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MODELING AND SIMULATION OF A CASSETTE FAN COIL SYSTEM FOR EFFICIENT COOLING IN INDOOR ENVIRONMENTS

Flávio Júnior Santiago Silva
Diego de Lima Sousa
Iago Gomes da Costa
Anne Oliveira
Carlos Henrique Matiolo
Heloisa da Silva Pinheiro
José Viriato Coelho Vargas
Sustainable Energy Research and Development Center (NPDEAS), Department of Mechanical Engineering, UFPR - Federal University of Paraná, CP 19011, 81531-980 Curitiba, PR, Brazil.
e-mails:flaviojrsantiago@gmail.com, wbalmant@gmail.com, diegofine@gmail.com, iago_gome_costa@gmail.com, annewcaroline@hotmail.com, carlos.matiolo@gmail.com, helo.rps@gmail.com, vargasjvcv3@gmail.com.

André Bellin Mariano
Sustainable Energy Research and Development Center (NPDEAS), Department of Electrical Engineering, UFPR - Federal University of Paraná, CP 19011, 81531-980 Curitiba, PR, Brazil.
e-mail:andrebmariano@gmail.com.

Abstract. *With the development of economy and industry, the demand for energy has increased significantly. The annual energy consumption due to air conditioning and refrigeration equipment corresponds to approximately 10% of the energy produced worldwide. In this context, it has attracted an increasing deal of interest thanks to such advantages as utilization of low-grade heat sources and environment-friendly working fluid pairs. In addition to using working fluids with low environmental impact compared to traditional refrigeration systems, and thereby making absorption refrigerators commercially competitive. This article presents a mathematical modeling of an absorption refrigeration system to simulate the air conditioning of two spaces, with the fan coil system being the main component to be evaluated. The development of the mathematical equations of the model was based on the application of the principles of mass and energy conservation in transient regime for the refrigerator components, reservoir, fan coil, and the environments, where each component was modeled as a simple control volume. The choice of the fan coil system to be evaluated in the model is due to its importance in air conditioning of environments, especially in commercial and residential buildings. After the development of the mathematical model, a computational code was elaborated in Fortran®90 language in order to obtain numerical responses for each control volume and perform analyses based on the parameters of the system. The mathematical model resulted in a system of ordinary differential equations (ODEs), which were solved through time integration with the aid of the Runge-Kutta-Fehlberg method of 4th/5th order, interacting simultaneously. Thus, the mathematical modeling of the fan coil system was able to provide valuable information for the analysis of the refrigeration system performance and its importance in air conditioning of environments. After completing the mathematical modeling of the system, the simulated results will be presented, focusing specifically on cassette type fan coils. These simulations will provide detailed information about two main aspects: the environment in which the fan coils are installed and the mass flow rate of the system. Additionally, a parametric analysis will be conducted to evaluate different configurations and parameters related to the fan coil. These simulated results are of utmost importance to understand the behavior and performance of cassette type fan coil, allowing for in-depth analysis and assisting in decision-making for optimization and improvement of the overall system.*

Keywords: *Air conditioning, Mathematical modeling and Fan coil system.*

1 INTRODUCTION

Heating, Ventilation, and Air Conditioning (HVAC) systems are widely used to create suitable indoor environments. However, these systems account for a significant proportion of building energy consumption. Given the global energy shortage and the increase in living standards, a more efficient use of energy in HVAC systems is desirable (Zhang, 2013).

The construction sector accounts for approximately 40% of global annual final energy consumption. Heating or cooling a closed space to maintain thermal comfort is one of the main sources, representing up to 60-70% of energy consumption in buildings (Gamarra, 2018). The increasing trend in energy consumption in buildings will persist in the coming years due to expanding built areas and related energy demands.

Half of the energy consumption and carbon emissions in the construction sector are attributed to refrigeration systems (Sores, 2017). These refrigeration systems are widely used in buildings, industries, and vehicles to maintain comfortable temperatures and for industrial processes requiring controlled temperatures. Air conditioning systems are a common example of refrigeration systems used in buildings. However, these systems typically utilize a vapor compression refrigeration cycle, which relies on non-sustainable energy sources (Whitman, 2016).

In addition to these systems, there are equipment components responsible for air distribution in indoor spaces, which directly influence the cooling system. Inefficient airflow distribution can increase the energy demand for space cooling, especially when the proportion of fresh air is low. This inefficiency in airflow can result in a significant increase in energy consumption, particularly when low-quality cooling sources are used (Johnson, 2016).

Based on this context, research is being conducted on other equipment components to improve the performance of refrigeration systems. One example of such equipment is cassette-type fan coils, which play an important role in efficient heat dissipation or cooling.

