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**ORGANIC RANKINE CYCLE FOR RESIDUAL HEAT TO POWER CONVERSION
IN NATURAL GAS COMPRESSOR STATION.
PART I: MODELLING AND OPTIMISATION FRAMEWORK****ORGANICZNY OBIEG RANKINA DO PRODUKCJI ENERGII ELEKTRYCZNEJ
Z CIEPŁA ODPADOWEGO W TŁOCZNI GAZU.
CZĘŚĆ I: MODEL MATEMATYCZNY SYSTEMU I SFORMUŁOWANIE ZADANIA OPTYMALIZACJI**

Basic organic Rankine cycle (ORC), and two variants of regenerative ORC have been considered for the recovery of exhaust heat from natural gas compressor station. The modelling framework for ORC systems has been presented and the optimisation of the systems was carried out with turbine power output as the variable to be maximized. The determination of ORC system design parameters was accomplished by means of the genetic algorithm. The study was aimed at estimating the thermodynamic potential of different ORC configurations with several working fluids employed. The first part of this paper describes the ORC equipment models which are employed to build a NLP formulation to tackle design problems representative for waste energy recovery on gas turbines driving natural gas pipeline compressors.

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

W artykule analizowano organiczny obieg Rankine'a (ORC) w wariantach podstawowym i z regeneracją ciepła, w celu odzyskiwania ciepła odpadowego w tłoczni gazu. Przedstawiono model matematyczny elementów systemu oraz sformułowano problem optymalizacji systemu, przyjmując maksymalizację mocy elektrycznej produkowanej w instalacji odzysku ciepła jako funkcję celu. Zadanie optymalizacji rozwiązano z wykorzystaniem algorytmu genetycznego. Celem badań było oszacowanie potencjalnych możliwości produkcji energii elektrycznej przy różnych konfiguracjach układu ORC oraz przy różnych czynnikach roboczych. W pierwszej części pracy przedstawiono uproszczony model matematyczny obiegu ORC, który posłużył do sformułowania zadania programowania nieliniowego, pozwalającego na rozwiązywanie typowych problemów projektowych instalacji odzysku ciepła z turbin gazowych w stacjach przetłaczających gaz.

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

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Notation

A	– area,
D	– diameter,
\dot{E}	– exergy rate,
h	– enthalpy,
L	– length,
\dot{m}	– mass flow rate,
p	– pressure,
\dot{Q}	– heat rate,
s	– entropy,
T	– temperature,
\dot{V}	– volumetric flow rate,
\dot{W}	– mechanical power.

Greek symbols

ε	– effectiveness,
ζ	– resistance coefficient,
η	– efficiency,
λ	– Darcy friction factor,
ρ	– density.

Subscripts

A	– air,
B	– boiler,
C	– condenser,
g	– saturated vapour,
P	– pump,
s	– isentropic,
S	– superheating,
T	– turbine,
0	– environmental conditions (reference state).

Acronyms

EG	– exhaust gas,
HTF	– heat transfer fluid,
WF	– working fluid.

1. Introduction

It is generally accepted that natural gas will remain among biggest energy sources for power generation, even at the time when alternative energies gain a sizeable share of the power sector. Natural gas offers a potential to provide flexible back-up for renewable energy sources and an advantage of a lower carbon intensity compared to coal when used to generate electricity. Therefore, its availability in Europe is continuously expanding through new LNG and pipeline

investment projects. An increased interest is now observed in studying the potential of improving the efficiency of energy conversion technologies in natural gas upstream applications (Szargut & Szczygieł, 2009; Łaciak, 2013), as well as natural gas midstream applications such as compressor stations (Gutiérrez & López, 2009; Saavedra et al., 2010; Chaczykowski, 2012) and pressure regulator stations (Howard et al., 2011; Kostowski & Usón, 2013).

Pipeline compressor station is a principal component of any gas transmission system. The process of gas compression in today's constructed pipeline systems is usually carried out in centrifugal compressors, and the choices for drivers can be gas turbines or electric motors. An important factor in favour of the gas drivers is the availability and reliability of the energy source. Failure statistics for the electric grids must be taken into account in the feasibility studies of the electric drivers, therefore gas turbines with centrifugal compressors are usually the preferred means of compressing the gas (Mokhatab et al., 2007).

