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Experimental Study on the Thermal Behavior of a Domestic Refrigeration Compressor during Transient Operation in a Small Capacity Cooling System

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ABSTRACT

Generally, domestic refrigerators and freezers are running in non-continuous operation mode most of the time, which is a necessity to match cooling capacity to thermal loads. In currently available domestic appliances it can be observed, that this matching is mainly realized in two different ways: On the one hand, a simple on/off control of the hermetic compressor is installed in lower priced appliances with limited energy efficiency for the mass market. On the other hand, modern top efficiency class appliances have a variable frequency controlled compressor installed. Both control strategies have a repetitive and transient change of thermodynamic states of the refrigerant in common.

For better understanding of these cyclic patterns in terms of internal temperature distribution, a state of the art domestic refrigeration compressor with a displacement of approximately 6 cubic centimeters is integrated in a commercial freezer. The compressor which has an on/off control is equipped with extensive measurement instrumentation. Several temperature probes are inserted and temperatures on surfaces inside and outside the compressor as well as refrigerant temperatures are logged for both cyclic and steady-state behavior. Finally, a comparison between transient experimental data and steady-state data from a standardized calorimeter test bench is done.

1. INTRODUCTION

Since a respectable part of the worldwide energy consumption can be attributed to household refrigerators and freezers, it is a necessity to increase the overall efficiency of cooling systems. These systems consist of several parts whereat the compressor plays the key role concerning the energy consumption. The energy consumption of hermetic compressors is mainly influenced by three aspects: the electrical, the mechanical and the thermodynamic efficiency. The electrical efficiency of the motor is usually about 90%, the mechanical efficiency, which is essentially affected by the bearing system, is also mostly around 90% and the thermodynamic efficiency is usually about 80 to 83% (Ribas *et al.*, 2008). Therefore, the thermodynamic analysis of reciprocating compressors for domestic refrigerators was the main focus of several different studies which were published in the last few years. Most of them were based on a quasi-static behavior of the hermetic compressor under certain standardized conditions.

Kara and Oguz (2010) investigated the thermal behavior of a small hermetic reciprocating compressor. They measured the gas and surface temperature on several positions while the compressor was tested in a calorimeter that enabled stabilized operating conditions (ASHRAE conditions) and compared the experiments with results by numerical models. Another work, wherein a thermal analysis of a hermetic reciprocating compressor was made, was published by Cavallini *et al.* (1996). The authors investigated several computational models and compared them with experimental data. These experimental measurements had been carried out on a hermetic compressor which also was operating under steady conditions (ASHRAE conditions). In the field of thermal analysis of reciprocating compressors, transient conditions represent very often only the unsteady phenomena during one revolution as in the study of Longo and Caracciolo (2002). Similar investigations were done by Negrao *et al.* (2010).

To investigate the complex thermal phenomena inside reciprocating compressors, experiments and numerical calculations are almost always carried out at stabilized operating conditions with standardized boundary conditions. These conditions can differ from the real world operating conditions especially for non-variable speed compressors. If a compressor should be improved in order to increase the overall efficiency and therefore, to reduce the annual energy consumption of the entire cooling system, measuring data from a calorimeter test bench could lead to misleading potentials for improvements. Therefore, this work compares steady-state and transient temperature measurements of a fixed speed compressor (50 Hz) operating at different conditions.

2. EXPERIMENTAL SETUP

Experimental measurements are carried out with a commercial hermetic reciprocating compressor for domestic refrigerators with a displacement of approximately 6 cubic centimeters. The refrigerant flows through the suction pipe into the shell of the compressor. There it enters the suction muffler and flows through a reed valve into the cylinder. The compressed refrigerant is then discharged and leaves the shell, passing the cylinder head, discharge mufflers and the serpentine.