1.1 Cassette-type fan coil

Cassette type fan coil units are cooling and heating units that are installed in the ceiling of indoor environments, typically in false ceilings. They are called "cassette" due to their rectangular or square shape resembling a box or compartment. These units are designed to provide efficient cooling or heating in commercial, residential, and institutional settings. Cassette type fan coil units consist of an indoor unit and an outdoor unit connected by refrigerant pipes. The indoor unit is installed in the ceiling and features a fan that distributes the cooled or heated air throughout the space. Additionally, the indoor unit contains cooling or heating coils, which are responsible for heat exchange with the air (Ranada, 2020).

These units often have multiple air ducts that can be directed in different directions, allowing for more even air distribution in the environment. Furthermore, some models of cassette type fan coil units have the capability to adjust the direction of the airflow automatically or manually, providing greater flexibility and adaptability. Cassette type fan coil units are popular in environments where the installation of central air conditioning units is not feasible or practical. They are commonly found in offices, shops, restaurants, hotels, and residences, where they offer an efficient and discreet cooling and heating solution (Zhu, 2023).

It is important to note that cassette type fan coil units are just one of many options available in the market for cooling and heating systems. The choice of the most suitable system will depend on the specific needs of the environment and the cooling or heating requirements (Jahanbin, 2022). One potential drawback of this type of system is the potential lack of inadequate temperature and airflow control, meaning the energy efficiency of cassette type fan coil units can be compromised if temperature and airflow control are not properly configured. Improper settings can lead to energy waste, such as unnecessarily cooling or ventilating areas or providing more air than necessary. In some cases, cassette type fan coil units may not operate efficiently under partial loads, meaning when the cooling demand is lower. This can result in excessive energy consumption when the system is operating at lower load levels (Dogan, 2022).

To address this, mathematical equations are available to simulate the air conditioning process in spaces. These simulations allow for the characterization of all parameters of a refrigeration system, identifying possible issues and even providing improvements for the systems, such as simulating fan coil units. Through these mathematical models, it is possible to conduct detailed analyses of the performance of refrigeration systems, considering factors such as thermal load, airflow distribution, and energy efficiency. Based on these simulations, it is possible to assess the impact of different configurations, optimize system designs, and identify improvement opportunities (Hernandez, 2022).

1.2 Mathematical models in refrigeration design.

Mathematical models play a fundamental role in the design of refrigeration systems, allowing for the prediction and analysis of the thermal performance of the involved equipment. These models are based on thermodynamic principles and the properties of refrigerant fluids, and can provide valuable information for optimizing the design and operation of refrigeration systems. One of the primary uses of mathematical models in refrigeration design is the simulation of the refrigeration cycle, which forms the basis of operation for many refrigeration systems, such as vapor compression cycles. These models enable the calculation of important parameters such as cooling capacity, energy consumption, efficiency, and system operating temperature (Wang, 2017).

Furthermore, mathematical models can be used to analyze the behavior of specific components within the system, such as compressors, condensers, evaporators, and heat exchangers. They allow for the simulation of different operating conditions, such as variations in thermal load and ambient temperature, to evaluate performance and determine necessary adjustments. Mathematical models are also essential for optimizing the design of refrigeration systems. Through the application of mathematical optimization techniques, it is possible to determine the best configurations and design parameters, considering specific constraints and objectives, such as minimizing energy consumption or maximizing cooling capacity (Liang, 2020).

In summary, mathematical models are powerful tools in refrigeration design, enabling the analysis and prediction of system thermal performance, simulation of various operating conditions, and optimization of the design. They play a crucial role in the pursuit of more efficient, economical, and sustainable solutions in the field of refrigeration.

2 METHOOLOGY

The process model comprises basically mass and energy balances and design equations for each process unit, and correlations to estimate process streams enthalpy. The complete model including physical relations and model implementation details (inequality constraints) is presented as supplementary material (Mazzei, 2014).

The following key assumptions are made to derive the mathematical model: i) the refrigeration system is 5 tons of refrigeration (TR). ii) the control volumes are refrigerator, reservoir, fan coil unit 1 and 2, and spaces. iii) each component is treated as **a control volume** (CV). iv) the energy balance in the refrigeration system will be performed only in the water side evaporator, in a single control volume. v) the system will be modeled under **transient operating** conditions. vi) any thermal fluid that enters the fan coil unit and conditions the room returns to the reservoir independently. vii) the working fluid considered is exclusively water and, viii) The coefficient of performance (COP) is defined by the ratio between the energy leaving the system, the heat transfer rate in the evaporator ($\dot{Q}_{evap}$), divided by the energy entering, the heat transfer rate in the generator ($\dot{Q}_{ger}$), and the electrical consumption generated by the pump ($\dot{W}_{bomb}$). Equation 1.

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{ger}.\dot{W}_{bomb}!} \tag{1}$$

2.1 Mathematical Equations

In this item, the mathematical equations that govern the physical phenomena in each control volume will be presented, as shown in Figure 2. It is crucial to highlight that each component of the system is individually modeled but they are coupled together to form the complete system. This means that the equations describing the behavior of each component are developed separately, considering the interactions between them to obtain a comprehensive representation of the system as a whole. This approach allows for a detailed analysis of each component and its influence on the overall performance of the cassette-type fan coil cooling system.

