Steady State Simulation of a CO₂ Heat pump system for Water Heating

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Abstract. A steady state simulation model for a CO₂ heat pump system was developed in order to evaluate system performance of water heating. A simple Macro Excel coding was develop to calculate the refrigerant condition in the heat pump pipe. The REFPROP 8.0 also use to find initial value of the pressure, enthalpy and refrigerant characteristic. The Macro Excel and REFPROP 8.0 was developed by linked both programme because it provide a similar features. The simulation was compared with experimental data of a laboratory scale CO₂ heat pump system. The pressure and temperature estimated by simulation were found to be approximately within 10% error between simulation and experiment. The COP obtained from the simulation was higher than that obtained by experiment and it was reduce as the effect of the mass flow rate of the refrigerant reduction for the performance analysis by the change of the closing angle of expansion valve.

Keywords: COP, Friction force, Heat pump, Heat-transfer coefficient

1. Introduction

A novel simulation model for CO₂ heat pump system was developed to evaluate performance for water heating. The model consisted of : (1) A compressor model using polytropic assumption, (2) Turbulent heat transfer model of Petukov-Gnielinski available for both gas and liquid phases depending on wide range of the Reynolds and the Prandtl numbers, (3) An expansion valve model using orifice flow assumption and (4) Thermophysical properties were quoted from REFPROP 8.0 tabulated beforehand.

Choi and Kim [1] analysis the expansion device Influencing to the performance of a heat pump using R407C. Because of the increasing focus on comfort and energy conservation, The thermostatic expansion valves (TXVs) was change to electronic expansion valves (EEVs). By varying refrigerant charge amount from -20% to +20% of full charge in a steady state, the effects of off-design refrigerant charge on the performance of a water-to-water heat pump was investigated. They found that the capillary tube system was relatively sensitive to refrigerant charge and outdoor load conditions. The capacity and COP of the EEV system had little dependence on the refrigerant charge, while those were strongly dependent on outdoor load conditions.

Rais, et.al [2] also investigated heat-pump cycle for air conditioning for both numerically and experimentally by evaluating the coefficient of performance (COP) under Japanese Industrial Standard.
(JIS B 8619:1999) and ANSI/ AHRI standard 750-2007 operating conditions. Two expansion valve coefficients $C_v(\varphi) = 0.12$ for standard operating conditions and poor operating conditions. The simulation and experiment comparison resulted the decreasing of the COP for standard operating condition is equal to 14 %, from 3.47 to 2.95 and a decrease of the cooling capacity is equal to 18 %, from 309.72 to 253.53 W.

An analytical model was developed for each component and combined by discretisation. The heat-pump system consisted of the following components: the compressor, condenser, evaporator, expansion valve, and corresponding tube connections for refrigerant flow, as shown in Fig. 1. The model was further simplified by applying basic equations for each component to shorten the simulation time

![Figure 1. The Schematic of Heat Pump system](image_url)

2. Simulation

2.1. Basic Equation and discretisation

Several assumptions were made when the refrigerant flow in the tube as The quasi-1-D steady, then the mass, momentum, and energy conservation equations and the equation of state tabulated from REFPROP 8.0[7] are given as follows:

\[
\frac{dG}{dz} = \frac{d}{dz}(A) = 0
\]

\[
u \frac{du}{dz} = \frac{dp}{dz} + \frac{dF}{dz}
\]

\[
\frac{d}{dz} \left( \frac{uA \left( h + \frac{u^2}{2} \right)}{2} \right) = \frac{dQ}{dz}
\]

\[
= (p, h)
\]

![Figure 2. The flow of refrigerant in a tube.](image_url)
Figure 2. shows the flow of refrigerant in a tube and it was assume as quasi one dimensional flow of refrigerant. By this assumption the equation (1)–(3) are discretised. The result of the discretisation was shows in equation (5)–(6), as it determinate as continuity equation and momentum equation.

\[ j \frac{u}{j} A_{j} + 1 \frac{u}{j+1} A_{j+1} = 0 \]  \hspace{1cm} (5)

\[ j \frac{u}{j} = j + 1 \frac{u}{j+1} \]  \hspace{1cm} (6)

\[ 0 = j \frac{u^2}{j} A_{j} + 1 \frac{u^2}{j+1} A_{j+1} \]  \hspace{1cm} (7)

Frequently (especially for flow processes), it is useful to express the first law as a statement about the rate of heat transfer and work for a control volume, the discretisation become equation (8) and (9) and it determinate as energy equation.

\[ p_2 = p_1 + \frac{u^2}{1} \left( \frac{1}{2} \right)^2 \frac{Q}{G} \]  \hspace{1cm} (8)

\[ h_2 = h_1 + \frac{u^2}{1} \left( \frac{1}{2} \right)^2 \frac{Q}{G} \]  \hspace{1cm} (9)

2.2. Calculations Flow

The flow of calculations was from the evaporator outlet, compressor, condenser and expansion valve. Since the variable conditions of upside flow are known, then \( p_2, F_{Fr}, Q \) can be computed, while \( u_2, p_2, h_2 \) can be determined through calculation algorithm, as follow.

