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Thermodynamic transient simulation of a combined heat & power system

Davide Borelli[a], Francesco Devia[a], Corrado Schenone[a], Alessandro Spoladore[a]*

[a]*Department of Mechanical Enginneering(DIME), University of Genoa, Genoa 16145, Italy*

Abstract

In this paper a numerical model aimed at studying dynamic behavior of CHP (Combined Heat and Power) plants is presented, paying particular attention to the components in which heat transfers take place. The analysis refers to a system powered by an internal combustion engine for a compression ignition type in cogeneration configuration, equipped with two heat extractors: the first one for coolant / water, the second one for exhaust gas / water. The numerical model has been implemented by using Matlab-Simulink software. After a description of the simplifying assumptions adopted for implementing the simulator, the model is exposed in detail with regards to each single element. Then simulation results are reported for two different operating conditions aiming to assess the effectiveness of the model in analyzing the dynamic behavior of CHP plants.

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Keywords: Dynamic analisys; CHP; Cogenerator; Matlab; Simulink

1. Introduction

The CHP energy generation and, in particular, the cogeneration technique in small scale, is a technology in strong development in recent decades [1, 2]. CHP plants allow to profit from two useful effects coming from the same thermal machine: mechanical work and heat, the latter of which in the ordinary configuration would be wasted.

* Corresponding author. Tel.: +390103532863.
E-mail address: alessandro.spoladore@edu.unige.it

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The current concept of SMART GRID [3], which involves the implementation of both a thermal and electrical network with production and utilization systems located in a capillary way in the network, foresees a large use of small size CHP.

This operating mode, in small and medium scale, it is often practiced by placing heat recovery units in the two main kind of standard thermal machines such as micro gas turbines (mGT) and internal combustion engines (ICE).

The choice of the type of machine, for various reasons, is mostly directed toward the ICE [4]. These machines, when compared to the mGT, ensure:

- a smaller "scale effect", which is characteristic of all thermal engines but has less effect on the ICE, which results in greater efficiency even for small installed powers;
- ease of maintenance, as it is easier to find skilled operators for the ICEs;
- lower cost of the machine with a good effect on the payback period.

The commercial CHP-ICE size range from 35 kW_e to 10 MW_e and this kind of plants can obtain a total efficiency, inclusive of the two useful effects, up to 98%.

These machines are usually derived from diesel fuelled compression-ignition engines, but they have been recently turning into feeding by natural gas or hybrid feeding by gas and oil. The heavy duty machines for static applications are optimized to operate at constant speed of rotation: the air flow is nearly constant and the load is controlled by varying the amount of injected fuel.

Fig.1 highlights the average breakdown of primary energy into ICE. The mechanical efficiency can reach up to 30-35% while the remaining part is transferred by radiation and convection (8-10%), by the cooling system (30-35%) and by the exhaust gases (28-33%).

The software used for implementing the numerical model is Matlab-Simulink [6]. It has been chosen because its primary interface is a graphical block diagramming tool and a customizable set of block libraries, which offers tight integration with the rest of the MATLAB environment.

With this tool, it is very easy to define a set of detailed characteristic curves for each components to evaluate the behavior of the whole plant in various conditions.

2. Description of the CHP model

The simplest CHP plant (Fig.2), in which an alternative engine is employed, is composed of three main components: the ICE, a water / refrigerant heat exchanger (HEX1) and a water / exhaust gas heat exchanger (HEX2).

The component ICE will provide the mechanical work, transformed into electrical energy from the generator, and the required heat flow to the thermal users.

