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Combined heat pump and power plant. Part I: thermodynamic analysis

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1. Introduction

Post-soviet countries have inherited backward, ineffective power production technologies and low quality blocks of flats. Due to this, as well as given relatively low living standards, the inhabitants are faced with difficulties to pay constantly increasing price for energy, including household heat and power energy, even though the price of natural gas and oil in Ukraine, Belarus and especially Russia is lower than the world market price. Another peculiarity of the post-soviet space is a multitude of large towns which, as a rule, have progressive central district heating systems and natural gas piping.

According to the principle of cogeneration, combined heat and power (CHP) plants produce electricity in summer, and heat and electricity – in winter. It is generally considered that CHP plants have advantage against other heat producing technologies; indeed, this is true as regards their utilization efficiency. However, as regards the price of both heat and power, this is not so forthright. As the CHP plants are constructed more for heating purpose, the power production in summer is not profitable and thus the overall profitability is possible at the expense of the heat consumers in winter season.

There is also an ecological aspect of the problem. In order to comply with the strict requirements to decrease NO_x, these regional CHP plants need to be modernized. As this will require additional considerable costs and given that these plants are too old, the modernisation can be not worthwhile.

The described situation urges to search for alternatives to old CHP plants. One of them is to abandon CHP technology and to reconstruct these plants to produce only heat by using modern condensing boilers, the utilization efficiency of which is almost 100%. In this respect, the biomass as a fuel has an advantage both from ecological and maybe even price point of view. Another alternative is the appliance of the heat pump (HP) technology. However, the HP development in the region is very weak due to low winter temperature and comparatively high investment costs.

Nevertheless, the post-soviet region already has two conditions for the appliance of the HP plant, one of which is the said district heating system. The second one is related to natural gas piping system of these towns. Namely, the power plants fuelled by natural gas (NG) are the most effective today. The gas turbine combined cycle (GTCC) power plants have advantage due to high efficiency and low investment costs. Mechanical efficiency of the GTCC heat engine oversteps 60% which means that the engine produces over 4.8kWh of mechanical energy from $1m^3$ natural gas burning. The GTCC power plants are suitable in town territories because burning NG does a limited damage to the surroundings. The appliance of GTCC heat engine instead of an old steam turbine means that the GTCC heat engine turns not only the power generator but the compressor of the HP as well. Due to the high efficiency of this engine, the cost of electricity and heat, in particular, would be sufficiently lower. The comparable high temperature of the waste heat received into the steam condenser determines high coefficient of performance (COP) of the HP.

HP is the most effective mode to produce thermal energy in a good many cases. Moreover, in the face of increasing price of mineral fuel as well as due to ecological reasons, it is not thermodynamically sensible to produce thermal energy for heating purpose by burning the fuel directly. The required heat of about 20°C is obtained inside the boiler with the temperature of over 1000°C.

The increasing interest in HP plant for district heating is witnessed by the results obtained from the analysis of the scientific literature, most of which appeared in last six seven years. However, no studies or analysis concerning the appliance of the GTCC heat engine for large HP plant was found.

Ajah et al. [1, 2] as well as Lunghi and Burzacca [3] examine the problem of utilization waste heat from chemical and refinery plants by increasing its temperature up to suitable for district heating system. The authors present simulation results of applying chemical or mechanical HP to increase fluid temperature. The authors [4-7] present the appliance of the HP for district heating by utilising the heat of geothermal warm water as well as the heat of solar radiation [8, 9]. Hepbasli [10] examins the issue of using ground heat source for district heating, and Holmgren [11] analyses utilization of different low temperature sources.

All the works mentioned above present investigation as regards the appliance of the HP for production of thermal energy of temperature suitable for district heating system. Most of them investigate mechanical HP system when electrical engine turns the compressor.

