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Rafał Kowalski*, Mariusz Łaciak, Krystian Liszka*,**,
Andrii Oliinyk*,**, Piotr Paszylk***

APPLICATION OF ORC SYSTEMS AT NATURAL GAS COMPRESSION STATION

1. INTRODUCTION

Transport of each kind of fluid requires external energy for overcoming the associated friction forces. This energy is supplied in the form of pressure. The natural gas transmission system has high pressures inside the pipelines. This is caused by high pressure drops accompanying transport at larger distances. For the sake of outbalancing these losses, natural gas is compressed in gas compressor stations.

From the point of view of energy management, compressor stations are a key element as they may be responsible for even 50% of total cost of transmission [1]. The issue of lowering the cost of gas compression was undertaken in numerous publications. One of the proposed solutions lied in defining a function of minimum cost of fuel on the basis of optimum division of loads between gas compressors in the cooperating gas compression stations [2]. Another solution was proposed by Ernst et al. [3]. They analyzed the work of gas compression station under the angle of compressed gas demand in time. They concluded that the cost of compression is redundantly increased by the partially loaded machines. This means that the whole compression station is designed for much higher capacities and pressures than normally encountered in everyday practice. They proposed in their work to make the systems work at full load in short periods of time, using the storage capacity of the pipelines, and in the remaining time, use them for electrical energy production [4]. These examples refer to the optimization of cost of natural gas compression, and this paper tackles the issue of energy efficiency of the compression processes. Increasing of the energy efficiency of compression will bring

* OGP GAZ-SYSTEM S.A.
** AGH University of Science and Technology, Faculty of Drilling, Oil and Gas, Krakow, Poland

about certain savings, though may turn out to be unprofitable because of the cost of implementation of this technology.

From the point of view of managing the energy efficiency of gas compression station, the heat losses caused by the operation of machines in the object are a key problem. The most popular machines driving the gas compressors are combustion engines and gas turbines driven with the transmitted medium. As a result of natural gas combustion, hot exhaust gases are emitted to the atmosphere, but which may be treated as a potential source of energy. The use of exhaust gases for feeding local heat supply networks or generation of electrical energy will increase the degree to which exergy of the burnt natural gas has been utilized. Interestingly, as indicated by Chaczykowski et al. [5], compression stations with gas turbines are the main source of exergy losses in the transmission system, and the loss level in the analyzed objects was higher than calculated exergy losses in pipelines, caused by the friction forces.

One of the options frequently analyzed in literature is managing waste heat from various industrial installations with the use of a system based on organic Rankine cycle (ORC). In their work, Campana et al. [6] stated that apart from metallurgical, cement and glass sectors the natural gas transmission system is one of the major sectors where ORC is applicable. These authors also assessed that this technology could be used for recuperating about 1300 MW of electrical energy in the compression stations in Russia and Europe. It is worth noting that the ORC-based technology of electrical energy production is still considered to be new and imperfect, which unfavorable affects the costs of the potential investments [4].

The possibilities of generating electrical energy with the ORC systems in Jarosław II gas compression station, localized in Jarosław, Podkarpacie Province, are presented in this paper. The calculations were performed on the basis of historical data, covering the time span of one calendar year.

2. ORC TECHNOLOGY

Analogous to the steam Rankine cycle, the organic Rankine cycle also consists of such components as: evaporator, expander, condenser and pump. The basic difference lies in the use of the organic component (e.g. hydrocarbons) instead of water as a medium. Organic factors begin to boil at much lower temperatures than water in a given range of pressure values. This causes that the ORC systems can make use of low- and medium-temperature energy sources (60–400°C) for electrical energy production. As compared to SRC, ORC has many advantages, i.e. is smaller, has lower investment and exploitation costs [7].

As already mentioned, the general principle of ORC very much resembles that of SRC, except that only one heat exchanger is needed for all three stages of heating:

heating, evaporation and overheating [8]. A simple schematic of organic Rankine cycle is presented in Figure 1. The following processes take place as the medium flows through the successive parts of the system [9, 15]:

- Stage 1 to 2 can be described with the decompression process. A fluid at high pressure and high temperature (stage 1) is directed to the expander, where the pressure and temperature values drop, and part of the generated mechanical energy can be used for, e.g. generator. The heat loss is negligible and the transformation can be treated as adiabatic.
- Stage 2 to 3 can be described with the use of an isobaric and exothermal transformation, during which the working medium gives off the heat to the cooling medium in the condenser. In this way the medium resumes its liquid form and the process continues at a constant pressure.
- Stage 3 to 4 is a compression stage. Cooled working agent is compressed with a pump. The transformation is treated as adiabatic because the heat losses are negligible.
- Stage 4 to 1 can be described with the use of an isobaric and endothermal transformation. Cooled and highly pressurized working medium (state 4) is directed to evaporator, where energy is obtained from waste heat. The working medium undergoes three stages: heating, evaporation and overheating.

The course of a simple organic Rankine cycle from 1 to 4 is presented in a temperature-entropy plot in Figure 2.

