Development of a simplified model for evaluating refrigeration capacity and power consumption of air conditioning units based on heat exchanger entropic temperature definition

Maxime Boulinguez, Jean Castaing-Lasvignottes, Olivier Marc Marc

DOI: 10.18462/iir.icr.2023.0781

Development of a simplified model for evaluating refrigeration capacity and power consumption of air conditioning units based on heat exchanger entropic temperature definition

Maxime BOULINGUEZ [a][b], Olivier MARC[a], Jean CASTAING-LASVIGNOTTES* [a]

[a] Piment Laboratory, Reunion University
Saint-Pierre, 97410, France,
[b] LEU Reunion
Saint-Pierre, 97410, France,
*Corresponding author: jean.castaing-lasvignottes@univ-reunion.fr

ABSTRACT

Air conditioning represents one of the main energy loads in buildings submitted to a tropical climate. In La Reunion, a French overseas territory located in the Indian Ocean close to Madagascar and Mauritius, it accounts for 40 % of the whole electricity consumption of the Island. Evaluating the latter as well as the amount of the refrigeration capacity is not straightforward, especially during the early design phase. Generally, and for bad safety reasons, oversizing HVAC system (compressors, heat exchangers, pumps fans, …) is very common. It results in operating conditions far from the optimum and leads to unexpected high electricity consumptions. Moreover, nominal working conditions provided by manufacturers are not always representatives of real conditions, often depending on climate, building thermal inertia and building use. To face these challenges, Building Energy Modelling (BEM) is widely used in consultancies but suffers from requiring a quite good and accurate description of the intended chiller or split which is generally not selected at that step.

What is aimed in this work is to present a simple phenomenological model suitable for any manufactured air conditioning machine and able to accurately evaluate refrigeration capacity and electricity consumption thanks to a very few numbers of parameters relative to the heat exchangers and to the compressor.

Keywords: Refrigeration, Model, simulation, parametric identification.

1. INTRODUCTION

Due to the increase on comfort requirement in buildings, particularly in hot climate where air conditioning is widely installed and to the new climatic challenges to come, a particular attention has now to be paid on the right sizing of these equipment in order to reduce both the consumed energy and the environmental impacts. This can be done from a static point of view by selecting the right devise to install or dynamically thanks to the use of Building Energy Modelling (BEM) tools that are now widespread used in consultancies. Whatever the case, a relatively high level of accuracy is required in consumption evaluation and in cooling capacity to avoid oversizing air conditioning systems which is very common in the building sector. Therefore, the development of numerical models able to estimate performances of chillers or air conditioners is required to select the most suitable ones and also to be easily implemented in global simulation tools. Three different ways to do so are reported in the literature (Afram and Janabi-Sharifi, 2014) and called black, white and grey approaches . The first one deals with empirically based or data driven models (by mean of artificial neural network or regression techniques for instance) that are generally very easy to setup and implement and that are consuming very low amount of computing time (Lee and Lu, 2010). These kinds of models suffer from being difficult to be generalized outside of the investigated domain, especially in a climatic context where severe conditions are becoming more and more frequent. White or physics-based models are on the contrary based on the detailed knowledge of the different processes occurring in a refrigeration cycle (American Society of Heating, 2017) and on the corresponding physical principles (fluid dynamics, thermodynamics, heat transfer, …). They are more computing time consuming and long to develop and calibrate. Grey box models are situated in between and their development is rapidly growing. They benefit from both approaches, combining physics knowledge for the basis of the model and data analysis to generally evaluate a few parameters that are tricky to physically determine (Mbaye and Cimmino, 2021). This is what is aimed in this paper: the development of a simple grey box model for evaluating refrigeration capacity, power consumption and performances of air conditioning systems based on physics and data provided by manufacturers catalogues or experiments.

