Modelling and Experimental Investigation of a Vapor Compression System under Steady State Regime

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***Abstract*—The main objective of this work is to establish a detailed modelling technique to predict the refrigerant conditions such as pressure and enthalpy of a Vapor Compression (VC) system. The steady state modelling techniques of VC systems suggested in many research works are usually not easy to reproduce due to lack of detailed methodology and to the multitude of analytical or computational schemes that could not be assessed objectively. This work has addressed this issue by introducing a modelling method developed from first principles and adaptable to different type of prediction problems. The validation of the model results with experiments was satisfactory. The model outputs such as the refrigerant evaporating pressure as well as the enthalpy at each junction are in agreement with experimental data. The proposed modelling technique could be adopted with other existing mathematical models of the components of a VC system. The modelling method could help to determine the optimal parameters of a VC system used to design and test optimal control strategies at low cost to improve the system's efficiency. This work could also be used for modelling of VC systems with complex configuration such as systems with single condenser and multiple evaporators.**

***Index Terms*—vapor compression, steady state, modelling, experiment**

I. Introduction

Manufacturers of Vapor Compression (VC) systems are working towards innovating more energy efficient systems to provide convenient cooling or heating effects. One way to address the aforementioned innovation is through a thorough understanding of the interaction between the parameters of VC systems. Critical processes could be identified through modelling in order to draw key design requirements [1]. Modelling of VC systems consists in investigating their variation so that optimal parameter values could be determined.

An early steady state heat pump model was proposed for system's performance simulation [2]. The model was subdivided in four components namely the condenser, thermostatic expansion valve, evaporator and compressor. The condenser model was subdivided in de-superheater, condenser and sub-cooler zones whilst the evaporator was subdivided in evaporating and superheating zones. The simulation results of the four model components were obtained with a FORTRAN subroutine. Some literature suggested a model predicting the component behavior in steady state regime of a refrigerating air-to-air system [3]. The evaporator and condenser models were derived from first principles along with empirical values. The compressor model was a hermetic type of compressor that evaluated the amount of heat absorbed and lost by the circulating fluid as well as the suction and discharge pressures. The aforementioned model was modified by incorporating a capillary expansion valve and a simpler compressor model [4]. The modified model was used for analytical evaluation of varying system performance due to component changes in heat pump system. FORTRAN subroutine was adopted for simulation of the condenser and evaporator. Subsequent improved model was proposed by enabling a simpler simulation of cooling operation, sub-cooling, superheating and pressure drop as well as an improvement in the convergence of simulation results [5].

Steady state modelling of air-to-air heat pump systems was performed using three FORTRAN simulation codes [6]. The first code could simulate various cycle arrangements of heat-pump to describe performance. The second code was used for evaluation of specific operating conditions of heat pump. The third code was used for simulation of heat pump operations. A more detailed steady state modelling of heat pump system [7] has been described by presenting the modelling principles with their benefits for simulation of an air-to-air heat pump system. The evaporator and condenser modelling were similar in the sense that identical assumptions such as the refrigerant stream heat transfer occurs only in the two-phase zone and uniformity of overall heat transfer coefficient along the heat exchangers. The evaporator model was subdivided in two heat transfer zones of the refrigerant stream whilst the condenser was subdivided in three zones. For both heat exchangers, pressure drop and heat transfer effects were correlated in each zone. The compressor on the other hand was modelled with manufacturer's data and performance curves.

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Steady state modelling of heat exchangers was conducted using a logarithmic mean temperature difference (LMTD) technique [8] to evaluate the rate of heat transfer between the circulating fluid and the heat exchanger shell. Unfortunately, this approach lacked accuracy in situations where the circulating fluid is changing phase. Additional model enabling performance simulation of air-to-air heat pump systems could also be found in the literature [9]. A limited amount of experimental data was used to validate such model and the compressor model was a map-based model developed with available manufacturer's data [10]. Zone model to evaluate the performance of an air-cooled condenser was suggested by subdividing the condenser in de-superheating, condensing and sub-cooling zones [11]. This method led to a set of non-linear equations resulting from the energy balance analysis in each zone for both the refrigerant and air streams as well as the effectiveness equation. The numerical results of the simulation converged with the experimental results.

Steady state behavior of a VC system with numerical approach was investigated using C++ code [12]. System equilibrium was numerically tracked by using input parameters [12]. Stable outputs were obtained by assuming the convergence of the refrigerant mass flow rate and the energy balance through the evaporator [12]. Attempt has been made to build a generic steady state model of fin and tube heat exchangers that could simulate any configuration of refrigerant circuits as well as evaluate the refrigerant distribution in the circuit and the rate of heat conduction through fins [13]. The steady state model was constructed on the basis of graph theory and a novel algorithm for computation called alternative iteration method was introduced to calculate separately the energy and momentum equations in order to reduce the simulation time of the model. The comparison of the modelling and experimental results of this investigation showed a ±10% error range.

