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DUAL ORC-BRAYTON POWER SYSTEM FOR WASTE HEAT RECOVERY IN HEAVY-DUTY VEHICLES

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Abstract: *Reducing the amount of energy required in industrial activities is one of the proven ways to achieve major cost savings, especially in the face of soaring energy prices. In the transport sector, besides the financial benefits, low energy consumption leads to the significant reduction of emissions of many pollutants. In this paper the new concept of dual power technology, dedicated to heavy road transport, was modelled and analysed by computer simulations. The combination of organic Rankine cycle and Brayton cycle was proposed, where the waste heat of fumes was recognized as a upper heat source, whereas the surrounding was adopted to be the lower one. Improvement of total energy conversion efficiency of the truck was the key success factor. Environmental friendly fluids (air and R123) were utilised. The operating parameters, power characteristics and energy streams (i.e. dispersion) of the system were evaluated, calculated and commented from the perspective of its theoretical profitability. The calculated net power capacity of analysed dual system was around 50 hp for 100% load. However, when the engine load is below 50% of nominal capacity, the power generation of combined system might be lower than in the case of single ORC system.*

Key words: *organic Rankine cycle, Brayton cycle, energy efficiency, road transport, waste heat utilization***.**

1. Introduction

Road transport, mainly due to the combustion of different liquid derivatives of petroleum (gasoline, diesel oil) and accompanying releases of toxic substances, has a negative impact on the natural environment. Therefore, many modern pro-ecological efforts and programs are strictly directed at the decreasing of emission factors and introducing environmental-friendly power technologies within the heavy duty trucks and cars (Merkisz et al., 2011; Merkisz-Guranowska and Pielecha, 2014). In relation to the mentioned fact, multiple of new solutions – new or improved combustion engines (i.e. Atkinson cycle), "clean" transport fuels or power sources (liquid biofuels, electricity generated in renewable energy sources, hydrogen), exhaust gas purification technologies (AdBlue system, Three Way Catalytic converters, high-efficiency particulate filters), waste heat utilization (thermoelectric generators, organic Rankine cycle implementation), reducing energy dispersion (new tires) – were analysed and adopted in a road transport industry. The positive efficiency of energy conversion process was used as a main objective in most of the referred technologies – the more mechanical power is generated from a unit of primary energy (fuel), the higher economic and environmental profits are achieved (Stelmasiak, 2011). The possible savings can be obtained in all main types of road vehicles – motorbikes, cars, lorries, buses. It is, understandably, crucial to include its technical specifications (temperatures of a waste heat streams, available power, technical arrangement) and limitations (mass, available space, operational safety) and identify the best solution for the long-term operational management of a two- and four-wheelers and trailers. Moreover, in order to strict meet emission standards and find the most favourable total energy balance, various states of engine's operation should be taken into account at the design stage - especially in the case of self-ignition engines (Chłopek, 2014).

The report (KOBiZE, 2015) conducted in case of *The Convention on Long-range Transboundary Air Pollution* (LRTAP) regulations, examined the emissions of different substances introduced from road transport in Poland in 2012 and 2013. Its results were compared with the assessments provided by (CSO, 2013) and presented in Table 1.

Despite the decline in total emissions by 2.6% in reference to 2012, in 2013 the road transport continued to be the main source of national NO_x releases – 31.9% in comparison to 30.5% from *Combustion in Energy Production and Transformation* (group 01 in SNAP methodology), 12.5% from *Other Transport and Mobile machinery* (SNAP 08) and 11.7% from *Combustion in Commercial, Institutional, Residential and Agriculture* (SNAP 02). In the case of dust (TSP, PM10 and PM2.5), where the road transport was responsible for 18.6% of annual emission of TSP in Poland (SNAP 02 – 40.1%, SNAP 08 – 2.3%, SNAP 01 – 8.8%), the degradation of natural environment was an effect of tires and brakes degradation and the abrasion of the road surface. The report (KOBiZE 2015) states, that the 3.3 % of reduction in TSP emission was obtained by a significant limiting of fuel consumption of road vehicles.

Many of the undertaken pro-environmental actions, leading to the energy efficiency improvements within the local road transport, are facing the ongoing increase in the number of vehicles. In 2012 in Poland, according to (CSO, 2015), the number of registered passenger car was 18,744 thousand. Respectively, the number of trucks in this period was 3,178 thousand. In both cases it meant a growth of 170-190% in comparison to the results of the assessments conducted in 2000. The road transport activity consumed in 2013 approximately 4,926 thous. m^3 of gasoline, 13,426 thous. m^3 od Diesel oil and 4,209 thous. m^3 of (LPG).

