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Ismail Yesilaydin
Arcelik A.S. Refrigerator & Compressor Plants, Turkey, ismail.yesilaydin@arcelik.com

L. Berrin Erbay
Eskisehir Osmangazi University, Turkey, lberbay@ogu.edu.tr

A Study on a Numerical Modelling of Discharge Line Flow Analysis of a Household Type Refrigerator Compressor

Ismail YESILAYDIN[1]*, L. Berrin ERBAY[2]

[1]Arcelik A.S., Compressor R&D Center
Compressor Plant, Organize Sanayi Bolgesi, 1. Cadde, Eskisehir, Turkey
Phone: +90 222 213 46 55, Fax: +90 222 213 43 44, ismail.yesilaydin@arcelik.com

[2]Eskisehir Osmangazi University, Engineering & Architectural Faculty,
Mechanical Engineering Department, Bati Meselik 26480, Eskisehir, Turkey
Phone: +90 222 239 37 50 / 3366, Fax: +90 222 229 05 35, lberbay@ogu.edu.tr

* Corresponding Author

ABSTRACT

The main component that determines the performance and efficiency of the refrigeration system is the compressor, which is the major energy consumer in a vapor-compression refrigeration system. In hermetic compressors for household refrigerators, one of the major factors that influences the total compressor efficiency is the flow efficiency of discharge line.

In order to find out the effect of the structural changes to flow losses and pressure fluctuations at discharge line, this paper presents a three-dimensional CFD model and experimental study of a hermetic reciprocating compressor. In the first phase of this study, transient numerical flow analysis was developed and it was determined whether or not it overlaps with the experimental measurements. The CFD model results were compared with the experimentally obtained pV indicator diagram and transient pressure measurements of discharge plenum by varying the rotation speed of the modelled inverter hermetic reciprocating compressor.

The result of the studies helps to identify the effect of the discharge line design parameters to the system performance.

1. INTRODUCTION

Today, increasing energy needs and the depletion of natural energy resources force people to find alternative energy sources and to use existing ones efficiently. Urbanization and increasing the quality perception of users leads using reliable, efficient, and noiseless devices. Governments and international associations have determined the declaration levels that they demand from the market according to these trends.

In the household type refrigeration market, that must respond to all these trends, hermetic reciprocating compressors are mostly used. Compressor is the main component that determines many performance criteria, especially efficiency and noise level of the refrigeration system. Lifetime, acoustic noise level, compactness, efficiency and price are the five fundamental requirements expected from hermetic reciprocating compressors (Rasmussen, 1997). In the design phase of a hermetic reciprocating compressor, pressure calculation is important since pressure pulsation affects the performance and noise of a compressor. Pressure pulsation is caused by unsteady flow due to the reciprocating action of the piston and by self-acting valves that open and close depending on the pressure difference (Ma, 2000). In an analytical and experimental study of compressor performance losses, Jacobs (1976) described pressure pulsations in compressor flow lines as one of the missing items in compressor performance. In the study, the relationship between the movement of the suction valve and the pressure pulsations in the suction manifold was investigated. Its indicated that pressure pulsations were affect the movement of valve and forced to reopen after closing the suction valve. For this reason, it was seen that there was a loss of 23% in compressor capacity and 12% in efficiency. (Jacobs, 1976)

On the other hand, the pressure pulsation in the compressor discharge line not only trigger the resonance of the discharge line elements, but also cause vibration-induced noise generated by the pressure fluctuations are transferred to the housing and cooling system components such as condenser. (Dreiman & Flora, 2000; Singh & Soedel, 1974)

Within the scope of this study, pressure fluctuations at discharge line of a hermetic reciprocating compressor were investigated both experimentally and numerically. Numerical analyzes on the modeling of the compressor discharge line has been carried out with a commercial CFD software package. In the study, dynamic mesh structure is used instead of FSI which is an expensive solution method in terms of time and resources. Compression of the piston modeled by the moving mesh structure. The opening and closing of the discharge valve is automatically carried out based on the pressure difference in the cylinder and discharge manifold. On the other hand, experimental studies have been carried out to compare numerical analysis. In the experimental side of the study, the current situation was examined using a commercial variable speed compressor. Compressor was instrumented with pressure transducer to obtain the pressure volume diagram. Pressure fluctuations in the cylinder head, which is the starting point of the discharge line, were measured at different operating speeds of the compressor under the ASHRAE LPB pressure condition. Condensation and evaporation temperatures are respectively 54.4 °C, -23.3°C. The subcool, superheat and the ambient temperature for the compressor were all set to 32.2°C.