Reviews of the available simulation technique for VC systems such as the model-based techniques and evaluation of thermodynamics properties of the refrigerant [14] has been reported. Innovative simulation techniques such as knowledge engineering methods and computational methods for nanofluids were also presented [14]. A novel simulation method of steady state modelling was suggested to improve the robustness, speediness and accuracy [15] of the simulation of ordinary as well as complex vapor compression systems. The modelling method adopted was a component-based algorithm [16]. This work led to more advanced means to determine the initial guess value for steady state modelling and the component evaluation time was reduced by about 40% whilst improvement on the robustness of the algorithm for solution tracking was achieved at 50%. Steady state model based on first principles and empirical equations for variable speed VC systems operating with R134a refrigerant [17] was reported. The model used input parameters to evaluate the condensing and evaporating pressures as well as the temperature of the secondary stream at condenser and evaporator outlets. The model predicted the performance of a chiller system with an error below ±10% [17].

Development of thermal as well as performance simulation of air-conditioning and refrigeration system of a service vehicle was undertaken using steady state mathematical modelling of vapor compression system [18]. The modelling results were in agreement with field and experimental data. The model was also used for investigation of system performance and acceptable results were used to implement a controller scheme to achieve optimized system performance. Significant improvement of the system coefficient of performance (COP) was noted under optimal control. Simulation method for complex configuration of VC systems [19] was reported. The simulation was developed from a component-based approach in which the components of the VC system were treated as black box and connected to one another through junctions and ports. The simulation was validated with experimental results with an error below ±10% [19]. Semi-analytical model of heat pump system used for water heating [20] has been introduced. The model evaluated the system performance without any detailed geometry of the components [20].The model was particularly useful in the sense that subcomponent models such as accumulator model were considered during simulation and the model's results were in agreement with experiments [20].

Regression-based modelling technique [21] for emerging air conditioning (AC) systems has been presented. The performance curves of the model were obtained from the system outputs or experimental data then used for energy simulation of a building. The regression-based modelling could be adopted as alternative modelling technique for emerging AC systems. Development of a model for performance simulation of a membrane based VC system [22] has been reported. This type of VC system has attracted a lot of attention in the research community as it could be an alternative for more environment friendly system for indoor air cooling. The performance simulation was undertaken with an initial state point model [23] and better results in terms of energy saving compared to commercial systems were found.

The models found in the literature are not usually easy to reproduce or implement due to their complexity as they either lack detailed derivation steps or experimental validation. This work is addressing this issue by using existing equations of the components of a VC system to provide a detailed model validated with experimental data. This work adopted the application of existing mathematical methods [24] to predict the parameter values of a VC system.

II. Steady State Modelling

Fig. 1 describes a conventional VC system with its junctions.

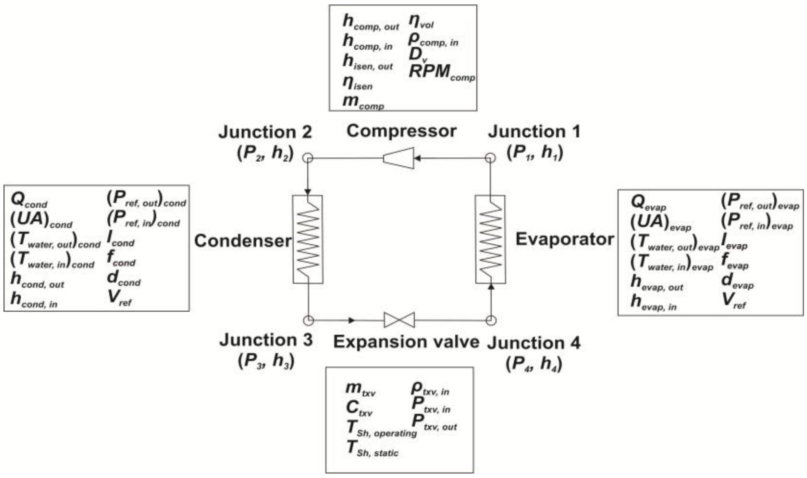


Figure 1. Junctions of a conventional vapor compression system

The compressor is used to raise the pressure of the circulating refrigerant, whilst the condenser rejects to the surrounding environment an amount of heat associated to the energy absorbed by the refrigerant during a cycle. An expansion valve is located downstream of the condenser in order to regulate the refrigerant flow prior to entering the evaporator whose function is to absorb heat from the surrounding environment to provide cooling effect. The junctions are used as interconnection between the components.

