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A parameter-estimation model for variable refrigerant flow heat pump systems

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Abstract

A variable refrigerant flow (VRF) model based on a simplified vapor compression cycle is presented. The outdoor unit and the indoor units are modelled separately. An optimization procedure is used to identify their parameters by minimizing the differences between the model predicted capacities and power input and manufacturer data. Calibration results show a root mean square error (RMSE) in range of 4%, 8.8% and 7.3% for the indoor unit total capacity, the outdoor unit capacity and power consumption respectively. Three configurations of VRF heat pump are evaluated using previously calibrated indoor unit and outdoor unit models. The results show that the model-predicted cooling capacity and power input are close to the manufacturer data with an average RMSE of 2.2% for both the power input and the cooling capacity.

Keywords: Variable refrigerant flow; Modelling and simulation, Model calibration

1. Introduction

Variable refrigerant flow (VRF) heat pump systems are growing in popularity for heating and cooling applications in buildings. When equipped with a heat recovery unit, VRF systems can provide simultaneous heating and cooling, with high efficiency, to different zones of the same building. To support their development, facilitate their design and evaluate their performance, simulation tools that model their behavior are necessary.

Lin et al. [1] gave a substantial review of recent development in modelling VRF heat pumps. VRF models range from detailed physics-based models to equation-fit models with varying levels of accuracy. Yan et al [2] developed a physical-based model for dual-evaporator air conditioning systems, including the effects of the pressure drop and the refrigerant piping length in the system performance. The model was developed by adding a second evaporator to an existing single-evaporator air conditioning system model. Hong et al. [3] developed a physical-based VRF model in EnergyPlus. The model validation using field test results of an instrumented house showed that the error associated with the VRF power demand is in range of 10%. Zhu et al. [4] implemented a modular transient VRF model in TRNSYS. Results showed an average deviation between simulation results and experimental data of 7.9%, 12.5% and 6.2% for the capacity, the power input and the coefficient of performance (COP), respectively. Cheung and Braun [5] proposed a component based gray box model for ductless multi-split systems with a variable frequency compressor. Parameters are first tuned using experimental data and then applied to the model. Comparisons between experimental data and simulation results after parameter tuning showed that the error in the power consumption is in range of 8%. More recently, Sun et al. [6] developed a general VRF model using graph theory. The model can simulate systems with arbitrary layouts and under various operating conditions including partial load operation. The model is validated using experimental data and the results showed deviations of 3% and 3°C for the capacity and the indoor unit (IU) temperature, respectively. While physical-based models have been shown to predict VRF behavior with acceptable accuracy, they are not widely used in energy simulation software such as EnergyPlus. They require inputs that are not readily provided in manufacturer literature and are typically computationally expensive. Some authors have proposed empirical equation-fit models using performance curves based on manufacturer data. Raustad [7] developed an equation-fit model implemented in EnergyPlus. A comparison of the model predicted energy use against field test results showed that the error is in range of 25% overall [8].

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Kim et al. [9] showed that a proper calibration of Raustad's model can improve its accuracy. Experiments in an office building showed that the error decreased from 32.3% to 15.7% for the non-calibrated and calibrated models, respectively. Torregrosa-Jaime et al. [10] have developed an improved equation fit-model based on Raustad's model that adjusts better to the manufacturer data. They proposed to evaluate the VRF power consumption directly as a function of the operating temperatures and the partial load ratio (PLR) while the original model uses intermediate parameters that are inferred from the manufacturer data (i.e. energy input ratio as function of the temperature, energy input ratio as function of the PLR). For partial load operation, the model still loses accuracy and shows considerable variations with the manufacturer's data.

This paper presents the development of a model for the simulation of VRF heat pump systems based on a simplified thermodynamic cycle. The indoor units (IU) and the outdoor units (OU) that form the VRF heat pump are modelled independently so that various VRF configurations with multiple IUs can be evaluated. Model parameters are inferred from a parameter-estimation procedure on available manufacturer data to minimize the error between the model-predicted performance metrics (e.g. capacity, power input) and the performance data from the manufacturer. Calibrated OU an IU models are then assembled into various VRF system configurations.

2. Methodology

The VRF model is based on a simplified vapor compression cycle as proposed by Jin [11]. The benefit in such cycle is to reduce the number of the model parameters in order to facilitate the calibration process and to reduce the computational time since only few refrigerant states are required. Here, the Martin-Hou equation of state for R410A is used to evaluate the thermodynamic parameters of the refrigerant [12]. Figure 1 presents a pressure-enthalpy diagram of the simplified vapor compression cycle that relies on the following assumptions:

- There is no subcooling in the condenser; the refrigerant leaves the condenser (point B) in a saturated liquid state.
- The refrigerant leaving the evaporator (point C) is in a superheated vapor state with a constant degree of superheating.
- The sensible heat transfer (from point A to point C) in the evaporator is neglected, as it is small compared to the latent heat transfer.
- The expansion process through the electronic expansion valve (EEV) is isenthalpic.
- The pressure drops within the heat exchangers as well as those in the refrigerant pipes are neglected.

