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An Instrumented Method for the Evaluation of Compressor Heat Losses in Heat Pumps On-Field

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Abstract

Estimating the performances of air-to-air heat pumps (HPs) on-field is problematic given that measuring accurately the enthalpy and mass flow rate of air is challenging. A method based on the compressor energy balance can be used to estimate the coefficient of performance (COP) of different HP types, including air-to-air, on-field. In this method heat transfer from the compressor shell towards the ambient air must be evaluated. In this paper, a non-intrusive method, which estimates scroll and rotary compressor heat losses towards the ambient air, is presented. The number and location of temperature sensors required by the method in order to estimate the external thermal profile of compressor shell, is specified. The developed method can be integrated in the on-field performance measurement method. Coefficient of performance values for scroll and rotary compressors are estimated in two operating conditions and compared with reference COP values obtained from intrusive measurements. The outcome of this study is the number and location of surface temperature sensors in scroll and rotary compressors along with appropriate correlations for heat transfer coefficients in order to accurately estimate COP values of air-to-air HPs using the on-field performance measurement method.

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Keywords: heat pump; coefficient of performance; heat losses; compressor; scroll; rotary; on-field; performance.

1. Introduction

Due to high theoretical efficiency of residential heat pumps (HPs), their development is essential when attempting to reduce heating energy consumption in dwellings. However, current methods to evaluate HP efficiencies have become problematic since efficiencies are measured and established in controlled laboratory conditions [1]. Performance values obtained in such conditions may differ from the ones obtained on-field due to several factors, such as installation quality, design of the heating system, and climatic conditions. Nevertheless, measuring accurately on-field heating capacity and coefficient of performance (COP) of air-to-air HPs is difficult since measuring air enthalpies and specifically air mass flow rate on-field is challenging [2].

Internal refrigerant method, presented and validated in the work of Tran *et al.* [3], demonstrates that air-to-air HP efficiencies can be evaluated using mass/energy balances of the refrigeration components, and the measurements are done on the refrigerant. Such method requires only non-intrusive sensors, and is therefore well-adapted for in-situ performance evaluation. The method requires estimating compressor heat transfer towards the ambient air, herein after referred to as compressor heat losses. In the work of Tran *et al.* [3] it was shown that the share of heat losses in the sensitivity index of heating power uncertainty predicted by the internal

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refrigerant method is up to 40%. Thus, compressor heat losses influence significantly the overall accuracy of the method, and a more accurate evaluation of these losses on-field is required.

In this article a non-intrusive method to evaluate compressor heat losses on-field is presented. First, correlations available from literature that account for physical characteristics of heat transfer and compressor geometry have been carefully selected and tested. Then, a CFD model, introduced in the work of Goossens *et al.* [4], was used to determine the zones on scroll and rotary compressor shell that contribute the most to compressor heat losses, by investigating their shell thermal profile and heat flux distribution. And finally, temperature values used to calculate compressor heat losses, $\dot{Q}_{amb}$, obtained from different sensor locations were tested in scroll and rotary compressors. Calculated $\dot{Q}_{amb}$ values were then compared to reference values in order to select the optimal location in terms of accuracy. Compressor heat losses obtained from various sensor locations were also integrated in the internal refrigerant method to calculate the COP values, and compared to reference values obtained from intrusive measurements.

C_a	oil mass concentration	(-)
D	shell diameter	(m)
g	gravitational acceleration	($m\ s^{-2}$)
h	specific enthalpy	($J\ kg^{-1}$)
h	heat transfer coefficient	($W\ m^{-2}K^{-1}$)
k	thermal conductivity	($W\ m^{-1}K^{-1}$)
L	zone length	(m)
$\dot{m}$	mass flow rate	($kg\ s^{-1}$)
Nu	Nusselt number	(-)
n	number of thermal zones	(-)
P	pressure	(MPa)
Pr	Prandtl number	(-)
$\dot{Q}$	heating power	(W)
Ra	Rayleigh number	(-)
T	temperature	(K)
$\dot{W}$	power consumption	(W)
x	characteristic length	(m)
α	thermal diffusivity	($m^2\ s^{-1}$)
β	thermal expansion coefficient	(K^{-1})
η	heat loss factor	(-)
ν	kinematic viscosity	($m^2\ s^{-1}$)
σ	standard deviation	(-)
σ	Stefan-Boltzmann constant	($W\ m^{-2}K^{-4}$)
Subscript		
amb	ambient	
comp	compressor	
dis	discharge	
in	inlet	
out	outlet	
o	oil	
r	refrigerant	
Abbreviations		
CFD	computational fluid dynamics	
COP	coefficient of performance	
HP	heat pump	
R407C	refrigerant type	

