MASS FLOW RATE EVALUATION OF A HERMETIC COMPRESSOR FROM A SMALL SCALE VAPOUR COMPRESSION REFRIGERATION SYSTEM

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The paper presents a mathematical model for mass flow rate evaluation of a hermetic compressor mounted on a small scale vapour compression refrigeration system used in the HoReCa industry. A description of the experimental setup components and operation is given. The mathematical model is described and explained. A corresponding program has been developed and implemented in Engineering Equation Solver software. The flowchart is given. Validation of the mathematical model has been carried out. Experimental data have been used as input data. Results point out mass flow and cooling capacity variation due to changing operating conditions. Comments are given and future work is discussed.

Keywords: reciprocating hermetic compressor, mass flow rate, experimental

1. Introduction

The compressor is a very important component of a vapour compression refrigeration system (VCRS). One could say that it is the "heart" of the refrigeration systems. Most of the small scale VCRSs use hermetic reciprocating compressors (HRC). A very important parameter which describes the operation of compressors

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in general, and that of HRC in particular, is the refrigerant mass flow rate. Using the value of the mass flow rate, the energy efficiency of a VCRS can be evaluated in real operating condition. The common VCRS that can be purchased from retailers, do not have a refrigerant mass flow meter installed and thus making their energy efficiency evaluation, in real operation conditions, very difficult. In laboratory conditions, a refrigerant mass flow meter could be installed. This is the easiest solution, but also the most expensive one. Another way of determining the mass flow rate for a given HRC is reported in the present paper. A straight forward mathematical model that helps to evaluate the mass flow rate is described. The mathematical model is based on the constructive parameters of the HRC and also on the thermodynamic properties of the refrigerant given by the real operating conditions of the VCRS. The results obtained with the proposed mathematical model are compared with the information presented by the manufacturer in the datasheet of the compressor.

The mathematical modelling and simulation of existing reciprocating compressors have been the subject of various papers, as presented below. In general, the simulation of the reciprocating compressors is done by taking into consideration the pressure drops, heat transfer processes and the refrigerant fluid dynamics. The results are compared with experimental data. Based on the results, if needed, the simulation is calibrated [1]. When it comes to VCRS, the mathematical modelling of the reciprocating compressor is carried out in closed connection with it. In other words, the thermodynamic properties of the refrigerant during the refrigeration cycle must be integrated into the mathematical equations which describe the operation of the reciprocating compressor. A good mathematical model of the reciprocating compressor can help to predict the general behaviour of the VCRS [2]. A key thermodynamic property of the refrigerant is its temperature. The value of the temperature of the refrigerant has great importance on the type of materials used for the construction of the reciprocating compressor. Measuring the temperature inside a reciprocating compressor during real-time operation can prove to be challenging [3]. The temperature difference between the reciprocating compressor parts can help to better understand the overall heat transfer mechanism [4]. Authors of [5] have developed a mathematical model which simulates the heat transfer for small HRCs. Two cases of HRC have been considered i) with semi-direct intake and ii) direct intake. The HRC has been divided into six control volumes. The results have been compared experimentally using R290 as a refrigerant. Authors point out that the value of heat transfer in the discharge line is up to 3-5 times larger than the one in the discharge chamber. For the temperature values, also, the authors show that there is a ±20% difference between the temperature values obtained with the model and the one obtained experimentally. The results obtained for the mass flow rate showed a ±10% when compared with the experimental data. In [6], the authors have developed a simulation program for the transient operation of a domestic refrigeration system. The results have been compared with experimental data. Good agreement has been obtained. Other researchers focused on the connection between the efficiency of HRCs and the type of refrigerant being used [7-9]. This type of work is mainly experimental but it can generate equations which better predict the efficiency of the HRCs [9]. In [10] and [11], authors have used a combined theoretical and experimental investigation method and applied it for different parts of the HRCs. A CFD analysis for a reciprocating compressor has been used by [12]. In [13] a simulation of reciprocating and screw compressors has been done. The aim to obtain the values for the refrigerant mass flow rate using experimental input data for the intake pressure and discharge pressure. The refrigerants used in the simulation are R12, R134a, R22 and R410A. The authors report a 5% difference between the mass flow rate values obtained by simulation and those obtained experimentally.