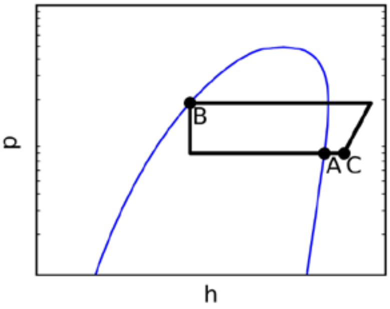

Fig. 1. P-h diagram of the simplified vapor compression cycle

3. Variable refrigerant flow heat pump model

3.1. Indoor unit model

The IU is likened to an air-refrigerant heat exchanger operating at a constant refrigerant temperature. The model parameters are the inside and the outside surface heat transfer coefficients (UA_{in} and h_cA_{out}). As shown in Figure 2, the model requires as inputs the air mass flow rate ($\dot{m}_{IU}$), dry bulb temperature ($T_{DB,IU}$), wet bulb temperature ($T_{WB,IU}$) and the total capacity ($\dot{q}_{IU}$). The outputs of the model are the sensible and latent capacities ($\dot{q}_{sen,IU}$ and $\dot{q}_{lat,IU}$) and the refrigerant evaporating (in cooling mode) or condensing (in heating mode) temperature (T_{ref}).

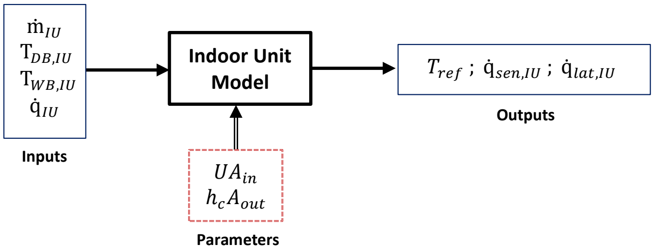

Fig. 2. Indoor unit model: parameters, inputs and outputs

The heat exchanger efficiency is calculated by the ε-NTU method:

$$UA = \frac{1}{\frac{1}{h_c A_{out}} + \frac{1}{UA_{in}}} \tag{1}$$

$$\mathrm{NTU} = UA/\dot{m}_{IU} c_{p,a} \tag{2}$$

$$\varepsilon = 1 - e^{(-NTU)} \tag{3}$$

where NTU is the number of transfer units, UA is the global heat transfer coefficient, ε is the heat exchanger efficiency, and $c_{p,a}$ is the specific isobaric heat capacity of the moist air.

When the indoor unit operates as an evaporator (i.e., in cooling mode), condensation may occur at the outside surface of the cooling coil. Depending on the operating conditions, the cooling coil may be fully dry, fully wet or partially wet, and each case will affect the heat transfer coefficient. Therefore, the model of the IU should take account of whether it operates as an evaporator or condenser. In heating mode, there is no condensation and the IU operates as a sensible heat exchanger. The heat transfer rate is given by:

$$\dot{q}_{IU} = \varepsilon \cdot \dot{m}_{IU} c_{p,a} \left(T_{DB,IU} - T_{ref}\right) \tag{4}$$

In cooling mode, the outside surface of the coil is considered to be fully wet as proposed by Jin [11]. The heat exchanger behavior follows the enthalpy potential method. The total heat transfer from the air stream to the coil surface is a combination of sensible heat transfer ($\dot{q}_{sen,IU}$) and latent heat transfer ($\dot{q}_{lat,IU}$):

$$\dot{q}_{IU} = \dot{q}_{sen,IU} + \dot{q}_{lat} \tag{5}$$

$$\dot{q}_{IU} = \varepsilon_{eff} \dot{m}_{IU} \left(h_a - h_{a,e}\right) \tag{6}$$

$$\varepsilon_{eff} = 1 - e^{(-NTU_{eff})} \tag{7}$$

$$\mathrm{NTU}_{eff} = UA_{eff}/\dot{m}_a c_{p,a} \tag{8}$$

where UA_{eff} is the effective heat transfer coefficient for the total (i.e. sensible and latent) heat transfer, h_a is the enthalpy of the moist air evaluated at the inlet air dry bulb and wet bulb temperature ($T_{DB,IU}$ and $T_{WB,IU}$), $h_{a,e}$ is the enthalpy of the moist air at the refrigerant evaporating temperature (T_{ref}), ε_{eff} and NTU_{eff} are the corresponding heat exchanger effectiveness and number of transfer units, respectively.

