

Numerical Investigation of Flow Losses through Discharge Line of Household Type Refrigerator Compressors

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Abstract. The technological developments, competition and energy policies are forcing the refrigeration market to increase the efficiency of their products as never before. The component that determines the performance and efficiency of the refrigeration system is the compressor, which is the major energy consumer in a compression refrigeration system. One of the essential elements of the total compressor efficiency is the discharge line flow efficiency that is subjected to this study. In this paper, the effects of design parameters on flow losses at discharge line of a hermetic reciprocating compressor were investigated with numerical flow analysis. At this study, three conceptual designs were modelled based on the discharge line design parameters such as line diameter, resonator volumes and line length. The pressure drop regions are determined by CFD analyses and they compared against the base model. Analyses are carried out by using commercial CFD software. Furthermore, the obtained numerical results were compared to experimental data and presented a good agreement in terms of pressure drop and discharge line flow efficiency.

1. Introduction

In recent years, the growing global need for energy and running out of the natural energy sources, prompt human being to find alternative sources and use the existing ones efficiently. By this way, energy saving practices and precautions will be reduced the effects of carbon dioxide emissions and global warming, which are serious threats to the future of the world.

According to electricity distribution and consumption statistics, 2.836.637 GWh of electrical energy is consumed by the countries of EU-27, 2010. The residential customers consumed 29.71% of this total electricity energy consumption. Only the refrigerators and freezers consume about 14.5% of the residential electric energy consumption. In this regard, energy consumption of residential refrigeration system is said to be 122.201 GWh approximately [1]. Total energy consumption by residential and commercial sectors in US is 21.7% and 17.8% respectively. Refrigerators and freezers consume 7.2% of total residential energy use and they are responsible for 11.9% of the residential electricity consumption [2]. As well as all over the world, regulations on energy saving are becoming increasingly stricter every year and also they have imposed on compressor manufacturers to design more efficient and environmentally friendly compressors.



One of the most important parts of refrigeration system is a compressor which is the heart of refrigeration systems. Basically, compressors determine the performance and efficiency of the cooling system. This basic component of the cooling system consuming the electrical energy and converts it to energy stored in the form of pressure of the evaporated refrigerant. Although there are lots of different compressor types to achieving this goal but in the competitive environment as a result of globalization, the hermetic reciprocating compressor is the most commonly used type of compressors in household type refrigerators.

There are five fundamental requirements of a hermetic reciprocating compressor such as lifetime, acoustic noise level, compactness, efficiency and price. By the way, it has been known that compressor consumes the greater part of the electrical energy that input power of the refrigeration system. Even though all of the fundamental requirements are important for compressor manufacturers, but the compressor efficiency is getting more important because of the energy prices and environmental awareness [3].

Studies for improving the efficiency of compressors are grouped under three main headings: increasing the efficiency of electric motor, reducing the mechanical losses and also reducing the thermodynamic losses due to the irreversibilities in the suction, compression and discharge process. Several investigations have been reported concerning improving the performance of household reciprocating compressors. McGovern and Harte [4] presented a new approach in the field of compressor irreversibilities analysis and quantified the exergy destruction in particular regions. Birari et al. [5] investigated the existing compressor design with alternative refrigerant. With the CFD simulations, they checked for the heat balance across the compressor domain and suction and discharge path pressure drops. Ozdemir [6] examined discharge tube's and discharge muffler's heating effects to the compressor performance by using conceptual designs. This paper describes the use of CFD (Computational Fluid Dynamics) in the design and development of an efficient discharge line of reciprocating compressor. For this purpose, CFD simulations are carried out for numerically investigate effects of the structural changes in the discharge line of a hermetic reciprocating compressor to the flow losses that occur at that region.

Within the scope of this study, the effects of some of the structural changes to flow losses at the discharge line of a household type hermetic reciprocating compressor were investigated by numerical analysis. For this study, three different conceptual designs, which based on the discharge line design parameters such as orifice diameter, resonator volumes and line length, were modelled. The pressure drop regions are determined by CFD analyses and they compared against the base model. Analyses are carried out by using commercial CFD software. Furthermore, the obtained numerical results were compared to experimental results.

2. Numerical analyses of discharge line

In this section, modelling the discharge path of a hermetic reciprocating compressor, discretization of the generated model and boundary conditions are discussed.