A. Compressor Modelling

The outlet condition and refrigerant mass flow rate at the compressor can be defined as follows:

$$h_{comp,out} = h_{comp,in} + \frac{h_{isen,out} - h_{comp,in}}{\eta_{isen}} \tag{1}$$

$$m_{comp} = \eta_{vol}\rho_{comp,in}D_v\frac{RPM_{comp}}{60} \tag{2}$$

$h_{comp,out}$ is the refrigerant enthalpy at compressor outlet in (kJ/kg). $h_{comp,in}$ is the refrigerant enthalpy at compressor inlet in (kJ/kg). $h_{isen,out}$ is the refrigerant isentropic enthalpy in compressor in (kJ/kg). η_{isen} is the isentropic efficiency of compressor. m_{comp} is the refrigerant mass flow rate in compressor in (kg/s). η_{vol} is the volumetric efficiency of compressor. $\rho_{comp,in}$ is the density of refrigerant at compressor inlet in (kg/m^3). D_v is the displacement volume of compressor in (m^3/s). RPM_{comp} is the rotational speed of compressor in (rpm).

B. Condenser Modelling

The heat rejected at the condenser could be evaluated as follows:

$$Q_{cond} = (UA)_{cond}[(T_{water,out})_{cond} - (T_{water,in})_{cond}] \tag{3}$$

Q_{cond} is the heat rejection in condenser in (kwh). $(UA)_{cond}$ is the overall heat transfer coefficient in condenser in $(kw/℃)$. $(T_{water,out})_{cond}$ is the water temperature at condenser outlet in (℃). $(T_{water,in})_{cond}$ is the water temperature at condenser inlet in (℃).

The outlet enthalpy of the refrigerant can be determined as follows:

$$h_{cond,out} = h_{cond,in} - \frac{Q_{cond}}{m_{comp}} \tag{4}$$

$h_{cond,out}$ is the refrigerant enthalpy at condenser outlet in (kJ/kg). $h_{cond,in}$ is the refrigerant enthalpy at condenser inlet in (kJ/kg).

The refrigerant outlet pressure could be expressed as follows:

$$(P_{ref,out})_{cond} = (P_{ref,in})_{cond} - f_{cond}\frac{2l_{cond}}{\pi d_{cond}^3}m_{comp}V_{ref} \tag{5}$$

$(P_{ref,out})_{cond}$ is the refrigerant pressure at condenser outlet in (kP_a). $(P_{ref,in})_{cond}$ is the refrigerant pressure at condenser inlet given in (kP_a). f_{cond} is the friction factor in condenser. l_{cond} is the condenser length in (m). d_{cond} is the inner diameter of condenser in (m). V_{ref} is the refrigerant speed in (m/s).

It is important to recall that the thermal behavior of the evaporator is similar to condenser's one but on the reverse side thus, its modelling equations have not been presented in this work.

C. Expansion Valve Modelling

The refrigerant mass flow rate at the thermostatic expansion valve can be derived as follows:

$$m_{txv} = C_{txv}(T_{Sh,operating} - T_{Sh,static})\sqrt{\rho_{txv,in}(P_{txv,in} - P_{txv,out})} \tag{6}$$

m_{txv} is the refrigerant mass flow rate in expansion valve in (kg/s). C_{txv} is the valve flow coefficient. $T_{Sh,operating}$ is the refrigerant operating superheat in (℃). $T_{Sh,static}$ is the refrigerant static superheat in (℃). $\rho_{txv,in}$ is the refrigerant density at expansion valve inlet in (kg/m^3). $P_{txv,in}$ is the refrigerant pressure at expansion valve inlet in

(kP_a). $P_{txv,out}$ is the refrigerant pressure at expansion valve outlet in (kP_a).

At steady state condition the refrigerant mass flow rate is balanced in the system and this could be described as follows:

$$m_{txv} = m_{comp} \tag{7}$$

Therefore, the refrigerant outlet pressure at the expansion valve is expressed as follows:

$$P_{txv,out} = P_{txv,in} - \frac{m_{comp}^2}{\rho_{txv,in}\left[C_{txv}(T_{Sh,operating} - T_{Sh,static})\right]^2} \tag{8}$$

III. Solution of the Steady State Modelling Equations

In this work pressure and enthalpy are determined to evaluate the refrigerant state at each junction. The inlet conditions are set at the inlet of each component whilst the outlet conditions have to be determined at the outlet.

A. Derivation of the Steady State Modelling Matrix

The refrigerant conditions (enthalpy and pressure) must be determined at each junction (Fig. 1). The unknown vector could be written as follows:

$$\overrightarrow{x_s} = [P_1 \quad P_2 \quad P_3 \quad P_4 \quad h_1 \quad h_2 \quad h_3 \quad h_4]^T \tag{9}$$

$\overrightarrow{x_s}$ is the output vector for steady state modelling. P_1 is the junction pressure at evaporator outlet or the predicted evaporating pressure in (kP_a). P_2 is the junction pressure at condenser inlet in (kP_a). P_3 is the junction pressure at condenser outlet or the predicted condensing pressure in (kP_a). P_4 is the junction pressure at evaporator inlet in (kP_a). h_1 is the junction enthalpy at evaporator outlet in (kJ/kg). h_2 is the junction enthalpy at condenser inlet in (kJ/kg). h_3 is the junction enthalpy at condenser outlet in (kJ/kg). h_4 is the junction enthalpy at evaporator inlet in (kJ/kg).

