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ABSTRACT

Nowadays, increasing the efficiency of white goods and small house appliances has become necessity as a result of the technological developments, competition and energy policies. As a basic component of the cooling system, compressor is the most fundamental element for determining the performance and efficiency of the system. In hermetic compressors for household refrigerators, one of the major factors of total compressor efficiency is the discharge line flow efficiency.

In this paper, the effects of some of the structural changes to flow losses at discharge line of a hermetic reciprocating compressor were investigated experimentally. For this study, three different conceptual designs were created based on the discharge line design parameters such as line diameter, resonator volumes and line length. By performing precise pressure measurements and calorimeter tests, performance criteria and pressure drop rates of conceptual designs were compared against the base model and each other of the conceptual design compressors.

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

1. INTRODUCTION

In recent years, the growing global need for energy and running out of the natural energy sources has forced the human beings to find alternative sources and use the existing ones efficiently. In this way, energy saving practices and precautions reduce the effects of global warming which is serious threat to the future of the world.

According to electricity distribution and consumption statistics, 2,836,637 GWh of electricity 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 (Bertoldi *et al*., 2012). 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 (EIA, 2007).

One of the most important parts of refrigeration system is compressor which is the hearth of refrigeration systems. Basically, compressors determine the performance and efficiency of the cooling system. This basic component of the cooling system consumes the electrical energy and converts it to energy stored in 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, these are 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 (Rasmussen, 1997). Studies for improve the efficiency of compressors are grouped under three main headings: increasing the efficiency of electric motor, reducing the mechanical loses and also reducing the thermodynamic loses due to the irreversibilities in the suction, compression and discharge process.

Several designs have been put forward in the literature to improve the performance of household reciprocating compressors. Rigola *et al*. (2002) experimentally investigated the thermal and flow characteristic of a hermetic reciprocating compressor by transient temperature and pressure measurements. They obtained a comparison of the experimental data and the numerical results by using an advanced detailed code. Ribas *et al*. (2006) presented a critical review of different approaches for reciprocating compressors thermal analysis. They emphasized that the thermodynamic efficiency was much lower than the electric and mechanical system efficiency of household reciprocating compressors. They also indicated that the major part of the thermodynamic losses was originated by viscous losses in the suction and discharge line. Ozdemir *et al*. (2013) experimentally and theoretically examined heat transfer processes between the components of a hermetic reciprocating compressor. In the experimental part of the study, they made conceptual designs on the components of the hermetic compressor to identify the thermal characteristics of the components. At the end of the study Ozdemir generated a network which has included all kind of heat transfer methods inside the hermetic reciprocating compressor. Kara and Oguz (2010) performed three different studies to understand the thermodynamic losses of household type refrigeration compressors. At the experimental part of his study, interaction between the crankcase design and the refrigerant temperature was investigated with detailed temperature measurements.

Within the scope of this study, the effects of some of the structural changes to flow losses at discharge line of a household type hermetic reciprocating compressor were investigated experimentally. 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 created. By performing precise pressure measurements and calorimeter tests; cooling capacity, COP (Coefficient of Performance) values and pressure drop rates of conceptual designs were compared against the base model and each other of the conceptual design compressors.

2. EXPERIMENTAL STUDY

2.1 Description of Conceptual Designs

The experimental studies are performed to identify the effect of the discharge line design parameters to the system performance of the compressor. For this purpose, four different designs were compared based on performance criteria of the compressors and the pressure drop rates of the discharge lines. Compressor A is the base model. The cooling capacity of this compressor is 200 W at ASHRAE conditions and the working fluid is isobutane (R600a). The first conceptual design, Compressor B, has an orifice diameter ($Ø_2$) which is 25 percent wider than the base value. At the second conceptual design referred as Compressor C, the orifice positioned between the opposite discharge muffler (Muffler 2) and the cylinder head. In this design, orifice is got closed to the discharge port and aligned to the opening of discharge valve. Discharge mufflers are not connected in series, one of the mufflers (Muffler 1) positioned as inactive. When the cross over tube between the active (Muffler 2) and inactive muffler (Muffler 1) is removed, we obtained the last conceptual design named as Compressor D. In order to better understanding, the flow path of the discharge system is depicted in Figure 1 and a one dimensional schematic comparison of the flow path of conceptual designs is given in Figure 2.

