Development of a Detailed Model of Hybrid System Composed by Air-to-Water Heat Pump and Boiler

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Abstract

Air-to-water heat pumps are one of the most promising and increasingly widespread solutions, despite some intrinsic drawbacks, such as their poor efficiency at low ambient temperatures and at high sink temperatures, e.g., in domestic hot water production. In this context, hybrid heat pump systems combining air-to-water heat pumps and boilers (HSs) have been proposed on the market, especially for the renovation of existing buildings, where high supply water temperatures are typically required. Even though HSs are off-the-shelf technology, the topic has recently gained interest in research. HSs consist of two generators, which must be designed with an integrated approach from the start. However, the performance improvement hinges on the availability of a detailed model able to accurately predict the HS performance. Most of the studies available in the literature use models based on performance maps that are not suitable for HS design. This study presents a new detailed model of a hybrid system, developed in the MATLAB environment. The model adopts a quasi-physical representation of the heat pump cycle and condensing boiler. The boiler model thermodynamically simulates the combustion process, using the Cantera solver and the Gri-Mech properties. The heat pump model simulates the thermodynamic cycle, using refrigerant properties obtained from CoolProp libraries. A detailed model for each main component of the system is developed. Component models are combined, thus allowing the user to consider the influence of single components or construction parameters on the overall HS performance. Individual component models were validated against software or performance data provided by manufacturers. The validation proved that the models of the single components can reproduce performance with high accuracy. Therefore, the model can be used for future studies involving HS design, to analyze the influence of construction choices on overall system efficiency.

1. Introduction

Hybrid systems (HSs) can be a promising solution for increasing the efficiency of heating systems, particularly for existing buildings that do not have as high levels of insulation as new buildings (Roccatello et al., 2022).

The way the two generators (boiler and heat pump) are combined is crucial for system efficiency. Therefore, the HS, consisting of two generators, must be designed with an integrated approach from the start. A detailed model of the system would be needed to study and develop a HS. This would allow the design of the individual components of the system to be optimized for their combination, to maximize the efficiency of the hybrid system under the chosen operating conditions.

Previous studies on the topic developed a HS model and used it to compare system performance with other solutions, e.g., monovalent systems with heat pump or boiler. Klein et al. (2014) applied a model of HS for simulating a building with different insulation levels, using TRNSYS software. They used a model of the hybrid system based on performance maps. Di Perna et al. (2015) adopted an experimentally derived HS model, for the comparison of HS performance with that of boiler or electric heaters. Bagarella et al. (2016) conducted simulations using TRNSYS to discuss the distribution between heat pump and boiler operation based on outdoor temperature. The HP model is based on the combination of performance map data for each component, while the evaporator was modeled using finned coil evaporator design software, which allows for estimation of the frost formation process on the evaporator. The boiler model ap-

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plies numerical correlations for determination of the efficiency. Dongellini et al. (2021) analyzed the primary energy consumption due to the HS with different heat pump sizes. The model of the HS is based on performance data provided by manufacturers. Li & Du (2018) compared different hybrid system configurations by simulating system performance under certain operating conditions. They adopted a model based on performance maps for the heat pump, and an average efficiency for the boiler. Park et al. (2014) developed a detailed HP model, based on individual models of system components, and a boiler model based on experimental correlations.

The literature review shows that most of the hybrid system models are based on performance maps. These models, as already discussed, are well suited for application to building simulations, as they provide the real-life behavior of components already available on the market. However, they are not suitable to be used for a new design of the system itself, but rather to analyze the performance of a given system over a certain period of time.

The goal of the present work is to develop a semi-physical model of a HS. The model is primarily based on the physical laws describing the processes occurring in the individual components of the system.

The modeling involved the main components of the heat generators. The models of the individual components were combined into the overall model of the single generator. The HP and boiler models can be retrieved as subroutines from the overall HS model, which contains the hybrid system logic.

The heat pump and boiler model development are based on the currently most adopted systems in HSs. The heat pump considered for modeling is a modulating heat pump, equipped with a plate heat exchanger (HX) for the condenser, and a finned-coil HX for the evaporator. The boiler considered is a condensing, modulating, natural gas-fired boiler.