The paper is structured as follows: Section 2 presents a brief description of the experimental setup and input data, Section 3 presents the mathematical model developed for the mass flow rate evaluation and cooling capacity estimation and Section 4 presents the validation of the mathematical model. Conclusions and future work are presented in Section 5.

2. Brief description of the experimental setup

The evaluation of the refrigerant mass flow rate is in a very strong connection with the type of the reciprocating compressor and the thermodynamic properties of the refrigerant. In this context, a brief description of the experimental setup is given first. The instrumentation is also mentioned with an emphasis on the thermodynamic parameters to be used in the mathematical model for the evaluation of the refrigerant mass flow rate.

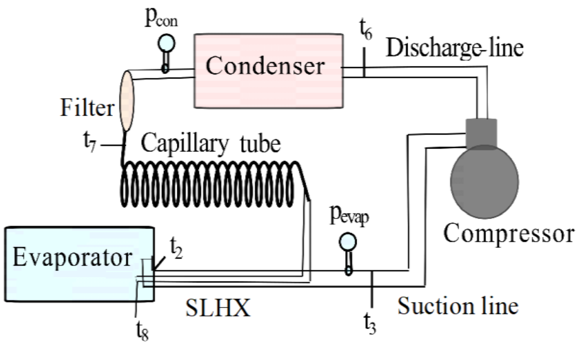

Fig. 1 Schematic of the experimental setup

Fig. 1 presents a schematic of the experimental setup. The experimental setup is a small scale VCRS comprised of the evaporator, hermetic reciprocating compressor, hot wall condenser, filter, capillary tube and a suction line heat exchanger (SLHX).

An overview image of the experimental setup is presented in Fig. 2. The experimental setup can be found in the Refrigeration Laboratory, Department of Thermodynamics, Engines, Thermal and Refrigeration Equipment, Faculty of Mechanical and Mechatronics Engineering from University POLITEHNICA of Bucharest. Basically, the experimental setup is a small scale VCRS, a small refrigerator used in HoReCa industry. As one can notice from Fig. 2 b), the condenser of the refrigeration system is not visible. This is because the type of condenser used is hot wall condenser. This means that the condenser is actually inserted in the sidewalls of the refrigerator. A detailed description of the experimental setup is not the aim of this work, but the main measurement points are presented below.

a) b)

Fig. 2. Overview image of the experimental setup a) front view b) back view

The experimental setup is an instrument in such a way that it allows to determine: the evaporating pressure p_{evap} *[bar]*, condensing pressure p_{con} *[bar]*, the temperature at evaporator outlet t_2 *[°C]*, temperature of the refrigerant after the SLHX t_3 *[°C]*, temperature of the refrigerant at the entrance into the condenser t_6 *[°C]*, temperature of the refrigerant at the entrance into the capillary tube t_7 *[°C]*, energy consumption *E [kWh]* and operation time τ *[h]*. The type of refrigerant used is R600a.

Fig. 3 The thermodynamic cycle of the experimental setup in the p-h diagram.