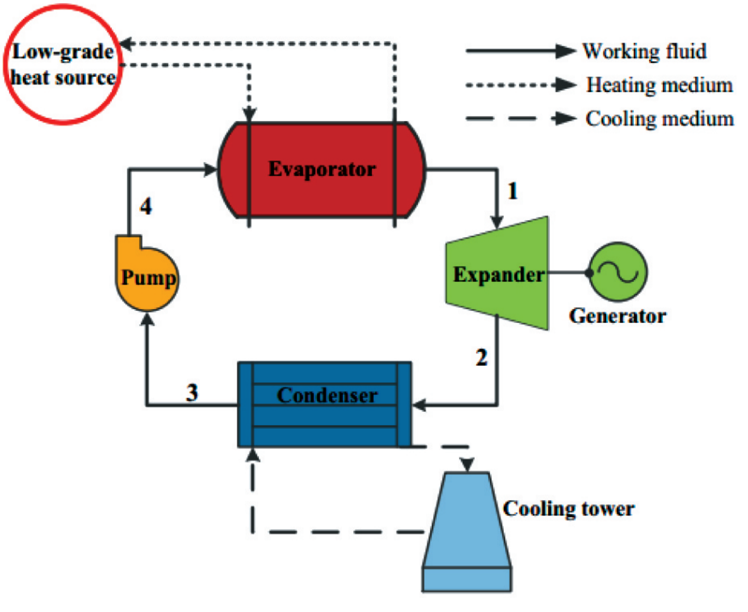


Fig. 1. Schematic of a simple ORC layout [7]

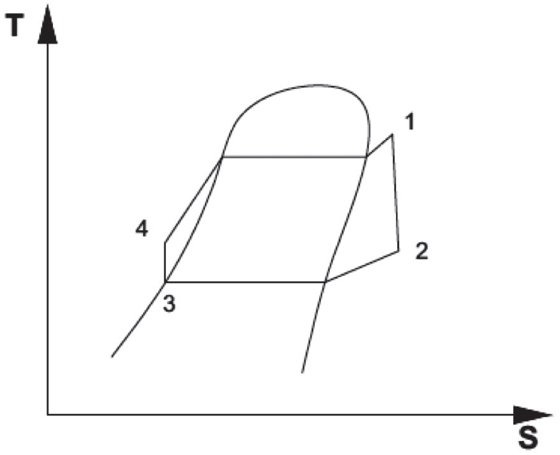


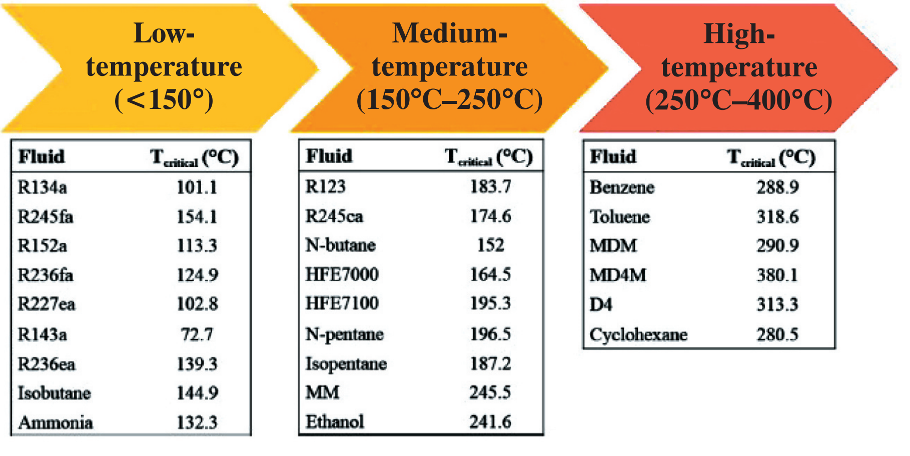
Fig. 2. T-S diagram of a simple ORC cycle [9]

One of the most important elements of designing ORC systems for a given installation is the correctly selected working medium. The optimum organic fluid will bring about higher efficiency of the system and maximize the energy obtained from waste heat. The thermophysical properties of a selected medium will have a great influence on the profitability, size of the applied devices, efficiency of expander, stability of the entire system, and environmental impact. The most important property of organic fluids is the low temperature and boiling pressure (at relatively high molar mass). Taking into account the inclination of the gaseous phase curve in the temperature vs. entropy plot (dS/dT), the analyzed organic fluids can be divided into wet, dry and isentropic. The tilt of the curve for wet fluids is negative, for isentropic is zero and for dry fluids – positive. A plot for dry fluid is presented in Figure 2. Dry fluids and isentropic fluids are preferred in simple ORC systems. This is caused by the fact that after passing through the expander, they are in the state of an overheated liquid. Thanks to this the risk of potential damage due to the condensing fluid is completely eliminated [7]. Besides, other important thermophysical properties should be also taken into account. Among the most important ones are:

- **Critical temperature.** The critical temperature of the working medium should be close to the maximum temperature of waste heat. Only in such a case the efficiency of the cycle will be on appropriate level [10].
- **Molar mass.** The drop of specific entalpy in the expander is inversely proportional to the molar mass of the decompressed medium. Hence a conclusion, that the higher is the molar mass, the smaller is the drop of entaply, and so the smaller number of decompression degrees for the expander, and so a lower is the cost of the investment [7].

