WASTE HEAT RECOVERY FROM AIR TURBO–COMPRESSORS IN MINING: A COMPARATIVE STUDY BETWEEN COGENERATION AND HEAT PUMP TECHNOLOGIES

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Abstract

The possibility of cogeneration and heat pump technologies implementation for waste heat recovery from mining turbo-compressors has been investigated. It has been calculated that heat pump implementation for turbo-compressor heat recovery enables a reduction of compressed air production costs by half. This decrease in costs lessens as cooling water inlet temperature increases. It has been found that instead of burning fossil fuel in coal-burned boilers to meet mines' needs for thermal energy, it is possible to recover compressor stations' waste heat by using heat pumps. They can produce 1.6–2.0 MW of useful heat in the form of hot water at a temperature of 50 °C from a single 1.5 MW turbo-compressor, thus reducing the fossil fuel consumption in mines considerably. Cogeneration technology implementation for compressor waste heat recovery has shown that maximum power generation (up to 75 kW) can be achieved, depending on the compressed air temperature leaving the recovery sections of the air coolers (intercooling temperature). It can reduce turbo-compressor energy consumption by up to 5 % and make it possible to save 540 000 kWh of electricity annually from a single 1.5 MW turbo-compressor.

Introduction

Air compressor stations which produce compressed air for mining machinery operation and different technological processes are one of the main energy consumers in coal and iron mines. Therefore, the efficiency of mining greatly depends on the efficient production, transmission and consumption of compressed air. It is known that along with the mechanical work produced, mining air compressors release large amounts of energy in the form of waste heat. In a typical compressor cooling system, all heat produced as a result of a compressor operation is lost to the environment, not only lowering the energy efficiency but also causing the environmental problems such as global warming and heat pollution. Since during the compressor operation the amount of waste heat produced is almost the same as the amount of electricity consumed, it is reasonable to try to recover this waste heat for power generation or useful heat production. Moreover, since a great amount of fossil fuel (basically coal) in Ukrainian mines is burned annually in boilers for heating or hot water supply, any measures for the reduction of fossil fuel consumption could bring economic and environmental benefits, increasing the efficiency of mining. That is why waste heat recovery from compressor stations can increase the efficiency of compressed air production, as well as that of mining overall.

In this paper cogeneration and heat pump technologies have been investigated to recover waste heat from air compressor stations. Since the temperature of compressed air after compressor stages can reach 110–140 °C, it would be interesting to know how much electricity can be generated, using cogeneration technology based on the organic Rankine cycle (ORC). As the temperature of the cooling water leaving the compressor cooling system can reach 30–35 °C, it is possible to introduce heat pumps that could meet the mines' needs for hot water.

The main objective of this study is to analyse and compare two technologies that can be implemented for waste heat recovery from mining compressor stations: cogeneration technology based on the ORC and heat pump technology.

Cogeneration technology

In this section the analysis of cogeneration technology implementation for waste heat recovery from a mining turbo-compressor is carried out. A combined power and heating cycle is proposed, to produce both electrical and thermal power by utilizing waste heat from a turbo-compressor K–250–61–4. The main objective of the analysis is to determine the conditions under which the maximum power output is reached.

Turbo-compressors K–250 and K–500 are the most wide-used compressors in Ukrainian mines. They are similar in aerodynamic and design. These compressors are 6-stage centrifugal devices, combined in 3 sections of uncooled stages (hereafter referred to as uncooled sections). To cool down the compressed air produced, there are 2 inter-stage coolers and one after-cooler. The compressors are equipped with synchronous 1600 and 3000 kW motors, respectively.

The atmospheric air is compressed by the turbo-compressor and cooled in the air coolers. In order to increase the temperature potential of the waste heat recovered, and to keep the outlet temperature of compressed air within a required range, the implementation of new types of air coolers is suggested. The new types of proposed air coolers consist of two sections: the first section is for waste heat recovery and the second one is for the further cooling of the compressed air to the required temperature. To convert the waste heat into power, the ORC is used. There are several advantages in using ORC to recover low grade waste heat, including smaller size, environmental soundness and great flexibility. The main advantage of the ORC is its superior performance in recovering low temperature waste heat [1]. The ORC has received increasing attentions for generating power from different heat sources such as geothermal, solar, waste heat of internal combustion engines and gas turbines [2–5].