In this paper the case of heavy road truck was examined. Featuring larger available spaces and notable power capacities, the technical potential of analysed power technology is higher. The results of the assessments (KOBiZE, 2015, Taylor AMPK, 2008) proved, that despite the significant differences in the numbers of each type vehicles the total emission of TSP from lorries is higher than the passenger cars. The releases of NO_x were at the same level, while SO_2 and NMVOC stood at half. The heavy road transport was responsible for 0.05% of Polish SO_2, 12.45% of NO_x, 1.83% of CO, 4.36% of NMVOC and 2.21% of TSP. The comparison of described assessments was posted in the Table 2.

Table 1. Contribution of transport in Polish pollutants emissions (above 5% in the total share)

Year	**Activity**	**NO_x**	**CO**	**NMVOC**	**TSP**	**PM_{10}**	**$PM_{2.5}$**	**HCB**	**PCB**
		Mg	*Mg*	*Mg*	*Mg*	*Mg*	*Mg*	*kg*	*kg*
2012	transport	271,735	621,584	144,772	78,460	23,819	21,090	2.1	77.8
	share	*33.1%*	*22.2%*	*22.9%*	*19.3%*	*9.7%*	*14.5%*	*15.4%*	*10.9%*
2013	transport	255,083	581,157	139,890	75,879	21,465	18,709	2	60
	share	*31.9%*	*20.2%*	*22.0%*	*18.6%*	*8.7%*	*12.9%*	*15.4%*	*7.9%*

Source: KOBiZE (2015).

Table 2. Emission of selected pollutants in 2012 in Poland and the contribution of both heavy and passenger road transport

sector	**SO_2**	**NO_x**	**CO**	**NMVOC**	**TSP**	**Pb**
	Mg	*Mg*	*Mg*	*Mg*	*Mg*	*Mg*
TOTAL	858,626	819,211	2,791,083	630,290	406,429	553,534
road transport	1,300	271,300	653,600	145,700	80,100	15,000
lorries with maximum weight over 3500 kg	400	102,000	51,000	27,500	9,000	-
passenger cars	600	103,200	486,800	45,400	6,900	14

Source: KOBiZE (2015), Taylor AMPK (2008).

Beyond the obvious environmental (emissions) and economic (fuel consumption) aspects, implementation of highly efficient power technologies in trucks and lorries is significant from a purely legal point of view. Stringent requirements concerning emissions of various substances from truck engines have a substantial impact on the adaptations of the new projects. New EU standard EURO VI established the maximum release of NO_x at the level of 400 mg per 1 kWh of net power (80% less than in EURO V). In the case of solid particles, the emission can not exceed 10 mg per 1 kWh (66% less than in EURO V).

In another part of the paper the concept of novel dual power unit, dedicated to the heavy truck reciprocating engine, was presented and compared to the most popular concepts of waste heat utilization in road vehicles. Later, the combination of air Brayton cycle and regenerative organic Rankine cycle (with a R123 as a working fluid and) was analysed in order to maximize the waste heat utilization. The operating parameters of fumes, air and R123 were identified and fitted to the engine. The power characteristic was defined and commented. Finally, the improvements of the energy efficiency factor (corresponding to the fuel savings) were predicted as well.

2. Waste heat recovery in vehicles

2.1. General description

The idea of utilizing the waste heat and improving total energy efficiency factor in vehicles is relatively known and, for the long time, was investigated by many of the scientists. Other concepts, concerning reducing the fuel consumption in cars and trucks, are directed towards the limiting the mechanical power usage – with no reuse of waste heat. They include implementation of aero parts (using the modified bodies and extra aerofoils), stop-start systems (limiting the idle speeds) and new types of tyres (improving the grip). In the case of heat recovery and its conversion into more useful form, 5 main technologies were indicated in literature: organic Rankine cycles (with various configurations and working fluids), thermoelectric generators (using Seebeck effect), turbocompounds (turbochargers) (Katsanos et al., 2012) combustion process optimization and hybrid technologies. It is crucial to introduce both the waste heat and mechanical energy (i.e. from braking) integrated technologies to obtain maximum fuel conversion factor. In this case, the thermal potential of fumes and various coolants should be analysed and utilized first. The quantity of exergy accumulated in fumes has the greatest value (one third of the energy delivered with fuel) of all energy streams in most of the road vehicles (Katsanos et al., 2012).

2.2. Implementation of additional thermodynamic cycles

By the application of additional heat exchanger the extraction of heat from waste streams is possible and justified. It can be an effective source (upper) of heat in numerous thermodynamic cycles, such as organic Rankine or Brayton ones. In both cases, the heat of fumes is transferred into the working fluid, which is heated, evaporated or superheated. The size, construction and thermal parameters of heat exchanger should satisfy the demand of upper source of heat. Heated stream can be further vaporised (in expander) and drives the work machine (electric generator). The 3-6% increase of energy conversion efficiency in vehicle can be granted in this case (Aghaali et al., 2015). The energy recovery by making new thermodynamic cycles is very popular in high capacity power engineering (i.e. combined cycles with both gas and steam turbines), but in the case of transport it have a several limitations (including size and complexity). It is necessary to optimise the total mass and size of all components (heat exchangers, turbine, electric generator, condenser, pipes, valves) to ensure the positive mass to power ratio and obtain the effect of reducing the fuel consumption.