2.1. Analysis

One of the major factors of total compressor efficiency is the discharge line flow efficiency. In order to determine the proper discharge line geometry for increase the performance of the compressor is intended. For this purpose, the effect of discharge lines which have different geometries on pressure drop due to the flow losses is examined with a commercial CFD software package ANSYS Fluent, which is based on the finite volume method. All the geometries in discharge line analysed that three dimensional and turbulent fluid flow pattern. Discharge process is assumed as steady state conditions and the thermal effects are neglected. Isobutane (R600a) is used as a refrigerant and in the simulations the fluid properties have been implemented as constant at mean temperature. At discharge line,

refrigerant gas flow is pulsating due to the opening and closing movements of the lead valves. In this study, refrigerant gas pulsation and also reed valve's movement are excluded.

2.2. Modeling

Four different exhaust lines are investigated and one of them is already being produced which can be seen in Figure 1.

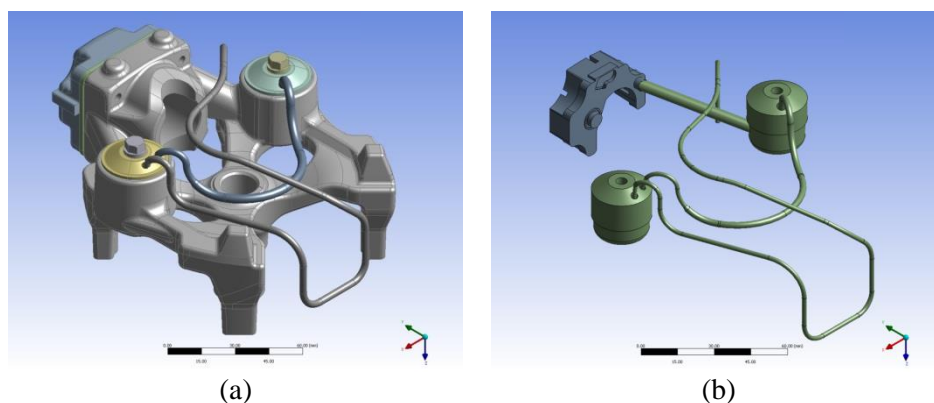


Figure 1. Structure of the discharge line: CAD assembly (a) and flow path (b).

The other models were generated from the original model. Producibility is significant during modelling the alternative discharge lines. Model 2 designed to see the effects of widening the connection bore. At model 3, connection bore connected the fluid to the other resonator. Thus, the resonator volume is extended and discharge line length is reduced. Model 4 has only one resonator and discharge line length is also reduced. These models are presented in Figure 2.

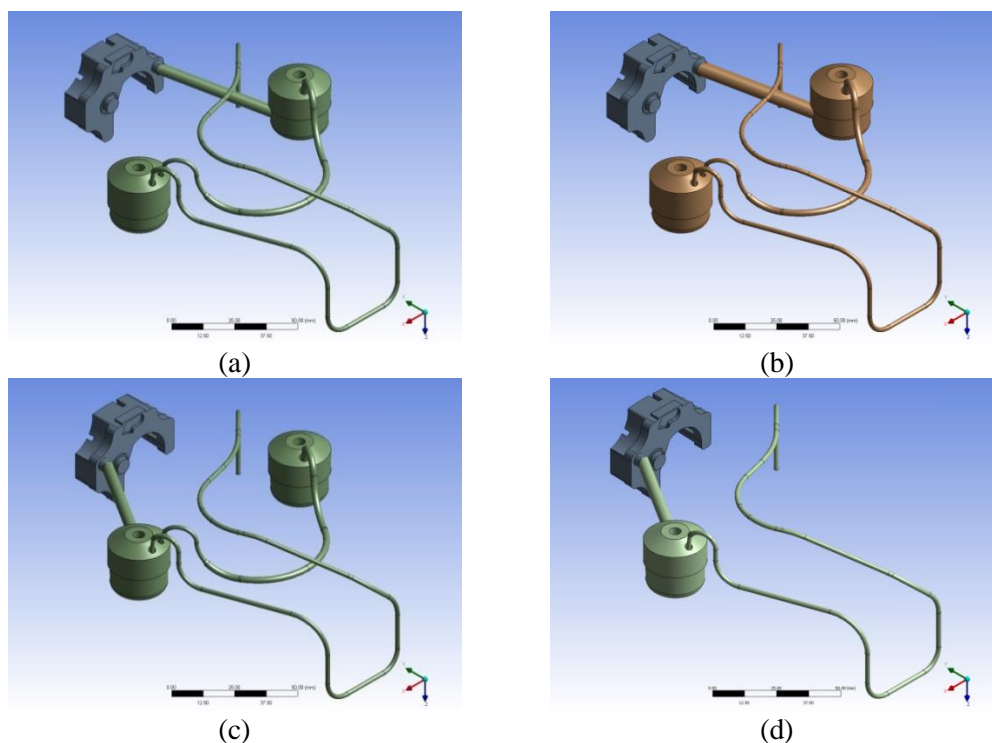


Figure 2. Discharge line designs: Model 1 (a), Model 2 (b), Model 3 (c) and Model 4 (d).

