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Title

Vapour quality determination for heat pumps using two-phase suction

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Vapour quality determination for heat pumps using two-phase suction

Abstract

For residential heat pumps, R32 becomes one of the most widely used refrigerant, thanks to an environmental impact much lower than R410A. However, with a classic suction superheat control, R32 induces higher discharge temperatures. Two-phase suction is thus generally allowed in rotary compressors to lower the discharge temperature. In theory, this process can also improve the heat pump performances. However, using two-phase suction for real-time performance optimization implies to identify and control the suction vapour quality. In this study, an iterative method using the compressor energy balance and the compressor efficiencies is presented to determine the suction vapour quality and refrigerant mass flow rate without using a flowmeter. An R32 air-to-air heat pump is then experimentally tested. The reference values are indirectly measured through the mass flow rate measurement and the compressor energy balance. A correlation is thus presented, giving the vapour quality from the operating parameters. If the volumetric or global compressor efficiencies are known, the proposed iterative method allows to calculate the vapour quality with a 0.4 % accuracy. Moreover, this method gives the refrigerant mass flow rate and heating capacity with a 1 % accuracy. The sensitivity of the method is evaluated considering a 10 % error on the volumetric or global efficiency values. In this case, the suction vapour quality values are kept within 2.5 % of the reference measurement. In addition of allowing a precise control of two-phase suction, this method can also make the in-situ performance measurement of these heat pumps possible.

Nomenclature

A	Compressor shell area	(m ²)
C	Compressor displacement	(m ³)
D	Diameter	(m)
ε_r	Relative error	
F	Compressor rotating frequency	(Hz)
h	Enthalpy	(J/kg)
k	Thermal conductivity	(W/(m.K))
L	Compressor height	(m)
\dot{m}	Mass flow rate	(kg/s)
\overline{Nu}	Nusselt number	
P	Pressure	(Pa)
\dot{Q}	Thermal power	(W)
T	Temperature	(K ou °C)
\dot{W}	Electric power	(W)
x_v	Vapour quality	
η	Efficiency	
ρ	Mass density	(kg/m ³)
σ	Uncertainty	
σ_{SB}	Stefan-Boltzmann coefficient	(W/(m ² .K ⁴))

Subscripts

0	Initialized value
amb	Ambient air

cond	Condenser
comp	Compressor
evap	Evaporator
D,1	Superior compressor shell surface
D,2	Inferior compressor shell surface
g	Global
in	Inlet
is	Isentropic
L	Lateral side of the compressor
losses	Compressor thermal losses
out	Outlet
ref	Reference
suc	Suction
surf	Compressor shell surface
tot	Total
vol	Volumetric

Abbreviations

<i>CEB</i>	Compressor Energy Balance
<i>COP</i>	Coefficient of Performance
<i>EEV</i>	Electronic expansion valve
<i>GWP</i>	Global Warming Potential

1. Introduction

Thanks to their high efficiency, heat pumps play a major role in greenhouse gas emission and energy consumption reduction in buildings in Europe. In order to reduce the impact of the refrigerant on climate change, the fluids with the highest global warming potentials (GWP) will progressively be phased out, as a consequence of the F-Gas regulation (EU) No 2014/517. Therefore, in particular for residential heat pumps, R410A is being replaced with R32, which has a 3-time lower GWP.

R32 can operate under the same temperature conditions than R410A without any major change in the thermodynamic cycle structure and can also improve the heat pump performances. Xu et al. (2013) experimentally demonstrated that the COP improvement using R32 instead of R410A on the same single-stage heat pump was between 2 and 9 %. However, controlling the expansion valve on suction superheat in the same way would lead to much higher discharge temperatures that could damage the compressor. Xu et al. (2013) showed that the discharge temperature using R32 was significantly higher than when using R410A in every operating condition, up to + 34 °C for an outdoor temperature of - 18 °C.

Thus, several patents were published with new control strategies. Using liquid injection cycles would allow to reduce the discharge temperature, but it would increase the manufacturing cost of the heat pump. Taira et al. (2003) recommended a method using the electronic expansion valve (EEV) to control directly the discharge temperature, and Yamada (2017) suggested to use the EEV to control the discharge superheat value, depending on the compressor frequency and the compression ratio. For both solutions, it leads to two-phase fluid at the compressor suction port in most operating conditions, which reduces significantly the discharge temperature. Taira et al. (2003) states that this wet compression cannot harm rotary-type compressors (rolling piston) as long as the suction vapour quality is higher than 0.75. Nowadays, two-phase suction is widely used in order to lower the discharge temperature in R32 air conditioners and heat pumps using a rotary compressor.

The effects of wet compression on the heat pump performances have been studied, but the effect of the discharge temperature control of heat pumps using two-phase suction on vapour quality is unknown. For example, Seong et al. (2017) showed that this process can improve the heat pump performances by lowering the compression ratio and increasing the isentropic efficiency. However, for that purpose, the suction vapour quality should be controlled to an optimal value depending on the operating conditions. In order to optimise the performances of heat pumps using two-phase suction, the real-time estimation of the suction vapour quality on an operating heat pump is necessary. Huang et al. (2018) presented a method based on Quasi Saturation Isentropic Compression in order to control the discharge temperature. With this method, the refrigerant can also be diphasic at the compressor suction, so the suction vapour quality was estimated using a fixed compressor global efficiency value. This method allows to compare the suction quality values on the same compressor under different operating conditions, but as the method precision is not established by the authors, it is not known if it is accurate enough for wet compression optimisation. Endoh (2014) presented a method to determine the suction vapour quality from the mass flow rate measurement and the compressor energy balance. It gives very good results, with a 0.5 % accuracy. However, using a mass flow meter on an operating heat pump is not always possible and can be expensive, so this method is not adapted for a real-time in-situ optimisation purpose.

When the fluid is in two-phase state at the compressor suction port, making the hypothesis that it is saturated leads the compressor energy balance to highly overestimate the mass flow rate. On the contrary, making this assumption to calculate the mass flow rate using the volumetric efficiency or the global efficiency gives underestimated results. The real mass flow rate value can thus be bounded using the compressor energy balance and the volumetric or the global efficiency with a vapour quality of 1. This paper presents two similar methods using iterations on suction vapour quality to make these calculations converge, allowing to obtain the suction vapour quality and the refrigerant mass flow rate without using a flow meter. An experimental study of an air-to-air heat pump using R32 in climatic chambers is then presented. The first objective is to measure the suction vapour quality from the mass flow rate measurement to obtain a correlation as a function of the operating conditions. The second one is to experimentally validate the iterative method, using the compressor energy balance and the volumetric or global efficiency of the compressor, and to compare the results with the reference measurement. The results and the main uncertainty factors of the method are discussed.