2. Methodology

The HS model consists of the logic that manages the interaction with the heating system – i.e., it determines whether a generator should operate and calculates the system setpoint – and decides which generator to operate based on the chosen control strategy. For a more detailed description of the logic of the HS model, the reader can refer to Roccatello et al. (2022). This paper presents a different type of HS model, in which the subroutines containing the heat pump and boiler models are not based on performance maps, but on semi-physical models of components. The generator models are developed as the union of the system components. The heat pump model is the combination of the component models of the air-refrigerant and water-refrigerant heat exchangers (evaporator and condenser), compressor, and expansion valve. The boiler is modeled as the union of the combustion chamber and the flue gas-water HX. Figs. 1 and 2 show the schematic of the heat pump and boiler models, respectively.

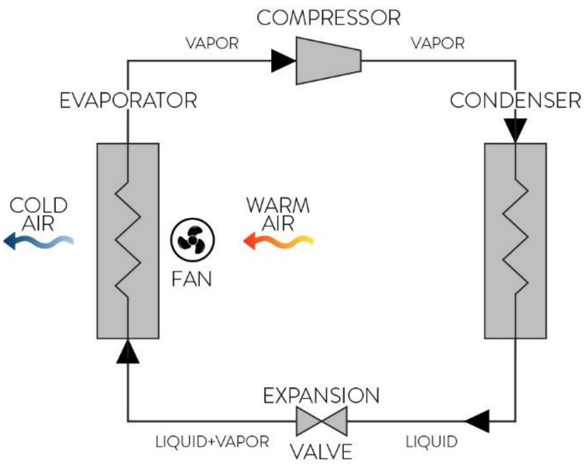

Fig. 1 – Schematic of heat pump model

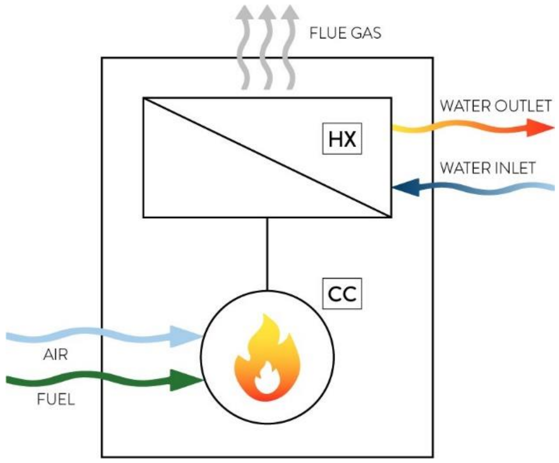

Fig. 2 – Schematic of boiler model

The following sections describe how the component models were developed, and subsequently combined, to generate the boiler and heat pump models.

2.1 Heat Pump Model

2.1.1 Condenser model

In this section, the heat transfer between refrigerant and water is modeled. The refrigerant exiting the compressor exchanges heat with water and undergoes condensation and subcooling. An example of a heat pump refrigerant cycle on a pressure/enthalpy diagram is shown in Fig. 3, in which the condensation process is highlighted.

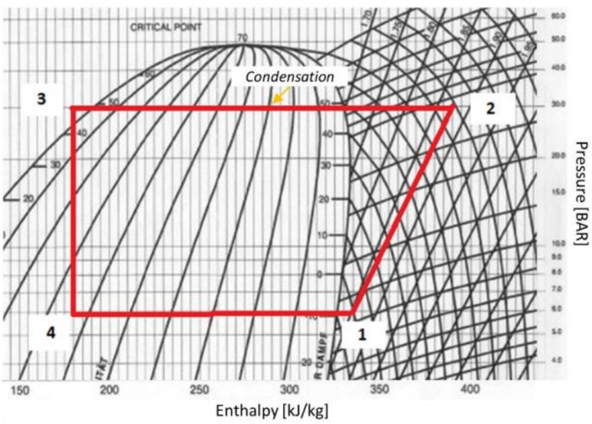

Fig. 3 – Condensation process represented on a pressure/enthalpy diagram

The condenser model refers to a plate HX, which is the most used type of exchanger for residential size heat pump systems.