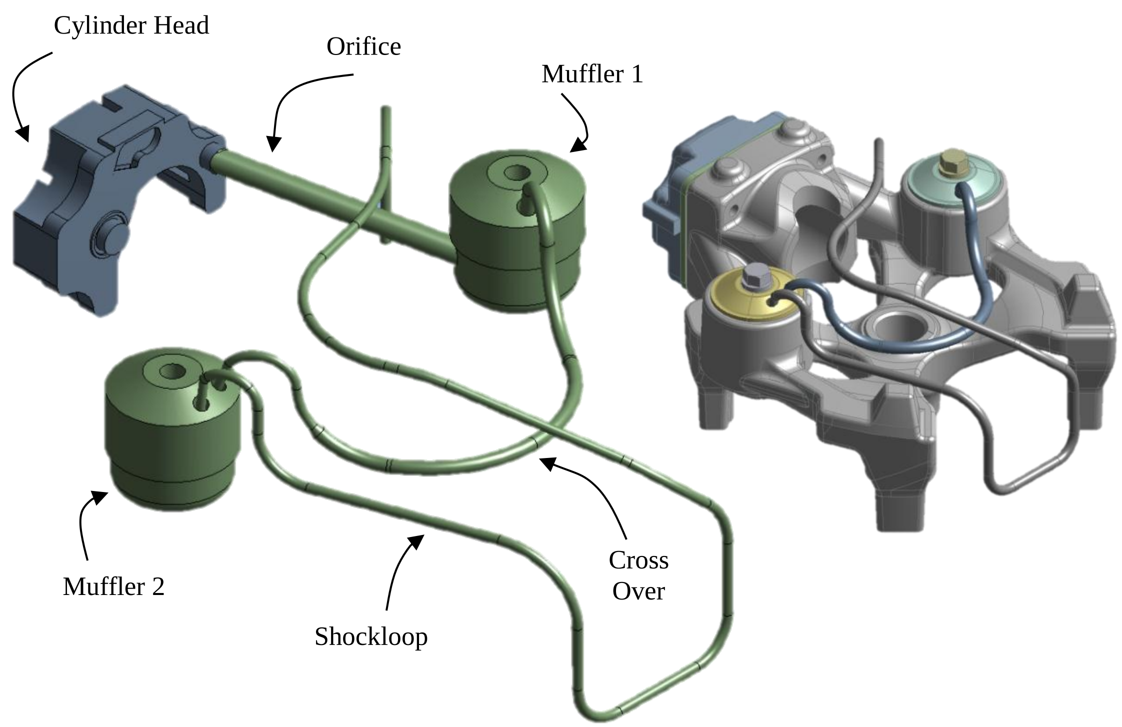


Figure 1: Internal flow path of the discharge system.

