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Concept for energy recovery in city buses based on organic Rankine cycle system

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Abstract

This paper presents a concept for energy recovery in city buses, based on organic Rankine cycle (ORC) system modeling. ORC modeling provides preliminary parameters which enable estimation of transient state energy flow, especially at those points in the system where it can mostly easily be recovered (exhaust, cooling and brakes). The paper describes an easy-to-use procedure for time-domain analysis of velocity waves in a real driving cycle. The results were combined with a dynamic system model of a city bus in block diagram environment, taking into account the influence of parameters assumed to have the greatest impact on the model's accuracy. Average fuel consumption and producer specifications (maximum vehicle speed, rate of acceleration) were used as reference parameters. A procedure was developed for tuning the simulation model according to the reference parameters. The results of the simulation, based on real measurements of Lodz city traffic parameters, are discussed and an estimate is made of the potential for energy recovery in city buses.

Keywords: City bus dynamics; Waste heat energy recuperation; Simulink

1 Introduction

Energy consumption is a significant issue for public transport providers. A considerable proportion of the energy supplied to internal combustions (IC) engines is wasted, as heat via coolant (30%) and exhaust (40%) systems [1–3]. Approximately 70% of energy provided via fuel to the vehicle is irrevocably wasted.

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However, a similar amount is generated by the braking system during a normal city driving cycle. Vehicle dynamics, determined by the driving cycle, have the greatest effect on energy consumption, especially in the case of city buses, which must stop frequently for passengers and due to traffic [4,5]. Therefore, analyzing the transient states of a vehicle as accurately as possible is crucial for efficient operation.

The global positioning system (GPS) technology has created new possibilities for vehicle dynamic simulation [6–8]. It is now possible to acquire real city driving data including the precise position of the vehicle (speed and acceleration can be calculated) in the time domain. This data can be used to set the vehicle's longitudinal position, and energy consumption parameters are estimated for this purpose. On the other hand, a relatively simple model of vehicle dynamics can be constructed in the commercial numerical computing and block diagram (Matlab/Simulink) environment, which can estimate the amount of heat wasted. In this paper, a system based on an organic Rankine cycle is proposed for recovery of waste heat in city buses.

2 Numerical model

A numerical model of city bus dynamics (CBD) can be defined as a mechatronic system, as it contains mechanically and thermodynamically coupled units with a feedback control system. The model below presents the flow of information by means of signals linked to the steering systems, and the flow of mechanical power between the engine, gearbox and vehicle, mainly by mass in forward motion. The model is based the procedure presented in [9,10].

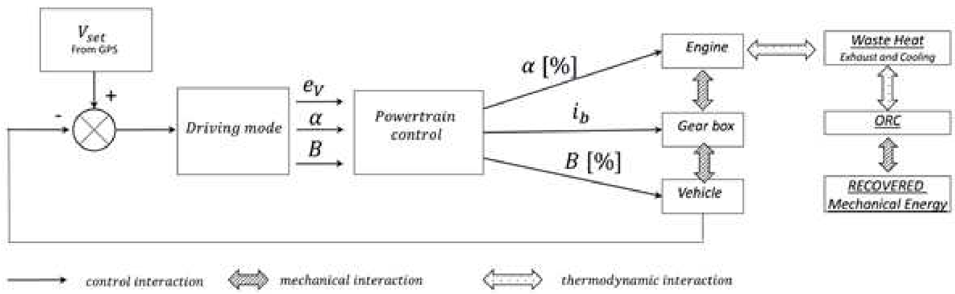

Figure 1: Model of city bus dynamics [1]; V_{set} – set vehicle speed, e_v – control error (difference between set and actual vehicle speed), α – engine load, B – intensity of braking, i_b – required gear ratio.