It is aimed to use the results of numerical and experimental studies in the calculation of pressure fluctuations in the design phase of a compressor. The generated numerical model helps to calculate the pressure fluctuations at the discharge line of a hermetic reciprocating compressor with minimum input data.

2. EXPERIMENTAL STUDY

The experimental studies are performed to identify the current situation of the pressure fluctuations at discharge line of hermetic reciprocating compressor and determine the boundary conditions to be used in numerical analysis. In addition, numerical analysis and experimental analysis were compared at the end of the study and the accuracy of the numerical model was determined. The compressor used in the study has 145 kcal/h capacity at ASHRAE conditions and the working fluid is isobutane (R600a). The flow path of the discharge line is depicted in Figure 1.

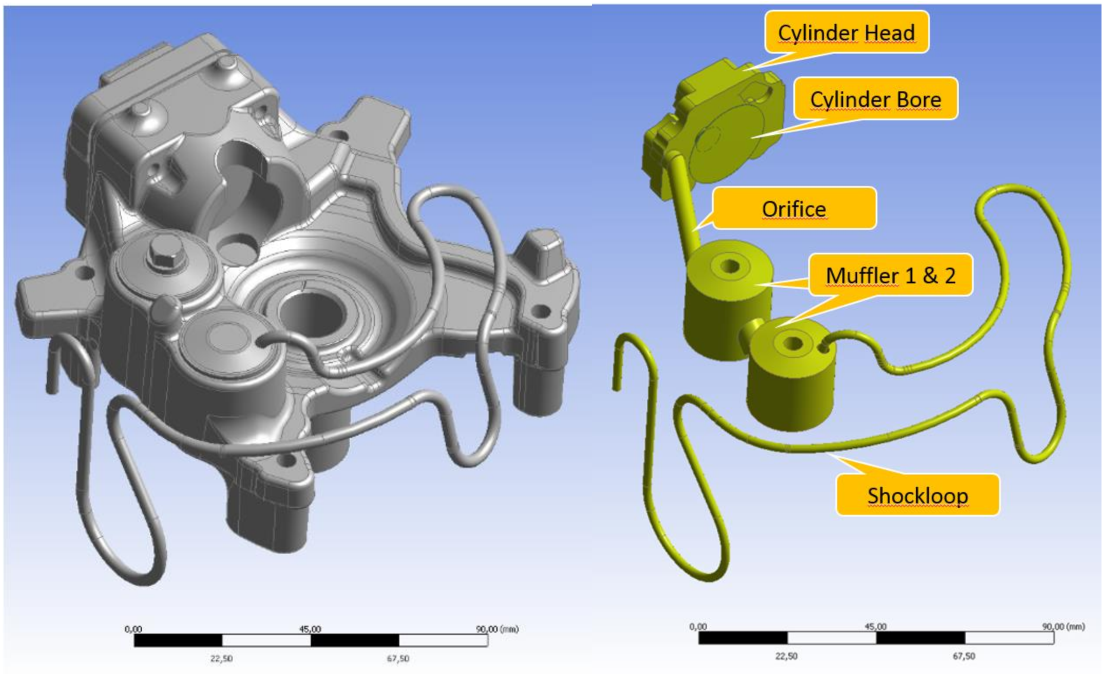

Figure 1: Internal flow path of the discharge system.

In the scope of the experimental study, the compressor was instrumented with pressure transducers and optical encoder to obtain the pV diagram and the pressure fluctuations in the discharge line (Figure 2). Piezo-resistive micro-transducers mounted in valve plate, suction plenum and discharge plenum generated the pressure measurements. The optical encoder that placed on the crankshaft provided us to determine the position of the crank and calculate the momentary cylinder volume.

The transducer has an accuracy of 0.1% FSO due to non-linearity and hysteresis. Pressure range is 0 – 17 bar absolute with a wide temperature range of -55°C to +175°C. Data acquisition were used with a digital scope and transferred to a computer. Software for acquisition and data management developed in a commercial software.