2.3. Turbocharger

This type of heat recovery is similar to many regenerative techniques in various thermodynamic cycles. Turbocharging is one of the most popular and justified ways to achieve higher thermal effectiveness within the vehicles with Diesel engine. Pressurized, hot fumes from the engine are directed to the turbine coupled with the air compressor. By increasing the pressure of air, higher mass stream can be introduced into the combustion zone. It leads to the reduction of emissions. Furthermore, the increase of engine power while reduction of mass (by 20%) is observed as well. When the turbocharger is installed in a heavy-duty vehicle, the fuel economy can be improved by about 3% (Arnold et al., 2001). However, this value must be easily enhanced and optimised - in passenger cars,

turbocharged diesel engines may reduce costs of fuel even up to 30–50% (Saidur et al., 2012).
The time delay between turbine start-up and first phase of acceleration (in the low engine speed range) is one of the significant issues of turbocompounds, which leads to the overheating and finally to permanent damages of bearings. Moreover, both the correct motor running and driving performance are hampered. In order to avoid mentioned disadvantages, the two-stage compressors (two units, each with different power capacity) should be installed. At lower speeds only the smaller one is used in order to enhance the torque. As a result, the fuel savings can be achieved in lower speeds, that is strongly recommended especially in town driving.

2.4. Electric power from AC/DC generators

When the fumes expanders are implemented in a car, the integration with electric generator is in most cases highly recommended. The additional source of electricity can be utilized in two ways: as a startup device (when the car is equipped with sufficient set of batteries) or power supply – in refrigeration (semi-trailers with refrigerators), household needs (recreational vehicles), lightning and more. The status of back-up unit can be obtained. The electric energy can be transferred outside and power other vehicles as well (in trams or trains - to the transmission network). Another key solution in electricity generation is KERS, which utilizes the mechanic energy of braking and transform it into the electric power. The energy dissipation and degradation of brakes are reduced. Moreover, the alternator load and fuel consumption can be significantly decreased. This system is currently used in small scale with different types of road vehicles.

2.5. Thermoelectric generators

This type of generators represents the set of devices that can be used to direct conversion of heat into the electricity. The Seebeck effect, discovered in 1821 by Thomas Johann Seebeck, is used in thermoelectric converters (i.e. thermoelectric generators, thermocouples). It involves generation of electromotive force between two areas with different temperatures, connected with two metals or semiconductors. In the case of road vehicle heat can be recovered from fumes, radiators or hot surfaces, but the direct industrial or commercial application within cars and trucks is nowadays limited, mainly due to relatively low performance capacities, material restrictions and economic weaknesses. When used in a proper way (identifications of crucial operating parameters are needed), thermoelectric devices can reduce the fuel consumption by 4.7% (Bang et al., 2016). They work quietly and have positive flexibility factors, especially in reduced engine loads. The low energy conversion efficiency is a main disadvantage.
The comparison of various waste heat energy solutions (turbocompounds, ORC units, thermoelectric generators, hybrid system) was presented in Table 3. Within the summary the main advantages and disadvantages of every technology were provided and fitted to the road transport.

Table 3. The comparison of different waste heat recovery technologies in road vehicles

System	Advantages	Disadvantages
Turbocompound	good BSFC reduction well known, relatively simple to introduce low volume, low cost	interaction to the engine limited BSFC at low loads low turbine efficiency with common technology
Organic Rankine cycle	very good BSFC reduction no interaction to the engine cooperates with various temperature ranges and heat sources (universal)	higher complexity high costs working fluids can be toxic or combustible
Thermoelectric generator	good BSFC reduction low weight no interaction to the engine	low efficiency large surface areas cost ineffective
Hybrid (engine + electric motor) system	lower fuel consumption more quiet	heavier (DC accumulators) higher complexity level high costs

Source: Aghaali et al. (2015).

3. Idea of dual Rankine-Brayton cycle

3.1. Technical concept

Analysed dual system executes both air supplied Joule-Brayton and organic Rankine cycles. This concept therefore varies from the conventional systems described in the literature. The integration of two mentioned cycles may be crucial to obtain higher thermal efficiencies in comparison to single ORC or Brayton cycle. In order to identify power capacities in different engine loads, system was modelled in the Mathcad 14.0 software (using available data from the specialised literature an publications) and analysed. Calculation model is presented in Fig. 1. and will be described later in this article.

