Design And Realization Of An Automated Test-Stand For Variable Capacity Household Compressors

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ABSTRACT

In Switzerland the efficiency of household refrigerators has to be proven by the manufacturer to get the European energy label for the devices. The efficiency is tested in the beginning of the lifetime of the fridge. However, the compressor might reach its best performance only after some hours of operation. Knowing precisely the compressor characteristics is necessary for the manufacturer to optimize the products. Therefore, an automated compressor test-stand was designed and built to test variable speed compressors on their evolution of isentropic and volumetric efficiencies over time.

The test-stand features three expansion valve sections to control the high- and low-pressure states, and the suction gas temperature of the compressor. The test setup was programmed to be automatically controlled. Additional heaters can balance the non-isenthalpic pressure drop through the different components (mass flow meter and expansion valves) and the large internal heat exchange of the compressor enabling high enough suction gas temperatures. Furthermore, the compressor is installed in a temperature-controlled chamber. The power consumption of the compressor is measured with and without the inverter to get additional information. The testing range is 20 to 200 W cooling capacity with pressures of the refrigerant R600a down to its corresponding evaporation temperature (-35°C) and up to its condensing temperature (55°C). The test-stand is equipped with high precision sensors to ensure uncertainties of volumetric and isentropic efficiencies smaller than 5%. This paper reports the design of the test-stand, the control of the system, as well as the first results.

1. INTRODUCTION AND MOTIVATION

The EU directive 1060/2010 (European Commission, 2010) demands that all energy-related products are labeled with an indication of, among others, their energy consumption. Domestic household refrigerators and freezers were the first products that have been subjected to this directive.
The energy label initially classified the energy efficiency from class G (low efficient) to class A (highly efficient). Various improvements of components and materials gradually led to additional efficiency classes currently standing at A+++ for the highest efficiency. Nowadays, Swiss regulation requires that every new fridge sold in Switzerland have at least an energy consumption that corresponds to the class A++ (EnV, 1998). Consequently, the sales figures of high efficient domestic household refrigerators were steadily increased while low efficient products phased out in the past years, as shown in Figure 1 (FEA, 2015).

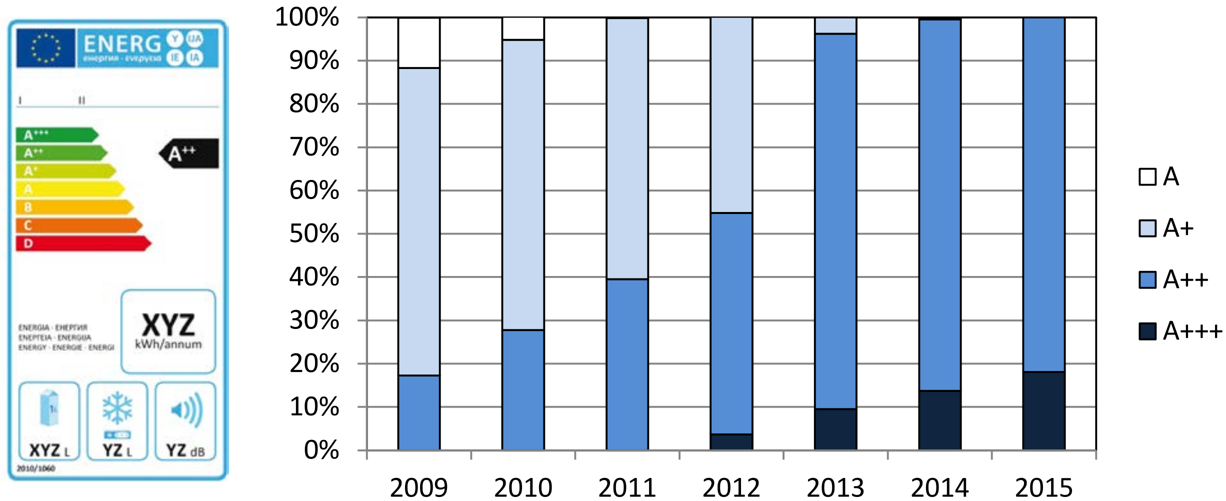


Figure 1: Energy label and development of sales figures of household refrigerators with different energy labels.