2. Compressor energy balance

For heat pumps with a controlled positive suction superheat value, a method has been previously developed to measure the mass flow rate and heating capacity on field using a light instrumentation. To avoid using a flowmeter, the refrigerant mass flow rate can be indirectly measured using only temperature and electrical power measurements. This method, developed by Tran (2012), relies on the compressor energy balance (equation 1).

$$\dot{W}_{comp} = \dot{m}(h_{comp,out} - h_{comp,in}) + \dot{Q}_{losses} \quad (Eq. 1)$$

The suction and discharge temperatures are measured, as well as the evaporating and condensing temperatures. The suction and discharge enthalpy, respectively $h_{comp,in}$ and $h_{comp,out}$, can then be computed from these measurements, as the fluid is superheated vapour. The compressor power input

\dot{W}_{comp} is also measured. \dot{Q}_{losses} represents the compressor heat losses for a hermetic compressor. In order to improve the compressor heat losses estimation, Niznik (2017) presented a measurement method using two temperature measurements for rotary-type compressors. One is placed on the compressor shell surface T_{surf} , and the other measures the ambient temperature T_{amb} . Niznik (2017) showed that the temperature of the air entering the outdoor unit was the best representation of the compressor ambient temperature, and that the temperature of the compressor shell is uniform enough to use only one sensor for rotary-type compressors. \dot{Q}_{losses} is thus calculated using equation 2, considering the convective and radiative heat losses. The emissivity of the compressor shell surface is supposed to be equal to 1.

$$\dot{Q}_{losses} = k \left(\frac{\overline{Nu}_L}{L} A_L + \frac{\overline{Nu}_{D,1}}{D} A_D + \frac{\overline{Nu}_{D,2}}{D} A_D \right) (T_{surf} - T_{amb}) + \sigma_{SB} A_{tot} (T_{surf}^4 - T_{amb}^4) \quad (Eq. 2)$$

Where Nu is the Nusselt number for the different sides of the compressor, k is the thermal conductivity of the air, L , D and A stand respectively for the length (height), diameter (1 and 2 for the top and the bottom surface respectively) and area for the lateral side, the top and the bottom of the compressor, σ_{SB} is the Stefan-Boltzmann constant for radiative heat transfer. The air properties are calculated at film temperature.

Noel et al. (2018) experimentally validated the compressor energy balance method on an air-to-water heat pump with a controlled positive suction superheat in stationary state, and concluded that the refrigerant mass flow rate and the heating capacity are obtained with an average 4.4 % accuracy compared to the water-side measurement.

However, for heat pumps allowing two-phase fluid at the compressor suction port to reduce the discharge temperature, the suction vapour quality and enthalpy are unknown. By comparing the calculation with an hypothesis of saturated fluid and using various theoretical vapour quality values, an estimation has been made on a few working conditions that the hypothesis of saturated fluid leads to a very high overestimation of the mass flow rate that can reach 20 % for a 0.97 vapour quality and about 40 % for a 0.94 vapour quality. The mass flow rate is needed to know the suction quality and enthalpy, or the vapour quality is necessary to calculate the suction enthalpy and the mass flow rate with the compressor energy balance. Therefore, another equation is needed to be able to make iterations on these variables.

3. In-situ vapour quality calculation method

When the fluid is in two-phase state at the compressor suction port, its vapour quality cannot be directly measured. Moreover, its evaluation through the void fraction measurement would not be accurate enough (Tran et al. 2012) and would need additional instrumentation. The compressor energy balance needs to be completed with another equation to be able to calculate the refrigerant mass flow rate and the suction vapour quality. The following paragraphs show how it can be done using the mass flow rate calculation from the volumetric or global compressor efficiency.

3.1. Mass flow rate calculation using volumetric efficiency

From the volumetric efficiency η_{vol} , the refrigerant mass flow rate can be calculated using equation 3.

$$\dot{m}_{\eta_{vol}} = \eta_{vol} \cdot \rho_{suc} \cdot C \cdot F \quad (Eq. 3)$$

ρ_{suc} is the refrigerant suction density, C the displacement volume of the compressor, and F the compressor rotating frequency. In order to calculate ρ_{suc} , the suction vapour quality is needed. Thus, its value can be initialized to $x_{v,0} = 1$, considering that the fluid is saturated vapour. Making this hypothesis when the fluid is actually under two-phase state underestimates the suction density, thus equation 3 underestimates the refrigerant mass flow rate. The subscript "0" is used for the chosen or calculated initial values of vapour quality or mass flow rate.

$$\dot{m}_{\eta_{vol}}(x_{v,0}) < \dot{m} \quad (\text{Eq. 4})$$

3.2. Mass flow rate calculation using global efficiency

The global efficiency is the ratio between the actual compression work required for the compression and the isentropic work. It considers both the isentropic losses and the heat losses. It is more adapted for studying a real compression than the isentropic efficiency, which only includes isentropic losses under adiabatic process. The refrigerant mass flow rate can be calculated from the compressor global efficiency η_g using equation 5.

$$\dot{m}_{\eta_g} = \frac{\eta_g \cdot \dot{W}_{comp}}{h_{comp,out,is} - h_{comp,in}} \quad (\text{Eq. 5})$$

\dot{W}_{comp} is the measured compressor power input. The suction vapour quality is needed to calculate the suction enthalpy, $h_{comp,in}$, and the corresponding discharge isentropic enthalpy, $h_{comp,out,is}$. As for the volumetric efficiency equation, the calculation can be initialised considering the refrigerant as saturated, with a vapour quality of $x_{v,0} = 1$. With this hypothesis, the mass flow rate is underestimated because the slope of the isentropic lines increases with a decreasing vapour quality, so the isentropic enthalpy difference decreases.

$$\dot{m}_{\eta_g}(x_{v,0}) < \dot{m} \quad (\text{Eq. 6})$$

3.3. Iterations on vapour quality

Using the volumetric or global efficiency with the hypothesis that the fluid is saturated underestimates the refrigerant mass flow rate. As seen in paragraph 2, the mass flow rate can also be calculated using the compressor energy balance using the same vapour quality initial value (equation 7).

$$\dot{m}_{CEB,0} = \frac{\dot{W}_{comp} - \dot{Q}_{losses}}{h_{comp,out} - h_{comp,in}(x_{v,0})} \quad (\text{Eq. 7})$$

However, it overestimates the suction enthalpy, so equation 7 overestimates the mass flow rate.

Since the calculation method is the same whether the volumetric efficiency or the global efficiency is used, the mass flow rate calculated with one or the other is called \dot{m}_{η} . Its value calculated with the hypothesis that the fluid is saturated ($x_{v,0} = 1$) is called $\dot{m}_{\eta,0}$.