Nomenclature

Acronyms

AC	after-cooler
C	condenser
CHP	compressor–heat pump
Ev	evaporator
G	generator
HP	heat pump
HWSS	hot water supply system
IC	inter-stage cooler
P	pump
PC	heat pump's compressor
T	turbine
Th	throttle valve
WH	water heater

Greeks

ΔT	temperature difference (K)
η_{gen}	electric generator efficiency
$\eta_{is.p}$	pump isentropic efficiency
$\eta_{is.t}$	turbine isentropic efficiency
$\eta_{mech.t}$	turbine mechanical efficiency
$\eta_{mot.p}$	motor efficiency
η_Q	total heat recovery efficiency
η_{Q_R}	heat recovery efficiency

Symbols

c_p	isobaric specific heat (kJ/kg*K)
$E_{d.tot}$	total exergy losses (kW)
$E_{in.tot}$	total input exergy (kW)
κ	polytropic exponent
L_{comp}	specific compression work (kJ/kg)
m_{air}	air mass flow rate (kg/s)
m_{wf}	working fluid mass flow rate (kg/s)
p_{in}	uncooled section inlet pressure (Pa)
p_{out}	uncooled section outlet pressure (Pa)
t_{h2}	hot water outlet temperature (°C)
t_{int}	intercooling temperature (°C)
t_{in}	uncooled section inlet air temperature (°C)
t_{out}	uncooled section outlet air temperature (°C)
Q	total heat transferred from the compressed air (kW)
Q_{cond}	heat removed from the condenser (kW)
Q_{HW}	heat transferred to hot water supply system (kW)
Q_{in}	heat pump condenser heat load (kW)
Q_{out}	heat pump evaporator heat load (kW)
Q_R	heat transferred from the compressed air (kW)
$W_{net.el}$	net power output (kW)
W_p	pump input power (kW)
W_t	turbine output power (kW)
x	vapor extraction rate

A schematic representation of cogeneration waste heat recovery system is shown in Fig. 1.

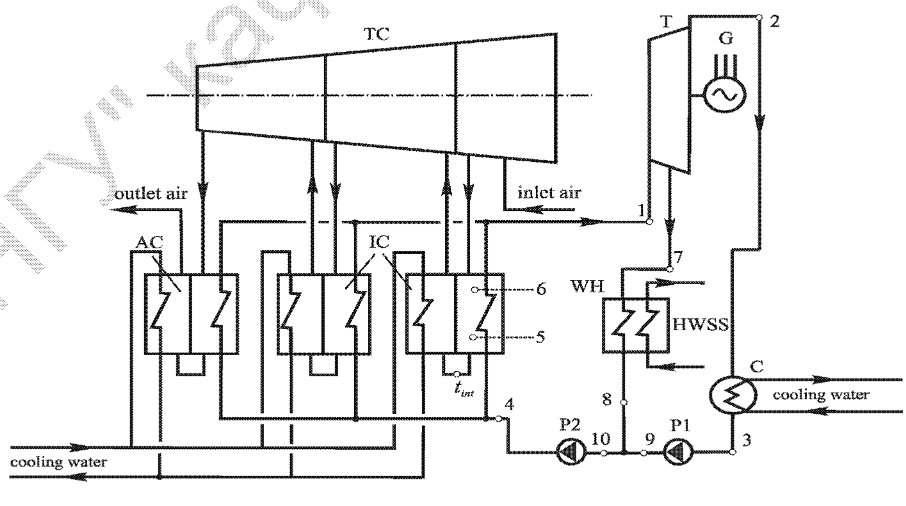

Fig. 1. Schematic diagram of cogeneration waste heat recovery system.

A certain challenge is to choose an appropriate organic working fluid which can meet all requirements and constraints imposed on the system. The working fluid should exhibit chemical stability at the operating pressure and temperature, environmental friendliness and low toxicity. In addition, it should be non-corrosive, non-flammable, be non-auto-igniting, as well as having good material compatibility [6]. Another characteristic that must be considered during the selection of the organic working fluid is its shape of the saturated curve. With respect to the slope of the saturated curve in the T–s diagram, all organic working fluids are divided into three groups: dry fluids have a positive slope; wet fluids have a negative slope; while isentropic fluids have an infinitely large slope. Basically, dry and isentropic fluids are more preferable, since they do not condense when the fluid goes through the turbine. The aim is to extend turbine blade service life by preventing their damage due to working fluid condensation. Since one of the main objectives of this study is to determine the conditions under which the highest value of power is obtained, it is desirable to choose the working fluid whose critical temperature is slightly higher than the temperature of the upper heat reservoir. In this case, the fluid evaporation enthalpy is minimal, and the working fluid flow rate is maximum, which influences the highest value of power output [7].

In view of the aforesaid, dry working fluids such as butane, pentane, isobutene, neo-pentane, R142b, R236ea and R245fa, whose critical temperatures are slightly above the outlet temperature of compressed air coming from the uncooled sections of the turbo-compressor have been chosen.

Table 1
Critical parameters and environmental properties of working fluids

	R142b	R236ea	R245fa	butane	pentane	neo-pentane	iso-butane
Critical temperature (°C)	137.1	139.3	154.05	151.98	196.55	160.6	134.67
Critical pressure (MPa)	4.07	3.502	3.64	3.796	3.37	3.1964	3.64
GWP	2400	1350	1020	0	0	0	0

Basis for GWP = 100 years, GWP of CO_2 = 1.

