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ORGANIC RANKINE CYCLE FOR RESIDUAL HEAT TO POWER CONVERSION IN NATURAL GAS COMPRESSOR STATION. PART II: PLANT SIMULATION AND OPTIMISATION STUDY

ORGANICZNY OBIEG RANKINA DO PRODUKCJI ENERGII ELEKTRYCZNEJ Z CIEPŁA ODPADOWEGO W TŁOCZNI GAZU. CZĘŚĆ II: SYMULACJA I OPTYMALIZACJA INSTALACJI

After having described the models for the organic Rankine cycle (ORC) equipment in the first part of this paper, this second part provides an example that demonstrates the performance of different ORC systems in the energy recovery application in a gas compressor station. The application shows certain specific characteristics, i.e. relatively large scale of the system, high exhaust gas temperature, low ambient temperature operation, and incorporation of an air-cooled condenser, as an effect of the localization in a compressor station plant. Screening of 17 organic fluids, mostly alkanes, was carried out and resulted in a selection of best performing fluids for each cycle configuration, among which benzene, acetone and heptane showed highest energy recovery potential in supercritical cycles, while benzene, toluene and cyclohexane in subcritical cycles. Calculation results indicate that a maximum of 10.4 MW of shaft power can be obtained from the exhaust gases of a 25 MW compressor driver by the use of benzene as a working fluid in the supercritical cycle with heat recuperation. In relation to the particular transmission system analysed in the study, it appears that the regenerative subcritical cycle with toluene as a working fluid presents the best thermodynamic characteristics, however, require some attention insofar as operational conditions are concerned.

Keywords: Gas pipeline, Compressor station, Waste heat, Energy recovery, Optimisation algorithms

W pierwszej części artykułu przedstawiono modele matematyczne elementów siłowni ORC, natomiast niniejsza, druga część artykułu, zawiera przykład ilustrujący efektywność różnych systemów ORC w instalacji odzysku ciepła w stacji przetłocznej. W wyniku lokalizacji w stacji przetłocznej, instalacja wyróżnia się pewnymi charakterystycznymi cechami, takimi jak stosunkowo duża wielkość systemu, praca przy niskich temperaturach otoczenia, zastosowanie skraplacza chłodzonego powietrzem. Obliczenia optymalizacyjne przeprowadzone dla 17 płynów pozwoliły na wybór odpowiednich czynników roboczych dla każdej konfiguracji obiegu, wśród których benzen, aceton i heptan wykazały najwyższą

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możliwość odzysku energii w obiegach nadkrytycznych, podczas gdy benzen, toluen i cykloheksan w obiegach podkrytycznych. Wyniki obliczeń pokazują, że dysponując strumieniem spalin z turbiny gazowej o mocy 25 MW, za pomocą benzenu jako czynnika roboczego, można uzyskać w obiegu nadkrytycznym z regeneracją ciepła maksymalną moc mechaniczną na wale turbiny wynoszącą 10,4 MW.W odniesieniu do systemu przesyłowego analizowanego w tej pracy najlepszym wariantem siłowni ORC z punktu widzenia charakterystyki termodynamicznej jest obieg nadkrytyczny z regeneracją ciepła przy zastosowaniu toluenu jako czynnika roboczego, jednak jego stosowanie mogłoby powodować problemy eksploatacyjne podczas użytkowania instalacji.

Słowa kluczowe: gazociąg, tłocznia gazu, ciepło odpadowe, odzysk energii, optymalizacja

Notation

c_p – specific heat,
$\dot{m}$ – mass flow rate,
p – pressure,
$\dot{Q}$ – heat rate,
T – temperature,
v – specific volume,
$\dot{W}$ – mechanical power.

Greek symbols

η – efficiency,

Subscripts

A – air,
C – condenser,
c – critical,
E – evaporator,
f – saturated liquid,
P – pump,
S – superheating,
T – turbine,

Acronyms

EG – exhaust gas,
HTF – heat transfer fluid,
PP – pinch point,
WF – working fluid

1. Introduction

A concise review of waste-heat-driven ORCs for gas turbines along with the specifics of the systems to be installed in compressor stations have been presented in the first part of this paper (Chaczykowski, 2016). The present investigation considers the technical viability of an ORC application in a gas compressor station on the basis of Yamal-Europe pipeline compressor data.

The main features of the possible bottoming cycle are determined, i.e. high-efficiency working fluids, plant layout, and possible nominal design parameters of the cycle. The studies are preliminary, therefore the alternative options for this application are compared on a thermodynamic basis, while economical considerations are left out.

The gas transmission system considered is a 680 km long Polish section of the Yamal-Europe pipeline with a nominal capacity to handle 90.1 million standard cubic meter gas per day. The gas is compressed in 5 compressor stations equipped with 3-4 parallel compressors each powered by a gas turbine rated at 25.4 MW. One is discussing the possibilities of upgrading these installations with ORC based power plants, and the performance of basic ORC configuration and the relevant alternatives need to be investigated on the basis of power generation potential. This locally produced electricity could be used internally at the site for powering induced draft natural gas coolers (Chaczykowski, 2012) and the remaining part of it could be sold in the open market into the electric grid.

2. ORC working fluid considerations

Appropriate selection of organic working fluid is critical to achieve high conversion efficiencies. Generally, dry and isentropic fluids, i.e. the fluids with non-negative slope of saturated vapor curve in T-s diagram are better for ORC systems. They present no risk of condensation when the fluid goes through a turbine, which otherwise could damage the blades. The application and performance of different working fluids have been studied by many researches, and some of the research studies attempt at providing indicators for the selection. Badr et al. (1985) identified thermal stability, availability, cost and safety requirements as primary factors to be satisfied in the screening process. From the thermodynamic point of view, ORC process requires a working fluid characterized by a low enthalpy of vaporization which allows the exhaust gas to be cooled to a significantly lower temperature compared to steam process (Larjola, 1995). Furthermore, Maizza and Maizza, (1996) considered high latent heat and a near vertical saturated liquid line, related to the low liquid specific heat, as a desirable working fluid characteristics. Such a fluid receives more energy from the source during the change of phase and allows us to maintain the Carnot-like shape of the cycle, without the need for the regenerative heating.

