents Kronecker delta.

Constants used in turbulence model were used as $C_{1\varepsilon}$ = 1,44, $C_{2\varepsilon}$ = 1.92 and C_μ = 0.09, and turbulence kinetic energy Prandtl number σ_k = 1.0 and turbulence energy dissipation Prandtl number were used as σ_ε = 1.3 [9].

2. 2. Boundary Conditions

Boundary conditions were needed to solve differential equations of the problem. For suction and discharge states of this compressor, two different states were considered and solved separately. Therefore, in suction state, since there was air input from suction channel, "inlet" velocity input was defined. In discharge state, since there was air output in output channel, "pressure outlet" was accepted as boundary condition in this region. On solid surface where the fluid contacted, no-slip condition was accepted and "wall" was applied as boundary condition. Technical properties of the air were presented in Table 2.

Table 2. Values of used fluid

Pressure [Pa]	Density [kg / m^3]	Temperature [K]	Air Input Velocity [m / sec]
101325	1.225	287.15	6.369

2. 3. Mesh Structure

As stated previously, in this study, 5 different models were created by changing the angle between cylinder axis and input and output pipe of reciprocating compressor. These models were analysed for both suction state and discharge state. Therefore, there were 10 different problem geometries in this study. Accordingly, mesh structure and component number in each geometry differs. For all these geometries, separate optimum mesh structure was determined, results were made independent of mesh structure, and simulations were conducted. To illustrate general form of mesh structure, as an example, mesh structure with 30^o angle of input pipe was presented in Figure 1.

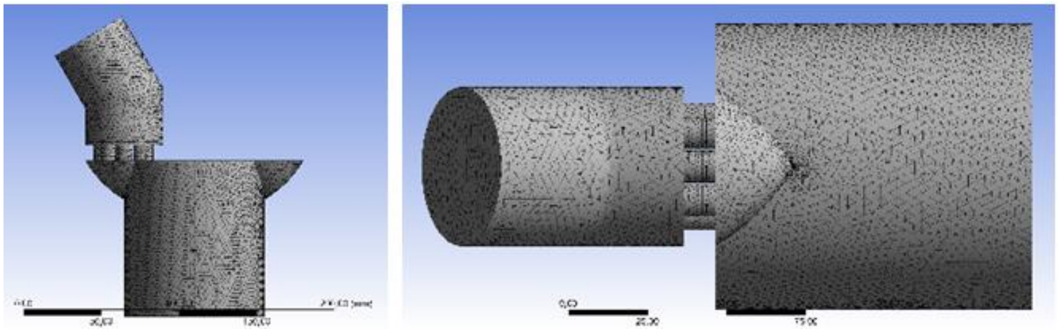
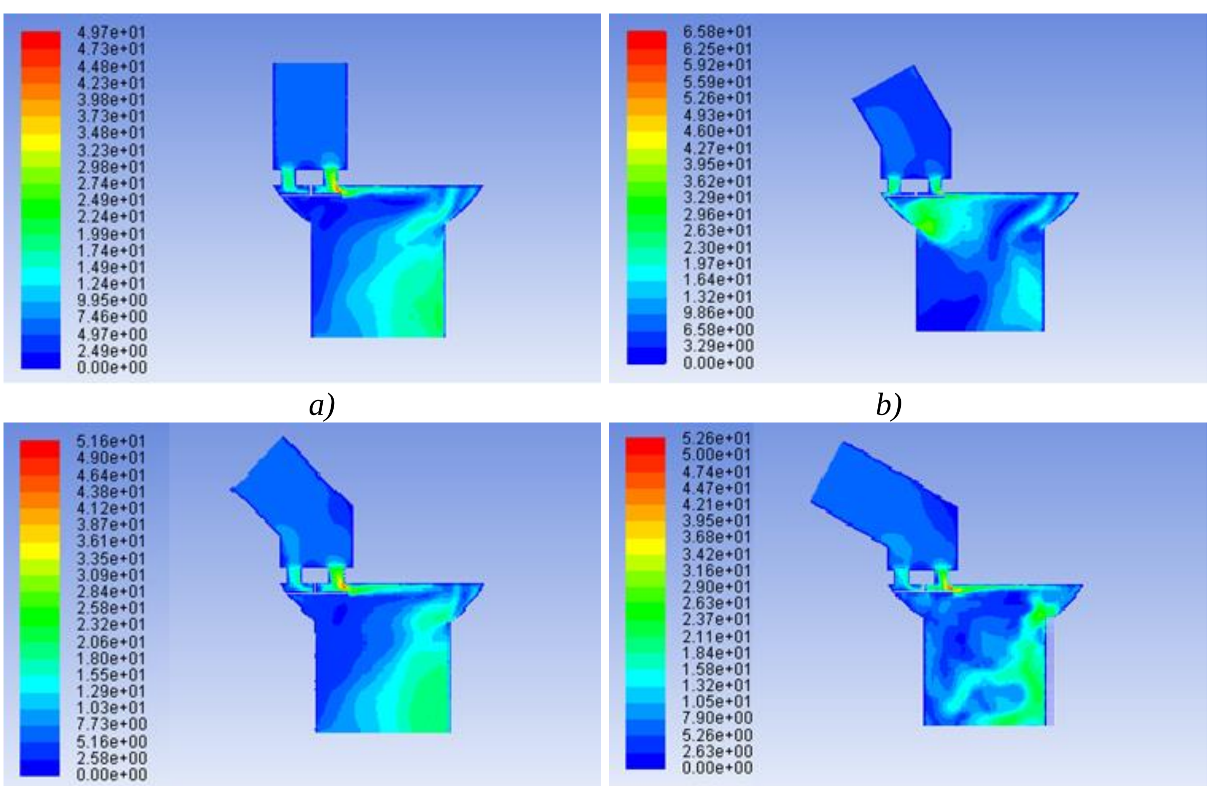