Fig. 2 shows the temperature–enthalpy diagram of the combined power and heating cycle. The cogeneration waste heat recovery system can run in two operating modes: a power mode, when only electrical energy is generated, and a heating mode, when more heat energy rather than electrical energy is generated.

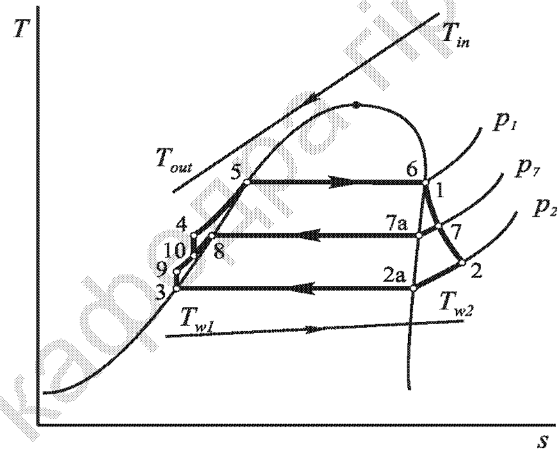

Fig. 2. Schematic T–s diagram of combined ORC.

When the proposed cogeneration system runs in the power mode, the working fluid leaves the condenser as saturated liquid (state point 3). Then, it is compressed by a liquid pump to the sub-critical pressure (state point 4). The working fluid is heated and boiled in the evaporator (process 4–5–1) until it becomes saturated vapor (state point 6) or slightly superheated vapor (state point 1). The saturated or slightly superheated vapor flows into the turbine and is expanded to the condensing pressure (state point 2). Two phase fluid passes through the condenser, where heat is removed until it becomes a saturated liquid (state point 3). In order to recover heat more completely and more flexibly, the system can run in the heating mode. In this case some amount of vapor is extracted from the turbine and directed to the water heater. The condensation temperature of the working fluid (7a) in the water heater is determined by the extraction pressure. Saturated vapor is pumped to the heat-recovery sections of the air coolers. When there is no need for hot water and there is zero vapor extraction, the system works in power mode (1–2–3–4–5–1), generating maximum electric power.

The main parameters that influence the performance of the combined cycle are turbine inlet temperature and pressure, vapor extraction rate (a portion of the vapor extracted from the turbine for the water heater), the temperature of the compressed air leaving the heat-recovery sections of the air coolers (intercooling temperature). These parameters effect the amount of waste heat recovered, power generated, heat recovery efficiency (the efficiency of waste heat recovery in the heat-recovery sections of the air coolers), total heat recovery efficiency (the efficiency of waste heat recovery in both sections of the air coolers) and exergy efficiency.

Turbo-compressor performance modeling

As initial data, an aerodynamic compressor performance curve [8], and air cooler characteristics [9], are used. To recalculate the compressor performance curve under new temperature conditions (which differ from the rated ones), the mathematical model [10] is used. This model is based on the assumption that the compression work of the uncooled sections remains equal at both rated and actual modes, if volume air flow rate also remains constant. In compliance with this assumption, pressure ratio for an i-uncooled section at a given inlet air temperature is expressed as:

$$\varepsilon_{ai} = \left[\frac{T_{in.ri}}{T_{in.ai}} \left(\varepsilon_{ri}^{\frac{k-1}{\eta_{pli}k}} - 1 \right) + 1 \right]^{\frac{\eta_{pli}k}{k-1}} \tag{1}$$

where $T_{in.ri}$ is rated absolute inlet air temperature for an i-uncooled section, $T_{in.ai}$ is actual absolute inlet air temperature, ε_{ri} is rated pressure ratio for an i-uncooled section, V_i is volumetric flow at the entrance to the i-uncooled section, η_{pli} is polytropic efficiency of an i-uncooled section, k is a politropic compression exponent.

If the compressor performance curve is given as $\varepsilon_i = f_i(V_i)$ and $\Delta T_i = f_i(V_i)$, outlet pressure and temperature for the i-uncooled section are expressed as:

$$p_{outi} = p_{ini}\varepsilon_{ai} \tag{2}$$

$$T_{outi} = T_{ini} + \Delta T_i \tag{3}$$

Energy and exergy analysis of the cogeneration waste heat recovery system

The total power of the pumps

$$\eta_{is.p} = \frac{h_{9is} - h_3}{h_9 - h_3} = \frac{h_{4is} - h_{10}}{h_4 - h_{10}} \tag{4}$$

$$W_p = m_{wf}[(h_9 - h_3)(1-x) + (h_4 - h_{10})] \tag{5}$$

The turbine power

$$\eta_{is.t} = \frac{h_1 - h_2}{h_1 - h_{2is}} \tag{6}$$

$$W_t = m_{wf}[(h_1 - h_7) + (h_7 - h_2)(1-x)] \tag{7}$$

The condenser heat rate

$$Q_{cond} = m_{wf}(h_2 - h_3)(1-x) \tag{8}$$

The total heat transferred from the compressed air

$$Q = m_{air}c_p\left(t_{out1} + t_{out2} + t_{out3} - t_{in2} - t_{in3} - t_{out}\right) \tag{9}$$

