UTILIZATION OF WASTE HEAT ENERGY OF THE EXHAUST GASES IN A DIESEL ENGINE USING THE ORC SYSTEM

Łukasz Buczek, Dmytro Samoilenko

Abstract

Thermal energy utilization systems using Organic Rankine Cycle (ORC systems) can be a way to increase the overall efficiency of internal combustion engines (ICE), and thus at the same time to reduce the emission of harmful compounds to the environment. The main two advantages of ORC systems are: the use of thermal energy, which is dissipated into the environment in the form of heat, i.e. fuel energy, which is not used by the ICE, and the lack of interference in the operation of the ICE. Additionally, high efficiency, low construction costs and high compatibility and flexibility of ORC systems mean that their installation on ICE exhaust systems is economically justified and simple. In the article below, the legitimacy of considering the above-mentioned solution was proven, the concept of an ORC system for a laboratory ICE was proposed, a diagram of the procedure during the design/construction of the system was presented, as well as the initial energy balance of the solution.

Keywords

diesel engine, ORC system, test stand

List of the most important abbreviations

ORC system – a system that uses the Organic Rankine Cycle
ICE – internal combustion engine

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Introduction

In the era of the growing problem of high air pollution, all sectors of the economy are constantly working to reduce the emission of harmful compounds to the environment. In 2019, in the European Union, the transport industry was responsible for 31% of the total CO_2 emissions to the environment, and in Poland for 21% [1]. These facts and the fact that the waste energy from the internal combustion engine dissipated by the exhaust system reaches even 30% of the total energy supplied to it, motivate to take up the topic of utilization of thermal energy dissipated by exhaust systems of internal combustion engines. Improving the heat balance of an internal combustion engine in favour of its efficiency brings measurable benefits in the form of savings due to lower fuel consumption, which is associated with lower emission of harmful compounds to the environment. Considering how much of the energy generated in the engine from the fuel is lost in the exhaust system, it can be assumed that even the use of a waste energy recovery system with a relatively low overall efficiency will be economically viable.

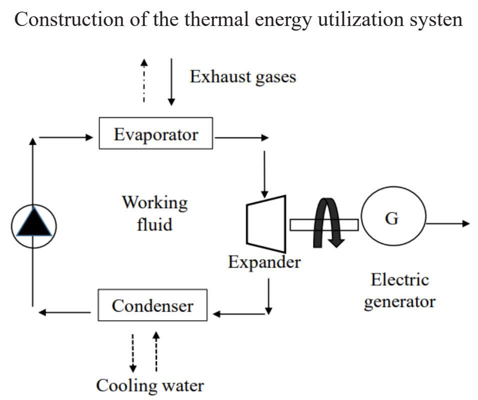
The compression-ignition engine exhaust temperature in the exhaust manifold reaches 700°C and decreases as it flows through the above-mentioned manifold, turbocharger and exhaust gas treatment systems. The temperature downstream of the exhaust after-treatment systems is between 200°C and 300°C. Therefore, the collection of thermal energy can only be realized by means of a system that allows energy transfer from low-temperature heat sources. One such system, the use of which has many advantages, is the system that uses the organic Rankine cycle.

Advantages of the ORC system

The ORC system is characterized by properties that make it suitable for use in the system for utilizing thermal energy from exhaust systems of internal combustion engines:

- it cooperates with low-temperature heat sources (work in the temperature range of the heat source: 100°C – 1050°C),
- works with time-varying parameters (temperature and exhaust gas flow rate) of the heat source,
- has high efficiency,
- its operation does not interfere with the operation of the internal combustion engine,
- it is flexible when it comes to its configuration of its installation,
- its maintenance costs are low – its components are characterized by low failure rate, and the system requires little maintenance,
- its degree of complexity is low, which translates into low construction costs.

Fig. 1. Schematic diagram of the ORC system for heat energy utilization [2]

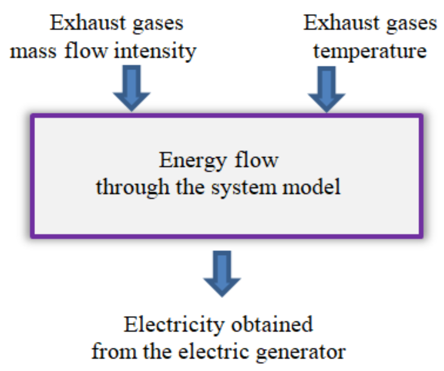


A typical ORC system consists of the components shown in Figure 1. These are: working liquid pump, evaporator, gas turbine, electric generator, condenser, hydraulic lines and working medium.