For example, an A+++, double-door, two-compartment household refrigerator with storage volumes of 203 liters (fridge: fresh food) and 73 liters (freezer: frozen food) must not consume more than 155 kWh/year (climate class tropical) at 5°C/-18°C (fridge/freezer) storage temperatures. This results in an average power consumption of 17.7 W. In order to fulfill this requirement, this specific appliance combines many system improvements stated by Bansal *et al.* (2011) as:

- advanced insulation by implementing vacuum insulation panels (VIP),
- isobutane (R600a) as refrigerant, and a
- variable speed compressor

The transition of existing, integrated household refrigerators to a better efficiency class is challenging since the outer dimensions are given. Consequently, increasing the insulation thickness directly reduces the storage volume which ultimately lowers the allowed energy demand to obtain the label. Therefore, this is not a preferred approach.

One key component of the appliance is the variable speed compressor. In order to increase the quality and efficiency of existing household refrigerators and to get additional design information, a compressor test-stand was designed and realized. Information about the transient evolution of volumetric and isentropic efficiencies during the first few hundred hours of operation, optimal conditions for high efficiencies, as well as inverter performances are key results obtained by the test-stand.

Compressor test-stands are commonly built to map and test compressors during development. This will also be possible in the presented test-stand. However, the main focus is to monitor the transient behavior of the efficiency over long periods of time and to analyze the effect of various parameters such as the suction gas temperature.

2. EXPERIMENTAL SET-UP

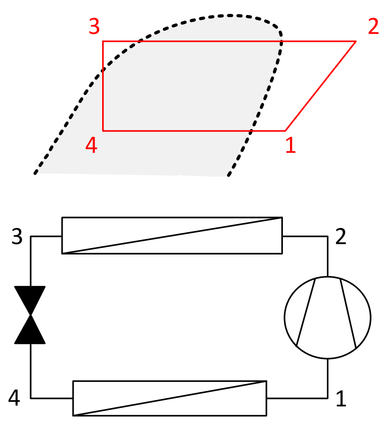
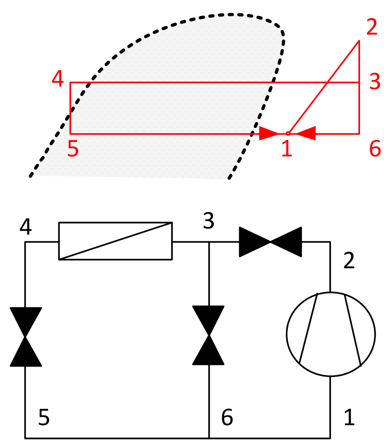
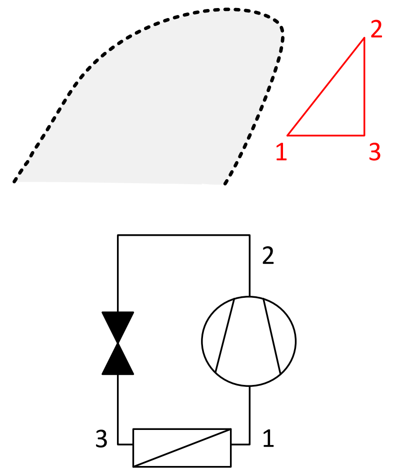
2.1 Test-stand design

The main specifications for the compressor test-stand are the following:

- Testing of compressors with cooling capacity ranging from 20 to 200 W
- Monitoring isentropic and volumetric efficiencies with uncertainty below 5%
- Testing conditions according to EN 12900 Household/CECOMAF
    - Ambient temperature 32°C
    - Suction gas temperature 32°C
- Condensing temperatures from 35 to 55°C
- Evaporation temperatures from -35 to -10°C

In order to fulfill these requirements three different options to design the compressor test-stand were considered. These concepts are schematically presented in Table 1.