The effective heat transfer coefficient UA_{eff} is given by:

$$UA_{eff} = \frac{1}{\frac{1}{h_c A_{out}} + \frac{c_{p,a,s}}{c_{p,a}} \frac{1}{UA_{in}}} \tag{9}$$

where $c_{p,a,s}$ is the specific isobaric heat capacity of saturated air evaluated at the coil effective surface temperature.

The sensible heat transfer rate is evaluated analogically to a sensible heat exchanger from the outside surface heat transfer coefficient, considering a uniform coil surface temperature:

$$\dot{q}_{sen,IU} = \varepsilon' \dot{m}_{IU} c_{p,a} \left(T_{DB,IU} - T_{surf}\right) \tag{10}$$

$$\varepsilon' = 1 - e^{(-NTU_a)} \tag{11}$$

$$\mathrm{NTU}_a = h_c A_{out} / \dot{m}_{IU} c_{p,a} \tag{12}$$

where NTU_a is the number of air side transfer units, ε' is the air side effectiveness and T_{surf} is the effective surface temperature of the coil, which corresponds to the moist air dry bulb temperature at the coil.

The effective surface temperature is inferred from the corresponding enthalpy of saturated air $h_{a,s}$:

$$h_{a,s} = h_a - \dot{q}_{IU} / \dot{m}_{IU} \varepsilon' \tag{13}$$

3.2. Outdoor unit model

The OU model is based on the simplified heat pump cycle previously presented in Figure 1. It consists of two components: a compressor and a heat exchanger. The heat exchanger model is the same as presented in section 3.1 for the indoor units. Depending on the operation mode, it is considered as a sensible heat exchanger (i.e., condenser in cooling mode) or as a latent/sensible heat exchanger (i.e., evaporator in heating mode). Frost accumulation is not considered in the model. Figure 3 shows a schematic chart of the OU model with its parameters, inputs and outputs.

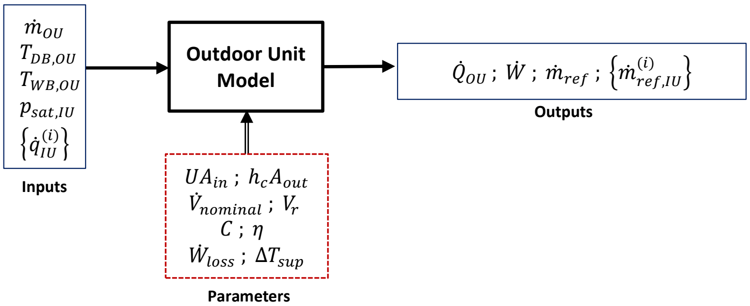

Fig. 3. Outdoor unit model: parameters, inputs and outputs

The scroll compressor is modelled according to Jin [11]. The compressor work is a result of isentropic compression at the built-in volume ratio followed by isochoric compression to the discharge pressure as described by Winandy et al. [13]:

$$\dot{W}_t = \frac{\gamma}{\gamma - 1} p_{eva} \dot{V}_{nominal} \left(\frac{\gamma}{\gamma - 1} \frac{p_{con}}{p_{eva} V_r} + \frac{1}{\gamma} p_r^{\frac{\gamma}{\gamma - 1}} - 1 \right) \tag{14}$$

where $\dot{W}_t$ is the theoretical compressor work, p_{eva} and p_{con} are the evaporating and condensing pressure, γ is the isentropic exponent of the refrigerant at the suction of the compressor, $\dot{V}_{nominal}$ is the nominal refrigerant volume flow rate, V_r is the built-in volume ratio between the discharge and the suction of the compressor and $p_r = V_r^{\gamma}$ is the built-in pressure ratio. In cooling mode, p_{con} is the refrigerant pressure in the OU heat exchanger and $p_{eva} = p_{sat,IU}$ is the pressure in the IU. In heating mode, $p_{con} = p_{sat,IU}$ is the refrigerant pressure in the IU and p_{eva} is the pressure in the OU heat exchanger.

The power input of the compressor is given by:

$$\dot{W} = \frac{\dot{W}_t}{\eta} + \dot{W}_{loss} \tag{15}$$

where η is the electromechanical efficiency of the compressor and $\dot{W}_{loss}$ are constant power losses.