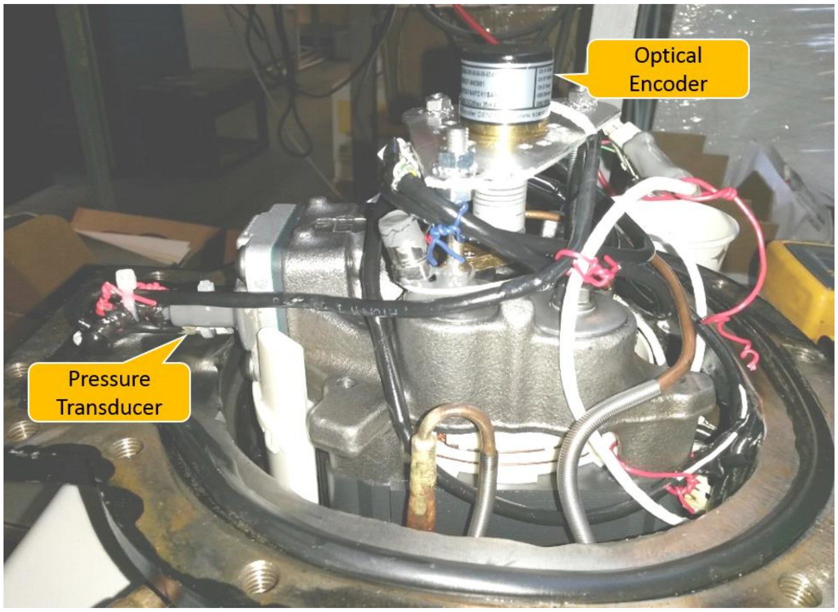

Figure 2: Instrumentation of pressure transducers and encoder.

After the instrumentation of compressor, the pressure measurements were performed while the compressors were running on a fully automated calorimeter system that can be stabilized the operating conditions at ASHRAE point.

3. NUMERICAL ANALYSIS

Numerical analyzes were carried out with the 3D discharge line geometry to understand the flow behavior and the pressure fluctuations at discharge line of hermetic reciprocating compressor. The flow volume that inspected in analysis, is delimited between the cylinder bore where the fluid is pressurized and the compressor discharge pipe. In the case of unsteady numerical analyzes, the gas flow is considered to be compressible.

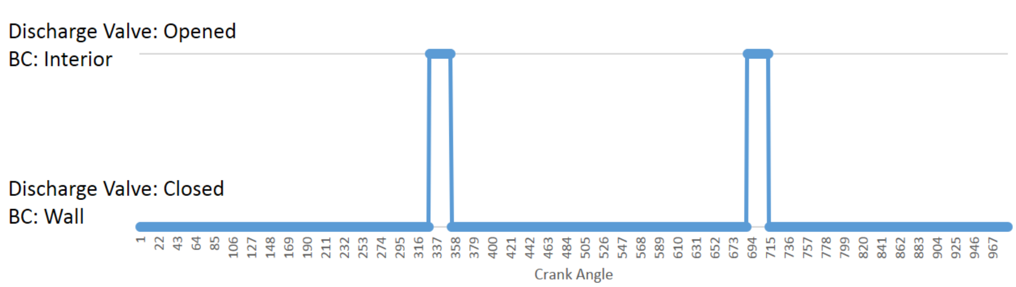

Figure 3: The relationship between the valve movement and the boundary condition with respect to the crank angle.

In the analysis, the movement of the piston is provided by dynamic mesh structure. The parameters of the crankshaft mechanism of the compressor are entered in dynamic mesh In-Cylinder module. Analyzes were performed at 2100, 3000 and 4500 rpm by changing crankshaft speed in this module, and necessary boundary conditions.

Opening and closing of the discharge valve is modeled by converting the flow line to wall and fluid interior boundary conditions. These changes in boundary conditions have been automated from dynamic mesh events depending on the pressure difference between the cylinder bore and the discharge plenum. For the flow condition through the discharge port, the flow area is determined by the maximum clearance that allowed by the discharge valve stopper. The relationship between the valve movement and the boundary condition is given in Fig. 3 with respect to the crank angle.