1. Assume the conditions of both up and downstream are, then \( \rho_1, \rho_2 = 1, F_{Frac} = F_1, Q = Q_1 \) would be obtained.
2. Input parameters (compressor inhalation volume, flow coefficient, piping length, pipe internal and external diameter, number of branch pipe, outside tube heat transfer coefficient, polytropic compression coefficient, sealed refrigerant amount)
3. Assume pressure (\( p \)) and enthalpy (\( h \)) at compressor inlet than search density (\( \rho_2 \)) using the condition equation (7)
4. Calculate downstream side flow of heat pump (from the expansion valve inlet flow tube 1~ evaporator ~ tube 2 ~ to compressor inlet) using Shah equation \( f = \frac{16}{Re} \) (\( Re \leq 2000 \))

\[ \alpha_i = 0.023 \frac{L}{D_i} Re_i^{0.8} Pr_i^{0.4} \left[ 1 + 3.8 \left( \frac{x}{1-x} \right)^{0.76} \left( \frac{p}{p_{cr}} \right)^{-0.38} \right] \] Blasius equation

\[ f = 0.079 \left( \frac{1}{Re} \right) \] and Gnielinski correlation equation

\[ f = 0.079 \left( \frac{1}{Re} \right) \] (\( Re < 2000 \))


\[ Nu_0 = 0.3 + \left( Nu_{lam}^2 + Nu_{turb}^2 \right)^{1/2} \]

5. Repeat the calculation to determining the density until obtain the smallest value than compare \( \rho_2 \) which obtained from \( \rho'_2 \). If error can be ignored, then the calculation \[
\left[ \frac{\rho_2}{\rho'_2} \right] \leq 1.0 \times 10^{-5} \] deemed \( \rho_2 = \rho'_2 \)

6. Determining the enthalpy at compressor outlet to get the density value by using Newton's method.

7. Calculate upstream side flow of heat pump (from compressor outlet ~ tube 3 ~ condenser ~ tube 4 ~ expansion valve)

8. The result of pressure and enthalpy at compressor outlet was used to calculate different pressure (\( \Delta p \)) at the expansion valve.

9. Repeat the calculation until obtain the correct value of pressure and the enthalpy at the expansion valve outlet was constant.

3. Experiment Setup

Heat pump unit consists of compressor, condenser, evaporator, and expansion valve which are connected to the pipe that is covered with insulator as shown in figure 3. The heat system was designed to allow the refrigerant flow under steady-state condition. The Cori-flow meter was used to measure refrigerant mass flow rate. The reciprocating compressor, which suck-press-release the gas according to the back and forth movement of the piston, was used. The expansion valve needle was used for decompression. The copper pipe tube was used with outside diameter of 1/4 inch Robin air brass manifold kit and insulated by a glass wool. All pipe was connecting as loop connection form compressor, condenser, expansion valve and evaporator at it was determine as tube 1, tube 2, tube 3 and tube 4. For mass flow rate meter connector, an extra tube were added as a branching pipe units

Figure 3. Experimental apparatus

4. Results and discussions

4.1. Performance evaluation of the hot water system

The performance of the system being studied the specific performance of heating Capacity that evaluated on the basis system COPs, which have been estimated for expansion valve coefficient and spiral heat exchanger calculation. The error tolerant was decided less than 10 % to compare the calculation of simulation and experiment observation. Table 1. shows The specific enthalpy and refrigerant pressure at each point are listed in table 1. The comparison of simulation and experimental
for work compression, heating capacity and COP are shown in table 2. And the $p$-$h$ diagram of the comparison are shown in Figure 4. We found the error of this calculation which shown in table 3, is within approximately 10%. Therefore the simulation result are working well with experimental.

Table 1. $p$-$h$ comparison of simulation & experiment

| No. | Simulation | Experiment |
|-----|------------|------------|
|     | $h$ [kJ kg\(^{-1}\)] | $p$ [MPa] | $h$ [kJ kg\(^{-1}\)] | $p$ [MPa] |
| 1   | 480.04     | 2.57       | 475.06               | 2.57       |
| 2   | 538.31     | 8.77       | 529.246              | 8.83       |
| 3   | 290.97     | 8.77       | 272.988              | 8.83       |
| 4   | 290.53     | 2.57       | 272.988              | 2.58       |

Table 2. Error comparison

| No. | % error | $h$ | $p$ |
|-----|---------|-----|-----|
| 1   | 1.57    | 7.28|
| 2   | 1.56    | 0.77|
| 3   | 6.85    | 0.79|
| 4   | 6.77    | 7.23|

Table 3. Comparison of simulation & experiment

| Performance specification | Experiment | Simulation |
|---------------------------|------------|-----------|
| Work of compression (W)   | 563.72     | 546.3     |
| Heating capacity (W)      | 2200.53    | 2251.5    |
| COP [-]                   | 3.90       | 4.12      |

Figure 4. $p$-$h$ diagram comparison of the simulation and experiment

The experimental and simulation results in standard operating condition showed loss of
pressure from the expansion valve outlet to the compressor inlet. The loss of pressure occur because of horizontal position of the pipe according to the simulation model in the evaporator. Hence the pipe used in this experiment was positioned vertically to compensate the difference of refrigerant pressure going to the compressor by gravity. According to performance evaluation of heat pump systems, the heating capacity and COP indicated no significant difference. Around 2200.53 and 2251.5 for heating capacity and 3.90 and 4.12 for the COP. For working compressor the result shows the experiment were lower than simulation, which were 563.72 and 546.3.