The implementation of Brayton cycle was directed to enable cooling exhaust gases to the temperatures that still allow to use the organic media with low value of critical parameters in second stage of heat recovery - Rankine cycle. A wide range of fluids that are chemically stable, safe and environmentally friendly determines enhanced low-grade waste heat utilization, even from the condenser of Brayton unit. The upper heat source of the system is heat obtained from flue gases generated from the engine of a truck engine. Heat exchanger is located after the EGTS system. The lower source of heat is the environment. Exhaust gases coming from a turbocharged combustion engine, after cleaning in the EGTS, are passing through the heat exchanger, where the heat is transferred to the compressed and purified air. Afterwards, heated air flows to the turbine, where it expands and where thermal energy is converted to mechanical one in the blading system. The air turbine is acts similarly to gas turbines. However, its big advantage is the lack of contact of the blades with the exhaust gas (containing acid components and solid particulates), which significantly promote the higher operation time without emergency situations (Goliński and Jesionek, 2009). Another advantage of air turbine is the ability to remove the media directly into the environment without a need of purification. The outlet air flows to another heat exchanger, where organic media is heated. Cool air is then removed to the atmosphere.

3.2. Selection of working fluid

For the organic Rankine cycle R123 fluid was selected (2,2-dichloro-1,1,1 trifluroethane; $C_2HCl_2F_3$). It is a colourless, widely available on the market refrigerant with a relatively low impact on the greenhouse effect – ODP (eng. *Ozone Depletion Potential*) of R123 equals 0.012 and GWP (*Global Warming Potential*) - 76. In comparison, R11 have ODP is equal 1 and GWP - over 5000. Its applicability is included in the Montreal Protocol and allows the possibility of using it in the new HVAC and related systems up to 2020 year (2030 for developing countries). It is normally stable (even under fire exposure). It does not react with water, nor burn under typical fire conditions (R123 is non-flammable). Contact with human skin can cause irritation with only minor residual injury (level 1 of health hazard according to NFPA 704 standard). From a thermodynamical point of view R123 is treated as a dry fluid, therefore no wet steam should occur under expansion cycle in the turbine. This fluid is also often a subject to analyses for the implementation of ORC using various types of heat sources (Li et al., 2013; Peris et al., 2015; Sprouse and Depcik, 2013)

Table 4. Parameters of R123

Parameter	Value
Molecular weight	152.93 kg/kmol
Triple point temperature	166 K
Boiling point	300.97 K
Critical point	456.8 K, 3.662 MPa, 550 kg/m^3
Minimum working temperature	166 K
Maximum working temperature	600 K
Maximum working pressure	40 MPa

Source: Younglove and McLinden (1994).

Saturated R123 steam expands in the turbine. Afterwards the fluid is cooled and condensed to a liquefied state in air cooled condenser. At the end it is pumped to a steam generator where it is heated and evaporated again. Within the cycle is internal heat regeneration is performed via an additional heat exchanger. Fluid utilized in the turbine is used for pre-heating its liquid state, contributing to an increase in the overall energy conversion efficiency of the ORC system.

4. Results and discussion

In order to perform the calculations, it was necessary to determine the parameters of the exhaust gases produced by the engine of a truck for its various

loads. To this end, the adopted parameters given in (Katsanos et al., 2012) shown in Table 5.
A number of assumptions related to the operation and efficiency of the individual components of the system has been made. They are summarized in Table 6.
Piping and fitting pressure losses were not taken under consideration. Moreover, due to the operating parameters features of R123, it is assumed that the temperature of saturated steam at the inlet to the turbine shall not exceed critical temperature for this medium. Value of this temperature is equal to 175°C. Exceeding this temperature causes changes in physical properties of the selected fluid. Heat provided to R123 steam generator comes from two sources - the outlet air from the air turbine and a partially cooled flue gas (having a temperature of 140°C). Due to the internal heat regeneration, the temperature of the outlet air is limited to the top R123 temperature. In respect to the dew point exhaust gases are cooled to 110°C.
As indicated earlier, the heat and power balance calculation of the Joule-Brayton was conducted in Mathcad 14.0 using ORC fluids thermodynamic properties library. These libraries were developed by scientists of the former Department of Boilers and Turbines at the Wroclaw University of Science and Technology. The results for various engine loads are summarized in Table 7.
Two technical concepts of waste heat utilisation from the exhaust gas were analysed and compared – As indicated earlier, the heat and power balance calculation of the Joule-Brayton was conducted in Mathcad 14.0 using ORC fluids thermodynamic properties library. These libraries were developed by scientists of the former Department of Boilers and Turbines at the Wroclaw University of Science and Technology. The results for various engine loads are summarized in Table 7. dual Joule-Brayton – ORC system (Fig. 1.) and single ORC system (without the J-B cycle - point 1e from Fig. 1. is then equal to 2e and 4f' to 1f). The results are summarized in Tab. 7., 8., 9a. and 9b. (where "BC" index – Brayton cycle). It can be easily observed that while engine power is decreasing, efficiency of the whole cycle is also decreasing. However, value of differences between additional power generation is higher for ORC part. It is because ORC inlet temperature is fixed and mass flow of fumes is lower for each case. Total sum of power loss (with engine power decreasing) is higher than for only-ORC system. It means that presented solution if more sensitive to load changes.