Thus, the real mass flow rate value is necessarily between the efficiency and the compressor energy balance results (equation 8).

$$\dot{m}_{\eta,0} < \dot{m} < \dot{m}_{CEB,0} \tag{Eq. 8}$$

As the suction vapour quality is needed to calculate the mass flow rate and vice-versa, it is necessary to make iterations using two different equations. Equation 8 shows how the mass flow rate value is bounded, so iterations can then be made by progressively decreasing the vapour quality value to make the efficiency and the compressor energy balance calculation converge. The iterations are stopped when the difference between the compressor energy balance and the volumetric (or global efficiency) is smaller than the measurement uncertainty.

Figure 1 illustrates the principle of these iterations on vapour quality, where \dot{m}_{η} is the mass flow rate calculated with one of those efficiencies and \dot{m}_{CEB} is the mass flow rate calculated with the compressor energy balance. Their initialised values calculated for $x_{v,0} = 1$ are respectively $\dot{m}_{\eta,0}$ and $\dot{m}_{CEB,0}$.

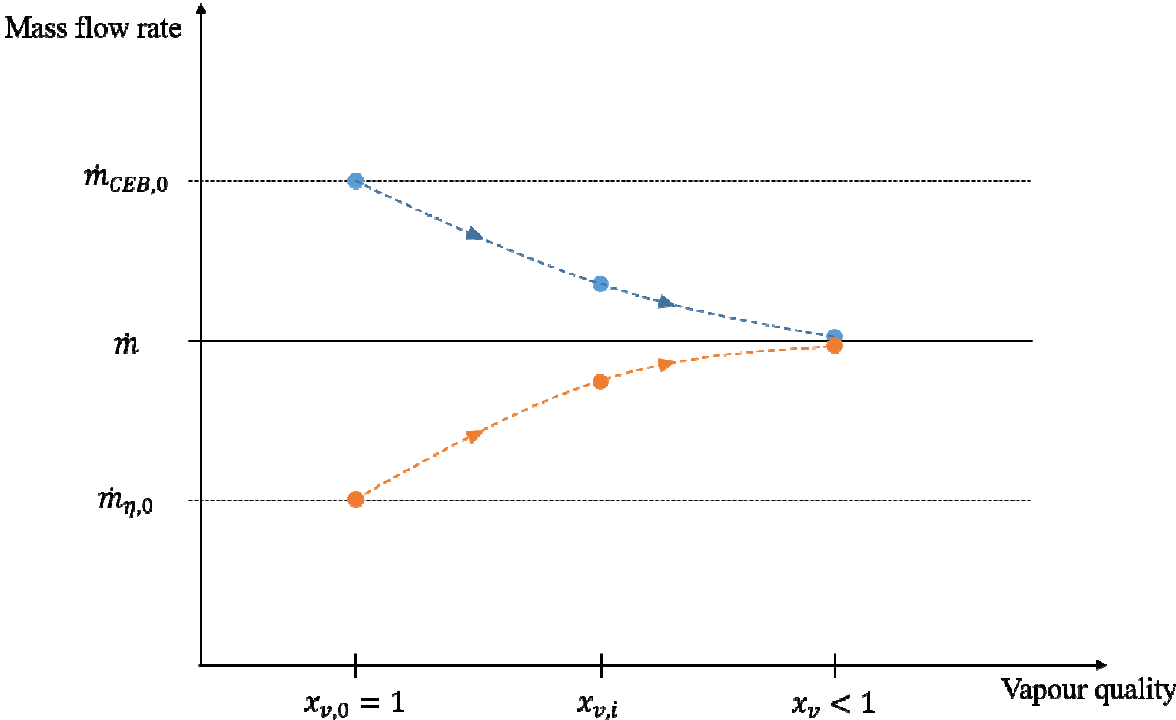


Figure 1: Principle of the Iterative method on suction vapour quality

4. Experimental set-up

A commercial 3.5 kW air-to-air heat pump has been instrumented and tested in order to study its two-phase suction control and the iterative methods presented before. The heat pump is composed of an inverter-driven rotary compressor, an electronic expansion valve, an evaporator and a condenser. The

working fluid is R32. In order to reduce the compressor discharge temperature, two-phase fluid is allowed at the compressor suction port.

4.1. Metrology

The heat pump was instrumented in order to have a reference measurement of the mass flow rate which can be used to obtain reference values of suction vapour quality and heating capacity.

