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The IC engine energetically combined with the steam micro-turbine

Abstract: The paper presents results of modeling a system for waste heat recovery from a cooling agent and exhaust gases in an IC (internal combustion) engine. Basic element of a system is a microturbine engine working in the Rankine cycle based system. Steam was applied as the working fluid in this system. Additionally, this waste heat recovery system consists of: exhaust gases – water- and engine coolant-water heat recuperators, a pump and a condenser. It was concluded, on the basis of thermodynamic analysis and energy balance, that the break power of the steam microturbine can reach 29% of the break power generated by the IC engine under optimal operation regime. It is significantly more than maximum benefit of approx. 7% in recovering waste energy obtained by applying a gas power turbine in the IC engine exhaust. The system described in this paper can be applied in high power output IC engines both tractional and stationary.

Key words: *IC engine, waste heat recovery, steam turbine, Rankine cycle*

Silnik tłokowy skojarzony energetycznie z parową mikroturbiną

Streszczenie: W artykule opisano model systemu energetycznego złożonego z tłokowego silnika spalinowego i mikroturbiny parowej pracującej w obiegu Rankina wykorzystującym odpadowe ciepła silnika tłokowego. W skład tego systemu oprócz silnika tłokowego wchodzą: dwa wymienniki ciepła (ciecz chłodząca silnik - medium robocze i spaliny silnika - medium robocze) konwertujące odpadowe ciepła z silnika do obiegu Rankina, mikroturbina parowa, skraplacz par medium roboczego i pompa obiegowa. Analiza numeryczna modelu tego systemu wykazala, że w optymalnych warunkach pracy systemu moc efektywna mikroturbiny parowej może osiągać do 29% mocy efektywnej silnika tłokowego. Jest to znacznie więcej niż można uzyskać stosując w silniku dodatkową turbinę mocy zasilaną spalinami i osiągającą moc nie przekraczającą ok. 7% mocy silnika tlokowego. Układ taki może znaleźć zastosowanie głównie w silnikach dużej mocy zasilanych paliwem ciekłym lub gazowym.

Słowa kluczowe: *silnik spalinowy, odzysk ciepła, turbina parowa, obieg Rankina*

1. Introduction

Among all the thermal machinery the internal combustion reciprocating engine is such a machine, which can convert chemical energy stored in fuel to useful work with the highest overall efficiency under the wide range of output power. This efficiency can exceed 45% as it is typical for high power natural gas fueled stationary engines working in CHP (Cogeneration of Heat and Power) sets. The rest of energy is heat, which sources are as follows: 20-27% exhaust gases, 12-16% engine cooling water, 4-7% lubricating oil cooling system, 8-12% intercooler. For instance the J420-GSA25 gaseous engine by Jenbacher while fueling biogas with chemical energy rate of 3.4 MW produces 1540 kW useful work with the overall efficiency of 45% whereas the waste heat of 1410 kW from the engine to external heating system stands for 41.5% with respect to the energy in the fuel flow. A power system consisted of the IC engine and a steam turbine working at the Rankine cycle for rational utilization of engine waste heat is particularly important in case the engine waste heat cannot be applied for heating or technological purposes. As example of

practical utilization of the engine waste heat an absorption refrigerator working on engine exhaust gases heat was described in [2]. The issues concerning effective engine waste heat recovery is the theme of the Sustainable Surface Transport (SST)-2011-RTD-1 competition under the 7th Framework Programme of the EU. The challenge within frames of this competition requires minimum 10% increase in engine overall efficiency by applying a waste heat recovery units. It is mainly focused on improvement the Rankine cycle based system working with water or other low-boiling fluids. However different solution of engine waste heat recuperation are also in use. It is confirmed, on the basis of computational analysis, that application of a heat pump is effective way to recover this waste heat [3]. Moreover, research on thermoelectric generators (TEG), directly converting heat into power, is also conducted. TEGs work effectively at high temperature of exhaust gases [4-7]. Such an investigation on TEG have been also carried out in Polish research centers [8]. Possibility of recovery both high temperature heat from exhaust gases and low temperature heat from cooling systems is the main advantage of applying the Rankine system for this

purpose. Results of research in this field confirm usefulness of the Rankine system for waste heat recovery in the IC engine [9]. Another solution is recovery exhaust heat and convert it to power directly through installing on the exhaust outlet a power generator and a gas turbine engine combined together as a single unit [10] or through injection of superheated steam, obtained in highly efficient exhaust-water evaporators, into a cylinder of a reciprocating engine [11]. Nowadays, systems for waste heat recovery are in common use in engines driving large ships and boats [12].

The mathematical model of the power system including the Rankine system with the steam tur-

bine and the IC engine is described in the paper. Additionally, potential improvement in overall efficiency by applying the Rankine system with a steam microturbine and heat exchangers is analyzed and discussed.