Table 1: Different approaches of compressor test-stands with simplified schematics

Calorimetric test-stand	Hot gas bypass test-stand	Hot gas bypass
		
Duggan *et al.* (1988)	Christen *et al.* (2006)	Zingerli and Ehrbar (2000)

The hot gas bypass test-stand (Christen *et al.*, 2006) was chosen since this concept features advantages considering versatility, controllable range of the suction gas temperature, refrigerant inventory, and low thermal inertia.
The first household compressor under test showed a significant internal desuperheating of the discharged gas. At some operating conditions, the refrigerant reached a two-phase state at the compressor outlet, as indicated in Figure 2. In addition, an external source of heating energy became necessary due to heat losses in the expansion processes through the valve sections 2-3, 3-6, and the Coriolis type mass flow meter (Figure 3).

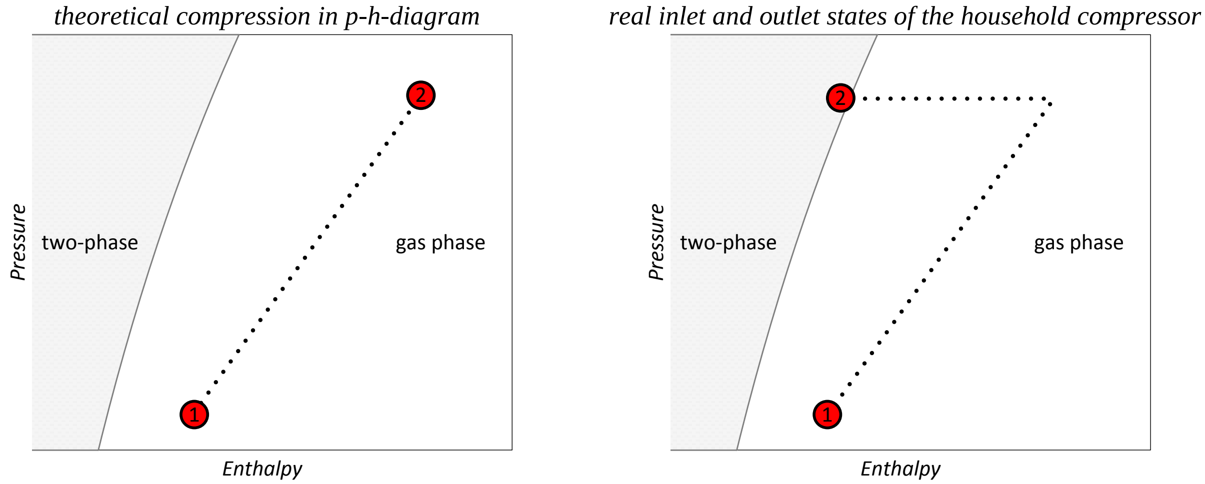


Figure 2: Compression only compared to real inlet (1) and outlet (2) states of a household compressor.