Initially, a first guess of the value of the condensation temperature is estimated. The heat of condensation (Q_{cond}) can be calculated using the following equation, which considers the enthalpy state of the refrigerant entering and leaving the condenser.

$$Q_{cond} = m_{ref}(h_2 - h_3) \quad (1)$$

Subsequently, heat transfer correlations are implemented for determination of the global heat transfer coefficient (Bergman et al., 2017). The equation for the calculation of the global heat transfer coefficient of the condenser (U_c) is reported here below:

$$U_c = \frac{1}{\frac{1}{h_c}+\frac{1}{h_w}} \quad (2)$$

in which h_c is the heat transfer coefficient of refrigerant, and h_w is the water heat transfer coefficient in the condenser. The calculation of h_w is performed according to the following equation:

$$h_w = \frac{k_{wr} Nu_{wc}}{D_c} \quad (3)$$

D_c represents the hydraulic diameter, while Nu_{wc} is calculated according to the following equation:

$$Nu_{wc} = C_{wc}(Re_{wc})^{wn}(Pr_{wc})^{\frac{1}{3}} \quad (4)$$

in which:

- Re_{wc} and Pr_{wc}: Reynolds and Prandtl number
- μ_{wc}: viscosity of the water

The values of C_{wc} and wn are evaluated as follows:

$$C_{wc} = \begin{cases} 0.718 & Re_{wc} \leq 10 \\ 0.348 & Re_{wc} > 10 \end{cases} \quad (5)$$

$$wn = \begin{cases} 0.349 & Re_{wc} \leq 10 \\ 0.663 & Re_{wc} > 10 \end{cases} \quad (6)$$

The refrigerant heat transfer coefficient (h_c) is evaluated in a different way when the refrigerant is the vapor phase and when it is in the condensation phase. In the first case it is calculated as:

$$h_{sc} = \frac{k_{sc} Nu_{sc}}{D_c} \quad (7)$$

in which k_{sc} represents the thermal conductivity of refrigerant.

In the condensation process, the heat transfer coefficient of the refrigerant (h_{tc}) is calculated as follows:

$$h_{tc} = \frac{k_{tc} Nu_{tc}}{D_c} \quad (8)$$

in which k_{tc} is the thermal conductivity of the refrigerant and Nu_{tc} is calculated according to the following equation:

$$Nu_{tc} = 0.0125\left(Re_{tc}\sqrt{\frac{\rho_{cl}}{\rho_{cv}}}\right)^{0.9}\left(\frac{x_c}{1-x_c}\right)^{0.1x_c+0.8}Pr_{cl}^{0.63} \quad (9)$$

- Re_{tc} is the Reynolds number of the refrigerant during condensation
- ρ_{cl} and ρ_{cv} are the refrigerant density of liquid and vapor
- x_c is the vapor quality
- Pr_{cl} is the Prandtl number of the refrigerant in the liquid-phase.

Finally, the calculation of the heat exchanged between the fluids in the condenser (Q_{cond}) is expressed by the equation:

$$Q_{cond} = U_c A_c \Delta T_c \quad (10)$$

where ΔT_c is the mean temperature difference between water and refrigerant, U_c the global heat transfer coefficient and A_c the condenser heat transfer area. The equation allows the adjustment of the value previously assumed for the condensation temperature.

2.1.2 Expansion valve

The process that the fluid undergoes after exiting the condenser is modeled as an isenthalpic expansion (Fig. 4). Hence, the following equation is obtained, which provides the input enthalpy conditions for the refrigerant evaporation phase.