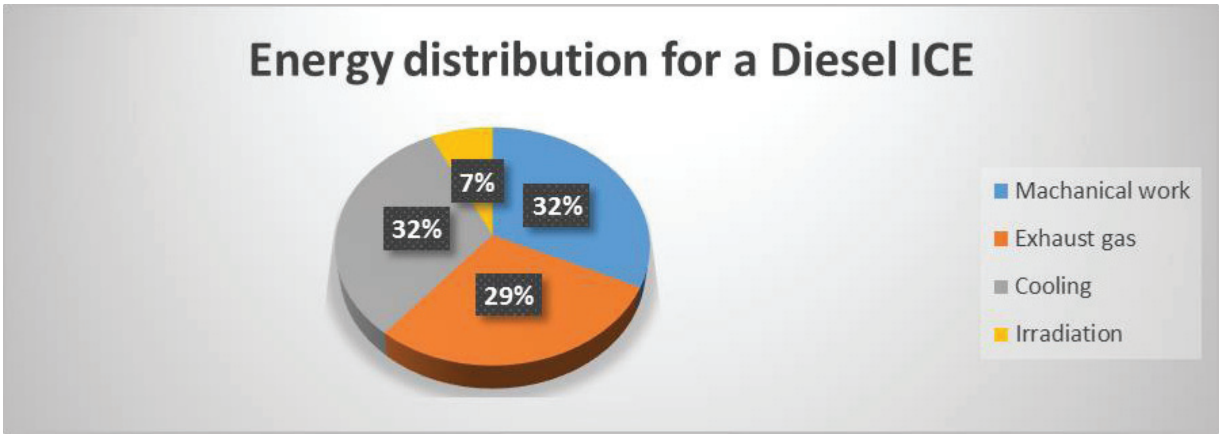

Fig.1. Mean Energy distribution for a Diesel engine (adapted from [5]).

The heat exchanger, HEX1, picks up the heat flow from the refrigerant transferring it to the water supply of the thermal network. The refrigerant is driven by a variable volume flow pump, which is controlled to maintain a set point temperature of the mass of coolant. In this model, HEX1 is representative of all the fluids which cool the engine.

The HEX2 picks the heat from the flow of exhaust gases which have very high temperature, which therefore makes it very advantageous to recover their thermal energy.

The water circuit, used as an energy vector to distribute the heat to the users, is equipped with a controlled variable volumetric flow pump to maintain supply temperature at a constant value.

The physical-mathematical model used for the dynamic analysis of the system is described below for each main component.

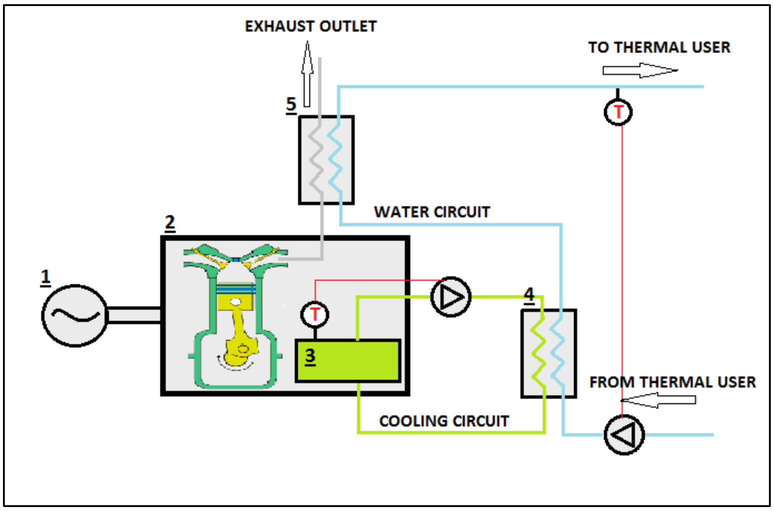

Fig.2. CHP plant scheme: 1) Electric generator, 2) ICE, 3) Coolant inertia, 4)HEX1 , 5)HEX2.

2.1 ICE emulator block

A detailed formulation for the ICE could involve the analysis of the real thermodynamic cycle and would require the modelling of the engine and the real combustion process [7].

This type of analysis would be highly computationally time-consuming as it would consider physical phenomena with a characteristic time scale definitely lower than the ones of the other components of the plant.

Commonly ICE, with similar technological features, has similar trends, in function of the load, of the characteristic parameters, like the mechanical and thermal power produced and the temperature of the exhaust gases.

In this model, the behavior of the ICE has been described using characteristic curve based on the percentage load. This approach allows to easily scale these curves to obtain the behavior of the different machines.

The "*ICE emulator*" requires as input the following parameters:

- required heat flow;
- mass and fluid proprieties of the refrigerant;
- removed heat flow by HEX1;

and it provides as output:

- refrigerant temperature;
- temperature and flow rate of the exhaust gas;
- generated electric power;
- percentage load of the machine.

The block ICE is made by three sub-blocks, each of which models a different physical phenomenon.