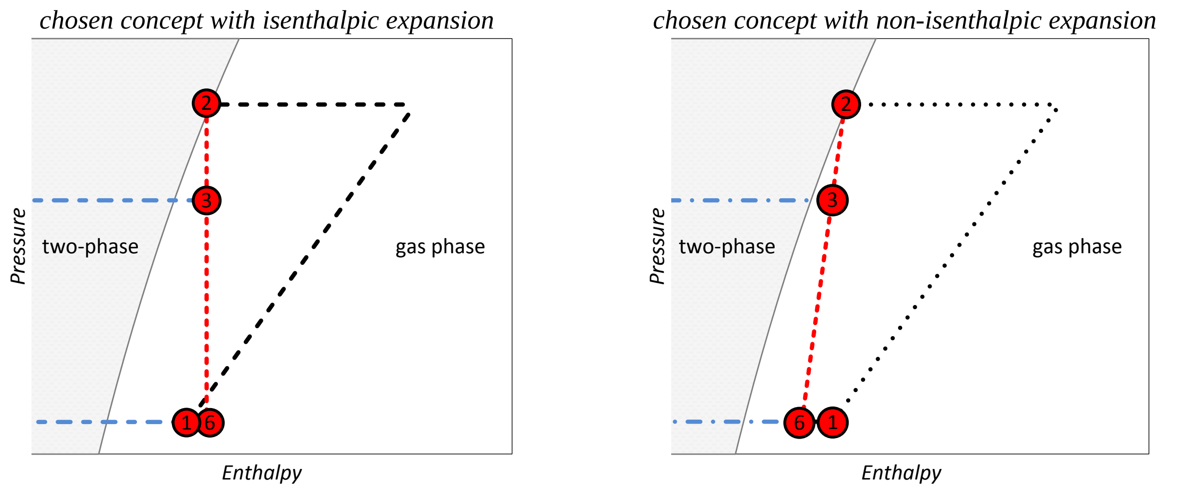


Figure 3: Influence of non-isenthalpic expansion through valves on the concept

Figure 4 shows the schematic of the final test-stand. The mass flow sensor measuring the refrigerant flow rate of the compressor was positioned before the split in the refrigerant line in point 3. The mass flow meter was originally giving inconsistent results due to an inhomogeneous flow at the sensor inlet which was caused by the two-phase state of the refrigerant. The refrigerant was in this state because the compressor has a large internal desuperheating capacity. Furthermore, due to the combination of the desuperheating and the non-isenthalpic expansion, the temperature at point 6 was lower than the target suction temperature. This made the implementation of heating stages necessary. As can be seen in the schematic, heating stages were integrated at the compressor outlet and upstream of the expansion valve (ExpV2-section). The heaters have a maximum heating capacity of 100 W each. Additionally, the high temperature section has been insulated and the mass flow meter itself was equipped with a 15 W electric heater. These modifications were necessary to ensure proper measurement of the mass flow rate. The heating stages 1 and 2 were realized using cartridge heaters that were clamped onto the refrigerant tube within the available space. In order to prevent the decomposition of the oil the maximum clamp-on-element temperature was limited to 105°C.