– **Viscosity.** Low viscosity of the organic fluid both in the gaseous and liquid phase, is advantageous because of smaller pressure losses in the installation due to the friction forces [6].
– **Specific volume.** Organic fluids have low specific volume. The lower is the specific volume, the higher is the density and so the lower is the volumetric flow rate. Thanks to this, the dimension of the heat exchangers, expander or pumps can be reduced, which directly lowers the cost of the investment [7].
– **Condensation pressure.** After passing through the condenser, the pressure should not be lower than atmospheric pressure. This will prevent the air from penetrating the installation and disturbing of the efficiency of the cycle [9].

It should be emphasized that apart from the thermophysical properties, attention should be paid to the environmental impact of the working medium. The fluid should have smallest possible environmental impact on the ozone layer and greenhouse effect potential [6]. Obviously there is no one fluid which would be optimum for all installations. Numerous research works are devoted to matching the medium with the ORC, depending on the place of destination of the installation or character of waste heat source (low- or medium-temperature). The division of organic fluids in view of the temperature of the source of waste heat, worked out by Rahbar at al., is presented in Figure 3.



Fluid	$T_{critical}$ (°C)
R134a	101.1
R245fa	154.1
R152a	113.3
R236fa	124.9
R227ea	102.8
R143a	72.7
R236ea	139.3
Isobutane	144.9
Ammonia	132.3

Fluid	$T_{critical}$ (°C)
R123	183.7
R245ca	174.6
N-butane	152
HFE7000	164.5
HFE7100	195.3
N-pentane	196.5
Isopentane	187.2
MM	245.5
Ethanol	241.6

Fluid	$T_{critical}$ (°C)
Benzene	288.9
Toluene	318.6
MDM	290.9
MD4M	380.1
D4	313.3
Cyclohexane	280.5

Fig. 3. List of fluids obtained from literature and categorized based on waste heat source temperature [7]

Apart from the working medium, the most important element of the ORC system is the expander. It is selected based on such factors as: work conditions of the system, type of organic fluid, expected range of generated power. Generally, owing to the character of work, the expanders can be divided into two major groups. The first of them consists of flow machines (e.g. axial and radial turbines), the other one – volume machines (e.g. screw expanders, snail expanders or reciprocating expanders) [7].

The advantages and disadvantages of exemplary expanders to be used in ORC, worked by Rahbar et al., are presented in Table 1.

Table 1

Comparison of chosen ORC expanders [7]

Type	Capacity range [kW_E]	Advantages	Drawbacks
Radial inflow turbine	<500	1. High efficiency. 2. Light weight. 3. Compact layout 4. Mature manufacturability. 5. High power to weight ratio. 6. Small number of moving parts	1. High rotational speed 2. High Mach number (possibility of local choking)
Axial flow turbine	>250	1. High power capacity 2. High efficiency at large scale. 3. Flexibility of operation under partial admission for off-design	1. Tight manufacturing tolerance requirements 2. Numerous number of blades per row with high manufacturing cost. 3. Large tip clearance and frictional losses if employed for small scale
Scroll expander	1–10	1. Light weight. 2. Low rotational speed. 3. No inlet and exhaust valves which reduce noise	1. Significant lubrication to operate without wear. 2. Most kinetically complex geometry compared to other volumetric expanders. 3. Under and over expansion losses if there is mismatch between ORC and expander nominal volume ratios. 4. Friction, suction and internal leakage losses
Screw expander	15–200	1. Tolerate two-phase flow 2. Low rotational speed	1. Lubrication requirement 2. Difficult manufacturing with tight tolerances. 3. Critical sealing requirements. 4. High cost. 5. Limited internal built-in volume ratio due to length of rotor
Reciprocating piston expander	20–100	1. Tolerate two-phase flow 2. Higher pressure ratio compared to other volumetric expanders. 3. Mature manufacturability	1. Large number of moving parts with high friction and wearing. 2. Heavy weight 3. Torque impulse. 4. Requires precise timing for inlet and exhaust valves. 5. Critical balancing requirements

3. MATHEMATICAL MODEL OF ORC SYSTEM

The working out of a mathematical model was preceded by the following assumptions [11]:

- Compression and expansion of the working medium is an adiabatic process.
- For the simplicity of calculations, heat losses and pressure drops in particular elements of the system are negligible.
- The study concentrates on the efficiency of ORC, assuming that it operated in pseudo--steady conditions.
- Constant temperature and pressure of working medium on the evaporator outlet were assumed. Thus, assuming the constant compression on the pump, one can determine the temperature of organic fluid on the entry to the evaporator.