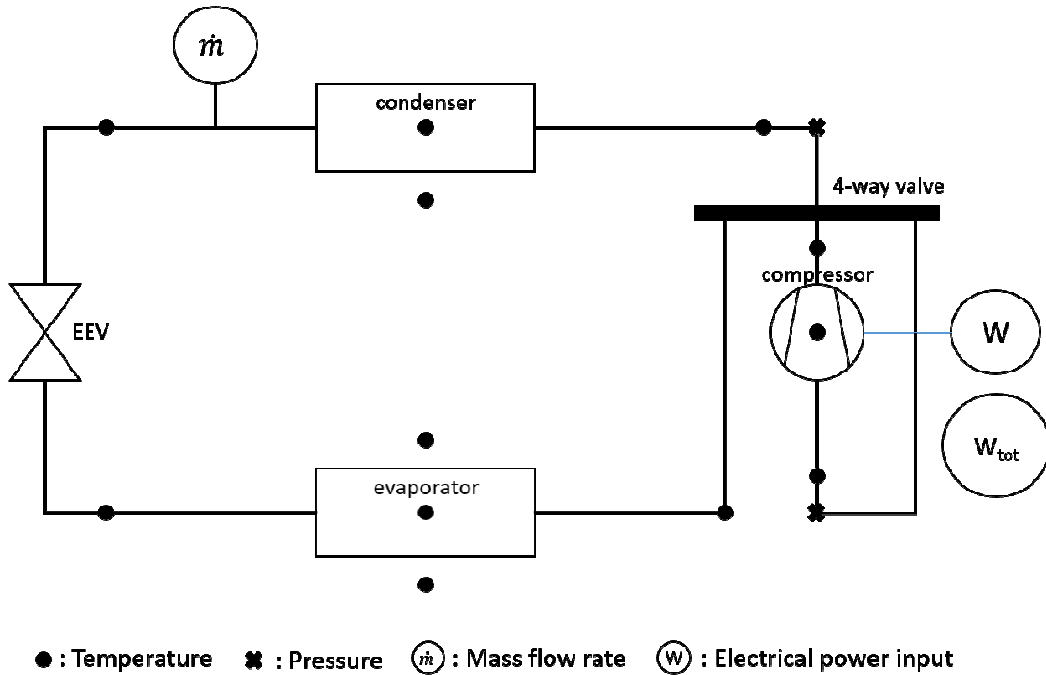


Figure 2: Instrumentation installed on the air-to-air heat pump

The refrigerant temperature is measured with surface Pt100 sensors. Their measurement uncertainty is considered to be 0.8 K, taking into account the uncertainty caused by the surface measurement (LNE, 2004). The air temperature is measured with Pt 100 sensors with a 0.1 K measurement uncertainty. The compressor input power and the total heat pump power are measured with power meters with a 0.5 % relative measurement uncertainty. Suction and discharge pressures are measured with a relative measurement uncertainty of 0.25 %. The refrigerant mass flow rate is measured with a Coriolis-effect flow meter placed at the condenser outlet, where the fluid is liquid. Its relative measurement uncertainty is 0.5 % of the mass flow rate value. The thermodynamic properties are calculated with CoolProp Library (Bell et al., 2014). According to Casari et al. (2019), the relative uncertainty on enthalpy values calculated using CoolProp is below 0.8 % for R410A. It is supposed to be equivalent for R32 considering the similarities between those fluids. A fraction of the compressor lubricating oil migrates into the refrigerant cycle. Niznik (2017), assumed that this oil circulation ratio represents 0.5 % of the refrigerant mass under any working condition, and experimentally verified that it leads to accurate mass flow rate results. The effects of the miscibility of the lubricant in the refrigerant are neglected. This value is used to calculate the refrigerant enthalpy values, according to equation 9.

$$h = (1 - C_{oil}) \cdot h_{ref} + C_{oil} \cdot h_{oil} \quad (\text{Eq. 9})$$

Table 1 sums up the tools used for measurement and calculation and their associated uncertainties. The acquisition time step is 10 s.

Measurement	Product	Measurement uncertainty
Mass flow rate	Coriolis 83F15-AD2S	0.5 %
Fluid temperature	Surface Pt100	0.8 K
Air temperature	Pt100	0.1 K
Pressure	Pressure transmitter	0.25 %
Power	Power meter	0.5 %
Thermodynamic properties	CoolProp	0.8 %

Table 1: Measurement and calculation uncertainties

4.2. Operating conditions

Different outdoor and heat pump set point temperatures were tested in order to make the air-to-air heat pump operate under various functioning temperatures, compressor frequencies and thermal loads. 60 combinations of operating conditions have been studied. The outdoor temperature ranged between $-10\text{ }^{\circ}\text{C}$ and $+10\text{ }^{\circ}\text{C}$, and the compressor rotating frequency ranged between 30 and 100 Hz.

For each operating condition, some dynamic phases could occur. In particular, for outdoor temperatures below $5\text{ }^{\circ}\text{C}$, the heat pump needed to engage defrosting sequences, with an inverted cycle operation during a few minutes. These phases, as well as the compressor starting phases, were excluded from the results. This study focuses only on steady-state operating phases, and the values of the variables of interest represented after are the mean values calculated during each of the 60 steady-state phases. Each operating condition was set during at least 2 hours in order to let the heat pump parameters stabilise.

4.3. Reference Values

Heating capacity

Thanks to the mass flow rate, temperature and pressure measurements, the reference heating capacity of the heat pump can be calculated with equation 10.

$$\dot{Q}_{cond,ref} = \dot{m} \cdot (h_{cond,in} - h_{cond,out}) \quad (\text{Eq. 10})$$

\dot{m} is the measured mass flow rate, $h_{cond,in}$ the refrigerant enthalpy at the condenser inlet, and $h_{cond,out}$ the enthalpy at the condenser outlet.

Considering the different measurement and calculation uncertainties, the uncertainty of the reference heating power can be calculated with the uncertainty propagation equation (equation 11).

$$\sigma_{\dot{Q}_{cond}} = \sqrt{\sum_i \left(\frac{\partial \dot{Q}_{cond,w}}{\partial x_i} \right)^2 \sigma_{x_i}^2} \quad (\text{Eq. 11})$$

For each operating condition, the relative measurement uncertainty of the reference heating capacity is below 1 % of its value.

Suction vapour quality

In order to obtain a reference value of the suction vapour quality, the compressor energy balance is used as well as the measured mass flow rate \dot{m} (equation 1, recalled below).

$$\dot{W}_{comp} = \dot{m}(h_{comp,out} - h_{comp,in}) + \dot{Q}_{losses} \quad (Eq. 1)$$

The compressor power input \dot{W}_{comp} and its heat losses \dot{Q}_{losses} are estimated (Niznik, 2017). The discharge temperature and pressure are measured, allowing to calculate the discharge enthalpy value $h_{comp,out}$. The suction enthalpy is then deduced from the compressor energy balance, according to equation 12.

$$h_{comp,in} = h_{comp,out} - \frac{\dot{W}_{comp} - \dot{Q}_{losses}}{\dot{m}} \quad (Eq. 12)$$

The suction vapour quality can then be determined from the measured suction pressure P_{suc} and enthalpy values (Eq 13), using CoolProp library.

$$x_{v,ref} = f(P_{suc}, h_{comp,in}) \quad (Eq. 13)$$

The vapour quality relative measurement uncertainty, calculated from table 1 and equation 10, is below 1.2 % of its value for every steady-state operating condition considered. The uncertainties caused by the assumptions needed to calculate the compressor heat losses are difficult to estimate. However, from 6 operating conditions, Niznik (2017), experimentally estimated the compressor heat losses uncertainty to be 10.4 %, making the hypothesis that it is equal to the experimental root mean square error. A 10.4 % uncertainty on the compressor heat losses leads to a 0.3 % uncertainty on the suction vapour quality calculated with equations 12 and 13.

At first sight, the main parameters leading to a high discharge temperature should be a high compression ratio, a high compressor frequency and a high heating capacity. Therefore, these conditions should lead to the lowest vapour quality values. Figure 3 shows the suction vapour quality as a function of the compression ratio.

