Increasing Energy Efficiency of a Combined Head and Power Plant Using the Heat Storage

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***Abstract*—The demand of heat of the urban consumers registers rather great variations from one season to another. The thermal load curve flattening, in hot and intermediate seasons, is possible due to the accumulation of heat during the periods of time when the heat consumption is low and its utilization during the thermal load peak periods. The paper analyzes the integration of heat accumulation using water sensible heat in a Combined Heat and Power Plant (CHPP) from Romania, with Reciprocating Gas Engines (RGE) and hot water boilers (HWB). The solution selected presupposes the adaption of an existing heavy fuel oil tank. Utilization of heat accumulation in the CHPP makes it possible both to diminish the time of HWB utilization and the increase in the energy produced in cogeneration, and the utilization of the RGE at maximum continuous load.**

***Index Terms*—cogeneration, energy efficiency, internal combustion engines, heating, heat storage.**

I. Introduction

The heat sources should monitor the consumption needs of the consumers connected to them. In residential/urban applications, the heat demand is the sum between the heat demand (only in winter and according to the exterior temperature) and the demand of hot water for domestic utilization (all year round, mainly according to hours during the day). Considering Romania's climate conditions the heat demand represents more than 80 % of the total annual heat delivered. Overall, there results that, regardless of the season, there are rather great relative variations of the hourly demand of heat depending on the average daily demand.

Combined Heat and Power Plant (CHPP) have two types of heat supply sources [1, 2]: a) installations for the combined production of electricity and heat that cover the base and semi-base zone of the thermal load curve and b) installations for the exclusive production of heat covering the semi-peak and peak of the same curve (HWB – hot water boilers).

The heat demand variability is the main problem of the CHPP–DH (district heating) systems [3]. Profitability of a CHPP heavily depends on the price of electricity on the market, which influences the availability of cheap CHPP plant heat [4, 5].

Thermal energy storage (TES) increases the efficiency and operating flexibility of the CHPP - DH systems [6].Thermal energy storage (TES), can be coupled with CHPP by means of a smart operation scheme to match the production, and consumption profiles, and result in better values [3], increasing revenues and thermal production capacity of the CHPP [7].

The use of thermal energy storage during power plant operation has been discussed and analyzed in various researches and studies including both short-term thermal storage [5], long-term thermal storage [8], and use together with cogeneration. The medium for the storage of energy can be solid, liquid in two phases (liquid plus vapors), or gaseous [9].The paper analyzes the advantages of introducing a heat accumulation system into the hot water network of the CHPP with Reciprocating Gas Engines (RGE) and hot water boilers, for increasing energy efficiency using water sensible heat storage. Due to the great number of variables, the effects cannot be pointed out on a general case but only in the case of a specific CHP plant with concrete imposed data.

II. CHPP with RGE – Case Study

The example chosen for the case study is that of a CHPP from Romania that produced energy in cogeneration by means of two reciprocating gas engines each of which develops an electric power of 6.805 MW (representing 47.3 % of primary energy, calculated at LHV) at nominal load and that enable the recovery of its heat losses of a heat flow rate de 5.566 MW (representing 38.69 % of the input flow, defined as above). The exclusively heat producing capacities are 2 HWB of 58 MW and 29 MW (Fig. 1). The heat carrier is hot water and the variation of the delivered heat flow rate is ensured through the modification of both the heat flow rate (heat carrier) and the hot water temperatures on the return (t_{supply}) and return (t_{return}) of the supply network.

In summer, the thermal level is set by the temperature of the water for domestic consumption consequently the temperature variations (Δt) between the supply (t_{supply}) and return (t_{return}) are very small. In winter for the radiators in the dwellings to be able to yield greater thermal powers, the hot water average temperature inside them should increase.

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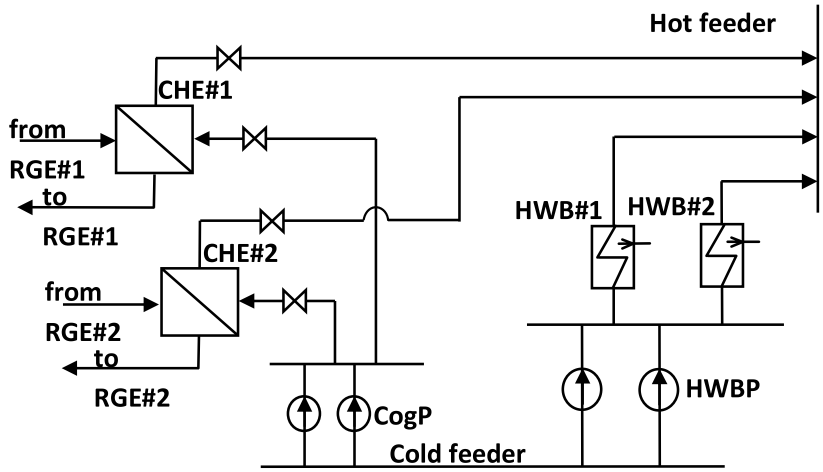

