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Numerical Evaluation of Performance Curves of a High Frequency Microcompressor

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ABSTRACT

This work presents a methodology for the numerical simulation of reed valves of a small-size reciprocating compressor (approximately 200 mm in length, 60 mm in diameter), oil free and operating at high frequency. A fluid-structure interaction methodology is employed, where the fluid domain is modeled by the tridimensional transient Navier-Stokes equations, while the solid domain is governed by a beam model considering geometrical contacts between valves and stoppers (valve plate, piston, etc.) Results are shown for mass flow, pump power consumption and valve losses for evaporating temperatures ranging from -15 degrees Celsius to +40 degrees Celsius, and condensing temperatures from +45 degrees Celsius to +85 degrees Celsius. Finally, numerical results were compared to experimental data.

1. INTRODUCTION

Cooling and refrigeration of portable systems and electronic equipments such as IT systems, displays, projectors, industrial enclosures and cooled vests demands a large cooling capacity and small size at evaporating temperatures much higher than that found on typical refrigeration systems. For example, in electronics cooling it is important to maintain the temperature as low as possible, but above the dew point. In some cases the evaporating temperature is just a bit lower than the ambient temperature as a way to ensure that no condensation of moisture will occur. Typically, electronic equipments work surrounded by air at 30 to 50 [°C].

Possamai et al. (2008) demonstrated that a linear motor with a reciprocating piston, both operating at frequencies a way higher than the traditional 50/60 [Hz] is the most suitable solution for small scale cooling and refrigeration systems, with the benefit that the compressor could operate without any oil (oil free).

Shioni et al. (2009, 2010, 2011) applied numerical methods to accurately predict the behavior of a reciprocating compressor with reed valves, showing very good agreement between the numerical and the experimental results in terms of pressure-volume diagram, valve displacement and pressure pulsations at the suction line.

Takemori (2009) adapted the methodology developed by Shioni et al. (2009, 2010, 2011), replacing the single degree of freedom parallel motion of the valves by a beam model. Moreover, Takemori (2009) explored the influence of various numerical parameters (advection and transient schemes, convergence criteria and temporal discretization) on the behavior of a miniaturized compressor operating at 350 [Hz].
That said, the objective of this work is to evaluate the performance of a high frequency miniaturized compressor (Figure 1, Figure 2 and Figure 3) under a wide range of evaporating and condensing temperatures, focusing on temperatures typically found on electronics cooling applications.

**Figure 1:** Photography of the high frequency miniaturized compressor. The weight of the compressor with inverter is 1300 grams.

**Figure 2:** Top view of the compressor showing main dimensions.

**Figure 3:** Back view of the compressor showing main dimensions.

All results presented correspond to an oil free miniature reciprocating compressor with these specifications:
- Operating frequency: 335 [Hz]
- Volumetric displacement: 0.267 [cm$^3$]
- Dead volume: 0.0351 [cm$^3$]
2. METHODOLOGY

2.1 Numerical procedure
The numerical procedure follows a fluid-structure interaction methodology (Takemori, 2009) where the fluid domain is modeled by the tridimensional transient Navier-Stokes equations, while the solid domain is governed by a beam model considering geometrical contacts between valves and stoppers (valve plate, piston and discharge stopper).

2.1.1 Fluid domain: The fluid domain includes the suction line, chamber, tubes and orifices, the cylinder, and the discharge orifices, chamber and line (Figure 4).

![Fluid domain diagram](image)

*Figure 4: Fluid domain.*

The compressor has a total of four valves: Two suction valves and two discharge valves (Figure 5). With that configuration it was possible to achieve a very good tradeoff between fluid flow area and valve stiffness and natural frequency.
The compressor works with R134a modeled with Aungier Redlich Kwong equation of state (ANSYS, 2011). The energy equation considers all terms, including the viscous work. Based on the research of Shiomi (2011), the SST turbulence model was chosen.

The suction line has as boundary condition pressure equals to the evaporating pressure and temperature of 60 °C. The discharge line has an outlet boundary condition with a pressure set equals to the condensing pressure. Piston displacement follows a sinusoidal motion, since the compressor works with a resonant mechanism. All walls were considered adiabatic.

2.1.2 Solid domain: The mathematical model that governs the solid domain is the classical pure bending Euler beam model. As a means to facilitate the input of data, the beam properties are replaced by two only: Natural frequency and equivalent stiffness (Figure 6).

2.1.3 Fluid-solid coupling follows an explicit scheme (Figure 7) considering variable time step in order to reduce the computational time. One time step value is used for compression and expansion, other for suction process and another for discharge process.
2.2 Experimental setup
Measurements were made according to the procedure described by Shiomi (2010). The experimental setup (Figure 8) is composed by control valves (CV), mass flow meter (FM), heat exchangers (HX), thermocouples (TC) and pressure transducers (PT). The measurement is carried out in a way that the R134a can flow through the high and low pressure lines in the superheated state, as indicated in Figure 8b. According to this arrangement, refrigerant enters the compressor (C) at point 1 and is compressed to point 2. After that, the fluid is cooled and expanded (2-3-4), and the mass flow is measured. The refrigerant is cooled again and throttled to the evaporation pressure (6). Finally, the suction line temperature is adjusted by an electrical heater (EH1), completing the operating cycle.

3. RESULTS
A comparison between the experimental data and the results predicted by the numerical procedure (Figure 9) shows a very good agreement. The estimate is a bit higher probably due to a bigger dead volume on the experimental setup.
Analyzing the results for the condensing temperature of 85 [°C], it is possible to notice that the experimental and the estimated results differ for evaporating temperatures above 25 [°C]. It occurs due to a limitation on the experimental side, where the electric motor can't supply the necessary power to the pump, and the electronic control of the compressor compensates that with a reduction on the volumetric displacement and an increase on dead volume. This reduction leads to lower mass flow rate.

It is possible to evaluate an ideal mass flow rate based solely on volumetric displacement, operation frequency, dead volume, pressure ratio, specific heat ratio and density at suction (Gomes, 2006). Figure 10 shows the ideal and the estimated mass flow rate. Notice that the difference between these results is relatively small at a condensing temperature of 45 [°C], and becomes larger when the condensing temperature rises. This behavior can be explained by a difference on dead volume. Figure 11 shows the ideal mass flow rate with a bigger dead volume (0.055 instead of 0.035 [cm³]). It is clear now that both results are very similar.
Figure 11: Mass flow rate for evaporating temperature ranging from -5 to +40 °C and condensing temperatures ranging from +45 to +85 °C.

In both cases - experimental and numerical - the fact that the effective dead volume is larger than the geometric dead volume needs to be better understood.

Knowing the pump power consumption is important to design or redesign the electric motor. Figure 12 shows the indicated power for the various conditions of evaporating and condensing temperatures.

Figure 12: Indicated power (or pump power consumption).

Figure 13 shows suction and discharge losses as a percentage of indicated power. As a typical value is 8% (Shiomi, 2011), it is important to analyze in detail the flow field in order to identify the causes of the significant losses.
4. CONCLUSIONS

This work presented the performance curves of a high frequency miniaturized compressor using both numerical simulation and experimental measurements. The results showed a very good agreement between these two methods. Moreover, the numerical analysis showed that the suction and discharge losses are considerably higher than typical values. A detailed study of the flow field can provide important information to improve the compressor efficiency.

Finally, some issues should be addressed on future works. An ideal mass flow rate model showed that the performance of the compressor can be considerably higher, and it needs to be better understood.

REFERENCES

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