The investigated compressor, running with R600a, is equipped with extensive measurement instrumentation. In order to measure the temperature profile during a duty cycle, which is in a range of several minutes, T-type thermocouples are used. Thirteen thermocouples are distributed on several positions inside the compressor to get a meaningful statement of the thermal behavior. The description of the locations of these thermocouples is given in Table 1. Additionally, the temperatures of the suction pipe (T14) and discharge pipe (T15) outside the compressor are measured. To get the outer surface temperature distribution of the compressor, another six thermocouples are placed around the shell. These measuring positions are described in Table 2. Furthermore, two pressure transducers are placed at the cylinder head (P1) and the suction muffler neck (P2). Figure 1 shows drawings of the compressor, wherein the temperature and pressure measuring positions are depicted.

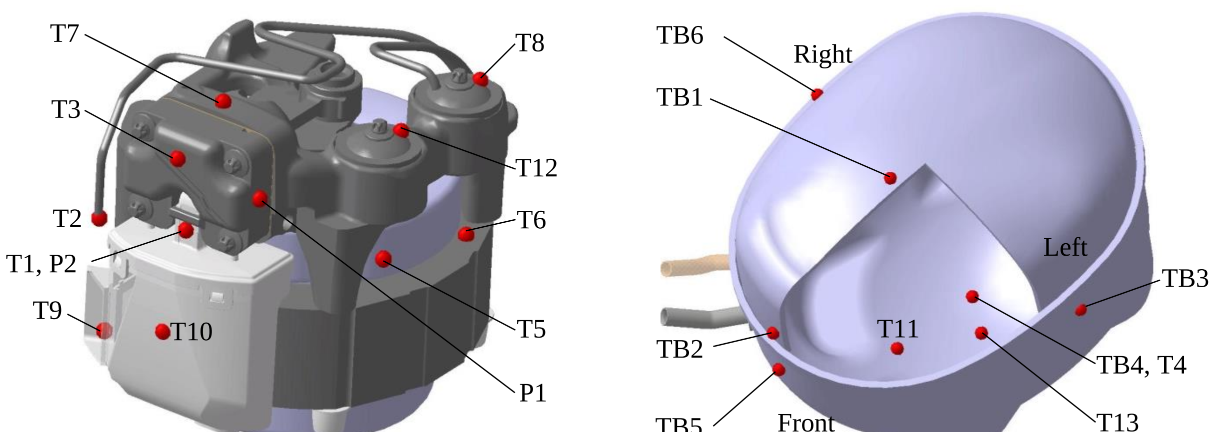

Figure 1: Temperature and pressure measuring positions

Due to the requirement of hermetical tightness of the compressor, every cable-feedthrough represents a potential vulnerability. Therefore, the Teflon®-insulated sensor wires are passed through the shell with a specially sealed feedthrough. The left side of Figure 2 shows the compressor without the upper part of the shell. One of the challenges is to arrange all the measurement instrumentations in a way that the influence on the behavior of the compressor is as small as possible. Thermocouples which measure a surface temperature inside the shell are fixed with epoxy resin which can be seen in Figure 2 (right).

Table 1: Thermocouple measuring positions at the compressor

No.	Temperature measuring position description	Location
T1	Suction muffler neck (gas)	-
T2	Serpentine end (surface)	-
T3	Cylinder head outside (surface)	Front
T4	Shell inside (surface)	Bottom
T5	Stator windings (surface)	Top
T6	Stator lamination (surface)	Top
T7	Cylinder (surface)	Top
T8	Discharge muffler-2 (surface)	Top
T9	Suction muffler intake tube (gas)	-
T10	Suction muffler wall (surface)	Centre
T11	Oil	Centre
T12	Discharge muffler-1 (surface)	Top
T13	Shell (gas)	-

Table 2: Thermocouple measuring positions at the shell (outside)

No.	Temperature measuring position description	Location
TB1	Shell cover (surface)	Middle
TB2	Shell cover (surface)	Front
TB3	Shell (surface)	Left
TB4	Shell (surface)	Bottom
TB5	Shell (surface)	Front
TB6	Shell cover (surface)	Right

To get steady state-data, the compressor is running on a fully automated calorimeter test bench under four different stabilized operating conditions. Thereby, ambient temperature is set to 32 °C and the adjusted evaporation and condensation temperature can be seen in Table 3. Each test takes several hours, whereat only the data which is recorded during the last 50 minutes is analyzed to ensure steady-state conditions.