Figure 1. Mesh structure of problem

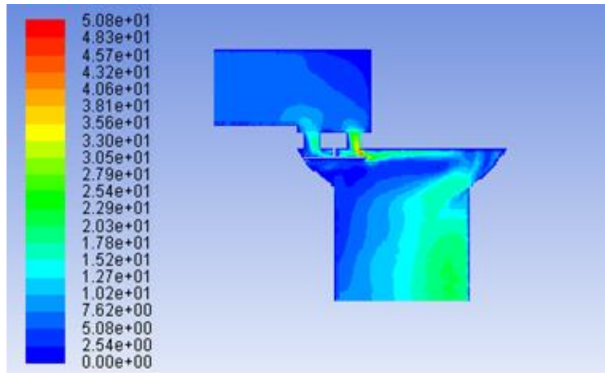
3. FINDINGS AND DISCUSSION

In this study, flow structure inside the cylinder of industrial type compressor was analysed for suction and discharge states. For this purpose, geometric dimensions and working principles of a real reciprocating compressor used commercially was used as reference. Suction and discharge processes of this compressor were analysed separately. For each process, angle of input and output pipes where fluid entered the piston and cylinder axis were changed as 0°, 30°, 45°, 60°, 90° and 5 different geometric models were determined. These models were numerically simulated with 6 mm valve openings. Some of the results obtained from analysis were presented below.