Since the sensible heat transfer to the refrigerant is neglected in the evaporator, the evaporator heat transfer rate is evaluated from the enthalpy difference between points A and B of Figure 1:

$$\dot{Q}_{eva} = \left(\frac{\dot{V}_{nominal}}{v_{suc}} - \dot{m}_{leak}\right)(h_A - h_B) \tag{16}$$

where $\dot{Q}_{eva}$ is the total evaporator heat transfer rate, v_{suc} is the specific volume at the suction of the compressor, $\dot{m}_{leak}$ is the leakage mass flow rate in the compressor. Based on Chen et al. [14], the leakage mass flow rate can be estimated as a function of the condensing and evaporating pressures:

$$\dot{m}_{leak} = C\, p_{con}/p_{eva} \tag{17}$$

where C is the leakage coefficient. The total condenser heat transfer rate, $\dot{Q}_{con}$, can then be obtained by an energy balance:

$$\dot{Q}_{con} = -(\dot{Q}_{eva} + \dot{W}) \tag{18}$$

In cooling mode, $\dot{Q}_{con}$ and $\dot{Q}_{eva}$ are the OU heat rejection ($\dot{Q}_{OU}$) and the sum of the IU capacities $\left\{\dot{q}_{IU}^{(i)}\right\}$ respectively. In heating mode $\dot{Q}_{eva}$ is the OU heat absorption ($\dot{Q}_{OU}$) and $\dot{Q}_{con}$ is the sum of the IU capacities $\left\{\dot{q}_{IU}^{(i)}\right\}$.

3.3. Assembled VRF model

For the assembled VRF model, it is important to ensure that the refrigerant is always at the same pressure in each IU since many IUs with different operating conditions can be involved. The operating conditions are then evaluated by iteration. The refrigerant mass flow rate in the IUs is adapted to meet the required capacity. Figure 4 presents the calculation procedure for the VRF model in cooling mode that follows this sequence:

1. Initialize the IU and the OU operating conditions using the nominal values of the IU capacity, dry bulb temperature and wet bulb temperature and the OU capacity, power consumption dry bulb temperature and wet bulb temperature. These nominal values correspond to the rated conditions and performances in accordance to the ANSI/AHRI standard 1230. They can be found in the product datasheets.
2. Evaluate each IU refrigerant temperature ($T_{ref,IU}^{(i)}$) using Eq. 4.
3. Evaluate the minimum IU refrigerant evaporating temperature:

$$T_{sat,IU} = \min\left[T_{ref,IU}^{(i)}\right] \tag{19}$$

4. Evaluate the IU capacities $\dot{q}_{IU}^{(i)}$ using Eq. 4 and the minimum evaporating temperature.
5. Evaluate the refrigerant condensing temperature at the OU using Eq. 4.
6. Evaluate the evaporating pressure p_{eva}, the condensing pressure p_{con} and the enthalpies (h_A and h_B) from the refrigerant equation of state.
7. Evaluate the capacity ($\dot{Q}_{eva}$), the heat rejection ($\dot{Q}_{con}$) and the power consumption ($\dot{W}$) using Eqs. 14 – 18.

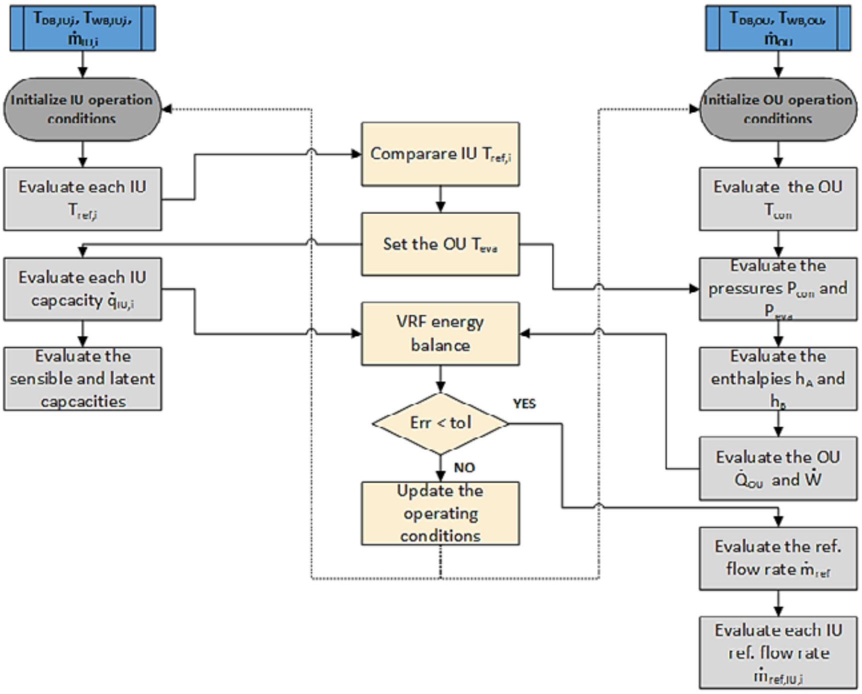

Fig. 4. Layout of the VRF heat pump model in cooling mode

8. Evaluate the errors in the VFR energy balance:

$$\Delta\dot{Q}_{eva} = \dot{Q}_{eva} - \sum_i \dot{q}_{IU}^{(i)} \tag{20}$$

$$\Delta\dot{W} = \dot{W} - \dot{W}^{-} \tag{21}$$

$$\Delta\dot{Q}_{con} = \dot{W} - \dot{Q}_{eva} - \dot{Q}_{con} \tag{22}$$

where $\dot{W}^{-}$ is the power consumption at the previous iteration.