2. Description of the thermodynamic model of the heat recovery

In the figure 1 the scheme of the power system consisted of the reciprocating IC engine and the Rankine system with a steam turbine, low- and high-temperature recovery units (heat exchangers) is presented.
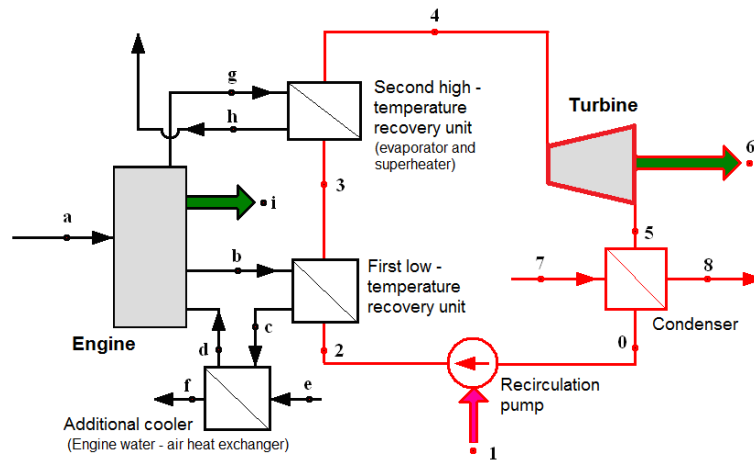

Fig.1. Scheme of the combined IC engine-turbine system based on the Rankine system for waste heat recovery from both exhaust gases and the engine cooling system of the IC engine

2.1. Energy balance for the IC engine

The energy rate N_{pal} present in the fuel flowrate $\overset{\circ}{m}_{pal}$ consumed by the engine can be determined as follows (LHV_f – lower heating value of the fuel):

$$N_{pal} = LHV_f \cdot \overset{\circ}{m}_{pal} \qquad (1)$$

The coefficient ξ_{ch} of heat losses from the engine cooling system expressed by the equation

$$\xi_{ch} = \frac{N_{ciep\ chl}}{N_{pal}} \qquad (2)$$

is ratio of the total heat (cooling of heads, cylinder liners and lub. oil) denoted as $N_{ciep\ chl}$ transferred to the engine cooling system upon the chemical energy rate N_{pal} stored in fuel flow.

The coefficient ξ_{spal} of heat losses from the hot exhaust gases given by

$$\xi_{spal} = \frac{N_{ciep\ spal}}{N_{pal}} \qquad (3)$$

is expressed as ratio of heat $N_{ciep\ spal}$ in the exhaust gases flow after a turbocharger upon the N_{pal}.

The coefficient of other losses ξ_{inne} in the IC engine identified as

$$\xi_{inne} = \frac{N_{inne}}{N_{pal}} \qquad (4)$$

is defined as ratio of other power losses N_{inne} over the N_{pal}. Thus, the break efficiency η_e of the IC engine can be determined by the following equation

$$\eta_e = 1 - \xi_{chl} - \xi_{spal} - \xi_{inne} \qquad (5)$$

which takes into account all the coefficients of losses from the cooling system ξ_{chl}, exhaust gases ξ_{spal} and other sources ξ_{inne}. The break power N_e of the engine can be determined as product of the break efficiency η_e and the energy rate N_{pal} in the fuel flow burnt in the engine (6).

$$N_e = \eta_e \cdot N_{pal} = \eta_e \cdot LHV_f \cdot \overset{\circ}{m}_{pal} \qquad (6)$$

The coefficient x given by the equation

$$x = \frac{\overset{\circ}{m}_m}{\overset{\circ}{m}_{pal}} \qquad (7)$$

is introduced to make thermodynamic analysis of the combined heat and power system consisted of the IC engine and the steam turbine working in the Rankine cycle. The x is the ratio of the Rankine fluid flowrate $\overset{\circ}{m}_m$ upon the mass fuel flowrate $\overset{\circ}{m}_{pal}$. In this analysis a fluid applied for the Rankine system is water/steam.

2.2. Condenser

In the condenser the Rankine fluid (water) condenses itself. Temperature t_0 of the condensed fluid is assumed. If the whole fluid (steam) is condensed into liquid (water) in the condenser (dryness fraction X = 0), then enthalpy of the condensate at temperature t_0 can be determined from the product of the specific heat $c_{p\,m}$ of the condensate and its temperature t_0 (8).

$$i_{m0} = c_{p\,m} \cdot t_0 \qquad (8)$$

Someone assumes that absolute pressure p_0 of the condensate in the condenser equals pressure p_5 in the outlet of the turbine and can be determined as saturation pressure of the Rankine fluid at temperature t_0 (9).

$$p_5 = p_0 = p_{satur(t=t_0)} \qquad (9)$$

2.3. Recirculation pump

The recirculation pump pumps incompressible fluid with the volumetric flowrate $\overset{\circ}{V}_m$ and density ρ_m from the condenser (pressure p_0) through heat recovery units (pressure $p_2 = p_3$) to the turbine (pressure p_4). The overall efficiency of the pump is denoted here as η_p. It is assumed that there is no pressure drop between the pump output and the turbine input, so it yields:

$$p_2 = p_3 = p_4 \qquad (10)$$

The pump requires the power N_l to drive it. It can be calculated as follows:

$$N_1 = \frac{1}{\eta_p} \cdot \overset{\circ}{V}_m \cdot (p_2 - p_0) = \frac{1}{\eta_p} \cdot \frac{\overset{\circ}{m}_m}{\rho_m} \cdot (p_2 - p_0)$$
$$= \frac{1}{\eta_p} \cdot \frac{x \cdot \overset{\circ}{m}_{pal}}{\rho_m} \cdot (p_2 - p_0) \qquad (11)$$

Total power losses ΔN_l in the pump expressed by the equation (12)

$$\Delta N_1 = (1 - \eta_p) \cdot N_1 \qquad (12)$$

are converted into heat, which is absorbed by the fluid recirculated in the Rankine system. This heat increases the total enthalpy of the fluid from i_{m0} at the condenser exit to i_{m2} at the pump outlet. The total enthalpy is given by the equation (13)

$$i = c_p \cdot T + \frac{w^2}{2} = c_p \cdot T_{spietrz} \qquad (13)$$

where: c_p is a specific heat at constant pressure, T – temperature in K, w – velocity of the flow and $T_{spietrz}$ – substitute temperature in K.