a) *b)*

c) *d)*

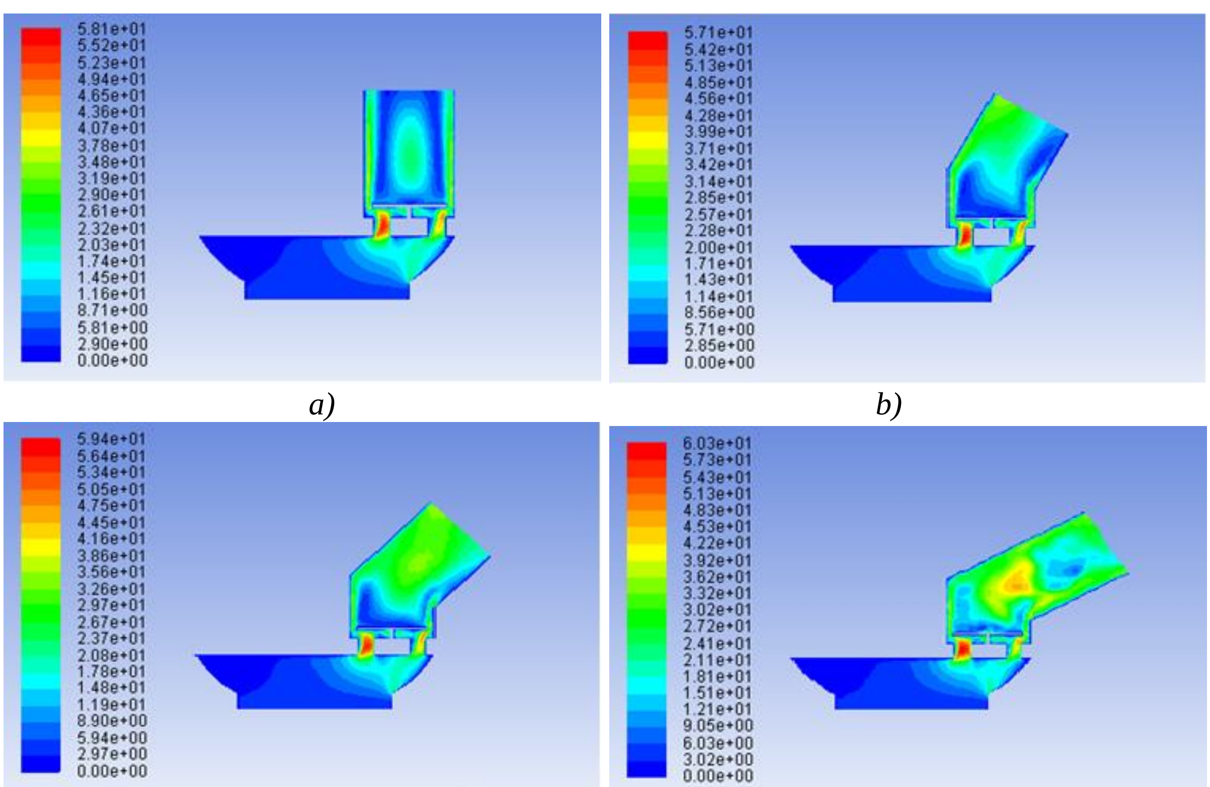


e)

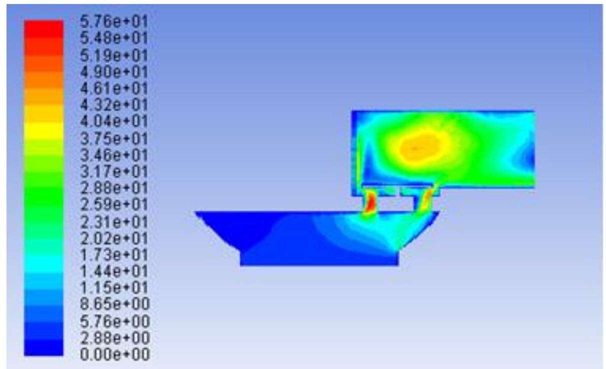
Figure 2. Velocity counter distribution for different input pipe angles for suction process a) 0°, b) 30°, c) 45°, d) 60°, e) 90°

Velocity counter distributions for different input and output pipe angles were presented in Figure 2 for suction process and in Figure 3 for discharge process. In suction process, only input pipe was included inside solution domain, and output pipe was neglected as output valve was closed. In discharge process, there was an opposite situation and input pipe was neglected as input valve was closed.