The possibility of improvement of the performance of ORC systems by adopting organic-fluid zeotropic mixtures as a working media has been investigated (Angelino & Colonna Di Paliano, 1998; Borsukiewicz-Gozdur & Nowak, 2007). It has been shown that for different compositions of the mixtures, better matching of the working-fluid and the source heat capacities may be achieved as an effect of non-isothermal phase change in the evaporator. Another option for better capacitance rate matching is the application of supercritical cycles. Compared with subcritical ORCs, supercritical cycles more efficiently utilize low-grade waste heat sources with gradient temperature (Chen et al., 2006; Karellas & Schuster, 2008).

Hung (2001) and Mago et al. (2007) presented a second-law analysis of the performance of ORC using different working fluids. The results show that dry working fluids must be operated at saturated conditions, since superheating dry organic fluids increases total exergy loss of the system, the cycle thermal efficiency remaining approximately constant. In fact, it is the rate at which the constant pressure lines in enthalpy-entropy diagram diverge that determines the impact of superheating. Constant pressure lines for some dry working fluids are nearly parallel, leading to decreased, unchanged or marginally improved cycle efficiencies in the superheat region (Hung et al., 1997; Chen et al., 2010; Chaczykowski, 2012).

Liu et al. (2004) analysed the influence of the critical temperature on ORC system efficiency. It has been concluded that high critical temperature of the working fluid allows higher temperature heat sources to be used without superheat and therefore at greater efficiency. This conclusion was confirmed in the study by Rayegan and Tao (2011). Furthermore, the authors identified vapor expansion ratio across the turbine as one of the important factors influencing the operation of an ORC system. High vapour expansion ratios together with small specific work result in greater number of stages and large turbine size. Working fluids with high density allow for the reduction of turbine size.

Working fluid selection in recent simulation studies on ORC applications is usually conducted through thermodynamic properties database mining. By screening of the large sets of fluids, the arbitrariness in the selection is limited. From a rather different perspective, Papadopouls et al. (2010) presented a general systematic design procedure of estimating working fluid performance potential based on computer aided molecular design. This approach goes one step further by assigning fluid performance measures divided into four categories: thermodynamic, environmental, safety and process related, which are then used as an evaluation criteria in the procedure of subsequent selection of molecules. It has been concluded, that the methodology leads to identification of both conventional and novel fluid molecular structures and avoids the consideration of non-optimal choices.

3. Simulation data and assumptions

3.1. Exhaust heat source

The gas turbines, rated mechanical drive of 25 MW, have been installed in five compressor stations of the Yamal-Europe pipeline to facilitate relatively stable compressor loads all year round. According to official data, the net outlet power of the unit is 25.40 MW, while the temperature and the mass flow rate of the hot exhaust gases are 543°C, and 80.4 kg/s, respectively. The quantities indicate relatively good quality residual heat parameters, which are confirmed in practice by pipeline operator, so the turbines seem to be well suitable for bottoming cycle applications.

Using the above data, and given the exhaust gas composition: N_2 75.4%, O_2 13.6%, CO_2 3.4%, H_2O 6.6%, Ar 1.0% (mole fractions), the rejected waste heat, compared with the atmospheric temperature of 15°C, may be estimated as equal to 46.2 MW. The figure is for one compressor, which is one-third or half of total plant, depending on the compressor station under consideration, since one unit in every station is on a stand-by mode (Chaczykowski, 2012). Needless to say, due to lacking any heat recovery system, this significant amount of exergy is now lost.

3.2. Heat transfer processes

The net power output is a function of working fluid flow rate, heat transfer fluid flow rate and air flow rate. Therefore, the consideration of heat transfer processes in the heat recovery oil heater, working fluid boiler and air condenser is necessary to fulfil ORC calculations.

Synthetic fluid, which has a temperature range (–35÷330)°C has been selected for the analysis, since the outdoor design temperature for winter conditions in the area where the compressor stations are located is -22°C. The correlations between enthalpy and temperature of the heat transfer fluid (Dowtherm Q) are given in (NREL, 2008).

The diagram in Fig. 1a. depicts composite curves for heat transfer fluid (HTF) and working fluid (WF) providing insight into cycle design. The maximum heat transfer fluid temperature was set to 330°C to assure chemical stability of the fluid. The minimum temperature difference between the heat transfer fluid and the working fluid at pinch point 1 (ΔT_{PP1}) was kept constant at 10°C, while the minimum temperature difference between the turbine exhaust gas (EG) and the heat transfer fluid (ΔT_{PP2}) was assumed to be 30°C.

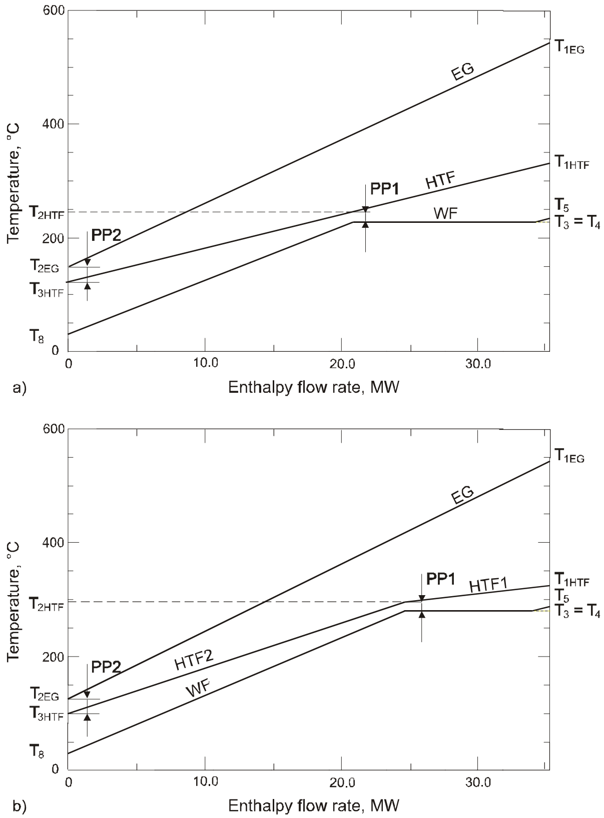

Fig. 1. Thermal matching of exhaust gas (EG), heat transfer fluid (HTF), and working fluid (WF) in ORC (regenerative configuration with recuperator): a) single HTF cycle, b) two separate HTF cycles in economizer and evaporator, PP–pinch point

In order to make a valid comparison between the various cycle configurations, each option was compared assuming the same exhaust gas rates and ambient air conditions. The system loss of 1.5% in the heat recovery oil heater was assumed, therefore the rate of heat delivered by the hot exhaust gases was reduced by 693 kW.