2. REFRIGERATION CYCLE

2.1. Thermodynamic cycle

The studied vapor compression cycle if composed of a compressor, a condenser, an expansion valve and an evaporator. Height points can be defined on the refrigerant loop (Figure 1), leading to height different transformations whose associated transferred energies are summarised in Table1.

The refrigerant mass flow rate flowing in the loop is given by $\dot{m}_{ref}$ and the energy balance applied to the global cycle leads to the following: $\dot{W}_{comp,\,real} + \dot{Q}_{dsh} + \dot{Q}_{cond} + \dot{Q}_{sc} + \dot{Q}_{evap} + \dot{Q}_{sh} = 0$. The Coefficient of Performance, defined as the ratio of the useful heat capacity over the energy required by the compressor to drive the cycle, is given by: $COP = \frac{Q_{evap}}{W_{comp,\,real}}$

Given that, Point is the ideal output of the compressor (considering an isentropic transformation), Point 3 (real output) is then calculated thanks to the definition of the isentropic efficiency of the compression given by: $\eta_{is} = \frac{h_2 - h_1}{h_3 - h_1}$

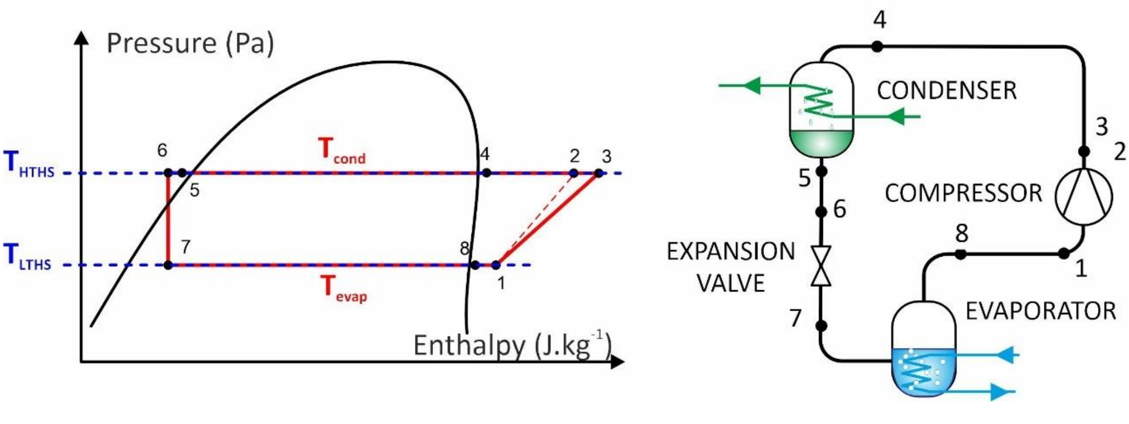

Figure 1: Refrigeration cycle represented on a pressure enthalpy diagram and corresponding scheme

Table 1. Detailed transformations occurring in a refrigeration cycle

Path	Name	Corresponding energy flux
1-2	Isentropic compression	$\dot{W}_{comp,\,is} = \dot{m}_{ref}.(h_2 - h_1)$
1-3	Real compression	$\dot{W}_{comp,\,real} = \dot{m}_{ref}.(h_3 - h_1)$
3-4	Desuperheating in the discharge line	$\dot{Q}_{dsh} = \dot{m}_{ref}.(h_4 - h_3)$
4-5	Desuperheating + condensation + subcooling	$\dot{Q}_{cond} = \dot{m}_{ref}.(h_5 - h_4)$
5-6	Extra Subcooling	$\dot{Q}_{sc} = \dot{m}_{ref}.(h_6 - h_5)$

6-7	Expansion	$h_7 = h_6$
7-8	Evaporation + superheating (usefull)	$\dot{Q}_{evap} = \dot{m}_{ref}.(h_8 - h_7)$
8-1	Extra Superheating (unusefull) in the suction line	$\dot{Q}_{sh} = \dot{m}_{sh}.(h_1 - h_8)$