At discretization stage, tetrahedral and hex mesh is generated for discharge line. The smallest and the biggest element sizes are $5.5{\cdot}10^{-5}$ m and $1{\cdot}10^{-3}$ m respectively. The total number of elements is approximately 800,000 which is enough for mesh independency. Dynamic mesh structure and the 3D geometry of cylinder bore and discharge plenum are shown in Figure 4.

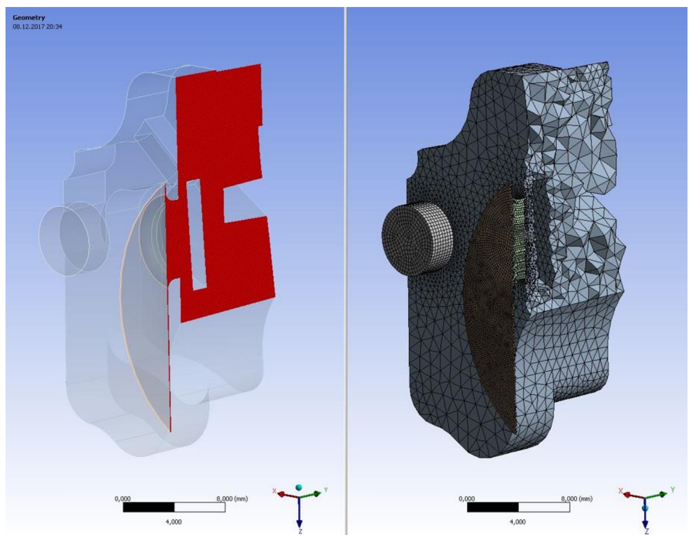

Figure 4: Dynamic mesh structure and the 3D geometry of cylinder bore and discharge plenum.

The behavior 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.

Isobutane (R600a) is used as a refrigerant and in the simulations the fluid properties have been implemented as constant at mean temperature of operating speed condition. The refrigerant is considered as a real gas and assumed to be pure without oil. Table 1 shows the refrigerant properties for an operating condition.

Table 1: The refrigerant properties.

Properties	Isobutane (R600a)
Density (kg/m3)	Ideal Gas
Specific Heat Cp (j/kg-K)	2151.4
Thermal Conductivity (W/mK)	0.02586
Dynamic Viscosity (kg/ms)	9.43017e-6
Molecular Weight (kg/kgmol)	58.1222

For the boundary conditions of discharge tube pressure outlet condition was set. There is no fluid inlet section in the system due to the fact that the suction flow line is neglected in the simulation. An initial cylinder pressure that obtained from pV experiments was set at the end of the suction phase of each cycle. Wall temperatures of cylinder and other discharge line areas are assumed to be constant and are the temperature values obtained by experimental measurements in steady state condition. The defined boundary conditions are summarized in Table 2.

Table 2: Boundary conditions.

Zone	Boundary Condition
Discharge Valve	Wall / Interior
Piston	Mowing Wall
Shockloop Out	Pressure Outlet
Cylinder	Wall
Discharge Line	Wall

4. RESULTS

As a result of numerical analysis, pV diagram and pressure fluctuations at discharge line are obtained. It can be seen that the pV results of the analysis are consistent with the experimental measurements except for the suction phase which is neglected and not included in the analysis. In Figure 5, experimental measurements and numerical analysis results are given at 3000 rpm operation speed.