Virtually all gas turbine installations in pipeline compressor stations operate in a simple cycle, in which the waste heat is rejected into the atmosphere, representing appreciable exergy loss. One should not overlook the fact that exhaust gases from a gas turbine in mechanical drive application can have a temperature of above 550°C, depending on the actual load and the environmental conditions. Accordingly, the compression of natural gas can be considered as an industrial process from which large amount of thermal energy, classified as low-grade heat, is available for conversion. As a matter of fact, the interest in heat recovery systems in gas compressor stations has recently increased. Fifteen existing organic Rankine cycle (ORC) plants were reported in the U.S. and Canada in 2009, nine of them had been commissioned in the last two years. The first plant in Europe started up in 2009 at Almendralejo compressor station in Spain. All above systems use natural gas as a fuel to cogenerate mechanical power for pipeline compressors and electrical power for the grid.

The amount of energy that can be recovered from the exhaust of a gas turbine is a function of temperature and volume of the flue gases, which in turn depend on turbine power. According to the report by INGAA (Hedman, 2008), gas turbines appear to offer the most potential for viable heat recovery projects, provided that the total station capacity is at least 11 MW with 5250 hours of annual operation (60% load factor). The report estimated that 90-100 compressor stations in the U.S. gas transmission systems meet these constraints, representing approximately 500 to 600 MW of potential power generation capacity.

The paper (Part 1 and 2) describes the major features of the ORC technology, plant modelling and optimisation framework, and finally presents a case study to demonstrate the procedure of initial equipment sizing.

2. Bottoming ORCs for gas turbines

Much research has been undertaken on ORCs to utilize low temperature waste heat. By contrast, relatively little attention has been focused on performance evaluation and optimisation of bottoming ORC in gas turbine applications, for the simple reason that water is the working fluid most often used in CHP plant cycles. In the paper by Najjar and Radhwan (1988) the authors showed that combining Brayton cycle realized in a recuperated gas turbine and bottoming ORC resulted in global thermal efficiency of combined cycle exceeding 45%. As a working fluid R22 was chosen.

Larjola (1995) studied the ORC system powered with exhaust gases at 425°C. Employing toluene as a working fluid, the ORC process efficiency of 26% was obtained.

Hung (2002) presented a conceptual design of a triple cycle power generation system which included Brayton cycle combined with steam Rankine cycle and ORC. The condenser heat and the residual heat of the exhaust gases were considered as a heat source for the ORC system. It has been illustrated that proper combination of the ORC with the steam cycle can result in triple cycle thermal efficiency exceeding 60%.

Lee and Kim (2006), and Invernizzi et al. (2007) investigated the possibility of recovering the thermal power of the exhaust gases in micro-gas turbine applications. The temperature of the exhaust gases from micro-gas turbines is much lower than that from large gas turbines, and a steam bottoming cycle is generally inappropriate to accomplish acceptable plant performances. This is caused by the fact that turbine cooling is not feasible both economically and technically in such applications, and the maximum turbine inlet temperature must be lower. In addition, the pressure ratios are relatively low, and virtually all applications employ recuperation. The studies have shown that the combined cycle global efficiency of 39% can be achieved based on the micro-gas turbine of 30% electrical efficiency. In the studies reported were R123 and MM (esa-methyl-disiloxane) as highest-efficiency working fluids.

Research study on bottoming ORCs for gas turbines has been presented by Chacartegui et al. (2009). Analysed were medium and large scale power plant applications of regenerative ORC. The results showed that ORC is an interesting option when combined with high efficiency heavy duty gas turbine, characterized by low exhaust temperatures. Among the working fluids analysed, toluene and cyclohexane have shown the highest global efficiencies of combined cycle, which were above 58% and 57%, respectively.

Saavedra et al. (2010) studied the effect of condensation temperature on the performance of ORC system utilizing residual heat from 2.6 MW gas turbine in natural gas compressor station. Three heat rejection options were analysed: air-cooled condenser at 50°C, and two variants of the water-cooled condenser, at 35°C and 90°, respectively. Among the fifteen working fluids considered, at the condensation temperature of 35°C, the highest power output of 1.22 MW was observed for toluene, while at the condensation temperatures of 50°C and 90°C, the highest power outputs of 1.08 MW and 0.78 MW, respectively, were obtained for pentafluorobenzene. The ORC system was modelled using Aspen HYSYS process simulator.

3. Specifics of ORC systems with respect to compressor stations

The widespread use of steam cycles in industrial CHP plants results from its cost-effectiveness, since the turbine's exhaust heat can be utilized to provide steam/hot water for the plant or an adjacent industrial and commercial users. Pipeline compressor stations, however, have very few thermal energy requirements, are typically located in isolated areas, and it is usually impossible to implement the heat delivery to district heating systems.