On the basis of the energy balance for the pump identified as

$$\overset{\circ}{m}_m \cdot (i_{m2} - i_{m0}) = \Delta N_1 = (1 - \eta_p) \cdot N_1 = \frac{1 - \eta_p}{\eta_p} \cdot \frac{\overset{\circ}{m}_m}{\rho_m} \cdot (p_2 - p_0) \qquad (14)$$

the enthalpy i_{m2} can be determined as follows

$$i_{m2} = i_{m0} + \frac{1 - \eta_p}{\eta_p} \cdot \frac{p_2 - p_0}{\rho_m} \qquad (15)$$

2.4. Heat balance for the first low-temperature recovery unit

The low-temperature recovery unit is to recover heat from the engine cooling agent (water) and transfer it to the Rankine fluid (water). It warms up the fluid from temperature t_2 to temperature t_3 (at the unit exit) under constant pressure $p_2 = p_3$. Following the rules of heat transfer, temperature of the fluid cannot exceed maximal temperature t_b of the cooling water from the engine. These temperatures can at most equal each other at a highly efficient heat exchanger with efficiency η_{wm}. Thus, enthalpy i_{m3} of the fluid warmed up in this first recovery unit can reach the enthalpy at temperature t_b which is temperature of hot water from the engine cooling system (16).

$$i_{m3} \le i_m\big|_{(t_3 = t_b)} \qquad (16)$$

Thermal power N_w from the combustion process to the cooling water (17)

$$N_w = \xi_{chl} \cdot LHV_f \cdot \overset{\circ}{m}_{pal} \qquad (17)$$

usually cannot be completely transferred to the Rankine fluid. Useful part $N_{u\,w}$ of this power, which can be transferred to the fluid, is expressed by the equation (18)

$$N_{u\,w} = \eta_{wm} \cdot \xi_{chl} \cdot LHV_f \cdot \overset{\circ}{m}_{pal} \cdot \frac{i_{w(t=t_b)} - i_{w(t=t_2)}}{i_{w(t=t_b)} - i_{w(t=0\,C)}} \qquad (18)$$

and depends on the efficiency η_{wm} of the recovery unit and the enthalpy change from temperature t_b of the engine hot water to temperature t_2 of the cold fluid in the Rankine system.

Thermal power $N_{u\,m(2 \div 3)}$ taken by the fluid in the first recovery unit (19)

$$N_{u\,m(2\div3)} = \overset{\circ}{m}_m \cdot (i_{m3} - i_{m2}) = \overset{\circ}{m}_{pal} \cdot x \cdot (i_{m3} - i_{m2}) \quad (19)$$

depends on the fluid flow rate $\overset{\circ}{m}_m$ and total enthalpy change of the fluid from temperature t_2 of the cold fluid at the unit entrance to temperature t_3 at the unit exit. Potentially the maximal thermal power $N_{max\,m(2\div3)}$ expressed by the equation (20)

$$N_{\max m(2\div3)} = \overset{\circ}{m}_m \cdot (i_{m3(t=t_b)} - i_{m2(t=t_2)})$$
$$= \overset{\circ}{m}_{pal} \cdot x \cdot (i_{m3(t=t_b)} - i_{m2(t=t_2)}) \quad (20)$$

can be obtained if temperature t_3 of the warmed fluid equals temperature t_b of the hot water delivered from the engine to the recovery unit.

There are 3 cases, with respect to heat balance, in relation between the useful thermal power $N_{u\,w}$ carried by the cooling water recirculated in the first recovery unit and the maximal power $N_{max\,m(2\div3)}$ available to be absorbed by the Rankine fluid.

Case 1: If the power $N_{u\,w}$ is higher than the maximal power $N_{max\,m(2\div3)}$ (21, 22, 23)

$$N_{u\,w} > N_{\max m(2\div3)} \quad (21)$$

$$\eta_{wm} \cdot \xi_{chl} \cdot LHV_f \cdot \overset{\circ}{m}_{pal} \cdot \frac{i_{w(t=t_b)} - i_{w(t=t_2)}}{i_{w(t=t_b)} - i_{w(t=0\,C)}} > \quad (22)$$
$$> \overset{\circ}{m}_{pal} \cdot x \cdot (i_{m(t=t_b)} - i_{m(t=t_2)})$$

$$\eta_{wm} \cdot \xi_{chl} \cdot LHV_f \cdot \frac{i_{w(t=t_b)} - i_{w(t=t_2)}}{i_{w(t=t_b)} - i_{w(t=0\,C)}} > x \cdot (i_{m(t=t_b)} - i_{m(t=t_2)})$$
$$(23)$$

then only part of the heat from the engine cooling system can be utilized for warming up the fluid and heat excess which is not absorbed by the Rankine fluid has to be utilized outside the Rankine system or dissipated by an additional cooler, which is necessary in this case.

Case 2: If the thermal power $N_{u\,w}$ equals the maximal thermal power $N_{max\,m(2\div3)}$,

$$N_{u\,w} = N_{\max m(2\div3)} \quad (24)$$

then the whole energy from the engine cooling water can be transferred to the Rankine system and an additional cooler is not required.

