EXPERIMENTAL INVESTIGATION OF TRANSIENT FLOW IN RECIPROCATING COMPRESSORS

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Abstract. *Measurements of velocity and pressure inside the suction chamber of a refrigeration reciprocating compressor were carried out to analyze transient effects during the suction process. Due to the high speed compressor chosen for the analysis, a hot-wire system and a piezoelectric pressure transducer were necessary to properly characterize phenomena that occur on very small time scales. On the other hand, because the compressor adopts R134a as the working fluid, a special method was developed to calibrate the hot-wire sensors. The time interval in which the valve is either open or closed is identified from experimental data for valve displacement. A detailed description of the flow is carried out with reference to the aforementioned results.*

Keywords*: compressor, suction process, hotwire, pressure transducer, flow*

1. INTRODUCTION

A schematic view of a reciprocating compressor and its indicator diagram is shown in Figures 1 and 2, respectively. At point A, the piston is at the top dead center of its stroke (TDC) and cannot move any closer to the valve plate. As the piston moves downward, the vapor left in the cylinder clearance is expanded to fill the increasing volume in the cylinder. This expansion process takes place along line A-B.

When the piston reaches point B, the cylinder pressure is reduced below the pressure of the vapor in the suction line. Once this occurs, the pressure difference gives rise to a force that eventually becomes greater than the spring force and the valve opens and low pressure refrigerant flows into the cylinder. As the piston continues moving along its intake stroke from B to C, the cylinder is filled with vapor from the suction line. During this period, the pressure in the cylinder remains almost constant, at a value slightly below that in the suction line. The flow of vapor into the cylinder continues until the piston reaches the bottom dead center of its stroke at point C (BDC). Then, the pressure difference between the suction chamber and the cylinder becomes negligible and the valve spring actuates, closing the suction port.

As the piston moves toward TDC, the pressure of the vapor in the cylinder increases along line C-D. When the piston reaches point D, the pressure of the vapor in the cylinder has increased beyond that in the discharge line. This pressure difference forces the discharge valve to open against its spring force, with high-pressure vapor exiting the cylinder through the discharge port and flowing into the discharge line. The flow of the vapor through the discharge valve continues as the piston moves from D to A. During this period, the pressure in the cylinder remains almost constant, at a value that is higher than the discharge pressure. When the piston reaches point A (TDC), the compression cycle is finished and the crankshaft of the compressor has rotated a complete revolution.

The flow in the suction muffler plays an important role in the compressor performance since it affects the flow head loss, gas superheating and the valve dynamics. Therefore, knowledge about the fluid flow in the suction muffler along the compressor operation cycle allows the identification of the main inefficiency sources. However, the experimental analysis of this phenomenon is not a trivial task, requiring elaborate techniques to characterize the steep property transients that are present in the flow.

Laser Doppler velocimetry (LDV) is a non intrusive technique usually applied to measurements in IC engines, which does not depend on the fluid properties to determine the local flow velocity. However, the seeding of particles in the flow makes it difficult to obtain data with the acquisition rates necessary in high speed compressors. Witze (1981) compared the main differences between hot-wire anemometry (HWA) and LDV through an extensive study in an internal combustion engine with a specially constructed engine head, operating at 600 rpm. Among the advantages of HWA, the author indicated the possibility of characterizing flows with low turbulent intensity and low compression rate. However, a major disadvantage of HWA is its dependence on the property of the working fluid, in addition to being intrusive and not allowing a simple determination of the flow direction (Bruun, 1995). The author also pointed out that, despite the possibility of calibrating the hot wire probe for different temperature ranges, the choice of analytical models is more appropriate to take into account this temperature dependence.