An alternative to recovering heat to provide thermal energy is to convert it into mechanical energy through a Rankine cycle, and deliver it to an additional compressor or an electric generator. In Europe, bottoming Rankine cycles in which the steam turbine was powering additional pipeline compressor were reported in Germany and Italy in mid-1980s. According to Leslie et al. (2009), four bottoming steam systems were installed at pipeline compressor stations in the

U.S. between 1968 and 1970, and another such system was built in the early 1980s. Between 1992 and 2000 in Ontario, Canada, five steam bottoming systems were installed in compressor stations to enhance the output of the adjacent combined cycle power plants. In one of the systems in Alberta, Canada, initially constructed to work with a steam cycle, freezing problems led to the installation of a system which uses an organic working fluid.

Above examples are rare, largely because the gas turbines powering natural gas compressors are about an order of magnitude smaller than their electric utility plant's counterparts, which leads to small steam systems, whose unit cost (per kW) is significantly higher. Furthermore, the direct drive of the natural gas compressors leads to frequent non-design operating conditions of the turbine, since gas turbines in pipeline applications, in which seasonal load fluctuations occur, allow for the compressors to operate at reduced load, and a decrease in exhaust gas temperature and volume is observed. According to Hung (2001), at heat source temperatures below 370°C conventional steam Rankine cycle fails to allow for efficient conversion of heat into mechanical energy. In such cases ORC is considered as a better bottoming cycle option, for the energy recovery from variable-temperature heat sources in medium to small scale power generation applications in particular. In the study by Nguyen et al. (2010) it was shown that for a given heat source at a temperature between 100°C and 225°C, the cycle using water performed the poorest in terms of the power generation potential when compared to ammonia, propane, isopentane, benzene and heptane. In the application discussed by Gutiérrez and López (2009), it was seen that ORC system generated more electricity than steam bottoming plant on an annual basis, when seasonal load fluctuations occurred in the pipeline. Bronicki and Schochet (2005) discussed additional advantages of ORCs over steam cycles with reference to natural gas compressor stations: (i) simplicity of the turbine plant, control system, and smaller size of plant equipment (turbines, pipes and condensers) as an effect of a lower specific volume of organic fluid, (ii) suitability for air-cooled applications (in areas where in-situ water resources do not exist) as a consequence of a smaller condenser size, and (iii) low-pressure process enabling remote, unattended operation of the plant, leading in turn to lower operating cost under current regulations in most U.S. states, in which a licensed steam plant operator is required.

Gutiérrez and López (2009) pointed out that bottoming ORCs in compressor stations contribute to power generation, and as such promote the independence from external energy supply, which follows the precepts of the EC law on the promotion of cogeneration.

Safety and failure consequences are some of the main concerns that should be addressed in the design of compressor station facilities. It has been assumed that diathermic oil is used as an intermediate heat transfer medium between gas turbine exhaust gases and organic fluid (Fig. 1). The system's intermediate heat transfer medium is typically adopted in applications, in which the temperature levels are relatively high, e.g. biomass power and heat plants, solar ORCs. Synthetic oil is usually used as a heat transfer fluid, which circulates through a heat recovery oil heater and a series of heat exchangers in an ORC plant. As a result of the oil thermal inertia, the intermediate heat carrier fluid ensures higher stability for the operation of the ORC system. The heat carrier provides additional safety for the system operations, since it reduces risks related to the flammability of the working fluid and makes it possible to operate the oil heater at atmospheric pressure. Furthermore, the collection of thermal energy from more than one gas turbine exhaust system is possible, which is particularly important in parallel arrangement of the compressors, in which some of the units are on a standby during the part-load operation conditions of the pipeline. The advantages of using the intermediate heat transfer medium are at the expense of a lower global system efficiency, resulting from additional exergy losses in the heat exchange process. What is

more, there are two instead of one pinch point limitations: one at the beginning of vaporization and another between heat transfer fluid and turbine exhaust gas, which can affect ORC system performance. In order to reduce the pinch point constriction at the beginning of vaporization, an adapted plant design with separate heat transfer fluid cycles in economizer and evaporator can be made, as discussed by Drescher and Brüggemann (2007).