Thermodynamic model of the thermal energy utilization system

From the point of view of the thermodynamic model of the thermal energy utilization system presented in Figure 2, the main engine parameters that affect the energy utilized are: exhaust gas mass flow rate and exhaust gas temperature. The exhaust mass flow rate is calculated from the physical parameters and engine operating parameters, while the exhaust gas temperature is measured with a thermocouple.

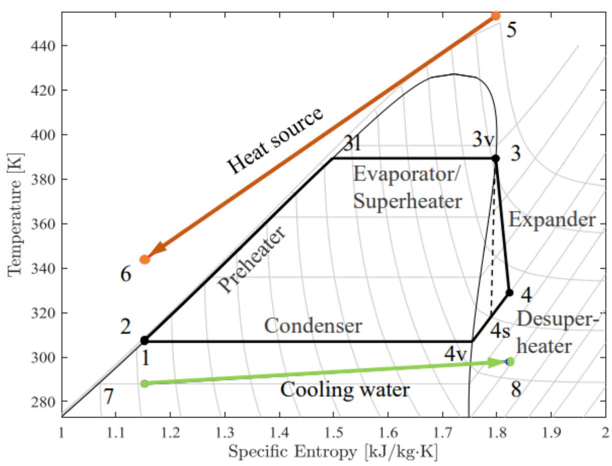
Fig. 2. Model including input and output data of the ORC system



Source: own study.

The operation of the ORC system is illustrated by the graph of temperature as a function of entropy, presented in Figure 3. Between the points 1 and 2 on the graph, the working liquid is pumped by the pump. Between points 2 and 3, the working liquid is heated and evaporated in the evaporator. Between points 3 and 4, the gas energy is transferred to the gas turbine, which drives the coupled electric generator. Between points 4 and 1, the gas is cooled and condensed in the condenser.

Fig. 3. Temperature-entropy diagram for the ORC system [2]



The curve between points 5 and 6 shows the change in temperature and entropy of the heat source (exhaust gases of the internal combustion engine), while the curve between points 7 and 8 shows the change in temperature and entropy of the coolant used to cool the working medium of the system in the condenser.

The description of the thermodynamic model of the system is a set of equations describing: heat fluxes transferred by individual elements of the system and the exergy of individual elements of the system. The following output equations can be used to make a preliminary estimate of the energy recovered with the ORC system:

$$\dot{Q}_{evap} = \dot{m}_{wf}(h_3 - h_2) = \dot{m}_{hs}C_{p,ex}(T_5 - T_6) \quad (1) \qquad [2],$$

$$\dot{W}_{exp,ind} = \dot{m}_{wf}\left(h_3 - h_{4,is}\right)\eta_{is,ind} \cong \dot{m}_{wf}(h_3 - h_4) \quad (2) \qquad [2],$$

where: $\dot{m}_{wf}$ – mass flow rate of the system working fluid; h – working fluid enthalpy in the selected system operating point; $\dot{m}_{hs}$ – exhausts mass flow intensity; $C_{p,ex}$ – specific heat of exhaust gases; T – exhaust gas temperature at the selected system operating point; $\eta_{is,ind}$ – isentropic efficiency of a gas turbine.

Equation no. 1 is the equation describing the heat flux flowing through the system's evaporator, and the equation no. 2 is the equation describing the heat flux flow through the system's gas turbine. After taking into account the efficiency of the gas turbine and the electric generator coupled with the gas turbine, the energy recovered by means of the ORC system can be calculated.

Research object

Measurements of temperature and mass flow intensity of the exhaust gases were made during empirical tests of a V-type diesel engine equipped with a prototype of a controlled turbocharger with variable geometrical parameters. The centrifugal turbine of the turbocharger was controlled by changing the surface area of the smallest cross-section of the inlet channel (A) in the bladeless deflector [3]. The specification of the test object is presented in Table 1. By using the measurements from the tests carried out on the engine with the built-in turbocharger prototype, it is possible to investigate the effect of temperature and mass flow intensity of exhaust gas on the value of energy recovered from the exhaust system and to create a preliminary design of the ORC system based on the real data, without making additional measurements.

Tab. 1. Specification of a V-type diesel engine [3]

Parameter	Value
Rated power	125 kW
RPM for rated power	2100 RPM.
Number of cylinders/engine type	6/ V-shaped
Piston diameter	130 mm
Piston stroke	115 mm
Boosting system	Turbocharger with air cooler
Fuel supply system	Direct injection with Bosch high pressure pump
Engine cubic capacity	9.15 dm^3

Calculation of energy recovered using the ORC system

In order to calculate the energy recovered using the ORC system, one needs to enter the values of individual parameters into equations 1 and 2. The values of mass flow intensities and exhaust gas temperatures for various control settings of the variable geometry of the turbocharger are presented in Table 2.