An adapted plant design with two separate heat carrier fluid cycles (Fig. 1b) was considered where necessary. The results showed that in case of certain working fluids, the pinch point bounds of the single heat transfer fluid cycle reduced the amount of heat delivered to the ORC system and above modification improved the overall performance of the plant.

The assumed heat sink technology was a forced draft air cooler. The air enthalpies were evaluated from the property relations at given p-T conditions. The calculations were carried out for a fixed condensing temperature of 30°C, which was consistent with an ambient temperature of 15°C. It has been assumed that the pinch point temperature difference in the condenser is 5°C

3.3. Pre-selection of working fluids

Based on the interaction between vapour pressure, process maximum pressure and higher temperature level of the cycle, the pre-selection of working fluids was performed. Possible toxicity of the fluid can be more easily resolved in the type of application considered here, thanks to already existing compressor station hazardous area zoning and open-air plant localization. The preferred working fluids should have negligible ozone depletion potentials and relatively low global warming potentials. Consequently, banned by Montreal protocol the CFC refrigerants were discarded from the analysis, and so were the HCFC refrigerants, phased out by the EC Ozone Regulation. In view of the potential future climate contribution of HFC refrigerants, HFCs were also not considered here, and the study focuses on 17 dry and isentropic organic fluids presented in Table 1. Except for acetone, the fluids are hydrocarbons, which can be considered as low-cost high-performance working fluids, compared to traditional refrigerants.

TABLE 1

Preselected working fluids for ORCs (properties data based on 30°C)

ASHRAE number	Name	Molecular weight (kg/kmol)	T_c (°C)	P_c (MPa)	Liquid c_p (kJ/kgK)	Latent heat (kJ/kg)
	Acetone	58.08	234.95	4.70	2.16	529.10
	Benzene	78.11	288.90	4.89	1.71	429.54
R-600	Butane	58.12	151.98	3.80	2.47	356.30
	Butene	56.11	146.14	4.01	2.34	354.80
	Cis-butene	56.11	162.60	4.23	2.27	387.20
	Cyclohexane	84.16	280.49	4.08	1.85	389.24
	Heptane	100.20	266.98	2.74	2.26	362.01
	Hexane	86.18	234.67	3.03	2.27	362.17
R-600a	Isobutane	58.12	134.66	3.63	2.46	323.33
	Isobutene	56.11	144.94	4.01	2.41	353.25
	Isohexane	86.18	224.55	3.04	2.26	344.00
R-601a	Isopentane	72.15	187.20	3.38	2.30	341.60
	Neopentane	72.15	160.59	3.20	2.34	299.25
R-601	Pentane	72.15	196.55	3.37	2.34	362.40
R-290	Propane	44.10	96.74	4.25	2.78	326.70
	Toluene	92.14	318.60	4.13	1.72	409.90
	Trans-butene	56.11	155.46	4.03	2.35	376.09

Among the quantities shown in Table 1, the molecular weight suggests the density of the fluid, thus the size of the facility, while the critical point suggests the possible temperature and pressure ranges of the plant. In brief, fluids with high density, low liquid specific heat and high latent heat are expected to give high turbine work output. It can be observed that acetone, benzene and toluene are promising working fluids with relatively high critical temperature, high latent heat and low liquid specific heat.

The possible operating temperature and pressure ranges of the plant are more specifically presented in Table 2, which shows working fluid melting point, auto-ignition temperature, fluid maximum temperature, maximum evaporation temperature, and the minimum pressure of the cycle (the condenser pressure). Taking into consideration nominal outdoor temperature in the area where compressor stations of the Yamal-Europe pipeline are located, working fluids with melting point temperature above –22°C would require special anti-freezing precautions. The values of the maximum temperature of the fluid were taken from REFPROP 8.0 database, and the limits should be near the point of decomposition. Thermal stability concerns, however, are the main source of uncertainty associated with the evaluation of fluid's suitability for ORC applications. The stability data published in the open literature are sparse, but the consideration of the kinetics of the working fluid decomposition can give insight into the expected decomposition rates (Schroeder & Leslie, 2010). The maximum process temperature was set to the smaller of the values of maximum allowable temperature of the working fluid and auto-ignition temperature, except in cases of benzene, propane and toluene, in which the maximum process temperature was limited to 320°C, assuming the temperature difference of 10°C at the so-called hot end of the superheater. The auto-ignition temperatures are considered here as guides, and in the final analysis the maximum process temperature should be low enough to guarantee safety. The maximum evaporation temperature corresponds with the evaporation pressure limit, which was set 0.1 MPa below critical pressure. The values in the last column of Table 2 represent minimum cycle pressures, determined by the fluid saturation condition at the condensing temperature of 30°C. These figures provide certain background on the operation-related problems caused by infiltration of gases. The condensation pressure above atmospheric pressure is preferable, since air infiltration at sub-atmospheric pressures reduces system efficiency. Consequently, high vacuum operating conditions lead to a high-maintenance system and should be avoided. For the fluids considered, the highest vacuum operating conditions has toluene, being the only fluid with minimum vapour pressure below 5 kPa. The condensation design temperature could be raised if necessary to maintain the minimum vapour pressure, but such a modification would be at the expense of a lower power output.

3.3.1. Design parameters

Under subcritical operation, the vapour pressure-temperature relation is ill-conditioned near critical pressure that makes the system unstable. Drescher and Brüggemann (2007) suggested setting the evaporation pressure limit 0.1 MPa lower than critical pressure, and this assumption was also made in the present study. Low heat exchange coefficients of the fluids in vapour phase lead to large heat exchangers. In order to reduce eventual equipment expenses, the superheating limit was set to $\Delta T_S \leq 5°C$.