Table 5. Exhaust gases parameters

Case	**Engine load** *%*	**Engine power** ***kW***	**Exhaust stream** ***kg/s***	**T_{1e}** *°C*	**p_{1e}** ***bar***
C1	100	366.6	0.4945	397.8	4.14
C2	75	277.8	0.4058	354.3	3.87
C3	50	183.6	0.2993	306.6	3.79
C4	25	90.0	0.1784	285.3	2.47

Source: Katsanos et al. (2012).

Table 6. Components parameters assumptions (on the basis of own engineering knowledge)

Joule – Brayton cycle		
Parameter	**Value**	**Unit**
Compressor efficiency	95	*%*
Number of compressor stages	3	-
Inlet compressor temperature	10	*°C*
Turbine total efficiency	90	*%*
Generator efficiency	97	*%*
Heat exchanger pressure drop	20	*Pa*
Outlet turbine pressure	1.2	*bar*
organic Rankine cycle		
Turbine efficiency	75	*%*
Mechanical efficiency	95	*%*
Dryness fraction of steam	saturated steam	-
Feed pump efficiency	85	*%*
Temperature accumulation in heat exchanger	5	*K*

Table 7. The results of Joule – Brayton cycle balance calculations (symbols in reference to Fig.1.)

Case	**Engine load** *%*	**P_{netBC}**	**$\dot{m}_{1a}$**	**t_{4a}**	**p_{2e}**	**η_{netBC}**
		kW_{el}	kg/s	°C	MPa	%
C1	100	14.37	0.520	234	0.394	10.0
C2	75	7.20	0.425	200	0.385	7.6
C3	50	1.77	0.311	162	0.359	3.3
C4	25	0.23	0.180	150	0.227	0.8

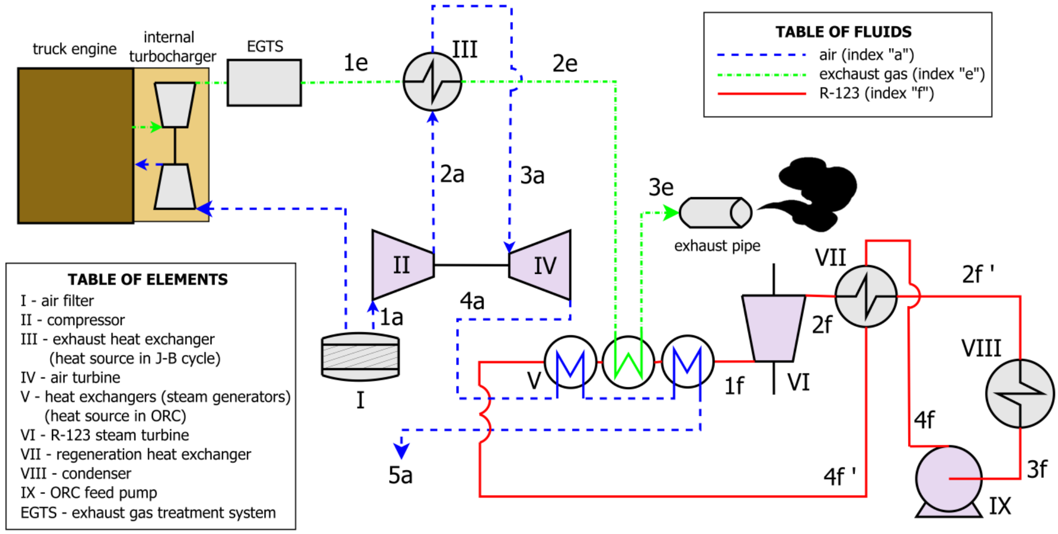

Fig. 1. Concept of the dual system (with the parameters and their locations)

Table 8. Fluid parameters at specific cycle points – dual cycle case (symbols in reference to Fig.1.)

Case	Engine load	$\dot{m}_{1f}$	P_{IX}	η_{netORC}	P_{netORC}	η_{netBC}	P_{netBC}	ΣP_{net}	η_{total}
	%	kg/s	kW	%	kW	%	kW	kW	%
C1	100	0.55	1.70	18.03	21.86	10.0	14.37	36.23	23.0
C2	75	0.38	1.18	18.03	15.13	7.6	7.20	22.33	21.0
C3	50	0.23	0.45	17.43	8.61	3.3	1.77	10.38	16.0
C4	25	0.12	0.19	16.47	4.33	0.8	0.23	4.56	14.0

Table 9a. Exhaust gas parameters at specific cycle points – non dual cycle case (symbols in reference to Fig.1.)