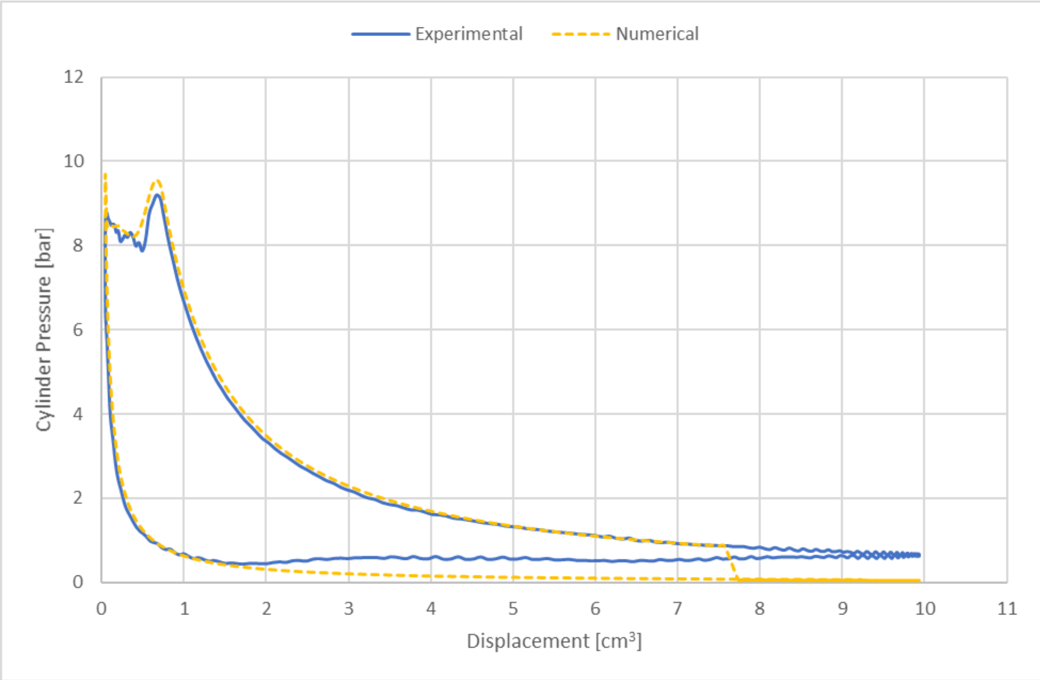

Figure 5: Experimental and numerical pV results at 3000 rpm.

As can be seen from the pV diagram, the pressure definition for the inside of the cylinder was made in the 7.5 cc cylinder volume area in the numerical analysis. All the numerical analysis results are examined in the 3rd cycle of computation.

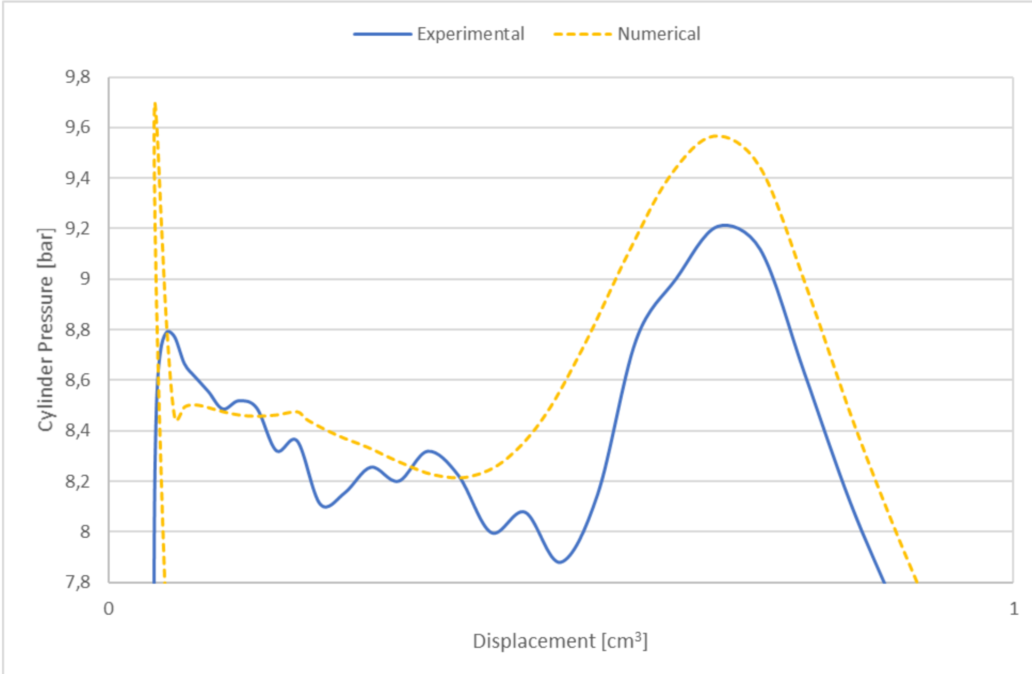

Figure 6: Discharge phase of the pV cycle.

When we closely examine the exhaust phase at Figure 6, there is a difference of about 0.2 bar between the numerical analysis and the experimental result in the overshoot zone. Furthermore, unlike the experimental measurements, an instantaneous increase in the cylinder pressure was observed with the closing of the discharge valve at the end of the discharge phase.