The transient measurements are done with the same compressor installed in a commercial freezer with a net capacity of approximately 100 liters. Several different measurements are carried out such as varying the ambient temperature, varying the cycle duration or varying the duty-cycle ratio. Table 4 describes the operating conditions of the respective transient measurements whereat evaporation and condensation temperatures refer to the ON-period of the compressor.

Table 3: Evaporation and condensation temperature of calorimeter measurements

Measurement	Evaporation temperature T_{eva} [°C]	Condensation temperature T_{con} [°C]
Calo -23/45	-23.3	45.0
Calo -23/55	-23.3	55.0
Calo -37/49	-37.0	49.0
Calo -34/50	-34.0	50.0

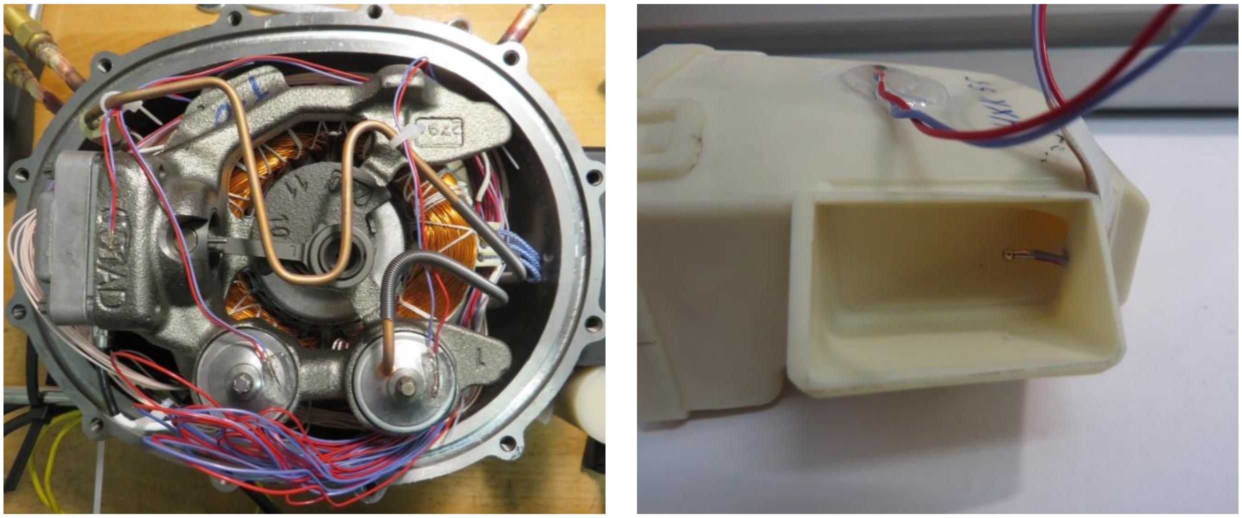

Figure 2: Compressor without upper shell (left), suction muffler details (right)

Table 4: Operating conditions of transient measurements

Measurement **(T_{amb}/t_{cyc}/r)**	**Ambient temperature** **T_{amb}** **[°C]**	**Cycle duration** **t_{cyc}** **[min]**	**Duty-cycle ratio** **r** **[%]**	**Evaporation temperature** **T_{eva}** **[°C]**	**Condensation temperature** **T_{con}** **[°C]**
Real 32/30/67	32	30	67	-33.8	50.0
Real 32/45/67	32	45	67	-34.9	50.0
Real 32/105/80	32	105	80	-37.2	49.1
Real 25/30/65	25	30	65	-31.2	41.2

3. RESULTS AND DISCUSSION

In the following chapter, the experimental data is presented and discussed. All measurements are carried out with the same hermetic compressor which is either running on a calorimeter test bench or in a commercial household freezer. Several operating conditions of the compressor are tested and effects on temperature distribution are investigated. Finally, a comparison between steady-state and transient operating behavior is done.