$$h_3 = h_4 \quad (11)$$

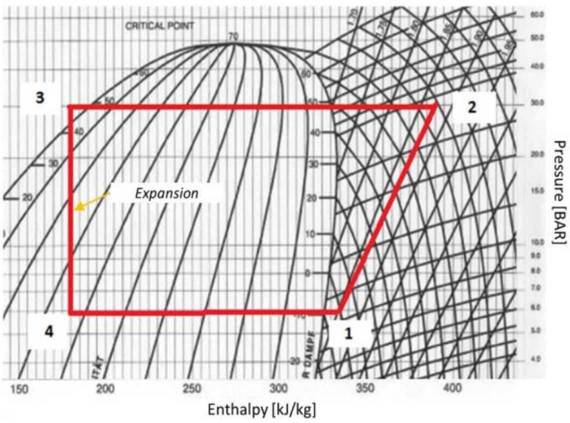

Fig. 4 – Expansion process represented on a pressure/enthalpy diagram

2.1.3 Evaporator

This section describes the modeling of heat transfer between air and refrigerant. The refrigerant process through the evaporator is shown in Fig. 5, on the pressure/enthalpy diagram. In the heat exchanger, the refrigerant undergoes an evaporation and a superheating. The type of HX considered is a finned-coil HX, which is widely used in air source heat pumps.

Initially, a first-guess value of evaporation temperature is assumed, which allows estimation of the heat exchanged during the evaporation process (Q_{ev}), hence the enthalpy difference between the outlet state and the inlet state at the evaporator. Referring to the diagram in Fig. 5, this can be expressed by the following equation:

$$Q_{ev} = m_{ref}(h_1 - h_4) \quad (12)$$

where m_{ref} represents the refrigerant mass flow rate.

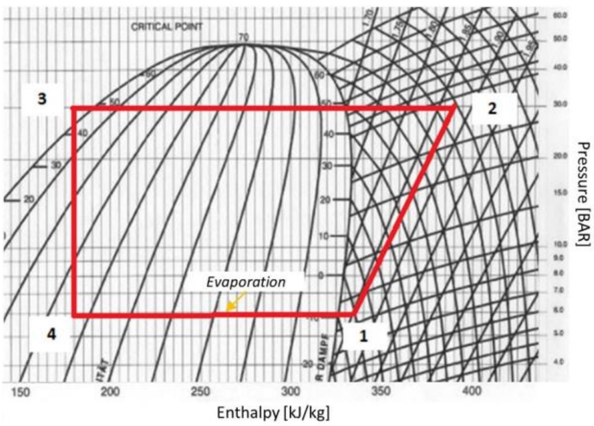

Fig. 5 – Evaporation process represented on a pressure/enthalpy diagram

Subsequently, the heat transfer correlations are implemented for the determination of the global heat transfer coefficient (Bergman et al., 2017).

h_a represents the air-side heat transfer coefficient:

$$h_a = \frac{j_a \rho_a u_m c_a}{Pr_a^{2/3}} \quad (13)$$

in which:

- is the density of the air
- is the specific heat of the air
- j_a is the heat transmission factor
- u_m is the maximum wind speed
- Pr_a is the Prandtl number of the air

j_a is calculated according to the following equation:

$$j_a = 0.0014 + 0.2618 Re_a^{-0.4}\left(\frac{A_{af}}{A_a}\right)^{-0.15} \quad (14)$$

in which:

- Re_a is the Reynolds number

- A_{af}/A_a is the ratio between the surface area of the tubes with and without fins.

The maximum wind speed (u_m) is calculated by Eqn. (15):

$$u_m = u_f \frac{s_h s_v}{(s_h - D_{te})(d_1 - d_2)} \quad (15)$$

where:

- u_f is the fan wind speed
- s_h is the tube spacings in the horizontal direction
- s_v is the tube spacings in the vertical direction
- D_{te} is the diameter of the tubes
- d_1 is the are the thickness of the fins
- d_2 is the spacing of the fins.

After that, the refrigerant heat transfer coefficient (h_e) is calculated. The value of the refrigerant heat transfer coefficient differs if the refrigerant is in the evaporation or in the superheating phase.