When Figure 2 was analysed, all models for suction processes showed similarities for general flow behaviour. Velocity of air increased while passing from valve and propagates inside the cylinder with a large volume. For input pipe 30^0, obtained velocity distribution after valve showed difference than other angles.

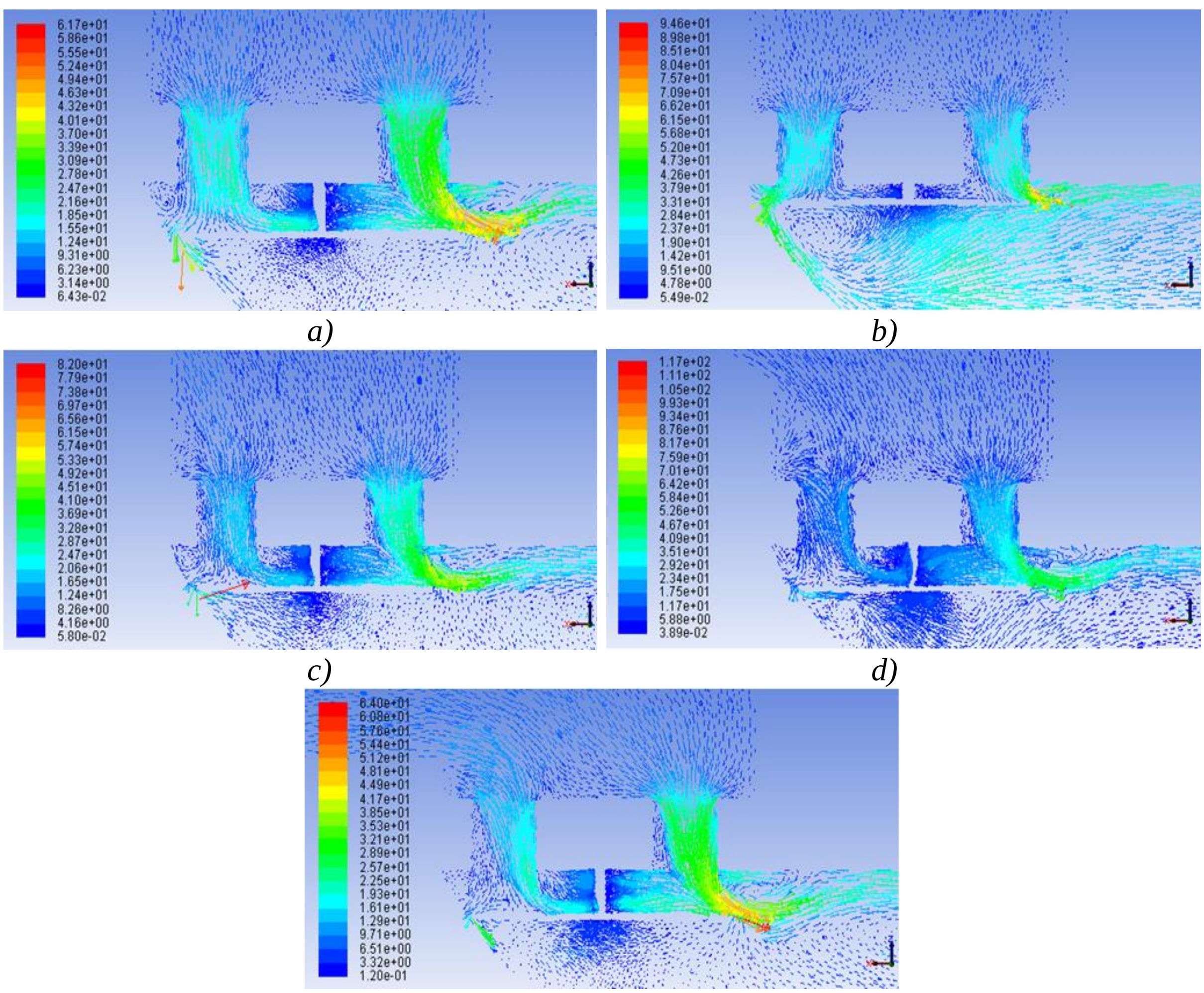


c) *d)*



e)