The block "*characteristic curves*" implements the performance curve *Heat to Power* (Fig.3), particularly used in CHP systems, which describes the value of the heat flow and electric power generated for each load value of the machine.

In this case the CHP is controlled to obtain a desired heat flow. The function provides that this value is given as input and the model calculate, by interpolating the curve, the machine load and the electric power generated.

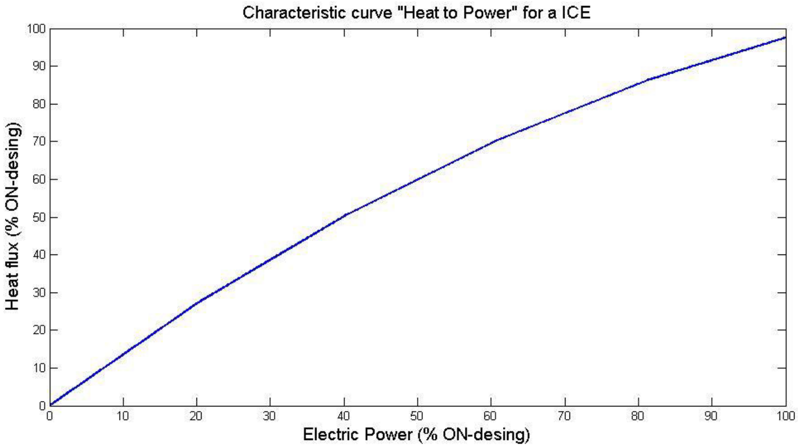

Fig.3. Characteristic curve “Heat to Power” for a Diesel ICE.

The block "*coolant thermal capacity*" calculate the temperature of the refrigerant. The quota of heat transferred to the coolant is calculated from the total heat flow, according to the relationships previously described (Fig.1) and it be set in an half of the total heat release by the ICE.

The energy balance applied to the mass of the coolant gives the temperature of the fluid (Eq.1).

$$m_{coolant} Cv_{coolant} \frac{dT_{coolant}}{dt} = Q_{in} - Q_{HEX1} \tag{1}$$

where Q_{in}represents the portion of the heat flux transferred to the refrigerant, Q_{HEX1} is instead the heat flow rate removed by the exchanger and the product $m_{coolant}$*$Cv_{coolant}$ is the thermal capacity of the coolant [kJ/K].

The block "*Exhaust gas*" calculate the temperature and the flow rate of the exhaust.

The one modelled is a compression ignition engine that works at constant rotational speed. The flow rate of the burnt gases is therefore less variable with the engine load and it is assumed constant. The outlet temperature of the exhaust is assumed to be linearly dependent by the load (Fig.4).

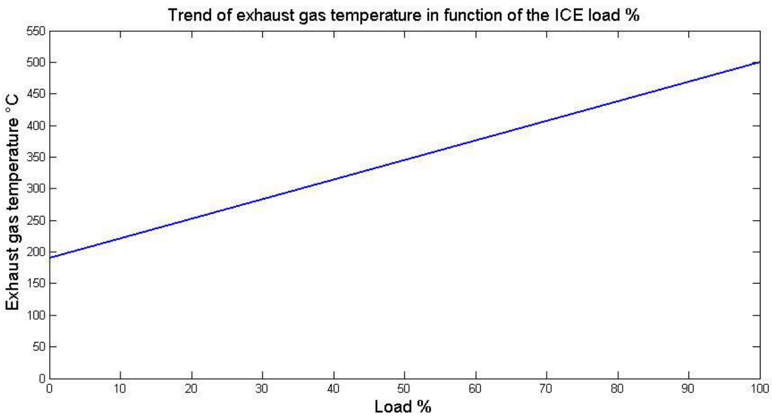

Fig.4. Exhaust gas temperature versus ICE load.

2.2 Heat exchangers

The model of the CHP analyzed involves the use of two different heat exchangers operating with the two fluids considered: coolant and exhaust gas.

Both components operate with water as secondary fluid, which constitutes the energy vector for the transport of thermal energy from the CHP to the thermal users.