3.1. Steady-state data of the calorimeter test bench

Figure 3 shows a chart where the arithmetic mean values of the measured temperatures can be seen. Considering the outer surface temperatures of the shell (TB1 – TB6), it can be observed that the left (TB3) and the right side of the shell (TB6) show the highest temperatures. The lowest temperatures of the shell occur at the middle of the shell cover (TB1). One would expect that the temperatures of the three mentioned positions are more similar and higher than the other ones (TB2, TB4 and TB5) because they are close to the discharge line which has the highest temperature level inside the compressor. One explanation for this phenomenon could be the influence of the air flow inside the calorimeter test bench on the heat transfer between compressor shell and ambient air. When comparing the calorimeter-datasets among each other, it should be kept in mind that different operating conditions lead to different mass flow rates.

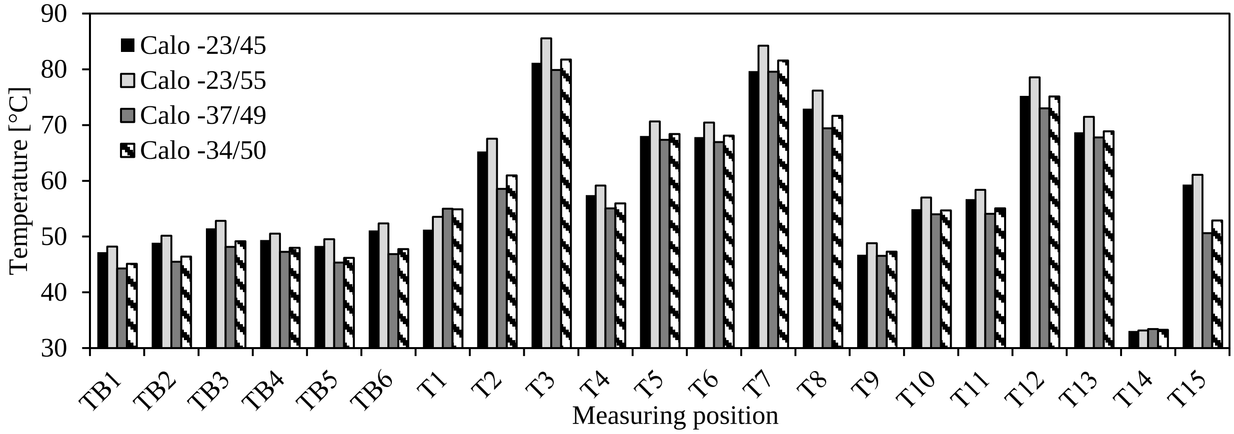

Figure 3: Arithmetic mean values of temperatures at different measuring positions - calorimeter data

The highest temperatures occur along the discharge line. From the cylinder head (T3) to the second discharge muffler (T8) a temperature drop of approximately 10 K can be observed for all four operating conditions. Along the serpentine, the temperature drops for another 8-10 K. The temperature of the gas inside the shell (T13) and temperatures of the electric machine (T5 and T6) are at similar levels, leading to the assumption that measuring position T13 is influenced by the motor. Due to the limited space inside the compressor, it is hardly possible to find a position, where influences from other parts could be fully excluded. The heating of the suction gas (T9, T1) is about 4.5 K for the measurements "Calo -23/45" and "Calo -23/55", and approximately 8 K for the other two operating conditions. This discrepancy results from the different mass flow rates.

3.2 Transient experimental data

To gain transient experimental data, the hermetic compressor is installed in a commercial freezer with a net capacity of approximately 100 liters. Thereby, cycle duration and duty-cycle ratio are varied. In this work, the duty-cycle ratio is defined as the duration of the ON-cycle divided by the total cycle duration. Additional to the three measurements at ambient temperature of 32 °C, another measurement at 25 °C is done to investigate the influence of different ambient temperatures.