- Control Volume 1 (CV1) - Refrigerator
- Control Volume 2 (CV2) - Reservoir
- Control Volume 3 (CV3) - Fan coil unit 1
- Control Volume 4 (CV4) - Space 1
- Control Volume 5 (CV5) - Fan coil unit 2
- Control Volume 6 (CV6) - Space 2

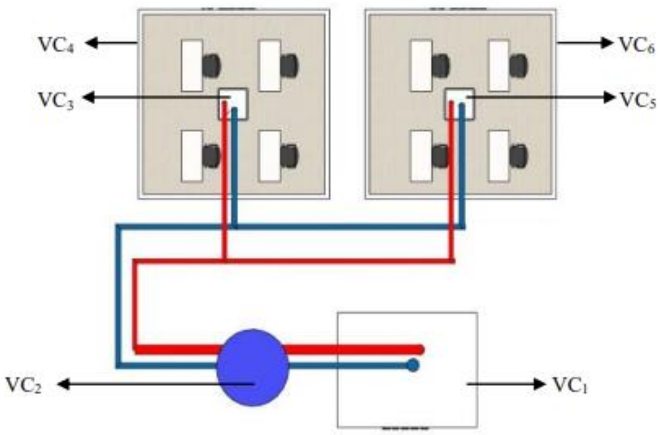


Figure 1: Operation scheme of the system.

2.1.1 Control volume 1

Initially, a mass and energy balance were performed in CV_1, identifying the mass and energy flows entering and leaving this first control volume, as well as the heat loss that occurs in the process, as shown in equation (2). Subsequently, the remaining processes occurring in the refrigerator are modeled based on the same principle.

After identifying the elements entering and exiting the volume, equation (3) demonstrates the application of the mass and energy balance. In this equation, the input parameters are represented by the energy produced in the refrigerator and the mass flow rate ($\dot{m}_r$) of the thermal fluid entering the refrigerator, the specific heat capacity (C_w) and the initial temperature of the thermal fluid (T_r) upon entering the refrigerator. The output parameters of the refrigerator are the same as the input parameters: the water flow rate ($\dot{m}_r$) at which the fluid exits the refrigerator, the specific heat capacity, and the outlet temperature of the thermal fluid. It is worth noting that the cooling capacity of the refrigerator ($\dot{Q}_r$) is 17.580 kW. On the other hand, equation (4) models the variation of the temperature of these same parameters as a function of time.

$$\boldsymbol{E_r} = M_r.C_w.T_r \tag{2}$$

$$\frac{dE_r}{dt} = \dot{m}_r.C_w.T_r - \dot{m}_r.C_w.T_r.\dot{Q}_r \tag{3}$$

$$\frac{dT_r}{dt} = \frac{\dot{m}_r.T_r - \dot{m}_r.T_r}{m_r} - \frac{\dot{Q}_r}{m_r.C_w} \tag{4}$$

2.1.1 Control volume 2

For the equation of CV_2, this control volume is characterized by two stages. In the mathematical modeling of this component, we consider the same factors as the previous control volume but only change the input parameters for a water reservoir. We have an input of the thermal fluid and its output towards the Fan coil unit (CV_3), as well as the return of this fluid from CV_3 to CV_2 and the output of the thermal fluid towards the refrigerator (CV_1). Equation (5) was developed to guide the mathematical modeling of control volume 2. In this equation, ($\boldsymbol{E_{re}}$) represents the energy produced in the reservoir, the specific mass flow rate of the thermal fluid entering the reservoir ($\dot{m}_{re}$), the specific heat capacity ($\boldsymbol{C_w}$) and the temperature at which the fluid is produced (T_{re}) in CV_2.

$$\boldsymbol{E_{re}} = m_{re}.C_w.T_{re} \tag{5}$$

$$\frac{dE_{re}}{dt} = \dot{m}_{re}.C_w.T_r - \dot{m}_{re}.C_w.T_{re} + \dot{m}_f.C_w.T_f - \dot{m}_f.C_w.T_{re} + U_{re}.A_{re}.(T_a - T_{re}) \tag{6}$$

$$\frac{dT_{re}}{dt} = \frac{\dot{m}_{re}.T_r - \dot{m}_{re}.T_{re} + \dot{m}_f.T_f - \dot{m}_f.T_{re}}{m_{re}} - \frac{U_{re}.A_{re}.(T_a - T_{re})}{m_{re}.C_w} \tag{7}$$

Equation (6) integrates CV_2 over time, considering the input parameters such as mass flow rate ($\dot{m}_{re}$), specific heat capacity ($\boldsymbol{C_w}$) and temperature at the inlet of the thermal fluid in the reservoir (T_{re}). The fluid originates from CV_1, and the equation also describes the fluid leaving the reservoir ($\dot{m}_{re}.C_w.T_{re}$). CV_2 receives the return fluid from the fan coil unit, and at this point, the mass flow rate of the fluid coming from the fan coil unit ($\dot{m}_f$), specific heat capacity ($\boldsymbol{C_w}$) and temperature of the thermal fluid in the reservoir (T_{re}), are calculated, taking into account the Global Reservoir Coefficient (U_{re}) and the reservoir area (A_{re}), which affect the ambient temperature (T_a) and reservoir temperature (T_{re}).