The heat transferred from the compressed air in the heat-recovery sections

$$Q_{R_i} = m_{air}c_p\left(t_{out\,i} - t_{int}\right) \qquad i = 1, 2, 3 \tag{10}$$

$$Q_R = \sum_{i=1}^{3} Q_{R_i} \tag{11}$$

The heat transferred to the hot water supply system

$$Q_{HW} = m_{wf}(h_7 - h_8) \tag{12}$$

The net power output

$$W_{net.el} = W_t \eta_{mech.t} \eta_{gen} - \frac{W_p}{\eta_{mech.p} \eta_{mot.p}} \tag{13}$$

Electrical efficiency

$$\eta_{el} = \frac{W_{net.el}}{Q_R} \tag{14}$$

Heat recovery efficiency

$$\eta_{Q_R} = \frac{W_{el} + Q_{HW}}{Q_R} \tag{15}$$

Total heat recovery efficiency

$$\eta_Q = \frac{W_{net.el} + Q_{HW}}{Q} \tag{16}$$

The exergy of the state point is expressed as:

$$E_i = m[(h_i - h_0) - T_0(s_i - s_0)] \tag{17}$$

The exergy losses of each component can be found from an exergy balance.

$$E_d = \sum_i E_{in} - \sum_i E_{out} - W \tag{18}$$

The overall exergy efficiency is defined as:

$$\eta_{ex} = 1 - \frac{E_{d.tot}}{E_{in.tot}} \tag{19}$$

For the performance simulation of the cogeneration system, assumptions are made as follows:
1) Pipe pressure drop and heat loss to the environment are negligible.
2) The condensing temperature is 30 °C. The cooling water inlet temperature is assumed to be 20 °C
3) Temperature difference in the condenser and pinch point temperature difference are considered to be 10 °C and 5 °C respectively.
4) Isentropic and mechanical efficiencies for the turbine are assumed to be 0.8 and 0.97, respectively; generator efficiency is 0.97. Isentropic and mechanical efficiencies for the pumps are assumed to be 0.7 and 0.97, respectively; motor efficiency is 0.97.

The thermodynamic properties of the working fluids were calculated by REFPROP 7.0 [11], developed by the National Institute of Standards and Technology of the United States. The ORC simulation was performed using a simulation program written by the author with MathLab.

As a result of the turbo-compressor performance simulation, it was calculated that under the given conditions the temperature of compressed air leaving the uncooled sections are 122, 132, 124 °C respectively, and its mass flow rate is 5.013 kg/s. With these values as input data for waste heat recovery system simulation, the following results have been obtained.

The influence of intercooling temperature on power output and waste heat recovery is show on the diagrams of Fig 3, 4.

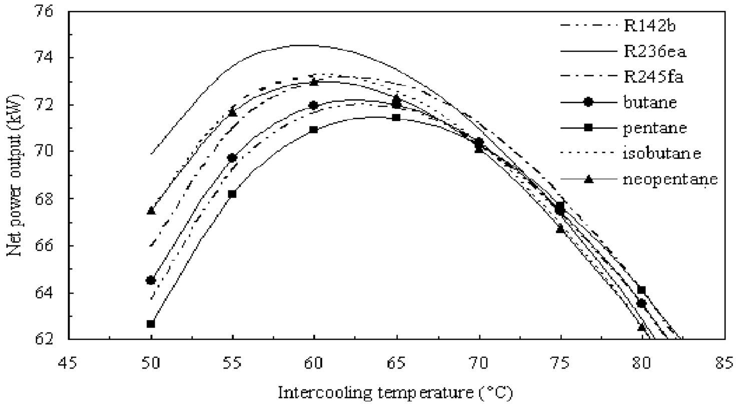

Fig. 3. Variation of net power output with various intercooling temperature.