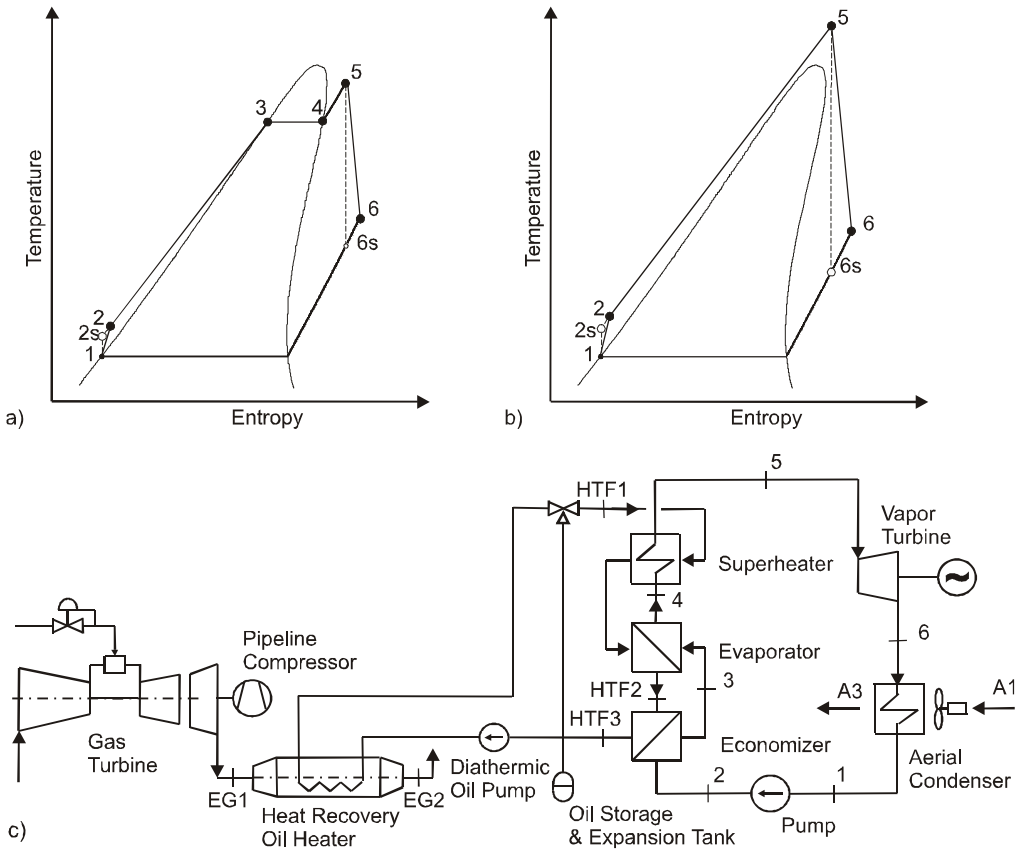


Fig. 1. Basic ORC configuration, a) subcritical, b) supercritical, c) flow-sheet of subcritical ORC plant, EG-exhaust gas, HTF-heat transfer fluid, A-air

Waste heat recovery projects in compressor stations are usually implemented on a retrofit basis, and there might be a necessity for the new equipment to be sited in some distance from the waste heat-to-oil heat exchanger due to lack of available space or station's hazardous area zoning. Therefore, the circulation pump power must be taken into consideration in the calculation of the net power output of the ORC plant. The installation itself would have to comply with the provisions contained in explosive atmospheres legislation.

4. Determination of the ORC design parameters

It is apparent that pressure and temperature at the inlet of the vapour turbine in the ORC plant have an impact on the overall system efficiency and the net power output. Several research studies investigating the performance of the ORCs are available at present in the scientific literature. Most of the works focus on determination of ORC parameters by sensitivity analysis using thermal efficiency of the cycle as a criterion. The system thermal or energy efficiency is defined as the ratio of the net power output to the heat input to the engine. Exergy efficiency, defined on the basis of the second law as the ratio of system total exergy used to the system total exergy available, was also used as a criterion for calculation of cycle parameters in the research studies conducted in the literature. The results of second law analysis help us to choose working fluids and system configurations that enable the recovery of a greater portion of input exergy of the cycle. Other design parameters, such as evaporator pitch point temperature difference or condensing temperature will also have impact on the results of parametric study of the cycle. Finally, variable operational conditions, such as ambient temperature, waste heat source temperature and mass flow rate, will have a significant effect on the ORC plant power output.