Figure 1. Scheme of analyzed residential CHPP, with RGE and HWB (HWBP – HWB pumps, CogP – cogeneration pumps, CHE – heat exchanger).

In summer, the thermal level is set by the temperature of the water for domestic consumption consequently the temperature variations (Δt) between the supply (t_{supply}) and return (t_{return}) are very small. In winter for the radiators in the dwellings to be able to yield greater thermal powers, the hot water average temperature inside them should increase. This is achieved by increasing both the supply and return temperature. In order to maintain the flow rate within reasonable limits the temperature increase slope on the supply will be higher than that on the network return.

The analysis has considered the data measured on three time intervals corresponding to: the cold season (winter), hot season (summer) and intermediate season (spring and autumn). The temperatures t_{supply} and t_{return} have been measured every day in the analyzed time intervals and the temperature difference between the supply and return ($\Delta t = t_{supply} - t_{return}$) has been determined. The curve of the temperature difference between the supply and return (Δt) has been drawn (Figure 2).

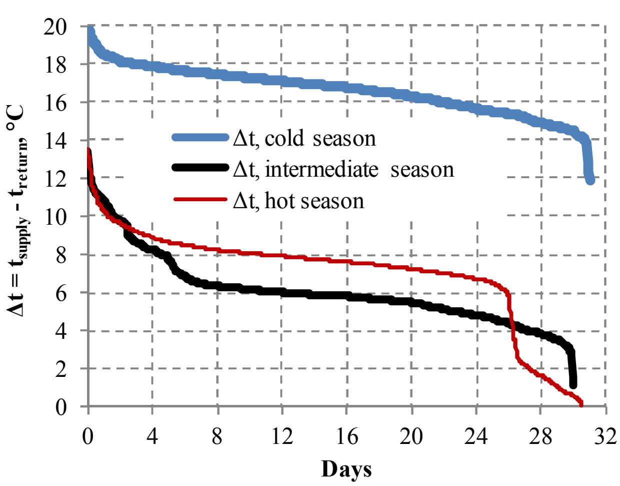

Figure 2. $\Delta t = t_{supply} - t_{return}$ (°C) vs. time (in days).

In a similar way, the hot water volumetric flow rates supplied to the thermal consumer (Q_v) have been measured and the curve of the flow rates has been drawn (Figure 3).

By means of t_{supply}, t_{return} and of Q_v the thermal power supplied to the thermal consumer ($\Phi_{thermal\ consumer}$) has been calculated [10]:

$$\Phi_{thermal\ consumer} = Q_v \cdot \rho_{water} \cdot c_p \cdot (t_{supply} - t_{return}) \qquad (1)$$

where: Q_v is the hot water volumetric flow rate, in m^3/s, ρ_{water} is the water density, in kg/m^3, c_p is the isobar specific heat of water, in kJ/ (kg·K).

By integrating in time the curves of the $\Phi_{thermal\ consumer}$ (Figure 4) the thermal energy supplied to the thermal consumer in each of the analyzed periods is obtained.

In Table 1, it is presented statistical data of hot water supply temperatures and return water temperature, on every season (winter season and summer).

TABLE I. WATER TEMPRATURE O SUPPLY AND RUTURN ON EACH SSEASON

Name	Supply Temperatures		Return Temperatures	
	Winter (°C)	***Summer*** (°C)	***Winter*** (°C)	***Summer*** (°C)
Maxim	73.514	67.955	56.368	73.514
Average + standard deviation	71.613	64.006	54.490	71.613
Avereage	70.217	60.701	53.579	70.217
Average - standard deviation	68.821	57.395	52.667	68.821
Minim	65.400	51.280	49.760	65.400
Standard deviation	1.396	3.305	0.911	1.396