The main steps in the model can be summarized as follows:

- downloading measurement data collected via GPS as a V_{set} simulation time function (V_{set} is the demanded vehicle speed, proposed model selcts vehicle dynamic parameters in order to track set vehicle speed);
- comparison with actual vehicle velocity and estimation of control error;
- selection of driving mode on the basis of the value set for acceleration calculated as a derivate of the set velocity. Driving mode is selected between acceleration and braking. The driving mode block also transports the control error, defined as the difference between actual and set vehicle velocity $e_V = V_{set} - V_{actual}$;
- the powertrain control block is a control loop feedback mechanism. It consists of three independent feedback loops which control the engine load and degree of braking. Powertrain control blocks also realize shift logic based on shifting lines, which switch the required gear ratio, i_b;
- the engine block estimates engine torque as a function of engine load and the angular velocity including engine inertia. Two modes of energy transport are indicated: mechanical (arrow dashed, Fig. 1) and thermodynamic (clear arrow, Fig. 1), as well as signal transmission (solid line, Fig. 1);
- the gear box and vehicle block estimates the influence of automatically chosen gear ratios and movement resistances (rolling, inertia of masses in linear and angular movement, aerodynamics).

3 Recording city bus movement parameters in real life conditions

The research focused on city buses (Lodz line 86), travelling between two Lodz districts, Polesie (more specifically, between its two parts, Retkinia and Stare Polesie) and Śródmieście. The registered cycles comprised almost the whole route, from Maratońska St to Plac Dąbrowskiego terminal, and the return trip in the opposite direction. The choice of bus route was made based on the following criteria:

- at least part of the route should be within the city centre (which is typical of most bus routes in Lodz city region);
- the route cannot run along streets where major roadworks are being carried out, as frequent traffic jams affect traffic flow and measurement results;
- the route cannot run along streets with bus lanes (so called ‘buspasy’), as these only represent a fraction of the city bus routes.

The 'Logger GPS' mobile phone application was used to record measurements. This easy-to-use program enables chosen movement parameters, such as latitude and longitude, elevation above sea level, GPS time and speed, to be measured as GPS traces which is are saved in the phone's memory. One of the advantages of using a mobile phone is its power independence, as opposed to other devices, such as car navigation systems, which have a very short built-in battery life and so would have required extra charging for research purposes. The chosen device makes it possible to carry out measurements for up to several hours without extra charging. Another reason why the Logger system was selected is the possibility of choosing a sampling rate of 1 Hz. The selected Logger records GPS traces in the well-known GPX format, which makes processing easier.

It was decided that recordings of the selected routes would take place at various times of the day and on different days of the week, so that they would reflect a range of traffic situations. The research was carried out between Monday and Friday, not during rush hour (because of traffic jams), from Monday to Friday at peak rush hour, and at weekends (Saturday and Sunday, regardless of the time of the day).

4 Model of the city bus dynamics

The block diagram of the city bus dynamics (CBD) model in Simulink (Fig.1) was built in the Matlab R2014a environment, Fig. 2. The proposed model was

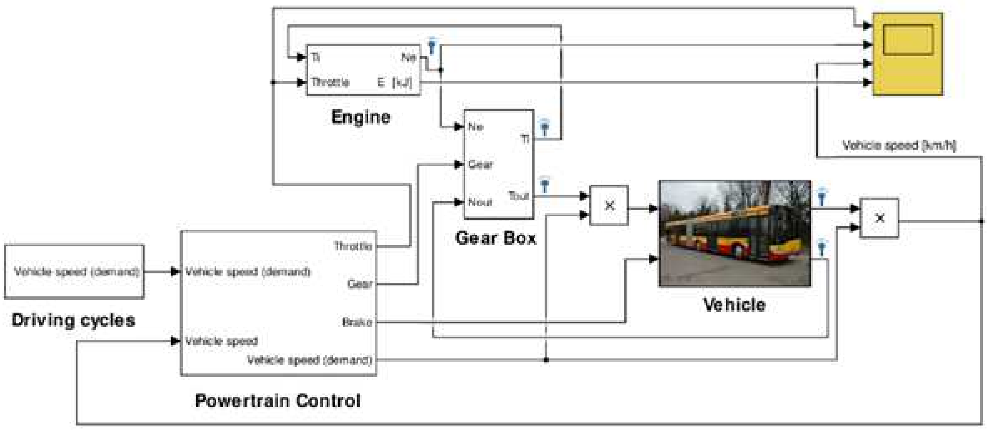

Figure 2: Simulink model of the city bus powertrain [11].

based on a tutorial model, the so called *Modeling an Automatic Transmission Controller* [11]. This model consists of six equations, two of which define engine dynamics and vehicle dynamics and four others describe how the powertrain's elements transmit power between each other.