Part-load efficiencies of working fluid pumps and vapour turbines can be derived from their characteristic maps provided by manufacturers, or from analytic expressions available in the literature. Detailed calculations of part load performances of evaporators and air cooled condensers are usually performed by adopting NTU- ϵ and LMTD methods to calculate the heat and mass balance. Mathematical models for the turbine, evaporator, air cooled condenser and pump, developed to address operational issues in an ORC plant are presented in the study by Sun and Li (2011).

This study concentrates on the optimisation problem under design conditions. The optimisation of the systems was carried out with turbine power output as the variable to be maximized. In order to make a valid comparison between the various cycle configurations each option was compared assuming the same exhaust gas parameters and ambient air conditions. Consequently, exhaust gas temperature and flow rate, ambient air temperature, and working fluid condensing temperature were assumed to be constant.

In the studies by Chacartegui et al. (2009), Dai et al. (2009), Wang et al. (2009), the calculation of the parameters of the ORC system was carried out by means of the genetic algorithm (GA). GA based search method was also used in this study for the solution of ORC parameter selection problem. GAs are adaptive heuristic search algorithms, considered to be an important methodology in the development of search methods (Goldberg, 1989; Michalewicz, 1996), which has already received attention in many research fields, including thermal engineering. The parameter selection problem solved by the GA based method in this study consists in finding the “point” \mathbf{x} defining the ORC model parameters that maximize the quality measure $f(\mathbf{x})$, i.e. the net power output of the system, subject to the constraints resulting from allowable design criteria. Detailed problem formulations vary as to the configuration of the ORC systems and will be presented later in the following section.

5. ORC model

From a thermodynamic viewpoint the ORC consists of four identical processes, as in the case of steam Rankine cycle: compression of the working fluid through a pump, isobaric heat addition through an evaporator, expansion of the high temperature and high pressure fluid through

a turbine, and isobaric heat rejection through a condenser. Common modifications of the steam Rankine system implement steam reheating and feed-water regeneration, which increase thermal efficiency of the cycle. However, the studies on ORCs conducted in the literature reveal that integration of additional processes into the ORC not always leads to a considerable improvement in engine performance. For this reason, the evaluation of the three ORC configurations is carried out in this study and the following gives a brief discussion of the theory for the analysis of the systems. The analysis was based on the steady-state rate balances of mass and energy. Heat losses and pressure drops in the heat exchangers and pipes were neglected, except from the heat losses in the heat recovery oil heater. In order not to compromise the relative simplicity of the plant, only the three relatively simple configurations, i.e. the basic cycle model, the model of regenerative cycle with recuperator, and the model of regenerative cycle incorporating both recuperator and open feed-water heater (turbine bleeding) were considered in the present study.

5.1. Basic ORC configuration

One of the specific ORC plant features which uniquely satisfies market needs is its simplicity. Despite of relatively low levels of achievable efficiency, basic cycle configuration (Fig. 1) will be described in this section and further considered as a “benchmark” for comparison between alternate cycle configurations.

Given the condensation temperature T_1 , the values of enthalpy and entropy at pump inlet are obtained from the properties of a saturated liquid. For an isentropic process in a pump, flash calculations are performed for determining the enthalpy at pump exit using the equation of state, based on the assumption of isobaric heat addition in the boiler. The enthalpy and entropy at turbine inlet are obtained from property relations, given the turbine inlet pressure p_5 and temperature T_5 . By analogy, for an isentropic process in a turbine, the enthalpy at turbine exit is determined from flash calculations assuming isobaric heat rejection through a condenser. The energy rate balance for the components of the system, i.e. turbine, condenser, pump, and boiler, yields

$$\dot{W}_T = \dot{m}_{WF} (h_6 - h_5) = \dot{m}_{WF} \eta_T (h_{6s} - h_5) \quad (1)$$

$$\dot{Q}_C = \dot{m}_{WF} (h_6 - h_1) \quad (2)$$

$$\dot{W}_P = \dot{m}_{WF} (h_2 - h_1) = \frac{\dot{m}_{WF}}{\eta_P} (h_{2s} - h_1) \quad (3)$$

$$\dot{Q}_B = \dot{m}_{WF} (h_5 - h_2) \quad (4)$$

where \dot{m}_{WF} is the working fluid flow rate, h_{6s} is the working fluid enthalpy at turbine exit after an isentropic expansion, η_T is the isentropic efficiency of the turbine, η_P is the isentropic efficiency of the pump, and h_{2s} is the working fluid enthalpy at pump exit if the compression were isentropic.