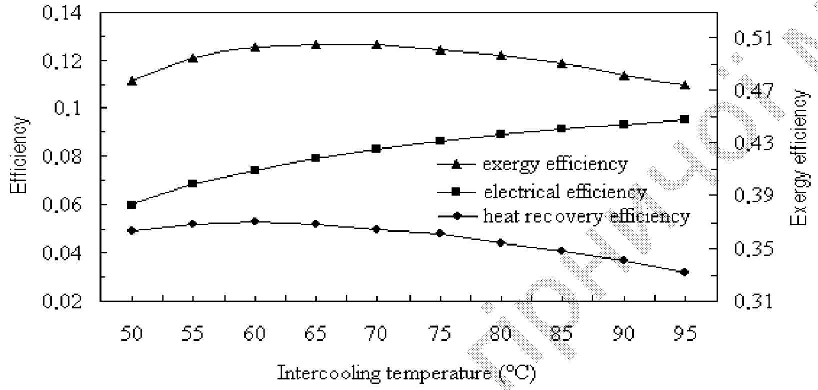

Fig. 4. Variation of different efficiencies with intercooling temperature.

Fig. 3 shows the variation of net power output with the various compressed air temperatures leaving the heat-recovery sections. The power output is the product of mass flow rate of the working fluid and the enthalpy drop. As the intercooling temperature increases, the mass flow rate of the working fluid decreases and the enthalpy of the inlet fluid to the turbine increases. As can be seen from the graph in Fig. 3, there is an optimum intercooling temperature for maximum power. The highest power is produced by R236ea at intercooling temperature of 60°C that almost coincides with maximum exergy efficiency. However, the electrical efficiency at the optimum intercooling temperature is not at maximum. The electrical efficiency increases with the temperature; at higher temperature the working fluid absorbs less heat and produces less power but has higher electrical efficiency Fig. 4.

Fig. 5 illustrates the variation of electrical efficiency and the heat recovery efficiency with various vapor extraction rate when the system runs in heating mode.

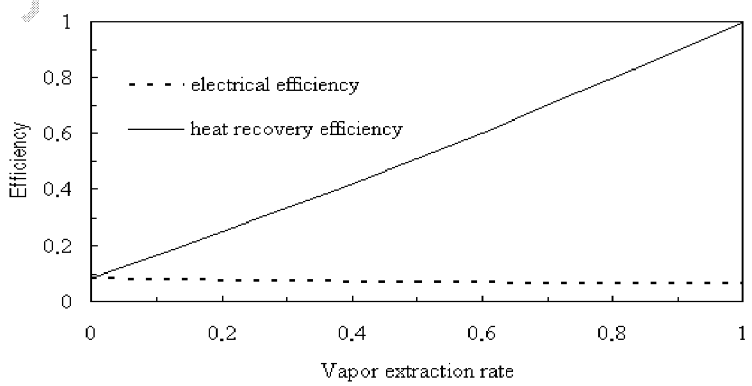

Fig. 5. Variation of electrical efficiency and heat recovery efficiency with the various vapor extraction rate.

Fig. 5 shows the variation of electrical efficiency and heat recovery efficiency with the various vapor extraction rate. It can be observed that electrical efficiency slightly decreases as vapor extraction rate increases. However, heat recovery efficiency increases dramatically due to heat usage for hot water supply.

Compared with other working fluids, R236ea produces the highest power from waste heat delivered from turbo-compressor K–250–61–4. There is an optimum intercooling temperature which lies within the range (60–65°C) for maximum power output. When the system runs in power mode, the maximum power output is 75 kW; heat recovery efficiency and total heat recovery efficiency are 0.074 and 0.053, respectively. When the system runs in heating mode, 891 kW of heat and 38 kW of electric power can be generated; heat recovery efficiency and total heat recovery efficiency are 0.99 and 0.65, respectively. R236ea shows a better performance over other working fluids, but not notably. Therefore, other factors, such as turbine size and heat exchanger surface area combined with economic modeling should be taken into account when deciding what working fluid to choose for waste heat recovery from mining turbo-compressors.

Cogeneration technology implementation for waste heat recovery from a single 1.5 MW turbo-compressor can reduce compressor energy consumption by up to 5 % and make it possible to save 540 000 kWh of electricity and 6, $42 \cdot 10^6$ kWh of heat energy annually, which is equal to about 925 tons of coal.

Heat pump technology

Since the amount of electricity that can be generated from the given heat source is significantly lower than the amount of heat energy, it would be reasonable to determine the feasibility of only heat energy production by using heat pumps. In this section the analysis of heat pump technology implementation for waste heat recovery is carried out.

Fig. 6 shows the diagram of heat pump implementation for waste heat recovery from a turbo-compressor.