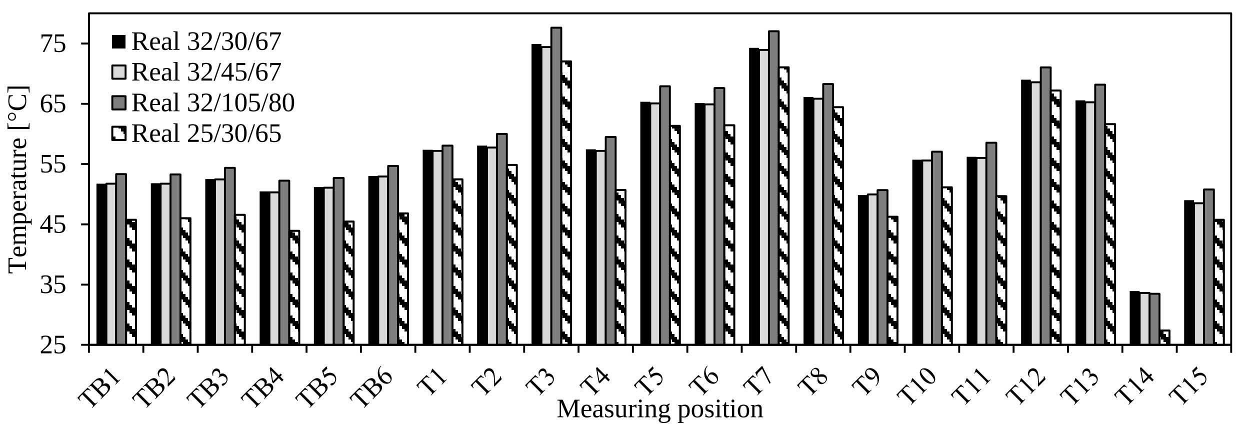

Figure 4: Arithmetic mean values of temperatures at different measuring positions (transient data, total cycle duration)

Figure 4 shows measured temperatures which are averaged over the total cycle duration. A closer look at the outer surface temperatures of the shell (TB1-TB6) shows that the lowest temperature occurs at the bottom (TB4) and the highest at the right and the left side of the shell (TB3 and TB6). This temperature distribution indicates the influence of the hot and cold compressor components on the surface shell temperature more pronounced than those determined by the steady-state measurements on the calorimeter test bench.

Similar to the calorimeter measurements, it can be seen, that the temperature drops approximately 8.5 K between the cylinder head (T3) and the second discharge muffler (T8) and another 8 K along the serpentine. The suction gas temperature increases by 7.5 K from the intake tube to the neck of the suction muffler for measurements at an ambient temperature of 32 °C and increases by 6.5 K for the measurement at 25 °C ambient temperature. Comparing the real-world measurements among each other, rather small differences can be seen when varying the cycle duration (Real 32/30/67 and Real 32/45/67). A similar thermal behavior but at higher temperature level (1.5-2.5 K higher) can be found when increasing the duty-cycle ratio up to 80 percent (Real 32/105/80). Due to the control logic of the commercial freezer, the total cycle duration has also increased. Decreasing the ambient temperature from 32 °C to 25 °C leads to an approximately 6 K lower shell surface temperature. However, the temperature level reduction of the discharge line, the suction gas and the electrical machine is only about 1.7-3 K, 4.5 K and 4 K, respectively.

Figure 5 illustrates differences between temperatures averaged over the ON-period, OFF-period and total cycle, on the basis of the transient measurement "Real 32/30/67". Concerning the outer surface temperature of the shell, it can be noted, that the deviation between ON- and OFF-period is smaller than 1 K except TB4 (shell bottom) which seems to correlate stronger with the oil temperature (T11). Inside the compressor, the same small temperature deviation can be observed for components with a high thermal mass such as the electric machine and the two discharge mufflers. Temperatures at measuring positions T3 and T7 (cylinder head and cylinder) show a significant deviation between ON- and OFF-period because they are directly affected by the heat generation resulting from the compression of the refrigerant. The temperature drop of the serpentine (T2) is similar to that of T3 and T7. Further considerable differences appear at measuring positions T1, T9 and T10 (suction muffler neck, suction muffler intake tube and suction muffler wall) which show the heating of the suction line during the OFF-period.