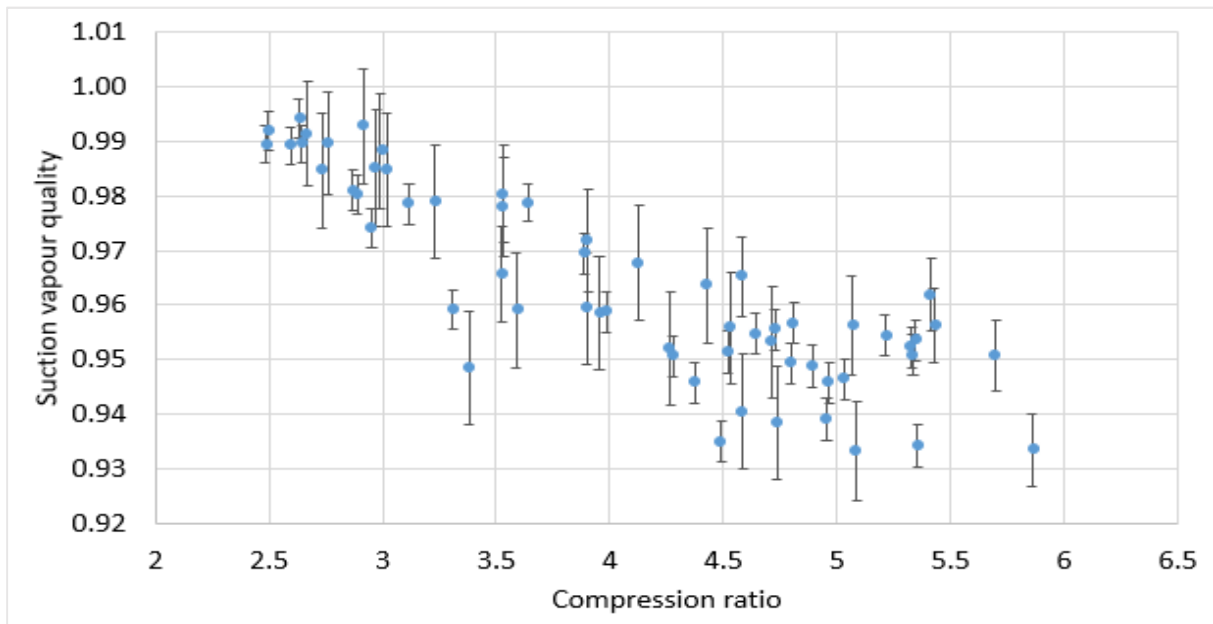


Figure 3: Reference suction vapour quality as a function of the compression ratio

Figure 3 shows that the suction vapour quality tends to decrease with the compression ratio, but it also shows that it cannot be resumed to it. To have a better modelling of the suction vapour quality, the evaporating temperature, the condensing temperature and the compressor frequency are taken into account to determine a correlation (equation 14).

$$\begin{aligned}
 x_v = & 1,008 - 1,491.10^{-3}T_{evap} + 1,852.10^{-3}T_{cond} - 1,787.10^{-3}F \\
 & + 1,128.10^{-4}T_{evap}^2 - 6,169.10^{-5}T_{cond}^2 - 2,586.10^{-6}F^2 \\
 & + 1,23.10^{-4}T_{evap}T_{cond} - 1,098.10^{-5}T_{evap}F + 4,521T_{cond}F
 \end{aligned}
 \tag{Eq. 14}$$

Figure 4 represents the vapour quality calculated with equation 14 compared to the vapour quality measurement.

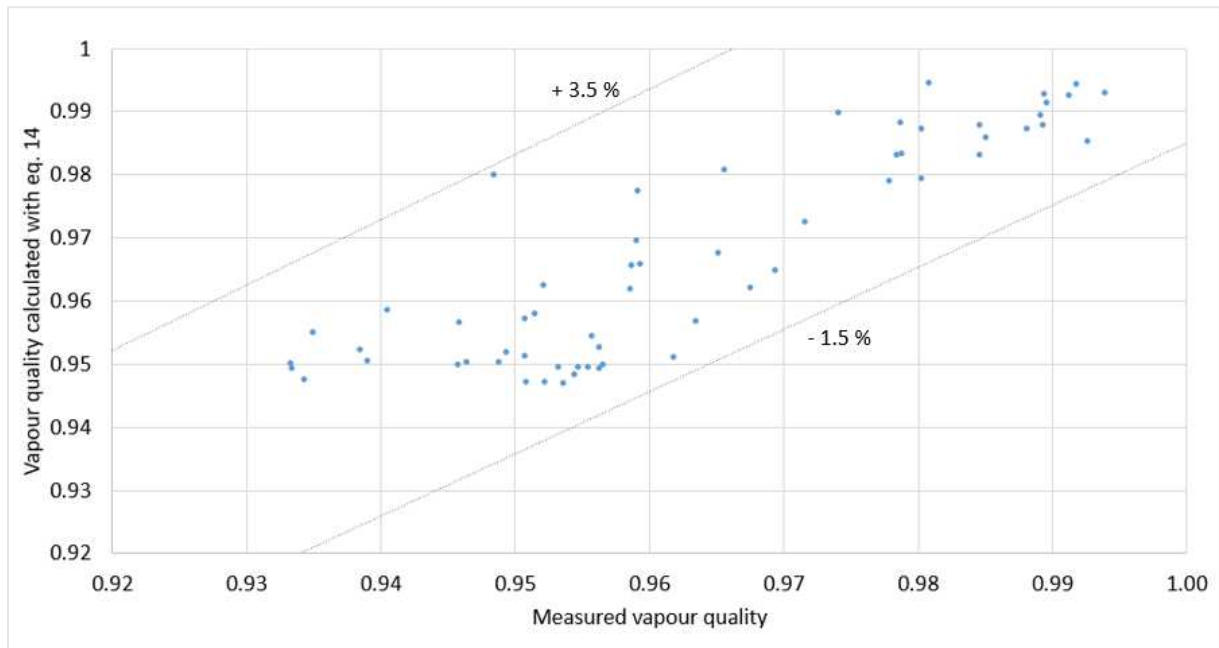


Figure 4: Comparison of vapour quality from the model (Eq. 14) and from the measurement

This relationship between the suction vapour quality and the operating conditions could be useful to have a simple estimation of the suction vapour quality in steady-state operation of a heat pump using two-phase suction. However, its accuracy needs to be evaluated when used on another heat pump that might have a different control logic of the discharge temperature.

Volumetric and global compressor efficiencies

The compressor volumetric efficiency can be calculated from equation 15.

$$\eta_{vol} = \frac{\dot{m}}{\rho_{suc} \cdot C \cdot F}
 \tag{Eq. 15}$$

With \dot{m} the measured mass flow rate, ρ_{suc} the suction density, C the compressor displacement, and F the compressor frequency. ρ_{suc} is calculated from the measured suction pressure and vapour quality, calculated from equation 13.

The compressor global efficiency considers the irreversibility and the heat losses of the compression. It can be calculated using equation 16.

$$\eta_g = \frac{\dot{m}(h_{comp,out,is} - h_{comp,in})}{\dot{W}_{comp}} \quad (\text{Eq. 16})$$

With \dot{m} the measured mass flow rate and \dot{W}_{comp} the measured compressor power input. The suction enthalpy $h_{comp,in}$ is calculated from the measured suction pressure and vapour quality, calculated from equation 13. The corresponding isentropic discharge enthalpy $h_{comp,out,is}$ is then deduced from $h_{comp,in}$ and the measured discharge pressure.