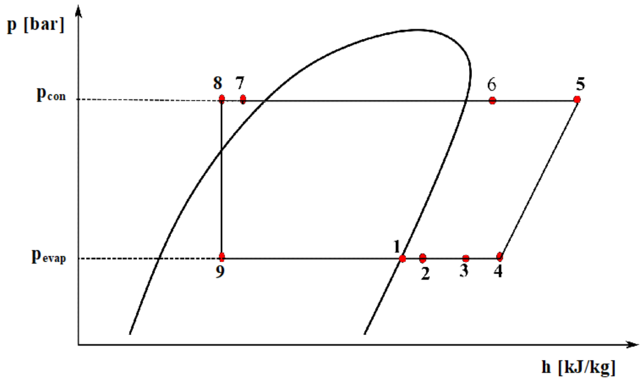

Fig. 3 presents the thermodynamic cycle of the experimental setup in the p-h diagram. The operation of the experimental setup is presented below in connection with Fig. 1 and Fig. 3. The refrigerant enters into evaporator having the thermodynamic state 9. In the evaporator, the refrigerant undergoes two consecutive processes namely: an evaporating process 9-1 followed by a superheating process 1-2. With the thermodynamic state 2, the refrigerant enters into the SLHX where it undergoes another superheating process 2-3. With the thermodynamic state 3, the refrigerant enters into the capsule of the HRC where it absorbs heat from the electric motor of the compressor an undergoes yet a different superheating process 3-4. This superheating process is not experimentally evaluated in this paper but it is taken into consideration by adopting a superheating degree between states 3 and 4. With the thermodynamic state 4, the refrigerant enters into the compressor cylinder where the compression process 4-5 takes place. After the compressor discharge, the refrigerant flows to the condenser. On the pipe which connects the compressor and the condenser, the refrigerant undergoes a subcooling process 5-6. With the thermodynamic state 6, the refrigerant enters into the condenser where the heat rejection takes place. In the condenser, usually, the refrigerant undergoes also a subcooling process. This process is not evaluated experimentally in the present work. On the pipe that connects the condenser and the filter and also in the filter the refrigerant undergoes another subcooling process due to heat rejection to the surroundings. The temperature of the refrigerant after the filter and at the entrance into the capillary tube is measured in state 7. This temperature allows the estimation of the subcooling degree between the end of the condensing process and the entrance into the capillary tube. In the capillary tube, the expansion process takes place and this process is overlapping with another subcooling process that occurs in the SLHX, 7-8. For simplicity and ease of understanding the two processes subcooling 7-8 and expansion in the capillary tube 8-9 are presented separately. The SLHX is a concentric heat exchanger in which the capillary tube is placed inside the suction line. Here a heat exchange between the hot refrigerant flowing through the capillary tube and the cold refrigerant flowing in the suction line takes place. The first is subcooling and the latter is superheating.

The heat load inside the refrigerator is simulated using a static heater with adjustable power. The nominal power of the heater is 150 W and it is connected to the power regulator. The power of the static heater can be regulated between 0 and 100%.

The next section of the paper presents the mathematical model for the evaluation of the refrigerant mass flow rate.

3. Mathematical model for the evaluation of the refrigerant mass flow rate and cooling capacity estimation

The mathematical model [14] will start from the expression of the mass flow rate:

$$\dot{m} = \rho_4 \cdot \dot{V}_{real} = \frac{1}{v_4} \cdot \dot{V}_{real} \tag{1}$$

In Eq. (1) ρ_4 *[kg/m^3]* is the density of the refrigerant at the compressor inlet, state 4 from Fig. 3; $\dot{V}_{real}$ *[m^3/s]* is the real volume flow rate of the HRC and v_4 *[m^3/kg]* is the specific volume of the refrigerant in state 4.

The real volume flow rate of the compressor $\dot{V}_{real}$ *[m^3/s]* can be computed using the following general relationship:

$$\dot{V}_{real} = \lambda \cdot \frac{\pi \cdot D^2}{4} \cdot S \cdot \frac{n_r}{60} \cdot i \cdot z \tag{2}$$

Where λ *[-]* is the volume flow reduction coefficient; D *[m]* is the cylinder diameter; S *[m]* is the piston stroke; n_r *[rot/min]* is the compressor speed; i *[-]* is the number of cylinders and z *[-]* is the number of compressors. For the experimental setup presented in the paper, there is only one compressor and thus $z = 1$, having only one cylinder $i = 1$. Eq. (2) can be rewritten for the present case as:

$$\dot{V}_{real} = \lambda \cdot \frac{\pi \cdot D^2}{4} \cdot S \cdot \frac{n_r}{60} \tag{3}$$

In Eq. (3) one can notice that $\dot{V}_{real}$ *[m^3/s]* depends on the constructive parameters D *[m]* and S *[m]* and on the functional parameter n_r *[rot/min]*. For a given VCRS, the case of the present work, these values are known.

The volume flow reduction coefficient λ *[-]* as to be determined as presented below.

$$\lambda = \lambda_0 \cdot \lambda_p \cdot \lambda_e \cdot \lambda_T \tag{4}$$

Where: λ_0 *[-]* is the volume flow reduction coefficient due to the existence of the dead space volume; λ_p *[-]* is the volume flow reduction coefficient due to the pressure losses in the distribution system of the compressor during the intake process; λ_e *[-]* is the volume flow reduction coefficient due imperfect sealing of the cylinder and λ_T *[-]* is the volume flow reduction coefficient due do the superheating of the refrigerant at the cylinder intake.