A comparison between discharge line properties is presented in Table 1 where the distance of exhaust port between connection bore, diameter of connection bore and number of resonator volumes are given.

Table 1. Alternative dimensions of discharge lines.

Property	Unit	Model 1	Model 2	Model 3	Model 4
Distance of Exhaust Port between Connection Bore	δ	a	a	0.37 a	0.37 a
Diameter of Connection Bore	\varnothing	b	1.25 b	b	b
Number of Resonator		2	2	2	1

2.3. Discretization and boundary conditions

At discretization stage, tetrahedral mesh is generated for all domains of discharge line by using the ANSYS Mesh Platform. Figure 3 shows the applied mesh form. The smallest and the biggest element sizes are $1.9715 \cdot 10^{-5}$ m and $2 \cdot 10^{-3}$ m respectively. The total number of elements is approximately 7 million, which is enough for mesh independency.

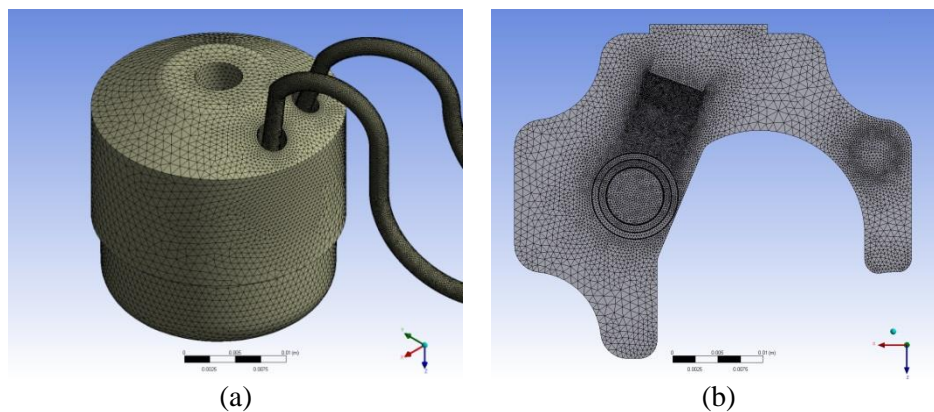


Figure 3. Image of applied mesh: resonator (a), cylinder head group (b).

For the simulations, ASHRAE standard test conditions were used (54.4 °C condensing temperature, - 23.3 °C evaporating temperature). Table 2 shows the refrigerant properties and boundary conditions used in the simulation.

Table 2. Properties and boundary conditions.

Refrigerant Properties (R600a)	Density (kg/m ³)	10.992
	Dynamic Viscosity (kg/m-s)	1.0629e-05
	Mass Flow Inlet (kg/s)	6.256e-04
Boundary Conditions	Pressure Outlet (Pa)	0
	Operating Pressure (Pa)	761300

The behaviour of the turbulent flow has been considered using the Realizable k-Epsilon model with Enhanced Wall treatment. The turbulence parameters, such as turbulence intensity, were calculated and specified at appropriate zones. SIMPLE algorithm was used for the velocity pressure coupling.

3. Results and discussion

The pressure drop which is the main flow parameters, at the discharge path of the compressor is considered as the comparable result. The pressure drop through the discharge line which is extending from valve plate exhaust port to the discharge tube are compared according to inlet pressure of the discharge line in Figure 4. Due to the output pressure is zero; inlet pressure means the lost pressure.