Figure 5 represents the mean volumetric and global compressor efficiencies as a function of the mean compression ratio for every steady-state operating condition studied.

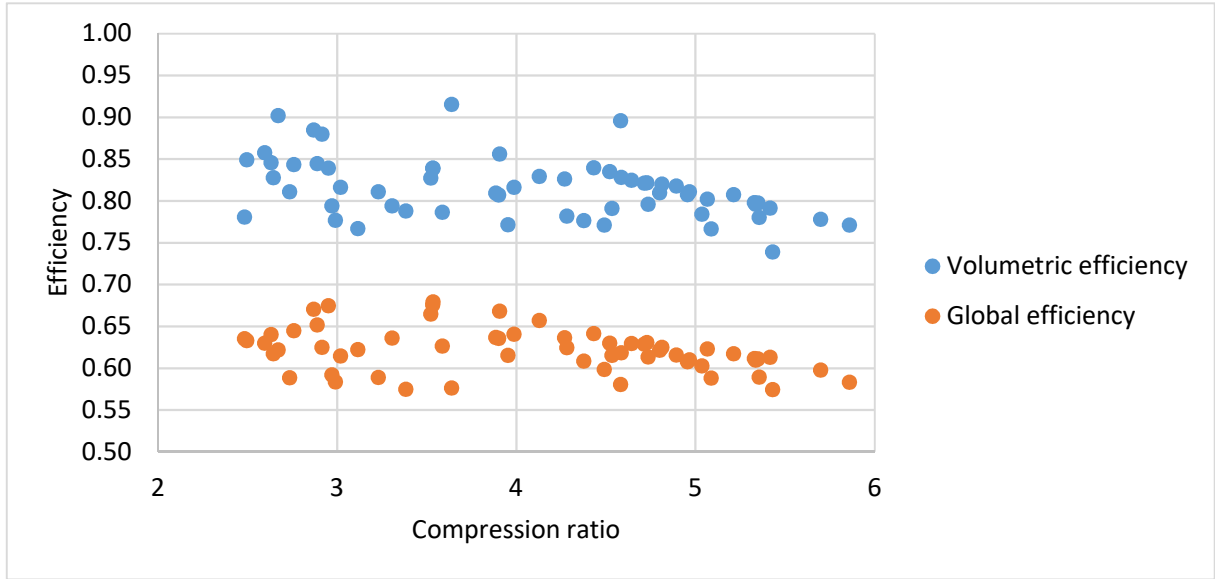


Figure 5: Volumetric and global efficiency as a function of the compression ratio

Considering every other parameter fixed, the volumetric efficiency decreases for high compression ratios. However, in this case, the compressor rotating frequency and the suction vapour quality depend on the heat pump control and are influenced by the compression ratio. A high compressor rotating frequency has a positive effect on volumetric efficiency, and Seong et al. (2017) showed that a decreasing vapour quality can increase volumetric efficiency to a certain extent. Therefore, figure 5 is a consequence of the combined effects of these parameters.

As for the compressor volumetric efficiency, figure 5 does not represent a usual compressor performance map for global efficiency, which would be established with a fixed frequency in specified operating conditions. It is a consequence of the combined effects of the evolution of the compression ratio, the compressor frequency and the suction vapour quality.

5. Results and discussion

5.1. Experimental results

The iterative method presented in paragraph 3 is applied to the 60 steady-state phases set on the air-to-air heat pump, using the volumetric efficiency (equation 3) or the global efficiency (equation 5), and the compressor energy balance (equation 1). In every operating condition, it converges to a mass flow

rate value and a suction vapour quality value. The relative error of the average estimated suction vapour quality can be calculated according to equation 17.

$$\varepsilon_r = 100 * \frac{(x_v - x_{v,ref})}{x_{v,ref}} \quad (\text{Eq. 17})$$

Figure 6 shows the estimated suction vapour quality using the volumetric efficiency (blue) and the global efficiency (orange), as a function of the reference vapour quality calculated with equation 13.

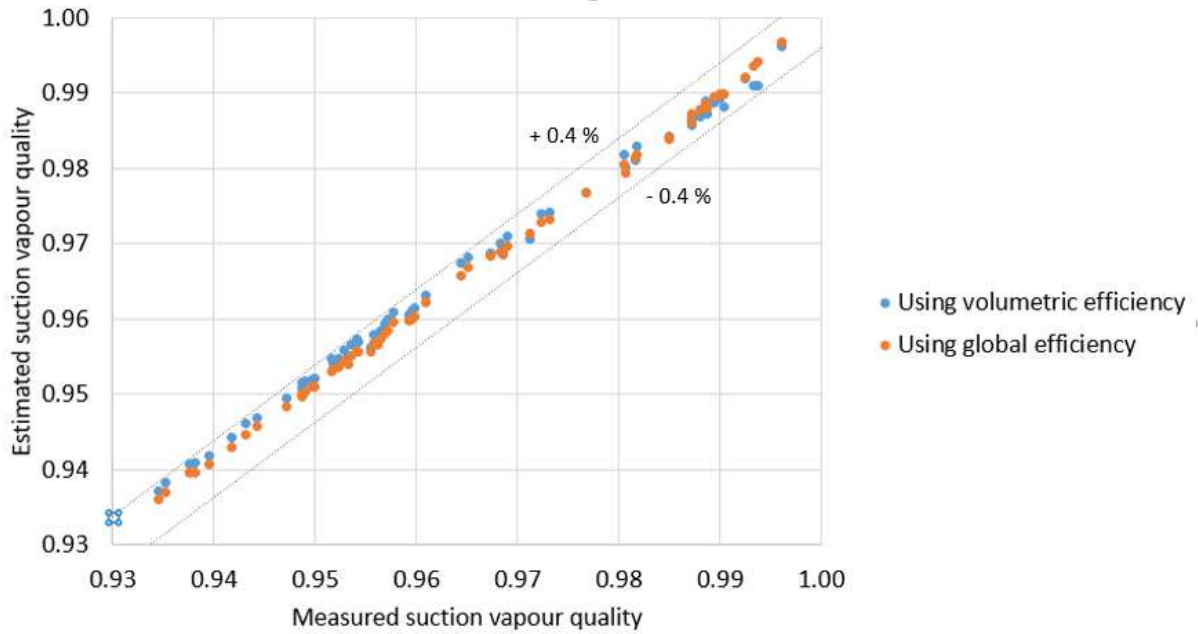


Figure 6: Suction vapour quality estimated with the iterative method compared to the reference measurement

In the 60 operating conditions tested, the estimated vapour quality from the compressor energy balance and the volumetric or global efficiency is always within 0.4 % of the suction vapour quality obtained from the mass flow rate measurement.