The volume flow reduction coefficient due to the existence of the dead space volume λ_0 *[-]* can be determined as:

$$\lambda_0 = 1 - \varepsilon_0 \cdot \left(H_c^{1/k} - 1\right) \tag{5}$$

In Eq. ε_0 [-] is the dead space volume coefficient and it can take values between *0.02* and *0.1*, depending on the cylinder size; *k [-]* is the adiabatic exponent and it depends on the type of refrigerant used. H_c *[-]* is the compression ratio and it can be computed as:

$$H_c = \left(p_{con} + \Delta p_{dis}\right) / \left(p_{evap} - \Delta p_{in}\right) \tag{6}$$

In Eq. Δp_{dis} is the pressure loss in the discharge process and Δp_{in} are the pressure losses in the intake process. These pressure losses can be determined as:

$$\begin{aligned} \Delta p_{dis} &= \left(0.08 \div 0.12\right) \cdot p_{con} \\ \Delta p_{in} &= \left(0.06 \div 0.1\right) \cdot p_{evap} \end{aligned} \tag{7}$$

For the present study, $\Delta p_{dis} = 0.08 \cdot p_{con}$ and $\Delta p_{in} = 0.06 \cdot p_{evap}$.

The volume flow reduction coefficient due to the pressure losses in the distribution system of the compressor during the intake process λ_p *[-]* can be determined as:

$$\lambda_p = 1 - \left(\Delta p_{in} / p_{evap}\right) \tag{8}$$

The volume flow reduction coefficient due to imperfect sealing of the cylinder λ_e *[-]* and the volume flow reduction coefficient due do the superheating of the refrigerant at the cylinder intake λ_T *[-]* are evaluated for a given compressor. For example, λ_e *[-]* is around 0.96 and λ_T *[-]* around 0.93 for larger compressors. For the present case, the considered values are $\lambda_e = 1$ and $\lambda_T = 1$. Thus, Eq. (4) which gives the volume flow reduction coefficient λ *[-]* can be rewritten as follows:

$$\lambda = \lambda_0 \cdot \lambda_p \tag{9}$$

Knowing the refrigerant mass flow rate, an estimation of the cooling capacity of the experimental small VCRS can be done as presented next.

The cooling capacity $\dot{Q}_0^{\exp}$ *[W]* is given by:

$$\dot{Q}_0^{\exp} = \dot{m} \cdot (h_2 - h_9) = \dot{m} \cdot q_0 \quad (10)$$

Where h_2 *[kJ/(kg K)]* and h_9 *[kJ/(kg K)]* are the refrigerant specific enthalpies at the inlet and outlet of the evaporator, respectively.

As one can notice, the mathematical model is in very close connection with the thermodynamic cycle presented in Fig. 3. Next, a discussion regarding how the thermodynamic cycle can be established starting from the experimental data is given.

The evaporating pressure p_{evap} *[bar]*, the condensing pressure p_{con} *[bar]*, the temperature at the evaporator outlet t_2 *[°C]*, the temperature of the refrigerant after the SLHX t_3 *[°C]*, the temperature of the refrigerant at the entrance into the condenser t_6 *[°C]* and temperature of the refrigerant at the entrance into the capillary tube t_7 *[°C]* are determined experimentally. With these values, the following thermodynamic states of the refrigerant can be determined, in connection with Fig. 3.

$$\begin{aligned}
&State\,1 \rightarrow p_{evap} \cap x = 1 \rightarrow t_1; h_1; v_1; s_1 \\
&State\,2 \rightarrow p_{evap} \cap t_2 \rightarrow h_2; v_2; s_2 \\
&State\,3 \rightarrow p_{evap} \cap t_3 \rightarrow h_3; v_3; s_3 \\
&State\,4 \rightarrow p_{evap} \cap (t_3 + \Delta t_{sh}) \rightarrow h_4; v_4; s_4 \\
&State\,5 \rightarrow p_{con} \cap (s_4 = s_5) \rightarrow h_5; v_5; t_5 \\
&State\,6 \rightarrow p_{con} \cap t_6 \rightarrow h_6; v_6; s_6 \\
&State\,7 \rightarrow p_{con} \cap t_7 \rightarrow h_7; v_7; s_7 \\
&State\,8 \rightarrow p_{con} \cap h_8 \rightarrow t_8; v_8; s_8 \\
&State\,9 \rightarrow p_{evap} \cap (h_9 = h_8) \rightarrow t_9; v_9; s_9; x_9
\end{aligned} \quad (11)$$