As already mentioned, the process of heating of the working medium can be divided into three stages: heating, evaporation and overheating. The pinch point temperature, being a difference between waste heat temperature on the evaporator outlet and temperature of working medium at the entry to the evaporator, is an important element of this stage. Its value usually stays within the range of 3–7 K, as the efficiency of the heat exchange process at that time is highest at that time. One should remember that the lower is the pinch point temperature, the bigger is the size of the heat exchanger, and so the waste heat resistance. Assuming the value of the pinch point temperature and the waste heat drop in the heat exchanger, the heat stream, which is transmitted to the working medium, is calculated with equation (1) [13]. Knowing the stream and the temperature of organic fluid at the entry to the exchanger, one may calculate the temperature on the outlet of the exchanger (state 1).

$$Q_{4-1} = m_h \cdot (h_{in} - h_{out}) = m_f \cdot (h_1 - h_4) \tag{1}$$

where:

m_h – mass stream of waste heat [kg/s],
h_{in} – specific entalpy of waste heat at evaporator entry [J/kg],
h_{out} – specific entalpy of waste heat at evaporator outlet [J/kg],
m_f – mass stream of working medium [kg/s],
h_4 – specific entalpy of working medium on the evaporator entry [J/kg],
h_1 – specific entalpy of working medium on evaporator outlet [J/kg].

The pressure and temperature of the working medium after expanding can be calculated on the assumption of adiabatic expansion and drop of pressure. The knowledge of these data allows for determining the entalpy of organic fluid at the expander outlet (state 2). The value of entalpy is needed for calculating power generated in the expander.

The calculation can be based on equation (2) [9]. Despite the fact that the process is treated as an isentropic transformation, the efficiency of the device should be accounted for in the calculations [9, 14].

$$W_t = m_f \cdot (h_1 - h_2) \cdot \eta_t \tag{2}$$

where:

- m_f – mass stream of working medium [kg/s],
- h_1 – specific entalpy of working medium at expander entry [J/kg],
- h_2 – specific entalpy of working medium at expander outlet [J/kg],
- η_t – efficiency of expander [–].

Potential power of generator producing electrical energy can be calculated with equation (3):

$$P = W_t \cdot \eta_g \tag{3}$$

where η_g – efficiency of generator [–].

The isobaric process of heat flow from the working medium to the cooling medium can be divided into three phases (analogous to heating): cooling, condensation, cooling up. Steady temperature of organic fluid after passing through the evaporator was assumed. In reality, such a solution would be possible by, e.g. applying regulation of stream of the cooling medium. The parameters of the cooling system can be calculated on the basis of the required drop of temperature of the cooling medium, i.e.: pinch point temperature, size of the heat exchanger, mass stream of the cooling medium [11]. Heat stream directed to the cooling medium can be calculated with equation (4) [13].

$$Q_{2-3} = m_c \cdot (h_{out} - h_{in}) = m_f \cdot (h_2 - h_3) \tag{4}$$

where:

- m_c – mass stream of cooling medium [kg/s],
- h_{out} – specific entalpy of cooling medium at evaporator outlet [J/kg],
- h_{in} – specific entalpy of cooling medium at evaporator entry [J/kg],
- m_f – mass stream of working medium [kg/s],
- h_2 – specific entalpy of working medium at evaporator entry [J/kg],
- h_3 – specific entalpy of working medium at evaporator outlet [J/kg].

Assuming that the compression of the working medium is an adiabatic process and knowing the required pressure of organic fluid at the pump outlet, one may calculate its temperature at the pump outlet. The coefficient of volumetric expandability and adiabatic compressibility were used in calculations. The required power for the pump to increase the pressure of working medium can be calculated with equation (5):

$$W_p = \frac{m_f \cdot (h_4 - h_3)}{\eta_p} \tag{5}$$

where:

- m_f – mass stream of working medium [kg/s],
- h_4 – specific entalpy of working medium at pump outlet [J/kg],
- h_3 – specific entalpy of working medium at pump entry [J/kg],
- η_p – efficiency of pump [–].

The net power of ORC is calculated with equation (6):

$$W_{net} = P - W_p \tag{6}$$

where:

- P – electric power produced by generator [W],
- W_p – electric power supplied to pumps [W].

The thermodynamic analysis of most processes lies in calculating thermal efficiency of the system. Using the energy analysis of the process according to the first and second law of thermodynamics, we do not get information about the irreversibility of the process. The purpose of this paper is describing work parameters of ORC system, which was applied to increase the energy efficiency of natural gas compression. For this reason the exergy efficiency seems to be a better parameter for describing the simulated system. Moreover, a detailed analysis of exergy losses of each element of the system can provide information which elements of the system with their work parameters can be optimized [13]. The exergy efficiency can be calculated with equation (7) [11]:

$$\eta_{ex} = \frac{W_{net}}{m_h \cdot \left[(h_{in} - h_0) - T_0 (s_{in} - s_0) \right]} \tag{7}$$

where:

- m_h – mass stream of waste heat [kg/s],
- h_{in} – specific entalpy of waste heat at evaporator entry [k/kg],
- h_0 – specific entalpy of waste heat in ambient conditions [J/kg],
- T_0 – ambient temperature [K],
- s_{in} – specific entropy of waste heat at evaporator entry [J/(K·kg)],
- s_0 – specific entropy of waste heat in ambient conditions [J/(K·kg)].