Therefore, equation (7) deals with the temperature of the thermal fluid, integrated over time. It includes the same input and output equations of CV_2, as well as adding the term ($U_{re}.A_{re}.(T_a - T_{re})$ which represents the process where the fluid leaves the reservoir at a lower temperature towards the cassette fan coil unit (CV_3). The fan coil unit receives this fluid at a lower temperature, conditioning the environment, and then returns to the reservoir with a temperature closer to the ambient temperature. This takes into account the Global Reservoir Coefficient, which measures a system's capacity to transfer heat between two mediums relative to the reservoir area, the ambient temperature that cools the environment, as well as the temperature of the reservoir water.

2.1.2 Control volume 3

Control Volume 3, which corresponds to a cassette-type hydronic fan coil unit, utilizes a chilled water system for cooling the environments. It is designed to be compact and robust, located within the space where the conditioning will take place. For the mathematical modeling of CV_3, we consider the same factors as in Control Volumes 1 and 2, but with a modification to the energy-producing component, which is the fan coil unit. In this case, we consider the mass flow rate (m_f) f the fluid entering the fan coil unit, the specific heat capacity (C_w) and the temperature (T_f) at which the thermal fluid enters the fan coil unit, as presented in equation (8).

$$\boldsymbol{E_f} = m_f.C_w.T_f \tag{8}$$

$$\frac{dT_f}{dt} = \frac{\dot{m}_f.T_{re}}{m_f} - \frac{\dot{m}_f.T_f}{m_f} - \frac{U_f.A_f.(T_s - T_f)}{m_f.C_w} \tag{9}$$

In equation (9), we perform the integration of the thermal fluid temperature in the fan coil unit (dT_f) with respect to time (dt). In this process, the thermal fluid from the reservoir ($\dot{m}_f.T_{re}$) enters the fan coil unit ($\dot{m}_f$) and the cooling of a space occurs (T_s). We take into account the temperature at which the thermal fluid enters the fan coil unit (T_f) and the area of the fan coil unit (A_f). Subsequently, the fluid exits the fan coil unit at ambient temperature and returns to the reservoir. In this equation, we also consider the Global Coefficient determined for the fan coil unit.

2.1.4 Control volume 4

When modeling Volume Control 4 (VC_4), the following information is taken into consideration: the calculated thermal load for a space ($\dot{Q}_s$), which is 28.967 BTU/h or 8,49 Kw, the ambient temperature (T_s), which is assumed to be 30ºC, and the amount of heat removed from the space by the fan coil unit ($\dot{Q}_f$).

$$\boldsymbol{E_s} = m_s.CV_{ar}.T_s \tag{10}$$

$$\frac{dT_s}{dt} = \frac{U_s.A_s.(T_a - T_s)}{m_s.CV_{ar}} - \frac{U_f.A_f.\left(T_s - T_f\right)}{m_s.CV_{ar}} + \frac{\dot{Q}_s}{m_s.CV_{ar}} \tag{11}$$

The main equation for VC_4, presented as Equation 10, describes the energy generated by the space ($\boldsymbol{E_s}$), the specific heat of air, volume, and constant (CV_{ar}), long with the standard temperature defined for a room (T_s). This equation is crucial in guiding Equation 10, where the temperature of the thermal fluid entering the space is integrated over time (dt). Initially, the equation is developed for the space, considering the global coefficient for that space, the area of the space (A_s) which directly influences the cooling process, and the corresponding temperatures between the room and the external environment ($T_a - T_s$). This expression is then divided by the flow rate of the thermal fluid entering the space and the air constant ($m_s.CV_{ar}$). Next, we have the equation that models the fan coil unit within the space, taking into account the global coefficient for the fan coil unit (U_f), the area of the fan coil unit (A_f), and the temperature difference between the space and the fan coil unit ($T_s - T_f$). his expression is also divided by the flow rate of the thermal fluid entering the space and the air constant. Finally, the thermal load generated by the space ($\dot{Q}_s$) is considered.

For the modeling of the remaining volume controls, the same ammonia absorption chiller was chosen, with a cooling capacity of 5 TR, as well as the 500-liter reservoir. As illustrated in Figure 1, each fan coil unit has a separate inlet and outlet for the thermal fluid, connecting to the reservoir. As a result, the cooling capacity in the space is reduced since the thermal fluid has already been used to cool another space previously.

