DESIGN OF WASTE HEAT RECOVERY SYSTEMS BASED ON SUPERCRITICAL ORC FOR POWERFUL GAS AND DIESEL ENGINES

Oleksii Rudenko[1], Leonid Moroz[2], Maksym Burlaka[3], Clement Joly*[4]

[1]SoftInWay, Engineering,
Burlington, MA, USA
o.rudenko@softinway.com

[2]SoftInWay, Engineering
Burlington, MA, USA
l.moroz@softinway.com

[3]SoftInWay, Engineering
Burlington, MA, USA
m.burlaka@softinway.com

[4]SoftInWay, Engineering
Burlington, MA, USA
clement.joly@softinway.com

* Corresponding Author

ABSTRACT

Nowadays the scientific world community is strongly concerned about problems of efficiency increase and emissions reduction of Internal Combustion Piston Engines (ICPE). The equipment of ICPE with Waste Heat Recovery Systems (WHRS) is an effective solution for the aforementioned problems. This paper focuses on finding the maximum possible heat recovery from the available high and low temperature waste heat flows of a powerful ICPE to produce the maximum amount of additional power while decreasing the load on the engine's cooling system.

Having considered and analyzed existing works devoted to the development of WHRS the most effective ideas were combined to design several thermodynamic cycles for new WHRS of a powerful piston engine (here a G3612 CAT gas petroleum engine is considered). The proposed WHRS is based on a Supercritical Organic Rankine Cycle (SORC) using R245fa as the working fluid where heat is extracted from the waste heat sources by a refrigerant at different pressure levels. Internal recuperation is used to further improve the cycle performances and increase the waste heat recovery. The thermodynamic analysis of the new WHRS showed that up to 19.73% of power boost for the internal combustion engine can be achieved without burning additional fuel which represents significant gains in terms of specific power.

In order to quantify the estimation of the performances for proposed cycles the design of a traditional, high efficiency, a WHRS based on double pressure water steam cycle for the same engine's conditions was performed. This comparison of performances between the steam cycle and the SORC R245fa cycles confirmed a high potential for the designed cycles.

1. INTRODUCTION

Internal combustion piston engines are among the largest consumers of liquid and gaseous fossil fuels all over the world. Despite the introduction of new technologies and constant improving of engines performances they still are relatively wasteful. Indeed, the efficiency of modern engines rarely exceeds 40-45% (Seher et al. (2012), Guopeng et al. (2013)) and the remainder of the fuel energy usually dissipates into the environment in the form of waste heat. The heat balance diagram of typical engine is given in Figure 1. As is evident from Figure 1, besides the mechanical work energy the heat balance includes a heat of exhaust gas, a heat of charge air, a Jacket Water (JW) heat, a heat of lubricating oil and a radiation heat. The energy from all the heat sources except the last one (radiation), due to its ultra-low waste heat recovery potential, can be used as heat sources for WHRS (Paanu et al. (2012)) and are considered here.

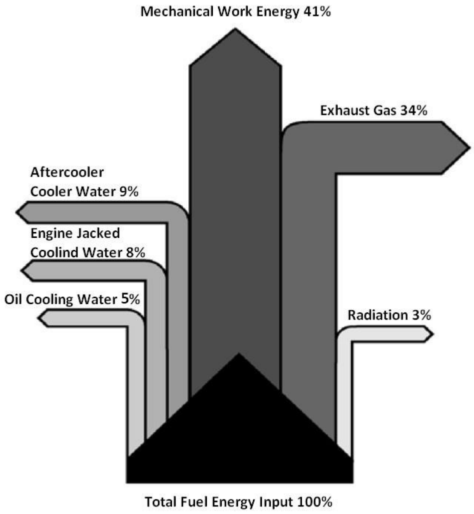

Figure 1: Typical heat balance diagram for CAT engine (Caterpillar (2011))

Waste heat utilization is a very current task because it allows to reduce the harmful influence of ICPE operation on the environment as well as to obtain additional energy and to reduce the load on the engine's cooling system. Different WHRS can produce heat energy, mechanical energy or electricity and combinations of the converted energy forms exist as well. In general, the type of WHRS to be used is determined by the engine type, fuel cost, available energy customers and other factors. In the present paper only WHRS for mechanical power and electricity production were considered because these kinds of energy are preferable for this type of applications and they can be easily converted into other forms of energy.