These methods also provide mass flow rate values \dot{m}_η , so the heating capacity can be calculated according to equation 18.

$$\dot{Q}_{cond,\eta} = \dot{m}_\eta \cdot (h_{cond,in} - h_{cond,out}) \quad (\text{Eq. 18})$$

Where $\dot{Q}_{cond,\eta}$ is the heating capacity calculated with the iterative methods using either the volumetric or the global efficiency. Figure 7 shows $\dot{Q}_{cond,\eta}$ compared to the reference heating capacity $\dot{Q}_{cond,ref}$ (equation 9).

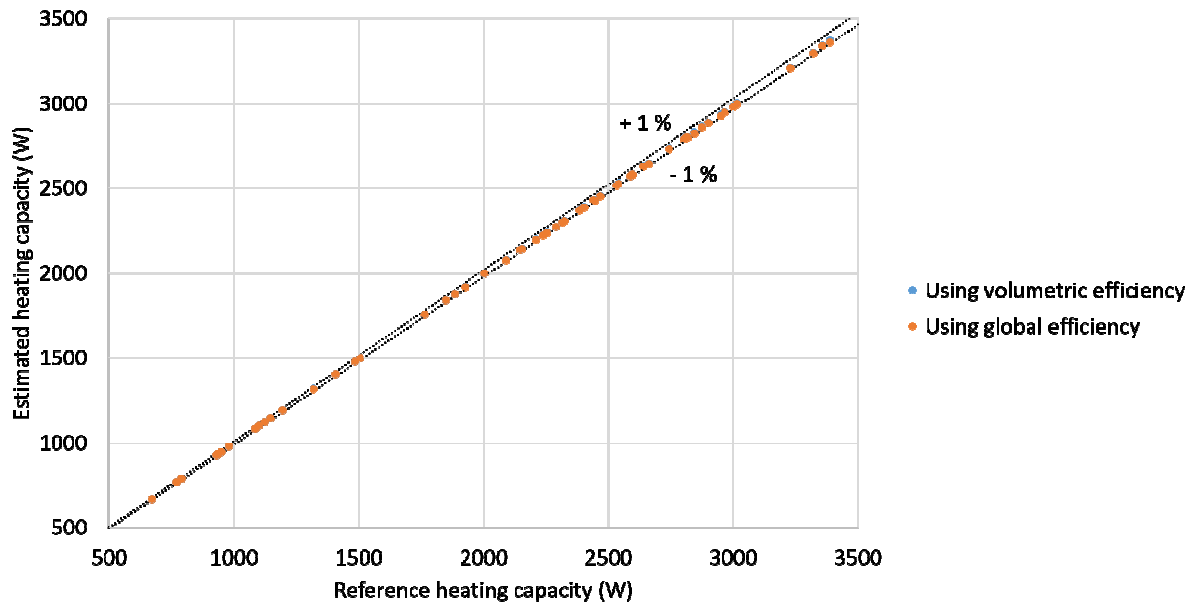


Figure 7: Heating capacity estimated using the iterative method compared to the reference value

Both iterative methods give results within 1 % of the reference heating capacity.

Therefore, the presented iterative method gives very good results of vapour quality and mass flow rate compared to the reference measurements when the compressor volumetric or global efficiencies are known.

5.2. Main uncertainty factors

The volumetric efficiency equation (Eq. 3) requires also the knowledge of the compressor displacement and the rotating frequency of the compressor. The latter can be measured on field by measuring the electrical frequency of the compressor power input. If the method is implemented by the manufacturer, it is also possible to use the frequency value set by the compressor control. The manufacturer also knows the compressor displacement. However, for an independent implementation of the method, the displacement is not always easily available. In that case, it would be very complicated to make an estimation of the compressor displacement, and it would be preferable to use the global efficiency equation.

Concerning the compressor global efficiency equation (Eq. 5), it requires the compressor power input measurement, which is also necessary for the compressor energy balance.

In the results presented on figures 6 and 7, the compressor efficiencies were known from the mass flow rate measurement. Compressor manufacturers generally provide performance maps, giving the compressor efficiency as a function of the functioning temperatures, for given rotating frequencies and fixed suction superheat conditions. It would be possible to use these correlations to estimate the volumetric or global efficiency on field, and use the methods presented in this paper. However, the difference in efficiencies with real operation under wet compression will thus not be considered properly, leading to additional bias as compared to the results presented in Section 5.1. Generic correlations or methods could not be found to consider the effect of wet compression.

In the case of an independent on-field measurement, where compressor maps are not always available, it could be possible to use a published correlation made for a similar compressor. The uncertainty on the obtained vapour quality value would probably be larger than using the specific compressor data, but it could at least give a good representation of the effect of the control logic on suction vapour quality.

In order to have an estimation of the sensitivity of the iterative method, the calculation have been made introducing an under and overestimation of the efficiency values of 10 %. Figure 8 shows the results of the method on suction vapour quality, using volumetric or global efficiency with a 10 % relative error, compared to the reference measurement.