Case	Engine load	$\dot{m}_{1e}$	t_{2e}	t_{3e}	p_{2e}	p_{3e}	i_{2e}	i_{3e}	Δi
	%	kg/s	°C	°C	bar	bar	kJ/kg	kJ/kg	kJ/kg
C1	100	0.49	398	110	4.14	1.05	848.9	534.1	314.8
C2	75	0.41	354	110	3.87	1.05	799.3	534.1	265.2
C3	50	0.30	307	110	3.79	1.05	747.0	534.1	212.9
C4	25	0.18	285	110	2.47	1.05	722.7	534.1	188.6

Table 9b. Fluid parameters at specific cycle points – non dual cycle case (symbols in reference to Fig.1.)

Case	Engine load, %	Q_{inORC}	t_{1f}	$\dot{m}_{1f}$	η_{netORC}	P_{netORC}
		kW	°C	kg/s	-	kW
C1	100	155.70	175	0.71	0.18	28.07
C2	75	107.65		0.49		19.41
C3	50	63.73		0.29		11.49
C4	25	33.66		0.15		6.07

The results of simulations performed in Mathcad software were used also to develop the characteristics defining the amount of the additional power gained from both systems depending on engine load, fig 2. It can be seen that there is a point where characteristic intersect each other. This means that the presented solution is economically beneficial only to a certain values of engine power. As there is no possibility of sudden disabling of the system (due to the continuous circulation of medium and high temperature), the system should be activated only in perspective of a longer transit of truck at a constant speed.

It was also crucial to determine the losses in the system and their values. For this reason Sankey's charts for different load cases were elaborated. They are summarized in Fig. 3a – 3d. It may be clearly observed, that the greatest loss of energy resulting from discharge to the environment. This is due to the adoption of a relatively high temperature (110°C) of exhaust gases from the system. This temperature is to protect components of the system before condensation of exhaust gases.

To assume the fuel savings when total energy potential from dual system is transferred to transmission shaft, the energy efficiency factors were used. The nominal load was taken under further considerations. Two cases of 500 hp (375 kW) truck were compared: 1) combination of 450 hp (337.5 kW) Diesel engine (net power) and 44 hp (33 kW) from ORC-Brayton system, 2) single 494 hp (370.5 kW) Diesel engine. In the first case, total energy efficiency factor, related additional electricity transmission and different mechanical loses, was set as 0.88. Therefore, in both cases, the total driving power was 494 hp (370.5 kW). Diesel consumptions per 10 km for two different power classes engines (450 hp and 494 hp) were assessed on the basis of (WNRI, 2012) - the values of 4.4 litres per 10 km and 4.5 litres per 10 km, respectively, were assumed.

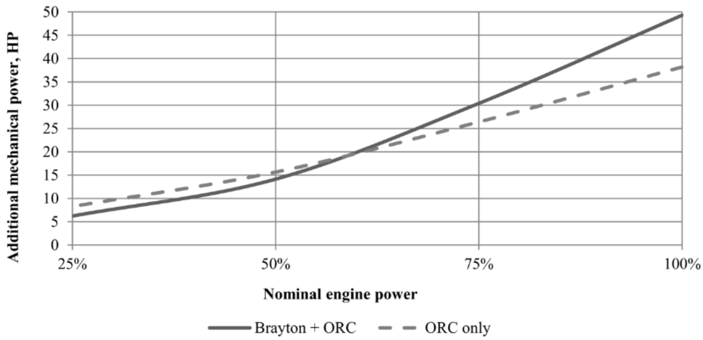

Fig. 2. Comparison of Brayton-ORC (Σ power from Tab. 8. in HP) and single ORC (P from Tab. 9b. in HP) additional power capacities