2. Compressor heat losses

2.1. State.of-the-art of analytical compressor heat loss modeling

One approach is to model heat losses as a function of compressor power input, $\dot{W}_{comp}$ as did Fahlén [5], depicted in the following equation:

$$\dot{Q}_{amb} = \eta \dot{W}_{comp} \tag{1}$$

where η is the heat loss factor, assumed to be constant and known, and equal to 0.08. However, experimental data obtained from compressor test bench has shown that ambient heat losses are not always equal to 8% of the power input: η ranges 7-13% and 5-47% depending on the operating condition and rotation speed in scroll and rotary compressors, respectively.

Compressor heat losses can be represented using the shell and ambient temperature relationship, as in the work of Tran *et al.* [3], as follows:

$$\dot{Q}_{amb} = a\left(T_{shell} - T_{amb}\right) + b\left(T_{shell}^{4} - T_{amb}^{4}\right) \tag{2}$$

where T_{shell} is the shell temperature, T_{amb} is the ambient temperature, and factors a and b are obtained from compressor geometry and the nature of heat transfer. Factors a and b represent convective and radiative heat exchange, respectively. It was assumed that the shell temperature is homogenous and equal to discharge temperature, T_{dis}. However, shell temperature may depend on many factors, such as pressure ratio, suction pressure, discharge pressure/temperature. Correlations have been developed in previous works establishing a relationship between T_{shell} and other variables related to the operating conditions. For instance, Kim and Bullard [6] proposed a linear dependence between T_{shell} and T_{dis}. A linear correlation was suggested by Duprez *et al*. [7], where T_{shell} is a function of evaporation and condensation temperatures. An analysis of experimental data over a wide range of operating conditions conducted by Li [8] resulted in an observation that the pressure ratio is insufficient to accurately predict the shell temperature, and that discharge pressure is also required. A non-linear correlation was suggested, where T_{shell} is a function of the two mentioned variables.

Establishing a correlation, that accurately represents T_{shell} taking into account the physical aspects of heat exchange and compressor geometry, yet remaining relatively simplified in terms of unknown parameters, and having a good extrapolation capacity, is challenging. In addition, assuming that shell temperature is homogenous is problematic, specifically in the case of scroll compressors with compression chamber on top, as strong temperature fluctuations take place along the shell.

2.2. Average Nusselt number correlations

Heat exchange between the exterior envelope of the compressor and environment occurs through radiation and convection. Typically, heat exchange by conduction can be considered negligible. It is assumed that convection can either be forced, natural, or mixed. Natural convection is believed to occur when compressor exposed to surrounding air has no ventilator generating ambient air velocity in its close proximity, *i.e.* no external force is generating air movement, unlike in the case of forced convection. Change in fluid temperature (in this case air) results in change of density, and in a gravitational field the lighter fluid is pushed upwards by buoyancy force, which is then replaced by a denser and cooler fluid. This results in air movement, thus heat transfer by convection becomes possible. On the other hand, mixed convection occurs when the buoyancy forces in forced flows are non-negligible, in other words neither natural nor forced convection dominate. Whether natural, forced, or mixed convection takes place around the compressor envelope depends on the configuration of compressor inside the heat pump exterior/interior unit.

Dimensionless Reynolds number, obtained from the following equation, is used to determine whether the flow is in laminar or turbulent regime when the fluid movement is generated by an external force:

$$Re_{x} = \frac{v_{air}\, x}{\nu} \tag{3}$$

where x is the characteristic length, which in our case is either the shell diameter, D, or length (vertical along the compressor shell), L, of the configuration, v_{air} is the air velocity, and υ is the kinematic viscosity.