In expressions (11), Δt_{sh} *[K]* is the superheating degree which takes place in the capsule of the HRC. It cannot be determined on the present experimental setup but it is taken into consideration by adopting a value of $\Delta t_{sh} = 15$ *[K]* [14]. The value for the enthalpy in state 8 can be computed from the equality:

$$h_7 - h_8 = h_3 - h_2 \quad (12)$$

Knowing the value for h_8 *[kJ/(kg K)]* will give the subcooling degree Δt_{sc} *[K]* in the SLHX, as:

$$\Delta t_{sc} = t_7 - t_8 \quad (13)$$

Based on the mathematical model presented before, a program has been developed in Engineering Equation Solver software [15]. A flow chart of the program is given below.

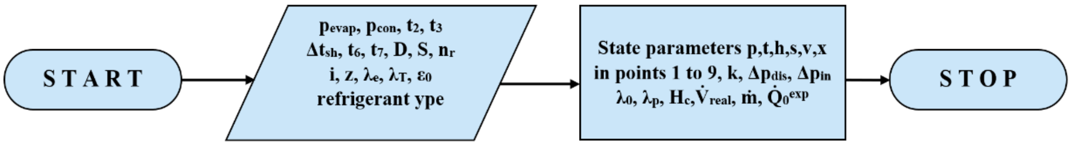

Fig. 4 Flowchart of the EES program

4. Validation of the mathematical model and results

The constructive characteristics of the HRC mounted on the experimental setup have been determined by dismounting an identical one. The type of compressor is the GMCC model LBP SZ55C1J. The refrigerant used is R600a. Detailed technical data about the compressor can be found at [16].

The following have been determined: diameter of the cylinder $D=0.021$ *m*; piston stroke $S=0.016$ *m* and the dead space volume coefficient has been adopted to be $\varepsilon_0=0.02$. For the small HRC type compressors, the speed is usually $n_r=3000$ *rot/min*.

The validation of the mathematical model will be done by comparing the cooling capacity $\dot{Q}_0$ *[W]* returned by the EES program presented in Fig. 4 adapted to the data given by the manufacturer of the compressor [16]. The input data in the EES program is the SHRAE test data mentioned by the manufacturer [16] which is: evaporating temperature $t_{evap} = -23.3\ ^oC$, ambient temperature $t_{amb} = 32.2\ ^oC$, condensing temperature $t_{con} = 54.4\ ^oC$, suction temperature $t_4 = 32.2\ ^oC$ and the subcooling temperature $t_7 = 32.2\ ^oC$.

The producer of the compressor, for the test conditions mentioned before, is specifying a cooling capacity of $\dot{Q}_0 = 85\ W$ and the cooling capacity obtained with the mathematical model is $\dot{Q}_0^{exp}=76.19\ W$. From the results, one can notice that there is a 10 % relative error between $\dot{Q}_0$ and $\dot{Q}_0^{exp}$. These values show a good agreement between the two. The result is also similar to the data reported in [5].

Table 1

Experimental data and results

Experimental test number	Heater Load [%]	Experimental data						
		p_{con} [bar]	p_{evap} [bar]	t_9 [^{o}C]	t_2 [^{o}C]	t_3 [^{o}C]	t_4 [^{o}C]	t_7 [^{o}C]
Test 1	25	6.53	0.47	-27.3	-23.07	18.03	28.3	36.88
Test 2	50	6.46	0.47	-27.3	-22.85	17.42	27.42	36.47
Test 3	75	6.41	0.47	-27.3	-24.81	13.82	23.8	35.22

The experimental input data for the mathematical model is presented in Table 1. Table 2 presents the obtained results. The experimental data presented in table 1 are very different from the test conditions indicated by the compressor manufacturer [16].