Tab. 2. Parameters of exhaust gases at the outlet of a controlled turbine with engine operation parameters: Ne = 100 kW and n = 1550 rpm (T_0 = 297°C; P_0 = 99458 Pa) [3]

Parameter		The area of the smallest cross section of the turbine inlet channel A, mm^2		
		A = 2312	A = 2065	A = 1442
Exhausts mass flow intensity, kg/s	$\dot{m}_{hs}$	0.159	0.168	0.179
Exhausts pressure at the outlet of the turbine, Pa	p_{OT}	99835.4	99835.4	99835.4
Exhausts temperature at the turbine outlet, K	T_5	853	822	796.2
Speed of exhaust gases at the outlet from the turbine, m/s	C_{OT}	63.6	64.6	66.5

The missing values necessary for the calculation of energy were selected based on the literature sources using interpolation, taking into account the parameters of the engine on which the measurements were made. For example, with the exhaust gas temperature at the outlet of the turbine T5 = 853 K (A = 2312 mm^2) the following necessary values can be assumed:

$T_6 = 753\ [K]$ – temperature of the heat source when the system is not operating,
$T_3 = 793\ [K]$ (for calculations it was assumed that the degree of overheating is 0, so $T_3 = T_{3l}$),
$T_{4s} = 728\ [K]$,
$T_{2a} = 736\ [K]$,
$\eta_{exp} = 0{,}70\ [-]$ – gas turbine efficiency,
$\eta_{elg} = 0{,}93\ [-]$ – efficiency of the electric generator.

In order to estimate the energy recovered by means of the ORC system, a program was written in the MATLAB software (Figure 4), which enables to examine the influence of modifying the values of individual parameters of the system, on the final result (the value of the energy obtained on the electric generator of the system).

