Design and operation strategies for a gas turbine based CHP scheme used for district heating

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1. Introduction

The purpose of this paper is to recommend design and operation strategies for a gas turbine based CHP (combined heat and power) scheme used for district heating (community heating). Residential heat consumers are usually supplied with hot water at a temperature that can reach 120 – 150 degrees Celsius. The water is heated first in a heat recovery water boiler (HRWB) using exhaust gases from the gas turbine (GT) and then, if necessary and only during wintertime, in a peak load water boiler.

The overall heat demand of a group of residential Rumanian consumers consists in space heating and warm water supply, both components being variable in time (during a day, during a season, during a year). Recent measurements showed that its minimum value (between 8 and 16 %) is reached during summer.

Under those circumstances, when the overall heat load decreases, the base heat load decreases as well. Hot water flow rate can either be kept constant or becomes variable (increasing or decreasing). The decrease of the base heat load is leading to GT part load operation and is taking place simultaneously with the increase of air temperature. This particular aspect influences the ratio between the electric power generated by GT and the amount of heat recovered from the exhaust gases flow rate corresponding to that power output, known as cogeneration index, and consequently overall CHP plant performances.

Both design and operation of such a CHPP must consider measures to improve plant performances and especially plant financial benefits.

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2. Computer model

A computer model of the entire heat supply system (CHPP, thermal network, thermal substations and heat consumers) is used to substantiate the way that variation of hot water parameters (flow rate and temperatures) and air temperature influence CHPP annual performances.

When the gas turbine electric output decreases, exhaust gas temperature and in some cases exhaust gas flow rate decreases as well. A specific gas turbine (TGC435CT) part load behaviour made public by the manufacturer (TUMA TURBOMACH SA) is used as a part of this computer model:

$$\begin{aligned}
&P_{\max} = 5{,}8 - 0{,}035 * t_a \\
&P = a * Q_B - P_0 \\
&D_g = 22{,}67 - 0{,}07464 * t_a \\
&t_g = (0{,}1625 * t_a + 58{,}5) * P + (1{,}655 * t_a + 154{,}0) \\
&P_0 = 28{,}4 * \left(\frac{t_a}{1000}\right)^2 + 2{,}86 * \left(\frac{t_a}{1000}\right) + 1{,}667 \\
&a = 0{,}408 - 0{,}00011 * t_a
\end{aligned} \tag{1}$$

Without supplementary firing, the amount of heat recovered from the exhaust gases depends upon exhaust gases and hot water flow rates and inlet temperatures. Gases flow rate decrease is a technical characteristic of a specific gas turbine. Instead, hot water flow rate decrease or increase can be decided by the heat supply system operator, provided the water pumps are properly controlled.

The amount of heat recovered from the exhaust gases depends upon exhaust gases flow rate and temperature and heat exchanger effectiveness as follows:

$$Q_{CG} = \varepsilon_{RC} * D_g * c_g * (t_g - t_w) \tag{2}$$

Hot water flow rate can be related to the space heating energy demand by means of a numeric coefficient [1]:

$$\frac{D_i}{D_{i0}} = \left(\frac{Q_i}{Q_i^{\max}}\right)^r = x_i^r \tag{3}$$

Its value must be positive $0 \le r \le 1$ and constitutes a distinct way to ensure heat load control during annual system operation. From a technical viewpoint *r* can be greater than 1 ($r \ge 1$), but its value is limited by the increase of hot water temperature during annual operation. We must emphasize that numeric coefficient r influences only a part of the entire hot water flow rate. This part corresponds only to the space heating demand and varies with *r* only during wintertime. The rest of the hot water flow rate during wintertime and the whole hot water flow rate during summertime are not influenced by *r*.

A thermal substation is a heat transformer consisting in two or three water-water heat exchangers located between the heat source (CHPP) and the heat consumers. Its scheme is influencing water temperatures and flow rate within the system. In this case there are two heat exchangers, one for space heating and the other for warm water, supplied in parallel with hot water coming from CHPP. Both heat exchangers are operated at different temperatures and heat loads than the rated values (design point) during a year. The thermal substation model for such situations is also based on heat exchangers effectiveness.

Air temperature range and heat load profile correspond to an aria near Bucharest.