4. PARAMETERS OF WORKING CYCLE AND PRESENTATION OF CALCULATIONS

As mentioned in the Introduction, the paper is devoted to waste heat at gas compression station Jarosław II in the Podkarpacie Province. There are five piston compressors, each of them powered by a separate combustion engine. Averaged daily rotations of each of the engines and average ambient temperature were used as entry data. The data were obtained from the archivation system and their scope covers the entire year. The calculations were performed with the use of an application REFPROP [16].

The ROC work parameters were based on the stream and temperature of waste gases generated in the course of natural gas combustion. Knowing the composition of burnt gas and model of engine powering the compressor, the composition of waste gases, their stream and ambient temperature of 298.15 K were determined on the basis of stoichiometric calculations and the documentation of the engine. Thus obtained values of stream and temperature of waste gases were modified, depending on the loading of the engine and ambient temperature on the specific day. The composition of waste gases, their stream and temperature are presented in Table 2.

Table 2

The composition of exhaust gases and their flow and temperature

Component of wet waste gases	Molar participation
Carbon dioxide	5.10
Oxygen	9.40
Nitrogen	73.55
Steam	11.95
Stream and temperature at maximum loading of engine and ambient temperature of 298.15 K	
M_h = 4.7 kg/s	
T_{IN} = 633 K	

The description of parameters started from determining the constant temperature and pressure of organic fluid after condensation. Knowing these data and assuming constant pressure in the pump, one could determine the temperature of the working medium on the heat exchanger entry. The stream of the working medium was defined in such a way that after passing through the evaporator at a pressure of about 1.7 MPa it had a temperature of about 535 K (not less than 525 K, assuming the drop of waste heat of 60 K). Constant parameters of the system are presented in Table 3. The remaining parameters vary depending on the stream of heat supplied to the system (temperature and waste gas stream).

Table 3
Constant parameters assumed for ORC

Parameter	Value/name
Applied organic fluid	Toluene
Pressure and temperature after condenser (stage 3)	709.05 kPa; 469 K
Pressure and temperature after pump (stage 4)	1 716.98 kPa; 470.37 K
Pressure after passing through evaporator (stage 1)	1 716.98 kPa
Pressure after passing through expander (stage 2)	709.05 kPa
Mass stream of organic fluid	1.5 kg/s
Isentropic efficiency of pump	0.7
Isentropic efficiency of expander	0.4
Efficiency of generator	0.8

The simulated cycle for a given measurement point is presented in Figures 4 and 5 in the form of temperature/entropy and pressure/entalpy plots. The stages 1, 2, 3 and 4 correspond to the stages described in chapter 2. A point is a moment, in which working fluid begins to change to a gaseous phase. Point b describes a place in the system, where the working fluid begins to be overheated, and point c denotes a place in which the working fluid begins to condense. The temperature, pressure, entalpy and entropy values at successive points are presented in Table 4.

Table 4
Parameters describing ORC as shown in Figures 4 and 5

Stream of waste gases: 3.66 kg/s Temperature of waste gases at exchanger entry: 628.75 K Exergy efficiency of cycle: 7.15%				
Point	Temperature [K]	Pressure [kPa]	Entalpy [kJ/kg]	Entropy [kJ/(kg·K)]
1	540	1 716.98	591.81	1.22
2	485	709.05	510.72	1.12
c	470	709.05	481.72	1.06
3	469	709.05	185.60	0.44
4	470.37	1 716.98	188.94	0.44
a	525	1 716.98	326.14	0.71
b	525	1 716.98	556.77	1.15

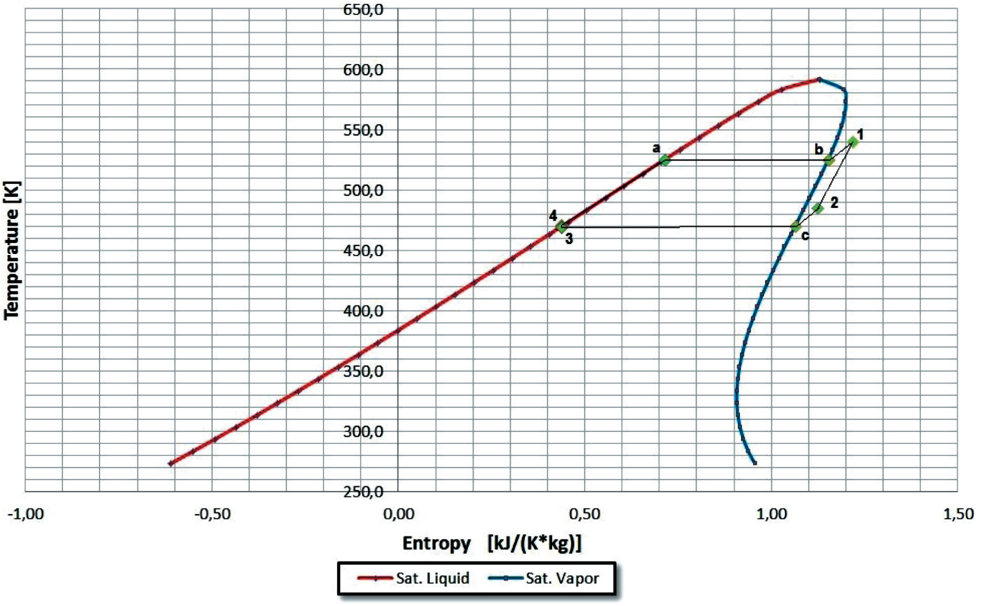


Fig. 4. Temperature-Entropy diagram of Organic Rankine Cycle represented by toluene