The calculation of the cycle efficiency must involve the estimation of the power consumed by the circulation pump of the heat transfer fluid. Neglecting the thermal expansion of the fluid we obtain

$$\dot{W}_{P,\text{HTF}} = \dot{m}_{\text{HTF}} \int_{P_{\min}}^{P_{\max}} v dp = \frac{8\bar{\lambda} \dot{m}_{\text{HTF}}^3 L}{\rho_{\text{HTF}}^2 \pi^2 D^5} \quad (5)$$

where $\bar{\lambda}$ is the average value of Darcy friction factor, L is the heat transfer fluid line length, ρ_{HTF} is the average value of heat transfer fluid density, and D is the diameter of the heat transfer fluid line.

We consider here that ambient air will serve as a coolant for the power cycle in a forced draught air cooler. The overall system efficiency requires taking into account the power consumed by the air fans

$$\dot{W}_{P,A} = \Delta p_A \dot{V}_A / \eta_A \quad (6)$$

where Δp_A is the pressure drop in the air coolers, \dot{V}_A is air volumetric flow rate, and η_A is the efficiency of the air fan. The pressure drop can be expressed in terms of a constant loss coefficient defined as

$$\zeta = \frac{\Delta p_A}{\rho_A w_A^2 / 2} \quad (7)$$

where w_A is the air velocity. The air velocity and the volumetric flow rate correspond to the mass flow rate

$$\dot{m}_A = \rho_A \dot{V}_A = \rho_A w_A A \quad (8)$$

Combining Eqs. (6)-(8) the power consumed by the air cooler is calculated as

$$\dot{W}_{P,A} = \frac{\zeta \dot{m}_A^3}{2\rho_A^2 A^2 \eta_A} \quad (9)$$

Accordingly the net power output is

$$\dot{W}^{\text{plant}} = \dot{W}_T - \dot{W}_P - \dot{W}_{P,\text{HTF}} - \dot{W}_{P,A} \quad (10)$$

The heat transfer fluid flow rate \dot{m}_{HTF} , working fluid flow rate \dot{m}_{WF} , and air flow rate \dot{m}_A are obtained from the analysis of heat transfer processes presented in Part 2 of this paper.

The primary interest in this study is to compare the cycles of different configurations and working fluids employed in terms of the highest power output, when utilizing the same exhaust heat source and at a constant heat rejection conditions. The turbine inlet pressure and temperature are chosen to be variables. Denoting the design variables as $\mathbf{x} \equiv (p_5, T_5)^T$, and the objective function as $f(\mathbf{x}) = \dot{W}^{\text{plant}}$, the NLP formulation of the ORC parameter selection problem in the case of basic configuration can be stated as

Find \mathbf{x} which

$$\max \dot{W}^{\text{plant}} \quad (11)$$

subject to

$$p_5 \leq p_5^{(U)} \quad (12)$$

$$T_5 \leq T_5^{(U)} \quad (13)$$

The pressure bound $p_5^{(U)}$ is imposed in order to limit safety measures, and the maximum process temperature $T_5^{(U)}$ results from the thermal stability or flammability limits of the working fluid. In case of the subcritical cycles, a constraint on the degree of superheating $\Delta T_S = T_5 - T_g(p_5)$ is set to reduce the heat transfer area requirement, and hence material expenses associated with the superheater. Therefore, the design variables are $\mathbf{x} \equiv (p_5, \Delta T_S)^T$, and the inequality constraint (13) is replaced by $0 \leq \Delta T_S \leq \Delta T_S^{(U)}$.

5.2. Regenerative ORC with heat recuperator

In case of dry working fluids, the positive slope of the saturated vapour line causes the fluid exiting the turbine to be in a saturated vapour state with some excess heat that can be recaptured. As a result, commercial ORC systems usually contain recuperators, which use thermal energy from turbine exhaust to preheat working fluid (Fig. 2). The primary effect of the additional internal heat exchange is the reduction in the mean temperature difference between heat transfer