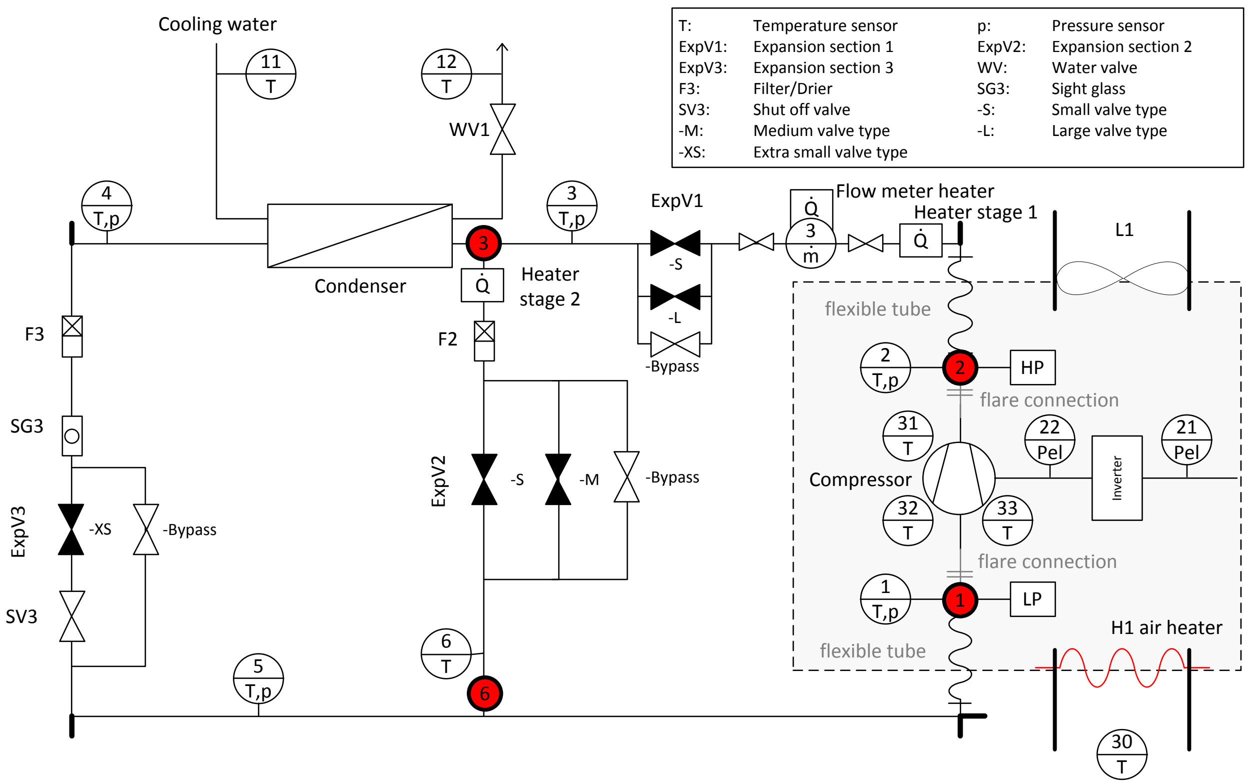


Figure 4: Schematic of the compressor test-stand

2.2 Operation modes and key components

The compressor test-stand can operate in two different modes: the gas cycle mode or the split stream mode. For low cooling capacities, with refrigerant mass flows below 0.3 g/s, and suction gas temperatures at 32°C the flow through the ExpV3-section is about 1/10 of the total mass flow, when operating in split stream mode. Even the smallest electronic expansion valve found in the actual market is too large and therefore not able to reliably control the mass flow. This led to severe fluctuation of the suction gas temperature. Hence, the condensing section is shut off and the suction gas temperature is controlled using the electric heater in stage 2 for small refrigerant flow rates.

The compressor is installed in an insulated compartment. Ambient air is drawn into the compartment by a fan, where its temperature is being controlled using two electric air heaters with 200 W capacity, each. The temperature of the air surrounding the compressor shell is calculated using the average of three temperature probes. To enable different compressor types to be easily placed in the compartment, the compressor is connected to flexible tubes using flare connections. The discharged gas passes the first heater stage and the mass flow is measured by the mass flow meter. Then, the refrigerant passes the ExpV1-section that controls the high pressure at the compressor outlet, since the pressure after the ExpV1-section is anchored by the condensing temperature. The secondary fluid in the condenser is tap water. The refrigerant flow may split up (depending on the operation mode) and one fraction is passing the condenser and expanded from liquid state to two-phase state by the ExpV3-section that controls the suction gas temperature in split stream mode. The rest of refrigerant is heated again by heater stage 2 and throttled by passing the ExpV2-section that controls the suction pressure of the compressor.

2.3 Controls

In order to meet the capacity of the compressor each expansion section features an array of expansion valves with different sizes. Each section may use the valves individually or any combination thereof to control the system. The KV-values of all expansion valves are summarized in Table 2 including controller settings calculated according to Chien, Hrones, Reswick (Haager, 2008).