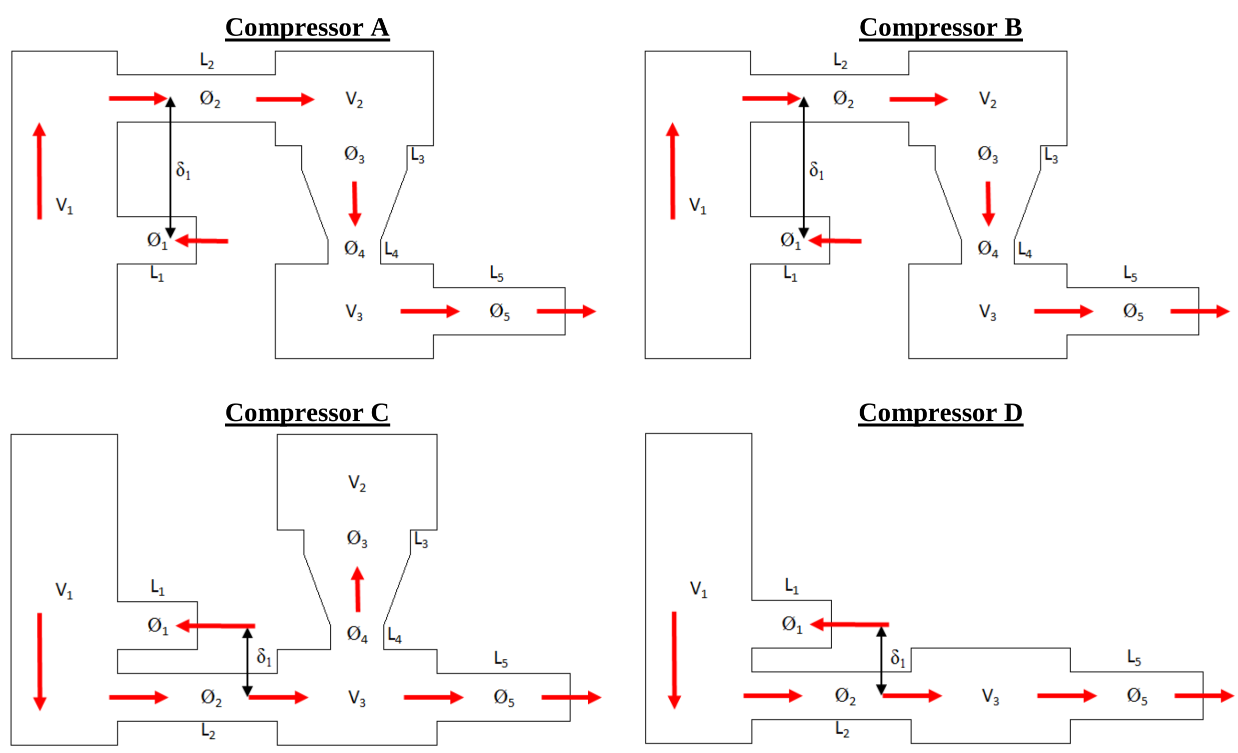


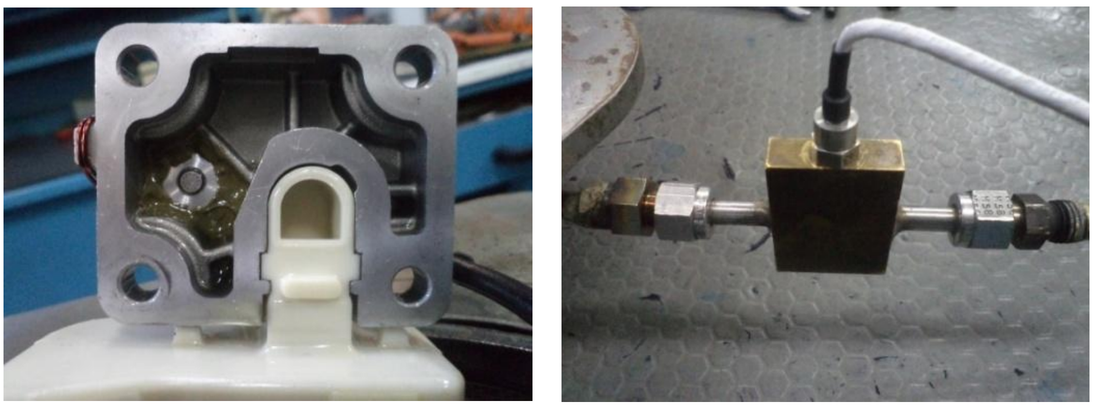
Figure 2: One dimensional schematic comparison of the flow path of conceptual designs.

Modified dimensions of the prototypes are given in Table 1.

Table 1: Modified dimensions of the prototypes.

Explanation	Symbol	Compressor Type			
		A	**B**	**C**	**D**
Distance between the orifice and the discharge port	δ_1	δ	δ	0.3 δ	0.3 δ
Orifice diameter	$Ø_2$	Ø	1.25 Ø	Ø	Ø
Number of discharge muffler		2	2	2	1

Per all compressor types, four prototypes have been created and instrumented with precise pressure transducers that were located between inlet and outlet of the discharge flow path. For the inlet side of the discharge flow path, pressure transducer was located in cylinder head (Figure 3a). The second pressure transducer was located on discharge pipe with a T-connection at the end of discharge line of the compressor (Figure 3b).