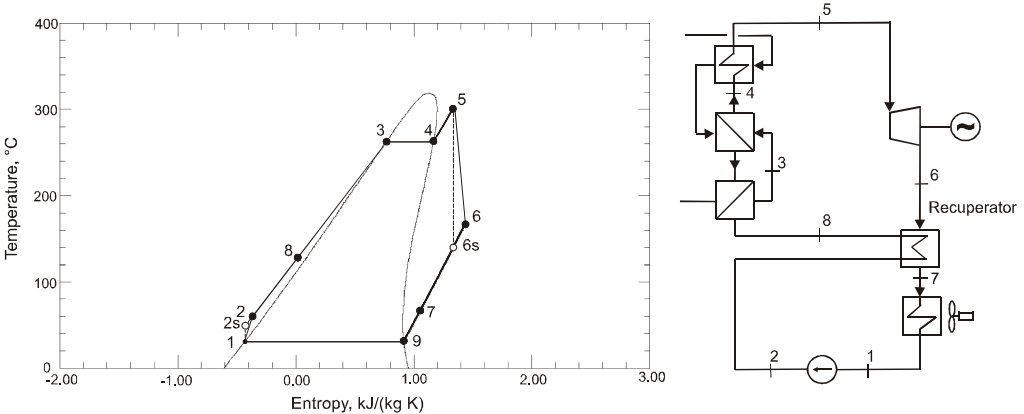


Fig. 2. ORC with recuperator (toluene as working fluid, the scale of processes 1-2 and 4-5 distorted for clarity)

fluid and working fluid in the boiler, leading to a lower exergy destruction. The presence of the recuperator makes a minor contribution to the increase in the complexity of the plant, however the use of additional equipment leads to increased capital costs. Recuperator is assumed to be an adiabatic, counter-current heat exchanger with a specified effectiveness. The turbine exhaust, which is the hot working fluid of the recuperator, is the stream with the minimum capacitance rate in this application. Accordingly, effectiveness is defined as

$$\varepsilon = \frac{h_6 - h_7}{h_6 - h_{7,\min}} \tag{14}$$

where $h_{7,\min} = h(p_1, T_2)$ is the minimum possible exit enthalpy for the smaller capacitance rate stream. The enthalpy at state 7 is

$$h_7 = h_6 - \varepsilon (h_6 - h_{7,\min}) \tag{15}$$

where ε – recuperator effectiveness. The enthalpy at state 8 is calculated from the energy rate balance for an adiabatic recuperator

$$h_8 = h_2 + h_6 - h_7 \quad (16)$$

The energy rate balance for the boiler is

$$\dot{Q}_B = \dot{m}_{WF} (h_5 - h_8) \quad (17)$$

The formulation of the ORC parameter selection problem remains unchanged.

5.3. Regenerative ORC with recuperator and feed-water heater

It has been assumed in this cycle configuration that similarly to the steam power plants an open feed-water heater (we use the term *water* despite the fact that organic liquid is processed) preheats the working fluid before it enters the boiler. The new component is essentially a direct contact heat exchanger, in which fraction of vapour extracted from the turbine mixes with the liquid exiting the pump (Fig. 3).

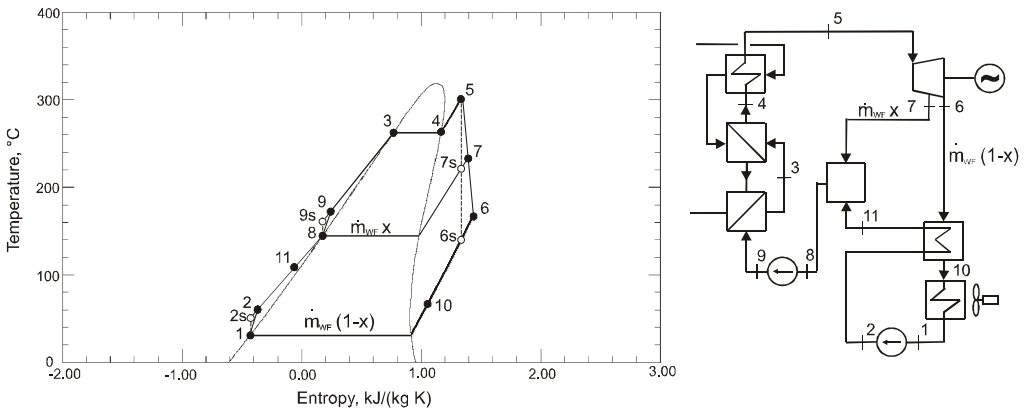


Fig. 3. ORC with recuperator and open feed-water heater (combined recuperation and turbine bleeding)

The enthalpy of organic liquid at state 11 (feed-water heater input) is calculated based on recuperator effectiveness

$$h_{11} = h_2 + \varepsilon (h_6 - h_{10,\min}) \quad (18)$$

where $\varepsilon = \frac{h_6 - h_{10}}{h_6 - h_{10,\min}}$.