There are several works, for example [12-14], wherein the case of the appliance of the heat engine in the HP system is analysed. However, they do not concern powerful heat engines, such as gas or steam turbines. Meanwhile Lowe [15] presents analysis of the virtual HP cycle coupled to the steam cycle of CHP plant and demonstrates a high COP of the HP. The general aim of the Luickx et al [16] is to search technological manner of employing night energy excess in the most densely habitable region of Europe. The appliance of massive HP is discussed in this article as well as in another one [17]. However, there is no mention of GTCC heat engine appliance in these articles, except in Lazzarin and Noro [18, 19] who make just a hint that GTCC heat engine could be applied in the HP system as a means for using excess of mechanical energy when demand of power is lower.

In their articles [18, 19], Lazzarin and Noro focus on the analysis of CHP plants; they draw a conclusion that if the efficiency of a power generating of CHP plant is lower than 24%, the total efficiency of the plant becomes lower than that of the condensing boiler. The authors also state that if condensation heat is recovered at a pressure higher than atmospheric, electrical efficiency generally decreases twice and is less than 20%. Lazzarin and Noro draw attention to the fact that the thermal energy generated by the HP is more efficient than that of the CHP plant. According to the authors, "heat pump coupled to condensing boilers is among the most modern and efficient heating technologies with an overall energy efficiency often better than the cogeneration in district heating".

2. CHPP plant cycle and COP

The HP condensing temperature (or pressure) depends on the required temperature of outgoing water, which leaves condenser and is directed to district heating grid. Winter conditions in East European countries (Poland, the Baltic republics, Ukraine, Belarus, etc.) require the outgoing water temperature of approximately 75°C at the average temperature of the three coldest months. However, there are cold waves when temperature drops up to minus 30°C or below. In Lithuania, for example, ten days occur statistically in winter season when the average temperature falls below minus 10°C. Naturally, the supplied water temperature must be much higher in these conditions. This issue will be discussed later.

Under the average winter temperature when the outgoing water temperature is 75°C, the returned water temperature would be about 20°C lower, i.e. 55°C. The condensing temperature for HP would be 3-5°C higher. Since the temperature of the compressed vapour before counter flow type condenser is 98°C (Fig.1), the condensing temperature can be even lower than the outgoing water temperature. Primal calculation gives $t_C = 71$°C.

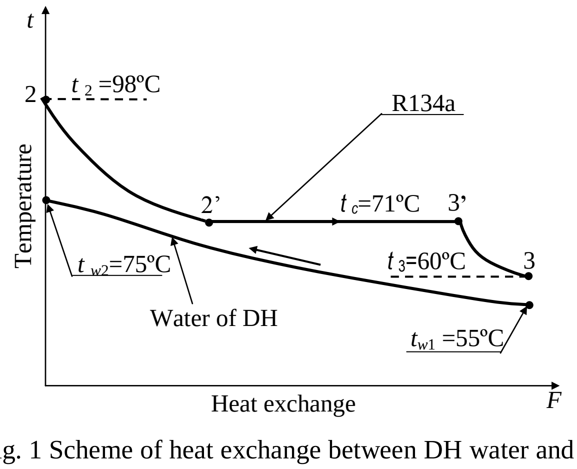

Fig. 1 Scheme of heat exchange between DH water and HP working fluid in the condenser

Refrigerant R134a is chosen as a working liquid for HP. However, another refrigerant namely R1234ze with similar thermal properties could be applied. The latter has similar thermal properties, so it is produced now as a substitute for R134a.

The evaporating temperature of R134a is several degrees lower than the water steam condensing temperature. High pressure steam expands in the turbine up to 0.05 bar which corresponds to water steam condensing temperature 32.9°C. Thus, the HP working liquid evaporates at 29°C at the beginning and 28°C – at the end of a heat exchanger, making comparatively small temperature difference.