For vehicle engines the WHRS based on Organic Rankine Cycle (ORC) are the most commercially developed (Paanu et al. (2012)). Because of strict restrictions on weight and dimensions, the mentioned systems typically operate on the base of a simple or recuperated ORC and utilize only high temperature waste heat from the exhaust gases and the exhaust gas recirculation. They usually produce mechanical power or electricity. More complex cycles and a larger number of heat sources are used for waste heat recovery from powerful internal combustion engines where additional weight and dimensions are not crucial factors. Waste heat from stationary, marine and another more powerful ICPE can be recovered using a typical steam bottoming cycle. Steam WHRS allow utilizing almost all a high temperature waste heat and partially utilizing a low temperature heat. The high efficiency steam WHRS are presented in (MAN Diesel & Turbo (2012), Petrov (2006)), they provide up to 14.5% of power boost for the engine.

From the existing works devoted to waste heat recovery range of problems the following methods of efficiency increase can be highlighted:

- Addition of the internal heat recuperation to a WHR cycle;
- Appropriate working fluid selection;
- Increment of initial parameters of bottoming cycle up to supercritical values;
- Maximize waste heat utilization due to the usage of low temperature heat sources;
- Bottoming cycle complexification or usage of several bottoming cycles with different fluids (Maogang (2011)).

This paper focuses on the development of new WHRS as an alternative to high efficiency steam bottoming cycles by accounting for the latest progress in the field of waste heat recovery. The application range of the proposed system extends to powerful and super powerful ICPEs.

2. DEVELOPMENT OF NEW WHRS

The goal of the present work is the development of a new, high efficiency WHRS for powerful and super powerful ICPEs based on ORC principles. To solve the assigned task, a thorough study of the currently existing works was performed and the best ideas were combined. The principles of the maximum waste heat utilization, maximum possible initial cycle parameters, recuperation usage and single working fluid were assumed as a basis for the new WHRS design.
It is well known that the fluid saturation temperature depends on the pressure and the higher the pressure level the higher the saturation temperature. This why the extraction of the waste heat from available sources by a refrigerant at different pressure levels is more effective to achieve a maximum waste heat utilization. The thermodynamic efficiency of a Rankine cycle mainly depends on its maximum cycle parameters (pressure and temperature) and minimum pressure. In the works (Jadhao and Thombare (2013), Braimakis et al. (2014)) is shown that ORC operation with a supercritical top pressure has a positive effect on cycle performances. The internal recuperation, in turn, increases WHRS efficiency due to returning part of the heat after the expander to the cycle.
The process of design of recovery system for waste heat flows from the G3612 CAT gas petroleum piston engine is described hereafter. The process is divided into 4 steps and includes: working fluid selection, definition of main cycle parameters, cycle design and thermodynamic simulation, preliminary design of High Pressure and Low Pressure Turbines (HPT, LPT, respectively). The used engine's data is given in Table 1 and the engine's heat balance is shown in Figure 1.

Table 1: Waste heat flows from G3612 CAT gas petroleum engine (Caterpillar (2011))

Energy Flow	Value, kW	Temperature Potential	Recoverability by WHRS
Total Input Heat From the Fuel	7192	High	-
Mechanical Work	2948.72	-	-
Heat Rejection to Exhaust (Recoverable Exhaust Heat at 120°C)	2445.28 (1644.62)	High	Yes
Heat Rejection to the Aftercooler	647.28	Middle	Yes
Heat Rejection to the Jacket Water	575.36	Low	Yes
Heat Rejection to the Oil Cooler	359.6	Low	Yes
Heat Rejection to the Atmosphere	215.46	Lowest	No

2.1 Working Fluid Selection

Unfortunately, a universal organic working fluid that can be used for a wide range of ORC does not seem to exist and the working fluid selection is one of the most important design steps. There are a lot of works devoted to the mentioned problem (Jadhao & Thombare (2013), Braimakis et al. (2014), DiCarlo & Wallace (2011), Jadhao & Thombare (2013), Nouman (2012)). As a rule, the working fluids are considered according to such criteria as thermodynamic properties, environmental impact, thermal stability and safety. Water, ethanol, R245fa and R134a are among the most popular organic working fluids at the moment. Besides, for a recuperated ORC it is recommended to use either an isentropic or a dry fluid. Here, the working fluid was selected according to its potential to remove heat from the selected sources, at different temperatures, in a pressure range from 1 to 45 bars.
Maogang (2011) used in his work for this purpose the combined thermodynamic cycle. It consists of two cycles: an ORC is used to recover the waste heat of the lubricant and exhaust gas and a Kalina cycle for the recovery of the waste heat of the low-temperature cooling water. Of course, the combined WHRS is effective enough but the use of 2 working fluids essentially complicates the system. For these reasons the Kalina cycle is eliminated here in order to simplify the system.
Based on aforementioned thoughts the R245fa (pentafluoropropane) was selected as the working fluid. Due to its low condensation temperature and relatively high decomposition temperature (higher than 250 °C (Honeywell (2014))) R245fa fits the basic criteria of this study. R245fa fluid properties

were calculated based on the NIST RefProp library (version 9.1). The fluid reference state corresponds to the International Institute of Refrigeration (IIR) convention.