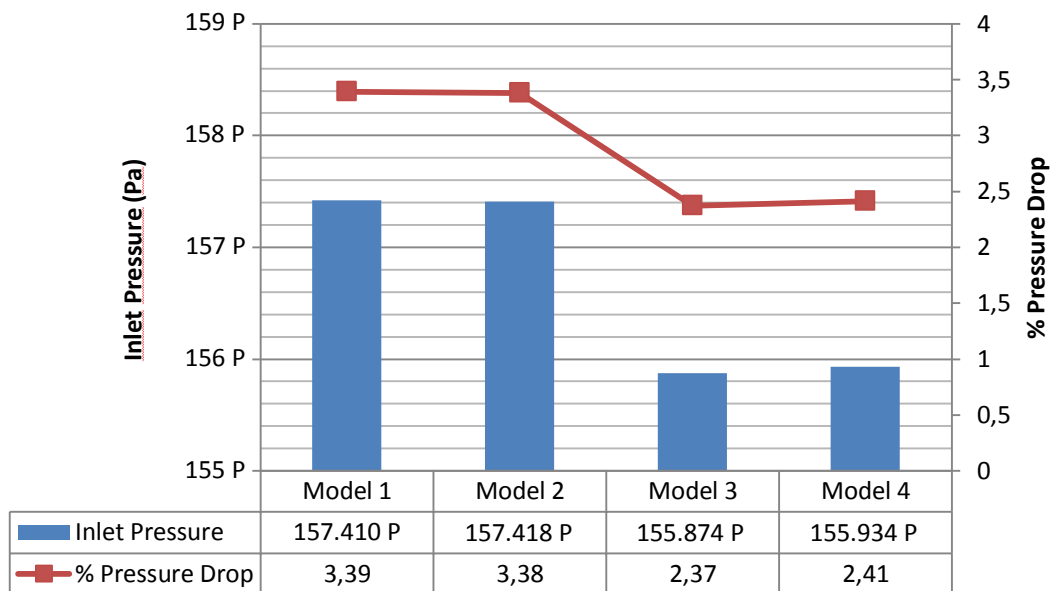


Figure 4. Inlet pressure and the pressure drop.

Figure 4 shows that the lost pressure of the Model 1 and 2 are greater than the others because of the flow across the single resonator and shortening the distance between the exhaust port and connection bore. Pressure drop of the both Model 1-2 and Model 3-4 are almost identical. Table 3 shows the comparison of the area weighted flow velocity at critical areas. It can be seen from the table, increasing the cross sectional area of the connection bore reduces the flow velocity about 0.19V m/s at this area. Aforesaid velocity reduction is reflected the outlet velocities.

Table 3. Area weighted flow velocities at critical areas.

Model	Inlet [m/s]	Connection Bore [m/s]	Outlet [m/s]
1	0.3433 V	0.5749 V	2.4957 V
2	0.3433 V	0.3815 V	2.3981 V
3	0.3433 V	0.5698 V	2.4745 V
4	0.3433 V	0.5381 V	2.4060 V

Figure 5 illustrates the streamlines of the discharge flow path. It is observed that there isn't any flow between the resonators at Model 3. Second resonator of the Model 3 pretends like it's excluded from the flow path. It is also observed that the streamlines between the exhaust port and connection bore is shortened at Model 3 and 4 with respect to Model 1 and 2.