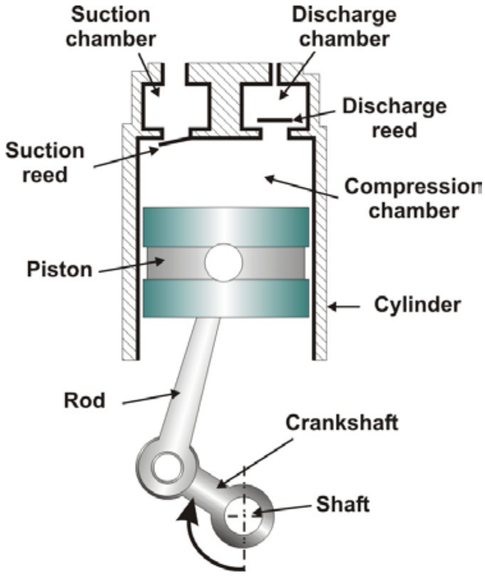

Figure 1. Compressor mechanism

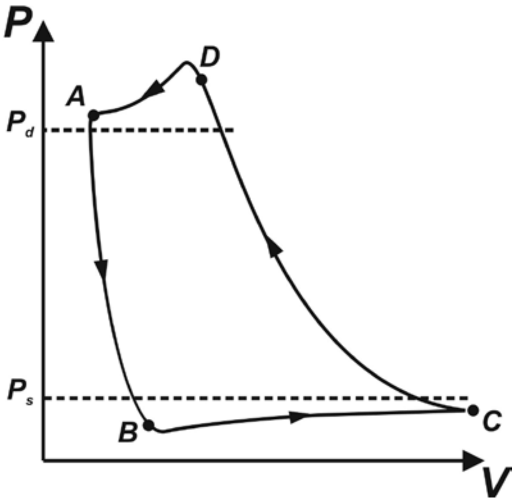

Figure 2. Compression cycle

Experiments of flow in inlet manifolds of IC engine were conducted by Bauer *et al.* (1998) with the purpose of analyzing heat exchange for different flow parameters. The instrumentation of the manifold was made with thermocouples, heat flux sensors positioned along the manifold, and next to the exit, a special port was arranged to fit, alternatively, a hot-wire, a cold-wire and a heat flux sensor. The authors concluded that the period in which the flow is stagnated contributes strongly to the total heat transfer to the fluid in the inlet manifold.

From the review given, it is clear that there is very little information about the fluid flow through suction mufflers of refrigerating compressors. In this paper, an experimental investigation of fluid flow transients is carried out in the suction muffler of a small high speed reciprocating compressor, operating at 3600 rpm. Results for valve displacement, velocity, and pressure in the suction chamber are used to analyze the gas flow.

2. EXPERIMENTAL PROCEDURE

A reciprocating compressor operating with R134a was selected for the analysis, being submitted to different operating conditions, represented by two pairs of evaporation and condensation temperatures: (-23.3ºC/54.4ºC) and (-35.0ºC/54.4ºC). The first condition defines the pressure in the suction and discharge lines as 1.149 bar and 14.71 bar, respectively. When the evaporation temperature is lowered to -35.0ºC, the suction pressure becomes equal to 0.6617 bar. A calorimeter facility was employed to investigate the compressor under the specified operating conditions. The uncertainties associated with measurements taken with the calorimeter are $\pm$3% for mass flow rate and power consumption. Further details of the experimental facility can be found in Pereira *et al.* (2008). The compressor was tested three times for each operating condition. The acquisition system is composed of a computer, a National Instruments converter analogue/digital (A/D), model 6071E, and a program for data acquisition and reduction developed using LabView 8.5.

Piezoelectric pressure transducers were selected for the measurements in the suction chamber, due to their high response frequency, small size and reliability regarding the hostile conditions inside the compressor. To correlate the pressure measurement with the crank angle, a sensing winding was assembled to the crankcase to collect the signal emitted by a magnet fixed to the crankshaft. The instantaneous crankshaft position is calculated taking into account the compressor mechanism characteristics. Small sensing windings were also assembled into the valve plate seat to give the valve lift according to the crankshaft position.