In both cases unknown temperature t_3 of the fluid is the same as known temperature t_b of hot water from the engine if someone assumes the recovery unit is an ideal counter-current heat exchanger with high efficiency. Then, total enthalpy of the fluid from the recovery unit can be computed as follows (25)

$$i_{m3} = i_{m(t=t_b)} \quad (25)$$

Case 3: If the useful power $N_{u\,w}$ is lower than the maximal power $N_{max\,m(2\div3)}$, available in the Rankine system (26, 27)

$$N_{u\,w} < N_{\max m(2\div3)} \quad (26)$$

$$\eta_{wm} \cdot \xi_{chl} \cdot LHV_f \cdot \frac{i_{w(t=t_b)} - i_{w(t=t_2)}}{i_{w(t=t_b)} - i_{w(t=0\,C)}} < x \cdot (i_{m(t=t_b)} - i_{m(t=t_2)})$$
$$(27)$$

than the whole heat from the engine cooling system can be utilized to warm up the fluid in the Rankine system, so an additional cooler is not required. Temperature t_3 of the fluid will be lower than temperature t_b of the hot water from the engine in this case. The total enthalpy of the fluid from this recovery unit can be determined on the basis of energy balance (28)

$$N_{u\,w} = N_{u\,m(2\div3)} \quad (28)$$

After substituting appropriate expressions to this equation for energy balance (29)

$$\eta_{wm} \cdot \overset{\circ}{m}_{pal} \cdot LHV_f \cdot \xi_{chl} \cdot \frac{i_{w(t=t_b)} - i_{w(t=t_2)}}{i_{w(t=t_b)} - i_{w(t=0\,C)}}$$
$$= \overset{\circ}{m}_{pal} \cdot x \cdot (i_{m3} - i_{m2}) \quad (29)$$

the total enthalpy i_{m3} of the fluid leaving the first recovery unit can be calculated as follows (30)

$$i_{m3} = i_{m2(t=t_2)} + \frac{\eta_{wm} \cdot \xi_{chl} \cdot LHV_f \cdot \frac{i_{w(t=t_b)} - i_{w(t=t_2)}}{i_{w(t=t_b)} - i_{w(t=0\,C)}}}{x}$$
$$(30)$$

The enthalpy $i_{m2(t=t2)}$, which is required for computing the total enthalpy i_{m3}, is calculated for known temperature t_2 of the condensed fluid (water) at elevated pressure.

In these cases fraction of heat transferred to the Rankine system from the engine cooling system can be determined with the equation (31)

$$\eta_{wyk\,chl} = \frac{N_{u\,m(2\div3)}}{N_w} = \frac{x \cdot (i_{m3} - i_{m2})}{LHV_f \cdot \xi_{chl}} \quad (31)$$

2.5. Heat balance for the second high-temperature recovery unit

The high-temperature recovery unit is to vaporize and warm up the fluid, which preliminary was warmed to temperature t_3 in the first recovery unit, to temperature t_4. As previously assumed following heat transfer principles, temperature t_4 cannot be higher than temperature t_g of hot exhaust gases at the entrance of the second recovery unit. These temperatures can be of the same value if an ideal counter-current heat exchanger of high overall efficiency η_{sm} is taken as the second recovery unit. Thus, enthalpy of the fluid warmed up in such the heat exchanger can be determined on the basis of temperature t_g of hot exhaust gases (32).

$$i_{m4} \leq i_m\big|_{(t_4=t_g)} \quad (32)$$

Thermal power N_s in the exhaust gases determined as follows (33)

$$N_s = \xi_{spal} \cdot LHV_f \cdot \overset{\circ}{m}_{pal} \qquad (33)$$

usually cannot be completely transferred to the fluid in the Rankine system. Useful thermal power $N_{u\,s}$ as a part of this exhaust gases power N_s (34)

$$N_{u\,s} = \eta_{sm} \cdot \xi_{spal} \cdot LHV_f \cdot \overset{\circ}{m}_{pal} \cdot \frac{i_{s(t=t_g)} - i_{s(t=t_3)}}{i_{s(t=t_g)} - i_{s(t=100\,C)}} \qquad (34)$$

which can be absorbed by the fluid in the second high-temperature recovery unit, depends on unit efficiency η_{sm} and the total enthalpy change Δi for exhaust gases at temperature t_g at the entrance of the recovery unit and temperature t_3 of the fluid coming into this unit. As confirmed in the section 2.4, temperature t_3 cannot be higher than temperature t_g of the hot exhaust gases entering the second recovery unit.

Thermal power $N_{u\,m(3\div4)}$ taken in the second recovery unit by the fluid recirculating in the Rankine system identified as (35)

$$N_{u\,m(3\div4)} = \overset{\circ}{m}_m \cdot (i_{m(t=t_4)} - i_{m(t=t_3)})$$
$$= \overset{\circ}{m}_{pal} \cdot x \cdot (i_{m(t=t_4)} - i_{m(t=t_3)}) \qquad (35)$$

depends on the fluid flow rate $\overset{\circ}{m}_m$ and the total enthalpy change of the fluid from temperature t_3 of the preliminary warmed fluid coming to the second recovery unit to temperature t_4 of vaporized and overheated fluid (steam) at the unit exit. Potentially maximal thermal power $N_{max\,m(3\div4)}$ (36)

$$N_{max\,m(3\div4)} = \overset{\circ}{m}_m \cdot (i_{m(t=t_g)} - i_{m(t=t_3)})$$
$$= \overset{\circ}{m}_{pal} \cdot x \cdot (i_{m(t=t_g)} - i_{m(t=t_3)}) \qquad (36)$$

can be obtained if temperature t_4 of vaporized and superheated fluid equals temperature t_g of hot exhaust gases in the second recovery unit.