The simulation have characteristics were the expansion valve and heat exchange can be used to determine the condition of the condenser by considering the expansion valve and condenser which affect the performance, reveal the existence of refrigerant pressure and change of temperature, and change of density and speed of sound. In two phases, more pressure loss occurred due to the friction in the tube. Moreover, the temperature decreased with the phase change.

4.2. Performance evaluation with the change of closing angle expansion valve

In order to analyse the effect of the closing angle expansion valve to the heat pump performance, it is very important to optimization the energy usage by analysing the energy requirements and heating capacities. In this section we analyse the different the performance of heat pump in various environment according to various expansion valve coefficient of the closing angle. The experimental and simulated resulted the performances coefficient of the heat-pump system and heating capacity are compared in Figs. 5 and 6.

Table 4. $p$-$h$ comparison of simulation and experiment at 3050 degree operating conditions

| No. | Simulation | Experiment |
|-----|------------|------------|
| 1   | h $[\text{kJ kg}^{-1}]$ | p [MPa] | h $[\text{kJ kg}^{-1}]$ | p [MPa] |
| 2   | 480.04     | 2.57       | 475.06     | 2.57      |
| 3   | 538.31     | 8.77       | 529.246    | 8.83      |
| 4   | 290.97     | 8.77       | 272.988    | 8.83      |
| 5   | 290.53     | 2.57       | 272.988    | 2.58      |
Table 5. Error comparison

| No. | % error | \( h \) | \( p \) |
|-----|---------|--------|--------|
| 1   | 1.57    | 7.28   |
| 2   | 1.56    | 0.77   |
| 3   | 6.85    | 0.79   |
| 4   | 6.77    | 7.23   |

Table 6. \( p-h \) comparison of simulation and experiment at 3060 degree

| No. | Simulation | Experiment |
|-----|------------|------------|
|     | \( h \) [kJ kg\(^{-1}\)] | \( p \) [MPa] | \( h \) [kJ kg\(^{-1}\)] | \( p \) [MPa] |
| 1   | 482.6      | 2.36       | 475.06 | 2.57 |
| 2   | 545.95     | 8.67       | 529.25 | 8.83 |
| 3   | 292.92     | 8.67       | 272.99 | 8.83 |
| 4   | 292.89     | 2.37       | 272.99 | 2.59 |

Table 7. Error comparison

| No. | % error | \( h \) | \( p \) |
|-----|---------|--------|--------|
| 1   | 7.90    | 1.58   |
| 2   | 1.75    | 3.15   |
| 3   | 1.77    | 7.3    |
| 4   | 8.14    | 7.29   |

Table 8. \( p-h \) comparison of simulation and experiment at 3068 degree operating conditions

| No. | Simulation | Experiment |
|-----|------------|------------|
|     | \( h \) [kJ kg\(^{-1}\)] | \( p \) [MPa] | \( h \) [kJ kg\(^{-1}\)] | \( p \) [MPa] |
| 1   | 489.12     | 1.88       | 481.76 | 1.99 |
| 2   | 574.48     | 9.04       | 556.17 | 9.28 |
| 3   | 271.62     | 9.04       | 265.69 | 9.27 |
| 4   | 271.60     | 1.89       | 265.69 | 2    |

Table 9. Error comparison

| No. | % error | \( h \) | \( p \) |
|-----|---------|--------|--------|
| 1   | 5.03    | 1.52   |
| 2   | 2.48    | 3.29   |
| 3   | 2.46    | 2.23   |
| 4   | 5.00    | 2.22   |
The error of p-h comparison of simulation and experiment was calculating, within approximately 10 % of the calculation shows in table 5,7 and 9. This result indicated that simulation are working well with experimental.

In the expansion valve, because the rate of refrigerant flow increases when the pressure difference was high. The flow rate were enhanced as the high-pressure value increases and the low-pressure value decreases. If the high-pressure value decreases and the low-pressure value increases, the flow rate was reduced. As the effect of the expansion valve closing angle to the COP and heating capacity, resulted that the COP and the heating capacity was decrease. This is cause by the flow rate of the refrigerant is reduce as closing the expansion valve.

5. Conclusions

In this study, the numerical simulation for the heat-pump system of water heating using CO2 was discussed. Within approximately 10% of the error calculation was found therefore simulated results are in reasonable agreement as compare to experimental. The result also show that the COP and heating capacity was decrease according to the effect of the closing angle of the expansion valve.

An optimizing study for the best calculation of total heat exchanger for spiral tube heat exchanger area has been carried out. The Zhukaukas correlation equation was changed to Gnielinski correlation equation and the result effect of heat transfer calculation on system performance has been presented a good result.

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