Fig. 4. Part of the program code for estimating energy recovered using the ORC system

```
% The efficiency of individual elements of the system
n_r=0.80;      % recuperator effectiveness
n_p=0.65;      % pump isentropic efficiency
n_exp=0.70;    % expander isentropic efficiency
n_elg=0.93;    % electric generator efficiency

% The heat input (rate)
m_hs=0.159                      % [kg/s]
                                % from the measurements
Cp_ex=1;                        % [kJ/(kg*K)]
                                % assumption: constant during this process
                                % variable depending on exhaust gas composition
T_5=853;                        % [K]
                                % temperature of the heat source when the ORC system is
                                % not working
                                % from the measurements
T_6=753;                        % [K]
                                % temperature of the heat source when the ORC system is
                                % working
                                % estimation based on the literature
Q_evap=m_hs*Cp_ex*(T_5-T_6)     % [kJ/s]

% The expander work
T_3=793;                                        % [K]
T_3l=T_3;                                       % [K]
                                                % assumption: SHD=0 (degree of
                                                % superheat)
T_4s=728;                                       % [K]
                                                % estimation based on the literature
Cp_wf                                           % [kJ/(kg*K)]
                                                % assumption: constant during this
                                                % process
T_2a=736;                                       % [K]
                                                % estimation based on the literature
m_wf=Q_evap/(Cp_wf*(T_3-T_2a))                  % [kg/s]
W_exp=m_wf*Cp_wf*(T_3-T_4s);                    % [kJ/s]
W_exp_elg=m_wf*Cp_wf*(T_3-T_4s)*n_exp*n_elg     % [kJ/s]
                                                % electric generator energy
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Source: own study.

The results generated by the above-mentioned program are presented in Table 3. The most important, from the point of view of the economic use of the system is the parameter marked W_exp_elg, which is equal to the power obtained on the electric generator and for the presented example amounts to ~ 12 kW. The result is therefore very promising, but it must be borne in mind that it is based on estimated calculations and it is known that the power obtained will be lower in the case of an engine equipped with an exhaust gas cleaning system.

Tab. 3. Results generated by the program for estimating the energy recovered by means of the ORC system for individual settings of the controlled turbine with the engine operating parameters: Ne = 100 kW and n = 1550 RPM (T_0 = 297°C; P_0 = 99458 Pa)

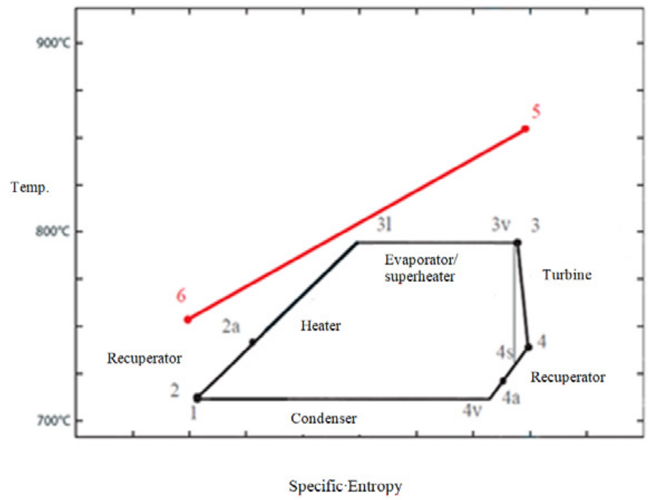
Parameter		**The area of the smallest cross section of the turbine inlet channel** ***A*****, mm²**		
		A **= 2312**	A **= 2065**	A **= 1442**
Exhausts mass flow intensity, kg/s	$\dot{m}_{hs}$	0.159	0.168	0.179
Exhausts gas temperature at the outlet of the turbine, K	T_5	853	822	796.2
The heat flux flowing through the system evaporator, kJ/s	$\dot{Q}_{evap}$	15.90	11.59	7.73
Mass flow intensity of the working liquid, kg/s	$\dot{m}_{wf}$	0.28	0.20	0.14
Electrical energy obtained from the generator, kW	W_{exp_elg}	11.80	8.61	5.74

Source: own study.

The results of estimated recovered energy calculations for three different settings of a turbocharger with variable geometry are presented in the last row of Table 3. It is easy to see that as the temperature of the heat source decreases, the value of the recovered energy decreases, and with a temperature drop of ~ 60°C, decreases of the energy recovered is ~ 6 kW. The second conclusion that arises is that the change in exhaust mass flow intensity compared to the temperature change is less significant from the point of view of the energy recovery system. During measurements, the mass flow intensity increases while the temperature of the heat source decreases, which results in the value of the energy recovered also decreasing. After taking into account the highest value of flow mass intensity m_hs = 0.179 kg/s and the highest value of exhaust gas temperature T_5 = 853 K, the value of recovered energy will be ~ 12 kW, which means that the change will be ~ 1 kW compared to the results for m_hs = 0.159 kg/s and T_5 = 853 K. Ultimately, the amount of energy recovered is directly proportional to the temperature difference of the heat source when it is in operation and when it is not in operation as well as to the mass flow intensity of the heat source.

Figure 5 shows a temperature graph as a function of entropy for the ORC system, which operates at the exhaust gas temperature at the turbine outlet T5 = 853 K (value A = 2312 mm^2 in a controlled turbine).

Fig. 5. Temperature-entropy diagram of the ORC system for the turbine setting A = 2312 mm^2



Source: own study.

Test stand for installing the ORC system

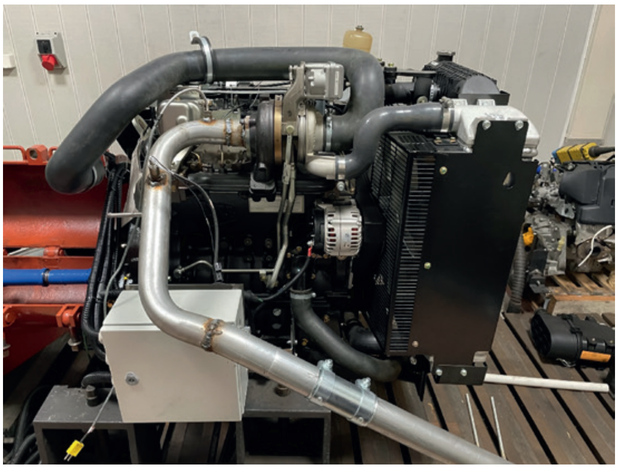
The engine, on which the tested system, for the utilization of waste energy from the exhaust system will be installed, is a compression-ignition 444 TA4i-81 I1 engine manufactured by Perkins, of 4.4 l displacement, maximum power 81 kW (at 2200 rpm) and a maximum torque of 420 Nm (at 1400 rpm). The engine is designed for industrial applications (mainly as a power unit for various types of working machines) and meets the European standard for the emission of toxic compounds in exhaust gases Stage IIIB and American Tier 4. It has been installed on the engine dynamometer in the Laboratory of the Vehicle Institute, as shown in the Figure. 6.

Fig. 6. 444 TA4i-81 I1 engine by Perkins in the IPiMR laboratory

Source: own study.