9. Evaluate the sum of the heat rejection and power consumption absolute relative errors:

$$Err = \left|\Delta\dot{Q}_{con}/\dot{Q}_{con}\right| + \left|\Delta\dot{W}/W\right| \tag{23}$$

10. If Err is above the tolerance, update the VRF state for the next iteration:

$$\dot{Q}_{eva}{}^{+} = \dot{Q}_{eva} + \delta\Delta\dot{Q}_{eva} \tag{24}$$

$$\dot{Q}_{con}{}^{+} = \dot{Q}_{con} + \delta\Delta\dot{Q}_{con} \tag{25}$$

$$\dot{W}^{+} = \dot{W} + \delta\Delta\dot{W} \tag{26}$$

$$\dot{q}_{IU}^{(i)+} = \dot{Q}_{eva}{}^{+}\frac{\dot{q}_{IU}^{(i)}}{\sum_i \dot{q}_{IU}^{(i)}} \tag{27}$$

where δ is a relaxation coefficient. A relaxation coefficient of 0.7 and a tolerance of 10^{-8} are used in this study. The new VRF state is then used to reinitialize the IU and OU before the next iteration, starting from step 2.

11. If Err below the tolerance, evaluate the refrigerant mass flow rate for each IU using the refrigerant routines and Eq. 17:

$$\dot{m}_{ref} = \frac{\dot{V}_{nominal}}{v_{suc}} - \dot{m}_{leak} \tag{28}$$

$$\dot{m}_{ref,IU}^{(i)} = \dot{m}_{ref} \frac{\dot{q}_{IU}^{(i)}}{\sum_i \dot{q}_{IU}^{(i)}} \tag{29}$$

4. Calibration procedure

Parameters to the models are unknown, as manufacturers do not typically provide values of the VRF heat pump parameters. Here, parameters are estimated by calibration of the models over the published manufacturer performance data. The IU model has two parameters (UA_{in} and $h_c A_{out}$), while the OU model has 8 (UA_{in}, $h_c A_{out}$, $\dot{V}_{nominal}$, V_r, C, η, $\dot{W}_{loss}$ and ΔT_{sup}). For IUs, manufacturers usually provide data of latent and sensible capacities at different airflow temperatures (dry bulb and wet bulb) and flow rates. For OU, manufacturers provide the power consumption in addition to latent and sensible capacities at different airflow temperatures. Regarding IU, sometimes the air flow rate is given as a set of two or three values (corresponding to the fan speed control, e.g., Low, Med, Hi) and it is not clearly indicated which value is used to generate the data. In such case, the calibration is performed successively with the different values of air flow rate. The air flow rate that leads to the minimal error between the model and the data is retained.

4.1. Indoor unit

Component models for IUs as well as calibration routines are implemented in Python. The parameters that minimize the sum of squared errors between the model and the manufacturer data are identified using the Sequential Least Squares Programing method (SLSQP) from the Python SciPy library:

$$SSE_{IU} = \sum_j \left[\left(\frac{\dot{q}_{sen,IU}^{(j)} - \dot{q}_{sen,data}^{(j)}}{\dot{q}_{sen,data}^{(j)}} \right)^2 + \left(\frac{\dot{q}_{lat,IU}^{(j)} - \dot{q}_{lat,data}^{(j)}}{\dot{q}_{lat,data}^{(j)}} \right)^2 \right] \tag{30}$$

where for the set of data points (j): $\dot{q}_{sen,IU}^{(j)}$ and $\dot{q}_{lat,IU}^{(j)}$ are the IU model sensible and latent capacities, $\dot{q}_{sen,data}^{(j)}$ et $\dot{q}_{lat,data}^{(j)}$ are the latent and the sensible capacities form the IU manufacturer data

For the calibration procedure, it is important to choose proper guess values for the parameters to ensure that the result converges in any case. Cimmino et al. [12] proposed a procedure based on the nominal values of the capacity and power input to evaluate the guess parameters. This approach was implemented assuming a pinch value of 5K in the heat exchanger. The initialization of the IU parameters follows this sequence:

1. Set the refrigerant temperature difference : $\Delta T = 5\,°C$ (31)
2. Evaluate the IU heat transfer coefficients: $h_c A_{out} = UA_{in} = 2\dot{Q}_{nominal}/\Delta T$ (32)

4.2. Outdoor unit

While an OU embodies only a compressor and a single heat exchanger, a second heat exchanger is included in the calibration model to complete the cycle and evaluate the OU capacity and power input. The heat transfer of this heat exchanger is not taken into account in the cost function but its heat transfer coefficient is still calibrated. Moreover, the exchanger is assumed to operate in sensible heat transfer mode in the air side and the air mass flow rate is set to the same value as the one given by the manufacturer for the outdoor unit heat exchanger. The same method as for the IU is used to minimize the sum of squared errors between the model and the manufacturer data for the OU:

$$SSE_{OU} = \sum_j \left[\left(\frac{\dot{Q}_{OU}^{(j)} - \dot{Q}_{data}^{(j)}}{\dot{Q}_{data}^{(j)}} \right)^2 + \left(\frac{\dot{W}^{(j)} - \dot{W}_{data}^{(j)}}{\dot{W}_{data}^{(j)}} \right)^2 \right] \tag{33}$$

where for the set of data points (j): $\dot{Q}_{OU}^{(j)}$ and $\dot{W}^{(j)}$ are the OU model heat rejection and power consumption respectively and $\dot{Q}_{data}^{(j)}$ and $\dot{W}_{data}^{(j)}$ are the heat rejection and power consumption from the OU manufacturer data.

For the calibration procedure, the guess values assume a pinch value of 5K in the IU and OU heat exchangers. The heat transfer coefficient of the inside and the outside surface are set at the same initial value. The leakage mass flow rate is set at 1% of the refrigerant mass flow rate. The guess value of the electro-mechanical efficiency is chosen to be $\eta = 0.95$ and the superheating, $\Delta T_{sup} = 5$K. The nominal values of the temperatures are the one at the rated condition.

The guess values of the OU parameters are evaluated with this following sequence in cooling mode:

1. Evaluate the refrigerant temperature:

$$T_{eva} = T_{DB,in,eva,nominal} - 5°C \tag{34}$$

$$T_{con} = T_{DB,in,con,nominal} + 5°C \tag{35}$$

2. Evaluate the evaporator heat transfer rate at nominal conditions:

$$\dot{Q}_{eva,nominal} = \dot{W}_{nominal} - \dot{Q}_{con,nominal} \tag{36}$$

3. Evaluate the evaporating pressure p_{eva} and condensing pressure p_{con} from the refrigerant equation of state.
4. Evaluate the specific volume and isentropic exponent at the compressor suction from the refrigerant equation of state using the condensing pressure p_{con} and the suction temperature $T = T_{eva} + \Delta T_{sup}$
5. Evaluate the volume ratio:

$$V_r = (p_{con}/p_{eva})^{\frac{1}{\gamma}} \tag{37}$$

6. Evaluate the nominal refrigerant volume flow rate:

$$\dot{V}_{nominal} = (\dot{m}_{ref} + \dot{m}_{leak})v_{suc} \tag{38}$$

$$\dot{m}_{ref} = \dot{Q}_{eva,nominal}/(h_A - h_B) \tag{39}$$

$$\dot{m}_{leak} = 0.01\dot{m}_{ref} \tag{40}$$

7. Evaluate the leakage coefficient:

$$C = \dot{m}_{leak}/(p_{con}/p_{eva}) \tag{41}$$

8. Evaluate the constant part of the power losses with the previously evaluated parameters and the theoretical power (Eq. 14):

$$\dot{W}_{loss} = \max(0, \eta\dot{W}_{nominal} - \dot{W}_t) \tag{42}$$

9. Evaluate the condenser and evaporator heat transfer coefficients:

$$\{UA_{eva}, UA_{con}\} = \dot{Q}_{eva,nominal}/5°C \tag{43}$$

10. Evaluate the evaporator internal and external heat transfer coefficients:

$$h_c A_{out,eva} = UA_{in,eva} = 2UA_{eva} \tag{44}$$

5. Results

5.1. Indoor unit calibration results

A commercial IU of 3.5 kW nominal capacity is used to verify the model and the calibration procedure in cooling mode. Values of sensible and latent heat transfer rates are given for different sets of dry bulb and wet bulb temperature. The temperature range goes from 22.7°C to 31.1°C for a total of 24 data points. Two air mass flow rates (0.104 kg/s and 0.168 kg/s), corresponding to two fan speeds, are given in the data. The calibration is performed with each of the two air mass flow rates and the best result is retained. The guess values and the calibration result of each parameter are shown in Table 1. Figure 5 shows the comparison between the model predicted and the manufacturer provided values of the IU sensible, latent and total capacities.