2.2 Model implemetation

To initiate the simulation of the mathematical modeling, it was necessary to define some of the input parameters of the model. The values of these parameters are presented in Table 1 and were thoroughly described during the mathematical development of each volume control. The global heat transfer coefficients were determined based on literature use to calculate the overall heat transfer coefficient and the heat transfer through a multi-layered wall (Engineering ToolBox, 2003). With this, it will be possible to analyze the influence of these parameters on the environment's air conditioning, as well as the inlet temperatures of the thermal fluid in each component. Subsequently, a parametric analysis will be conducted to evaluate the operational performance of the volume controls.

2.2.1 Numerical method

The numerical problem to be solved consists of the numerical integration of ordinary differential equations (ODEs) that describe the behavior of the system. The mathematical model was computationally implemented using the Fortran®90 programming language. The ODEs are integrated explicitly in time using adaptive time-stepping and the 4th/5th order Runge-Kutta method (KINCAID and CHENEY, 1991). In numerical terms, the time step size is automatically adjusted based on the local truncation error, keeping it below a specified tolerance of 10-6. The developed computational code includes stopping criteria that allow calculations to be performed until a predetermined final time or until a steady-state condition is reached (MARTINHO, 2013).

Table 1: Input parameters for model simulation.

Parameters	Symbols	Value	Unit
Refrigerator heat transfer	$\dot{Q}_r$	17.580.3	kW
Refrigerator volume	V_r	0.005	m^3
Mass flow of fan coil	$\dot{m}_f$	100	kg/s
Reservoir volume	V_{re}	0.5	m^3
Fan coil volume	V_f	0.005	m^3
Fancolete global coefficient	U_f	100	w/m²k
Fan coil area 1	A_f	2	m^2
Fan coil area 2	A_{f2}	2	m^2
Global Coefficient of Spaces	U_s	1	w/m²K
Space area 1	A_{sl1}	48	m^2
Space area 2	A_{sl2}	17	m^2
Global Cooler Coefficient	U_r	500	w/m²K
Refrigerator Saturation Temperature	T_{st}	-18,9	ºC

3 RESULTS

After finalizing the mathematical modeling of the system, the simulated results will be presented, with a specific focus on cassette-type fan coils. These simulations will provide detailed information about two main aspects: the environment in which the fan coils are installed and the mass flow rate of the system. Additionally, a parametric analysis will be conducted to evaluate different configurations and parameters related to the fan coil. These simulated results are of utmost importance to understand the behavior and performance of cassette type fan coil, allowing for in-depth analysis and assisting in decision-making for optimization and improvement of the overall system.

3.1 Mass flow rate of the fan coil

Figure 3 shows the variation of the fan coil mass flow rate in a specific environment, analyzed based on the variation of the ideal thermal fluid flow rate adopted in the methodology of this study. The results observed in Figure 00 demonstrate that initially, an adequate flow rate for the fan coil is 100 kg/s, where the coefficient of performance (COP) is 0.0057 (COP = 0.57). Subsequently, different values were applied to determine the optimal flow rate, which provides the best performance of the fan coil in cooling the room.

In the analyzed system, lower COP values were observed when the flow rate was increased above 100 kg/s, indicating a decrease in performance. Similarly, when the fan coil flow rate was reduced to lower values, such as 0.001 kg/s, the COP was 0.02. On the other hand, when the flow rate was increased to more than 10 kg/s, the COP also tended to decrease. In the range of flow rates between 0.1 m3/s and 0.9 m3/s, the coefficient of performance showed better performance in relation to the fan coil flow rate. Therefore, the ideal flow rate for the fan coil was determined as 1 kg/s, resulting in a COP close to 0.12.

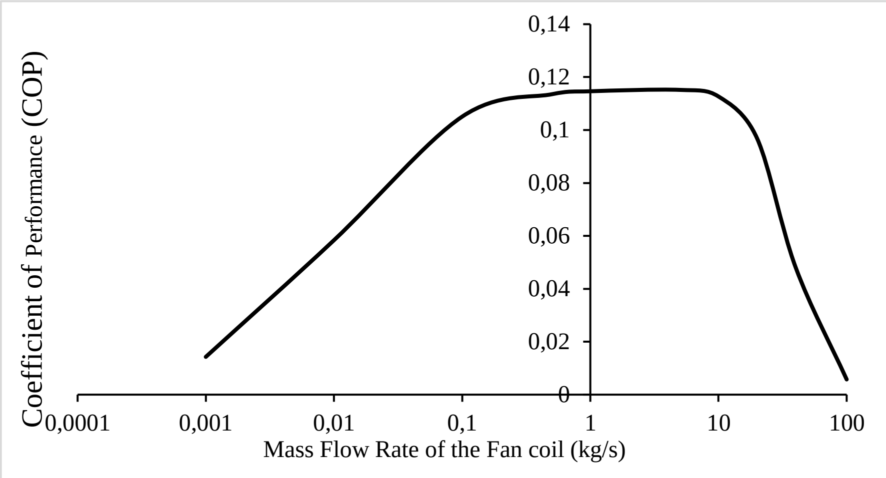


Figure 2: Parametric analysis based on the mass flow rate of the fan coil.