A classical counterflow shell & tube heat exchangers is used. The heat exchanger needs, as input, the following geometrical parameters:

- diameter, length, thickness and number of internal pipes;
- diameter of the shell;

and the following fluid-dynamic parameters :

- thermal properties of the fluids as a function of the temperatures;
- thermal conductivity of the pipes;

and the following variables of the plant for both flow:

- mass flow rate;
- inlet temperature.

The model of HEX is composed of two main blocks, the first contains the specific equations of the counterflow exchanger and the second calculates the convective heat transfer coefficients.

The block "*characteristic curves*" calculates the outlet temperature of the fluids using the *ε-NTU* method [8]. The efficiency ε is calculated as a function of the *NTU*, Number of Transfer Units, and the ratio of the thermal capacity *R* (Eq.2).

$$\varepsilon = \frac{Q_{real}}{Q_{max}} = f\left(\frac{U\,A_{HEX}}{E_{min}}, \frac{E_{min}}{E_{max}}\right) = f(NTU, R) \tag{2}$$

Where E_{min} and E_{max} are respectively the minimum and maximum thermal capacity between that of the two streams and it is calculated from the product of the flow rate per the specific heat, and A_{HEX} is the heat exchange surface.

U is the total transmittance comprehensive of the two convective terms due to the streams, a conductive term due to the thickness of the metal and a supplemental Fouling resistance R_{FO} (Eq.3).

$$U = \left(\left(\frac{1}{h_{f1}} + \frac{1}{h_{f2}} + \frac{Th_{metal}}{k_{metal}} + R_{Fo}\right)^{-1}\right) \tag{3}$$

The value of ε is therefore expressible in the function of these two dimensionless numbers using the Eq.4, the graphical result is plotted in Fig.5.

$$\varepsilon = \frac{1 - e^{-NTU(1-R)}}{1 - R * e^{-NTU(1-R)}} \tag{4}$$

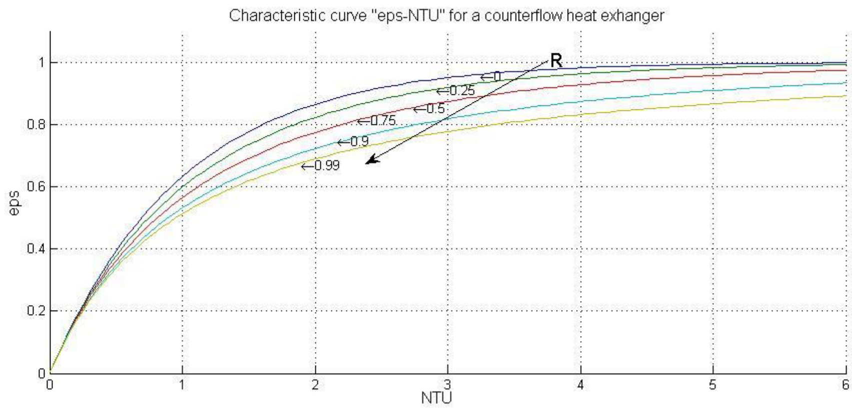

Fig.5. Graphical results of the Eq.4.

From ε it is possible to compute the outlet temperatures by Eq.5 and 6 or by 5bis and 6bis.

If $E_{f1}=E_{min}$

$$T_{f1,out} = T_{f1,in} - \varepsilon\ (T_{f1,in} - T_{f2,in}) \tag{5}$$

$$T_{f2,out} = T_{f2,in} + R\ (T_{f1,in} - T_{f1,out}) \tag{6}$$

else if $E_{f2}=E_{min}$

$$T_{f2,out} = T_{f2,in} + \varepsilon\ (T_{f1,in} - T_{f2,in}) \tag{5bis}$$

$$T_{f1,out} = T_{f1,in} - R\ (T_{f2,out} - T_{f1,in}) \tag{6bis}$$

The block "*convective coefficient calculation*" evaluate the Nusselt number (Nu), the Reynolds number (Re), and the Prandtl number (Pr). The formulation of dimensionless numbers can be seen in equations 7 to 9.

$$Nu = \frac{h\ D_{eq}}{k_f} \tag{7}$$

$$Re = \frac{v\ D_{eq}}{\mu_f(T_{avg})} \tag{8}$$

$$Pr = \frac{\mu_f(T_{avg})\ cp_f}{k_f} \tag{9}$$

It has been used a correlation valid for fully developed flows and cylindrical geometries (Eq.10), the coefficients *A*, *m* and *n* are derived as a function of the flow regime in force in the ducts.

$$Nu = A * Re_D^m Pr^n \tag{10}$$

3. Analysis of the results

The numerical model developed by means of the criteria described above was used to simulate the operation. Two different relevant tests to analyze the behavior of the CHP were selected.