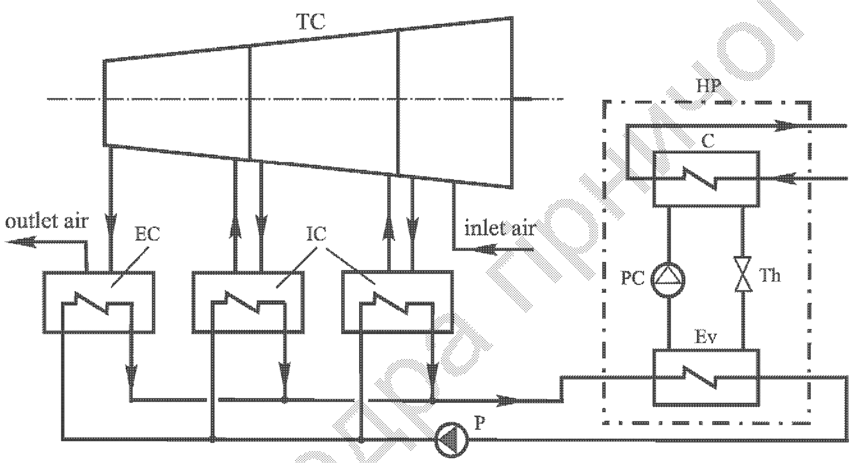

Рис. 6. Schematic diagram of heat pump implementation for waste heat recovery.

According to the diagram in Fig. 6 the water in the turbo-compressor cooling system circulates in a closed loop. It is heated in the turbo-compressor air coolers and cooled down in the heat pump evaporator. In a typical compressor cooling system the temperature of the inlet cooling water depends on the ambient air temperature. In case of heat pump technology implementation, this temperature can be considerably increased or decreased with respect to the ambient temperature.

The temperature of the inlet cooling water greatly influences the cost of compressed air production and affects the operation of the heat pump. However, the nature of this influence is ambiguous. On the one hand, the increase of cooling water inlet temperature leads to increasing COP. But on the other hand, when cooling water inlet temperature increases, compressed air outlet temperature and (as a result compression work) also increase. This results in turbo-compressor performance decrease. Therefore, one of the main objectives of this research is to determine the temperature mode of cooling water circulation loop at which the efficiency of Turbo-compressor–Heat Pump (THP) system is highest.

The production cost of 1 м^3 of compressed air $C_{air.hp}$ and benefit *B* gained from the hot water produced by the heat pump have been chosen as the efficiency index of THP system performance. While calculating the production cost of 1 м^3 of compressed air, the following expenditures have been taken into consideration: the cost of electricity consumed by the turbo-compressor $C_{el.tc}$, and by the heat pump $C_{el.hp}$, heat pump depreciation cost $C_{dc.hp}$ and the cost of heat energy C_h.

$$C_{air.hp} = \frac{C_{el.tc} + C_{el.hp} + C_{dc.hp} - C_h}{V_{year}} \quad (20)$$

where V_{year} is the amount of compressed air produced annually.

The benefit gained from heat pump technology implementation for waste heat recovery has been estimated as a difference between the cost of compressed air produced by the turbo-compressor without heat pumps $C_{air.comp}$, and after heat pump technology implementation $C_{air.hp}$.

$$B = \left(C_{air.comp} - C_{air.hp}\right)V_{year} \tag{21}$$

When the pressure on the inlet and outlet side of the compressor, inlet air temperature, cooling water flow rate and its temperature on the inlet side of the air coolers t_{w1} are given, the problem of turbo-compressor mode calculation comes in determining the cooling water outlet temperature t_{w2} and the temperature of the compressed air on the outlet side of the air coolers, the discharge temperature and pressure of the compressed air on the outlet side of the uncooled sections. Also to be considered are the amount of heat rejected in the air coolers Q_{in} (which is heat pump evaporator heat load) and power consumed by the turbo-compressor's motor W_{comp}.

The values t_{w2} and Q_{in}, obtained through compressor operation mode calculations, are input data for heat pump mode calculations and energy analysis.

The objective of the heat pump operation mode calculations when t_{w1}, t_{w2}, Q_{in}, t_{h2} are given involves the calculation of heat pump thermodynamic cycle parameters, heat pump electric power input W_{hp} and its thermal power output Q_{out} (which is heat pump condenser heat load) as well as COP.

For the CPH system performance simulation, the following assumptions are made:

1) Cooling water inlet temperatures t_{w1} varied between 5 and 30°C.
2) Atmospheric pressure is assumed to be 0,1 MPa, and ambient temperature is 15°C.
3) Discharge absolute pressure is 0.9 MPa.
4) Cooling water mass flow rate is considered to be 11.0 kg/s.
5) Hot water outlet temperature is considered to be 50°C.
6) Cost of heat energy is assumed to be 0.095 UAH/MJ; heat pump capital costs are considered to be 820$/kW.
7) Temperature differences in the evaporator and in the condenser are assumed to be 5°C.
8) Isentropic, mechanical and motor efficiencies for the heat pump are assumed to be 0.75, 0.97 and 0.94 respectively.
9) All heat exchangers in the cycle (evaporator and condenser) are considered to be shell-and-tube counter flow heat exchangers.
10) As working fluid, refrigerant R134a is used.

The cooling system of a turbo-compressor K–250–61–5 has been chosen as a source of low-grade heat for the heat pump.