2.2 Main ORC Parameters

The design parameters of the cycle components used in this study are presented in Table 2. The maximum cycle temperature was limited to 240 °C to avoid fluid decomposition. The maximal cycle pressure was set to 45 bars. The subsequent pressure increase does not lead to essential cycle performance increase but it leads to an increase of the WHRS production cost and complication of the HPT design. At the HPT inlet the R245fa is at a supercritical pressure which allows for an increase in the cycle efficiency. The comparison of performances between the SORC and subcritical ORC is given in (Jadhao and Thombare (2013), Braimakis et al. (2014)). For this case the condenser pressure was fixed to 1.3 bars to prevent air ingression into the closed cycle.

Table 2: Design parameters of cycle components

Parameter	Units	Value
Efficiency of HPT and LPT	-	0.8
Efficiency of High and Low Pressure Pumps	-	0.8
Minimal Pinch Point for Heat Exchangers	°C	10
Hydraulic Losses in the Pipelines and Heat Exchangers	-	Ignored

2.3 WHRS Cycle Design and Thermodynamic Simulation

At the initial stage of the ORC design waste heat flows from the G3612 CAT engine (see Figure 1 and Table 1) were considered to provide optimal heat utilization with moderate system complexity. In Figure 2 the distribution of the waste heat flows according to their temperatures and the option of their utilization to generate superheated working fluid for ORC are shown.

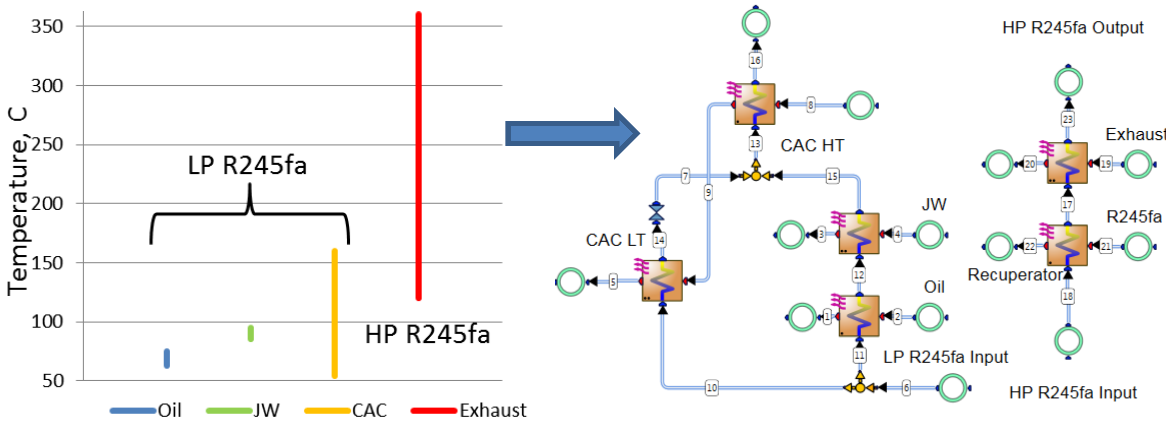

Figure 2: The distribution of heat flows according to their temperatures and the option of their utilization (where LP and HP – low pressure and high pressure; LT and HT – low temperature and high temperature; CAC – charge air cooler)

As is clear from Figure 2, the waste heat from the lubrication oil, the JW and the charge air has a lower temperature potential than the exhaust. It is then rational to use the heat from these sources to preheat the LP flow of R245fa to the necessary conditions. The heat exchangers which transfer heat from the oil and the JW to the low pressure flow of R245fa are connected in series according to their temperature ranges. The Charge Air Cooler (CAC) is divided into two temperature zones; The low temperature CAC (CAC LT heat exchanger in Figure 2) which operates in parallel with the JW and the oil coolers and the high temperature CAC (CAC HT) which is intended for the whole LP flow of R245fa superheat. The pressure of the LP R245fa loop was determined so that the temperature of the working fluid at high temperature CAC outlet is slightly higher than the saturation temperature at the given pressure. Due to the use of a dry working fluid the expansion process in the LPT will take place in the superheated region. In the designed ORC the low pressure loop operates under 7 bars of pressure.