A single 5 μm diameter tungsten wire sensor (55P11) was employed for velocity measurements. Measurements of instantaneous velocity were conducted with a DANTEC MiniCTA system (54T30). The sensor was calibrated by using a DANTEC calibration unit (90H10), with a temperature sensor (55A76) to register any significant variation during the process.

It should be mentioned that a number of aspects may affect measurements of velocity with anemometry sensors. For instance, Weiss *et al.* (2005) has shown that the presence of oil droplets in the flow can increase the sensor diameter with running time, leading to a reduction of its frequency response. Such a contamination process requires testing and cleaning the wire steadily in order to keep a significant frequency response. This is particularly relevant in compressors because oil is always present in the suction system.

The calibration of the hot-wire sensor was carried out with a DANTEC calibration unit (90H10), in which the sensor is exposed to a range of flow velocities typically found in the suction chamber. A number of 20 different velocity levels was employed to adjust a calibration curve to express the velocity, *U*, as a function of the voltage indicated in the anemometry system, *E*. During the calibration procedure, pressure and temperature prevailing in the flow were also registered so as specify the fluid properties at each velocity level.

Tetrafluoroethane (R134a) is the working fluid of the compressor selected for the present study, which is a common choice for household refrigeration systems. Therefore, after the hot-wire sensor was calibrated in air, the resulting calibration curves were made dimensionless (Kramers, 1946), by characterizing the fluid flow conditions through Reynolds and Nusselt numbers. Figure 3 shows the dimensionless calibration curve.

As illustrated in Fig. 4, the anemometry sensor was assembled at the entrance of the suction chamber, allowing measurements of velocity and temperature. A small tap hole (1 mm diameter) was used to connect the suction chamber to a larger orifice (7 mm diameter), in which the piezoelectric pressure transducer is mounted.

The first step in the experimental procedure is to submit the compressor to an adequate vacuum condition, in order to remove air, humidity and any other contaminant inside the system. Then, the system receives a charge of refrigerant and the flowmeter reading is set to zero. After the compressor is switched on, a period of approximately 6 hours is needed to establish a fully periodic operating condition because of compressor thermal inertia. When this condition is satisfied, data for energy consumption and mass flow rate are acquired during a period of 10 minutes. Finally, data for temperature, velocity and pressure in the suction chamber and displacement of suction valve are collected for 240 operation cycles of the compressor, allowing the evaluation of an average value for each quantity.

A total of 1000 measurements were acquired to properly characterize the compressor operation cycle. Since the compressor operates at a velocity of 3600 rpm, an acquisition rate of 60 kHz was needed. The 240,000 experimental data collected along 240 cycles were submitted to a statistical analysis, with ensemble average results being expressed as function of the crankshaft angle.

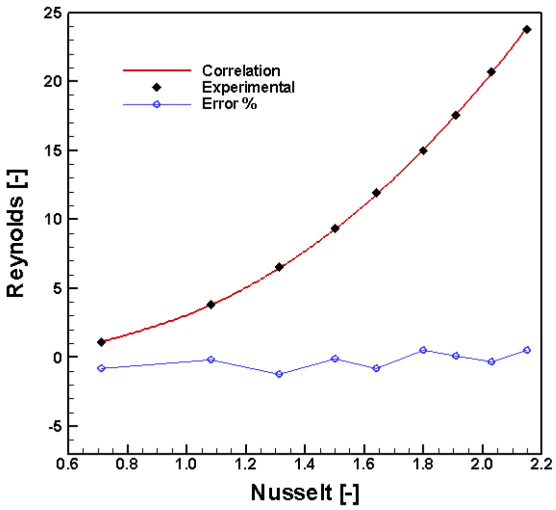

Figure 3. Dimensionless calibration curve.