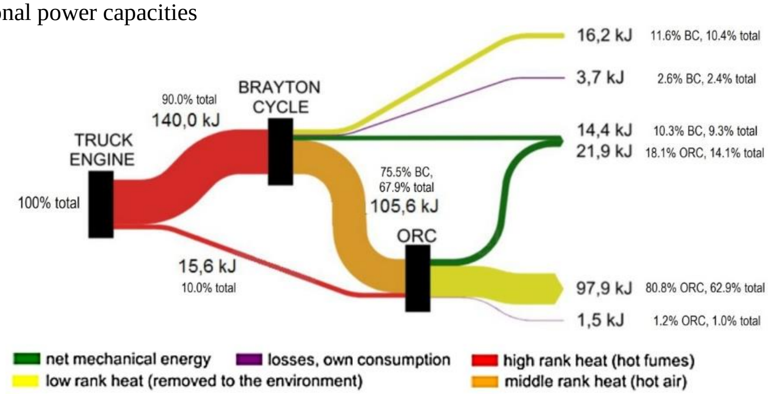

Fig. 3a. Sankey diagram for the dual ORC-Brayton system at 100% load

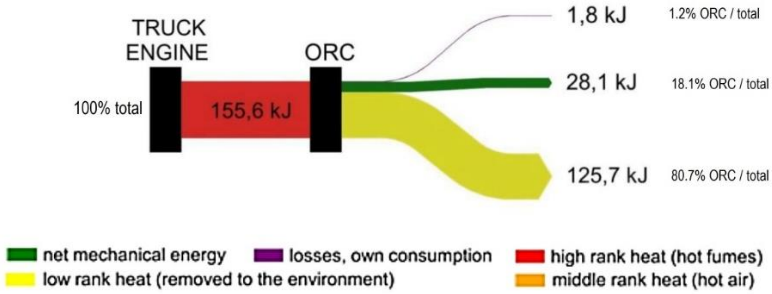

Fig. 3b. Sankey diagram for the single ORC system at 100% load

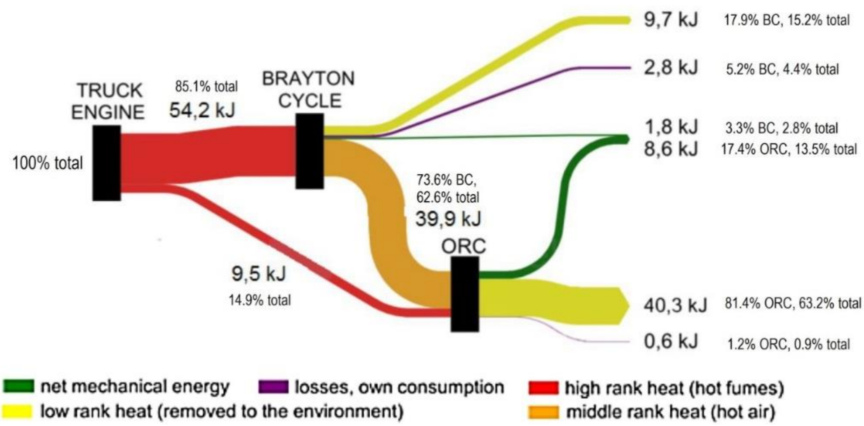

Fig. 3c. Sankey diagram for the dual ORC-Brayton system at 50% load

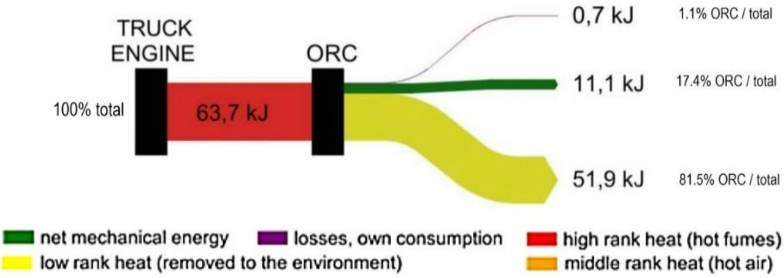

Fig. 3d. Sankey diagram for the single ORC system at 50% load

5. Selection of ORC expander

During the design cycle turbines powered by organic fluids there are a plenty of conditions that must be met in order to achieve optimum efficiency and trouble-free operation in future. This requirements are directly connected to features of the organic fluids. High density of the refrigerant at the turbine inlet and low mass flow rate demands the use of special design aimed at the fulfilment of the minimum height of the blade condition. In case of axial turbines, in order to minimize secondary flow losses and retaining an appropriate relative roughness value, the minimum height of the blade must be above 15 mm (Tuliszka, 1973). To fulfil this condition, it is necessary to use partial arc admission for the first few stages of turbine. Unfortunately, application of partial arc admission adversely affects the efficiency. However, there are methods, which can reduce the losses generated by this design. These include the use of plenum chambers, anti-wind age rings and stators clocking, (Stępień, 2013).

Another characteristic feature of organic fluids is low value of the speed of sound. In presented system, it is equal 86 m/s at the inlet and 136 m/s at of the turbine outlet. Owing to the fact that there is a Mach number limit (Ma = 0.96) (Costal et al. 2015), below which shockwaves do not affect the efficiency, it is therefore necessary to assume low enthalpy drop at each stage during the design process. Authors estimates indicate that in order to achieve the enthalpy drop for the entire machine and to meet the Mach speed limit condition it is necessary to use 7 axial. In the first three stages partial arc admission is required. The ratio of the specific volume between the inlet and outlet of the turbine 64. Such a large increase in specific volume results in the need for long and 3D twisted blades. The cost of a pure axial turbine, consisting of a number of precision made components may be too high for wide application in road transport. An alternative to axial turbines can be inflow radial turbines, which are widely used in the automotive industry. Their main advantage is small size (compensated by high RPM) while maintaining high efficiency. This design allows to use small height of the blade at the inlet to the turbine without a need of using partial arc admission. Furthermore, a larger enthalpy drop can be applied. Unfortunately, some of its design features speaks against this solution. Although some designs consisting of more than one radial inflow stages are known, (Kang 2016, Han et al. 2014), the complication of the whole machine must be taken into account. This is basically caused by the necessity of using reversing channels with complex geometry. In addition to the raising costs of a unit, reversing channels generate pressure losses resulting in reduction of overall efficiency of the machine. For the realization of a given enthalpy drop it is required to 2 or 3 radial stages. However, if the two stage design will be chosen an exceed of the Mach number limit will occur which can result in louder turbo-set operation.