The high pressure loop of the WHRS operates at a maximum pressure of 45 bars. The ORC working fluid is heated through two heat exchangers connected in series (internal recuperator and exhaust gas exchanger). The internal recuperator is used to return to the cycle the part of the heat rejected into condenser. The developed distribution of heat flows for this CAT engine allows utilizing the extra heat in the ORC cycle; almost all of the waste heat can be transferred to the WHRS, the CAC has minimum heat utilization (584.4 kW from the 647 kW available heat). Two alternative cycle concepts for this CAT engine were designed: with separate turbines and with a shared LPT. Both concepts were simulated with the use of the heat balance calculation tool AxCYCLE™ (SoftInWay Inc. (2014)).

2.3.1 Dual loop SORC concept with separate turbines

The flow diagram of the dual loop SORC with the separate turbines is presented in Figure 3. The cycle consists of 6 heat exchangers, 2 turbines (HPT and LPT), 2 pumps (HPP and LPP) and the condenser. Both turbines operate with the same backpressure – 1.3 bars. The flows of R245fa are mixed at the condenser inlet and split at its outlet. The temperature – entropy diagram for the presented cycle is shown on Figure 4. The process 1-2-3-4-5-1 corresponds to the high pressure loop operation and the process 10-20-30-40-10 is for the low pressure loop operation.

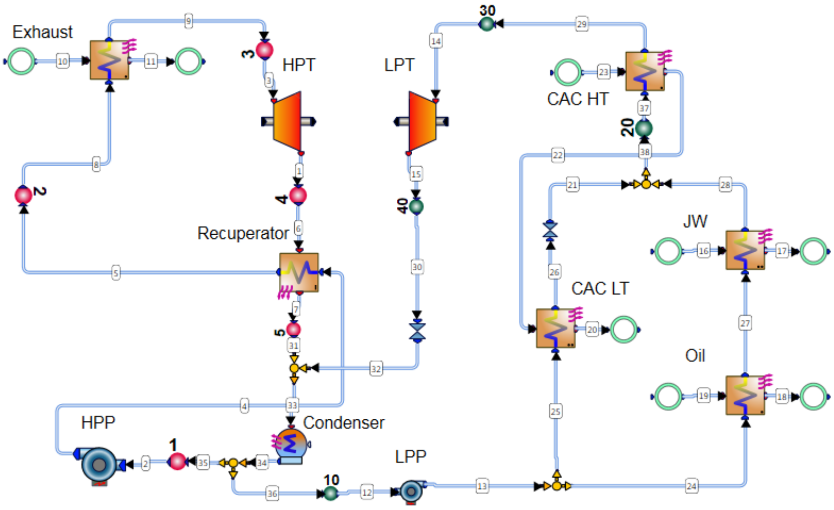

Figure 3: The flow diagram of the SORC with separate turbines

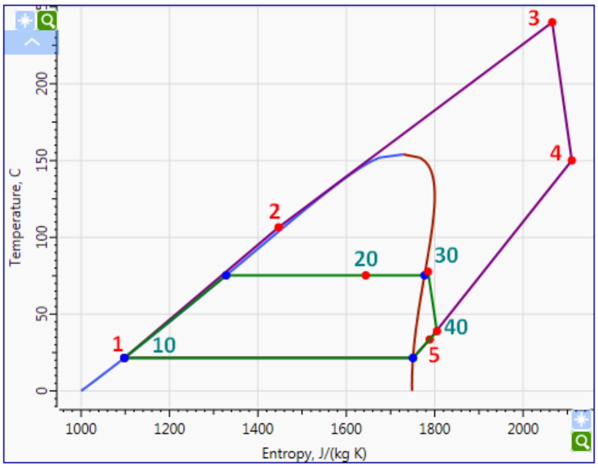

Figure 4: The t-s diagram for the SORC with separate turbines

2.3.2 Dual loop SORC concept with shared LPT

The flow diagram of the dual loop SORC with shared LPT is presented on Figure 5. The considered cycle has the same set of components as the previous one. Unlike in the previous cycle, here the working fluid expands in the HPT up to the pressure of the LP loop (7 bars). After that the flows from the different loops mix and expand in the shared LPT. This means that the mass flow through the LPT is equal to the sum of the HP and LP flows. Flows are again split at the condenser outlet.