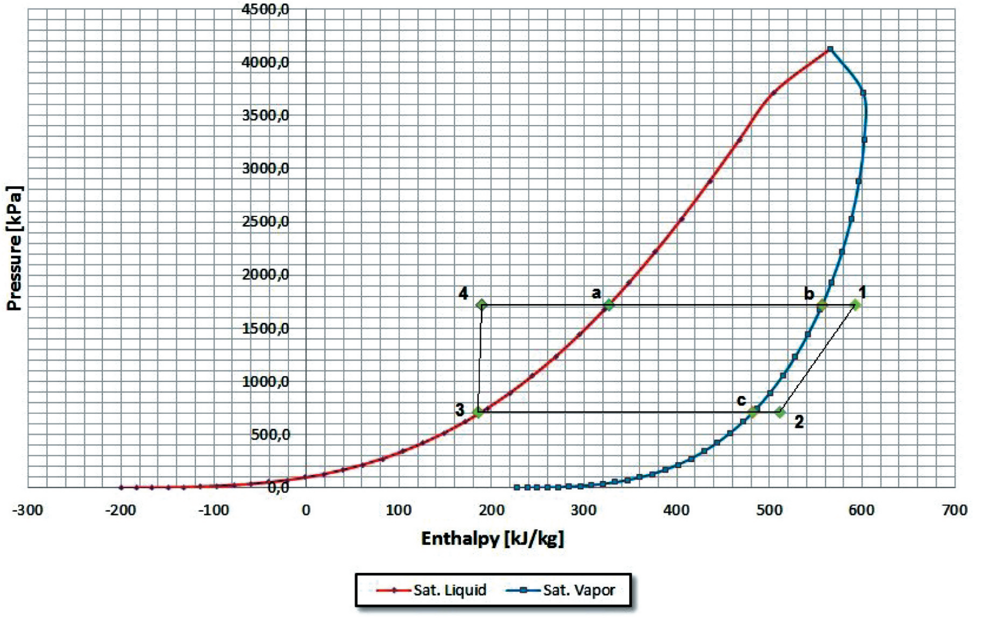


Fig. 5. Pressure-Enthalpy diagram of Organic Rankine Cycle represented by toluene

Basing on the model described in chapter 3, the power of ORC generating electrical energy for each machine was calculated. The net power (see equation (6)) and exergy efficiency of ORC are presented in Table 5. This table shows also the number of days over the year, when the system can operate. One of requirement for operating of the ORC system is work of combustion engine which is driving the compressor – only then the waste heat is produced.

Table 5

Calculation results of the net electric power output of the ORC

Engine No.	Minimum net power value of ORC	Maximum net power value of ORC	Average net power value of ORC	Number of days in the year when the system can operate	Average exergy capacity
	W_{net_min} [kW]	W_{net_max} [kW]	W_{net_av} [kW]	n [–]	η_{ex} [%]
1	31.85	33.67	31.89	168	7.11
2	28.61	33.81	32.04	168	7.08
3	28.68	33.71	31.96	142	7.07
4	28.59	33.65	31.80	108	7.06
5	29.05	33.71	32.05	154	7.04

The analysis of this table reveals that work parameters of the simulated ORC system are highly stable for each of the engines. The ambient temperature has small impact on the waste gases temperature and relatively stable number of rotations of engine while it works. The differences are observed in the number of days over the year when the system is operational. The cumulative net values of electric power produced by the system in successive days of the year are presented in Figure 6. The participation of electric power generated from waste heat of the engines is also accounted for.

The correct interpretation of above plot can be done on the assumption that machines operate at the same time during the day. The rotations of the engine (entry data) are averaged for a given day. Even if we have an averaged number of rotations of an engine for all machines, there is no certainty that they were operating simultaneously. With this assumption, the analysis of the plot reveals that the guaranteed net value totals to about 32 kW for the whole year (one machine). The power totals to about 65 kW during operation of two machines for over half a year. When three machines are operating simultaneously, the cumulative electric power from ORC would reach 90 to 100 kW.