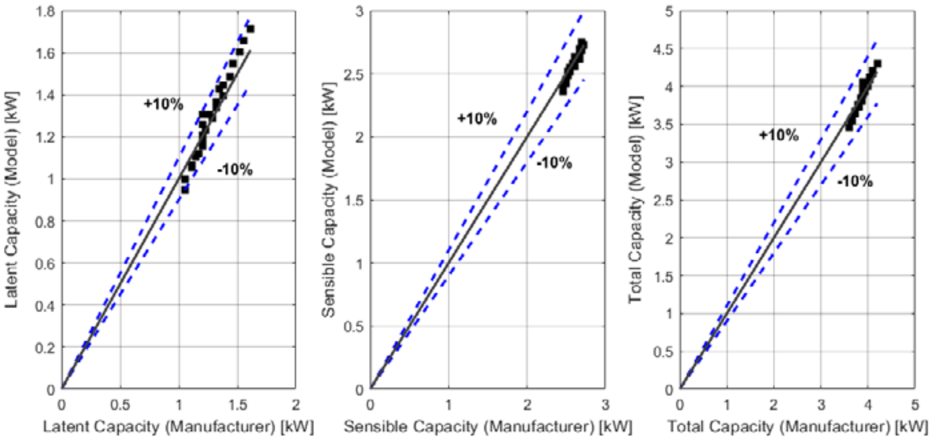

Fig. 5. Comparison of manufacturers and calibrated IU capacities

Table 1. IU parameters calibrated with manufacturer data

Parameters	Guess values	Calibrated values
UA_{in} $[W/K]$	1406.4	1543.1
$h_c A_{out}$ $[W/K]$	1406.4	1567.6

The root mean square error (RMSE) is 4.9%, 1.5% and 1.9% for the latent, sensible and total capacities, respectively. The maximum difference between values from the model and the manufacturer data is 10.3%, 4% and 4% for the latent, sensible and total capacities, respectively. The error on the latent capacity is caused by the assumption of a coil that is completely wet. The results are acceptable in the context of energy simulation, considering that the total capacity is the most relevant performance metric in this case. Various IU sizes from different manufacturers have been also verified and similar results have been obtained.

5.2. Outdoor unit calibration results

The OU model and calibration method is also verified against manufacturer data. Data from a 10.5 kW nominal capacity outdoor unit is used to calibrate the model in cooling mode. The nominal coefficient of performance (COP) of the OU is 3.37. As the OU heat exchanger operates as a condenser, only the external heat transfer coefficient ($h_c A_{out}$) is calibrated. Table 2 shows the guess values and the calibration result of each parameter and Figure 6 shows the comparison between the model and the manufacturer data of power and capacity.

An RMSE of 2.1% and 2.3% is obtained for the power input and the capacity, respectively. The maximum differences are 8.8% and 7.3% for the power input and the capacity, respectively. Similar results have been obtained with OUs of different capacities and from different manufacturers.

Table 2. OU parameters calibrated with manufacturer data.

Parameters	Guess values	Calibrated values
$UA_{in}\ [W/K]$	-	-
$h_cA_{out}\ [W/K]$	2491	1016.1
$\dot{V}_{nominal}\ [m^3/s]$	0.001	0.0026
$V_r\ [-]$	2.16	2.45
$C\ [kg/s]$	0.0091	0.0
$\eta\ [-]$	0.95	0.99
$\dot{W}_{loss}\ [W]$	358.7	0.0
$\Delta T_{sup}\ [°C]$	5.0	0.5

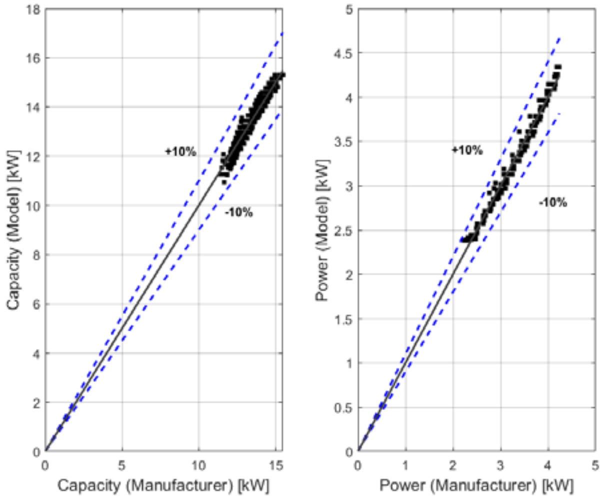

Fig. 6. Comparison of manufacturers and calibrated power and capacity of the OU

5.3. VRF model results

The VRF model is verified by assembling an OU and several IUs. Since manufacturers do not communicate data of assembled VRF units, the performance metrics of the model are compared to the available manufacturer data for the corresponding OU. Previously calibrated OU and IUs are used to build different configurations of VRF units. Table 3 and Table 4 show the parameters of the IUs and OU considered. The parameters are obtained by calibration, while the air mass flow rates are the values given by the manufacturer and used during the calibration process. Three cases corresponding to three VRF configurations are evaluated.