TEST A corresponds to a very slow transient and it is aimed at evaluating the CHP behavior as a sequences as a quasi-steady state conditions; TEST B represents a common operating condition which could be representative of a morning start-up followed by a critical shut-down.

3.1 Test A results

The "TEST A" provides an analysis of the behavior of a CHP-ICE, of 500kW$_e$, in all operating points, calculated minimizing the transient effects. To do this, the model has been tested by varying the electrical load, with a very slow increase, in a linear way by starting from the minimum technical conditions (20% on-design load) to the design conditions (Fig.6).

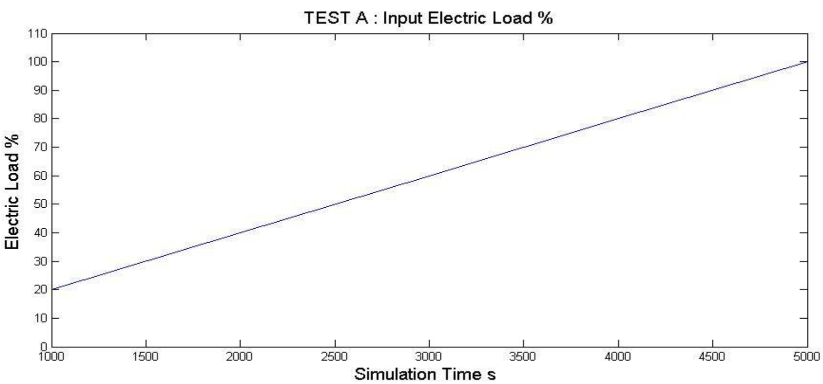

Fig.6. Electric power demand in TEST A.

The first campaign of simulations involved the analysis of the behavior in the off-design of a CHP-ICE machine capable of delivering 500 kW_e. The basic parameters, snatched from the technical details, for the design operating point, were as follows:

- recovered heat flux recovered 475 kW_t;
- produced electrical power 500 kW_e;
- exhaust mass flow rate;
- ratio between heat power recovered by coolant and by exhaust gas.

The heat exchangers are sized to work properly at design condition. The temperature of the exhaust gases, produced by the ICE, are only slightly affected by size of the machine: for this reason a typical trend for it was chosen (Fig.4).

The boundary conditions and the control system have been set in order to simulate the connection to a district heating network (DH). It was assumed that the return flow temperature from the DH system, not modelled in this paper, is at a 60°C. In this condition the system acts on water mass flow rate to maintain the output temperature to the DH at a constant level of 90°C.

The second control unit operates to maintain the temperature of the storage at 90°C (water / glycol composed) by varying the flow rate of fluid as a function of the different operating conditions. In Fig.7 the mass flow rate of water and coolant are shown.

It is observed that the water flow rate grows proportionally with the required load. The refrigerant flow rate is proportional to the load but it also presents different gradient between the phase of low and high operativity: this difference can be attributed to a change in the flow regime and the consequent abrupt of the convective coefficient.

Fig. 8 highlights the trend of temperatures in the coolant / water heat exchanger (HEX1); given the characteristic of the control system it is observed that the two inlet temperatures are almost constant. In fact, the inlet temperature of the refrigerant is controlled by means of the variation of the flow for coolant and thus varies around 90°C. It is observed that the temperature gap, from the side of the coolant, decreases very to the increase of the required load this is because the control system also provides for the increase of the water flow with a consequent variation of the characteristics of the exchanger.

Fig.9 highlights the temperatures of the fluids in the exhaust gas / water heat exchanger.

In this case the thermal water is almost constant as it is controlled by the outlet temperature and the inlet temperature, equal to that for output from HEX1, varies very little with the load.

The thermal jump of the gas is instead very variable because the temperature of the combustion gases is considerably influenced by the load.