Analogically to Reynolds number, a dimensionless Rayleigh number for natural convective heat exchange, is used to estimate the flow regime of air, and is calculated as follows:

$$Ra_x = \frac{g\beta\left(T_{shell} - T_{amb}\right)}{\nu\alpha} x^3 \tag{4}$$

where g is gravitational acceleration, β is thermal expansion coefficient, ν is kinematic viscosity, and α is thermal diffusivity. Material properties are considered at film temperature as follows:

$$T_{film} = \frac{T_{shell} + T_{amb}}{2} \tag{5}$$

For ideal gases (in our application air can be assumed to behave as an ideal gas), thermal expansion coefficient is calculated as depicted in the following equation:

$$\beta = \frac{1}{T_{film}} \tag{6}$$

A literature review of available correlations has yielded the following correlations for isothermal surfaces, which are best fitted to the nature of flow and compressor geometry in question. For the lateral part of the cylinder Churchill and Chu [9] correlation for **natural** convection on vertical plates was used:

$$\overline{Nu}_L = \left(0.825 + \frac{0.387 Ra_L^{1/6}}{\left(1 + \left(0.492/\mathrm{Pr}\right)^{9/16}\right)^{8/27}}\right)^2 \qquad \left(10^9 \le Ra_L < 10^{12}\right) \tag{7}$$

for turbulent flows, where Pr is Prandtl number and L is the lateral zone length. And for laminar flows the correlation is adapted, as follows:

$$\overline{Nu}_L = \left(0.68 + \frac{0.67 Ra_L^{1/4}}{\left(1 + \left(0.492/\mathrm{Pr}\right)^{9/16}\right)^{4/9}}\right)^2 \qquad \left(Ra_L \le 10^9\right) \tag{8}$$

Only the correlation for laminar flows was used in this study, as the Rayleigh number fell within the limits of laminar flow for two operating conditions investigated.

McAdams' correlation for **natural** flows on hot horizontal circular plates facing up was used for compressor top and bottom surfaces [10]:

$$\overline{Nu}_D = 0.54 Ra_D^{1/4} \qquad \left(10^5 \le Ra_L \le 2\cdot 10^7\right) \tag{9}$$

for laminar flow and for turbulent flow, as follows:

$$\overline{Nu}_D = 0.14 Ra_D^{1/2} \qquad \left(2\cdot 10^7 \le Ra_L \le 3\cdot 10^{10}\right). \tag{10}$$

Correlations for both flows, turbulent and laminar, were employed in the calculations, depending on the temperature gradient between the shell top plate and ambient air.

The following correlation, suggested by Churchill and Bernstein [11] for **forced** flows across cylinders, was employed for the lateral side of the compressor shell:

$$\overline{Nu}_D = 0.3 + \frac{0.62\,\mathrm{Re}_D^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1+\left(0.4/\mathrm{Pr}\right)^{2/3}\right]^{1/4}} \qquad \left(\mathrm{Re}_D < 10^4\right) \tag{11}$$

and for horizontal plates, a correlation for average Nusselt number developed for **forced** laminar flows along flat plates was used [10]:

$$\overline{Nu}_D = 0.664 Re_D^{1/2} Pr^{1/3} \qquad \left(10^3 < Re_D < 5\cdot 10^5\right) \tag{12}$$

Convective heat transfer coefficient is then obtained from the following equation:

$$\bar{h}_x = \frac{\overline{Nu}_x k}{x} \tag{13}$$

where k is the thermal conductivity of air. Heat losses towards the ambient air can then be obtained from the following equation:

$$\dot{Q}_{amb,i} = \bar{h}_{c,i} A_i \left(T_{shell,i} - T_{amb}\right) + \sigma A_i \left(T_{shell,i}^4 - T_{amb}^4\right) \tag{14}$$

where A_i is the compressor shell area and σ is the Stefan-Boltzmann constant. Compressor surface emissivity is considered to be equal to unity and the emissivity of surrounding air can be neglected.