Supercritical operation was also analysed, although, for the sake of limiting the safety measures, as well as to reduce material expenses, the maximum process pressure in ORC plant has been fixed at 8.4 MPa, which is the maximum operating pressure of the compressor station plant.

TABLE 2

Operational limits of the ORC components for preselected working fluids

Working fluid	Melting point (°C)	Ignition point (°C)	Maximum T (°C)	Maximum T_{eva} (°C)	Minimum P_{con} (kPa)
Acetone	–95	465	276	234	37.9
Benzene	6	498	362	287	15.9
Butane	–138	365	302	150	283.4
Butene	–185	385	252	145	344.9
Cis-butene	–139	324	252	161	250.3
Cyclohexane	7	260	427	279	16.2
Heptane	–91	285	327	264	7.8
Hexane	–95	225	327	232	25.0
Isobutane	–160	460	302	133	404.7
Isobutene	–140	465	277	143	353.9
Isohexane	–153	264	277	222	34.6
Isopentane	–160	420	227	185	109.2
Neopentane	–17	450	277	159	200.6
Pentane	–129	309	327	195	82.0
Propane	–190	450	352	95	1079.0
Toluene	–93	480	427	317	4.9
Trans-butene	–105	324	252	154	273.3

As discussed in Part 1 of this paper, the power required by the heat transfer fluid circulation pump has been taken into consideration in the calculation of the net power output of the ORC plant. It has been assumed for the purpose of this study that diathermic oil line has a total length of 300 m. The turbomachinery efficiencies were selected to be 85% and 80% for the pump and turbine, respectively, and the recuperator effectiveness was set to 80%.

The overpressure at the discharge section of the air fan was assumed 2 kPa and the compression in the air fan was assumed to be isothermal. Accordingly, for a forced draft air-cooler configuration, the figures for inlet air pressure and temperature were 0.1033 MPa and 15°C, respectively. The air components were N_2, O_2 and Ar, with mole fractions of 79%, 20% and 1%, respectively.

The ratio of resistance coefficient ζ to the squared cross-sectional area of the air flow path A^2, necessary for the determination of the cooler power consumption $\dot{W}_{P,A}$, is a function of cooler geometry. It was set to a constant value of $1.2 \cdot 10^{-7}$ m^{-4}, which is technically valid for the scale of the application considered here. Air fan efficiency was assumed 60%.

For each combination of cycle configuration and working fluid employed the genetic algorithm (GA) based search routine was executed to determine the cycle design parameters. GAs are based on the evolutionary ideas of natural selection and genetics and operate on the encoded versions of the parameters (genotypes) in the form of a string of digits (chromosome). At the beginning, the population of individuals is randomly generated – first generation, since the GA operators act on a population rather than on a single individual. Over the generations, the selection of individuals based on a performance evaluation (i.e. objective function evaluation) is carried out. In brief, the GA algorithm consists of three basic operations: reproduction, crossover and mutation. After the reproduction population is determined based on performance evaluation, the crossover operator acts on a pair of chromosomes to produce a pair of offspring chromosomes, followed by the low-probability mutation operator, which randomly modifies some characteristics of the individuals produced by the crossovers. The crossover and mutation operators are applied only if a probabilistic tests yield true, therefore a crossover and mutation rates must be given as input data.

GA based optimisation routine PIKAIA 1.2 (Charbonneau, 2002) was used in this study. The crossover probability and mutation probability were set to 85% and 20%, respectively. The population counted 200, and the GA was run for 100 generations. The saturation properties, as well as the properties at the respective p-T states of the cycles were obtained from REFPROP 8.0 database (Lemmon et al., 2007).

4. Simulation results and discussion

4.1. Basic ORC configuration

The calculation results of the shaft-power output of the basic ORC configuration under the same residual heat and ambient air conditions are presented graphically in Fig. 2. It can be observed that benzene, toluene, acetone and cyclohexane are the most promising working fluid candidates for this application in terms of maximizing the cycle power output. The correlation between latent heat, critical temperature and power output can be confirmed by these results. With regard to thermodynamic appropriateness of the fluids, benzene, toluene and cyclohexane showed the highest energy recovery potential in subcritical cycles, while benzene again, acetone and heptane in supercritical cycles. Supercritical operation appears not to increase the power output in a significant manner when the above working fluids are used. However, it has a visible effect on power output for the case of fluids with low critical temperatures, like propane and isobutene, as a result of a better matching of working-fluid and heat transfer fluid composite curves. Missing data regarding supercritical cycles of cyclohexane, hexane and toluene result from the fact that cyclohexane and hexane have the critical temperature above auto-ignition temperature, and toluene has the critical temperature merely 1.4°C below cycle maximum temperature of 320°C, which causes the supercritical cycle expansion in the two-phase conditions.

Table 3 gives in more detail the summary of simulation results for determination of working fluids which provide highest power output in basic ORC configuration. The supercritical cycle using benzene showed the highest shaft-power output of 10.31 MW, against 10.05 MW and 9.86 MW of subcritical cycles using benzene and toluene, respectively.

Regarding supercritical cycle parameters, it can be noted that the obtained solutions have a maximum temperature constraint active, therefore it can be concluded that the highest power output requires maximum temperature of the fluid at turbine inlet.

Thermal efficiencies corresponding to the maximum net power output are higher in case of supercritical cycles, compared to the efficiencies of subcritical cycles with the respective working fluids employed. Due to process limitations regarding the amount of the recovered heat, there might be no strong correlation between thermal efficiency and exergy efficiency/power output, as can be exemplified by the case of subcritical and supercritical cycles of heptane.

Regarding subcritical cycles, it can be noted that benzene and toluene required adapted plant design with two heat transfer fluid cycles to utilize more heat from the exhaust gas. The advantage of double cycle design was particularly visible in case of toluene, for which adopted plant design caused the increase in the shaft-power output by 0.88 MW. The figure for benzene was 0.21 MW, which can make additional heat transfer fluid cycle incapable of being profitable.