(a) (b)

Figure 3: Localization of pressure transducers.

2.2 Equipment and Test Conditions

In this sub-section, the technical specifications of the equipment are presented. The measurement signal was generated by a piezo-resistive micro-transducer. The transducer has an accuracy of 0.1% FSO due to non-linearity and hysteresis. Pressure range is 0 – 1.7 MPa 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 Labview™ language.

After the prototyping, 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; namely +54.4°C condensation and -23.3°C evaporation temperatures. The subcool, superheat and the ambient temperature for the compressor were all set to 32.2°C. The calorimeter and the pressure measurement tests take about four hours in order to ensure the thermal stabilization of all the components inside the compressor. For the calculations the last 60 minute's data was collected by DAQ system which collects the pressure data at intervals of 0.5 seconds.

3. RESULTS

One of the most important components in terms of thermodynamic attitude of a hermetic reciprocating compressor is the discharge line. The discharge line does not only put the refrigerant out of the compressor cylinder to the cooling appliance, but also reduces the pressure pulsations of refrigerant. On the other hand, due to the hot and high pressurized refrigerant within the discharge passage, heat transfer occurs from the discharge line to the other compressor components and the refrigerant inside the shell. According to this point of view, discharge line should be as short as possible. So, this study intends to reduce the length of discharge passage by using the conceptual designs that the details mentioned above.

The results obtained in steady state regime for pressure drop rates of conceptual designs are presented in Table 2. As it is given in Table 1, Compressor B column in Table 2 is including the results of widening the orifice between the Muffler 1 and Cylinder Head. Widening the orifice reduced the pressure drop rate by 1.37%. Respectively, reducing the number of discharge muffler and using of an inactive discharge muffler reduced the pressure drop rate by 13.76% and 16.01%.

Table 2: Measured average pressure.

Transducer Location	Average Pressure			
	Compressor A	**Compressor B**	**Compressor C**	**Compressor D**
Pressure Drop [MPa]	P	0.986 P	0.839 P	0.862 P
Difference from Base [%]	**0**	**-1.37**	**-16.01**	**-13,76**

During the operation of reciprocating compressor, the refrigerant is sucked into the cylinder and compressed into the discharge passage as the reciprocation of piston. Because of this process, compressor cannot discharge the refrigerant continuously; pulsation occurs at discharge flow. As it is known, pressure pulsation is one of the noise sources of the hermetic compressor and amplitude of the pressure pulsation is desired as low and smooth as possible. The behavior of the pressure pulsation according to compressor designs is shown in Figure 4.