The solution of the system modelling should verify the following residual system of conservation equations at each junction:

$$\begin{bmatrix} (P_{ref,out})_{evap} \\ P_{comp,out} \\ (P_{ref,out})_{cond} \\ P_{txv,out} \\ h_{evap,out} \\ h_{comp,out} \\ h_{cond,out} \\ h_{txv,out} \end{bmatrix} = \begin{bmatrix} P_1 \\ P_2 \\ P_3 \\ P_4 \\ h_1 \\ h_2 \\ h_3 \\ h_4 \end{bmatrix} \tag{10}$$

$P_{comp,out}$ is the refrigerant pressure at compressor outlet in (kP_a). $h_{txv,out}$ is the refrigerant enthalpy at expansion valve outlet in (kJ/kg).

The outlet refrigerant conditions could be expressed in terms of the pressure and enthalpy at each junction as followed:

$$\begin{cases} (P_{ref,out})_{evap} = P_4 - f_{evap}\dfrac{2l_{evap}}{\pi d_{evap}^3} m_{comp} V_{ref} \\ P_{comp,out} = P_2 \\ P_{txv,out} = P_3 - \dfrac{m_{comp}^2}{\rho_{txv,in}\left[C_{txv}(T_{Sh,operating} - T_{Sh,static})\right]^2} \\ (P_{ref,out})_{cond} = P_2 - f_{cond}\dfrac{2l_{cond}}{\pi d_{cond}^3} m_{comp} V_{ref} \\ h_{evap,out} = h_4 + \dfrac{Q_{evap}}{m_{comp}} \\ h_{comp,out} = h_1 + \dfrac{h_{isen,out} - h_1}{\eta_{isen}} \\ h_{cond,out} = h_2 - \dfrac{Q_{cond}}{m_{comp}} \\ h_{txv,out} = h_4 \end{cases} \tag{11}$$

B. Steady State Modelling Matrix

The refrigerant pressure at the compressor outlet and enthalpy at the expansion device outlet must be set so that the modelling outputs could be determined. Substituting and re-arranging Eq. (11) into Eq. (10) leads to:

$$\begin{bmatrix} -1 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & -1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & -1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & -1 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & (\eta_{isen}-1) & -\eta_{isen} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & -1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} P_1 \\ P_2 \\ P_3 \\ P_4 \\ h_1 \\ h_2 \\ h_3 \\ h_4 \end{bmatrix} = \begin{bmatrix} f_{evap}\dfrac{2l_{evap}}{\pi d_{evap}^3} m_{comp} V_{ref} \\ P_{comp,out} \\ f_{cond}\dfrac{2l_{cond}}{\pi d_{cond}^3} m_{comp} V_{ref} \\ \dfrac{m_{comp}^2}{\rho_{txv,in}\left[C_{txv}(T_{Sh,operating} - T_{Sh,static})\right]^2} \\ -\dfrac{Q_{evap}}{m_{comp}} \\ -h_{isen,out} \\ \dfrac{Q_{cond}}{m_{comp}} \\ h_{txv,out} \end{bmatrix} \tag{12}$$

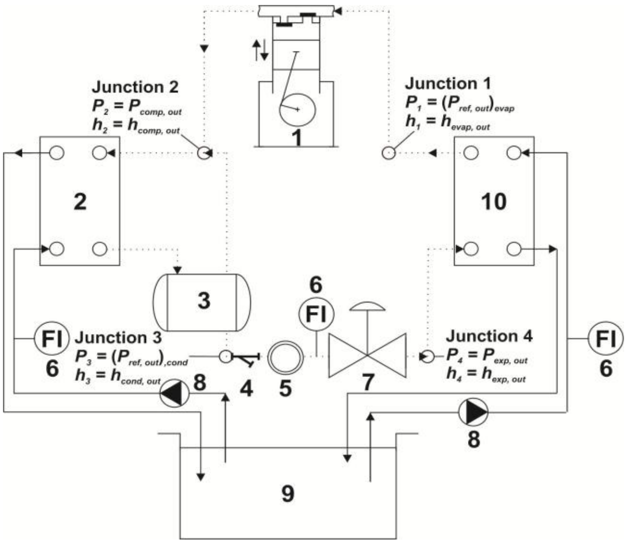
IV. Experimental Investigation of a VC System

The experimental setup is formed by the 4 main components of a conventional VC system. The compressor consists of a reciprocating piston electrically driven by a 3-phase inverter motor.

The condenser and evaporator units consist of plate heat exchangers using re-circulated water at room temperature as secondary fluid. The expansion device consists of a thermostatic expansion valve.