Table 2: Control devices and settings

			Controller settings		
section	**control device**	**size**	**P-gain**	**I-gain**	**D-gain**
ExpV1	S L Needle valve (bypass)	KV: 0.009 m^3/h, at OD 100% KV: 0.024 m^3/h, at OD 100% KV: 0.33 m^3/h, at OD 100%	-0.0001	0.05	-
ExpV2	S M Needle valve (bypass)	KV: 0.009 m^3/h, at OD 100% KV: 0.014 m^3/h, at OD 100% KV: 0.08 m^3/h, at OD 100%	0.09	0.5	-
ExpV3	XS	KV: 0.003 m^3/h, at OD 100%	-0.0005	0.009	-
Heater stage 2	Cartridge heater	100 W	3.3	80	-
Air heater	Finned heating coil	2x200 W	7	5	-

The compressor is set to a constant rotational speed according to the test point. As input signal to the inverter a frequency-modulated signal corresponding to the desired rotational speed is generated using an Arduino Uno Board. A conversion uncertainty of 1% was taken into account for uncertainty analysis.

2.4 Instrumentation and measurement

In order to measure the compressor's isentropic and volumetric efficiency under 5%, the precisions of the probes are crucial.

Table 3: Measurement device uncertainty

Measurement point	**Measurement device**	**Absolute uncertainty**	**Relative uncertainty**
T_1	PT100, 1/3 DIN	0.1 K	0.0017 K/°C·|T|
T_2	PT100, 1/3 DIN	0.1 K	0.0017 K/°C·|T|
p_1	Piezo resistive pressure transducer	0.0015·300 kPa	
p_2	Piezo resistive pressure transducer	0.0015·1000 kPa	
$\dot{m}_3$	Coriolis mass flow meter		0.5%
$P_{el,21}$	Power Analyzer	0.364 W	(0.2+0.05/PF·f)%
n	Conversion of rotational speed (DAQ-System – Arduino Uno – Inverter)		1%

As can be seen from Figure 5 the temperature probes for ambient temperature measurement (T_{31}, T_{32}, T_{33}) are inserted in flexible coolant pipes to enable the individual placement. This is necessary because the outer dimensions of the different compressor types do vary significantly. At the compressor's suction and discharge line the temperature measurement is taken using an in-tube PT100-temperature sensor and the pressure is measured close to that location. Figure 5 shows the flare connections as well as the temperature and pressure measurements at the discharge line with insulation removed.

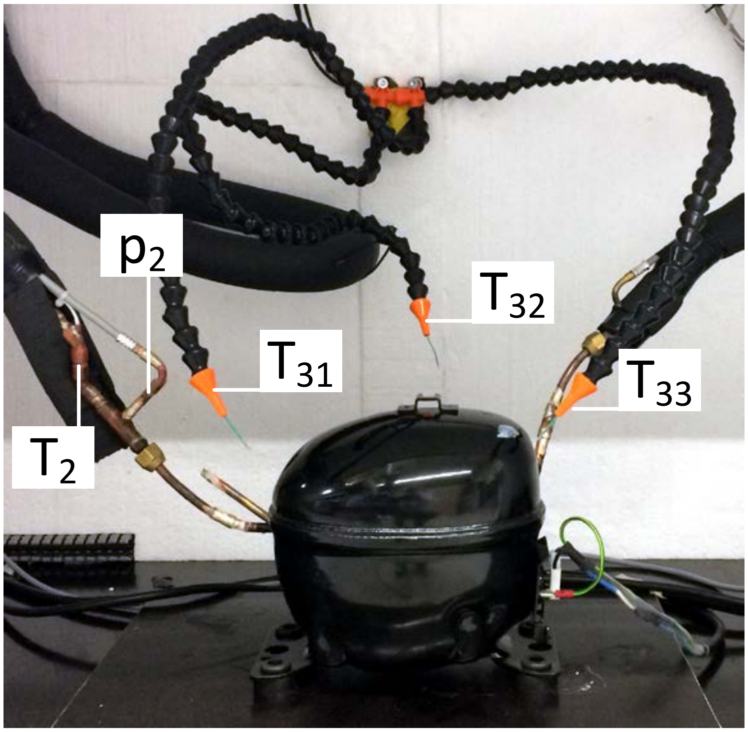


Figure 5: Compressor and sensors in insulated test compartment

In addition to the overall power consumption $P_{el,21}$, the power input to the compressor motor only ($P_{el,22}$) was acquired by a three channel power analyzer to rate the efficiency of the inverter.