The temperature – entropy diagram for the cycle with the shared LPT is shown on Figure 6. In the process 1-2-3 the heat is transferred to the high pressure R245fa flow (45 bars), the process 3-4 is the expansion in the HPT, the process 10-20-30 corresponds to the heat addition to the low pressure flow (7 bars), at point 5 the flows from the different loops mix, between 5 and 6 – the expansion in the LPT occurs and for 6-7 happens a heat transfer in the internal recuperator.

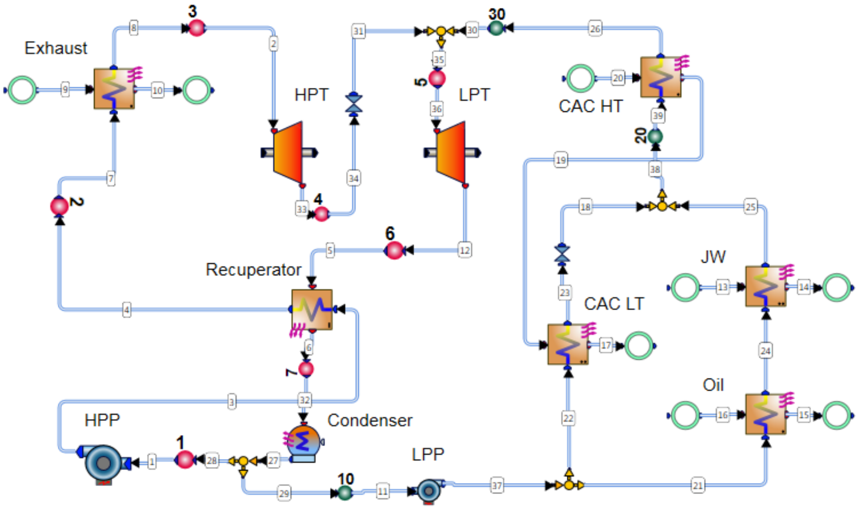

Figure 5: The flow diagram of the SORC with shared LPT

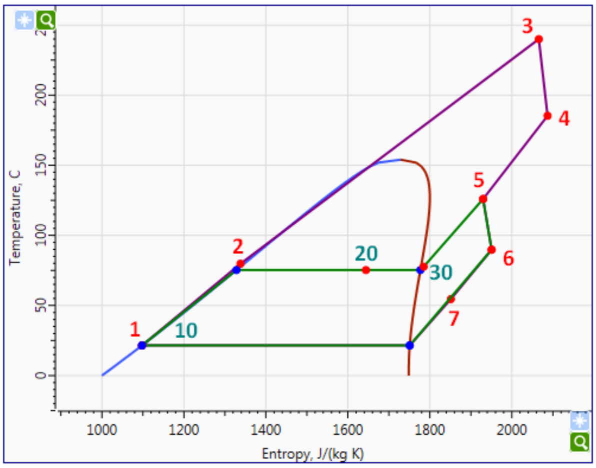

Figure 6: The t-s diagram for the SORC with shared LPT

2.3.3 Performance comparison

In order to quantify the estimation of the performances for the proposed embodiments of the SORC the comparison of their performances with the integral parameters of WHRS based on the double pressure water steam cycle was performed.

The bottoming steam cycle was designed according to materials from (MAN Diesel & Turbo (2012)) for the same CAT engine's conditions given in Table 1. The design parameters used for the steam cycle components are presented in Table 2. The maximum cycle temperature and pressure were limited to 258 °C and to 9 bars, respectively. The pressure at the HPT outlet was set to 3 bars and the condenser pressure was taken as 5.7 kPa. The steam cycle flow diagram and its process in the t – s coordinates are shown on Figure 7. As is evident from Figure 1Figure 7, the use of a steam WHRS allows covering only two sources of the engine's waste heat. This is connected to the thermodynamic properties of water. In this cycle a huge part of the heat is absorbed by the working fluid in the two-phase region under pretty high temperatures (175.35 °C at pressure of 9 bars and 133.52 °C at 3 bars respectively) and the heat of most of the low-temperature sources remains unclaimed. The use of heat from the low temperature sources as well as the steam mass flow is limited by the high temperature source (by the exhaust gas temperature and mass flow).