The bus under consideration was Solaris Urbino 18 bus equipped with a DAF PR228 engine with maximum power of 231 kW and maximum torque of 1300 Nm available at constant angular velocity 115– 178 rad/s. The curb weight of the bus is 16 000 kg and the gross vehicle weight is 28 000 kg (for further calculations, it was assumed that the maximum mass of vehicle was 22 000 kg, which is 79% of the maximum weight) [12]. The main gear ratio is 6.2 and the gear ratios for subsequent gears of automatic gearbox are respectively: 3.43, 2.01, 1.42, 1.00, and 0.83. Universal engine torque characteristics, T_e , were modeled as a 3D map against angular engine velocity, ω_e, and engine load percentage, α. This is a quasi-static analysis and was therefore extended by a dynamics model of the engine itself, taking into account its inertia. The universal engine torque characteristics were based on the external characteristics of the engine, assuming linearity for fractional loads.

It was assumed that the basic driving cycle of the research object was torque-determined motion, transferred from the engine to the driving wheel. Total indispensable energy expenditure, expressed as the product of supplied fuel, G_L, and its caloric value, W_d, referred to as total energy intensity, is defined by total energy consumption in motion and energy losses in the engine and the powertrain

$$G_L W_d = E + \Delta E_e + \Delta E_p \ , \tag{1}$$

where: E – energy consumption during motion, ΔE_e – losses resulting from energy transformation in the engine, ΔE_p – losses incurred during energy transmission from the engine to the driving wheels, moreover

$$G_L W_d = \frac{E}{\eta_p} \ , \tag{2}$$

where η_p is the, i.e., driving efficiency, the product of engine efficiency, η_e, and drivetrain efficiency, η_t

$$\eta_p = \eta_e \eta_t \ . \tag{3}$$

The engine model was built based on a '2D look at table', by downloading the universal torque characteristics. The input values for his block are the engine load and angular velocity, α, and ω_e, respectively, while the output value is the engine torque, T_e,

$$I_e \dot{\omega}_e = T_e - T_1 \ , \tag{4}$$

where: I_e – engine moment of inertia, $\dot{\omega}_e$ – angular engine acceleration, T_e – engine torque, T_1 – torque of the pump in the torque converter.

The torque converter was modeled based on two zero-dimensional characteristics: coefficient K and dynamic gear shift, i_d, (both via kinetic shift, i_k. The following characteristics were formulated based on criteria for selecting a torque converter as well as producer data for the gear box used in Voith Diwa D864.5 buses. The following equations show the correlations between basic torque converter parameters:

$$T_1 = \frac{\omega_e^2}{K^2} , \tag{5}$$

however dynamic gear ratio is described by

$$i_d = \frac{T_2}{T_1} , \tag{6}$$

where: T_1 – the torque on the output shaft torque converter, T_2 – torque on the input shaft torque converter. K factor and i_d are function of the transmission ratio between input and output torque converter shafts.

Gear shifts are performed by gears and have the following effect on the relations between input and output parameters:

$$T_{out} = i_{gear\ box} T_{in} , \tag{7}$$

where: $i_{gear\ box}$ – gear box ratio, T_{in} – gear box input torque, T_{out} – gear box output ratio. Powertrain total gear, vehicle inertia and changes in resistance affect vehicle dynamics as shown by the relation

$$I_v \dot{\omega}_{wheel} = i_{total}(T_e - T_{load}) , \tag{8}$$

where: I_v – vehicle inertia with reduced mass ratio versus gear, $\dot{\omega}_{wheel}$ – wheel angular acceleration, i_{total} – drivetrain overall gear, T_{load} – vehicle load moment, including braking moment (a percentage of the maximum braking force) and resistance (inertia which is reduced mass to forward motion, rolling resistance and aerodynamic resistance via vehicle speed squared and its aerodynamic parameters).