In this part, neither extra subcooling nor extra superheating is considered, leading to h_6=h_5 and h_1=h_8. When such a cycle is drawn on a ln(P)-h diagram, the main assumption concerning evaporation and condensation temperature is that they correspond to respectively the Low-Temperature Heat Source (LTHS) and the High-Temperature Heat Source (HTHS). This leads to ideal heat transfer conditions in the two heat exchangers of the cycle. As an example, a Vapour Compression Cycle (VCC) for air conditioning application, working with R32 refrigerant and (COOLPROP(Bell et al., 2014) or EES(Klein, 1993)) submitted to:

- an evaporating temperature of 0°C,
- a condensing temperature of +35°C,
- a useful superheating of 3°C (realized in the evaporator)
- a subcooling of 2°C realized in the condenser
- an isentropic efficiency of 80 % for the compressor

This leads to a COP value of 5.11 and coordinates of major points of Table 2

Table 2. Thermodynamic characteristics of main point of the cycle

N°	Temperature	Pressure	Specific enthalpy	Quality
	°C	bar	kJ/kg	%
1	3	8.132	519	100
2	65.43	21.9	559.3	100
3	73.46	21.9	569.4	100
6	33	21.9	261.2	-100
7	0	8.132	261.2	0.1941

2.2. Machine cycle

When considering a real refrigeration machine (i.e. composed of non-ideal heat exchangers and of a real compressor), the cycle differs from the previous one mainly because of heat transfer occurring in the heat exchanger, leading to a condensation temperature higher then the one of the outside air at High Temperature Heat Source (HTHS) and to an evaporation temperature lower than the one of the Low Temperature Heat Source (LTHS), itself represented by room air temperature in the case of air conditioning for instance (Figure 2). These changes contribute to the increase of the pressure ratio the compressor being submitted to and due to a higher condensing pressure and a lower evaporation pressure than in the previous case. The main consequence is a higher amount of power consumed by the compressor to drive the cycle and therefore a lower COP. Generally, the mass flow rate of refrigerant is also impacted by the pressure ratio and contributes to a lower refrigeration capacity.

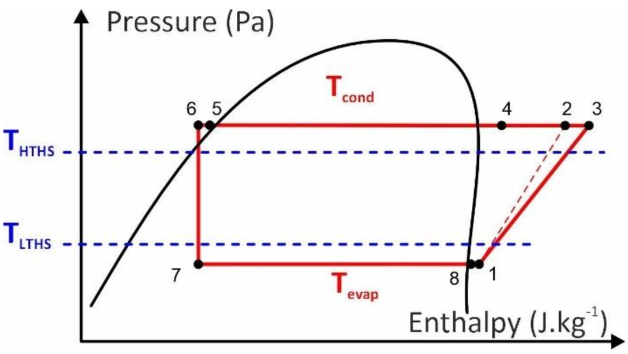

Figure 2: Refrigeration machine represented on a pressure enthalpy diagram

Since there now exists a temperature difference between refrigerant and heat source temperatures due to heat exchanger, those later have to be integrated in the model. This is made possible by the classic method considering UA value for each exchanger and by introducing a representative temperature for the refrigerant called the isentropic temperature. This allows to simplify the global model by considering only one equivalent temperature for the refrigerant on the hot side (between point 3 and 6) and the same for the cold side (from point 7 to 1). These temperatures are calculated thanks to the following equations taking into account the irreversibilities of the system (Invernizzi, 2013):

$$\tilde{T}_{evap} = \frac{h_1 - h_7}{s_1 - s_7} \text{ and } \tilde{T}_{cond} = \frac{h_3 - h_6}{s_3 - s_6} \tag{1}$$