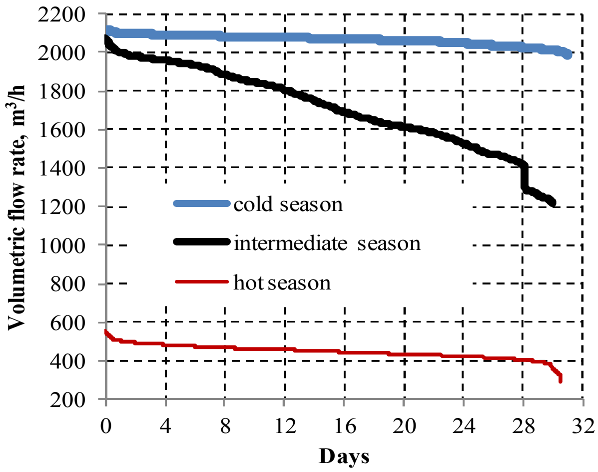

Figure 3. Volumetric flow rate (m^3/h) vs. time (in days).

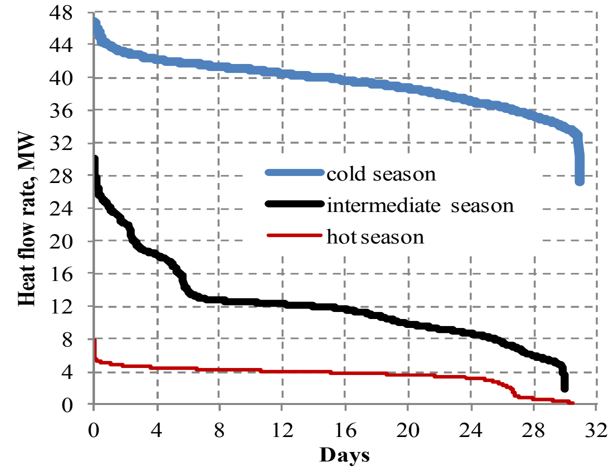

Figure 4. Heat flow rate (MW) vs. time (in days).

By referencing the thermal energy supplied to thermal consumer to the period of observation the average powers characteristic of the analyzed periods of time ($\Phi_{\text{thermal average}}$) are obtained: 39.358 MW for the cold season, 12.213 MW for the intermediate season, and 3.540 MW for the hot season.

Based on the analysis in Figure 4 and on the $\Phi_{\text{thermal average}}$ there appears that:

- During the cold season, $\Phi_{\text{thermal consumer}}$ is much higher than the thermal power that can be recovered from the two engines (11.132 MW), consequently it is necessary to use HWB during the entire period of time analyzed.
- During the intermediate season $\Phi_{\text{thermal average}}$ (12.213 MW) is comparable with the thermal power that can be recovered from the two engines (11.132 MW), during the analyzed period of time there existing both an interval of time when HWB are utilized, but also cases when the thermal power that can be recovered from the two engines maximum continuous load, surpassing the thermal consumer demand. There are also instances when $\Phi_{\text{thermal consumer}}$ is of the same order of magnitude as the thermal power recovered from a single engine.
- During the hot season $\Phi_{\text{thermal average}}$ (3.54 MW) is below the value of thermal power recoverable from a single engine, at maximum continuous load.

III. Justification of the Chosen Solution

The traditional equipment solution is that in summer the demand be covered only by the cogeneration installation while in winter the latter operate together with the HWB. The important daily variations of the thermal load may cause inconveniencies in the two types of seasons. In summer, during nighttime the demand of heat for hot water preparation for domestic consumption is much lower than during daytime.

In the big cities, such variations can be partially compensated by the thermal inertia of the big district heating networks.

In the small and medium size towns the short district heating networks made up of pipes with smaller diameters, have reduced thermal inertia and the cogeneration installations may be forced to operate at low loads with low performance indices.

At the beginning and end of the heating period, there are great variations of the heat demand during daytime. During the daytime, as during the nighttime periods of the hot season,, it can be lower than the heat flow rate installed in cogeneration. During nighttime the heat demand increases and it is necessary to start the HWB. In both situations, the implementation of a heat accumulator within the hot water network has beneficial effects. During the daytime, the cogeneration capacities can be maintained at the nominal load producing more electricity and heat with the best performance indicators, while the generated surplus of heat can be stored. During nighttime, the increased heat demand can be covered from the thermal storage, thus avoiding operating the HWB and producing heat exclusively.