3. Model results

The computer model generated a great amount of useful data. Every value was obtained as a function of air temperature and CHPP corresponding heat load. Considering the duration for each air temperature, average annual values of several CHPP parameters and performance indices could be calculated. CHPP parameters calculated as functions of air temperature are:

a) Hot water and exhaust gases flow rates and temperatures;
b) Power output and heat recovered from exhaust gases;
c) Cogeneration index and CHPP overall efficiency;
d) Cumulative sum of CHPP income.

Data displayed in table 1 show how much and when cogeneration index is influenced by the numeric coefficient *r*. A comparison of the last three columns of table 1 illustrates that they are slightly different only in wintertime, when the numeric coefficient *r* is influencing the amount of hot water flow rate. During summertime, the three values of the cogeneration index are identical because hot water flow rate and all other CHPP parameters are the same.

The first two columns of table 1 also show the relation between air temperature and CHPP heat load for an aria near Bucharest.

Values in the last three columns of table 1 correspond to a maximum to minimum base heat load ratio equal to 2.1.

Table 1

Cogeneration index

Air temperature (degrees Celsius)	Relative CHPP heat load	Numeric coefficient r		
		0	0,5	1
-15	1,000	0,674	0,674	0,674
-12	0,937	0,661	0,660	0,659
-9	0,859	0,647	0,645	0,644
-6	0,780	0,633	0,631	0,629
-3	0,693	0,620	0,617	0,614
0	0,606	0,607	0,604	0,600
3	0,511	0,593	0,592	0,587
6	0,407	0,580	0,580	0,576
9	0,288	0,570	0,570	0,568
12	0,136	0,492	0,492	0,492
15	0,130	0,467	0,467	0,467
18	0,125	0,446	0,446	0,446
21	0,120	0,420	0,420	0,420
24	0,115	0,392	0,392	0,392
27	0,110	0,366	0,366	0,366
30	0,105	0,333	0,333	0,333
33	0,100	0,296	0,296	0,296
36	0,095	0,247	0,247	0,247

Annual cogeneration index and CHPP overall efficiency were also calculated (tables 2 and 3). The results of these calculations are extremely comprehensive.

Table 2

Annual cogeneration index

Maximum to minimum base heat load ratio	Numeric coefficient r =				
	0	0,25	0,5	0,75	1
1,05	0,572	0,572	0,571	0,571	0,570
1,57	0,551	0,551	0,550	0,550	0,549
2,10	0,523	0,523	0,523	0,522	0,521
2,62	0,502	0,502	0,501	0,501	0,500
3,14	0,483	0,482	0,482	0,481	0,480

It is obvious that the value of the numeric coefficient r has a very small influence on both annual average values. The computer model is able to obtain a specific optimal value or a range for the numeric coefficient r in order to insure maximum electricity supplied to the grid. Its target is the difference between the annual amount of electricity produced and the annual amount of electricity consumed to drive hot water pumps. This specific optimal value or range does exist and it is influenced by the way that cogeneration index decreases when hot water flow rate decreases. This is the case for cogeneration steam turbines [2]. If the cogeneration index does not decrease at all or not enough, the annual amount of electricity supplied to the grid does not reach a maximum value at a specific value of the numeric coefficient *r*. By consequence, the numeric coefficient *r* has not an optimal value or range. This is the case of a gas turbine operated in cogeneration mode.

Table 3

Annual CHPP overall efficiency

Maximum to minimum base heat load ratio	Numeric coefficient r =				
	0	0,25	0,5	0,75	1
1,05	0,849	0,849	0,849	0,849	0,850
1,57	0,843	0,843	0,843	0,843	0,844
2,10	0,831	0,832	0,832	0,832	0,833
2,62	0,821	0,821	0,821	0,822	0,822
3,14	0,810	0,811	0,811	0,811	0,812

It is also obvious that the annual overall efficiency of the combined heat and power plant is in the same situation versus the numeric coefficient r.

The conclusion is that from a technical viewpoint, heat load control can be realised by any r value between 0 and 1 and even greater then 1 without influencing the annual amount of electricity supplied to the grid by CHPP. There are however two conditions:

a) Variable speed drive for the electric motors driving water pumps;
b) Effective control of hot water distribution to each thermal substation.