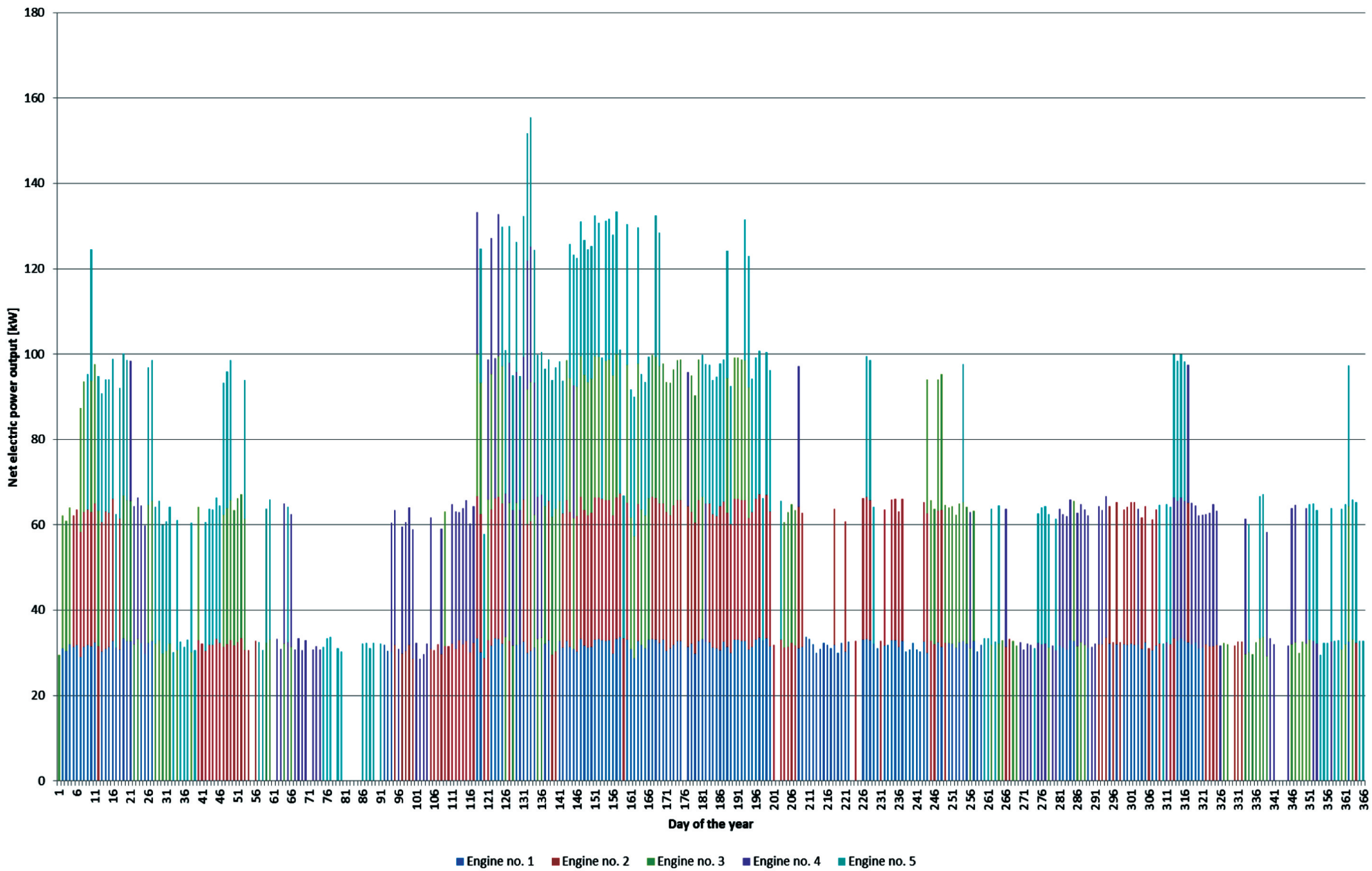


Fig. 6. Presentation of net electric power output from waste heat of each engine

The other question is the amount of electrical energy which could be generated by such systems. This is connected with the time of operation of compressor aggregates during a day and would be most important when performing economic analysis of such systems.

For the sake of showing the influence of entry data on the operation of ORC system, a dependence of exergy efficiency on ambient temperature and number of engine rotations is presented in Figures 7 and 8 (values in the plot calculated for engine No. 1).