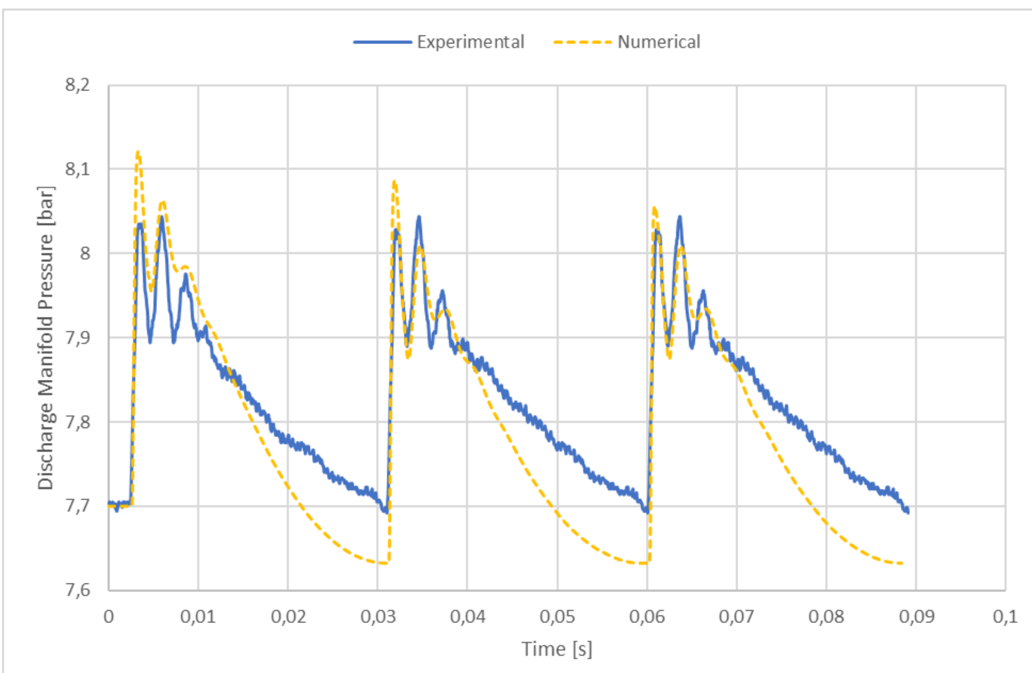

Figure 7: Pressure fluctuations at discharge plenum. (2100 rpm)