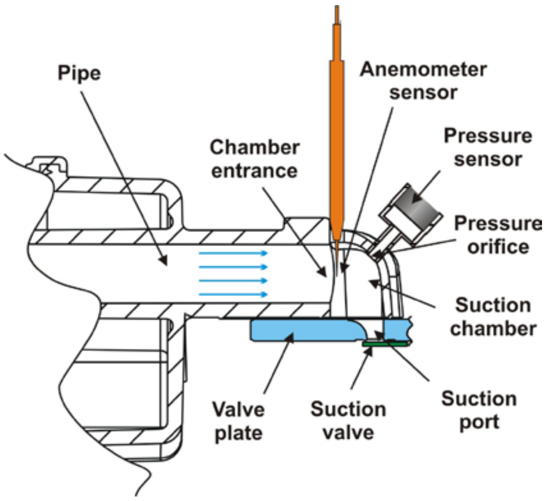

Figure 4. Detail of the anemometry sensor assembly.

3. RESULTS AND DISCUSSION

Figure 5 shows experimental results for pressure, velocity and valve displacement. It is observed that the valve opening occurs at the crankshaft angle of 230°, being followed by a decrease in pressure and an increase of velocity in the suction chamber entrance (Figure 4). The measurements indicate velocity levels of 30 m/s at the crankshaft angle 264°, when the valve reaches its maximum opening. There are also noticeable small fluctuations of velocity even when the valve is closed, probably caused by pressure waves. Naturally, the anemometer probe is insensitive to the flow direction and, therefore, one cannot distinguish between positive and negative.

Figure 6 was prepared to closely analyze pressure and velocity in the suction chamber during the valve opening. It is interesting to notice the subtle changes in the velocity signal inclination between the angles of 249 and 255°, also observed in the raw data for each cycle, which are probably linked to flow perturbations. It is observed a phase delay between the opening of the valve and the beginning of the flow in the duct, probably related to the gas inertia in the suction chamber.

The pressure in the chamber decreases initially because gas is supplied to the suction valve, until it reaches pressure P_1, from which it starts to rise. At the crankshaft angle of 262°, such an increase of pressure causes the flow deceleration and, hence, a reduction of velocity at the suction chamber entrance. After the angle 285°, the pressure in the chamber is observed to decrease, probably due to a lower mass flow rate in the suction entrance compared to the suction port.

As shown in Fig. 6, from point D_2, the resulting force from the pressure field causes the valve to open again, followed by an increase of flow velocity at the suction chamber entrance, as indicates point V_2. From this point on, the pressure drop in the suction chamber promotes again the flow acceleration, in the same way as observed for the valve opening at the angle 230°.

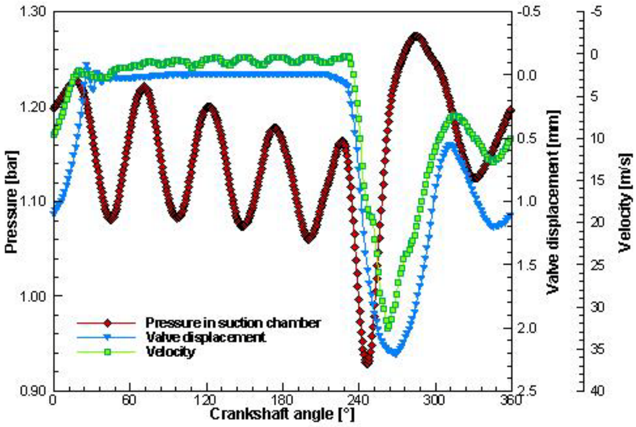

Figure 5. Measurements of velocity, pressure and valve displacement; operating condition -23.3°C/54.4°C.