Figure 3. Velocity counter distribution for different input pipe angles for discharge process
a) 0°, b) 30°, c) 45°, d) 60°, e) 90°



a) *b)* *c)* *d)* e)

Figure 4. Velocity vector distribution for different input pipe angles for suction process
a) 0°, b) 30°, c) 45°, d) 60°, e) 90°

Figure 3 had similar results. For discharge process, in all models, general flow structure observed inside cylinder was similar. However, flow velocity distribution discharged from output valve showed differences due to different output pipe angles.

Velocity vector distributions for different input and output pipe angles were presented in Figure 4 for suction process, and in Figure 5 for discharge process. While these figures were plotted, regions with valves that showed intense velocity changes were emphasised. As seen from figures, in both suction and discharge state, change in input and output pipe angles affected direction and amplitude of velocity vectors around valve regions. When suction process was analysed, velocity vectors obtained for 30° input region showed difference than other states.

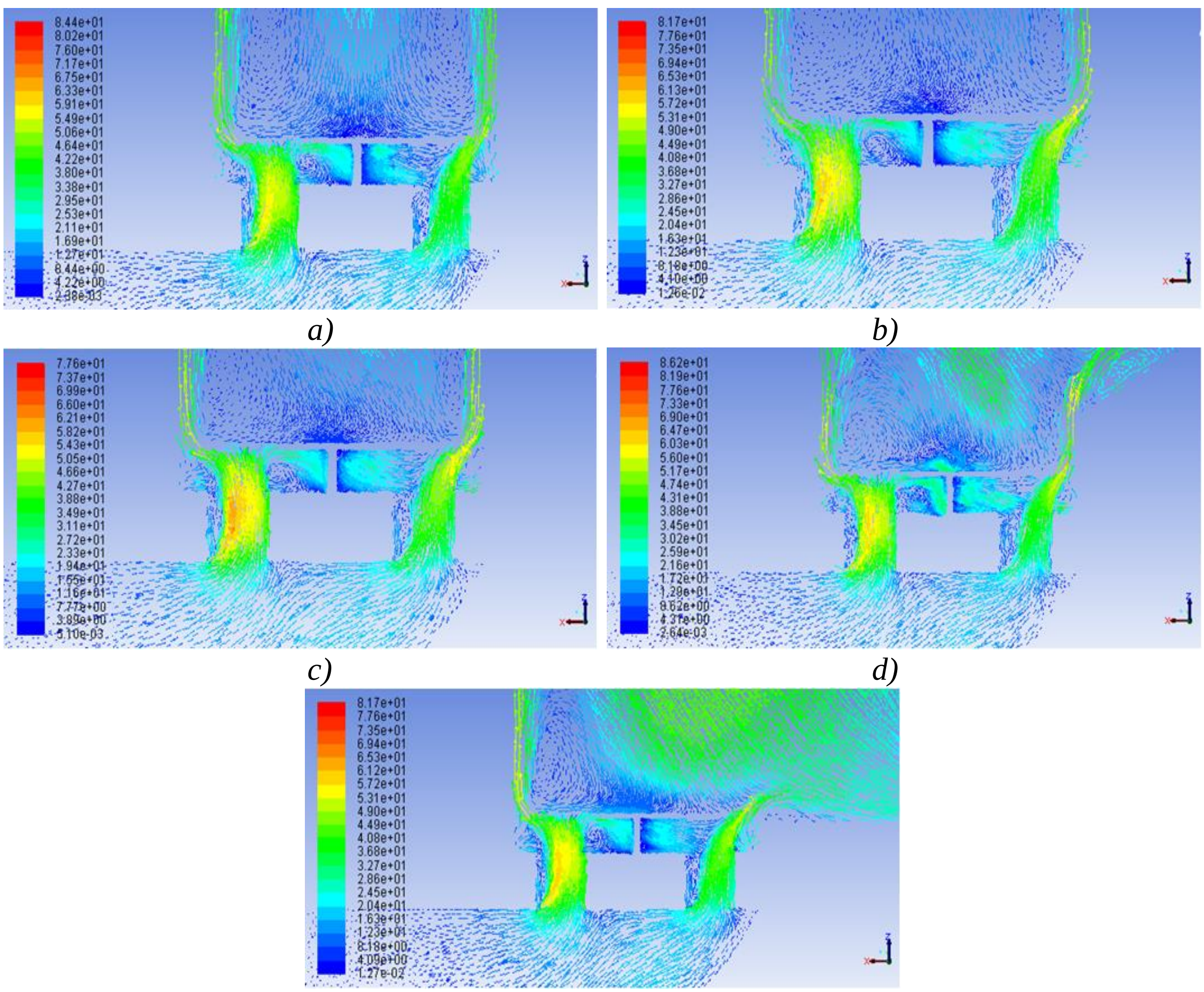


Figure 5. Velocity vector distribution for different input pipe angles for discharge process a) 0°, b) 30°, c) 45°, d) 60°, e) 90°

In Figure 6, for different input angles, average velocity change of air that passed from valve and entered into cylinder in suction process was showed. As seen from figure, minimum air input velocity was observed for the geometry with input angle of 0°. Input velocity increased with increased input angles. Although velocity slightly decreased for the geometry with input angle of 45°, at 60° input velocity was at maximum level. However, as input angle increased, input velocity decreased sharply, and decreased back to input velocity of geometry with 0°. In reciprocating compressor, filling suction air inside the cylinder in the shortest time possible enables more efficient working conditions. Therefore, it could be said that model with 60° input pipe-cylinder angle is more suitable for optimum working conditions.

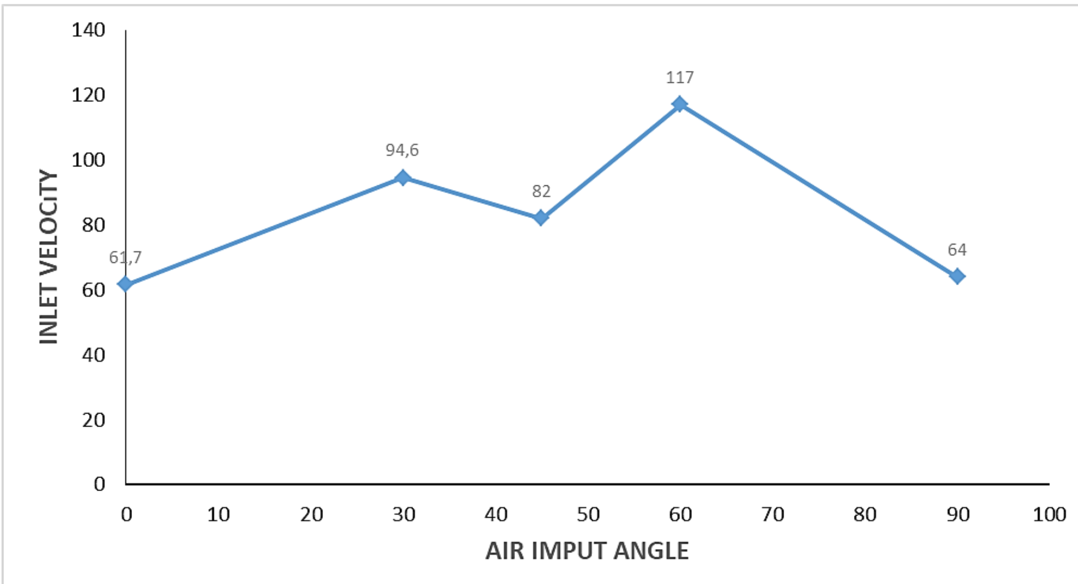


Figure 6. Cylinder air inlet velocity change for different input angles.