The amount of vapour that goes into the feed-water heater can be determined from the energy balance equation

$$\dot{m}_{WF} x (h_7 - h_8) = \dot{m}_{WF} (1-x) (h_8 - h_{11}) \quad (19)$$

By rearranging Eq. (19) the fraction of the working fluid extracted from the turbine is given by

$$x = \frac{h_8 - h_{11}}{h_7 - h_{11}} \quad (20)$$

The gross turbine power, pump power, and the energy rate balance for the boiler can be determined as

$$\dot{W}_T = \dot{m}_{WF} \eta_T [(h_5 - h_{6s}) + x(h_{6s} - h_{7s})] \quad (21)$$

$$\dot{W}_P = \frac{\dot{m}_{WF}}{\eta_P} [(1-x)(h_{2s} - h_1) + (h_{9s} - h_8)] \quad (22)$$

$$\dot{Q}_B = \dot{m}_{WF} (h_5 - h_9) \quad (23)$$

The design variables are pressure and temperature at the turbine inlet, and the ratio of pressure differences defining the turbine bleed point $\kappa = \frac{p_7 - p_6}{p_5 - p_6}$. Therefore, the decision variables are $\mathbf{x} \equiv (p_5, T_5, \kappa)^T$, and the parameter selection problem for this configuration can be formulated as follows:

Find \mathbf{x} which

$$\max \dot{W}^{\text{plant}} \quad (24)$$

subject to

$$p_5 \leq p_5^{(U)} \quad (25)$$

$$T_5 \leq T_5^{(U)} \quad (26)$$

$$0 \leq \kappa \leq 1 \quad (27)$$

5.4. Efficiency of heat-to-power conversion process

The expression describing thermal efficiency of the cycle is

$$\eta_{\text{thermal}} = \frac{\dot{W}^{\text{plant}}}{\dot{Q}_{\text{EG}}} \quad (28)$$

where \dot{Q}_{EG} is the rate of heat delivered by the exhaust gas stream. The rate of the exhaust heat was calculated from the equation

$$\dot{Q}_{\text{EG}} = \dot{m}_{\text{EG}} (h_{1\text{EG}} - h_{2\text{EG}}) \quad (29)$$

where \dot{m}_{EG} is the gas-turbine exhaust gases flow rate, $h_{1\text{EG}}$ is the enthalpy of exhaust gases at heat recovery oil heater inlet, and $h_{2\text{EG}}$ is the enthalpy of exhaust gases at heat recovery oil heater outlet.

The overall system exergy efficiency is given by the ratio of the system total exergy used to the system total exergy available. System total exergy used is assumed to be equal to the amount of the recovered exergy in the form of the produced net power output, and total exergy available is assumed to be the supplied exergy of the exhaust heat stream

$$\eta_{\text{exergy}} = \frac{\dot{W}^{\text{plant}}}{\dot{E}_{1\text{EG}}} \quad (30)$$

Neglecting the effects of kinetic and potential exergies, the physical part of exergy associated with exhaust gas flow is expressed as

$$\dot{E}_{1\text{EG}} = \dot{m}_{\text{EG}} \left[(h_{1\text{EG}} - h_{0\text{EG}}) + T_0 (s_{1\text{EG}} - s_{0\text{EG}}) \right] \quad (31)$$

The exergy of the air stream at the condenser inlet is not taken into account in calculation of the available exergy of the system, since the air fan is assumed to be within the system's boundary, as included in Eq. (10).

When analysing the performance of waste heat recovery plant on a thermodynamic basis, the ultimate goal is to maximize the rate of recovered exergy. For a given rate of exergy of the exhaust heat source, the net power output is correlated with exergy efficiency; the cycle with the highest net power output is the one that converts the most of the available exergy of the heat source into mechanical work. The net mechanical work output is correlated with thermal efficiency of the cycle, provided that a fixed amount of source heat is considered. However, the amount of source heat, determined from the initial and final enthalpy of the exhaust gases in the heat recovery oil heater is conditioned by the processes taking place within the ORC system, which is why thermal efficiency is not strongly correlated with exergy efficiency.

6. Conclusions

The first part of this paper has introduced a modelling framework for ORC systems in which the performance of the equipment is given as an input data. The models represent a generic formulation that can be incorporated into a global optimisation routine to address the preliminary design problems of the energy recovery systems. The design variables of a certain waste energy recovery problem (i.e. equipment sizes) can be optimised to obtain a solution that maximizes a given task objective (e.g. net power output, cycle thermal efficiency, exergy efficiency of the system). In the second part of this paper, the implementation of this modelling framework to address the design of ORC plant in natural gas compressor station will be presented.

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