On the t-s diagram (see Figure 7) between 1 and 2 the total water flow is preheated. After this the flows splits, in the process 2-3-4 the heat is transferred to the high pressure loop and the process 2-8 corresponds to the heat addition to the low pressure flow. The processes 4-5 and 6-7 are for the expansion in the HPT and LPT, respectively.

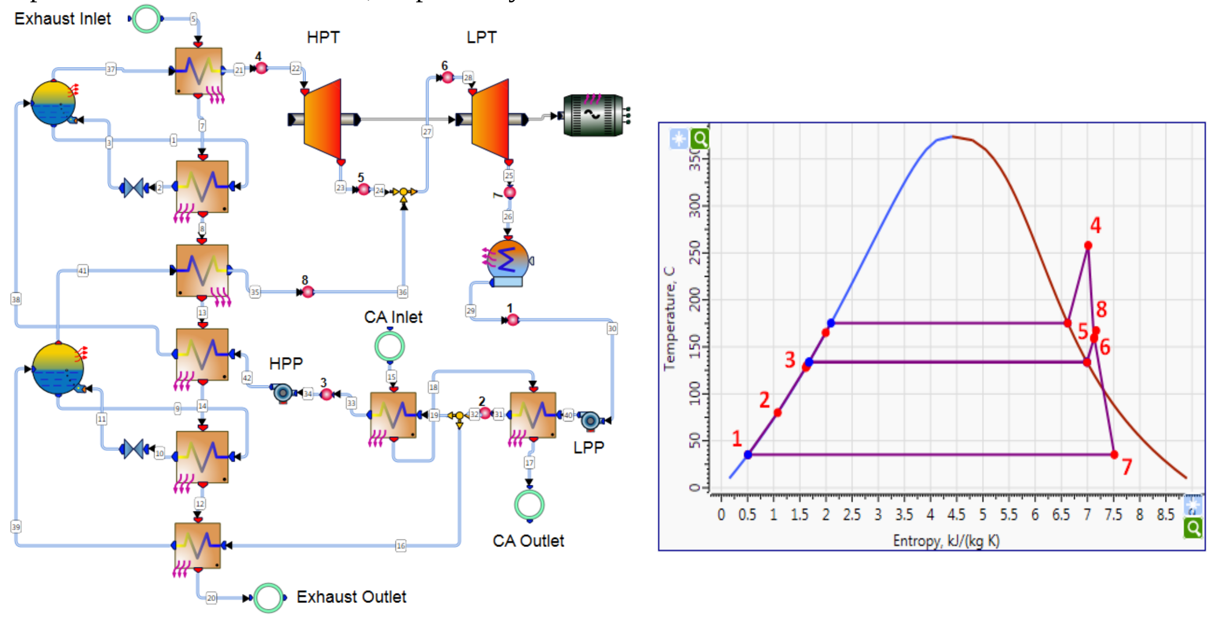

Figure 7: Flow diagram and process in t-s coordinates of the steam cycle

Table 3: The main thermodynamic parameters and the calculated performances of considered cycles

Parameter	Unit	Cycle Embodiments		
		R245fa with Separate Turbines	**R245fa with Shared LPT**	**Steam Cycle**
Total Mass Flow	kg/s	12.53	11.77	0.63
Pressure at HPT inlet	bar	45.0	45.0	9.0
Temperature at HPT inlet	°C	240.0	240.0	258.0
Pressure at HPT outlet	bar	1.3	7.0	3.0
Temperature at HPT outlet	°C	150.0	185.3	158.8
Pressure at LPT inlet	bar	7.0	7.0	3.0
Temperature at LPT inlet	°C	77.7	126.0	160.0
Pressure at LPT outlet (Condenser Pressure)	bar	1.3	1.3	0.057
Temperature at LPT outlet	°C	39.0	89.8	35.2
Saturation Temperature at Condenser Pressure	°C	21.5	21.5	35.2
Total Heat Transferred to Cycle	kW	3162.46	3162.46	1750.96
Net Power Production	kW	575.824	531.102	395.728
Power Boost for the CAT Engine	%	19.53	18.01	13.42
Total System Efficiency (ICE+WHRS)	%	49.01	48.38	46.50