3. METHODOLOGY AND EXPERIMENTAL RESULTS

3.1 Uncertainty Analysis

Uncertainty analysis was based on the method outlined in Taylor and Kuyatt (1994) applying the law of propagation of uncertainty. The following contributions to the uncertainty were considered:

- Distribution of fluctuations of the measurement values in terms of the one sided 95% confidence interval of the one sided t-distribution.
- Sensor accuracy according to Table 3 and propagation of sensor accuracy through calculated properties.
- Uncertainty of rotational speed setting of 1%.

The following contributions to the uncertainty were neglected:

- Uncertainty of the property calculation routines (Coolprop, Bell, 2011).
- Manufacturing tolerances of the displacement of the compressor.

3.2 Checking the uncertainty of the compressor efficiencies

A compressor using R600a was tested at several operating conditions (Table 4) from low to high cooling capacity.

Table 4: Uncertainty check: Test runs

Test run	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
n [s^{-1}]	20.0	20.0	26.7	26.7	26.7	26.7	33.3	33.3	33.3	50.0
p_1 [kPa]	88.8	36.6	36.6	88.8	36.6	88.8	88.8	36.6	88.8	36.6
p_2 [kPa]	464.5	464.5	464.5	464.5	772.4	772.4	464.5	772.4	772.4	772.4

Test run	**11**	**12**	**13**	**14**	**15**	**16**	**17**
n [s^{-1}]	50.0	50.0	50.0	75.0	75.0	75.0	75.0
p_1 [kPa]	88.8	36.6	88.6	36.6	36.6	88.8	88.8
p_2 [kPa]	464.5	464.5	772.4	464.5	772.4	772.4	604

Using this data the isentropic and volumetric efficiency was calculated according to equations (1) and (6) and the law of propagation of uncertainty was applied.

$$h_1 = f(R600a, p_1, T_1) \tag{1}$$

$$s_1 = f(R600a, p_1, T_1) \tag{2}$$

$$\rho_1 = f(R600a, p_1, T_1) \tag{3}$$

$$h_{2s} = f(R600a, p_2, s_1) \tag{4}$$

$$\eta_{isentropic} = \frac{h_{2s} - h_1}{P_{el,21}} \tag{5}$$

$$\lambda_{volumetric} = \frac{\dot{m}}{V_{disp} \cdot \rho_1 \cdot n} \tag{6}$$

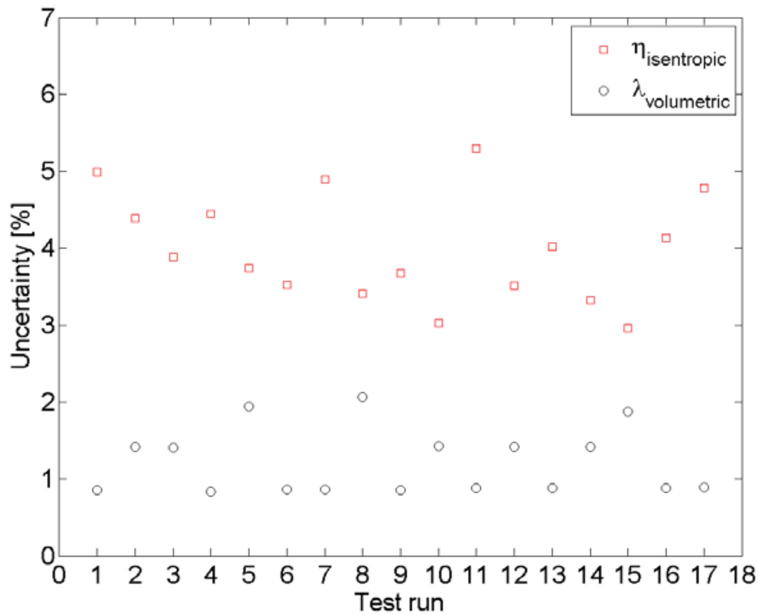


Figure 6: Resulting relative uncertainty considering the propagation of all uncertainties according to Table 3

The resulting relative uncertainties are shown in Figure 6. The uncertainties are under 5% for 94% of the values for the isentropic efficiency and under 3% for every value of the volumetric efficiency measured.