On the heat transfer fluids side, the heat fluxes are simply calculated by their corresponding heat balances and in the end, the following set of three coupled equations must be solved for each exchanger:

$$\begin{cases} \dot{Q}_{evap} = \dot{m}_{ref}.(h_8 - h_1) \\ \dot{Q}_{evap} = UA_{evap}.\dfrac{(T_{LTTF,in} - \tilde{T}_{evap}) - (T_{LTTF,out} - \tilde{T}_{evap})}{ln\dfrac{T_{LTTF,in} - \tilde{T}_{evap}}{T_{LTTF,out} - \tilde{T}_{evap}}} \\ \dot{Q}_{evap} = \dot{m}_{LTTF}.Cp_{LTTF}(T_{LTTF,in} - T_{LTTF,out}) \end{cases} \text{ and } \begin{cases} \dot{Q}_{cond} = \dot{m}_{ref}.(h_3 - h_6) \\ \dot{Q}_{cond} = UA_{cond}.\dfrac{(\tilde{T}_{cond} - T_{HTTF,in}) - (\tilde{T}_{cond} - T_{HTTF,out})}{ln\dfrac{\tilde{T}_{cond} - T_{HTTF,in}}{\tilde{T}_{cond} - T_{HTTF,out}}} \\ \dot{Q}_{cond} = \dot{m}_{HTTF}.Cp_{HTTF}(T_{HTTF,out} - T_{HTTF,in}) \end{cases} \tag{2}$$

Lastly, when considering a real compressor, one has to account for its size (represented by its swept volume at least) and for its volumetric efficiency that contributes to lower the mass flow rate flowing through it. This latter depends now on suction conditions (at point 1), the value of the volumetric efficiency (generally depending on pressure conditions) and on the specific compressor being installed.

The machine is then represented by a few parameters: UA values for the two heat exchangers, an isentropic and a volumetric efficiency as well as a swept volume for the compressor. The model proposed in our case to simulate its behaviour when submitted to given conditions requires to define the inputs: the high and low input Temperature of both heat transfer fluid $T_{HTTF,\,in}$ and $T_{LTTF,\,in}$, their both heat capacities CP_{HTTF} and CP_{LTTF}, and their both mass flow rate $\dot{m}_{HTTF}$ and $\dot{m}_{LTTF}$. The most important outputs are the different exchanged heat fluxes, the power consumed by the compressor and the corresponding COP. Since the working fluid is also part of the inputs, all the points of the cycle are also calculated as well as phase change temperatures T_{cond} and T_{evap}. Concerning heat transfer fluids, the output temperatures are also calculated by the model.

If the swept volume of the compressor is generally given by manufacturers, only an order of magnitude of UA values can be evaluated and so as for the isentropic and volumetric efficiencies. Nevertheless, manufacturers are generally giving performances for a limited set of selected conditions and if necessary experiments can be done to access to these values. Therefore, the model can be used indirectly to identify the missing characteristics by mean of criterion minimization techniques. This method is being described in the following part.

3. PARAMETRIC IDENTIFICATION

3.1. Identification based on manufacturer's data

Since the objective of the method is to access to the missing main characteristics of the refrigeration machine, the easiest and cheapest way to do so is to take advantage of manufacturer's data provided generally in their catalogue or their websites. As an example, a chiller from Carrier (AQUASNAP 30 RW/RWA model 160) working with R407C and cooled by water has been used in the following. 3 different points have been selected and are summarized in Table 3.

Table 3. Selected data from manufacturer for identification process

N°	Input/Output chilled temperature (°C)	Input/Output cooling temperature (°C)	Refrigeration Capacity (kW)	COP	Consumed power (kW)
A	12/7	30/35	162	4.538	35.7
B	13/8	30/35	168	4.693	35.8
C	15/10	30/35	180	5.014	359

Both EES software which integrates minimization tools and fluid properties and a dedicated tool based on Excel and the use of the integrated Solver coupled with COOLPROP, a fluid property library, have been used and tested to identify the missing parameters. The criterion ε to be minimized is the sum of the absolute relative differences of the power consumed by the machine, the refrigeration capacity and the COP.