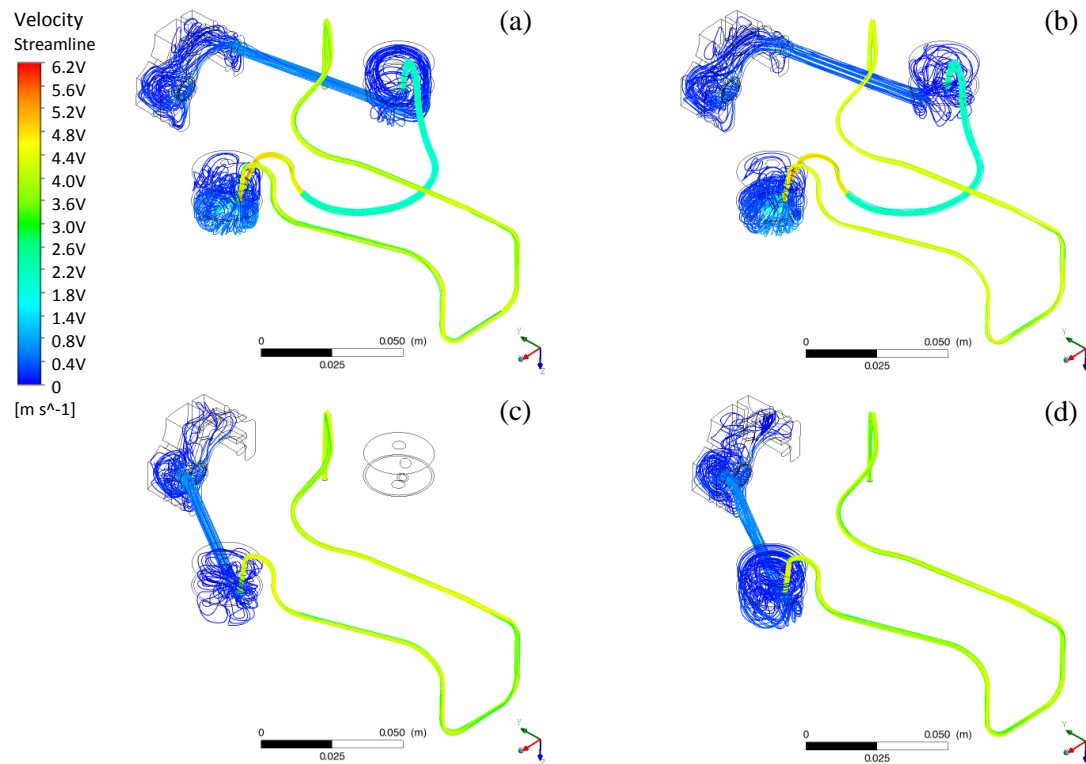


Figure 5. Streamlines: Model 1 (a), Model 2 (b), Model 3 (c) and Model 4 (d)..

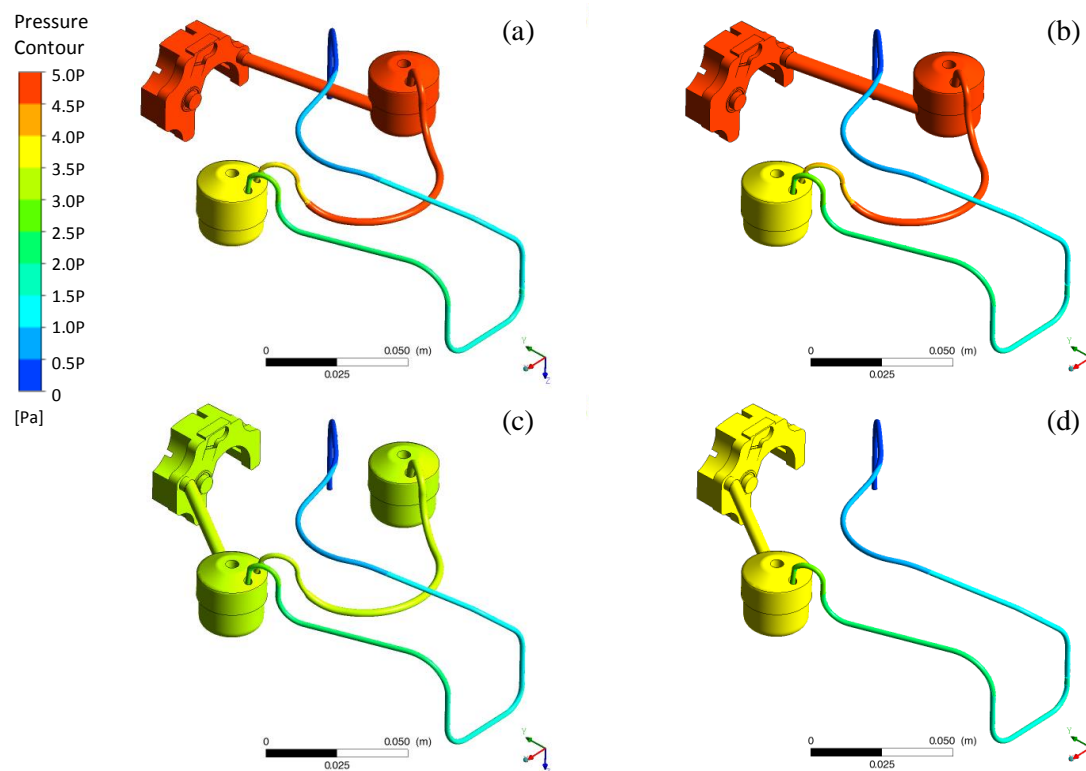


Figure 6. Pressure distribution plot: Model 1 (a), Model 2 (b), Model 3 (c) and Model 4 (d)..

Figure 6 shows the pressure distribution contour plot. It can be seen that there isn't a significant pressure drop between discharge port and first discharge muffler as against the shockloop area. For all models, the maximum pressure drop occurs in the shockloop flow path. Model 3 and 4 doesn't include tube between the resonator volumes on the direct flow path. For this reason the pressure drop that occurs in this area is prevented with these designs. It is also observed that the Model 3 has greater pressure drop against Model 4. It can be concluded that enlargement of the resonator volume with connection tube increased the pressure drop.