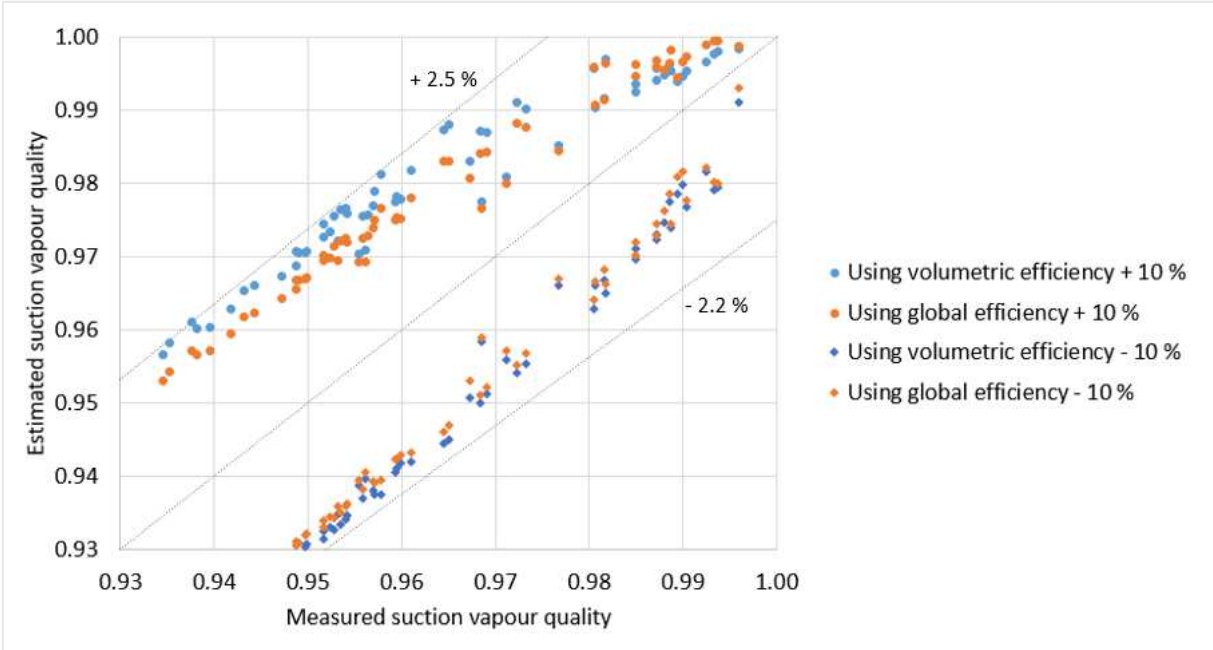


Figure 8: Vapour quality results with a 10 % error on volumetric and global efficiency

Figure 8 shows that even if a 10 % error is made on the estimation of the compressor efficiencies, the results on suction vapour quality are kept within 2.5 % of the reference values. It can be noted that an underestimation is slightly preferable than an overestimation. Also, the error is reduced when the vapour quality gets close to 1, as the errors made on suction enthalpy and density at the first iteration are smaller.

The effects of a 10 % efficiency error on the heating capacity estimation are represented on figure 9.

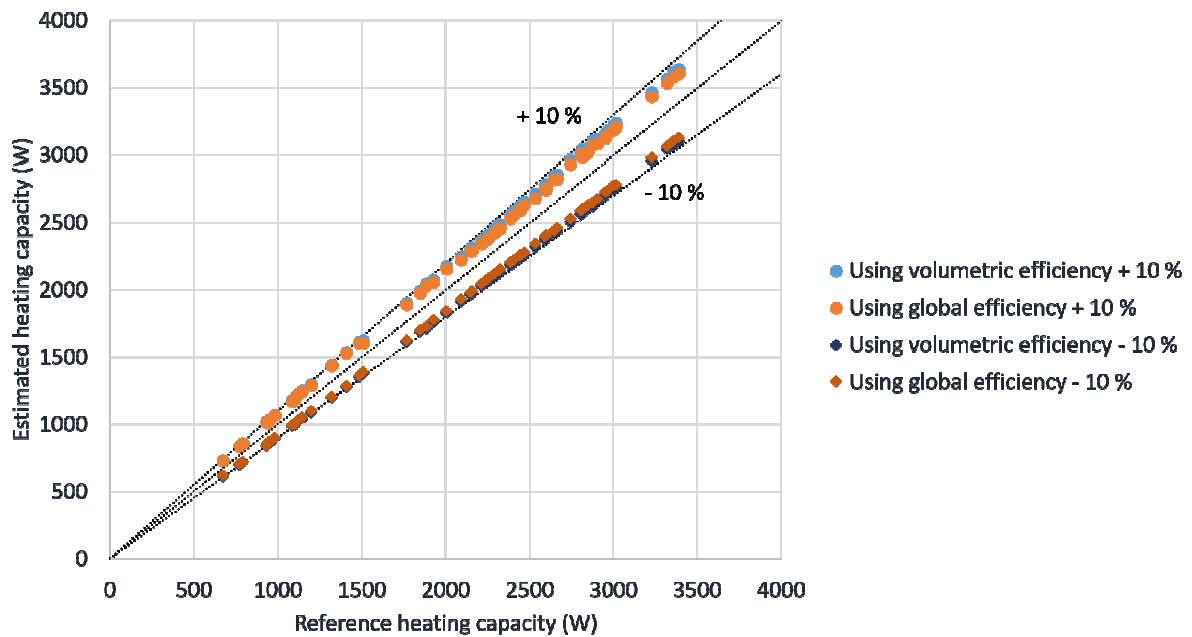


Figure 9: Heating capacity results with a 10 % error on volumetric and global efficiency

The relationship between an error on the efficiency values and the error on heating capacity is more direct than for suction vapour quality (i.e. 10 % error on the efficiency leads to about 10 % maximum error on the capacity). Therefore, in a performance assessment purpose, the compressor efficiencies estimation needs to be accurate enough to have a correct heating capacity evaluation. In any case, even with a 10 % error on the efficiencies, this iterative method gives far better results than those of the compressor energy balance used with the saturated vapour hypothesis when the fluid is in two-phase state at the compressor suction port.

6. Conclusion

A method has been presented in order to calculate the suction vapour quality and the refrigerant mass flow rate using the compressor volumetric or global efficiency and the compressor energy balance, for a heat pump using two-phase suction to reduce discharge temperature. An experimental study has been carried out to measure the suction vapour quality and to test the method on a recent air-to-air heat pump using R32.

First, a correlation giving the suction vapour quality in steady-state operating conditions was presented. It can be used to have a simple model of the two-phase suction control from the evaporating and condensing temperatures and the compressor frequency. The possibility to generalise it to other similar heat pumps needs to be evaluated.

Then, the experimental study showed that it is possible to obtain the suction vapour quality and the refrigerant mass flow rate making iterations on suction vapour quality and using the compressor energy balance. This method could allow compressor manufacturers to control precisely the two-phase suction process, in order to optimise the heat pump performances.

The method requires the compressor efficiencies knowledge, which can be obtained using the compressor performance maps provided by the manufacturer. Ideally, these maps should also be corrected to account for the effect of wet suction, but presently this requires refrigerant mass flow

measurement as generic correlations or methods to account for the impact of wet compression on global and volumetric efficiencies could not be found. These maps might however not always be available for an independent implementation, so the sensitivity of the method to the efficiency accuracy was estimated using 10 % under and overestimated volumetric and global efficiency values. The estimated suction vapour quality is kept within 2.5 % of the reference value, while the heating capacity estimation is close to 10 %. In any case, this method allows to make a much better estimation of the heat pump heating capacity compared to the compressor energy balance when the fluid is in two-phase state at the compressor suction port.

Therefore, in addition of two-phase suction optimisation, the proposed method to evaluate the mass flow rate could make in-situ performance monitoring possible for these heat pumps without needing a mass flow meter.

The experimental testing of the method was realized in steady-state operating conditions of an air-to-air heat pump in heating mode. It would be interesting to extend this study to the cooling mode. Moreover, its extension to transitory operating phases could allow to better understand the suction vapour quality variation after a compressor start-up, and to have a more accurate performance assessment during these phases.

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