$$\varepsilon = \left|\frac{\dot{Q}_{evap,calc} - \dot{Q}_{evap,man}}{\dot{Q}_{evap,man}}\right| + \left|\frac{\dot{W}_{comp,calc} - \dot{W}_{comp,man}}{\dot{W}_{comp,man}}\right| + \left|\frac{COP_{calc} - COP_{man}}{COP_{man}}\right| \tag{3}$$

Identified values for UA, isentropic and volumetric efficiencies in reported in Table 4. Both used tools differ in terms of minimization methods (residual for EES, genetic algorithm, simplex or gradient for Excel/Solver) but gave the same results. A graphic comparison of manufacturer and calculated values obtained for these three cases is made Figure 3.

Table 4. Identified Values

US_{cond} (kW. K^{-1})	US_{cevap} (kW. K^{-1})	η_{vol}	η_{is}
23	30	0.72	0.76

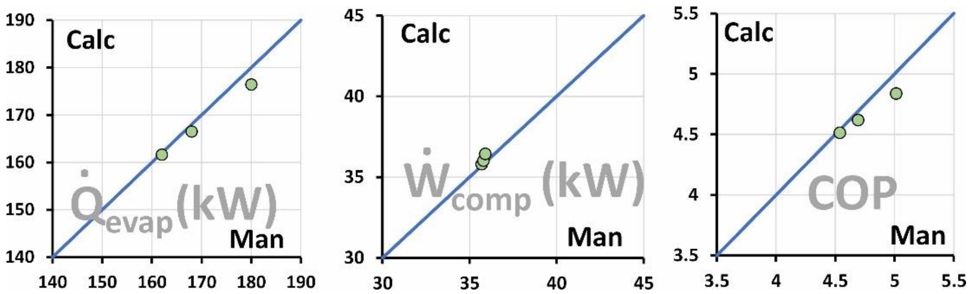

Figure 3: Obtained results during the identification process: comparison between model and manufacturer's data

To validate the method, another set of data extracted from the manufacturer's catalogue has been used. 6 other cases were selected and the main characteristics are summarised in Table 5. Of course, the model machine remains the same and the difference is in the cooling water temperature entering the condenser. The model, in this case is used as direct, i.e. parameters are set constant and equals to the identified ones, and the outputs are $\dot{Q}_{evap}$, $\dot{W}_{comp}$ and COP. A graphic comparison of results is proposed Figure 4 where a quite good relationship between the manufacturer's data and calculation can be observed.

Table 5. Selected data from manufacturer for validation process

N°	Input/Output chilled temperature (°C)	Input/Output cooling temperature (°C)	Refrigeration Capacity (kW)	COP	Consumed power (kW)
D	12/7	35/40	154	3.84	40.1
E	13/8	35/40	159	3.955	40.2
F	15/10	35/40	170	4.218	40.3
G	12/7	40/45	144	3.193	45.1
H	13/8	40/45	150	3.319	45.2
I	15/10	40/45	160	3.532	45.3