Only velocity streamlines or pressure distribution plots don't make sense for complex geometries like compressor flow path. For expanding on the numerical results, both pressure and velocity on centreline are illustrated in Figure 7 for Model 1.

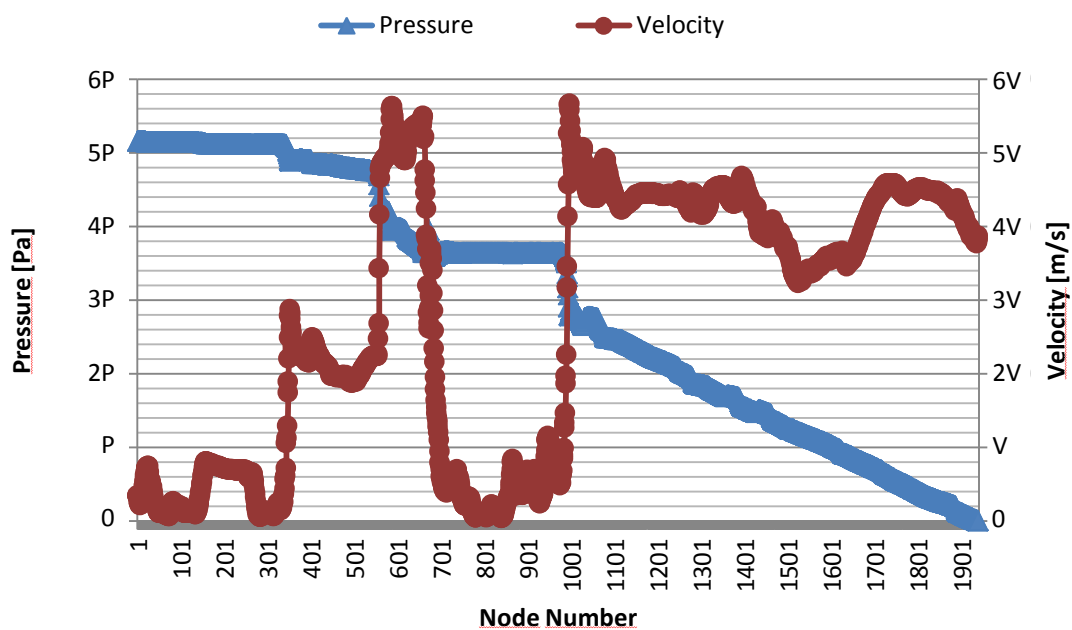


Figure 7. Pressure and velocity at centreline for Model 1.

In this figure, pressure drop areas can be recognised with the node numbers through the whole flow path. For this model, pressure value doesn't vary significant amount up to 319th node. In this section, refrigerant passes from first resonator to the connection tube. For this throttling effect velocity is increased up to 3V. Entrance of the connection bore at 158th node, velocity is increased up to 0.4 V. At the second throttling area, which is on the connection tube, is increased the velocity up to 5.6 V at 589th node. At the second resonator velocity is reduced to approximately V between the 661st and 994th nodes. After this region refrigerant passes to the shockloop and pressure reduces linearly with increased velocity.

Figure 8 shows the pressure and velocity at centreline for Model 2. It can be found that the characteristics of the pressure and velocity are the same with Model 1. For this model, it can be observed that the velocity is less than the Model 1 through the connection bore.