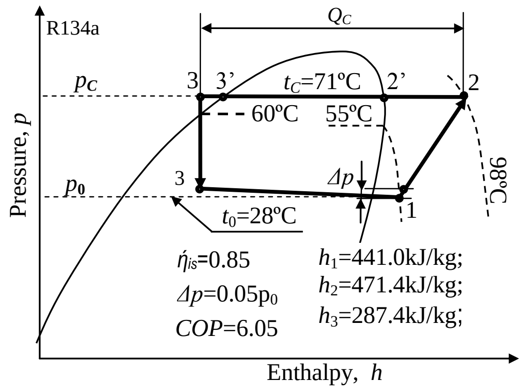

Fig. 2 Cycle of HPP and enthalpies of typical points

Two factors determine such a small difference of the fluids temperatures: isothermal heat exchange process and the factual absence of fouling on both sides of the surfaces. The said concerns also the HP condenser and justifies such a small temperature difference between working fluid and water (see t_3 and t_{W1} temperatures in the Fig. 1).

The efficiency of gas compression process is very important for the COP. Compressor must be as effective as possible. The isentropic coefficient η_{is} determines this effectiveness which reaches value of 0.85 for the modern powerful turbo compressors.

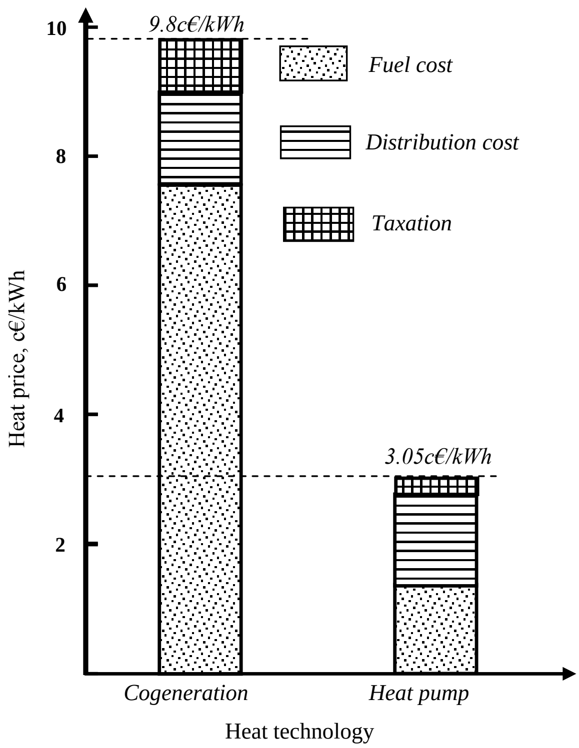

Fig. 3 Composition of heat price of different technologies

As could be seen from Fig. 2, the HP cycle efficiency is high enough (COP = 6.05) to get very competitive heat price compared to cogeneration and condensing boiler technology. This is mainly due to high evaporating tempe-rature and low compression ratio. The COP magnitude 6.05 means, that 1m^3 of natural gas by using HP produces (4.8×6.05=) 29.0 kWh of heat suitable for district heating. The price of 1 m^3 natural gas in Lithuania, for example, is 0,4€, so fuel cost for heat by heat pump technology makes only (0.4 €/29.0=) 0.0138 €/kWh, which is significantly lower compared to the corresponding price today, which is 0.0757 €/kWh (Fig. 3). On the other hand, the 0.0757 €/kWh cost includes also other costs of heat production plant, such as operation, depreciation, profit, etc. However, all of them make about only tenth of the value 7.57.

3. Required powerfulness of a GTCC heat engine

Obviously, the mechanical power necessary for turning of the compressor does not determine the overall powerfulness of GTCC heat engine. This is due to a high mechanical efficiency of GTCC (η_m = 0.60-0.62) and also high COP of HP. According to the Fig. 2, only one sixth of the produced heat energy makes mechanical power; and the amount of low potential heat necessary for the HP system is five times larger than mechanical power. Waste heat of the GTCC heat engine makes only 40% of the heat supplied to cycle and about half of all heat energy obtained by burning natural gas (Fig. 4).