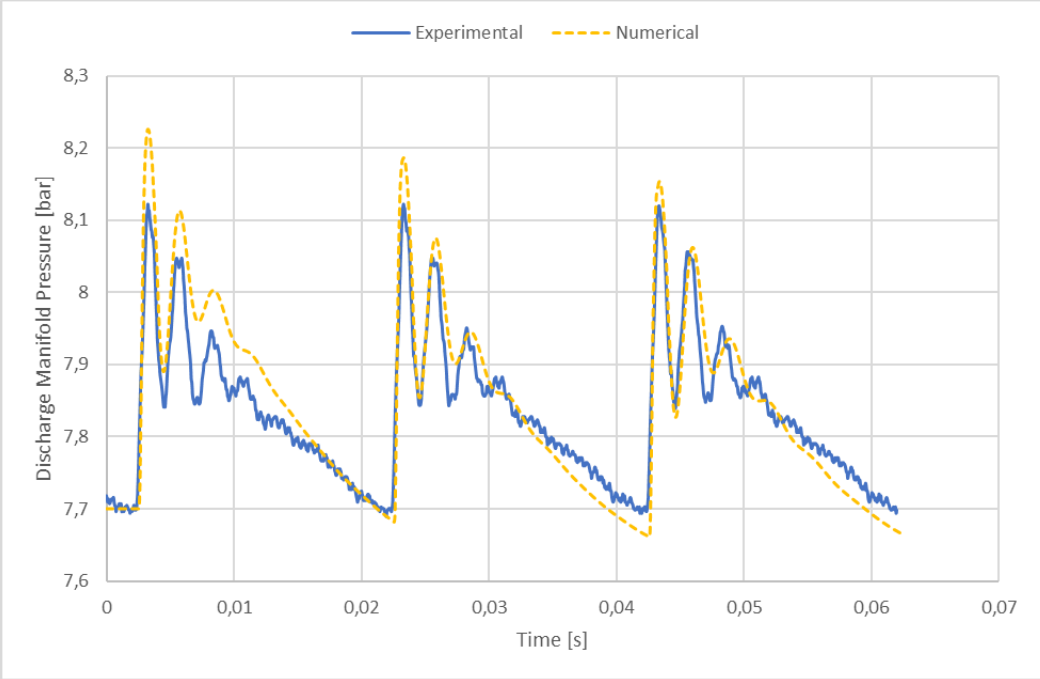

Figure 8: Pressure fluctuations at discharge plenum. (3000 rpm)

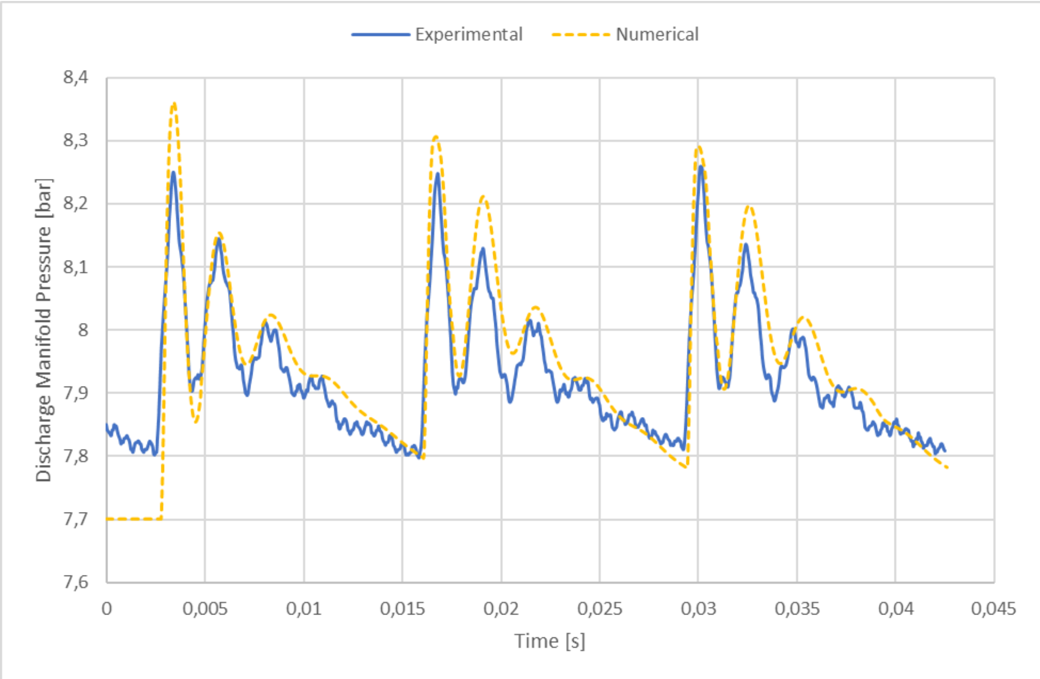

Figure 9: Pressure fluctuations at discharge plenum. (4500 rpm)

The discharge plenum pressures obtained from the numerical analysis are compared with the experimental measurements carried out at the operating speeds of 2100, 3000 and 4500 rpm in Figures 7, 8 and 9 respectively. It has been verified that the numerical analysis results are in good agreement with the experimental values for the 3rd cycle of computation.

5. CONCLUSIONS

This paper presented a three-dimensional CFD model developed to investigate of pressure fluctuations and the flow behaviour at discharge line of a hermetic reciprocating compressor. Developed numerical solution included cylinder bore, discharge valve and discharge mufflers with shockloop tube. The experimental data have shown that the numerical model can predict pressure fluctuations at the discharge line at different operating speeds. Based on the results of this study it will be extended that:

- The effect of geometric changes in the discharge flow line on pressure fluctuations will be examined.
- The movement of the discharge valve will be examined experimentally and compared with the opening and closing times in the analyses.

NOMENCLATURE

CFD	Computational Fluid Dynamics	(–)
FSO	Full Scale Output	(–)
LBP	Low Back Pressure	(–)
p	Pressure	(bar)
V	Volume	(cc)

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