The refrigerant heat transfer coefficient in the superheating phase (h_{se}) is calculated as follows:

$$h_{se} = \frac{k_{se} Nu_{se}}{D_{te}} \quad (16)$$

in which k_{se} is the refrigerant thermal conductivity. 〖Nu〗 _se is calculated as:

$$Nu_{se} = \frac{(f_{se}/8) Re_{se} Pr_{se}}{1.07 + 1.27 \left(\frac{f_{se}}{8}\right)^{0.5} (Pr_{se}^{\frac{2}{3}} - 1)} \quad (17)$$

in which Re_{se} and Pr_{se} are the refrigerant Reynolds and Prandtl number, and f_{se} is the friction coefficient, calculated according to the equation:

$$f_{se} = (1.82 ln Re_{se} - 1.64)^{-2} \quad (18)$$

The heat transfer coefficient of the refrigerant in the condensation phase is calculated according to the following equation:

$$h_{te} = h_{el} \left\{ \left[(1 - x_e) + 1.2 x_e^{0.4} (1 - x_e)^{0.01} \left(\frac{\rho_{el}}{\rho_{ev}}\right)^{0.37} \right]^{-2.2} + \left[\frac{h_{ev}}{h_{el}} x_e^{0.01} \left(1 + 8 (1 - x_e)^{0.7} \left(\frac{\rho_{el}}{\rho_{ev}}\right)^{0.67}\right) \right]^{-2} \right\}^{-0.5} \quad (19)$$

where:

- h_{el} is the heat transfer coefficient of the refrigerant liquid-phase
- h_{ev} is the heat transfer coefficient of the refrigerant vapor-phase
- ρ_{el} is the density of the refrigerant liquid-phase
- ρ_{ev} is the density of the refrigerant vapor-phase
- x_e is the vapor quality of the refrigerant.

Thus, the calculation of the evaporator global heat transfer coefficient (U_e) is performed according to the following equation:

$$U_e = \frac{1}{\frac{1}{h_e} + \frac{1}{h_a}} \quad (20)$$

Finally, the calculation of the heat exchanged between the fluids in the evaporator (Q_e) is expressed by the equation:

$$Q_{ev} = U_e A_e \Delta T_e \quad (21)$$

where ΔT_e is the mean temperature difference between air and refrigerant and A_e the evaporator heat transfer area.

2.1.4 Compressor

Given the geometric and operational complexity of the component, the compressor was modeled using performance data provided by the manufacturers, to avoid great inaccuracy of the heat pump model.

The compressor model adopts polynomial correlations, which allow for the estimation of the refrigerant mass flow rate and compressor power input, as a function of suction and discharge pressure, which correspond to the evaporating and condensing pressures in the heat pump model, if neglecting pressure drops.

In addition, the polynomial correlations are a function of the compressor frequency, i.e., they allow for the modeling of a variable-speed compressor, and thus for the development of a modulating heat pump model.

The correlations used in the model are reported here below (Copeland Select Software). The variable X represents either the refrigerant mass flow rate or the compressor power input. S and D are the evaporating and condensing temperatures, respectively, expressed in °C, while C0 – C9 are the specific coefficients for the compressor provided by the manufacturer.

X = C0 + C1*S + C2*D + C3*S ^2+ C4*S*D + C5*D^2 + C6*S ^3 + C7*D*S ^2+C8*S*D^2 + C9*D^3 (22)

2.1.5 Model development

The flow chart in Fig. 6 describes the rationale of the heat pump model. At the beginning, initial values of the evaporating and condensing temperatures are guessed. Based on these values, the compressor model estimates the refrigerant mass flow rate and power input. These values are used as inputs by the condenser model, which estimates the heat exchanged in the condenser and adjusts the value of the condensing temperature. Similarly, the evaporator model allows for the adjustment of the first guess evaporating temperature value through an iterative procedure. Finally, the outputs of the model, i.e., heating capacity and power input, are released, allowing for the COP calculation.

2.2 Boiler Model

The boiler was modelled by subdividing the system into its major components, namely combustion chamber (CC) and HX.

The model of the CC is based on a thermodynamic equilibrium simulation of the combustion process carried out using the Cantera solver (Cantera). The inputs to the combustion chamber model are the fuel mass flow rate and air mass flow rate (or excess air). Through the modeling of the combustion process, the adiabatic flame temperature is calculated, i.e., the temperature that the gas mixture would ideally reach in the absence of heat loss. The model considers the heat losses of the combustion chamber (Q_{loss_cc}), based on information provided by the manufacturer. The outputs of the combustion chamber model are the temperature and mass flow rate of the flue gas.