The main thermodynamic parameters and the calculated performances of the aforementioned cycles are summarized in Table 3. It can be seen that the steam cycle has the lowest net power production of the three embodiments studied. This fact can be explained by the lowest total amount of waste heat transferred to the WHR cycle. At the same time this cycle has the largest thermal efficiency; it produces 395.728 kW from 1750.96 kW of transferred heat. However, for a WHRS cycle the Net Power Production (NPP) is more attractive since the waste heat is free.
The SORC with separate turbines has a higher NPP due to its higher internal heat recuperation. In that cycle the working fluid temperature at the recuperator inlet (HPT outlet) is 150 °C versus 90 °C in the cycle with the shared LPT. It allows increasing the working fluid mass flow at the HPT inlet (6.02 vs. 5.25 kg/s) which therefore leads to a rise in the mechanical power production. However, the operation of the turbine of the HP loop in the pressure range of the low pressure turbine (cycle concept with

separate turbines) is not a very rational solution. Despite the NPP, the cycle embodiment with the shared LPT is more preferable in the terms of HPT design, production costs and off-design operation. The results of the preliminary design of the HPT and LPT for the SORC with shared LPT are given below.

2.4 Preliminary Design of High Pressure and Low Pressure Turbines

In order to estimate performance and dimensions of these turbines, it was decided to perform a preliminary design for them using SoftInWay turbomachinery design/analysis tool AxSTREAM (Moroz et al. (2005) and (2006)). The boundary conditions for the turbines design are presented in Table 4.

Table 4: Boundary conditions

Parameter	Unit	HPT	LPT
Inlet total pressure	bar	45.00	7.000
Inlet total enthalpy	kJ/kg	621.87	515.63
Static pressure at outlet	bar	7.000	1.300
Mass flow rate	kg/s	5.249	11.77

Rough preliminary estimations showed that both turbines will be rather small in size. For example, the LPT turbine of axial type has 10 stages with a 1st stage nozzle height of about 2 cm and a constant hub diameter of 26 cm. The HPT turbine has up to 20 stages and even smaller blade heights and diameter. It is obvious that it is not reasonable to use axial turbines in these conditions. Taking into account the aforementioned preliminary results it was decided to select a radial turbine type for the further steps.
Turbines evaluation was performed in two steps:

1. Turbine design utilizing simplified 1D axisymmetric calculation models by automatic generation of thousands of designs and selection of the best design.
2. Turbine analysis utilizing precise 2D axisymmetric models to get realistic turbine performance.

The Mitrohin-Stepanov loss model was utilized for impeller losses calculation.
Thousands of turbine designs were automatically generated using AxSTREAM's preliminary design tool. At this step an optimum rotational speed and diameter were selected. After this, 2D calculations were performed for both turbines to obtain their performance. A meridional view of the turbines with some performance data and density field are shown on Figure 8. It should be noted that the turbines picture on Figure 8 are presented using a different scale. Real dimensions can be seen on the vertical and horizontal rulers on the left and the bottom sides of the pictures, respectively. Three-dimensional views of the turbines are presented on Figure 9. Both turbines are of a non-nozzles design type (only impellers and volute). The required inlet angle on impeller blades is achieved by using a special volute design. All performance data and crucial dimensions are collected in Table 5.

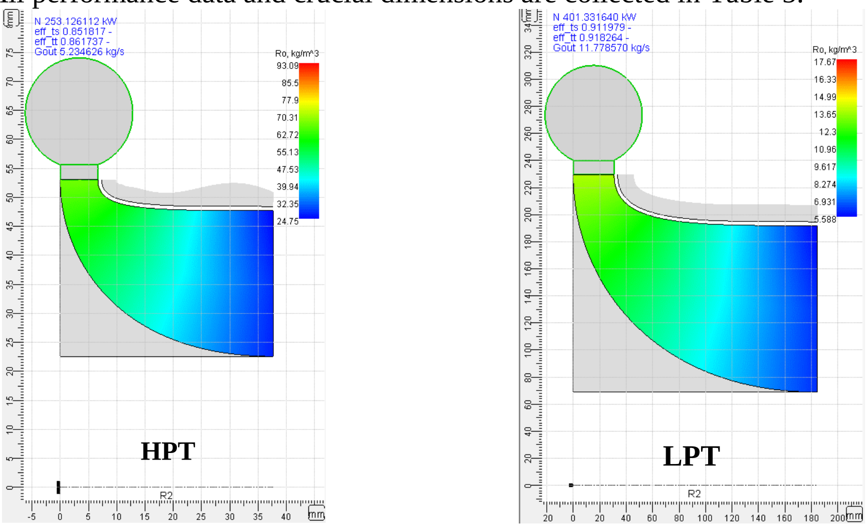

Figure 8: Turbines meridional view with density field
(where, eff_ts – turbine internal total-to-static efficiency, eff_tt – turbine internal total-to-total efficiency, N – turbine power, Gout – turbine mass flow rate, Roh – density**)**

The HPT is about 4 times smaller than the LPT but its shaft rotational speed is about 5 times higher than for the LPT. The HTP gives 211 kW of mechanical power while the LPT turbine provides about 395 kW. It should be noted that the obtained geometry and performance should not be considered as final. Final configurations might have slightly different performance and dimensions but the difference will not be significant.