The computer model results also show that the ratio between maximum and minimum base heat load (HRWB heat load) should not be greater than 1.5. Data displayed in table 3 show that if this ratio is greater that 1.5, the cumulative sum of CHPP income becomes negative at high air temperatures, during summer. The explanation is that every income corresponding to air temperatures greater than 12 degrees Celsius is negative. By consequence, annual profit will be higher if the GT unit is shut down and the hot water boiler is running during summer.

This situation changes for the better if instead of one GT we install two units. Those two GT units will have together the same rated power than the one. The advantage of this solution consists in the possibility that one of the two units will be operated only on winter. The other will be operated the whole year, so that each unit annual average relative heat load will be higher. The two GT units can be designed to use only one heat recovery unit (HRWB).

Table 4

Cumulative sum of CHPP income MEUR

Air temperature (degrees Celsius)	Maximum to minimum base heat load ratio				
	1,05	1,57	2,10	2,62	3,14
-15	0,042	0,030	0,024	0,021	0,018
-12	0,121	0,087	0,070	0,060	0,053
-9	0,229	0,165	0,134	0,115	0,102
-6	0,422	0,307	0,250	0,216	0,193
-3	0,676	0,496	0,407	0,353	0,317
0	0,965	0,716	0,592	0,517	0,467
3	1,230	0,924	0,771	0,679	0,617
6	1,452	1,106	0,933	0,830	0,760
9	1,587	1,230	1,052	0,945	0,860
12	1,607	1,259	1,043	0,913	0,812
15	1,623	1,275	1,029	0,881	0,769
18	1,637	1,286	1,012	0,846	0,722
21	1,650	1,289	0,992	0,811	0,679
24	1,660	1,288	0,970	0,778	0,637
27	1,668	1,282	0,949	0,746	0,599
30	1,674	1,276	0,931	0,721	0,571
33	1,677	1,269	0,917	0,703	0,550
36	1,678	1,264	0,909	0,694	0,539

4. Conclusions

Air temperature range and heat load profile mach an aria near Bucharest.

For a specific GT unit and a specific type of thermal substation, a computer model able to obtain a specific optimal value or a range for the numeric coefficient r was available. Its target was to insure maximum electricity supplied to the grid. This specific optimal value or range depends upon the way that cogeneration index decreases when hot water flow rate decreases. In the case of cogeneration steam turbines this decrease is important [2] and the optimal value for *r* is close to zero. In the case of GT and HRWB, the cogeneration index does not decrease enough to ensure an optimal value or range for the numeric coefficient *r*. As a consequence of this situation, the supply system operator is free to establish any *r* value.

The computer model results also show in what way the ratio between maximum and minimum base heat load influences CHPP annual benefit. Data displayed in table 3 show that for a ratio greater that 1.5, the cumulative sum of CHPP income becomes negative. It is the result of GT part load operation at high air temperatures, during summer. By consequence, annual profit will be higher if the GT unit is shut down and the hot water boiler is running during summer.

This situation changes for the better if instead of one we install two GT units. Those two GT units will have together the same rated power than the first one. The advantage of this solution consists in the possibility that one of the two units will be operated only on winter. The other will be operated the whole year, so that each unit annual average relative heat load will be higher. The two GT units can use only one HRWB.

This type of CHPP income analysis should be included in any feasibility study for gas turbine based CHPP used for district heating. There is however a difficult problem concerning real information and data on part load and different air temperature behaviour of a specific GT included in manufacturer's offer prior to feasibility study. For different reasons, offer information on the subject is often not complete or comprehensive.

5. Nomenclature (symbol list)

P – GT electric output;
P_0 – GT electric idle consumption;
P^{max} – GT maximum electric output for a specific air temperature;
Q_B – GT fuel consumption;
D_g – GT exhaust gases flow rate;
t_g – GT exhaust gases temperature;
t_a – air temperature;
t_w – hot water temperature;
D_i – hot water flow rate related to space heating demand;

D_{i0} - hot water flow rate related to space heating demand corresponding to minimum air temperature;

c_g – specific heat for exhaust gases;

Q_i – space heating demand;

Q_{i0} – space heating maximum demand;

ε_{RC} – HRWB effectiveness;

a - constant dimensionless coefficient.

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