Table 3. Calibrated indoor units used to build VRF configurations.

Indoor unit	Capacity $[kW]$	$h_cA_{out}\ [W/K]$	$UA_{in}\ [W/K]$	$\dot{m}_a\ [kg/s]$
A	2.6	1192	506	0.162
B	3.5	1567	1543	0.168
C	5.3	4312	3211	0.231
D	7	5723	3630	0.272

The units are assembled to maintain a capacity ratio between OU and IUs equal to one. In case 1, the OU is paired to 3 IUs (2A + 1C). In case 2, 3 IUs of same capacity (3B) are connected to the OU. In case 3, the OU is paired to 2 IUs (1B + 1D). Comparisons of the VRF input power and capacity between the calibrated model and the OU manufacturer data are shown on Figure 7. Table 5 shows that the RMSE is in range of 1.8% to 3.5% for the power input and 1.9% to 5.1% for the capacity.

Table 4. Calibrated outdoor unit used to build VRF configurations.

Parameters	Values
UA_{in} $[W/K]$	1834
$h_c A_{out}$ $[W/K]$	-
$\dot{V}_{nominal}$ $[m^3/s]$	0.002325
V_r $[-]$	2.807
C $[kg/s]$	0.0
η $[-]$	0.81
$\dot{W}_{loss}$ $[W]$	0.0
ΔT_{sup} $[°C]$	1
$\dot{m}_a$ $[kg/s]$	2.162

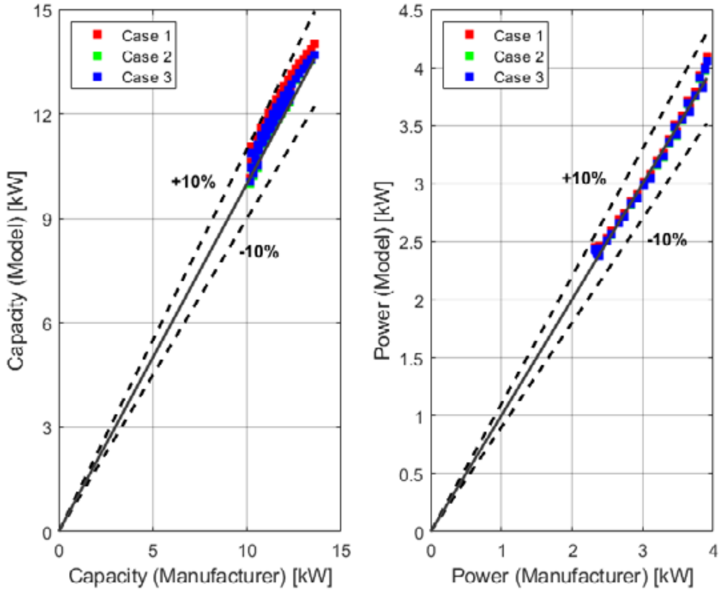

Fig. 7. Comparison of assembled VRF and manufacturer power and capacity

Table 5. VRF configurations and RMSE values compared to manufacturer data

Case	Configuration	Error on the Power		Error on the Capacity	
		Maximum	RMSE	Maximum	RMSE
1	OU + IUs {A, A, C}	5.2%	2.1%	8.3%	5.1%
2	OU + IUs {B, B, B}	4.4%	3.5%	6.3%	1.9%
3	OU + IUs {B, D}	4.4%	1.8%	6.2%	3.3%

6. Conclusions

A model for an air source variable refrigerant flow heat pump is presented. The model is based on a simplified vapor compression cycle to reduce the number of thermodynamic properties evaluation and maintain a low computational time. The IUs and OU are modelled separately so that various VRF configurations with multiple IUs can be considered. Models for the units and a calibration procedure were implemented in Python. The parameters of the models are calibrated by using manufacturer data and by minimizing the differences between the model's performance metrics and those given in manufacturer data.

Different configurations of VRF heat pumps were evaluated by using previously calibrated IUs and OU. Comparison with manufacturer data showed that the model predicts very closely the input power and the capacity of the outdoor unit. A maximum root mean square error of 2.9% and 2.8% were observed respectively for the capacity and the power input. Future work will be devoted to the implementation of a partially wet coil in the heat exchanger model, defrosting in heating mode, the extension of the model for partial load operation and heat recovery.

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