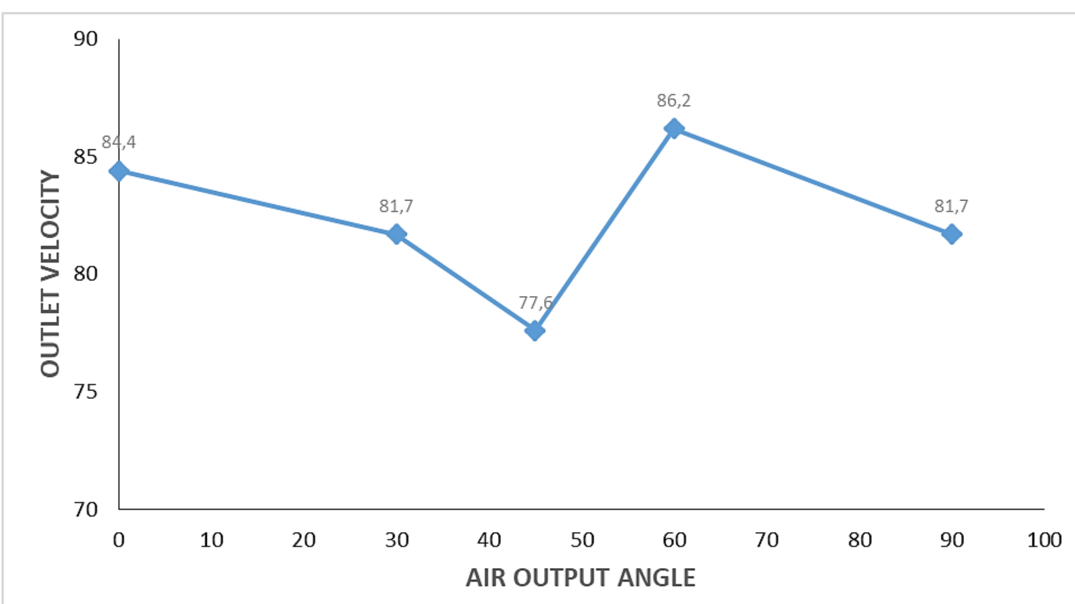


Figure 7. Cylinder air outlet velocity change for different output angles.

In Figure 7, for different output angles, average velocity change of air that passed from valve and discharged from cylinder in discharge process was showed. As seen from figure, minimum air output velocity was observed for output angle of 45°, and maximum air output velocity was observed for output angle of 60°. When output angles were 0° and 90°, average output velocities were similar. In reciprocating compressors, another factor that positively effect efficient working conditions is to discharge the air inside the cylinder in the shortest amount of time possible. Therefore, it could be said that model with 60° output pipe-cylinder angle is more suitable for optimum working conditions.

4. CONCLUSIONS

In this study, flow structure inside the cylinder reciprocating compressor was analysed for suction and discharge states. For this purpose, suction and discharge processes were considered separately for constant 6 mm valve opening in reciprocating compressor. For each process, angle of input and output pipes where fluid entered the piston and cylinder axis were changed as 0°, 30°, 45°, 60°, 90° and 5 different geometric model were created, and numerical simulations were made. As a result of analysis, it was observed that change of the angle between cylinder axis and input and output pipes affected flow structure and flow characteristic inside solution volume. For both suction and discharge processes, in the model with angle of 60°, maximum suction and discharge velocities were observed. In a reciprocating compressor, it is needed for suction and discharge processes to be completed in a fast and constant way. When this condition is considered, it could be said that among five different models analysed in this study, geometry that satisfies optimum working conditions was the model geometry with input and output angles of 60°.

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