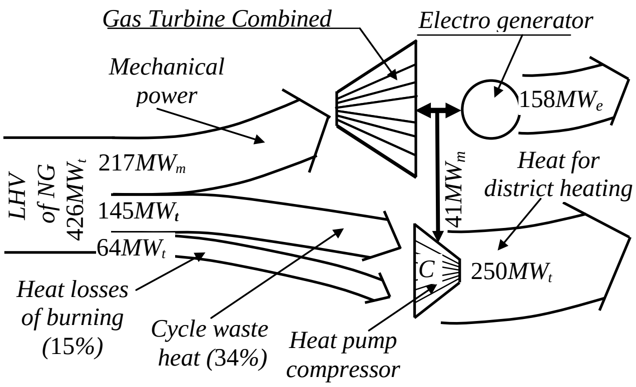

Fig. 4 Sections of primary energy of CHPP plant of 250 MW district heating demand

The overall or utilization efficiency of the GTCC η_t is 85%. Thus 49% of primary energy obtained by burning natural gas (209 MW of 429 MW) can be used for low potential heat requirements. According to the scheme presented in Fig. 5, for district heat demand Q = 250 MW (the case of Kaunas, for example), the HP requires 209 MW of waste heat, which is obtained from the GTCC heat engine together with heat from economizer ECN (Fig. 5). The HP compressor needs 41MW of mechanical power that makes only 19% of all mechanical power of the GTCC heat engine (217 MW). Another part is directed to electrical power production with efficiency η_E = 0.92 which includes all losses of generating electricity [18].

It is sensible to use formula for powerfulness P_E calculation of GTCC heat engine:

$$P_E = \frac{Q(COP-1)\eta_M \eta_T \eta_E}{COP(1-\eta_M \eta_T)} \,.$$

Thus, this is the variant when the GTCC heat engine of 200 MW of electric powerfulness is needed for 250 MW heat demand. The engine can produce annually about 1.2 TWh of electricity (with capacity factor 0.85) and almost the same amount of heat – 1.3 TWh (including average 50 MW of heat power for hot water in summer).

4. Scheme of CHPP plant and capacity reguliation

The scheme of the modernized CHP plant operating with GTCC heat engine which turns both electro generator and HP compressor is presented in Fig. 5. HP cycle operates without intercooler for preheating vapour before the HP compressor. Heat from economizer ECN is used for the vapour preheating in the AEP. The utilization efficiency of the GTCC is 0.85, so the waste heat amounts to at least 15% of low caloric value of natural gas [18, 19]. The counterflow heat exchangers ECN and AEP ensure about 55°C temperature (Fig. 2) of the HP working fluid at the entrance of the compressor C. So HP system consists of the condenser HPC, evaporator HPE, thermo valve TV system, compressor C and two additional exchangers ECN and AEP.

Water and steam distributor devices DD serve for heat capacity regulation. The device directs a bigger part of steam to PSC when the demand of district heat is lower (in the beginning of winter, for example). In this case, the water steam part, which condenses in the HPE, is over-cooled as well. Therefore, it makes the evaporating temperature lower; consequently the HP capacity decreases. The COP of HP cycle does not decrease much because the condensing temperature is lower under these conditions as well.

It is impossible to apply such a mode of regulation in case when demand of heat is higher the nominal. The HP evaporating and condensing temperatures must be increased by decreasing the cooling water mass flow in the PSC and the cooling tower. The temperature of the outgoing water from PSC, which is directed to cooling tower afterwards, will be higher, which increases water steam pressure in the PSC and HPE. The higher steam pressure in the HPE should also increase the HP working fluid pressure; consequently, the HP mass flow would increase as well as temperature in HPC (additionally, it would need to be optimised with the help of TV).