The refrigerant temperature is measured at the outlet of the thermostatic valve by an expansion valve thermocouple and at the outlet of the evaporator coil by a temperature sensing-bulb. Two electronics pressure transducers are installed at the inlet and outlet of the compressor to measure the refrigerant pressure on the low- and high-pressure sides of the compressor.

The data acquisition system is the RA1 software - data logger run by a Windows personal computer (PC). The software enables real time control and monitoring of all the measurement device outputs with a mimic diagram displayed on the PC screen.



List of components	
1	Compressor
2	Condenser
3	Receiver
4	Filter
5	Sight glass
6	Flow meter
7	Expansion valve
8	Water pump
9	Water reservoir
10	Evaporator

Figure 2. Experimental setup

Figure 3. Experimental setup image [25]

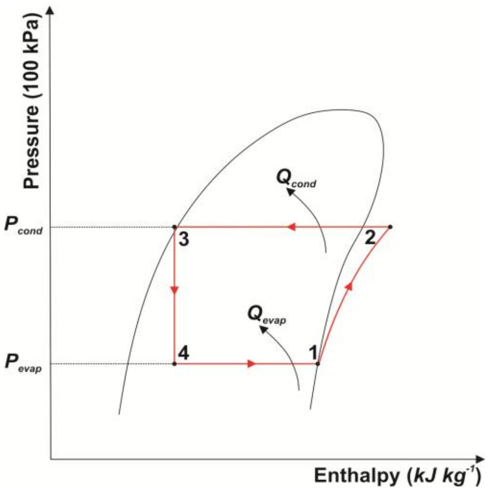


Figure 4. p-h diagram of the experimental setup

The parameters used for steady state modelling are listed as follows:

TABLE I. STEADY STATE PARAMETERS

Parameters	
$V_{ref} = 0.05\ m\ s^{-1}$	$\eta_{isen} = 0.65$
$f_{cond} = 1$	$h_{isen,out} = 258.36\ kJ\ kg^{-1}$
$f_{evap} = 1$	$D_v = 92.5.10^{-6}\ m^3\ s^{-1}$
$l_{cond} = 0.5\ m$	$RPM_{comp} = 1000$
$l_{evap} = 0.5\ m$	$\rho_{txv,in} = 1183.5\ kg\ m^{-3}$
$d_{cond} = 0.012\ m$	$C_{txv} = 10^{-6}$
$d_{evap} = 0.012\ m$	$T_{Sh,operating} = 4$ ℃
$\eta_{vol} = 4.6$	$T_{Sh,static} = 3.5$ ℃

V. RESULTS AND DISCUSSION

In Fig. 2, the refrigerant vapor R134a enters the compressor through junction 1 to undergo compression with its temperature rising above saturation (superheating) and exits through junction 2.