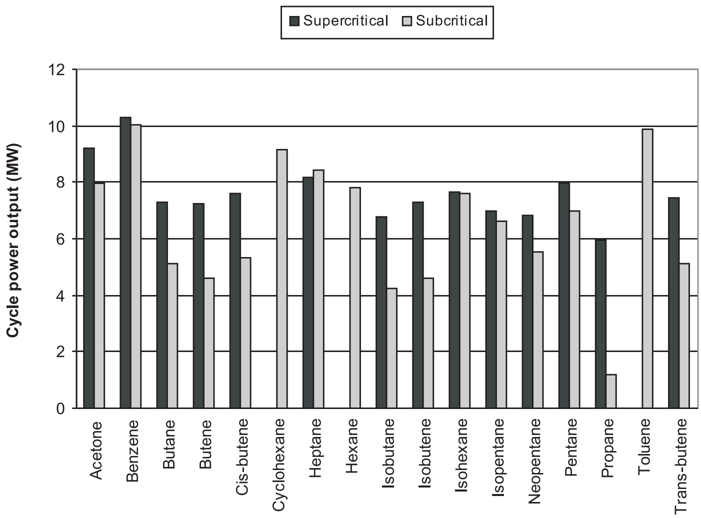

Fig. 2. Power output from basic ORC

TABLE 3

Flow parameters and performance parameters for basic cycle (selected working fluids)

Operational pressure	Working fluid							
	Benzene	**Benzene**	**Toluene**	**Acetone**	**Cyclohexane**	**Heptane**	**Heptane**	**Acetone**
	Super-critical	**Sub-critical**	**Sub-critical**	**Super-critical**	**Sub-critical**	**Sub-critical**	**Super-critical**	**Sub-critical**
P_5 (MPa)	5.33	4.79	2.43	4.80	2.95	2.64	3.26	4.42
T_5 (°C)	320.0	292.2	281.1	276.0	260.0	269.4	285.0	236.0
ΔT_S (°C)	—	5.0	4.1	—	5.0	5.0	—	—
v_6 (m^3/kg)	2.01	2.01	5.56	1.11	1.83	3.21	3.21	1.11
v_6 / v_5	396	416	573	126	245	578	831	146
$\dot{m}_{WF}$ (kg/s)	50.6	58.5	55.8	48.0	57.0	53.1	49.8	59.4
$\dot{m}_{HTF}$ (kg/s)	67.6	74.1/65.3 [a]	102.9/58.7 [a]	67.6	66.5	66.5	67.6	66.4
$\dot{m}_A$ (kg/s)	2090	2414	2200	2443	2134	1846	1735	3022
$\dot{Q}_{HFT}$ (MW)	39.71	41.41	41.50	39.73	41.47	41.48	39.78	40.89
$\dot{Q}_C$ (MW)	28.66	30.13	30.41	29.41	31.52	31.53	31.12	31.97
$\dot{W}_T$ (MW)	11.44	11.68	11.29	10.70	10.22	9.21	8.97	10.37
$\dot{W}_P$ (MW)	0.39	0.40	0.20	0.37	0.27	0.26	0.30	0.42
$\dot{W}_{P,HTF}$ (MW)	0.11	0.27	0.50	0.11	0.10	0.10	0.11	0.10
$\dot{W}_{P,A}$ (MW)	0.63	0.96	0.73	1.00	0.67	0.43	0.36	1.89
$\dot{W}^{plant}$ (MW)	10.31	10.05	9.86	9.22	9.18	8.42	8.20	7.96
η_{exergy} (%)	51.9	49.8	48.9	46.4	45.5	41.73	41.21	39.4
$\eta_{thermal}$ (%)	25.5	23.9	23.4	22.8	21.8	20.0	20.3	19.1

[a] two heat transfer fluid cycles

Figures for energy flow rates shown in table 3 indicate that power consumption by auxiliary equipment such as heat transfer fluid pumps and air condenser fans can be significant. Taking as an example subcritical cycle using acetone as a working fluid, the power consumption of the aerial cooler was 1.89 MW which is the most (18%) of the source power of 10.37 MW produced by the vapour turbine, among the presented fluids.

4.2. Regenerative ORC with heat recuperator

Another investigated aspect was the impact of heat regeneration on cycle net power output, and the possible reduction in the heat rates, the output heat rate in particular, as a consequence of the air-cooled condenser technology assumed for this application. The exhaust gas and atmospheric air inlet parameters were fixed for all cases.

The results of shaft-power output for regenerative ORC with heat recuperator are illustrated in Fig. 3. As can be observed, cycles using benzene, toluene and cyclohexane continue to have the highest power generation potential, however cycle regeneration has a minor effect on the increase of the power output. There is some evidence in the results to suggest that in the case of the regenerative cycle using fluids with low critical temperatures, like propane, isobutane and isobutene, supercritical cycles offer advantage over subcritical cycles of a higher increase in power output. Missing data regarding subcritical cycles of acetone and propane in Fig. 3 result from the uselessness of recuperator in case these fluids are used.

The results for the selected fluids in this cycle configuration are shown in Table 4. It can be noted that the highest shaft-power output was obtained for both supercritical and subcritical cycles of benzene, for which the figures were 10.40 MW and 10.19 MW, respectively, followed by subcritical cycles of toluene and cyclohexane, which showed 10.07 MW and 9.88 MW, respectively.

One of the specifics of ORC applications in compressor stations is that extended duration shutdown can occur as an effect of part-load operation of the pipeline. Accordingly, working fluids with high melting point, i.e. benzene and cyclohexane seem to be inappropriate for this particular application. Among the remaining fluids, toluene, acetone, heptane and pentane provide the highest power output with respect to the configuration with recuperator.