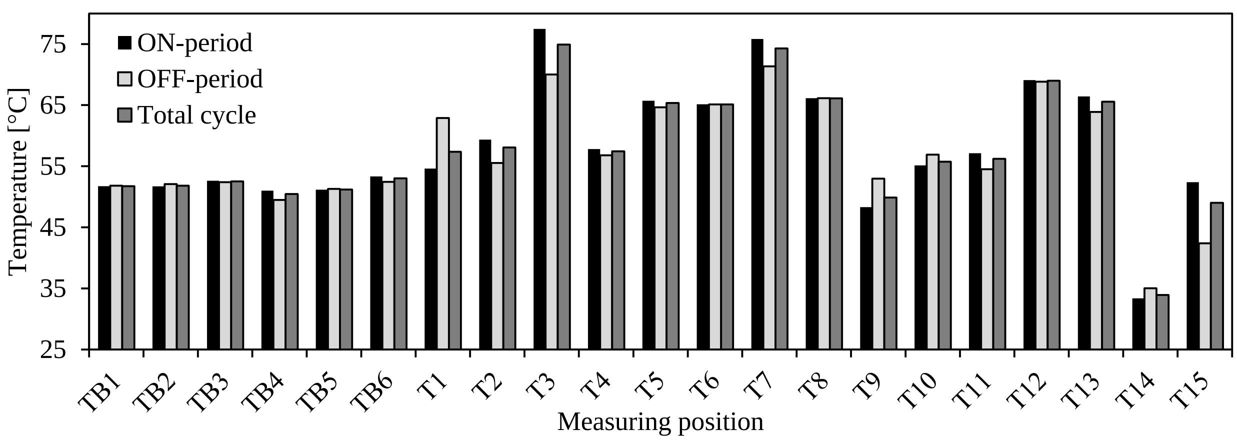

Figure 5: Averaged temperatures of transient measurement “Real 32/30/67” – comparison between ON- and OFF-period

3.3. Comparison between steady-state and transient data

In this section, a comparison between steady-state and transient measurement data is done for two different operating conditions. The adjusted evaporating and condensing pressure is 0.386 bar and 7.02 bar for the measurements "Real 32/30/67" and "Calo -34/50", and 0.386 bar and 6.68 bar for the measurements "Real 32/105/80" and "Calo -37/49", respectively. Figure 6 shows the time averaged temperatures of the measurements mentioned before. Thereby, the temperatures for the transient measurements are averaged over the entire cycle duration. Comparing the outer surface temperatures of the shell (TB1-TB6), a difference of approximately 5-7 K can be observed for both operating conditions. Based on the assumption of a similar input power to the compressor for transient and steady-state measurements, nearly the same or even a lower temperature level of the transient measurements could have been expected. The main reason for this discrepancy is the air flow over the compressor in the calorimeter test bench, which is needed to keep the ambient temperature constant at the desired level. During the transient measurements, the compressor is located in a small recess at the backside of the freezer. There, the air temperature close to the compressor is higher than the ambient temperature and heat transfer only occurs by natural convection and heat radiation.