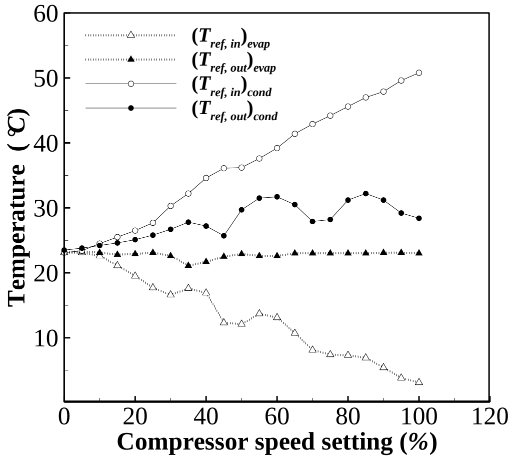


Figure 5. Temperature curves of the refrigerant

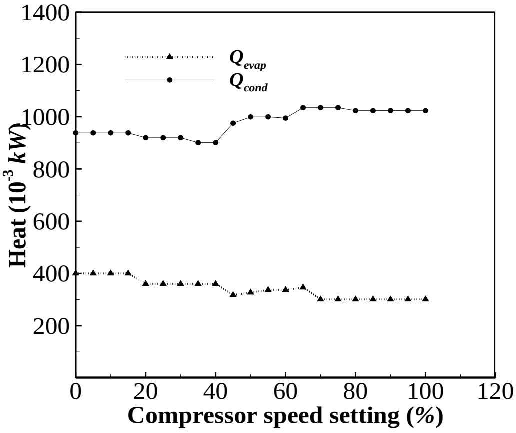


Figure 7. Heat curves of the refrigerant

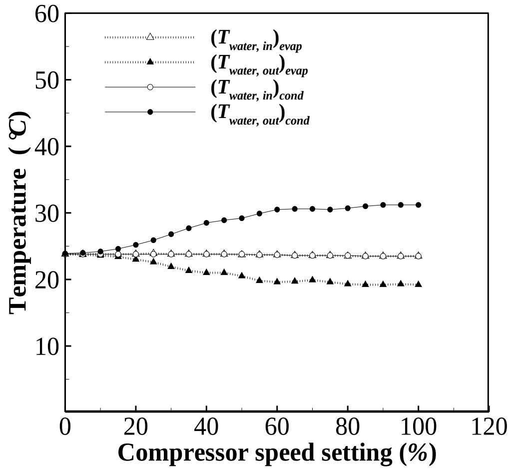


Figure 6. Temperature curves of water

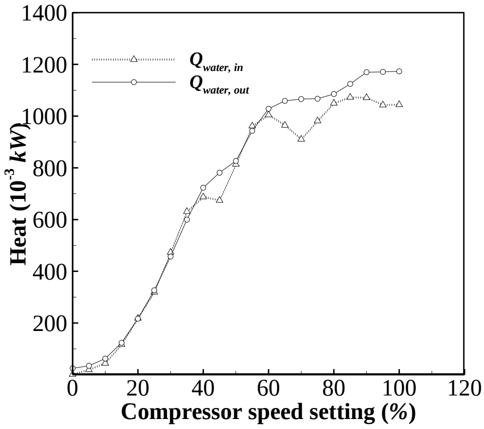


Figure 8. Heat curves of water

High temperature refrigerant enters the condenser through junction 2 to be cooled for superheat removal so that phase change from vapor to liquid could occur through latent heat removal at constant temperature and pressure. The refrigerant liquid exits the condenser through junction 3 and the heat removed is carried by the surrounding water stream in the condenser. The liquid stream of the refrigerant enters the expansion valve through junction 3 to undergo expansion with abrupt pressure drop followed by an exit through junction 4 in low temperature and pressure liquid - vapor mixture. Low temperature refrigerant mixture enters the evaporator through junction 4 to absorb energy from the surrounding water stream so that phase change to vapor could occur before returning to the compressor through junction 1 for a new cycle. The actual image of the experimental setup is given in Fig. 3 and Fig. 4 describes the thermodynamic cycle undergone by the refrigerant in a VC system

Fig. 5 presents the variation of the refrigerant temperature with the compressor speed setting within the VC system whilst Fig. 6 presents a similar distribution for the water streams surrounding the condenser and evaporator. The heat transfer distribution to and from the refrigerant is presented in Fig. 7 whilst its counterpart for the water streams is presented in Fig. 8. The model validation with experimental data for prediction of the condensing and evaporating pressures is presented in Fig. 9**.** The model validation with experiments to predict the junction enthalpies is presented in Fig. 10. The experimental setup reached steady state conditions after running for approximately 10 minutes with the water flow rate set at 50% of its full range through the condenser and at 70% of its full range through the evaporator. The compressor speed was varied with 5% step increment. The predicted junction pressures and enthalpies were determined from Eq. (12).

The data were recorded every minute and over 30 minutes for each range of compressor speed setting. The average value of each data range was plotted against their respective compressor speed setting. The collected data remained relatively unchanged for small variation of compressor speed setting. However, as the compressor speed setting was increased significantly, an important variation of the recorded data was observed.

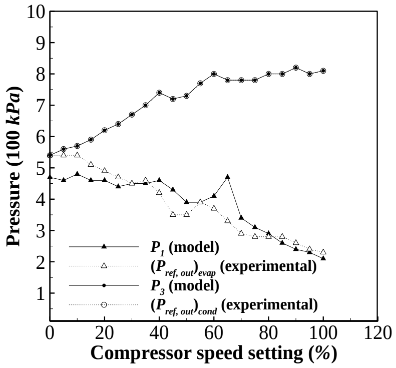


Figure 9. Pressure curves of the refrigerant

Re-circulated water entered the evaporator and condenser at an average room temperature of 23℃. The refrigerant temperature at the evaporator outlet decreased with increasing compressor speed and stabilized at about 60% compressor speed setting (Fig. 5). On the other hand, the refrigerant temperature through the condenser outlet increased as the speed of the compressor was raised and oscillated within 28℃ to 32℃ at 60% compressor speed setting (Fig. 5). As the compressor speed increased so did the refrigerant temperature through the condenser inlet (Fig. 5). Similarly, the inlet refrigerant pressure through the condenser also increased with increasing compressor speed (Fig. 9). The refrigerant enthalpy through the condenser inlet remained high (Fig. 10) because at this point the gas stream at high pressure and temperature carried an important amount of heat. Increasing compressor speed led the refrigerant temperature through the evaporator inlet to decrease (Fig. 5).