As discussed in the previous section, here there are also 3 cases corresponding to relation between the useful thermal power $N_{u\,s}$ of the exhaust gases and the maximal thermal power $N_{max\,m(3\div4)}$ available to be absorbed by the Rankine fluid flowing through this recovery unit.

Case 1: If the power $N_{u\,s}$ is higher than the maximal power $N_{max\,m(3\div4)}$ (37, 38, 39)

$$N_{u\,s} > N_{max\,m(3\div4)} \qquad (37)$$

$$\eta_{sm} \cdot \xi_{spal} \cdot LHV_f \cdot \overset{\circ}{m}_{pal} \cdot \frac{i_{s(t=t_g)} - i_{s(t=t_3)}}{i_{s(t=t_g)} - i_{s(t=100\,C)}} > \qquad (38)$$

$$> \overset{\circ}{m}_{pal} \cdot x \cdot (i_{m(t=t_g)} - i_{m(t=t_3)})$$

$$\eta_{sm} \cdot \xi_{spal} \cdot LHV_f \cdot \frac{i_{s(t=t_g)} - i_{s(t=t_3)}}{i_{s(t=t_g)} - i_{s(t=100\,C)}} > x \cdot (i_{m(t=t_g)} - i_{m(t=t_3)})$$
$$(39)$$

then only part of the heat from exhaust gases can be utilized to warm up, vaporize and overheat the fluid and the heat excess has to be utilized outside the Rankine system or thrown out to the environment together with partially cooled exhaust gases.

Case 2: If the power $N_{u\,s}$ equals the maximal power $N_{max\,m(3\div4)}$ (40)

$$N_{u\,s} = N_{max\,m(3\div4)} \qquad (40)$$

then the whole energy from the exhaust gases can be transferred to the Rankine system. In both cases wanted temperature t_4 of the fluid is the same as known temperature t_g of hot exhaust gases from the engine if someone assumes the second recovery unit is an ideal counter-current heat exchanger with high efficiency. Then, total enthalpy of the fluid from this recovery unit can be calculated as follows (41)

$$i_{m4} = i_{m(t=t_g)} \qquad (41)$$

Case 3: If the useful thermal power $N_{u\,s}$ of the exhaust gases is lower than the maximal thermal power $N_{max\,m(3\div4)}$ to be absorbed by the Rankine system (42, 43)

$$N_{u\,s} < N_{max\,m(3\div4)} \qquad (42)$$

$$\eta_{sm} \cdot \xi_{spal} \cdot LHV_f \cdot \frac{i_{s(t=t_g)} - i_{s(t=t_3)}}{i_{s(t=t_g)} - i_{s(t=100\,C)}} < x \cdot (i_{m(t=t_g)} - i_{m(t=t_3)})$$
$$(43)$$

than the whole heat $N_{u\,s}$ from the exhaust gases can be utilized to warm up the fluid in the Rankine system. In this case temperature t_4 of the fluid will be lower than temperature t_g of hot exhaust gases. The enthalpy of the fluid leaving this recovery unit can be determined on the basis of heat balance given by (44)

$$N_{u\,s} = N_{u\,m(3\div4)} \qquad (44)$$

After substituting appropriate expressions to this equation for heat balance in the second recovery unit (45)

$$\eta_{sm} \cdot \overset{\circ}{m}_{pal} \cdot LHV_f \cdot \xi_{spal} \cdot \frac{i_{s(t=t_g)} - i_{s(t=t_3)}}{i_{s(t=t_g)} - i_{s(t=100\,C)}} \qquad (45)$$

$$= \overset{\circ}{m}_{pal} \cdot x \cdot (i_{m4} - i_{m3})$$

the total enthalpy i_{m4} of the Rankine fluid flowing out the second recovery unit can be calculated as follows (46)

$$i_{m4} = i_{m\,3(t=t_3)} + \frac{\eta_{sm} \cdot \xi_{spal} \cdot LHV_f \cdot \dfrac{i_{s(t=t_g)} - i_{s(t=t_3)}}{i_{s(t=t_g)} - i_{s(t=100\,C)}}}{x} \tag{46}$$

The total enthalpy $i_{m3(t=t3)}$, which is required for calculating the total enthalpy i_{m4}, is determined from the heat balance for the first recovery unit (25, 30). Other necessary values should be assumed.

In these cases fraction of heat $\eta_{wyk\,spal}$ transferred from exhaust gases to the Rankine system can be determined with the equation (47)

$$\eta_{wyk\,spal} = \frac{x \cdot (i_{m4} - i_{m3})}{LHV_f \cdot \xi_{spal}} \tag{47}$$

2.6. Turbine

Isentropic power $N_{iz(4\div5)}$ of the turbine defined by the equation (48)

$$N_{iz(4\div5)} = \overset{\circ}{m}_m \cdot \left(i_{m4} - i_{m5\,iz(s_5=s_4)}\right)$$
$$= x \cdot \overset{\circ}{m}_{pal} \cdot \left(i_{m4} - i_{m5\,iz(s_5=s_4)}\right) \tag{48}$$

depends on the flow rate $\overset{\circ}{m}_m$ of the fluid flown through the turbine and the isentropic total enthalpy drop from the initial state $i_{m4(t4,p4,s4)}$ at point 4 before the turbine to final state $i_{m5\,iz\,(t5,\,p5,\,s5=s4)}$ at point 5_s lying on the constant pressure line $p_5 = p_0$ corresponding to pressure in the condenser.