IV. CHPP with Heat Storage

Improving the power plant operation is made, in this paper, by adapting an existing HFO (heavy fuel oil) tank, by 4836 m^3, to storage the heat, using the water sensible heat. The heat accumulator is of the mono-tank, bi-stock type with thermal stratification. The integration of a heat storage tank (HST) in the existing diagram (Figure 4) is presented in Figure 5. The main modifications of the existing scheme are the following:

- Moving the CogP from the district heating network return (Figure 4) on the supply (Figure 5);
- Utilization of new pumps (NewP) for covering the pressure losses in the CHE and the internal network;

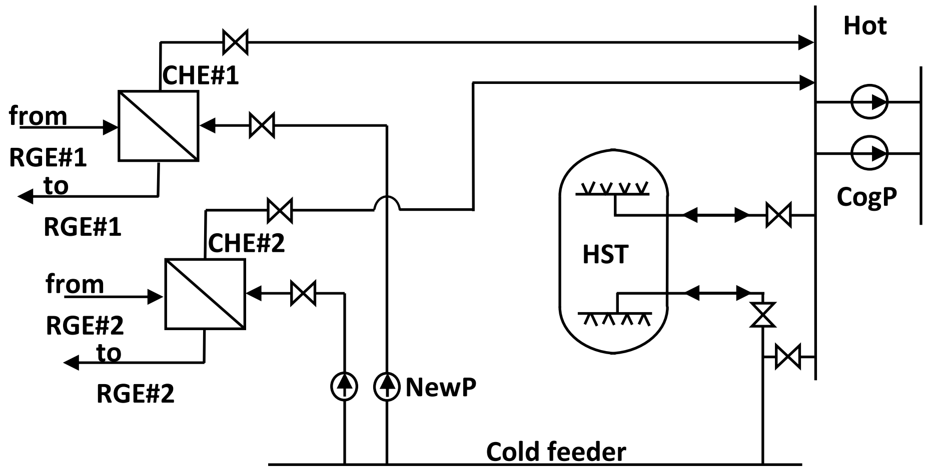

Figure 5. Scheme of propesed residential CHPP, with RGE, HWB and HST (NewP – new pumps)

- Integration of HST in parallel with CHE through a system of pipes and valves.

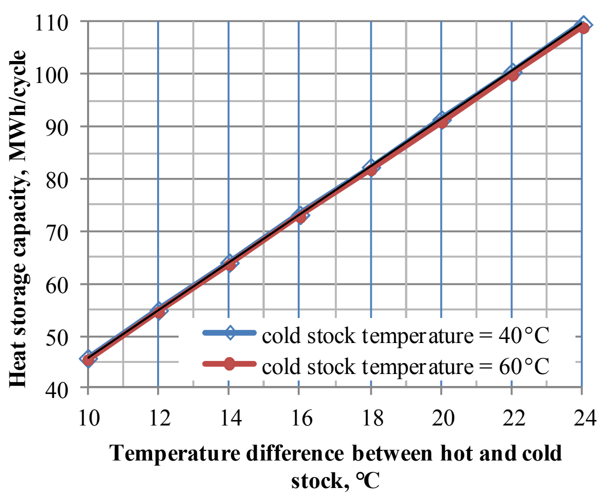

Figure 6. Heat storage capacity vs. temperature difference between hot and cold stock of HST, for differents cold stock temperatures

The heat storage capacity is influenced by the temperature difference between hot and cold stock of HST, and isn't depended by the cold stock temperatures (Figure 6). By increasing the temperature difference between hot and cold stock of HST with 10 °C (from 12 to 22 °C), the heat storage capacity increases by 45 MWh/cycle (from 55 to 100 MWh/cycle), considering a net storage capacity by 3990 m^3.

The HST functions in the following way:

- During loading, the HST takes over the hot water from the CHE outlet and dislocates the cold water in the cold feeder for supplying the CHE.
- During unloading, hot water from the HST is taken over through the CogP, the amount of dislocated water being taken over from the cold feeder.
- For regulating water temperature in the district heating network supply, a bypass between the cold feeder and the hot feeder is mounted.

Figure 7 presents the period necessary for storing the heat in the tank starting from the „completely unloaded" state up to „completely loaded" state, according to the temperature difference between the stocks when functioning with one or two MP, at different loads.