Fuel consumption, G_L, for distance L in movement with driving force, F, is

$$G_L = \frac{1}{\eta_p W_d} \int_0^L F dx , \tag{9}$$

while mileage fuel consumption, Q, (i.e., for a unit of the distance travelled) is described by relation

$$Q = \frac{E}{\eta_p W_d L} \; . \tag{10}$$

The proposed Eqs.(8)–(10) can be used according to the following assumptions:

a) the vehicle wheels are rigid, longitudinal and latitudinal elasticity are not considered;
b) rolling resistance is assumed as a vehicle speed function;
c) aerodynamic resistance is assumed as a front area and drag coefficient function of the squared vehicle speed;
d) the powertrain components are rigid, whole system stiffness is assumed as a first-order transfer function to suppress internal vibration;
e) inertia of rotating mass was covered by a mass reduced coefficient included at the vehicle inertia;
f) instantaneous states of engine torque characteristics are not considered;
g) the gear shift logic is simplified, and does not cover instantaneous states;
h) the braking force is estimated according to the vehicle dynamics, without considering the internal dynamics of this system (driver and braking system delay, the impact of lateral forces);
i) the unknown coefficient (reduced mass ratio against the subsequent gear shift change cycle is known);
j) real vehicle inertia moment is reduced to mass in forward motion;
k) simplified fuel consumption characteristics are estimated according to the procedure presented in the next paragraph.

5 Procedure for tuning the simplified CBD model

The proposed model is tuned using two independent tools to ensure high correspondence to real life conditions. These tools provide a method of estimating fuel consumption in the time domain. A similar method was proposed in previous papers by the author [9,10], as a part of the process of creating continuous variable transmission (CVT) drivetrain steering maps. The following parameters, available in reference books or provided by the producer, were selected for use in the analysis:

- reduced mass ratio against the subsequent gear shift change cycle,
- real vehicle inertia moment reduced to mass in forward motion,
- simplified fuel consumption characteristics.

The main steps in the two methods are:

- identification, selection and analysis of the model's key parameters to estimate the accuracy of the final results. In this case, it was decided to focus on the parameter of energy consumption;
- introducing real-life reference parameters to enable tuning of the model to real life conditions. In this case these were as follows:
  - acceleration time from stationary to velocities of 16.6 and 22.2 m/s;
  - average monthly fuel consumption by mileage: July 2013 – 54.77 dm^3/ /100 km, August 2013 – 23.79 dm^3/100 km;
  - average fuel consumption by mileage for a selected route measured by an on-board computer. For route C2 this was 54 dm^3/100 km;
  - registered movement trajectories under real life conditions, classified as routes C1 to C6;
- multiparameter optimization using the Simulink signal constraint. With this device, a matrix of probable solutions can be searched to obtain specific values relevant to real life conditions.

An optimization process was performed in several stages.

Stage I

Choice of drivetrain parameters The parameters which determined the vehicle's dynamics were 'tuned' in a simulation process of accelerating a bus from stationary to 16.6 or 22.2 m/s. The automatic regulating system α and B were disconnected. Constant $\alpha = 100\%$ was introduced, corresponding to the maximum engine load, while for the braking system a 0% constant was set, resulting in its deactivation in the simulation test. The vehicle accelerated in 18 s to a speed of 60 km/h (as stated in the producer's data information), then reached a maximum speed of 80 km/h and remained at constant speed. The outcome of the procedure provided carefully selected and adjusted simulation model parameters, defining the reduced mass ratio and reduced vehicle inertia. In this way, the dynamics of the model were tuned to ensure high correspondence to real life conditions.