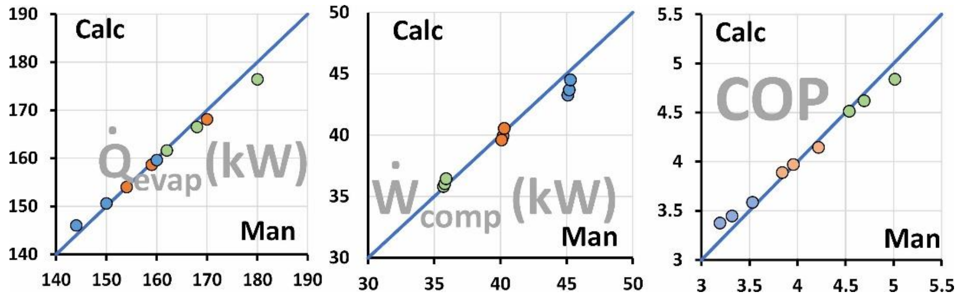

Figure 4: Comparison between model and manufacturer's data

Since the method appeared interesting, the same tests have been done for a commercial heat pump (Carrier AQUASNAP 61 WG model 020) working with R410A and cooled by water. As before, a set of data is used for identifying the missing parameters and another different set is used to validate the obtained values. All the investigated cases are summarized in the Table 6 where different working conditions are varied, such as the entering temperature on the condenser side and the entering temperature on the evaporator side. The cases used for identification are A, B and C (leading to the Table 7 containing the characteristics of the machine) and the others are used for validation. The performance indicator for heat pump cycle is called Coefficient Of Amplification and defined as: $COA = \frac{Q_{cond}}{W_{comp,\,real}}$.

Table 6. Selected data from manufacturer for identification and validation process in the case of the heat pump

N°	Input/Output Low temperature Heat source (°C)	Input/Output High temperature Heat source (°C)	Heating Capacity (kW)	COA	Consumed power (kW)
A	10/7	30/35	28.9	5.75	5.03
B	10/7	40/45	27.6	4.53	6.09
C	10/7	55/65	25.7	3.01	8.54
D	15/12	25/30	34.1	6.63	5.14
E	15/12	30/35	33.1	5.95	5.56
F	15/12	35/40	32.2	5.33	6.04
G	15/12	40/45	31.3	4.78	6.55
H	18/15	25/30	37	7	5.29
I	18/15	30/35	35.9	6.29	5.71
J	18/15	35/40	34.9	5.65	6.18
K	18/15	40/45	33.8	5.06	6.68

Table 7. Identified Values for the heat pump case

US_{cond} (kW. K^{-1})	US_{cevap} (kW. K^{-1})	η_{vol}	η_{is}
5.5	3.5	0.78	0.80

Figure 5 presents the whole comparison between manufacturer's values and calculated ones for all the investigated cases. Here again, the model tends to represent relatively accurately the behavior of the machine.

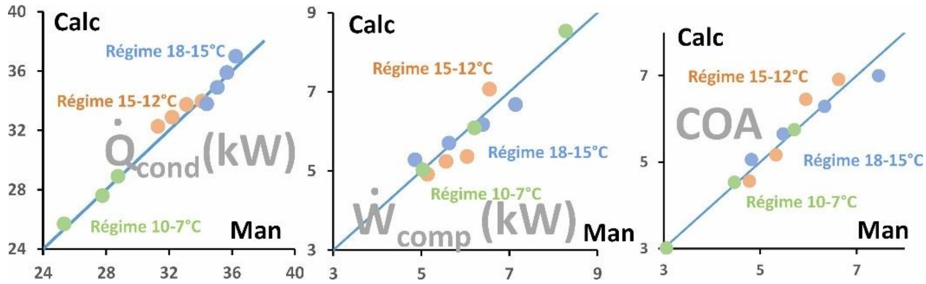

Figure 5: Comparison between model and manufacturer's data

3.2. Identification based on experiments

A few years ago, a test bench was installed in the laboratory to measure performances of individual air conditioning systems, mainly in the case where they are submitted to severe climatic conditions encountered in tropical environment in terms of temperature and humidity. It consists in two climatic chambers separated by a wall representing the interface between internal and external conditions, as in a house or a building (Figure 6). Composed of two units, tested split systems are installed on both sides of the wall. Thermal and hydric loads are provided by respectively an electrical heater and a vapour generator located in the internal room while heat generated by the condenser in the room representing external conditions is extracted by mean of an air to liquid heat exchanger, itself connected to a refrigeration machine.