The refrigerant pressure at the evaporator inlet also decreased with increasing compressor speed (Fig. 9). The refrigerant mixture carried less heat at low pressure and temperature through the evaporator inlet consequently its enthalpy remained low at this point (Fig. 10). At steady state operating conditions of the experimental setup, the heat rejected by the refrigerant through the condenser remained higher than the heat absorbed by the refrigerant through the evaporator (Fig. 7). The heat added to the evaporator and heat removed from the condenser by the secondary water stream had the same magnitude (Fig. 8) and became stable at high compressor speed (about 65% compressor speed setting). The curve trends of the refrigerant conditions through the condenser and evaporator confirmed that the outputs of the VC experimental setup are satisfactory and adequate for accurate predictions. Some parameter values for steady state modelling (Table 1) were used to determine the output parameters of the components of the VC model. The refrigerant pressure at the compressor outlet and its enthalpy through the expansion valve outlet were inputs to the system of conservation equations at each junction (Eq. (12)) so that the modelling outputs could easily be determined with a Gaussian elimination method. The prediction of the model outputs at each junction was performed by coding the steady state modelling matrix in Matlab using the steady state parameter conditions in Table 1 and by solving for the model outputs in Eq. (9) with a Gaussian elimination method. The results obtained from the Matlab code were plotted against the experimental data for model validation.

The predicted evaporating or low refrigerant pressure was in agreement with the experimental data (Fig. 9). However, the evaporating refrigerant pressure was determined by reversing the modelling outputs for each compressor speed setting in order to describe the expansion process occurring in the expansion device otherwise, the distribution of the predicted evaporating pressure would grow linearly with the predicted condensing pressure in a purely analytical point of view. The expansion device – evaporator junction and the evaporator – compressor junction had the same pressure which was the lowest refrigerant pressure in the system. One could refer to these junctions as the low-pressure side of the VC system (Fig. 1). On the other hand, the compressor – condenser junction and the condenser - expansion device junction had the same pressure called condensing pressure. These junctions could be referred to as the high-pressure side of the VC system (Fig. 1).

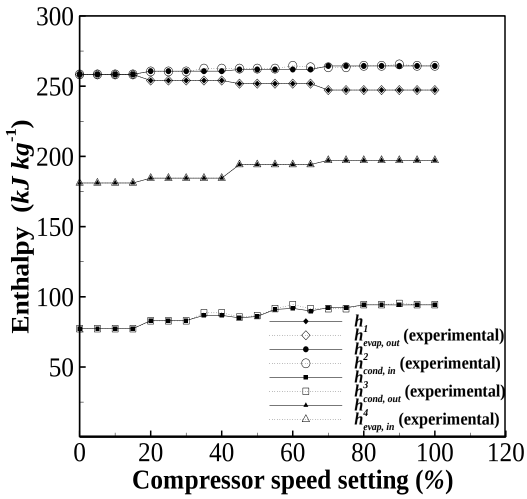


Figure 10. Enthalpy curves of the refrigerant

The junction enthalpy of the refrigerant at the evaporator outlet was in agreement with the experiments (Fig. 10) with an average value of $252\ kJ\ kg^{-1}$ and slightly lower than the refrigerant enthalpy of $262\ kJ\ kg^{-1}$ at the condenser inlet. The predicted outlet refrigerant enthalpy through the condenser was in agreement with experiments at an average of $87\ kJ\ kg^{-1}$ and confirmed that downstream the condenser heat is removed from the refrigerant by the circulating water stream to achieve condensing effects. The refrigerant enthalpy through the evaporator inlet was at an average of $190\ kJ\ kg^{-1}$ and in agreement with experiments. However, the assumption of isenthalpic process along the expansion device frequently adopted in the literatures could not be confirmed since the inlet and outlet refrigerant enthalpies through the expansion valve are different. The heat absorbed through the evaporator and removed through the condenser were assumed constant for modelling.

VI. Conclusion

A modelling technique for VC systems was developed using existing component modelling equations. A steady state modelling matrix was derived to determine the modelling outputs. The steady state matrix used a communication framework to exchange information with the VC system so that the only required parameters for modelling would be the junction pressure and enthalpy. The system of equations for steady state modelling was obtained by applying the conservation equations of energy and mass at each junction. Some input parameters were set in order to solve the system of equations. Experiments were performed, the recorded data were verified and confirmed relevant for model validation. The predicted evaporating pressure were in good agreement with the experimental data. The predicted junction enthalpies were also converging with the experimental results.

It was also found that the condensing pressure increased with increasing compressor speed setting. The evaporating pressure was inversely proportional to the condensing pressure due to the expansion process occurring at the expansion valve. The results also showed that at steady state conditions with constant water and refrigerant flow rates, more heat was removed from the refrigerant in the condenser than it was added to the evaporator.

The modelling results could be used to determine the optimal parameters of a VC system that are required to design optimal control strategies to improve the efficiency of VC systems. The modelling technique could also be adopted for different type of component modelling equations and its results supported with experimental results. The results obtained could be useful for improving the design of innovative control methods of VC systems.

Although not attempted in this work, the developed modelling method could be extended for modelling of complex configurations of VC systems such as systems with single condenser and multiple evaporators.

Conflict of Interest

The authors declare no conflict of interest.

Author Contributions

Conrad Sanama wrote the paper and Xiaohua Xia checked the paper.

All authors approved the final version.