The heat demand in summer regime is sufficiently lower and in fact is only related to hot water preparation. The scheme ensures effective operation in summer in case the working fluid in the HP system is changed by another fluid. R134a working fluid is used in winter regime, as could be seen from Figs. 1 to 3. For summer regime, i.e., for heat capacity lower by several times, the HP working fluid of much less volumetric capacity, namely, R123 or R11, is needed. At the same temperature of the district heating water, outgoing from HPC, the HP compressor with, for example, R123 fluid would produce six times less heat. Therefore, the construction of turbo compressor has to be designed for R134a because operating conditions with fluids of lower volumetric capacity are much easier.

Under summer conditions, the heat exchangers HPE, HPC and AEP would operate at almost 100% efficiency. Part of steam directed to HPE should be several times less, respectively. Heat capacity regulations in summer regime are of the same mode as in winter regime.

5. Heat pump cycle during cold waves and summer

In case of cold waves of regional winters, the demand of heat may even double. It means that both mass flow and, in particular, the temperature of supplied water increases. For example, in Lithuania, during an exclusively severe winter of 2012, the water temperature has been raised to 110°C. In order to supply the required heat capacity and temperature, the HP of CHPP plant must operate at much heavier conditions. Density of working fluid before and after the compressor must be much higher.

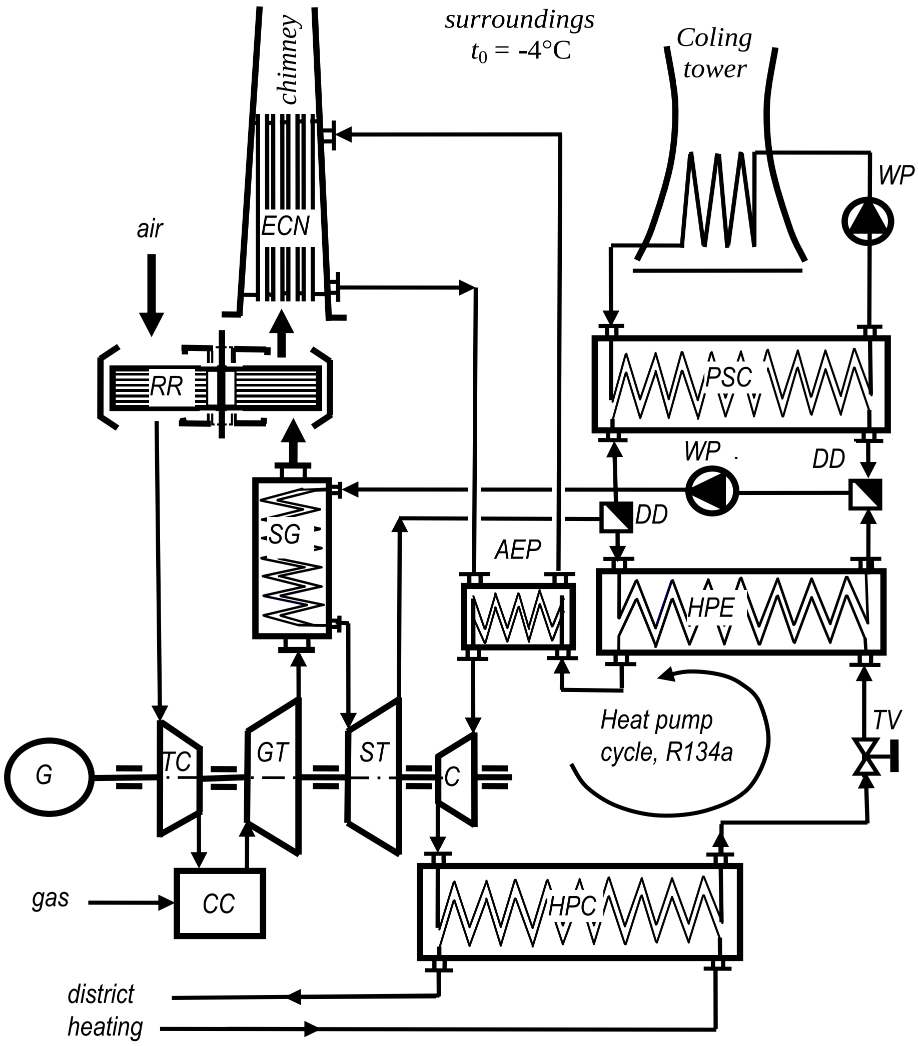

Fig. 5 Scheme of combined heat pump and power plant operating with GTCC. TC – turbo compressor of the GTCC heat engine; CC – combustion chamber; GT – gas turbine; ST – steam turbine; C – compressor of the heat pump (HP); HPC – heat pump condenser; HPE – heat pump evaporator; AEP – additional evaporator and preheater; TV – thermo valve; PSC – power station condenser; SG – steam generator; WP – water pump; RR- rotating regenerator; ECN – economizer; G – electro generator; DD – distributing device

In order to increase heat capacity twice, the pressure against the HP compressor must also be, roughly, double. The evaporating temperature of the working fluid R134a increases up to 56°C with corresponding pressure 15.3 bar. In its own turn, the condensing pressure increases up to 57 bar, which signifies a trans-critical regime of the HP (Fig. 6) and different heat exchange scheme in the condenser (Fig. 7). The strength of the heat pump condenser HPC must be much higher, which increases the condenser price. Adding the fact that extreme conditions require much higher capital costs for more powerful GTCC as well as for HP compressor, it is reasonable to not to construct HP to be used in extreme conditions but rather to apply the same cogeneration principle for one of the turbines.