Break power of the turbine N_6 follows the equation (49)

$$N_6 = \eta_{pol} \cdot \eta_{mech} \cdot N_{iz(4\div5)} = \eta_{og} \cdot N_{iz(4\div5)}$$
$$= \eta_{og} \cdot x \cdot \overset{\circ}{m}_{pal} \cdot \left(i_{m4} - i_{m5\,iz(s_5=s_4)}\right) \tag{49}$$

comes from product of the isentropic power $N_{iz\,(4\div5)}$ and the overall efficiency η_{og} of the turbine resulting from multiplying (50) the turbine polytropic efficiency η_{pol} with its mechanical efficiency η_{mech}.

$$\eta_{og} = \eta_{pol} \cdot \eta_{mech} \tag{50}$$

2.7. Benefits from combining the Rankine system with the IC engine

Technical benefits available from applying the Rankine system for heat recuperation from the IC engine can be expressed by the coefficient $\Delta\eta_e$ which is a ratio of net power N_6-N_1 from the Rankine system upon the chemical energy rate in the fuel N_{pal} (51, 52)

$$\Delta\eta_e = \frac{N_6 - N_1}{N_{pal}}$$
$$= \frac{\eta_{og} \cdot x \cdot \overset{\circ}{m}_{pal} \cdot \left(i_{m4} - i_{m5\,iz(s_5=s_4)}\right) - \dfrac{1}{\eta_p} \cdot \dfrac{x \cdot \overset{\circ}{m}_{pal}}{\rho_m} \cdot (p_2 - p_0)}{LHV_f \cdot \overset{\circ}{m}_{pal}} \tag{51}$$

$$\Delta\eta_e = \frac{x \cdot \left[\eta_{og} \cdot \left(i_{m4} - i_{m5\,iz(s_5=s_4)}\right) - \dfrac{1}{\eta_p \cdot \rho_m} \cdot (p_2 - p_0)\right]}{LHV_f} \tag{52}$$

The benefits can be also expressed by the coefficient μ which is defined as the net power N_6-N_1 upon the break power N_e of the IC engine (53) that finally expresses the relative increase μ in the break efficiency η_e (54) of the engine after combining it with the Rankine system.

$$\mu = \frac{N_6 - N_1}{N_e}$$
$$= \frac{\eta_{og} \cdot x \cdot \overset{\circ}{m}_{pal} \cdot \left(i_{m4} - i_{m5\,iz(s_5=s_4)}\right) - \dfrac{1}{\eta_p} \cdot \dfrac{x \cdot \overset{\circ}{m}_{pal}}{\rho_m} \cdot (p_2 - p_0)}{\eta_e \cdot LHV_f \cdot \overset{\circ}{m}_{pal}} \tag{53}$$

$$\mu = \frac{1}{\eta_e} \cdot \frac{x \cdot \left[\eta_{og} \cdot \left(i_{m4} - i_{m5\,iz(s_5=s_4)}\right) - \dfrac{1}{\eta_p \cdot \rho_m} \cdot (p_2 - p_0)\right]}{LHV_f}$$
$$= \frac{\Delta\eta_e}{\eta_e} \tag{54}$$

3. Input data for modeling

Combining the engine with the Rankine system does not significantly influence on engine performance, therefore all the input data for the engine are constant for the whole computational process. The data for the engine were taken from a typical modern turbocharged compression ignition engine used in a heavy duty vehicle (IMEP = 2.68 MPa at 1350 rpm).

Table 1 Input data for the model

Coefficient of engine cooling losses (from the engine and oil)	$\xi_{chl} = 0.141$
Coefficient of engine exhaust losses (after the supercharger)	$\xi_{spal} = 0.32$
Engine break efficiency	η_e=0.383
Temperature of hot cooling water from the engine	t_b=85°C
Temperature of hot exhaust gases from the engine after the supercharger	t_g=600°C
Lower Heating Value (LHV) of the fuel	LHV_f = 41.92MJ/kg
Main components of the exhaust gases	$N_2 = 0.729$ $O_2 = 0.0376$ $CO_2 = 0.10507$ $CO = 0$ $H_2O = 0.12825$
Fluid in the Rankine system	Water/steam
Temperature in the condenser	t_0=30°C
Overall efficiency of the recirculation pump	η_P=0.5
Pressure of the fluid after the pump	p_2 = 1.6 MPa
Efficiency of the low-temperature (first) recovery unit	η_{wm}=0.9

Efficiency of the high-temperature (second) recovery unit	η_{sm}=0.9		
Isentropic efficiency of the turbine	η_{pol}=0.8		
Mechanical efficiency of the turbine	η_{mech}=0.95		
Absolute pressure in the turbine inlet	$p_4 = 1.6\ MPa$		

Quantity x (the ratio of the Rankine fluid flow rate $\overset{\circ}{m}_m$ upon the mass fuel flow rate $\overset{\circ}{m}_{pal}$) is assumed as the variable for computation the individual modeling cycle.

4. Results of modeling

In the table 2 there are exemplary results of computation with the $x = 3.7$. Other results with the x varying from 2.5 to 5.1 are presented in the figures 2, 3, 4, 5, 6.