Low specific volume of the saturated vapour gives an indication of the smaller condenser size, which is related to the system capital cost. Toluene shows the highest specific volume of the vapour at the turbine exit, which indicates that it would require larger condenser. Furthermore, it has the lowest saturation pressure, as depicted in Table 2, which suggests that there would be a need for more frequent plant maintenance due to sealing difficulties. On the other hand toluene is the only fluid with the degree of superheating at maximum power output less than 5°C, which indicates that higher superheating is not favourable for increasing the power output. For subcritical cycle using toluene the effect of heat regeneration was an increase of the shaft-power output by 2.1% from 9.86 MW to 10.07 MW, accompanied by the reduction of the condenser heat output by 17.2% from 30.41 MW to 25.17 MW, and the reduction of cycle heat input by 11.9% from 41.50 MW to 36.56 MW. The possibility of the reduction of the condenser area or the decrease in power consumed by the electric motors driving the air fans is the primary advantage of the regenerative cycle over basic cycle configuration. The lower amount of input heat allowing for the reduction of the size of the heat exchangers is also of importance.

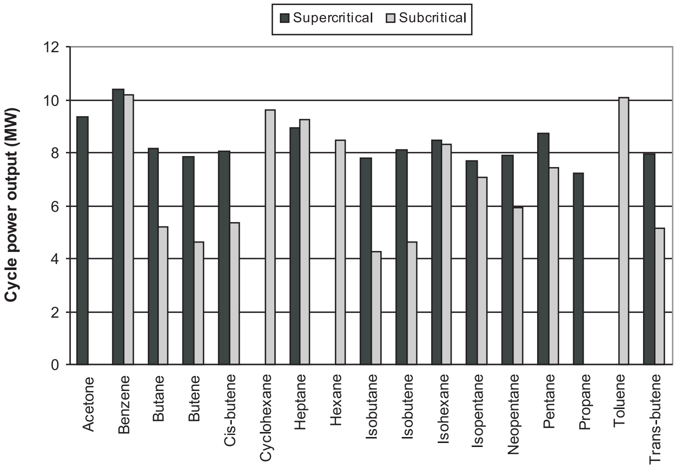

Fig. 3. Power output from regenerative ORC with recuperator

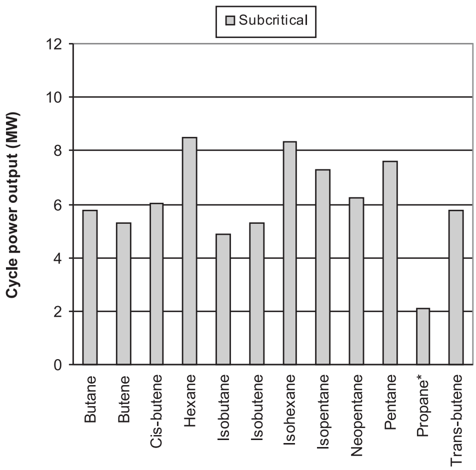

Fig. 4. Power output from regenerative ORC with recuperator and feed-water heater, *feed-water heating only

With respect to the supercritical cycles, heptane has the highest vapour expansion ratio of 1579, which inevitably indicates that it would be necessary to use a larger size turbine with a greater number of stages. Acetone, in contrast, had the lowest expansion ratio of 126, however it shows the highest condenser heat output of 26.75 MW.

Pentane is the working fluid frequently used in ORC systems run on low temperature heat sources. For the application under consideration, it shows the advantage of a relatively small condenser size resulting from the heat output of 22.04 MW, at the expense of a slightly lower, yet still high shaft-power output of 8.72 MW. Pentane has also the lowest underpressure at saturation condition of 82 kPa (see Table 2) among the best performing fluids listed in Table 4, which is the primary reason for selecting it as a working fluid in many commercially available ORC installations.

TABLE 4

Flow parameters and performance parameters for regenerative cycle with recuperator (selected working fluids)

Operational pressure	Working fluid							
	Benzene	Benzene	Toluene	Cyclohexane	Acetone	Heptane	Heptane	Pentane
	Super-critical	Sub-critical	Sub-critical	Sub-critical	Super-critical	Sub-critical	Super-critical	Super-critical
P_5 (MPa)	5.51	4.79	2.40	3.98	4.80	2.64	6.61	8.40
T_5 (°C)	320.0	292.2	280.0	283.5	276.0	269.4	320.0	308.9
ΔT_S (°C)	—	5.0	4.1	5.0	—	5.0	—	—
v_6 (m^3/kg)	2.01	2.01	5.56	1.83	1.11	3.21	3.21	0.41
v_6/v_5	424	416	562	414	126	578	1579	128
$\dot{m}_{WF}$ (kg/s)	51.7	59.6	57.5	58.6	48.8	59.3	51.8	51.1
$\dot{m}_{HTF}$ (kg/s)	72.6	75.6/67.6 [a]	104.5/61.2 [a]	71.5	69.8	73.1	78.4	51.1
$\dot{m}_A$ (kg/s)	2135	2461	2266	2193	2483	2065	1803	1779
$\dot{Q}_{HFT}$ (MW)	34.87	38.23	36.56	35.72	37.24	34.55	31.66	31.32
$\dot{Q}_C$ (MW)	23.67	26.73	25.17	24.98	26.75	24.54	21.99	22.04
$\dot{W}_T$ (MW)	11.62	11.91	11.60	11.11	10.87	10.30	10.30	10.15
$\dot{W}_P$ (MW)	0.41	0.41	0.20	0.38	0.37	0.29	0.63	0.86
$\dot{W}_{P,HTF}$ (MW)	0.14	0.29	0.53	0.13	0.12	0.14	0.18	0.18
$\dot{W}_{P,A}$ (MW)	0.67	1.02	0.80	0.72	1.05	0.60	0.40	0.39
$\dot{W}^{plant}$ (MW)	10.40	10.19	10.07	9.88	9.33	9.27	9.09	8.72
η_{exergy} (%)	52.3	50.5	49.9	48.9	46.9	45.9	45.7	43.8
$\eta_{thermal}$ (%)	29.2	26.2	27.0	27.1	24.6	26.3	28.1	27.2

[a] two heat transfer fluid cycles

4.3. Regenerative ORC with heat recuperator and feed-water heater

Results of power output obtained for the regenerative ORC with combined heat recuperation and turbine bleeding are shown in Fig. 4. Only 12 fluids are presented due to the fact that for the remaining 5 fluids, i.e. acetone, benzene, cyclohexane, heptane, and toluene, the fraction of the working fluid extracted from the turbine is either zero, or so small a value that it is not considered significant. In other words, the results of the optimisation of the cycle parameters indicate that feed-water heating leads to the decrease in power output when these fluids are used. It can be noted that only subcritical cycles benefit from feed-water regeneration in terms of the increase of power output.