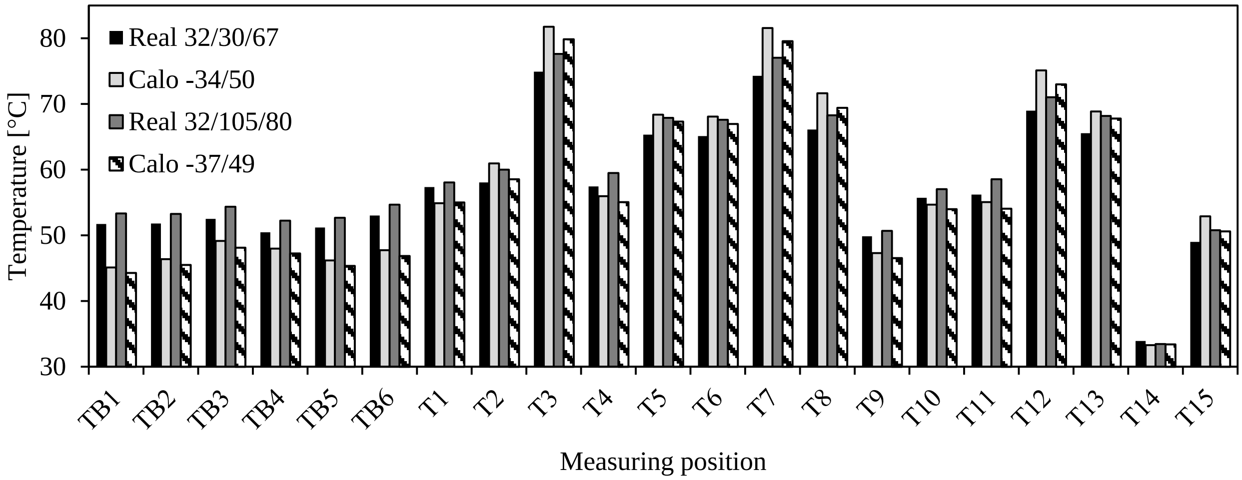

Figure 6: Comparison between transient and steady-state temperature measurements

Inside the compressor, higher temperatures along the discharge line (T3, T7, T8, T12) can be observed for steady-state measurements compared to transient ones, which is expected due to the OFF-period of the compressor. Similar differences can be seen for temperatures influenced by the cold suction gas. T1, T9, and T10 show values which are significant lower for steady-state measurements. Worth mentioning is that the average temperature level at operating condition 0.386 bar/7.02 bar (Real 32/30/67) is lower when comparing both transient measurements, whereas it is higher (Calo -34/50) when comparing both steady-state measurements.

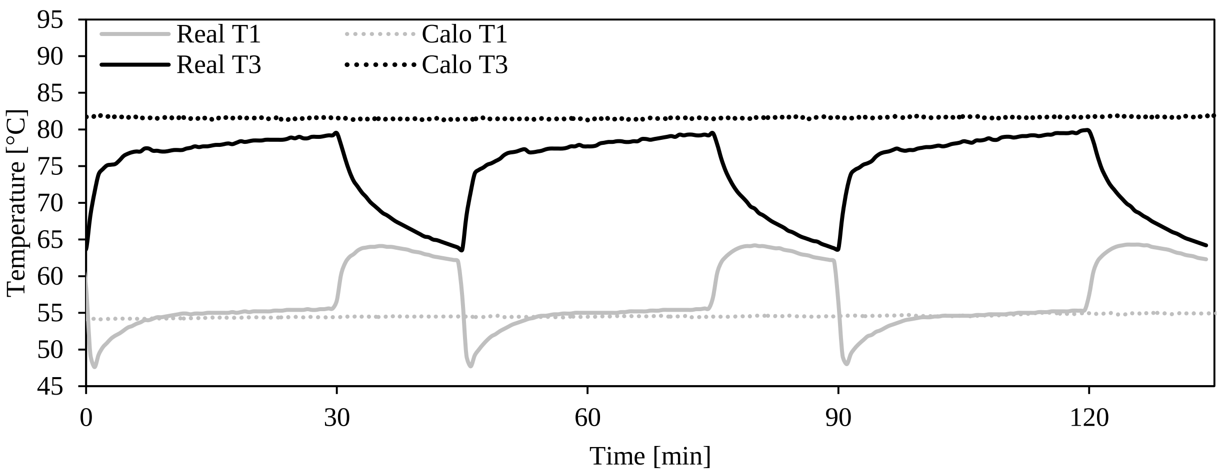

Figure 7: Temperature profile of measuring positions T1 and T3 – transient versus steady-state operating conditions