Stage II

Choice of powertrain control system parameters Two proportional-integral-derivative (PID) regulators were used in the proposed simulation model, in order to calculate the downloaded driving cycle (C1–C6) as a parameter defined by a steering signal: an automatic engine load degree regulation system α and braking power steering ratio B. In order to improve the quality of regulation, it was necessary to perform value tuning of the six PID regulator parameters (three for each regulator). An indicator of regulation quality was defined, as the maximum difference between the set value and the executed value, taking into account the requirements for exhaust tests, i.e., below 5% (1 m/s) maximum speed value.

Stage III

Choice of parameters related to fuel consumption A simplified characteristic was correlated with the following data: average monthly fuel consumption and fuel consumption by mileage for route C2.

6 Selected results of simulations

Based on the model presented above and a hybrid procedure of implementing the research results in real life conditions, a number of tests to further optimize the model were performed, related to the choice of solver and the time frame for the calculations. Solver ode 45 [17] was selected and a variable step, enabling a real-time test route consisting of 3500 s to be simulated in just a few minutes. Simulation tests were carried out for all six defined routes (C1–C6). Figure 3 presents example engine speed changes for a vehicle speed cycle and estimated energy consumption (cycles and average parameters calculated from the start of the cycle) for the first 200 s of cycle C2.

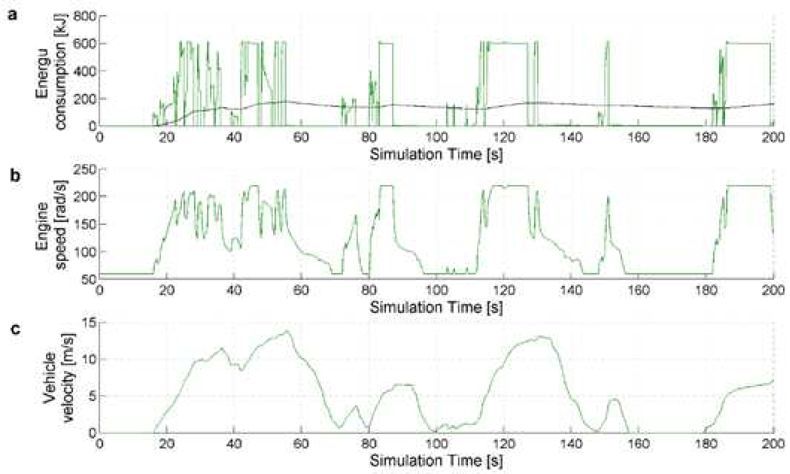

Figure 3: Energy consumption (a), engine angular speed (b), vehicle velocity (c) for cycle C2.

6.1 Energy consumption analysis of the city bus based on CBD model

The simulation test results were divided into two groups:

Group I This group (cycles C2 and C6) comprised bus cycles run from Monday to Friday, except during peak times. Cycles in this group showed higher energy consumption per kilometer due to higher traffic congestion levels, compared to weekends. This resulted in more frequent stops (not including regular bus stops, of which there are the same number in each cycle), and extra fuel consumption as the vehicles were set in motion again (during the initial acceleration phase).

Group II The second group (cycles C4 and C5) comprises only rush hour cycles (from 7 a.m. to 9 a.m. and from 3 p.m. to 5 p.m.) run from Monday to Friday. Energy consumption for this group is the highest.

Table 1 presents mileage fuel consumption estimates for individual cycles on each route. An analysis of energy consumption in all groups shows that total energy consumption in cycles run from Monday to Friday, not including peak times, was approximately 25% higher than in cycles run at weekends and approximately 16% lower than in rush hour cycles. Without dedicated bus lanes, total energy consumption in rush hour cycles was almost 55% higher than at weekends. Fuel consumption reached values of more than 600 kJ.

Table 1: Average energy provided at fuel consumption for individual cycles.