Based on the analysis Figure 7 the following remarks can be made:

- For high temperature differences between hot and cold stock of HST (e.g 23 °C) the complete charging time of HST can rich 24 hours, if one RGE is used at 75 % loading.
- If one RGE is used, the complete charging time of HST can decrease between 2 and 5 hours if the RGE is used at 100 % loading instead 75 % loading, for a temperature differences between hot and cold stock of HST between 10 and 24°C.
- For decreasing charging time of HST, the utilization of two RGE instead one can be taken into consideration, even if the demand of heat can be covered by using one RGE.

When the thermal stock is unloaded, the amount of cold that enters the tank is greater than that of the discharged hot water. When the thermal stock is loaded the amount of hot water that enters the tank is lower than that of the discharged cold water. Figure 8 presents the water variation of the amount of water exchanged between the heat storage tank and the hot water network every half hour of the loading – unloading cycle according to the difference in temperature between the stocks.

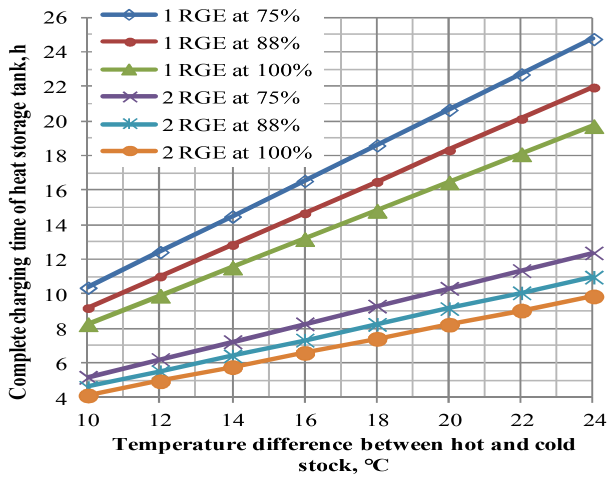

Figure 7. Complete charging time of heat storage tank vs. temperature difference between hot and cold stock of HST

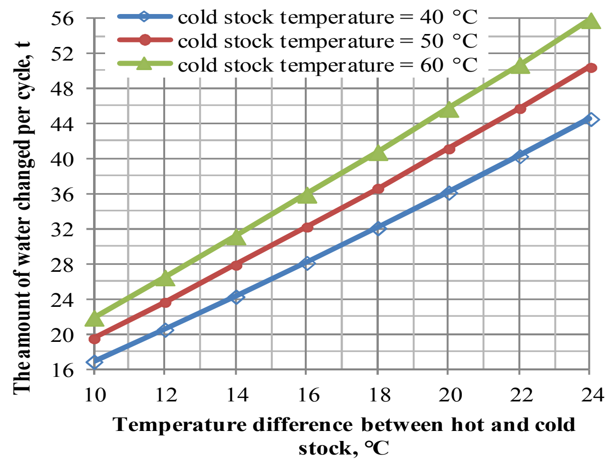

Figure 8. The amount of water changed per cycle vs. temperature difference between hot and cold stock of HST

The analysis of the results obtained in the paper point out that:

- During the cold season, the heat demand surpasses the amount of heat produced in cogeneration that is why the power plant uses two hot water boilers for meeting the demand. In this case, the heat storage is not used
- During the hot season, the heat demand is lower than the amount of heat that can be provided in cogeneration by the operation of a RGE at maximum load. In this case, the heat storage allows the engine to be used at a constant load (maximum load).
- In spring and autumn (intermediate mode), the heat storage reduces the duration of hot water boiler utilization, increasing the heat delivered in cogeneration and the power plant energy efficiency.

V. Conclusions

The paper underlines the benefits of heat storage integration using an existing heavy fuel oil (HFO) tank into the power plants with reciprocating engines. Integration of the heat accumulator into the CHPP circuit has positive effects only in a quota of the intermediate season and during the hot season.

Integration of heat storage in the analyzed power plant increases energy efficiency by decreasing the utilization time of HWB and by increasing the utilization time of RGE in cogeneration. In summer the RGE will be operate shorter periods, at nominal electric load.

As a result, RGEs can be used at times when electricity is sold more expensive, which will increase CHPP revenues. The heat demand can be assured using the heat accumulator without RGE operation.

In the intermediate season, the presence of the heat accumulator makes it possible to cover the higher night consumption with the heat produced in cogeneration and accumulated during the day.

As a result, the power plant energy efficiency is increased: 1) RGE will be able to be used all the time at nominal load, 2) The ratio of electricity versus heat in cogeneration will increase, 3) Annual share of cogeneration will increase ($MWh_{th\ in\ cogeneration}/MWh_{th\ total}$).

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