In this second analysis of the fan coil flow rate, we investigated the influence of Control Volume 1 when adopting a maximum cooling load in the environment. Additionally, we conducted a parametric analysis of this parameter in relation to the Coefficient of Performance (COP), testing different flow rates to determine the optimal flow rate for the system under maximum load. The loads varied from those established in the methodology to the limit of thermal comfort.

Figure 4 presents the parameters used for system operation, starting with flow rates above 100 kg/s and gradually increasing. Initially, we observed good performance when flow rates higher than those used in the methodology were adopted. The system achieved excellent Coefficient of Performance (COP) values when adopting a mass flow rate of 200 kg/s, and it even found an optimal flow rate of 300 kg/s with a COP of 0.69. Thus, we can affirm that the higher the mass flow rate adopted for this fan coil, the better its efficiency. However, it was also observed that the system's efficiency started to decline as increasingly higher flow rates were used. Therefore, it is understood that the ideal flow rate when the system operates under maximum load is 300 kg/s.

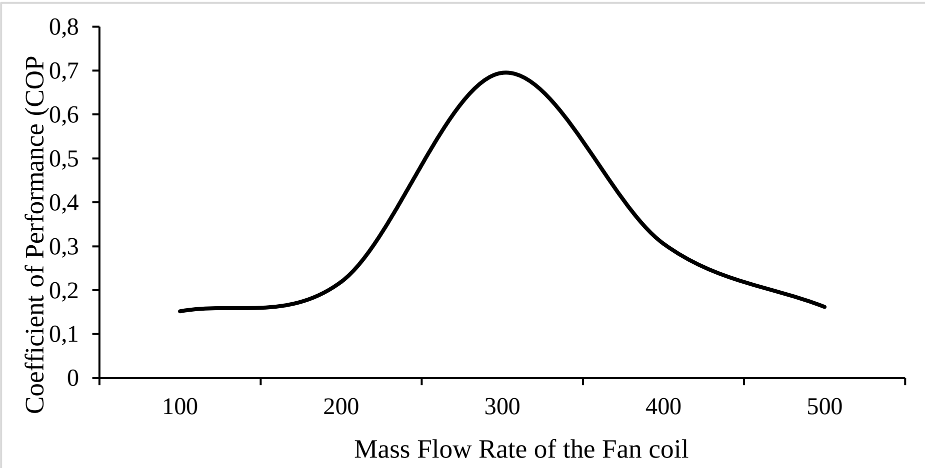
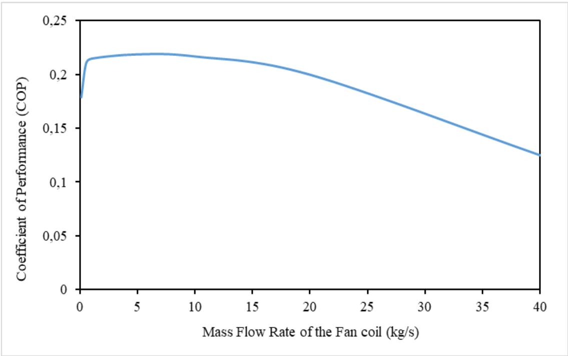


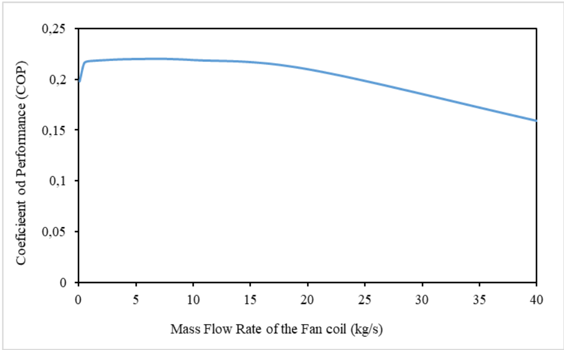
Figure 3: Parametric analysis based on the fan coil flow rate.