Table 2. Selected results of modeling the system consisted of the IC engine and the Rankine system for steam pressure of 1.6 MPa at optimal x equaled 3.7

	Quantity	Dimension	Value
1	Absolute pressure of the condensate in the condenser	kPa	4.246
2	Enthalpy increase of the fluid in the recirculation pump	kJ/kg	0.2828
3	Enthalpy increase of the fluid in the low temperature recovery unit	kJ/kg	231.74
4	Fraction of heat transferred from the engine cooling system to the Rankine system	1	0.1447
5	Enthalpy increase of the fluid in the high-temperature recovery unit	kJ/kg	3336
6	Fraction of heat transferred from the exhaust gases to the Rankine system	1	0.9202
7	Entire enthalpy increase of the fluid from the condensate state to vapors before the turbine	kJ/kg	3568
8	Temperature of vapors before the turbine	°C	600
9	Dryness fraction of the fluid before the turbine	1	1
10	Entropy of vapors before the turbine	kJ/(kg·K)	7.808
11	Enthalpy of vapors before the turbine	kJ/kg	3693
12	Enthalpy of the fluid after isentropic expansion in the turbine	kJ/kg	1894
13	Dryness fraction of the fluid after isentropic expansion	1	0.9198
14	Isentropic enthalpy drop in the turbine	kJ/kg	1333
15	Real enthalpy drop in the turbine	kJ/kg	1066
16	Enthalpy of the fluid after the turbine	kJ/kg	2627
17	Entropy of the fluid after the turbine	kJ/(kg·K)	8.674
18	Real dryness fraction of the fluid after the turbine	1	1
19	Absolute pressure of the fluid after the turbine	kPa	4.246
20	Enthalpy drop in the condenser	kJ/kg	2501
21	Total enthalpy drop in the turbine and the condenser	kJ/kg	3568
22	**Overall engine efficiency**	1	0.383

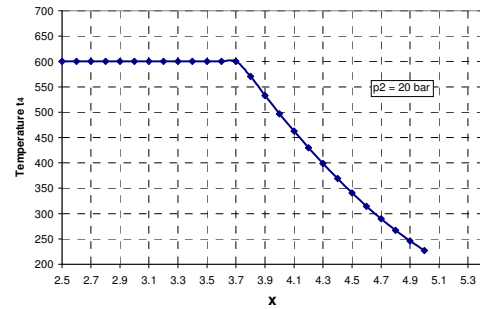| 23	**Overall efficiency of the system combined the IC engine and the Rankine system**	1	0.472
24	**Net break power of the turbine / Break power of the engine**	1	0.243

Fig. 2. Temperature t_4 of fluid vapors (steam) at the exit of the second recovery unit vs the coefficient x

With $x < 3{,}7$ temperature t_4 of the fluid equals temperature of the exhaust gases (fig. 2) according to the assumption that it can exceed this temperature if the recovery unit is an ideal highly efficient counter-current heat exchanger. Due to lower thermal capacity of the Rankine fluid flow rate, heat from the exhaust gases cannot be completely utilized (fig. 3).
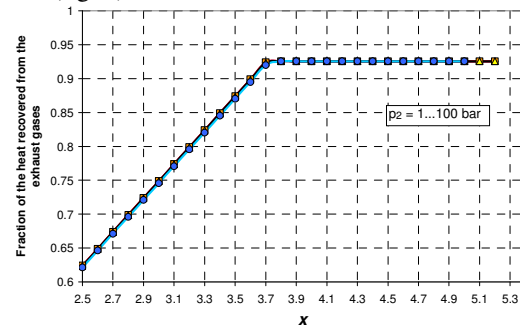

Fig. 3. Fraction of the heat from the exhaust gases to the fluid vs the x

Maximal fraction of the heat transferred from the exhaust gases to the fluid equals 0.927 and it is limited by temperature of 100°C. Remarkable lower is fraction of the heat taken by the fluid from the engine cooling water in the low-temperature recovery unit. As shown in the figure 4 this fraction is inline with the coefficient x.
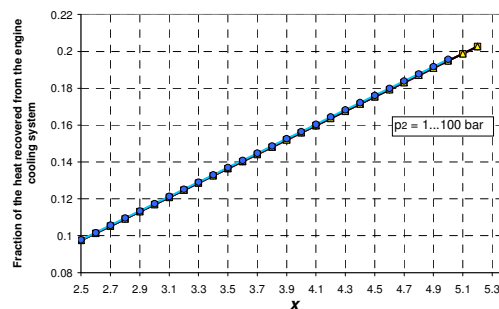

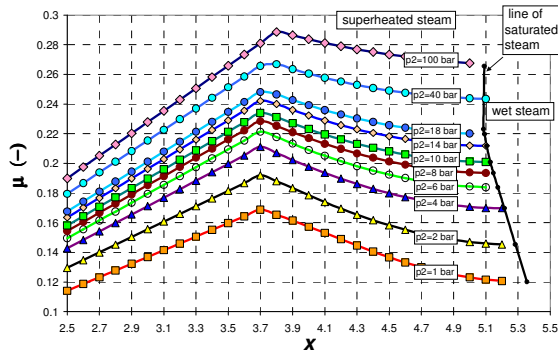Fig. 4. Fraction of the heat from the engine cooling water to the fluid vs the x

Fig. 5. Ratio μ of the break power from the Rankine system upon the break power of the IC engine (relative efficiency increase) vs the *x* with various pumping pressure p_2