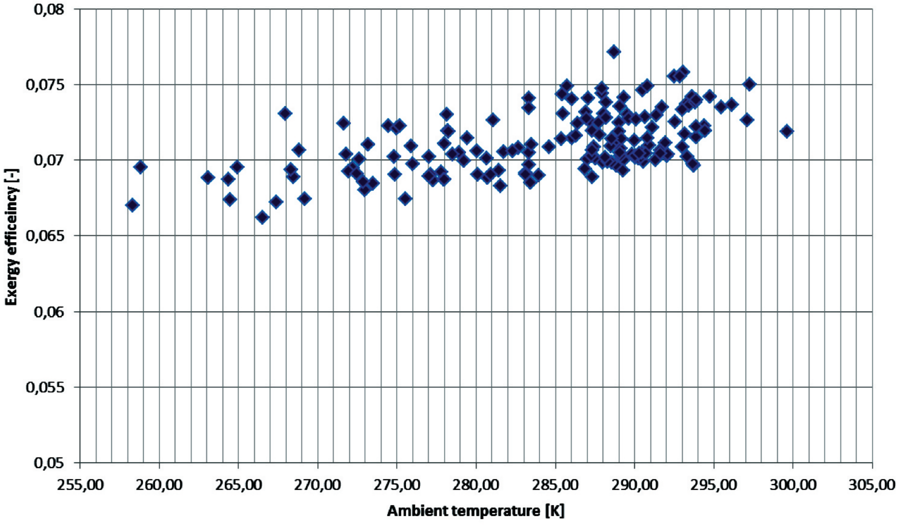


Fig. 7. The influence of ambient temperature on exergy efficiency of a cycle

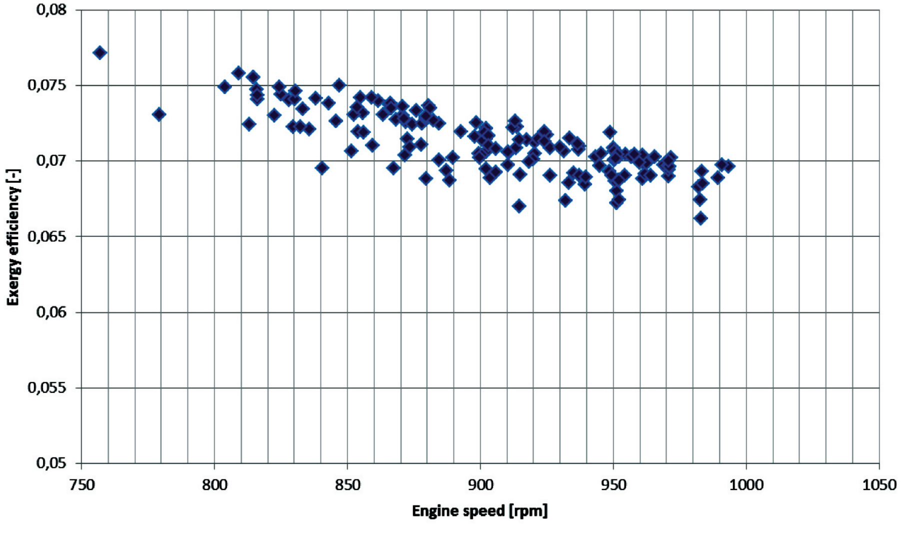


Fig. 8. The influence of engine speed on exergy efficiency of a cycle

The increasing efficiency of the cycle with the growing ambient temperature is visualized in Figure 7. This is connected with the fact that the increase of ambient temperature is associated with the growth of temperature of waste gases. On the assumption of a constant drop of waste heat in the evaporator, the higher temperature of the waste heat at the entrance to the heat exchanger, the greater the value of the energy transferred to the working fluid.

The analysis of figure 8 reveals that the higher is the speed of the engine, the lower is the exergy efficiency. This is caused by the proportionality of the waste gases stream and the engine speed. The bigger is the stream of waste gases, the higher is the exergy of the analyzed stream. On the other hand, a bigger stream of waste gases does not have any significant impact on the increased amount of heat transmitted to the organic fluid. In other words, an increased amount of heat transmitted to the working fluid does not compensate for the total growth of exergy which can be recuperated from waste gases.

5. CONCLUSIONS

The increase of energy efficiency of various industrial processes has grown in importance on the company's and also country's economy scale. The same is also true for the operators of gas transmission systems. The analysis of the performed calculations reveals that the heat stream obtained from waste gases ranged from 220 to 260 kW when each of the engines was operational. Obviously these values can change depending on the type of the engine, character and conditions of work of a given station, nonetheless the solution of the problem managing waste heat should be an important issue for gas industry in the coming years. The ORC system was proposed for the production of eclectic energy. On the assumption of pseudo-steady operation of the system, the average annual net electric power of about 32 kW could be obtained when feeding the ORC system with waste heat from a compressor engine.

One of the bigger problems encountered while managing the heat at the gas compression stations is the unstable and unpredictable characteristic of operation of particular engines. The conducted analyses reveal that in the case of simulated ORC systems the available electric power would be about 32 kW annually (at least one machine works). Assuming a simultaneous operation of three machines, the net electric power could even reach 90 to 100 kW. The obtained exergy efficiency of the cycle is very low and on average totals to about 7.06%. This mainly stems from the assumed efficiency of the pump, expander and generator. No detailed analysis of the selection of the elements of the system was made as the authors focused on highlighting the places, where energy efficiency of the compression can be increased and the associated technology.

Another important issue which has not been discussed in this paper is the quantity of electrical energy which could be generated by ORC systems. This value depends on time of work of the compressor engine, as only then the source of waste heat, i.e. waste gases, appears. Apart from the market energy prices, it is the amount of annually generated electrical energy which has the main influence on the profitability of the planned investment. Judging from the available publications, the ORC projects frequently fail to be implemented just for economic reasons. This should be a challenge for the scientists, who are encouraged to search for solutions increasing the exergy efficiency of the system.

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