The results for selected fluids in this cycle configuration are shown in Table 5. The highest energy recovery was shown by hexane, isohexane, and pentane. The increase in shaft-power output ranged from 0.01 MW for hexane and isohexane to 0.70 MW for cis-butene in relation to the cycle with heat recuperation only, so the changes observed were modest. The differences in heat rates were more notable, as the corresponding results of the decrease in heat input ranged from 0.93 MW for hexane to 3.70 MW for trans-butene, and the decrease in heat output ranged from 0.59 MW for hexane to 4.09 MW for pentane.

The main observation was that the net power output of the regenerative cycles with simultaneous heat recuperation and feed-water heating was not significantly higher compared to that of the cycles with recuperation only, so feed-water heating seemed to be beneficial only in terms of the reduced heat rates. For the subcritical cycle using pentane the effect of simultaneous heat recuperation and feed-water heating was the shaft-power output increased by 2.4% from 7.43 MW to 7.61 MW, the condenser heat output reduced by 7.3% from 29.34 MW to 27.21 MW, the cycle heat input reduced by 5.7% from 38.13 MW to 35.95 MW, compared to the cycle with heat recuperation only.

TABLE 5

Flow parameters and performance parameters for regenerative cycle with recuperator and feed-water heater (selected working fluids)

Operational pressure	Working fluid							
	Hexane	Iso-hexane	Pentane	Iso-pentane	Neo-pentane	Cis-butene	Trans-butene	Butane
	Sub-critical	Sub-critical	Sub-critical	Sub-critical	Sub-critical	Sub-critical	Sub-critical	Sub-critical
P_5 (MPa)	2.45	2.93	3.26	3.28	3.09	4.12	3.93	3.69
T_5 (°C)	225.0	227.1	199.4	190.2	163.6	165.9	159.1	155.3
P_7 (MPa)	0.25	0.32	0.53	0.66	0.98	0.98	1.05	1.14
T_8 (°C)	100.7	102.1	94.7	96.0	93.5	82.3	82.3	85.7
x	0.09	0.10	0.14	0.17	0.27	0.19	0.21	0.25
ΔT_S (°C)	5.0	5.0	5.0	5.0	5.0	5.0	5.0	5.0
v_6 (m^3/kg)	1.15	0.83	0.41	0.31	0.16	0.17	0.15	0.14
v_6/v_5	142	142	63	47	24	23	21	19
$\dot{m}_{WF}$ (kg/s)	69.2	73.2	82.0	89.8	113.9	98.9	102.6	106.2
$\dot{m}_{HTF}$ (kg/s)	71.8	72.0	71.2	71.4	71.1	70.1	70.1	70.4
$\dot{m}_A$ (kg/s)	2276	2264	2449	2461	2564	2781	2790	2736
$\dot{Q}_{HFT}$ (MW)	35.50	35.35	35.95	35.83	36.03	36.93	36.94	36.67
$\dot{Q}_C$ (MW)	26.06	26.08	27.21	27.39	28.54	29.30	29.54	29.40
$\dot{W}_T$ (MW)	9.80	9.73	9.32	9.06	8.25	8.46	8.23	8.11
$\dot{W}_P$ (MW)	0.36	0.46	0.58	0.63	0.75	0.83	0.83	0.84
$\dot{W}_{P,HTF}$ (MW)	0.13	0.14	0.13	0.13	0.13	0.12	0.12	0.13
$\dot{W}_{P,A}$ (MW)	0.81	0.79	1.00	1.02	1.15	1.47	1.49	1.40
$\dot{W}^{plant}$ (MW)	8.50	8.34	7.61	7.28	6.22	6.04	5.79	5.74
η_{exergy} (%)	42.1	41.4	37.7	36.1	30.8	29.9	28.7	28.5
$\eta_{thermal}$ (%)	23.5	23.1	20.8	19.9	16.9	16.1	15.4	15.4

Based on the observations related to steam Rankine cycles, previous research on ORCs assumed either arbitrary selected bleeding pressure values, or such that the temperature of the feed liquid was the average temperature of the cycle $T_8 = (T_5 + T_1)/2$, where T_1 is the condensing temperature (Desai and Bandyopadhyay, 2009). Then, the study with bleeding pressure equal to the turbine average pressure $p_7 = p_5 - (p_5 - p_6)/2$ is also available in the literature (Tchanche et al., 2010). However, it may be noted from Table 5 that in case of organic liquids the highest cycle power outputs were seen for the bleeding temperature slightly below the average temperature of the cycle, which corresponds with the bleeding pressure significantly below turbine average pressure. Furthermore, vapour superheating was seen as favourable for higher power output in case of all best performing fluids in regenerative cycles with simultaneous heat recuperation and feed-water heating.

It has been observed that thermal and exergy efficiencies may be increased significantly by heat recuperation and feed-water heating. For subcritical cycle using pentane, the thermal efficiency increased from 16.6% in case of basic cycle configuration to 19.1% in case of heat recuperation and 20.8% in case of simultaneous heat recuperation and feed-water heating. The corresponding results for exergy efficiency were 34.6%, 36.8%, and 37.7%, respectively.

5. Summary and Conclusions

Preliminary assessment of bottoming ORC systems for gas turbine in pipeline compressor station has been carried out in this study. As a heat source data, information concerning nominal operating conditions of the compressor driver of the Yamal-Europe pipeline was taken.

Commercially available waste heat recovery plants for pipeline compressor stations incorporate ORCs that utilize pentane as a working medium. The results of this study show that benzene, toluene, cyclohexane, and some other alkanes like heptane and hexane appear as a competitive working fluids from a thermodynamic point of view. The fact remains that benzene and cyclohexane are non-freezing fluids above -5°C and 6°C, respectively, which may limit or even preclude the possibility of their use in some applications. Consequently, toluene merit consideration for use in ORCs for gas turbines in mechanical drive applications. The widely-used pentane should be considered as a relatively good working fluid in terms of thermal performance, and as a much better working fluid regarding operational aspects. Unlike toluene, pentane offers the possibility for avoiding leakage of ambient air in the system.