Table 2

Results

Experimental test number	Heater Load [%]	Results			
		λ [-]	q_0 [kJ/(kgK)]	$\dot{Q}_0^{exp}$ [W]	$\dot{m}\cdot 10^{-3}$ [kg/s]
Test 1	25	0.7272	300.2	66.74	0. 2223
Test 2	50	0.7297	300.2	67.12	0.2235
Test 3	75	0.7318	297.5	67.55	0.227

From Table 1 one can notice that as the heat load increases the temperature of the refrigerant at the evaporator outlet (t_2) tends to increase. This means higher superheating degree in the evaporator. Having thermodynamic state 2, the refrigerant enters into SLHX where the superheating process 2-3 takes place. Experimental data shows that for each test, the temperature at the end of the SLHX is different which will yield different temperatures at the compressor intake (state 4). From Table 1, one can also notice that there is no variation in the evaporating (p_{evap}) and a small variation of the condensing pressures (p_{con}). The small variation of p_{con} will lead to slightly different values for the flow rate coefficient λ. As λ increases, the mass flow rate will increase. From Table 2, the results show that the heat load q_0 has similar values for all 3 tests. In these conditions, the change in the mass flow rate will yield a change in the cooling capacity $\dot{Q}_0^{\mathrm{exp}}$. If the results are looked at in a global way, one can say that the highest cooling capacity is obtained when the compression ratio is the smallest. The smallest compression ratio is obtained for test 3 followed by test 2 and test 1. Another important conclusion is that the cooling capacity of the small scale VCRS is not constant during operation. It is strongly influenced by operating parameters such as evaporating pressure, condensing pressure, and compressor intake temperature t_4.

The results obtained for the refrigerant mass flow rate using the model presented in the paper can be used to further evaluate the performance of the refrigeration system (coefficient of performance). The present work can be used for those VCRS where a refrigerant mass flow rate sensor is not present. This is the case of most domestic, commercial, and industrial refrigeration systems. Future work will involve calibration of the model by installing, if possible, a refrigerant flow meter on the experimental setup and thus comparing the results. Using the cooling capacity and the energy consumption, the real coefficient of performance of the VCR can be evaluated.

5. Conclusions

The paper presents a straightforward mathematical model that can be used to evaluate the refrigerant mass flow rate in the case of reciprocating compressors in general and hermetic reciprocating compressors in particular. The hermetic reciprocating compressor used in the present work is mounted on a small-scale experimental refrigeration system used in HoReCa industry. A schematic of the experimental setup is given, and the operation is explained in correlation with the thermodynamic cycle. A brief description of the experimental setup is given. The mathematical model is explained and based on it, a program has been developed. For the program, a flowchart is presented. According to the flowchart, the mathematical model is implemented in the Engineering Equation Solver software.

The mathematical model is validated using test data provided by the compressor manufacturer. The results showed a good agreement. The relative error between the data provided by the compressor manufacturer for the cooling capacity and the value obtained in the present study was about 10%. This error value has also been reported in other similar works.

After validation, the mathematical model was used based on the experimental data from the experimental setup as well as on the constructive parameters of the hermetic reciprocating compressor. The results indicate a difference in the mass flow rate and thus a difference in the cooling capacity. The highest value of the mass flow rate is obtained when the compression ratio is the smallest.

The compression ratio is not constant because the evaporating and condensing pressures are not constant. This will yield different values for the flow rate coefficient. Or, the higher the flow rate coefficient, the higher the mass flow rate and the higher cooling capacity are. An important conclusion is that during real operation conditions, the cooling capacity of the small-scale refrigeration system is not constant. It is strongly influenced by operating parameters.

The mathematical model can be used to evaluate the refrigerant mass flow rate. With this value, further performance indicators of the vapour compression refrigeration system can be determined. Future work will involve calibration of the model and installing a refrigerant mass flow meter sensor on the experimental setup, if possible, and thus comparing the results. The present mathematical model can be extended to another refrigeration system as most of them do not have a refrigerant mass flow meter installed and thus making the evaluation of their performance very difficult.

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