The influence of the cooling water temperature t_{w1} on the energy and cost parameters of THP system performance is shown on Fig. 7–9. Twenty-four-hours mode of operation has been chosen for the heat pump.

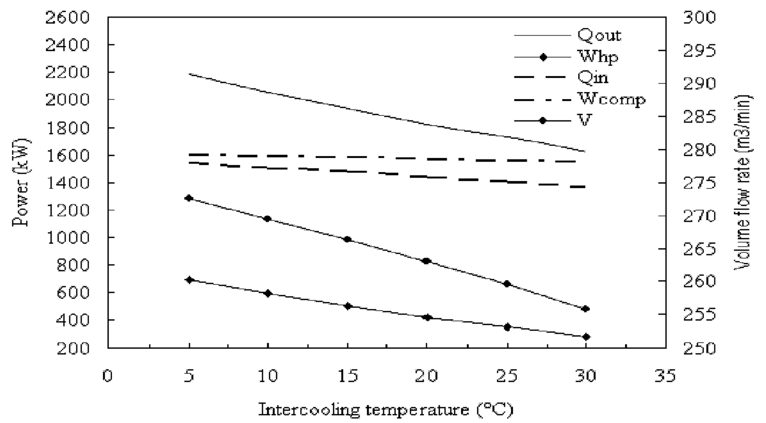

Fig. 7. Influence of cooling water inlet temperature on energy parameters of THP system.

Fig. 7 shows the influence of t_{w1} on compressor performance and energy parameters of the system, such as: condenser heat load Q_{out}, evaporator heat load Q_{in}, heat pump W_{hp} and turbo-compressor electric power consumption W_{comp}. From these results, it is evident that as t_{w1} decreases, all energy parameters increase. The increase of evaporator heat load Q_{in} can be explained by a better cooling of the compressed air and an increase in compressor performance. This, in turn, results in an increase in both the condenser heat load Q_{out} and electric power W_{hp} consumption. The increase of W_{hp} is also due to the decrease of COP as temperature t_{w1} decreases.

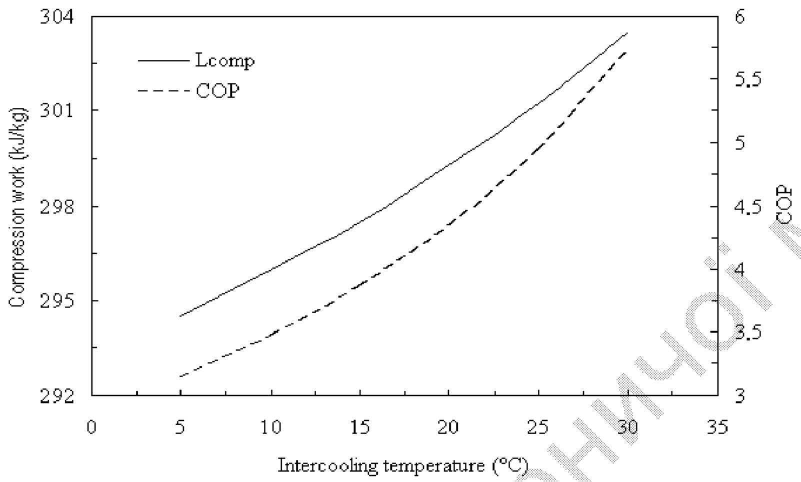

Fig. 8. Influence of cooling water inlet temperature on COP and compression work.

Fig. 8 illustrates that as t_{w1} increases, COP also increases. It is shown that at the point when temperature t_{w1} is 30°C and hot water outlet temperature is 50°C, COP can reach 5.7 which is considered to be a promising value. Fig. 7–8 show that in spite of the fact that as t_{w1} decreases, both compressor performance V and as a result electric power consumed by the turbo-compressor W_{comp} increase, specific compression work L_{comp} decreases. For a given turbo-compressor, the amount of waste heat recovered amounts to 1400–1550 kW. Heat pump heating capacity is 1600–2000 kW.

Fig. 9 shows the influence of t_{w1} on some economic parameters such as production cost of 1 m^3 of compressed air $C_{air.hp}$ and benefit B gained from the thermal energy produced.

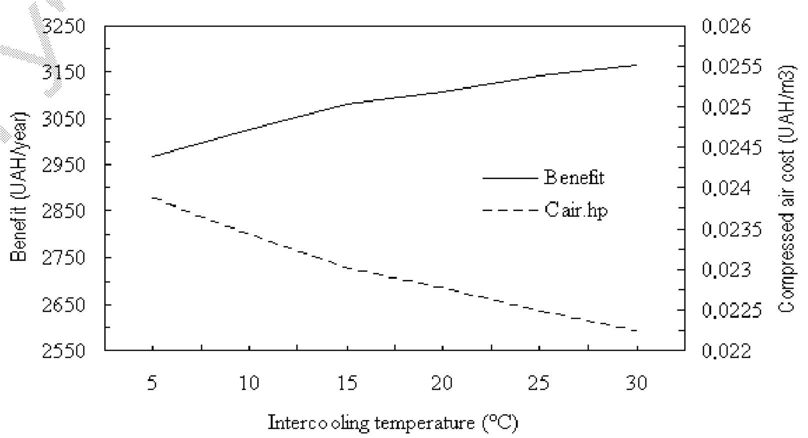

Рис. 9. Influence of cooling water inlet temperature on compressed air production cost and benefit gained from thermal energy generation.