3.2 Mass flow rate of the fan coil unit for two spaces

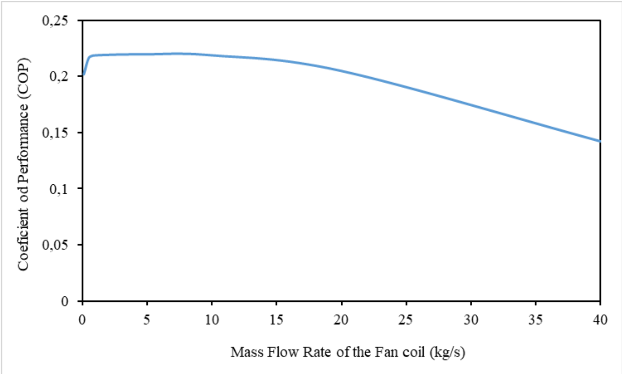
In the simulation for 2 spaces, mass fractions of the thermal fluid were determined for each environment. Initially, a standard value of 0.7 (7%) was set for VC_4, which is a $48m_2$ environment with higher foot traffic and a greater number of equipment, resulting in a higher heat generation. The remaining fluid was directed to the other environment. Therefore, Figure 05 shows the optimizations performed on the fan coil units (VC_3 and VC_5) for environments VC_4 and VC_6, according to the determined fraction for each environment, namely 0.2 (2%), 0.5 (5%), and 0.7% (7%) as a function of the coefficient of performance.



(A)



(B)



(C)

Figure 4: Mass flow rate of the fan coil unit for 0.2% (A), 0.5% (B), and 0.7%(C) fraction as a function of COP.

Figure 5 showcases the simulated model for mass flow rates ranging from 0.1 to 40 kg/s with a fraction of 0.2 (2%). Significantly meaningful values were observed when a mass flow rate lower than 40 kg/s was adopted. The system achieved a performance coefficient of 0.219 (COP = 0.21) when a flow rate of 5 kg/s was implemented.

In the analysis of Figure 5 (B), a fraction of 0.5 (5%) for the refrigerant water was utilized. The system exhibited optimal performance when the mass flow rate was 5 kg/s, resulting in a COP of 0.22. Similarly, in the case of a fraction of 0.7 (Figure 5 (C)), which served as the standard for this mathematical model, the system demonstrated its best performance with a flow rate of 5 kg/s and a COP of 0.22. These findings suggest that the system achieves superior performance in terms of COP when operating at a flow rate of 5 kg/s.

In Figure 6, the optimization of the mass flow rate of the fan coil unit is presented when the system considered the maximum load for VC6 of 18,000 kW, which corresponds to the maximum capacity required to maintain thermal comfort in accordance with the ABNT NBR 16401-2:2008 standards. Initially, when flow rates ranging from 0.1 to 10 kg/s were considered, the system exhibited lower performance than expected for an absorption refrigeration system, with a COP of 0.207 (COP = 0.2). Subsequently, variations were made starting from 40 kg/s, and it was observed that when the system adopted a flow rate of 80 kg/s, the model achieved better performance, resulting in a COP of 0.416 (COP = 0.4). Beyond this flow rate, the system's COP decreased. Hence, it can be concluded that the optimal flow rate for the system when VC6 operates at its maximum load is 80 kg/s.

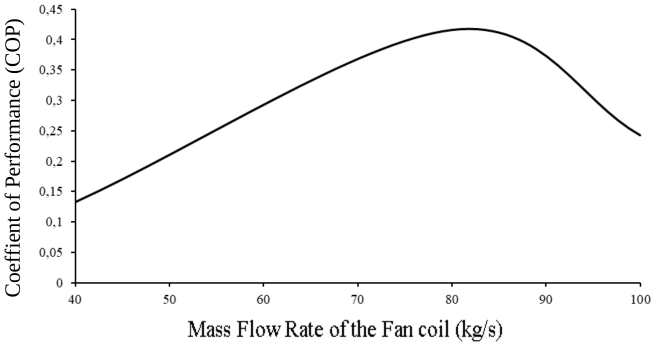


Figure 5: Optimization of fan coil unit mass flow rate under maximum load

4 CONCLUSIONS

Initially, mathematical equations were developed to simulate the application of a 5 TR refrigeration system, specifically focusing on fan coil units. This allowed for the prediction of system operation conditions during the cooling process of the environments. The developed model was applied to two spaces under different operating conditions, from the initial state until reaching steady-state. These simulations helped identify the mass flow rates of the fan coil units and understand their influence on space cooling. Additionally, a parametric analysis was conducted based on the coefficient of performance, and the maximum capacity of the spaces to achieve thermal comfort was predicted. It is expected that this model, developed in this study, can be utilized as a feasible and efficient computational tool for real-world simulations in environments, aiding in proper sizing and providing insights into the optimal operation of cassette-type fan coil units.

The main observations raised through the parametric analyses are as follows:

- And when values between 0.1 m3/s and 0.9 m3/s were assigned, the coefficient of performance showed better performance in terms of the fan coil unit mass flow rate for cooling a space.
- For the maximum thermal load in two spaces, the system adopted a flow rate of 80 kg/s, which demonstrated the best performance, resulting in a COP of 0.416 (COP = 0.4).
- When the model simulated different percentages of thermal fluid (0.2%, 0.5%, and 0.7%) in the spaces, it was observed that the system exhibited better performance in terms of COP when the system flow rate was 5 kg/s.
- The optimal flow rate for the system when VC6 operates at its maximum load is 80 kg/s.