Having considered both, advantages and disadvantages of axial and radial inflow designs authors suggest a combination of both solutions through the use of units consisting of one radial stage and 4 axial stages – figure 4. This concept would to provide favourable flow in the area of low specific volumes through the use of radial inflow stage (inlet parameters), and high efficiency through the use of longer blades of conventional axial design for intermediate and low pressure part of the turbine. Operation at higher RPM than synchronous will reduce the size of the entire unit, making it cheaper to manufacture.

An interesting alternative for realization of the expansion process can be obtained by choosing a radial outflow turbine. In terms of manufacturing issues it has a major advantage over its competition, which is a simplicity of the steam path. Rotors and stators of this type can be produced only with a 3-axis CNC milling machine. When it comes to mass production, these parts can be casted or forged. Low operating temperature allows the use of temperature resistant aluminium alloys, which can be easily subjected to the above-mentioned methods of manufacturing.

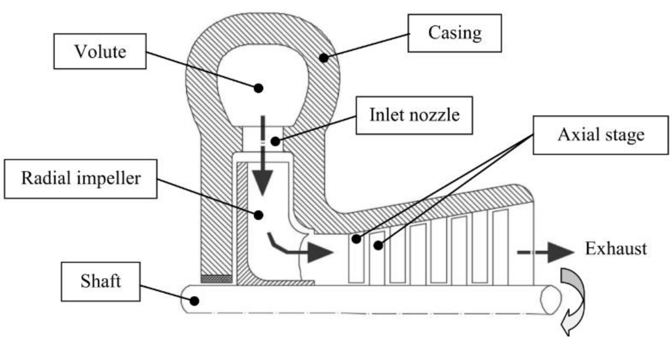

Fig. 4. The concept of ORC turbine

The main disadvantage of outflow turbines is the necessity to use more stages, which is related to the flow kinematics. The difference of circumferential velocity between inlet and outlet of the stage is always negative, thereby the output work of a single step is reduced. An unquestionable advantage over other solutions is that the passage area naturally increases along the expansion. For this design, in comparison to axial turbines, compactness is also advantageous, (Pini et al., 2013). Some reference state, that it is also possible to achieve higher efficiency, (Spadacini et al., 2015). What is more, in favor of outflow turbines speaks facilitated possibility of no axisymmetric endwalls contouring on both, hub and shroud walls. This can also be performed with 3 axis CNC milling machine. No axisymmetric endwalls contouring significantly reduces the generation of a secondary flow losses which is especially important when dealing with short blades (Knezevici et al., 2009).

6. Conclusions

The results quantified the theoretical influence of the energy recovery system implementation on efficiency of the truck engine. The utilization of dual system consisting of Joule-Brayton and ORC cycles, resulted in increasing of engine power output by 50 HP for maximum load. In addition, it was possible to use non-toxic working medium (air, R123), which is favour to this concept developed in comparison to other proposed solutions using this concept (Peris et al., 2015, Sprouse III and Depcik, 2013).

The comparison of power to load characteristics of both system (dual and single) showed that up to a certain point the dual system generates more power than the single RC cycle. This shows the justification of the developed concept. On the other hand, the calculations have also shown that in order to maintain maximum efficiency, dual system should be used only if there is a perspective of long ride at constant speed, without high differences in engine load. The greatest energy loss are caused both by the use of the Joule-Brayton's air cycle, and high outlet exhaust temperature.

The utilization of power (mechanical or electrical) generated in mentioned dual ORC-Brayton system might be implemented within both drive unit (i.e. as an additional source of electric power in a conventional-hybrid system) or external road-related devices (in a refrigerated trucks or tankers). The electricity generated may be stored in a batteries and supplies loading, unloading and reloading activities as well. It has been shown that, the covered fuel savings per 1000 km in following operating conditions shall be 10 litres. The resulting value is relatively low and proves the depreciates the legitimacy of its application as a direct drive unit in the current economic heavy - transport background. Even so, the potential of 35 kW in electricity is still justified to its technical utilization.

The future work is to develop a concept a cycle optimization algorithm, including obtain the optimal division of fumes stream between Brayton and ORC cycle. It is also crucial to find the lowest possible temperature of outlet fumes, to increase efficiency of whole the system.

Abbreviations

ORC	– Organic Rankine Cycle
SNAP	– Standardized Nomenclature for Air Pollutants
PM_{xx}	– Particulate Matter, where xx – maximum size of a particle in μm
LPG	– Liquid Petroleum Gas
TSP	– Total Suspended Particles
NMVOC	– Non-Methane Volatile Organic Compound
KERS	– Kinetic Energy Recovery System
EGTS	– Exhaust Gas Treatment System
RPM	– Rotations per minute
CNC	– Computer Numeric Control

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