A parametric study aiming at the maximization of the cycle net power output was conducted using genetic algorithm. The search was performed for each combination of cycle configuration and working fluid employed, resulting in the values of design parameters allowing for a better integration of cycle processes with exhaust heat stream from a thermodynamic point of view. The shaft-power outputs obtained for the most preferred fluids in cycles with simultaneous heat recuperation and feed-water heating (ranging from 8.50 MW to 5.74 MW) were considerably lower compared to the figures for the best performing fluids in regenerative cycles with heat recuperation only (ranging from 10.40 MW to 8.72 MW). As a result, when focusing on the net power output, the configuration with heat recuperation only should be preferred. The evaporator heat input and the condenser heat output seem to be the quantities favouring the configuration with feed-water heating. However, the benefits may not compensate for the drawbacks associated with the increased complexity of the plant.

The rejected waste heat of a 25 MW compressor driver may be estimated as equal to 46.2 MW at atmospheric temperature of 15°C, and the idealized analysis indicates that this significant amount of hitherto lost exergy allows for the production of a maximum of about 10.4 MW of shaft power by regenerative ORC system with recuperator. The actual efficiencies of the equipment as well as the heat losses and the pressure drops in the heat exchangers and piping should be further accounted for in a more conservative estimate of the possible energy recovery.

With regard to technological advancement, high capacity turboexpanders are widely used in natural gas industry applications, yet the fact remains that a 10 MW-class turboexpanders, suitable for the scale of application considered in this study do not have many commercial applications in the ORC industry, which may result in an undersized ORC system.

In summary, the study reported here has merely a thermodynamic aspect, and a potential investment opportunity has to be evaluated in economic terms. The final technical specifications result from a compromise between high power output and low investment costs of the plant.

References

Angelino G., Colonna Di Paliano P., 1998. *Multicomponent working fluids for organic Rankine cycles (ORCs)*. Energy, 23(6), 449-463.

Badr O., Probert S.D., O'Callaghan W.P., 1985. *Selecting a working fluid for a Rankine-cycle engine*. Appl. Energy, 21(1), 1-42.

Borsukiewicz-Gozdur A., Nowak W., 2007. *Comparative analysis of natural and synthetic refrigerants in application to low temperature Clausius-Rankine cycle*. Energy, 32(4), 344-352.

Chaczykowski M., 2012. *Thermal processes in natural gas compressor station for pipeline system efficiency improvements*. Scientific Papers of The Warsaw University of Technology, series Environmental Engineering, No. 60, Warszawa (in Polish).

Chaczykowski M., 2016. *Organic Rankine cycle for residual heat to power conversion in natural gas compressor station. Part I: Modelling and optimisation framework*. Arch. Min. Sci., Vol. 61, No 2, p. 245-258.

Charbonneau P., 2002. *Release Notes for PIKAIA 1.2, NCAR Technical Note 451+STR*. National Center for Atmospheric Research, Boulder.

Chen Y., Lundqvist P., Johansson A., Platell P., 2006. *A comparative study of the carbon dioxide transcritical power cycle compared with an organic Rankine cycle with R123 as working fluid in waste heat recovery*. Appl. Therm. Eng., 26(17-18), 2142-2147.

Desai N.B., Bandyopadhyay S., 2009. *Process integration of organic Rankine cycle*. Energy, 34, 1674-1686.

Drescher U., Brüggemann D., 2007. *Fluid selection for the Organic Rankine Cycle (ORC) in biomass power and heat plants*. Appl. Therm. Eng., 27(1) 223-228.

Hung T.-C., 2001. *Waste heat recovery of organic Rankine cycle using dry fluids*. Energy Convers Manage, 42(5), 539-553.

Hung T.C., Shai T.Y., Wang S.K., 1997. *A review of organic Rankine cycles (ORCs) for the recovery of low-grade waste heat*. Energy, 22(7), 661-667.

Karellas S., Schuster A., 2008. *Supercritical Fluid Parameters in Organic Rankine Cycle Applications*. Int. J. of Thermodynamics, 11(3), 101-108.

Larjola J., 1995. *Electricity from industrial waste heat using high-speed organic Rankine cycle (ORC)*. Int. J. Production Economics, 41(1-3), 227-235.

Lemmon E.W., Huber M.L., McLinden M.O., 2007. *NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 8.0*. National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg.

Liu B.-T., Chien K.-H., Wang C.-C., 2004. *Effect of working fluids on organic Rankine cycle for waste heat recovery*. Energy, 29(8), 1207-1217.

Mago P.J., Chamra L.M., Somayaji C., 2007. *Performance analysis of different working fluids for use in organic Rankine cycles*. Proc. IMechE, Part A: J. Power and Energy, 221(3), 255-264.

Maizza V., Maizza A., 1996. *Working fluids in non-steady flows for waste energy recovery systems*, Appl. Therm. Eng., 16(7), 579-590.

NREL, 2008. *The System Advisor Model (SAM Version 2.5) Reference Manual*. National Renewable Energy Laboratory, Denver.

Papadopoulos A.I., Stijepovic M., Linke P., 2010. *On the systematic design and selection of optimal working fluids for Organic Rankine Cycles*. Appl. Therm. Eng., 30(6-7), 760-769.

Rayegan R., Tao Y.X., 2011. *A procedure to select working fluids for Solar Organic Rankine Cycles (ORCs)*. Renew Energy, 36(2), 659-670.

Schroeder D.J., Leslie N., 2010. *Organic Rankine Cycle Working Fluid Considerations for Waste Heat to Power Applications*, ASHRAE Transactions, 116 PART 1, 525-533.

Tchanche B.F., Lambrinos Gr., Frangoudakis A., Papadakis G., 2010. *Exergy analysis of micro-organic Rankine power cycles for a small scale solar driven reverse osmosis desalination system*. Appl. Energy, 87(4), 1295-1306.