5 ACKNOWLEDGEMENTS

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REFERENCES

Chao Liang, C. Li, C. Shao, X. Li, B. 2020. Numerical analysis of the methods for reducing the energy use of air-conditioning systems in non-uniform indoor environments. Building and Environment 167. 106442. https://doi.org/10.1016/j.buildenv.2019.106442.

Dogan, A., Kayaci, N., Demir, H., Sevindir,. M. K. 2022. An experimental comparison of radiant wall and ceiling cooling system integrated with ground source heat pump and direct expansion fan coil system in a highly glazed office room. Energy and Buildings. 273. 112412. https://doi.org/10.1016/j.enbuild.2022.112412.

Engineering ToolBox, (2003). Overall Heat Transfer Coefficients. Available on: https://www.engineeringtoolbox.com/overall-heat-transfer-coefficient-d_434.html. (Data de acesso: 15/07/2021).

Gamarra, I.R L. Perez-Lombard, J. Ortiz, C. Pout, A review on buildings energy consumption information, Energy Build. 40 (2008) 394–398. [2] A.R.. Istrate, I. Herrera, C. Lago, J. Lizana, Y. Lechon, Energy and water consumption and carbon footprint of school buildings in hot climate conditions. Results from life cycle assessment, J. Clean. Prod. 195 (2018) 1326–1337.

Hernandez, F. F.,. Atienza-Marquez, A., Suarez, J. M. P., Cantalejo, J. A. B., Muriano, M. C G. 2022. Analysis of a HVAC zoning control system with an air-to-water heat pump and a ducted fan coil unit in residential buildings. Applied Thermal Engineering. 215. 118963. https://doi.org/10.1016/j.applthermaleng.2022.118963.

Jahanbin, A. 2022. Efficacy of coupling heat recovery ventilation and fan coil systems in improving the indoor air quality and thermal comfort condition. Energy and Built Environment. 3.478–495. https://doi.org/10.1016/j.enbenv.2021.05.005.

Johnson, Bill; Whitman, Bill; Tomczyk, John. Refrigeration and Air Conditioning Technology. 8th ed. Boston: Cengage Learning, 2016.

KINCAID and CHENEY, 1991). KIM, B. PARK, J. Dynamic Simulation of a Single-Effect Ammonia-Water Absorption Chiller. International Journal of Refrigeration, v. 30, p. 535-545, 2007.

L. Perez-Lombard, J. Ortiz, C. Pout, A review on buildings energy consumption information, Energy Build. 40 (2008) 394–398. A.R. Gamarra, I.R. Istrate, I. Herrera, C. Lago, J. Lizana, Y. Lechon, Energy and water consumption and carbon footprint of school buildings in hot climate conditions. Results from life cycle assessment, J. Clean. Prod. 195 (2018) 1326–1337.

Martinho, L. C. S. Vargas, J. V. C. Balmant, W. Ordonez, J. C. A Single Stage Absorption Refrigeration System Dynamic Mathematical Modeling, Adjustment and Experimental Validation. International Journal of Refrigeration, v. 68, p. 130-144, 2016.

Mazzei, M. S., Mussati, M. C., Mussati. S. F. 2014. NLP model-based optimal design of LiBreH2O absorption refrigeration systems. international journal of refrigeration 38, 58 e 7 0. http://dx.doi.org/10.1016/j.ijrefrig.2013.10.012.

Ranade, A., Provan, G., Mady, A. E-D., O'Sullivan, D. 2020A computationally efficient method for fault diagnosis of fan-coil unit terminals in building Heating Ventilation and Air Conditioning systems. Journal of Building Engineering. 27. 100955. https://doi.org/10.1016/j.jobe.2019.100955.

Whitman, Bill; Johnson, Bill; Tomczyk, John. Refrigeration and Air Conditioning Technology. 8th ed. Boston: Cengage Learning, 2016.

Yang Wanga, Y. Kuckelkorna, J. Zhaoc, F. Spliethoffa, H. Langd, W. 2017. A state of art of review on interactions between energy performance and indoor environment quality in Passive House buildings. Renewable and Sustainable Energy Reviews 72. 1303–1319. http://dx.doi.org/10.1016/j.rser.2016.10.039.

Zhu, X., Wang, H., Han, X., Zheng, C., Liu, J., Cheng, Y. 2023. Experimental study on the operating characteristic of a combined radiant floor and fan coil heating system: A case study in a cold climate zone. Energy and Buildings. 291, 113087. https://doi.org/10.1016/j.enbuild.2023.113087.

Zhang. L., Liu. X., Jiang, Y. 2013. Application of entransy in the analysis of HVAC systems in buildings. Energy 53, 332 and 342. http://dx.doi.org/10.1016/j.energy.2013.02.015.

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