Cycle no.	t[s]	L[km]	a_{aver} [m/s^2]	Average energy provided by fuel kJ/mileage energy provided by fuel [J/m]	Average fuel consumption [l/100 km]
C1	3274	16.5	0.57	70/21	48
C2	3066	16.9	0.60	92/30	54
C3	3172	17.5	0.55	62/20	49
C4	2919	18	0.64	105/36	60
C5	2941	17.3	0.62	99/34	58
C6	3019	16.8	0.59	84/28	52

After analysing the average energy consumption and fuel consumption each group, urban city bus route 86 Lodz operating Monday to Friday (except during rush hour) was chosen as the most representative cycle for route 86 buses in the Lodz city region. Two factors influenced the choice of this route. Firstly, of the three route 86 bus groups, this group use the highest percentage of all cycles run over the entire week. Secondly, average energy consumption values and fuel consumption in this group are very close to the average results when the other two groups are included (cycles registered at weekends and during peak times balance each other to a level that may be considered representative). Figure 4 shows the representative driving cycle of urban city bus 86 Lodz.

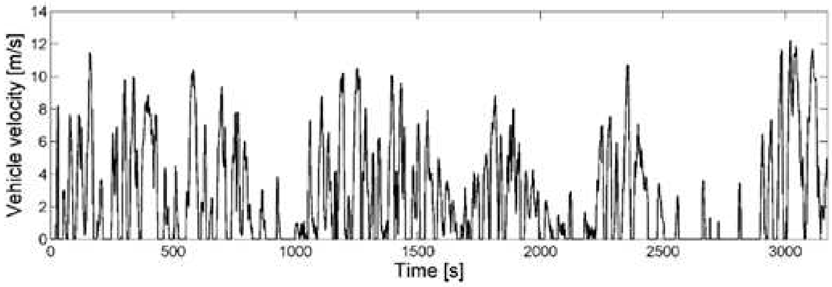

Figure 4: Urban city bus 86 Lodz driving cycle.

The total energy intensity in this representative cycle is 84 kJ, which is divided between: motion energy intensity (27.7 kJ), cooling system losses (25.2 kJ), exhaust system losses (21 kJ) and other losses (10.1 kJ). The sum of cooling system losses and exhaust system losses is approximately 46 kJ while average fuel consumption is around 52 l/100 km.

6.2 Preliminary ORC system location

Having decided to position the interface between the exhaust and the ORC system, measurements were taken of exhaust gas temperatures. Thermal camera images were made of the running side and top engine compartments (Fig. 5). The velocity and volume flow ratio of exhaust gas flows were also measured, along with temperature, for a range of different angular engine velocities (Tab. 2).

An alternative location for placement of components of the system without decreasing passenger space is in the roof (Fig. 6). Auxiliary components, such as the pump or controller, which do not require access to the exhaust system could be installed in a roof compartment.

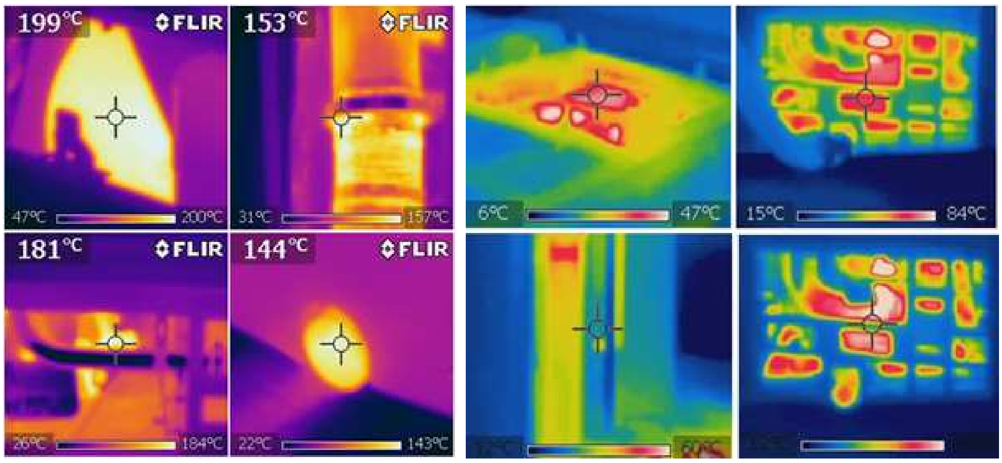

Figure 5: Selected thermal camera measurements of exhaust systems.