The results show that as t_{w1} increases, the production cost of 1 m^3 of compressed air $C_{air.hp}$ decreases and benefit B gained from thermal energy increases. From these results, it is evident that turbo-compressor specific compression work increases as t_{w1} increases. However, the economic benefit increases, which contributes to a decrease in production cost of 1 m^3 of compressed air. It should be stressed that the increase of t_{w1} and as a result COP, can result in a decrease in turbo-compressor performance that can influence the normal work of machinery that utilize compressed air.

Conclusions

Cogeneration technology implementation for waste heat recovery from a single 1.5 MW turbo-compressor can reduce compressor energy consumption by up to 5 %. It makes it possible to save 540 000 kWh of electricity and 6, $42*10^6$ kWh of heat energy annually, which is equal to about 925 tons of coal. In comparison, heat pump technology implementation can generate 1600–2000 kW of thermal power, which is equal to 1660 tons of coal. In addition, the production costs of compressed air can be almost halved.

The results of THP system simulation show that if compressed air and hot water are considered to have equal significance, certain recommendations can be made. It can be recommended to cool down the compressed air at a slightly higher temperature, when cooling water inlet temperature is 30–35°C just like in a typical cooling system. However, since the main aim of a compressor is to produce compressed air, preference should be given to those operation modes of the system under which the cooling water inlet temperature would not exceed the temperature specified for typical turbo-compressor cooling systems. In this case, however, an economic benefit gained from heat pump technology implementation for waste heat recovery from turbo-compressors still remains considerable.

Another important advantage of the technical solution proposed, is a closed circuit of the cooling water system. This prevents heat exchange surfaces of air coolers from fouling and contributes to an increase in interrepair maintenance period of the air coolers, as well as less spending on repairs. Moreover, waste heat recovery from compressor stations leads to the reduction of boiler plant emissions while burning fossil fuel to meet the mine's needs for heating and hot water. In addition, environmental heat pollution is also reduced significantly.

Although heat pump technology implementation for waste heat recovery from turbo-compressors is more beneficial than cogeneration in terms of coal saved, cogeneration technology is more flexible. It makes it possible to generate both heat and electricity according to the mine's needs. For instance, in summer time when demand for heat energy is low, the cogeneration waste heat recovery system can work in power mode. In winter, when demand for heat energy increases, the cogeneration system can run in heating mode, generating mostly heat energy.

References

1) Tamamoto T, Furuhata T, Arai N, Mori K. Design and testing of the organic Rankine cycle. Energy 2001;26:239-51.
2) H.D. Madhawa Hettiarachchi, Mihajlo Golubovic, William M. Worek, Yasuyuki Ikegami Optimum design criteria for n Organic Rankine cycle using low-temperature geothermal heat sources. Energy 2007;32(9):1698-1706
3) Agustín M. Delgado-Torres, Lourdes García-Rodríguez. Analysis and optimization of the low-temperature solar organic Rankine cycle (ORC). Energy Conversion and Management 2010;51(12):2846-2856.
4) IacocpoVaja, Agostino Gambarotta. Internal combustion engine (ICE) bottoming with organic Rankine cycle. Energy 2009;35(2):1084-1093.
5) Costante Invernizzi, Paolo Iora, Paolo Silva. Bottoming organic Rankine cyle for micro-gas turbines. Applied Thermal Engineering 2007;27(1):100-110.
6) V. Maizza, A. Maizza. Unconventional working fluids in organic Rankine cycles for waste energy recovery systems. Applied Thermal Engineering 2001;21(3):381-390.
7) Aleksandra Borsukiewicz-Gozdur, Wladyslaw Nowak. Comparative analysis of natural and synthetic refrigerants in application to low temperature Clausius-Rankine cycle. Energy 2007;32(4): 344-352.
8) Цейтлин Ю. А., Мурзин В. А. Пневматические установки шахт. М.: Недра, 1985.
9) Берман Я.А, Маньковский О.Н., Марр Ю.Н., Рафалович А.П. Системы охлаждения компрессорных установок. Л.: Машиностроение, 1984.
10) Рис В.Ф. Центробежные компрессорные машины. Л.: Машиностроение, 1981.
11) REFPROP Version 7.0, NIST Standard Reference Database 23, 2002.