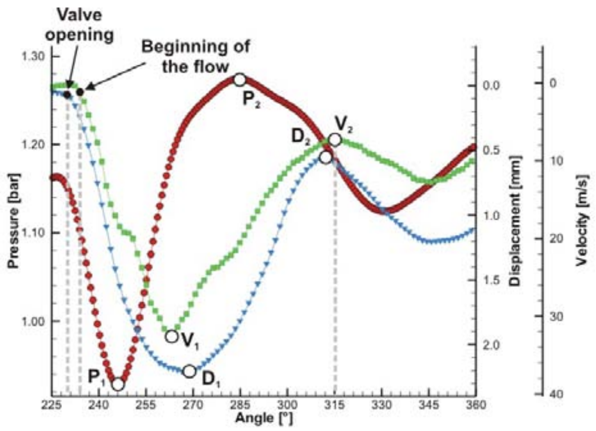

Figure 6. Detail of Figure 5, results for velocity and pressure during the valve opening; operating condition -23.3°C/54.4°C.

After the angle 330°, the suction chamber pressure starts to increase and the velocity at the chamber entrance to decrease, as the valve approaches its closing position. As can be seen from Fig. 5, at the angle 24° the valve finally reaches the closing position.

In Fig. 7 results for valve displacement are shown according to the operating conditions -23.3°C/54.4°C and -35°C/54.4°C. It is observed a difference between phase and amplitude in the curves for both conditions. The phase shift of 19° can be explained by a delay in the valve opening for the condition -35°C/54.4°C.

As Tab. 1 shows, the evaporation temperature of -35°C is associated with a lower evaporation pressure. It is well known that before the suction valve can open and allow vapor to flow into the cylinder, the high-pressure vapor left in the clearance volume must expand until its pressure drops below that of the vapor in the suction line. Thus, since the discharge pressure is the same in both operating conditions, the clearance vapor must expand to a lower pressure in the case of a lower suction pressure. As a consequence, there is a delay in the opening of the suction valve for the operating condition -35°C/54.4°C. For this condition, the angles in which the valve opens and closes are equal to 245 and 45°, respectively.

Table 1. Compressor operating conditions adopted in the experiments.

Operating condition (°C/°C) (1)	-23.3/54.4	-35/54.4
Suction pressure (bar)	1.150	0.656
Discharge pressure (bar)	14.71	14.74
Mass flow (kg/h)	5.07	2.20
Average suction temperature (°C)	51.3	57.0

(1): Evaporating/Condensing temperatures

The maximum valve displacement corresponds to 1.45 mm at 286°, with the valve being almost closed at the angle 340°. One explanation for this may be a simple comparison between the mass flow rate, which is lower in this condition.

Regarding pressure pulsation in the suction chamber, Fig. 8 also shows a phase shift between the curves for both operating conditions. The maximum pressure level in the second operating condition is 0.72 bar at 304°, and the minimum level is 0.53 bar at 264°. Measurements of velocity for both conditions, shown in Fig. 9, indicate that the conditions of maximum and minimum for the second condition are lower than those for the first condition, remaining with levels near 3 m/s during the second oscillation of the valve displacement. Results for pressure, velocity and valve displacement considering the second condition are displayed in Fig. 10, showing the phenomenon is quite similar to that observed for the first condition (Fig. 5).

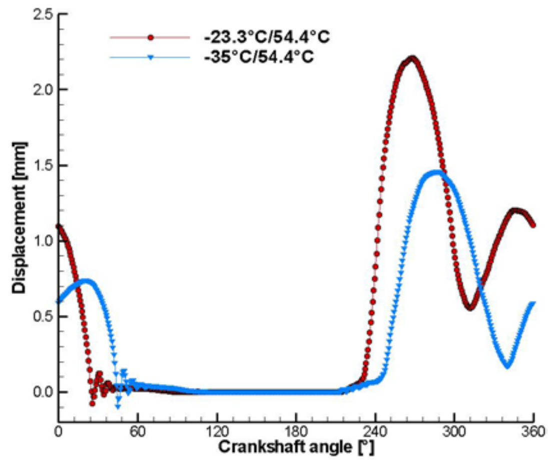

Figure 7. Suction valve displacement for both operating conditions.