Table 2: Results of exhaust gas measurement for Solaris Urbino 18 bus.

ω[rad/s]	Under load		Not loaded		T [°C]
	c_s [m/s]	V_s [m^3/s]	c_s [m/s]	V_s [m^3/s]	
80	6.8	0.081	6	0.070	55
100	7.6	0.090	7.7	0.091	61
120	13.5	0.160	9.8	0.120	72
150	24	0.285	13.5	0.160	94

where: ω – engine angular velocity, c_s – exhaust gas velocity, V_s – exhaust gas flow volume, T – exhaust gas temperature.

Figure 6: Preliminary ORC system location.

6.3 Example calculation of ORC waste heat recovery

The Rankine cycle is a thermodynamic cycle which converts heat into work. Heat is supplied externally to a closed loop, which usually uses water as the working fluid. The organic Rankine cycle (ORC) is a Clausius-Rankine cycle in which vapour of low boiling fluid is used instead of water as the working fluid. In recent years, ORC has become popular in energy production processes, due to the fact it enables exhaust heat with low energy and temperature to be used [1,2,3,14]. The organic Rankine cycle is a thermodynamic process in which heat is transferred to a fluid at a constant pressure. The fluid is vaporized and then expanded in a vapor turbine which drives a generator, producing electricity. The spent vapour is condensed into liquid and recycled back through the cycle. This thermodynamic cycle is similar to the reference Rankine cycle, itself based on the following sequence: evaporation using heat provided by the heat source; expansion, producing output power; condensation, discharging unused residual heat; fluid pressure augmentation, with pump. The organic working fluid is compressed with a pump, which forces the fluid through a regenerator. The regenerator enables preheating of the liquid working fluid by desuperheating the expanded vapor. The preheated working fluid is then evaporated, superheated and expanded in a turbine, which drives a generator. The desuperheated vapor is condensed in a condenser. If the temperature level of the condensation is sufficiently high, as for example in the case of biomass combustion, the waste heat can be used in a district heating net. If low temperature heat is used to drive the ORC, such as geothermal or waste heat, the condenser is cooled by means of cooling water. A model was developed to study the effects of the organic Rankine cycle system on energy recovery and to estimate its potential in downsized form.

Simulation of the system was constructed using commercial MATLAB/Simulink [18] and CoolProp software [19]. It was used to describe the processes and dependences occurring in the ORC system. CoolProp is an open-source, cross-platform, free property database based on C++. It includes pure fluids, pseudo-pure fluids and humid air properties, and was used to input working fluid data into the simulation.

Figure 7 illustrates the pattern of changes in electric power recovered by the ORC system turbine, obtained from simulation tests carried out on a model built in a similar manner to the CBD model presented in this paper. The input values are: turbine efficiency; pump efficiency; source properties (temperature, mass flow, pressure); coolant properties (temperature, mass flow, pressure); cycle liquid properties (R245fa, from CoolProp software). Output values are: thermal efficiency, heat input and output; work generated by the turbine and pump; cy-