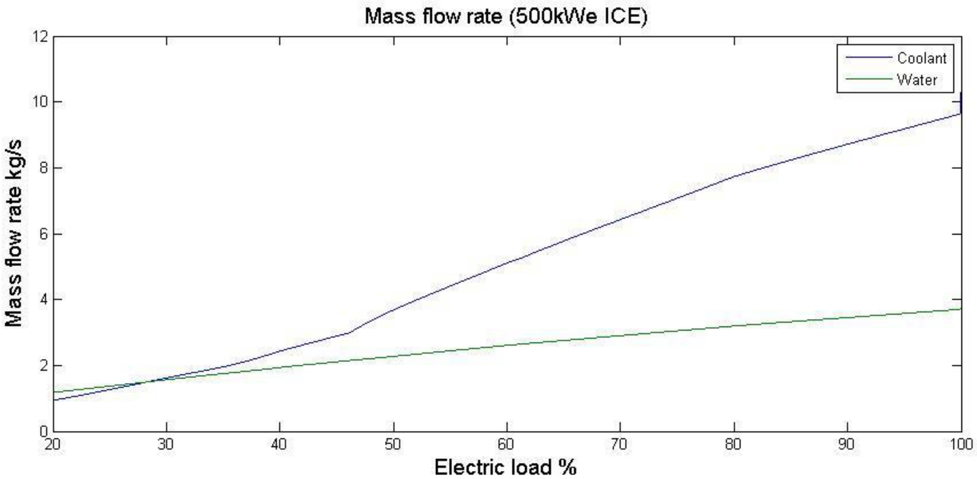

Fig.7.Trend of mass flow rates under varying load.

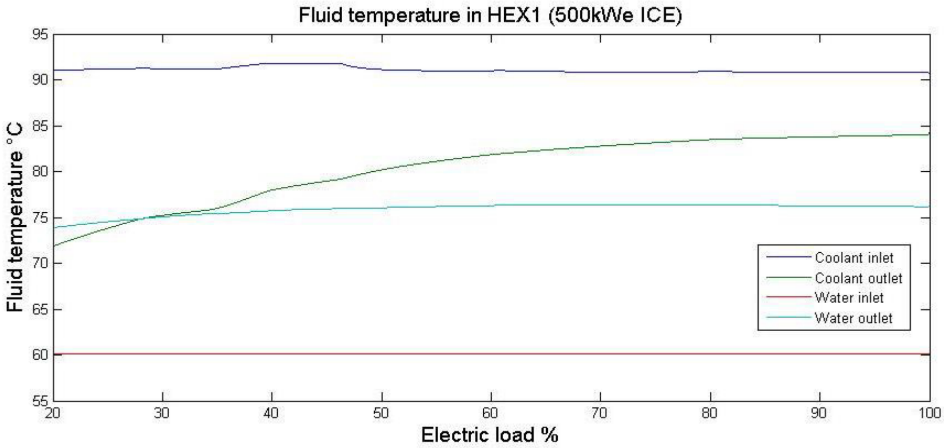

Fig.8. Temperature trend for input and output from HEX1.

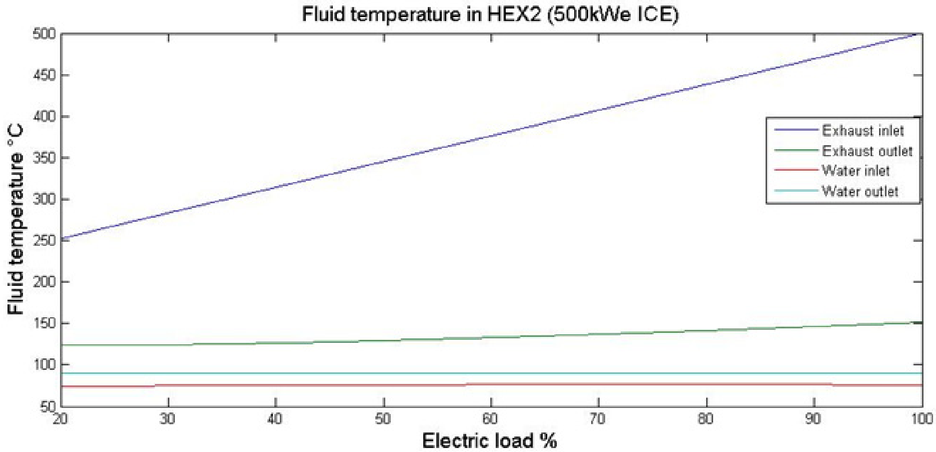

Fig.9.Temperature trend for input and output from HEX2.