Figure 7 shows the profile of the gas temperature at the suction muffler neck (T1) and the surface temperature at the cylinder head (T3) of the transient (Real 32/30/67) and the steady-state (Calo -34/50) measurement. Therein it can be seen, that the deviation between steady-state and transient temperature decreases during the ON-period of the compressor and reaches a value of approximately 1-2 K shortly before the compressor stops. The deviations of the time averaged temperatures are of course much more significant which can be seen in Figure 6. When the compressor is switched on, temperature T1 decreases very fast due to the cold refrigerant which is sucked in. Afterwards, the valve plate is heated and influences the gas at the muffler neck which can be seen in the rather fast increase of T1 until a state of equilibrium is reached. When the compressor stops and, therefore, the refrigerant mass flow is zero, the remaining gas in the neck is heated very fast and then cools down together with the surrounding solid parts.

4. CONCLUSIONS

This work deals with the investigation of the thermal behavior of a state-of-the-art compressor used in domestic refrigerators during steady-state and transient operating conditions. Due to the difficulty of sensor-signal feedthrough during operation of hermetic compressors, this investigation is not a standardized measurement and provides valuable temperature data. To gain measuring data, the non-variable speed compressor was operated in a fully automated calorimeter test bench and in a commercial freezer. The temperature was measured at nineteen measuring positions distributed around the shell and inside the compressor. Influences of the duty-cycle ratio, the cycle duration or the ambient temperature on the temperature level of the compressor were investigated at transient operating conditions. Furthermore, calorimeter measurements at four different stabilized operating conditions were compared. Finally the data of transient and steady-state measurements at two different operating conditions were compared. Based on these results the following major conclusions can be drawn:

- The temperature drop along the discharge line (without serpentine) is in the same range for all eight measurements and is almost independent of the refrigerant mass flow rate. On the contrary, the influence of different operating conditions on the heating of the suction gas and the cooling along the serpentine is more significant.
- The influences of the total cycle duration on the temperature levels are rather small. An increase of the duty-cycle ratio results in a similar thermal behavior but generally at a higher temperature level, whereas a variation of the ambient temperature results also in a slightly different thermal behavior.
- The comparison between transient and steady-state measurements shows, that the temperature level differs significantly due to different environmental conditions. Inside the calorimeter test bench, a forced air circulation keeps the exact ambient temperature, whereas inside the recess at the backside of the freezer, where the compressor is located, only buoyancy driven flow occurs. This leads to lower heat transfer coefficients and a higher air temperature close to the compressor.

REFERENCES

Cavallini, A., Doretti, L., Longo, G. A., Rossetto, L., Bella, B., Zannerio, A. (1996). Thermal analysis of a hermetic reciprocating compressor, Proceedings of the International Compressor Engineering Conference at Purdue, West Lafayette, USA, 535-540.

Kara, S., Oguz, E. (2010). Thermal analysis of a small hermetic reciprocating compressor, *Proceedings of the International Compressor Engineering Conference at Purdue*, West Lafayette, USA, paper 1307.

Longo, G. A., Caracciolo, R. (2002). Unsteady state analysis of a hermetic reciprocating compressor: heat transfer inside the cylinder and valve dynamics, Proceedings of the International Compressor Engineering Conference at Purdue, West Lafayette, USA, paper C4-4.

Negrao, C. O. R., Erthal, R. H., Andrade, D. E. V., Silva, L. W. (2010). An algebraic model for transient simulation of reciprocating compressors, Proceedings of the International Compressor Engineering Conference at Purdue, West Lafayette, USA, paper 1486.

Ribas, F. A., Deschamps, C. J., Fagotti, F., Morriesen, A., Dutra, T. (2008). Thermal analysis of reciprocating compressors - a critical review, *Proceedings of the International Compressor Engineering Conference at Purdue*, West Lafayette, USA, paper 1306.

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