2.3. Calculation method

As mentioned in sub-section 2.2, the correlations presented in Eqs. 7-12 are developed strictly for isothermal surfaces. While rotary compressors tend to have a homogenous temperature along the entire shell, scroll compressors with a compression chamber on top (motor assembly on the bottom) tend to exhibit strong temperature fluctuations between the low and high pressure zones. The investigation of thermal profiles and heat fluxes from compressor shell towards ambient air in scroll compressors in various operating conditions using a numerical simulation model introduced in [4], permitted to determine zones on compressor envelope that contribute to heat losses the most. Significant temperature fluctuations were noticed along the envelope of scroll compressors: a strong increase in temperature occurs at the level of compression chamber. In fact, 80% of heat losses in scroll compressors occur from the lateral zones that are at the level of compression chamber, discharge plenum, and the top horizontal plate, when $T_{cond} = 40$ ℃, $T_{evap} = 0$ ℃, $T_{amb} = 10$ ℃. Figure 1 depicts the heat transfer shell zones taken in consideration in scroll and rotary compressors, as well as the characteristic lengths for Rayleigh and Reynolds numbers.

With the aid of numerical simulation model introduced in [4], the average velocity of surrounding air is determined as $v_{air} = 1.2\ m/s$. A ratio between Gr_x/Re_x^2, where $Gr_x = Ra_x/Pr$, showed that heat is exchanged by means of mixed convection [10]. Nusselt number correlation for mixed flow along a horizontal heated plate facing up with a transverse buoyancy force, as listed below, was used for horizontal plates:

$$\overline{Nu}^{7/2} = \overline{Nu}_f^{7/2} + \overline{Nu}_n^{7/2} \tag{15}$$

and the correlation for cross-flow over an immersed cylinder with a transverse buoyancy was used for the lateral side:

$$\left(\overline{Nu} - 0.3\right)^4 = \left(\overline{Nu}_f - 0.3\right)^4 + \left(\overline{Nu}_n - 0.3\right)^4 \tag{16}$$

where $\overline{Nu}_n$ and $\overline{Nu}_f$ is the average Nusselt number for natural and forced convection, respectively.

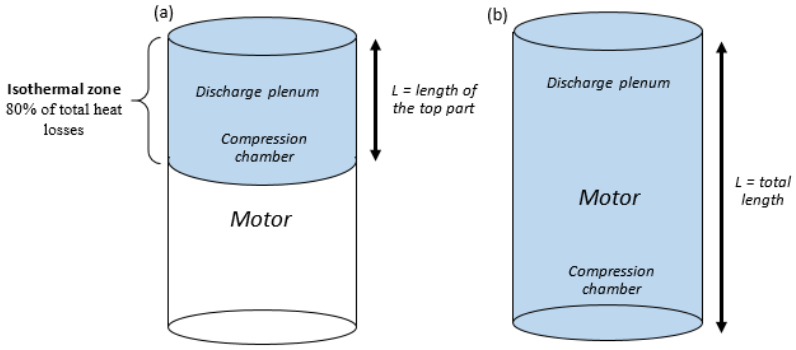


Fig. 1. Zones of scroll compressor envelope taken into consideration when estimating heat losses on-field for scroll (a) and rotary (b) compressors.

3. On-field instrumentation

3.1. Sensor locations for scroll and rotary compressors

During experimentations on scroll and rotary compressor test benches various number of different sensor locations were tested. Table 1 depicts at which internal component level was the sensor situated for scroll and rotary compressors, where T_{dis} is the refrigerant temperature at discharge, implying that zero sensors were placed on the compressor shell.

Table. 1. Temperature sensors placed on scroll and rotary shell

SCROLL	Discharge plenum	Top plate	-	T_{dis}
ROTARY	Center of motor assembly	Compression chamber	Discharge plenum	T_{dis}

Measured temperatures and their corresponding surface areas were then used to calculate the compressor convective and radiative heat transfer coefficients to obtain heat losses, as depicted in sub-section 2.2. The analysis was performed at two operating conditions, $T_{amb} = 10$ °C, $T_{evap} = 0$ °C, and $T_{cond} = 40$ °C and 60 °C, at 30 rps. The refrigerant fluid used was R407C. Temperature and mass flow measurements were acquired using thermocouples and a Coriolis flow meter.