These values are used as input for the model of the water-flue gas HX. An exchanger with unitary efficiency was considered, which is a good approximation, given the very high efficiency in recovering flue gas heat in the heat exchange process.

The value of flue gas outlet temperature determines whether condensation of water vapor in the flue gas occurs. If the flue gas outlet temperature is lower than the dew point, the condensation heat is recovered and transferred to the water. The logic adopted in the boiler model development is shown in Fig. 7.

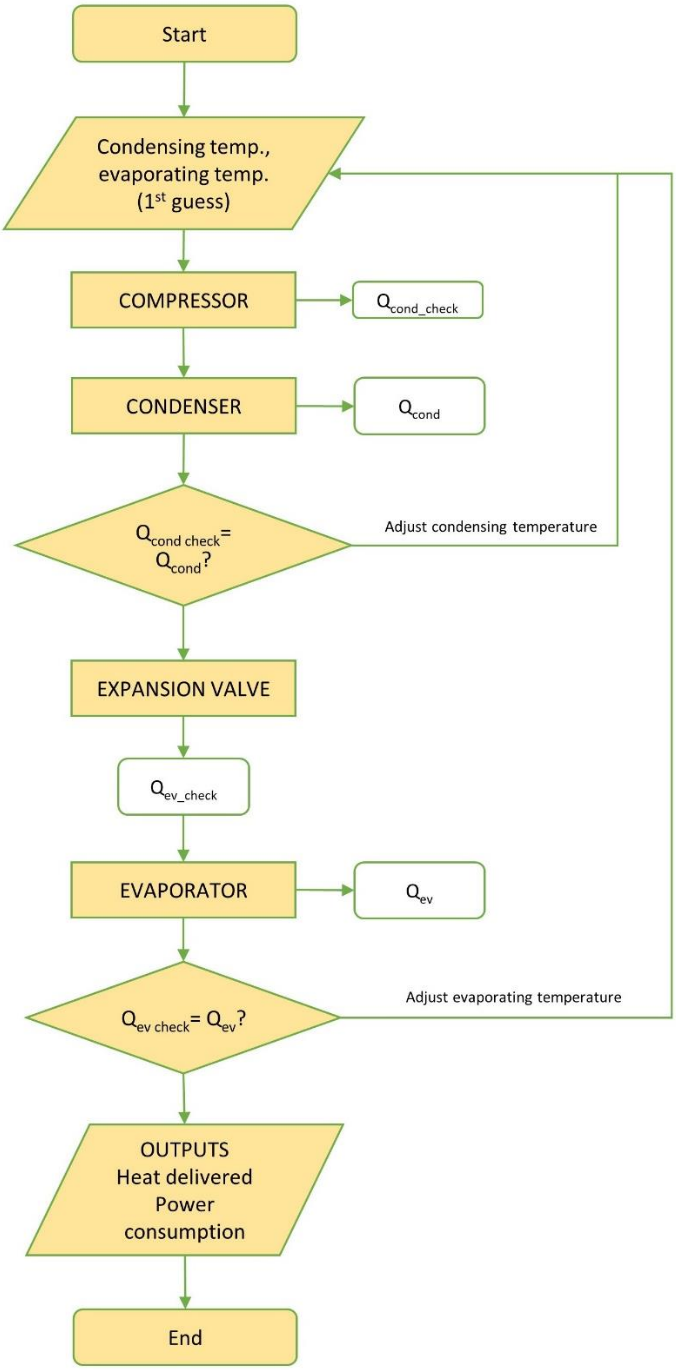

Fig. 6 – Heat pump model logic

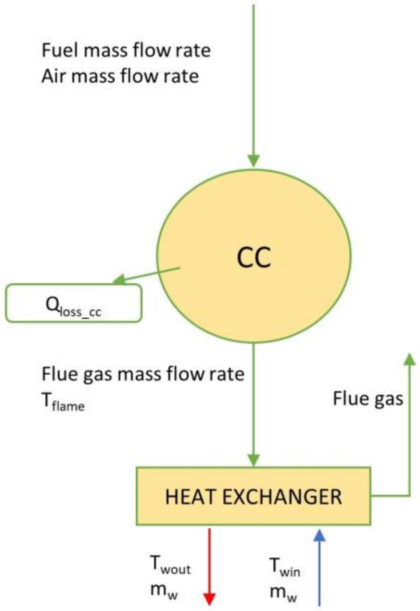

Fig. 7 – Boiler model logic

2.3 Model Validation

For the heat pump, validation of the HX models is presented, the compressor model being based on performance maps. The validation of the condenser model was carried out using the selection software provided by SWEP (SSP G8-SWEP), which provides the geometric parameters of the heat exchangers and allows the exchanged capacity to be evaluated by setting the operating conditions as input to the software.

The validation of the evaporator model was performed using performance data from a finned-coil HX obtained from NIST software (EVAP-COND). The software receives as input the geometric parameters and operating conditions and calculates the system performance in terms of exchanged capacity.

The validation of the boiler model was carried out using the data of heating capacity provided by the manufacturer, as a function of the fuel mass flow rate and water entering and leaving temperature.

3. Results And Discussion

In this section, the results of the validation of the component models are presented.

3.1 Condenser Model Validation

For a given heat exchanger, SWEP software returns the exchanged heating load by entering the operating condition data. The HX geometry and operating conditions are given as input to the model, which determines the overall heat transfer coefficient and heat load. Tables 1 and 2 show the geometrical and operating parameters related to the validation test carried out. Table 3 shows the results related to the heat exchanged between the fluids in the condenser obtained from the manufacturer's software and estimated using the model. It can be observed that the relative error in the estimate of the heat load is less than 5 %.