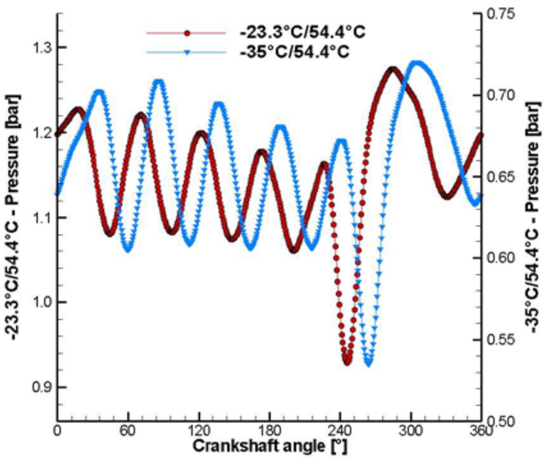

Figure 8. Pressure in the suction chamber for both operating conditions.

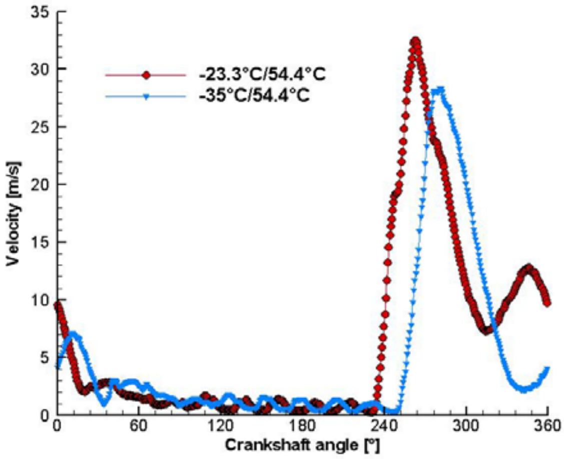

Figure 9. Instantaneous velocity in the suction chamber entrance.

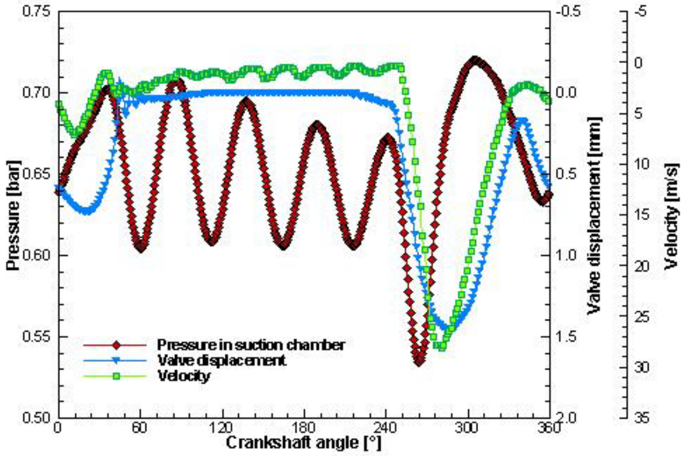

Figure 10. Measurements of velocity, pressure and valve displacement; operating condition -35°C/54.4°C.

4. CONCLUSIONS

An experimental setup was developed to investigate fluid flow transients in the suction chamber of high speed refrigeration compressors. To this extent, a hot-wire system with a 5μm diameter wire probe was adopted for velocity measurements. A calibration procedure was specially elaborated due to the impossibility of using the refrigerant as the working fluid for this purpose. Results for valve displacement and pressure pulsation in the suction chamber were also obtained to complement the fluid flow analysis. Measurements indicated abrupt variations in all flow properties, strongly linked to the valve motion. When the valve is open, velocity reach levels of 32 m/s, in comparison with 3 m/s when it is closed. In the former period, the fluid flow in the entrance of the suction chamber is related to the pressure fluctuations generated due the expansion and compression waves in the duct. It has been observed that the fluid and valve inertia are significant during the first opening of the valve resulting in a phase delay with the pressure fluctuation. Although there is a phase delay between the operating conditions tested in the present study, measurements indicate very similar phenomena.

5. ACKNOWLEDGEMENTS

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