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References

[1] P. Welsby, S. Devotta, and P. J. Diggory, "Steady and dynamic state simulations of heat-pumps. Part I: literature review," *Applied Energy*, vol. 31, pp. 189-203, 1988.

[2] C. C. Hiller and L. R. Glicksman, "Improving heat-pump performance via compressor capacity control analysis and test," *Massachusetts: MIT.*, 1976.

[3] G. L. Davis and T. C. Scott, "Component modelling requirements for refrigeration system simulation," *Purdue Compressor Technology Conference, West Lafayette*, pp. 401-408, 1976.

[4] R. D. Ellison, F. A. Creswick, "A computer simulation of steady-state performance of air-to-air heat pumps," *Tennessee: Oak Ridge National Laboratory*, 1978.

[5] R. D. Ellison, C. K. Rice, "Heat pump model update," *Tennessee: Oak Ridge National Laboratory*, 1979.

[6] J. E. Flower, "Analytical modelling of heat pump units as a design aid for performance prediction," *California: Lawrence Livermore Laboratory*, 1978.

[7] F. W. Ahrens, "Heat pump modelling, simulation and design," *Espinho: Nato, Advanced Study Institute on Heat Pump Fundamentals*, 1980.

[8] W. M. Kays, *Compact Heat Exchangers*, New York: McGraw-Hill, 1984.

[9] A. E. Dabiri, "A steady-state computer simulation model for air-to-air heat pumps," *ASHRAE Transactions*, vol. 88, no. 2, pp. 973-987, 1982.

[10] A. E. Dabiri, C. K. Rice, "A compressor simulation model with corrections for the levels of suction gas superheat," *ASHRAE Transactions*, vol. 87, pp. 771-782, 1981.

[11] M. L. Martins Costa, J. A. R. Parise, "A three-zone simulation model for air-cooled condensers," *Heat Recovery Systems and CHP*, vol. 13, no. 2, pp. 97-113, 1993.

[12] M. J. S. Lemos and E. L. Zaparoli, "Steady-state numerical solution of vapor compression refrigeration units," *International Refrigeration and Air Conditioning Conference*, West Lafayette, pp. 235-240, 1996.

[13] J. Liu, W. Wei, G. Ding, C. Zhang, M. Fukaya, K. Wang, and T. Inagaki, "A general steady state mathematical model for fin-and-tube heat exchanger based on graph theory," *International Journal of Refrigeration*, vol. 27, pp. 965-973, 2004.

[14] G. L. Ding, "Recent developments in simulation techniques for vapour-compression refrigeration systems," *International Journal of Refrigeration*, vol. 30, pp. 1119-1133, 2007.

[15] J. M. Winkler, "Development of a component based simulation tool for the steady state and transient analysis of vapor compression systems," *Maryland: Faculty of the Graduate School of the University of Maryland, College Park*, 2009.

[16] J. M. Winkler, V. Aute, and R. Radermacher, "Comprehensive investigation of numerical methods in simulating a steady-state vapor compression system," *International Journal of Refrigeration*, vol. 31, pp. 930-942, 2008.

[17] J. M. Belman, J. Navarro-Esbri, D. Ginestar, and V. Millian, "Steady-state model of a variable speed vapor compression system using R134a as working fluid," *International Journal of Energy Research*, vol. 34, pp. 933-945, 2010.

[18] F. Bagheri, "Development of efficient air conditioning and refrigeration system for service vehicles," *Vancouver: School of Mechatronic Systems Engineering, Faculty of Applied Science, Simon Fraser University*, 2016.

[19] M. Beshr, V. Aute, and R. Radermacher, "Steady state modeling of advanced vapor compression systems with multiple air and refrigerant loops," in *Proc. 12th IEA Heat Pump Conference, Rotterdam*, 2010, pp. 1-10.

[20] G. Panaras, E. Mathioulakis, and V. Belessiotis, "A semi-analytical refrigeration cycle modelling approach for a heat pump hot water heater," *International Journal of Sustainable Energy*, vol. 37, no. 4, pp. 393-414, 2018.

[21] J. Woods and E. Bonnema, "Regression-based approach to modeling emerging HVAC technologies in energy plus: A case study using a Vuilleumier-Cycle heat pump," *Energy & Buildings*, vol. 186, pp. 195-207, 2019.

[22] A. A. Bukshaisha and B. M. Fronk, "Simulation of membrane heat pump system performance for space cooling," *International Journal of Refrigeration*, vol. 99, pp. 371-381, 2019.

[23] S. A. Klein, "Engineering equation solver," *Madison: F-Chart Software*, 2017.

[24] C. Sanama, M. Sharifpur, and J. P. Meyer, "Mathematical modeling of orifice downstream flow under flow-accelerated corrosion," *Nuclear Engineering and Design*, vol. 326, pp. 285-289, 2018.

[25] Armfield Ltd., Instruction manual: RA1 vapour – compression refrigeration unit, *Hampshire: Bridge House*, 2009.

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