Table 1 – Input geometrical parameters used for validation test of plate heat exchanger model

Geometric parameters		
Total heat transfer area	m^2	1.57
Refrigerant channel volume	dm^3	0.0313
Water channel volume	dm^3	0.0301
Number of plates	units	58
Height	mm	324
Length	mm	94
Width	mm	90.7

Table 2 – Input operating parameters used for validation test of plate heat exchanger model

Operating parameters		
Operating parameters	°C	50
Condensing temperature	°C	56.2
Subcooling	°C	2
Condenser inlet temperature	°C	80
Refrigerant mass flow rate	kg/s	0.059
Water mass flow rate	kg/s	0.458

Table 3 – Heat load values provided by manufacturer and calculated by the model, and relative error, for validation test on plate heat exchanger model

Validation results		
Manufacturer's heat load	kW	9.6
Model heat load	kW	10.0
Relative error	%	4.7

3.2 Evaporator Model Validation

The model validation of the evaporator was performed, using EVAP-COND software, for a given finned-coil HX geometry. The heat exchanged between the two fluids calculated by the software was compared with the exchanged heat estimated by the model. The data for the finned-coil HX geometry considered for the validation are shown in Table 4 and the operating parameters used in the validation are reported in Table 5.

Table 4 – Evaporator geometric parameters

Geometric parameters		
Number of tubes	units	16
Number of rows	units	3
Tube length	mm	454
Inner diameter	mm	9.22
Outer diameter	mm	10.01
Tube pitch	mm	25.40
Depth row pitch	mm	22.23
Front area	mm	0.188
Heat transfer area	mm	3.8
Fin data		
Thickness	mm	0.2032
Pitch	mm	2.004

Table 5 – Evaporator operating parameters

Operating parameters		
Volumetric air flow rate	m^3/min	30
Evaporating temperature	°C	0
Superheating	°C	4.1
Air inlet temperature	°C	15
Inlet vapor quality	-	0.2
refrigerant mass flow rate	kg/h	0.0345

The results of the validation are shown in Table 6. The test shows a relative error below 10 % for the estimation of the heat exchanged between water and refrigerant in the evaporator.