4. Comparison with experimental data and discussion

Compressor energy balance, presented in the work of Tran *et al.* [3], was used to calculate experimental heat losses using the data obtained from compressor test bench, as follows:

$$\dot{Q}_{amb,\exp} = \dot{W}_{comp} - \dot{m}\left[\left(1 - C_g\right)\left(h_{r,comp,out} - h_{r,comp,in}\right) + C_g \Delta h_o^{T_{comp,in} \to T_{comp,out}}\right] \quad (17)$$

where $h_{r,comp,out}$ and $h_{r,comp,in}$ are refrigerant enthalpies at compressor outlet and inlet, respectively, and $\dot{W}_{comp}$ is compressor power input, $\Delta h_o^{T_{comp,in} \to T_{comp,out}}$ is the enthalpy change of oil and C_g is the oil mass fraction. Mass flow rate of refrigerant was measured directly with a mass flow meter during the tests. Error propagation formula, listed in the following equation, was used to determine the absolute error of experimental values:

$$\sigma_{\dot{Q}_{amb}} = \sqrt{\sum_i \left(\frac{\delta \dot{Q}_{amb}}{\delta n_i}\right)^2 \sigma_{n_i}^2} \quad (18)$$

where n_i is the measured variable, σ_{n_i} is the absolute uncertainty of the measured quantity. Wattmeter, flow meter, pressure and surface temperature sensor uncertainties were taken into consideration and their uncertainties are given in Table 2. Reference COP values were obtained from the following equation:

$$COP_{ref} = \frac{\dot{m}\left[\left(1-C_g\right)\left(h_{r,cond,in}-h_{r,cond,out}\right)+C_g \Delta h_o^{T_{cond,out}\rightarrow T_{cond,in}}\right]}{W_{HP}} \tag{19}$$

where $h_{r,cond,out}$ and $h_{r,con,in}$ are refrigerant enthalpies at condenser outlet and inlet, respectively, and $\dot{W}_{HP}$ is compressor power input, $\Delta h_o^{T_{cond,out}\rightarrow T_{cond,in}}$ is the enthalpy change of oil at the condenser.

Table 3 and 4 present convective heat transfer coefficients and the discrepancies between experimental and calculated heat loss values at different sensor locations for rotary and scroll compressors, at $T_{amb} = 10$ ℃, $T_{evap} = 0$ ℃, and $T_{cond} = 40$ ℃ and 60 ℃, respectively, at 30 rps, in mixed flows. The most optimal sensor location in terms of accuracy is at the middle of motor assembly level in rotary compressor, and for scroll compressors the measurement of discharge temperature is sufficient, *i.e.* no additional sensors are required. Figure 2 (a) compares experimental heat loss values with calculated values of scroll compressor obtained when shell temperature is assumed to be T_{dis}, expressed as a percentage of compressor power input for confidentiality reasons. Similarly, Figure 2 (b) compares experimental heat loss values with calculated values of rotary compressor obtained when shell temperature is measured at the middle of motor.

Relative deviations of COP values, obtained by integrating heat losses presented in Figs. 2 (a) and (b) for scroll and rotary compressors, respectively, from the reference values are 2% and 3% at $T_{cond} = 40\ °C$ and $60\ °C$, respectively, for scroll compressor and 5% and 3% at $T_{cond} = 40\ °C$ and $60\ °C$, respectively, for rotary compressor.