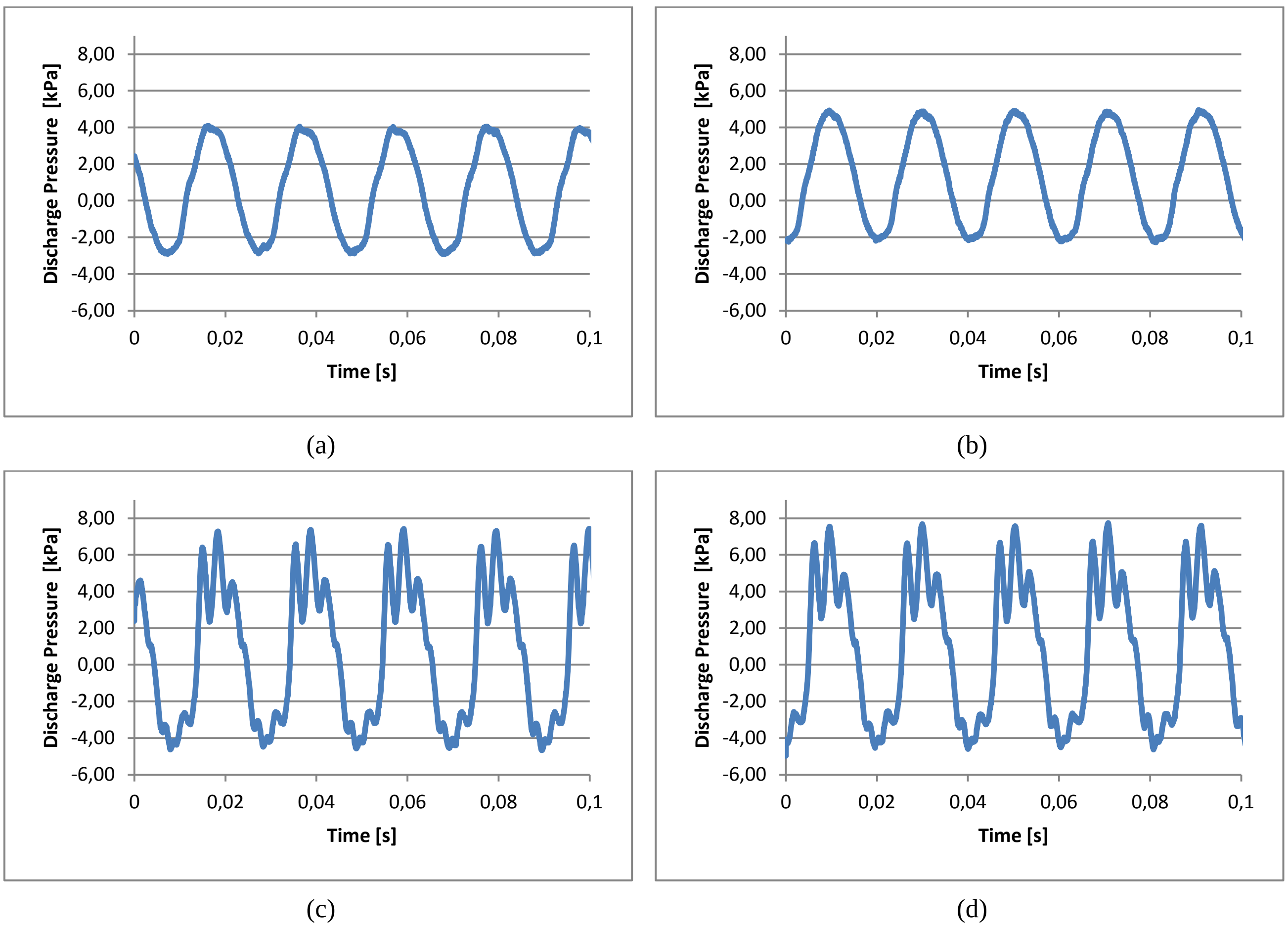


Figure 4: The behavior of the pressure pulsation according to compressor designs.

From Figure 4(a), one can observe that the amplitude of pressure pulsation concerned with the base model is quite smooth and it has low amplitude. The lowest and the smoothest amplitude are observed at the Compressor B (Figure 4b). Using an inactive discharge muffler and reducing the discharge passage length causes an oscillation in the discharge pressure (Figure 4c). As it can be seen in Figure 4(d), using an inactive muffler or connecting the discharge resonators as parallel doesn't affect the behavior of the pressure pulsation. Compressor D also has an oscillation in the discharge pressure.

The comparison of the pressure pulsation amplitude according to compressor designs is given in Figure 5. It is clearly seen that longer discharge passage results less pressure pulsation. According to the base design, Compressor C and D have approximately 0.7 times more pressure pulsation amplitude at the discharge process which has less throttling.