3.2 Comparison of measurements to the check point data of the data sheet

A new compressor has been installed and run for 250 hours for the quality assessment of the test-stand. Afterwards the check point conditions according to the compressor manufacturer data sheet (refer to Table 5) were tested. The resulting power consumption and mass flow rate were compared to the data sheet values (Table 6).

Table 5: Check point data

Check point	1	2	3	4	5
n [s^{-1}]	20.0	26.7	33.3	50.0	75.0
T_{evap} [°C]	-25	-25	-25	-25	-25
T_{cond} [°C]	55	55	55	55	55

Table 6: Check point conditions: Test results and comparison

-25°C/55°C		Data sheet		Measurement		Relative deviation	
		$P_{el,total}$ [W]	$\dot{m}$ [kg/h]	$P_{el,total}$ [W]	$\dot{m}$ [kg/h]	$P_{el,total}$ [W]	$\dot{m}$ [kg/h]
1	20.0 s^{-1}	25	0.45	24.4	0.458	-2.2%	1.9%
2	26.7 s^{-1}	32	0.61	32.0	0.626	0.0%	2.6%
3	33.3 s^{-1}	40	0.77	39.9	0.797	-0.2%	3.5%
4	50.0 s^{-1}	61	1.16	60.7	1.206	-0.5%	4.0%
5	75.0 s^{-1}	89	1.65	94.3	1.742	5.9%	5.5%

The measurement matches well with the check point data of the compressor. At the highest rotational speed a larger deviation can be noticed. However, since both the mass flow rate and the power consumption deviate in the same direction the coefficient of performance will not be negatively affected.

5. CONCLUSIONS

The design of a compressor test-stand for low capacity, variable speed household compressors is described. It has been noticed that various issues had to be taken into account when testing highly efficient household compressors.

- The compressor features large internal desuperheating of the discharged gas, which can lead to a refrigerant in two-phase condition at the compressor outlet.
- The initial superheat controlling approach is not applicable to some testing conditions due to the non-isenthalpic expansion processes.

Therefore, two electric heating stages have to be added to the test-stand and the mass flow sensor itself is heated to a temperature higher than the condensing temperature of the refrigerant. Hence, the compressor test-stand can operate in two modes:

- Gas cycle mode : High superheat conditions are controlled by the power of the heating stages.
- Split stream mode: Moderate and low superheat conditions are controlled by the fraction of the refrigerant passing by the condenser and expanding before being mixed back to the rest of the hot gaseous refrigerant.

The first test results show relative uncertainties of the isentropic and volumetric efficiencies of around 5% and 2% respectively. The check point data were similar to the data sheet values.

6. OUTLOOK

The next step is to rate the compressor test-stand using calorimetrically mapped compressors from the manufacturer. Then, various tests will be carried out in order to get data of the transient evolution of the compressor efficiencies and to further optimize the application in household refrigerators.

NOMENCLATURE

f	Frequency	kHz
h	Enthalpy	kJ/kg
KV	Flow coefficient	m^3/h
$\dot{m}$	Mass flow	kg/h
n	Rotational speed	s^{-1}
OD	Opening degree	%
p	Pressure	kPa
P_{el}	Power (electric)	W
PF	Power factor	-
T	Temperature	°C
V	Volume	cm^3
η	Isentropic efficiency	-
λ	Volumetric efficiency	-
ρ	Density	kg/m^3
LP	Low pressure switch	
HP	High pressure switch	

Subscript	
1...33	Index/location in schematic
2s	State point after isentropic compression
cond	Condensation
disp	Displacement
evap	Evaporation

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