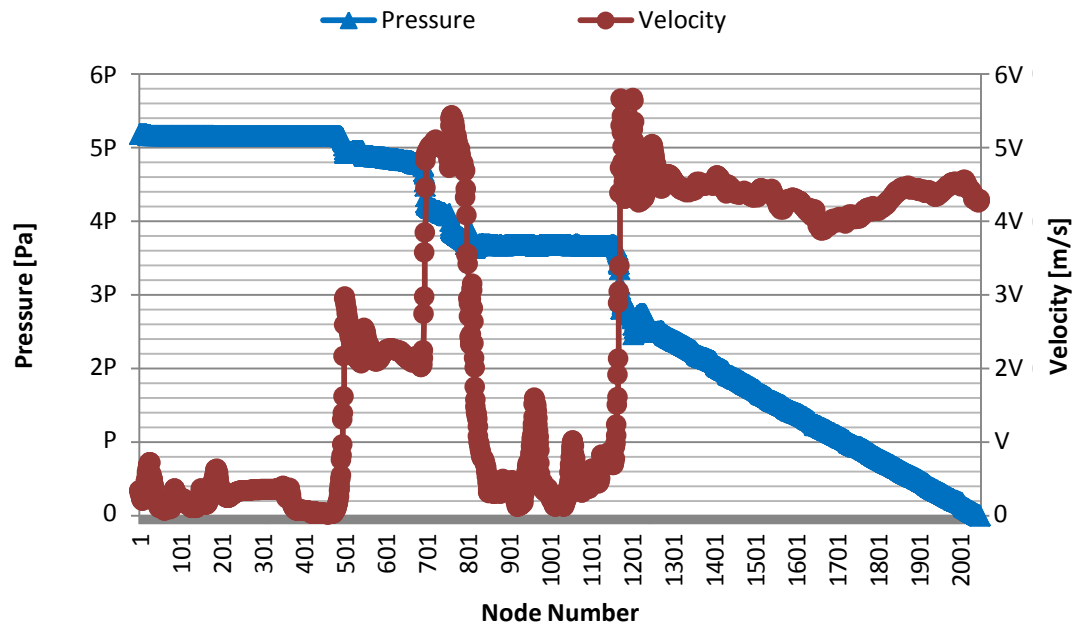


Figure 8. Pressure and velocity at centreline for Model 2.

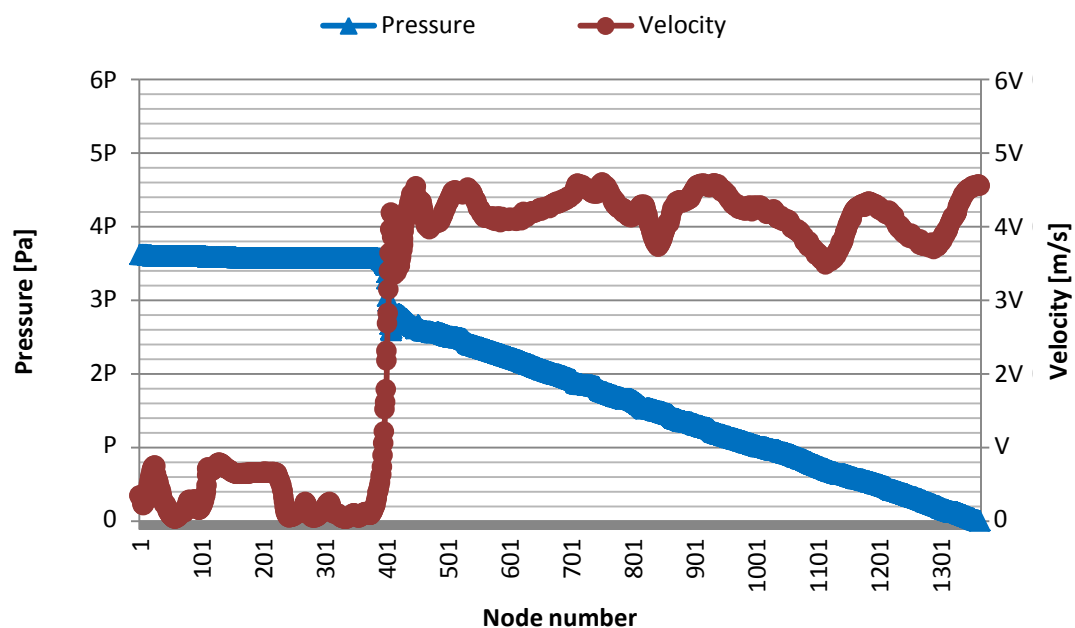


Figure 9. Pressure and velocity at centreline for Model 3.

In Figure 9 and 10 it can be observed that the flow path is simplified at these models. The drastic pressure drop is only occurred at the entrance of the shockloop. There isn't steady flow occurred between the resonators. For this reason it could not be obtained the effect of second resonator.