Table 6 – Results of model validation

Validation results		
EVAP-COND heat load	kW	6.0
Model heat load	kW	6.5
Relative error	%	7.0

3.3 Boiler Model Validation

For the validation of the boiler model, the performance data of a commercial model of boiler manufactured by the Immergas S.p.A company were considered, for which certified performance data are available. These data show the useful heating capacity produced by the boiler as a function of fuel mass flow rate and water inlet and outlet temperatures. The manufacturer also indicates the value of excess air used by the boiler.

The model receives the fuel mass flow rate, excess air, return temperature, and water flow rate as inputs, and estimates the useful power delivered to the water. The validation was performed considering the conditions of inlet water at 30 °C and outlet water at 50 °C. In this case, water vapor condensation in the flue gas occurs. Validation was performed for fuel mass flow rates varying from minimum to maximum. The operating conditions used for validation are summarized in Table 7. The graph in Fig. 8 shows the results of the validation. It can be observed that the maximum relative error on the estimate of the useful boiler heating capacity does not exceed 10 % for each value of fuel mass flow rate.

Table 7 – Operating conditions for boiler model validation

Operating conditions		
Inlet water temperature	°C	30
Outlet water temperature	°C	50
Fuel mass flow rate (max-min)	kg/h	3.69–0.43

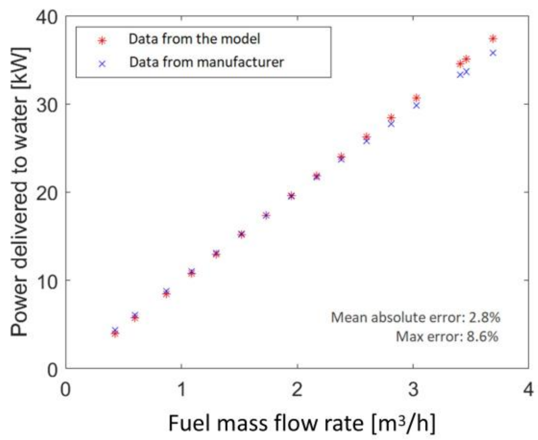

Fig. 8 – Validation results for the boiler model

3.4 Discussion

Regarding the validation of the heat pump components, the results presented show that the component models developed according to physical laws give results which are in line with the performance values declared by the manufacturers. Similarly, the validation of the boiler model showed how the physical model replicates with good accuracy the performance data provided by the manufacturer.
The heat pump and boiler models described can be used as subroutines of an overall HS model. Thus, this new model can be used to evaluate the influence that the choice of certain system components, or certain construction parameters, has on the overall efficiency of the hybrid system.

4. Conclusions

This paper presents the development of a new quasi-physical model of a hybrid system, based on the combination of the models of the individual components of the system. These models are based on physical laws, except for the compressor model, because of the high geometrical complexity of the component and the risk of introducing large errors by approximating its behavior with an analytical model.
For the heat pump, the evaporator and condenser model were developed using heat transfer correlations, which model the heat exchange between air and refrigerant, and water and refrigerant. The compressor model, on the other hand, is based on performance data provided by the manufacturers. The process occurring in the expansion valve is modeled as an isenthalpic expansion.
The boiler model is divided into a model of the combustion chamber and a model of the heat exchanger between flue gas and water. The combustion chamber model is based on the thermodynamic equilibrium simulation of the combustion process carried out using the Cantera solver.
After that, the validation of the models of the system components was presented, which provided results with acceptable accuracy to qualitatively estimate the behavior of a hybrid system.
Therefore, the model can be used for the detailed study of a hybrid system, and especially in the design process of the system itself. It will be possible to analyze the influence of the choice of certain components or construction parameters on the overall efficiency of the hybrid system.

Acknowledgement

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Nomenclature

CC	Combustion chamber
HS	Hybrid system
HX	Heat exchanger
Q_{loss_cc}	Heat losses in the combustion chamber
Q_{cond}	Heat exchanged in the condenser
Q_{ev}	Heat exchanged in the evaporator
m_{ref}	Refrigerant mass flow rate
m_w	Water mass flow rate
T_{win}	Inlet water temperature
T_{wout}	Outlet water temperature
T_{flame}	Temperature of flue gas exiting the combustion chamber

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