GTCC heat engine has two turbines. The steam turbine could operate partly as a usual cogeneration steam turbine as it is possible to change the pressure behind it. Our case does not require such a high pressure of steam compared with cogeneration regime as the heat is used for heating purpose indirectly. This heat is required for the low potential heat exchanger, i.e. evaporator HPE and is transformed into the heat of higher temperature and capacity and can be regulated as in case, when heat demand is not much higher the nominal (see section 4).

In Kaunas, for example, there are, as an average, 7-8 days during winter time with the temperature of about minus 15°C. The temperature of the supplied water must be increased up to 90°C; the heat requirement increases by

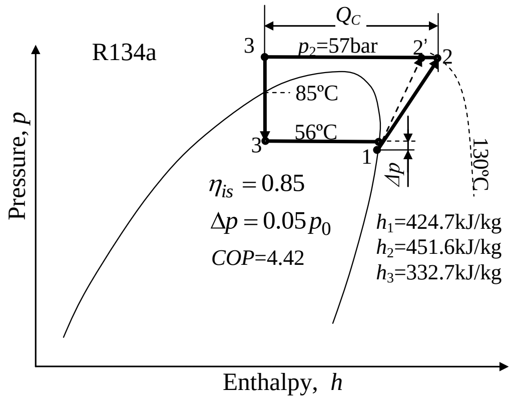

Fig. 6 Cycle of HPP and enthalpies at condition of a severe cold valve

about 50%. In order to satisfy these requirements, it is necessary to increase water steam condensing pressure in HPE up to 0,09bar. The pressure is quite normal for conventional steam turbines because it could occur during hot waves in summer. This higher pressure in winter can be reached by decreasing the cooling water mass flow in the central condenser PSC. Thus the GTCC heat engine will slightly lose its power; this loss can be compared to the one which occurs in summer with respect to winter.

In view of the above, it can be stated that the way to overcome extremely severe cold wave is to apply additional boiler for extra heating of DH water. The boiler could be fuelled by natural gas or bio fuel. It should be noted that the use of bio fuel complies with the strategic objectives of the region. An additional boiler could serve for heat water preparation in summer. GTCC heat engine would produce only electricity and could be stopped in summer when the production of electricity is loss-making. The cost of heat produced by bio fuel boiler is higher compared to the heat produced HP technology; however, the final price can be lower in case the sale of electricity requires to be subsidised by heat customers. The sale of heat in summer is much lower so the subsidy of electricity would increase the heat price considerably.