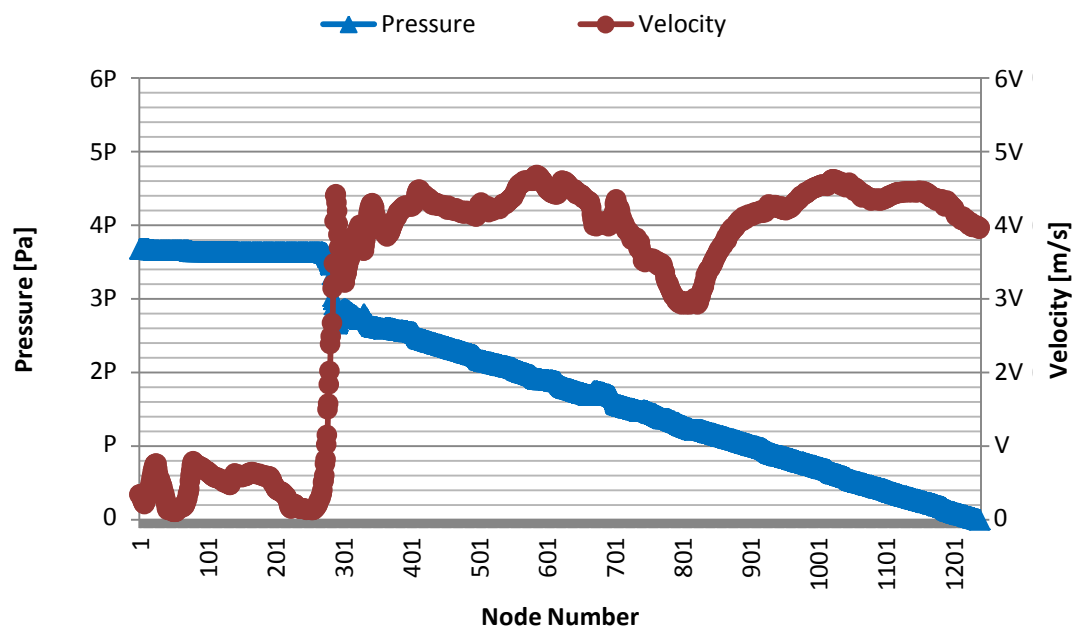


Figure 10. Pressure and velocity at centreline for Model 4.

In a comparison stage based on experimental study [7], it can be concluded that, the obtained numerical results have reasonable agreement in terms of pressure drop and discharge line flow efficiency. In Figure 11 comparisons of numerical and experimental results are presented. To improve the accuracy of the numerical results, pressure pulsations and thermal effect have to be taken into account.

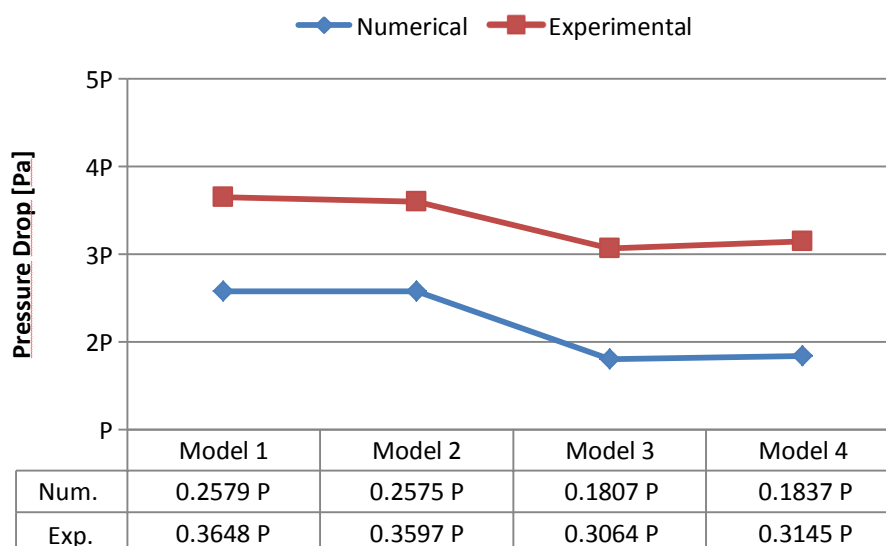


Figure 11. Comparison of numerical and experimental results.

4. Conclusion

In this study, numerical investigation of flow losses through discharge line and the behaviour of the pressure pulsation at the exhaust line of a reciprocating hermetic compressor were presented and the following conclusions were obtained:

- Reduction of the discharge flow path length reduces the pressure drop.
- The maximum pressure drop occurs in the shockloop flow path.
- Enlargement of the resonator volume with connection tube increased the pressure drop.
- Decreasing the discharge passage length might be affected the pressure pulsation value negatively. For this reason, decreasing the discharge passage length or resonator volume must also be reviewed in terms of acoustics.

Based on the results of this study it will be extended that:

- Investigation will be extended with the temperature and heat flux measurements.
- Pressure pulsation effect will be taken into account and correlated with acoustic measurements.

5. References

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