Figure 6: Test bench, outside view (left), cooled room (right, up), outside conditions (right, down)

The two rooms have their own sensors measuring air temperature and humidity and each unit is also equipped with temperature and humidity probes on the air side of the heat exchanger but also with anemometers so as to measure the volumetric flow rate of air passing through each unit (Figure 6). Energy consumption or amount of generated vapour are also monitored in each room. On the refrigerant side, the units are connected together through a specific facility, allowing the measurement of mass flow rate, pressure and temperature at few points of the refrigerant loop. Thanks to this equipment a certain number of tests have been done on different split systems under various conditions. It has been the case for a HITACHI 9000 model working with R32 and obtained experimental values are used to validate the above developed method when applied to a split system.

Table 8. Selected data from manufacturer for identification and validation process in the case of the heat pump

N°	Input temperature evaporating unit (°C)	Input temperature condensing unit (°C)	Cooling Capacity (kW) Exp./Mod.	COP Exp./Mod.	Consumed power (W) Exp./Mod.
A	25.85	34.89	2.08/2.15	3.81/3.7	547/579
B	25.01	25.03	2.08/2.17	4.69/4.43	445/492
C	25.05	29.86	2.097/2.15	4.31/4.14	487/520
D	25.02	34.83	2.145/2.11	3.62/3.67	593/575
E	26.48	29.93	2.496/2200	4.1/4.17	611/557

Table 8 summarizes 5 different conditions applied to this split. The results in terms of heat fluxes, power flux and COP are also part of the table. The first three points are used to identify the characteristics of the machine in terms of UA and of efficiencies whose values are reported Table 9. The last two points are used in the end to compare experiments and simulated results. The observed differences remain lower than 10 %, concluding to a rather good accuracy of the whole methodology in this case

Table 9. Identified Values for the test bench case

US_{cond} (W. K^{-1})	US_{cevap} (W. K^{-1})	η_{vol}	η_{is}
200	110	0.68	0.82

4. CONCLUSION

The objective of the present work was to develop an original grey box model for predicting COP, consumed power and useful power (cooling capacity in the case of a refrigeration cycle or heating effect in the case of a heat pump). The method leads to the definition of 4 parameters that are identified thanks to manufacturer's data one can access from manufacturer's catalogues for 2 examples and from experimental data obtained on a bench built to measure split systems performances. 3 different kind of machines have been studied, a chiller, a heat pump and a split system, each one working with a different fluid (R407C, R410A and R32). The validation process concludes to a rather good accuracy, whatever the investigated case. Next steps consist in first continuing harvesting data relative to split systems thanks to the test bench. Secondly, building a library of machines described through their 4 parameters is also aimed so as to be able to use them in Building Energy Modelling (BEM) and to select the best one able to maintain a required comfort at lower cost during the hot season for instance.

REFERENCES

Afram, A., Janabi-Sharifi, F., 2014. Review of modeling methods for HVAC systems. Applied Thermal Engineering 67, 507–519. https://doi.org/10.1016/j.applthermaleng.2014.03.055

American Society of Heating, R. and A.-C.E., 2017. 2017 ASHRAE handbook: Fundamentals : SI edition. ASHRAE, 2017, ©2017, Atlanta, GA.

Bell, I.H., Wronski, J., Quoilin, S., Lemort, V., 2014. Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp. Ind. Eng. Chem. Res. 53, 2498–2508. https://doi.org/10.1021/ie4033999

Invernizzi, C.M., 2013. Closed power cycles: thermodynamic fundamentals and applications. Springer, New York.

Klein, S.A., 1993. Development and integration of an equation-solving program for engineering thermodynamics courses: EES FOR ENGINEERING THERMODYNAMICS. Computer Applications in Engineering Education 1, 265–275. https://doi.org/10.1002/cae.6180010310

Lee, T.-S., Lu, W.-C., 2010. An evaluation of empirically-based models for predicting energy performance of vapor-compression water chillers. Applied Energy 87, 3486–3493. https://doi.org/10.1016/j.apenergy.2010.05.005

Mbaye, A., Cimmino, M., 2021. A parameter-estimation model for variable refrigerant flow heat pump systems. Presented at the 13th IEA Heat Pump Conference 2021: Heat Pumps –, Korea.