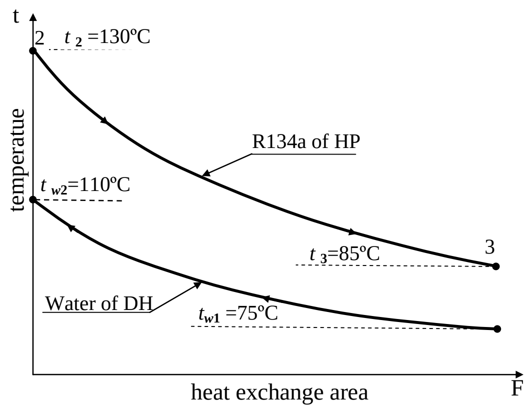

Fig 7 Scheme of heat exchange between DH water and working fluid of HP in transcritical cycle operation

6. Conclusions

Big post-soviet cities have progressive district heating grid and cogeneration power and heat plants fuelled by natural gas. However, heat consumers have to pay huge bills for heating due to the increased price of fuel. The presented combined heat pump and power plant can be cost-effective due to the heat pump technology, which is a much more effective means of heat production. The thermodynamic analysis of this technology proposes that the coefficient of performance is more than 6, which ensures not only a lower heat price but also a possibility to subsidise the sale of electricity. Such a high effectiveness of heat pump is conditioned by an effective GTCC heat engine and also high evaporating temperature of the HP working fluid in the power plant condenser. The developed scheme proposes a possibility to regulate heat capacity during winter and summer regimes. The formula developed for calculation of the powerfulness of the GTCC heat engine proposes the possibility to estimate various heat capacities for any big city with a district heating and natural gas piping. The cold wave problem is proposed to be solved by adding a supplementary boiler instead of entering into the trans-critical cycle regime of heat pump.

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V. Dagilis

KOMBINUOTA ŠILUMOS SIURBLIO IR ELEKTROS JĖGAINĖ. PIRMA DALIS: TERMODINAMINĖ ANALIZĖ

R e z i u m ė

Straipsnyje aprašyta pažangi kombinuota šilumos siurblio ir elektros jėgainė, kurioje tiek elektros, tiek šilumos gamybai panaudotas modernus kombinuoto ciklo šiluminis variklis. Ši nauja technologija užtikrina žemą šilumos savikainą ir galimybę subsidijuoti elektros pardavimą. Efektyvus šilumos siurblio darbas gaunamas dėka efektyvaus šiluminio variklio, taip pat dėka aukštos virimo temperatūros elektros jėgainės kondensatoriuje. Sukurta jėgainės schema leidžia reguliuoti šilumos našumą ir žiemą ir vasarą. Šilumos variklio galios skaičiavimo formulė leidžia jį paskaičiuoti įvairiems skirtingų miestų su centralizuoto šilumos tiekimo sistema ir gamtinių dujų tinklu šilumos poreikiams. Šalčio bangų problemos analizė rodo, kad užuot vertus šilumos siurblį dirbti virškritiniu režimu, geriau įrengti papildomą katilinę.

V. Dagilis

COMBINED HEAT PUMP AND POWER PLANT. PART I: THERMODYNAMIC ANALYSIS

S u m m a r y

The paper presents the thermodynamic analysis of an advanced heat pump and power plant which applies a modern heat engine of the gas turbine combined cycle for both heat and power production. This new technology ensures low cost of heat and a possibility to subsidise the sale of electricity. High efficiency of heat pump is conditioned by effective heat engine and high evaporating temperature of the heat pump working fluid in power plant condenser. The developed plant scheme provides for a possibility to regulate heat capacity during winter and summer regimes. The formula developed for the calculation of the powerfulness of the heat engine presents calculations for various heat capacities in any big city with a district heating and natural gas piping. The cold wave problem is proposed to be solved by adding a supplementary boiler instead of entering into the trans-critical cycle regime of the heat pump.

Keywords: heat pump, power plant, cogeneration.

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