Table 2. Measurement sensors and the associated uncertainties

Sensor	Uncertainty	
	Relative	Absolute
Mass flow	0.2%	
Temperature		0.6 K
Pressure		comp,in: 0.005 MPaG comp,out: 0.009 MPaG
Power input	0.08% of reading + 0.08% of range	

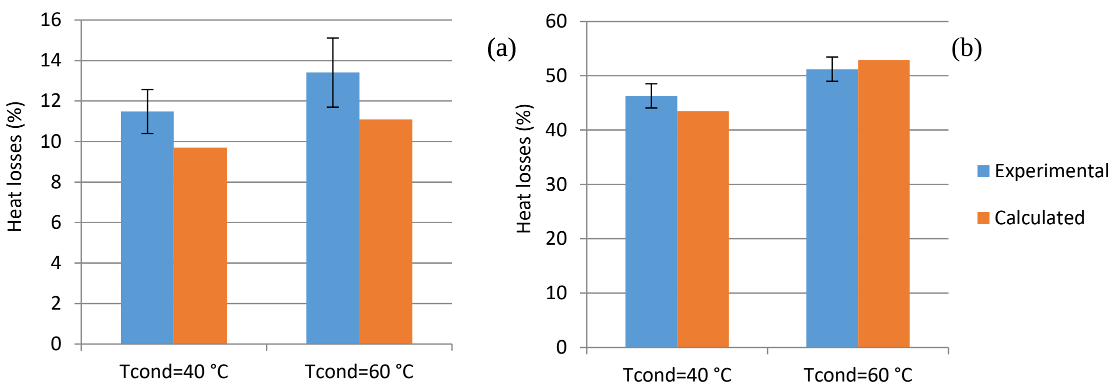
Table 3. Convective heat transfer coefficients and heat loss deviations in scroll compressors

	$T_{cond} = 40$ ℃			$T_{cond} = 60$ ℃		
T_{shell}	h_x (W m^{-2}K^{-1})		$\Delta\dot{Q}_{amb}$ (%)	h_x (W m^{-2}K^{-1})		$\Delta\dot{Q}_{amb}$ (%)
	top	lateral		top	lateral	
discharge plenum	10.69	8.67	24	10.90	8.78	26
top plate	10.71	8.70	20	10.92	8.79	22
fluid at discharge	10.73	8.70	15	10.94	8.80	17

Table 4. Convective heat transfer coefficients and heat loss deviations in rotary compressors

	$T_{cond} = 40$ ℃		$T_{cond} = 60$ ℃	
T_{shell}	h_x (W m^{-2}K^{-1})	$\Delta\dot{Q}_{amb}$ (%)	h_x (W m^{-2}K^{-1})	$\Delta\dot{Q}_{amb}$ (%)

	top/bottom	lateral		top/bottom	lateral	
discharge plenum	12.82	12.61	18	12.91	13.56	6
middle of motor	12.84	12.79	6	12.92	13.72	3
compression chamber	12.65	12.82	15	12.90	13.59	5
fluid at discharge	12.83	12.69	13	12.91	13.57	6



5. Conclusions

Heat losses were estimated using various surface temperature sensor location along the compressor envelope for scroll and rotary compressors in two operating conditions. The obtained heat losses were compared to experimentally derived values, and most optimal combinations in terms of result accuracy were selected. There are no additional sensors required to estimate heat losses of scroll compressor, temperature measurement of refrigerant at compressor outlet (discharge) is sufficient, in order to achieve a relative deviation from $\dot{Q}_{amb,exp}$ of 15% and 17% for $T_{cond} = 40\ °C$ and $60\ °C$, respectively. The relative deviation of COP using the performance measurement method from intrusively obtained reference value is 2% and 3% for $T_{cond} = 40\ °C$ and $60\ °C$,

Fig. 2. Experimental heat loss values with associated uncertainties in comparison with calculated values expressed as a percentage of compressor power input for confidentiality reasons, at two operating conditions in scroll (a) and rotary (b) compressors

respectively. On the other hand, for rotary compressors one additional sensor, at the level of middle of motor assembly is required to achieve a relative deviation from $\dot{Q}_{amb,exp}$ of 6% and 3% for $T_{cond} = 40\ °C$ and $60\ °C$, respectively. Thus, the relative deviation of COP value from COP_{ref} in rotary compressors is 5% and 3% for $T_{cond} = 40\ °C$ and $60\ °C$, respectively.

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