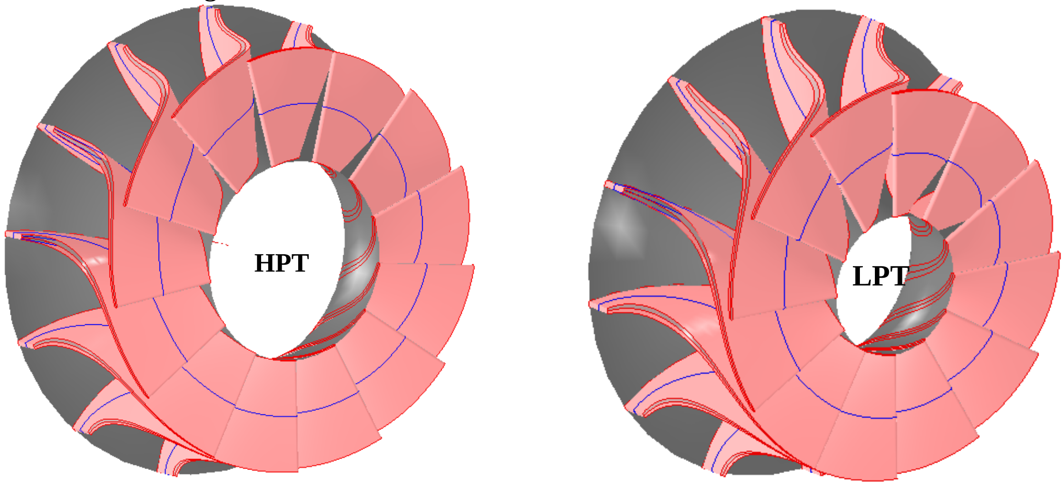

Figure 9: 3D view of the turbines

Table 5: Performance data and crucial dimensions of the turbines

Parameter	Unit	HPT	LPT
Internal total-to-static efficiency	%	85.18	91.33
Power	kW	211.4	394.6
Shaft rotational speed	rpm	40000	7914
Impeller diameter at inlet	mm	106.1	459.2
Mean impeller diameter at outlet	mm	70.3	260.6
Blade height at inlet	mm	6.7	30.9
Blade height at outlet	mm	25.1	122.8
Blade number		13	12

As is evident from Table 5, the obtained turbines efficiency values exceed the previously assumed values (see Table 2). The thermodynamic simulation of the SORC with shared LPT while accounting for the new turbines efficiencies gave 581.86 kW of net power production and 19.73% of power boost for the CAT Engine.

3. FUTURE WORK

The current paper represents a feasibility study and all the received results are not final. In the future, our works on the theme of WHR will be continued in the following directions: off-design WHRS simulation, control system design, estimation of cost and dimensions and more.

4. CONCLUSIONS

- The proposed concepts of dual loop Supercritical Organic Rankine Cycles are a promising technology for maximum waste heat utilization with moderate system complexity. The WHRS designed on the basis of the proposed concepts for powerful and super powerful ICPEs can be either adopted for electricity or mechanical power production.
- The simulation of the new WHRS in AxCYCLE showed that up to 19.73% of power boost for the G3612 CAT gas petroleum engine can be achieved without burning additional fuel which represents significant gains in terms of specific power.
- The comparison of performances between the traditional, high efficiency steam cycle and the SORC R245fa cycles confirmed a high potential for the designed cycles. Both proposed cycles embodiments have a higher net power production compared to the one for the steam cycle in net

power production due to the deep utilization of the low temperature waste heat sources that is not possible using steam.

- For one of the proposed ORC embodiments (the SORC with shared LPT) the evaluation of the turbines size and performance prediction was performed. The designed turbines have high efficiency levels with reasonable dimensions. The internal total-to-static efficiency of the HPT is equal to 85.18 % while it is 91.33% for the LPT. The maximum impeller diameters are equal to 106.1 and 459.2 mm, respectively. The strength characteristics and manufacturability of the designed turbines were not considered in the scope of the present study.

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