The engine installed in the laboratory does not have a «classic» exhaust system, it has been modified for the needs of the engine installation. Therefore, it is devoid of the exhaust gas treatment system and has a unique geometry, which is shown in Figure 7. This primarily causes that the temperature of the exhaust gases that will flow through the heat exchanger of the ORC system will be higher than for the exhaust system with the exhaust gas treatment system. This will not have a negative impact on the tests, because it will only affect the physical parameters of individual elements of the ORC system, while the principle of its operation and the procedure for the preparation, implementation and development of the experiment results will not change in relation to the engine with the exhaust gas treatment system.

Fig. 7. Exhaust system of the laboratory engine



Source: own study.

The tests course of action

It is obvious that before commencing any considerations, it is necessary to first confirm their legitimacy. In the case of the ORC system, the legitimacy of taking up the topic is confirmed by: the need for systems limiting the degradation of the natural environment, as described in the introduction, the advantages of the ORC system (section: Advantages of the ORC system) and the economic and energy analysis of the model, which shows that its construction is profitable. More precisely, the costs of building the system get repaid after a certain period of its operation, while the materials and energy used for its construction are much less detrimental to the environment than the energy utilized through the exhaust systems of internal combustion engines.

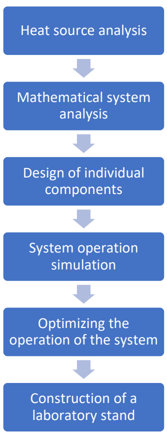
A simplified course of action in this experiment is shown in the diagram in Figure 8. The first point is the analysis of the heat source. Tt means, first of all, estimating what energy potential the heat source gives us, i.e. how much energy we are able to receive from it without disrupting its operation, and assessing whether and how we are able to install an ORC system on it. The article analyses the energy potential of the «twin» engine, which confirms the legitimacy of taking up the topic, while when it comes to the installation, the exhaust system of the laboratory engine fully allows for the installation of the thermal energy utilization system. As far as the installation on the exhaust systems with exhaust gas treatment systems is concerned, it is also possible thanks to the flexibility of the ORC system.

The second point in the diagram is the mathematical analysis of the system, which was conducted in the chapter: Calculation of energy recovered using the ORC system. Initial estimates of the energy recovered look very promising and encourage further development.

The next step that will be performed will be modelling the individual elements of the system. The first and most important will be the system heat exchanger mounted on the exhaust system of the internal combustion engine. Depending on its parameters, other components of the system will be selected.

Before starting to manufacture the elements, the simulation of selected elements and the operation of the system in ANSYS will be carried out, which will allow for a thorough check of the operation of the system and, at the modelling stage, modify its elements in such a way as to optimize its operation as much as possible.

Fig. 8. Plan of the experiment



Source: own study.

Conclusions

Based on the preliminary analysis of the ORC system as a method of utilizing the thermal energy dissipated by the combustion engine exhaust system, it can be concluded that the use of the system is justified because:

- it allows to utilize the unused energy from fuel, which is tantamount to increasing the efficiency of the entire system and reducing the emission of toxic compounds to the environment,
- it has a relatively high efficiency,
- it works with heat sources of variable parameters, which makes it perfect for cooperation with internal combustion engines,
- it does not interfere with the operation of the heat source, i.e. the internal combustion engine,
- its dimensions and flexibility in terms of construction allow to be fitted even in a mobile vehicle,
- the costs of its construction get repaid after a relatively short period of its operation,
- its components are not toxic to the environment and there are no problems with their subsequent disposal,
- its maintenance costs are very low.

After the initial analysis of the heat source, it was estimated that at the output of the ORC system, even ~ 12 kW of electrical power can be obtained with a power of ~ 100 kW of the combustion engine as a heat source. Estimates are subject to an approximation error for individual values, therefore the expected real results are about half lower and will be better approximated after designing the geometry of individual components of the ORC system and after conducting computer simulations of the ORC system operation.

The estimated calculations were carried out without taking into account the temperature drops in the exhaust gas treatment system, which the laboratory engine does not have. It should be noted, however, that the expected results for the engine with the exhaust gas treatment system built-in the exhaust system are also satisfactory.

During the calculations, it was noticed that as the temperature of the heat source decreases, the value of the energy recovered decreases, and that the change in the mass flow intensity of the exhaust gas compared to the temperature change is less significant from the point of view of the energy recovery system. During measurements, the mass flow intensity increases while the temperature of the heat source decreases, which results in the value of the energy recovered also decreasing. Ultimately, the amount of energy recovered is directly proportional both, to the temperature difference of the heat source when in and out of operation and to the mass flow intensity of the heat source.

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The next step in the research will be to design the geometry of the ORC system heat exchanger and to select other components of the system from among the solutions available on the market.

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