It is noted that the highest energetic benefit occurs at the *x* approximately equaled 3.7 for pressure in the range from 1 to 100 bar. The *x* of 3.7 is the limit value at which temperature t_4 of the fluid at the turbine inlet is still the highest and equaled temperature t_g of exhaust gases. At this temperature t_4 the fluid is at superheated vapor state. With the *x* higher than 5.1-5.3 (line of saturated fluid in the fig.5) fluid (water) does not vaporize completely in the second recovery unit that causes water passes over the turbine, therefore operating the Rankine system with the *x* higher than 5.1 is not reasonable. Furthermore, it shows that the *x* = 3.7 value corresponding to maximum of the μ is almost constant with pressure p_2 changing from 1 to 100 bar as it is plotted in the fig.5. In the fig.6 the coefficient μ against pressure p_4 before the turbine is depicted (p_2 is of the same value as p_4).
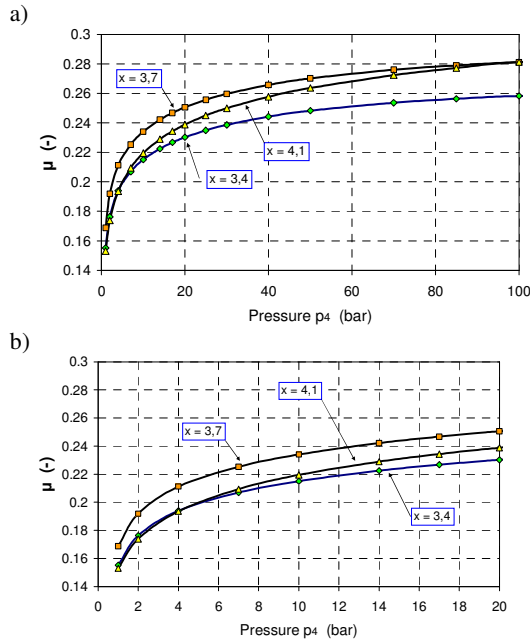
a)

b)

Fig. 6. Ratio μ of the break power from the Rankine system upon the break power of the engine against pressure p_4 in the turbine inlet (a), zoom of the fig.6a (b)

Someone can find that significant increase in the μ appears with pressure p_4 increasing from 1 to approximately 15 bar. Further pressure increase up to 100 bar does not contribute to remarkable improvement of the μ. Thus, applying a high pressure recirculation pump in the Rankine system combined the IC engine with power of several hundreds kW is not economically justified.

5. Conclusions

Conclusions from this preliminary analysis of the modeled Rankine cycle based system for recovery waste heat from the IC engine are as follows:

1. Fraction of waste heat transferred from the engine cooling system to the Rankine system increases with increase in the ratio *x* of the Rankine fluid flow rate over the flow rate of the fuel burnt in the engine. Maximal fraction of this utilized waste heat does not exceed 0.2, thus from this point of view it can be considered as insignificant issue unlike the heat absorbed from the exhaust gases.

2. Fraction of the heat absorbed in the exhaust gases - Rankine fluid heat exchanger (the high-temperature recovery unit) increases with increase in the ratio *x* (the Rankine fluid flow rate over the flow rate of the fuel) up to value of 3.7. With this ratio higher than 3.7 the fraction of the utilized heat is constant and equals 0.92.

3. Maximum in overall fraction of the total heat absorbed by the Rankine system in the both recovery units can be obtained with the ratio *x* equaled 3.7. The optimal value of the *x* is characteristic for the type of fluid applied in the Rankine system and the engine data. With the *x* decreasing below 3.7 the fraction of the heat absorbed from exhaust gases decreases. Moreover, the Rankine fluid flow rate also decreases that contributes to reduction in power output of the Rankine system. With increase in the *x* above 3.7 temperature of the Rankine fluid vapors goes down.

4. With the optimal *x* = 3.7 the whole expansion process in the turbine goes under the steam superheated. The relative increase *μ* in the engine break efficiency defined as the ratio of the net break power from the turbine upon the break power from the engine increases with increase in pressure of the steam to the turbine. The *μ* can reach 0.29 at pressure of the steam of 100 bar. It means that additional available power from the microturbine in the Rankine system can reach 29% of the break power of the engine without increase in the engine fuel consumption.

5. Investigation on applying low-boiling fluids other then water into the Rankine system makes it possible to obtain higher fraction of the low-temperature heat absorbed from the engine cooling system. Thus, potential increase in power output from the Rankine system is possible.

6.   Due to simplifications (e.g. assumed that heat recovery units are ideal highly efficient heat exchangers) the computed final results of heat recovered are the maximal values, which might be possible to obtain in the system consisted of the IC engine and the microturbine working with the Rankine system using heat recuperated from the engine coolant and the exhaust gases. On the basis of the conducted analysis someone concludes that overall efficiency of a propulsion system consisted of the IC engine can be remarkable improved by combining it with the microturbine.

Nomenclature/*Skróty i oznaczenia*

Δi Enthalpy change/*zmiana entalpii*
η Efficiency/*sprawność*
μ Relative efficiency increase/*względny przyrost sprawności*
ρ Density/*gęstość*
ξ Coefficient of losses/*współczynnik strat*
LHV   Lower Heating Value/*wartość opałowa*
N   Power, thermal power/*moc, ciepło*
X   Dryness fraction/*stopień suchości pary*
c_p specific heat at constant pressure/*ciepło właciwe przy stałym ciśnieniu*
i   Enthalpy/*entalpia*

$\overset{\circ}{m}_{pal}$ Fuel flowrate/*wydatek paliwa*
$\overset{\circ}{m}_{m}$ Rankine fluid flowrate/*wydatek cieczy obiegu Rankina*
p   Pressure/*ciśnienie*
s   Entropy/*entropia*
t,T   Temperature/*temperatura*
w   Velocity/*prędkość*
x   ratio of $\overset{\circ}{m}_{m}$ upon $\overset{\circ}{m}_{pal}$

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