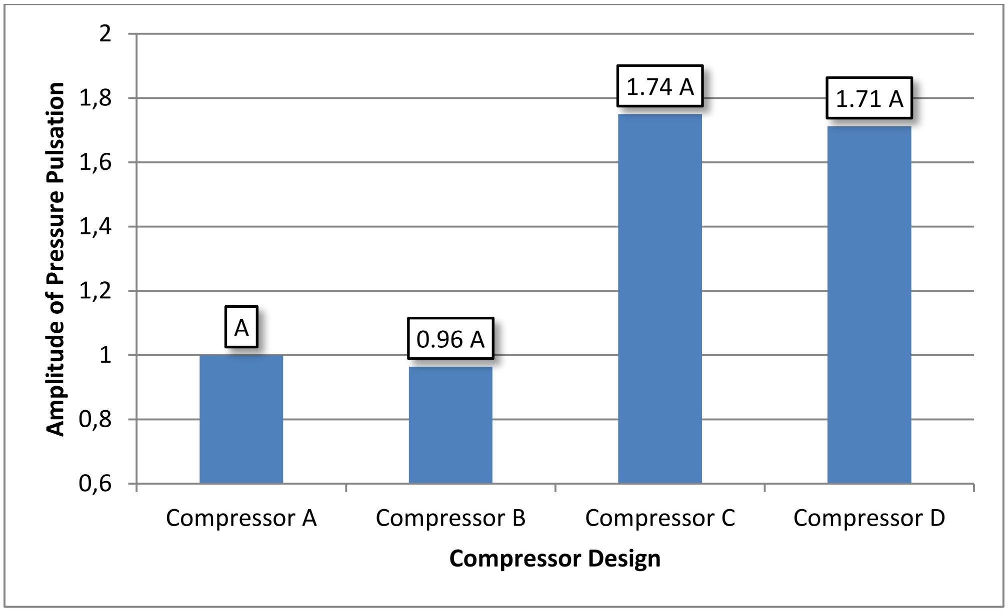


Figure 5: The amplitude comparisons of the pressure pulsation according to compressor designs.

Table 3 presents results for the main performance parameters of the prototype compressors. The comparison of the performance parameters comprises average of the four prototypes. The difference between the results of prototypes per each compressor design was around 2% for the cooling capacity and pressure pulsation.

Table 3: Comparison of the main performance parameters. (Evaporation: -23.3°C/ Condensation +54.4°C)

Compressor Design	Cooling Capacity Difference [%]	COP Difference [%]
Compressor A	0.00	0.00
Compressor B	-0.12	0.30
Compressor C	2.14	0.84
Compressor D	1.96	1.50

Results show the influence of the thermodynamic improvements on the main performance parameters. According to the results, widening the orifice at Compressor B doesn't affect the cooling capacity but it reduces the discharge losses and inlet power. For this reason the COP value increases by 0.30%. Cooling capacity of the Compressor C is increased by 2.14% due to reducing the discharge passage length by using one of the discharge muffler (Muffler 1) as inactive. Using the Muffler 1 as inactive increases inlet power compared to Compressor D. This case can be explained by temperature rise of the Muffler 1. The absence of flow in the Muffler 1 could be caused temperature rising in this region. Reducing the number of discharge muffler increased the cooling capacity and COP by 1.96% and 1.50%, respectively.

4. CONCLUSIONS

In this study, investigation of the effects on gas flow efficiency and the behavior of the pressure pulsation at exhaust line of a reciprocating hermetic compressor were presented and the following conclusions were obtained:

- From the pressure drop point of view, the length of the discharge passage is the determiner factor with reducing the pressure drop rate by 16.01%.
- In spite of the pressure drop rate, the smoothest and the lowest amplitude of pressure pulsation was achieved by widening the orifice between the cylinder head and the muffler. Based on the presented results, decreasing the discharge passage length affected the pressure pulsation value negatively. Although shortening the discharge passage causes an oscillation and increases the amplitude up to 0.7 times, these cases must be reviewed in terms of acoustics.
- The main performance parameter results show that the shortest discharge passage is the most efficient. Expanding the discharge muffler volume by connecting the resonators as parallel doesn't affect the pressure drop or pressure oscillation substantially. The absence of flow in the inactive muffler could be caused temperature rising in this region. Therefore, Compressor D case slightly increases the efficiency of the compressor value in comparison with Compressor C.

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

- The pressure drop results will be validated by CFD analysis.
- Experimental study will be extended with the temperature and heat flux measurements.
- Increasing the number of pressure measurement point will elaborate the pressure drop value with respect to detailed control volumes.
- Pressure pulsation measurements will be correlated with acoustic measurements.

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