3.2 Test B results

The "TEST B", has the aim to verify the dynamic behavior of the CHP system in a real operating condition. The electrical load input (Fig.10) varies to cope with an increase of the power demand (from the conditions of minimum to 70% load), similar to a typical morning trend, and a subsequent, unpredicted, shutdown (from 70% to zero) which a reduces the electric load with a very fast transient.

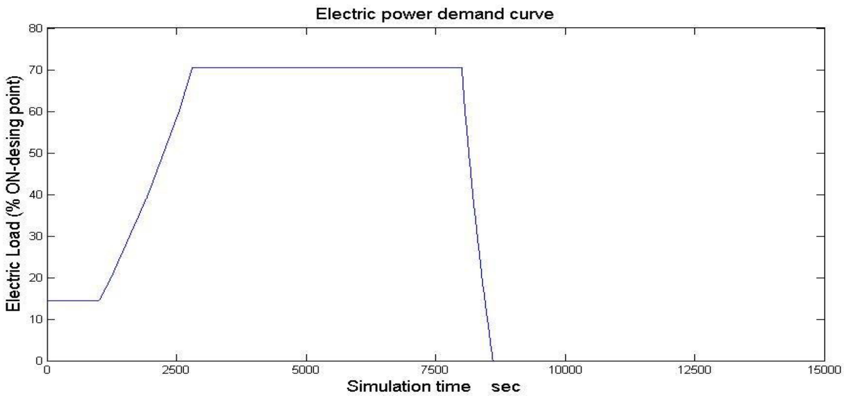

Fig.10. Electric power demand in TEST B.

In this second set of simulations, a non-monotonic curve of load demand is considered. This curve is adopted to describe the transient behavior in the phases of switching on and off the CHP.

Fig.11 shows the behavior of the mass flow rates. It is observed that, also in this case, the flow of water is strictly proportional to the required load and therefore it is constant in the phase in which the demand is constant. On the contrary, in the same period, the flow rate of refrigerant grows up and tends asymptotically to the design value.

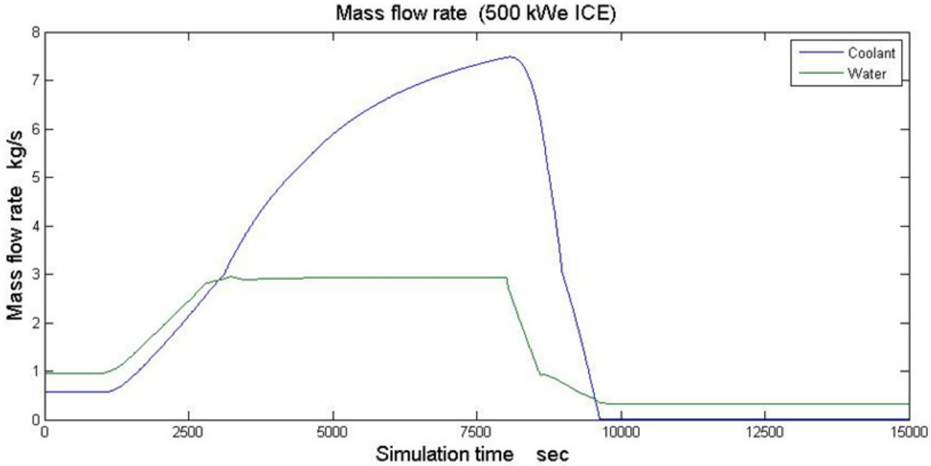

Fig.11. Mass flow rates.

Fig.12 describes the temperatures of the fluids in HEX1. It is observed that during the switch-off phase, even if the thermal load introduced into the exchanger is null, the heat exchanger continues to transfer heat from the refrigerant to the water until all the temperatures of both the fluids go to an equilibrium.

In Fig.13, the temperatures in the HEX2 are shown. It can be noticed that, in this case, the exhaust gas temperature remains high even in the condition of minimum and then is maintained the temperature for water output from the CHP in which flows a minimum.

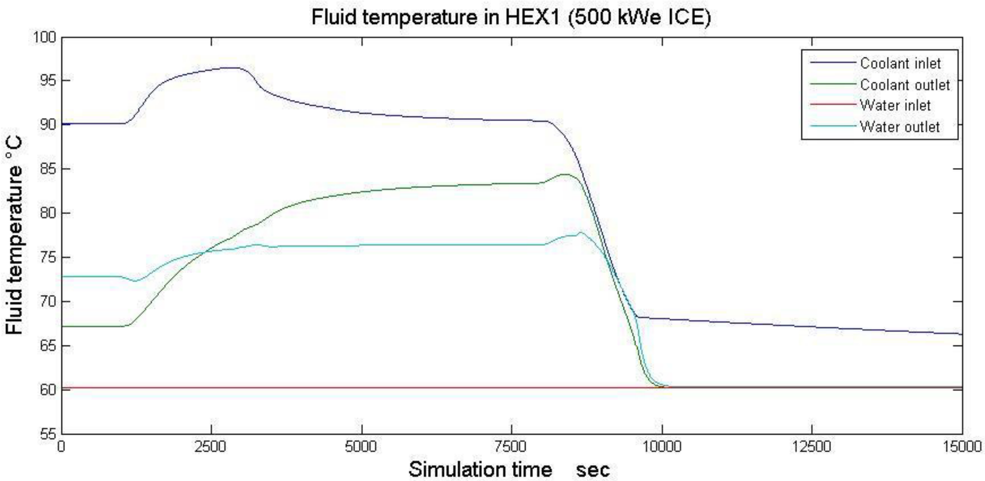

Fig.12 Temperatures in HEX1.

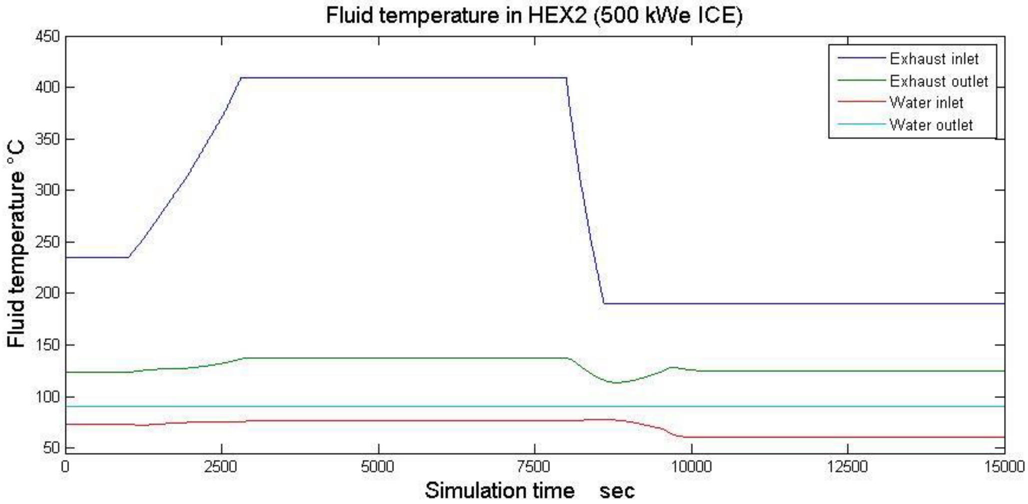

Fig.13 Temperatures in HEX2.

Looking at Fig.12 and Fig.13 it is can be noticed that the overshoots, which are caused by temporary unbalance of the mass flow rates, are limited to few degrees (5-10°C).

4. Conclusions

In this study, it has been developed a numerical model aimed at studying the dynamic behaviors of CHP systems and its related heating network.

The model has been implemented by Matlab-Simulink software with the explicit goal to develop and analyze the performance of the control system and its effects on supply temperatures of the connected DH.

The present numerical model has proved to be easily scaled in order to describe the behavior of Diesel CHP, non-turbo charged, of different sizes.

The simulator easily predicts the output temperatures and mass flow rates of the fluids passing through the heat exchangers. The control system is able to maintain the correct temperature at the users and to limit the overshoot in a range lower than 10°C.

Future work will include the possible implementation of separate heat exchangers for each cooling fluid.

Acknowledgements

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