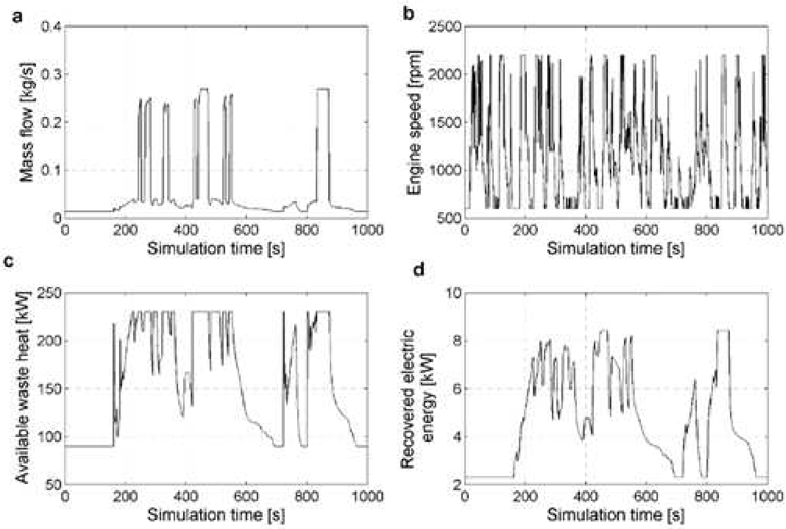

Figure 7: Selected parameter variations: a – mass flow, b – engine speed, c – available waste heat, d – recovered electric energy.

cle fluid flow; temperature, entropy, enthalpy, pressure [1,2,3,13,14]. The model was tuned in a process similar to that described for CBD. Results from experimental research carried out at municipal transport company at Lodz (MPK) city transport terminal in Nowe Sady St. were used. These included key data on temperature distribution along the length of the exhaust system (thermal imaging), exhaust gas flow versus engine load ratio (flowmeters), exhaust gas temperature changes in front of and behind the catalytic convertor versus engine load and torque (diagnostic computer from DAF engine producer). As a result of simulation model hybridization using these experimental data, a model of waste heat recovery from the exhaust fumes was constructed. Vehicle dynamics were estimated in a CBD model and implemented in the ORC model. As can be seen in Fig. 7 illustrating changes in electric power generated by the ORC system turbine (d), it is possible to estimate the parameters for individual driving cycles (a,b,c).

7 Preliminary ORC system assembly

Figures 8,9, and 10 present a preliminary ORC system assembly for the city bus. Measurements of the bus engine compartment enabled the construction of a computer aid design (CAD) model of the available space. The locations of particular elements indicated show that it is possible to implement the proposed system in the existing vehicle infrastructure. 3D models of the pump and scroll expander were available for download from the producer's webpage [20]. The producers of the plate heat exchanger (condenser), generator and buffer tank provided sufficient data to draw a simplified 3D model. The condenser is used to exchange energy between the ORC and engine cooling system. It utilizes outputs from the cooling subsystem such as in- and out-temperatures, pressure and mass flow. The coolest parts of this system have been considered for use. Only the size of the variable speed drive (VSD) [21] was based on proportions. Figure 8 shows an exhaust heat exchanger (evaporator) designed according to those fitted in Solaris buses. The required maximum pressure drop and required outlet of the system are defined. An evaporator module is used to simulate energy exchange between the heat source and ORC. It uses outputs from the source subsystem such as in- and out-temperatures, source pressure and source mass flow to measure available heat. Taking into pump values, specifically enthalpy and temperature of the ORC loop, and assuming that the liquid in the system is vaporized in this step, so setting quality at step 3 as equal to 1 (which is true for saturated vapour), the subsystem collects values for temperature, entropy, enthalpy and pressure in the evaporator. The ORC medium cooling system was omitted from the presented assembly but this system influence was included in the simulation as it was presented in Figs. 8 and 9.

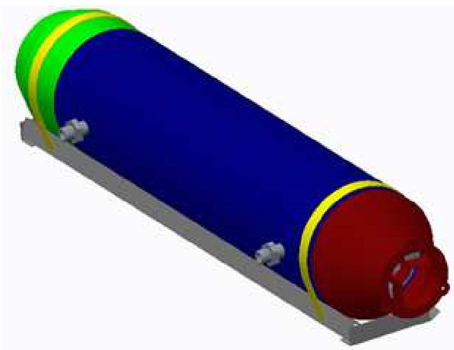

Figure 8: Exhaust heat converter – evaporator.

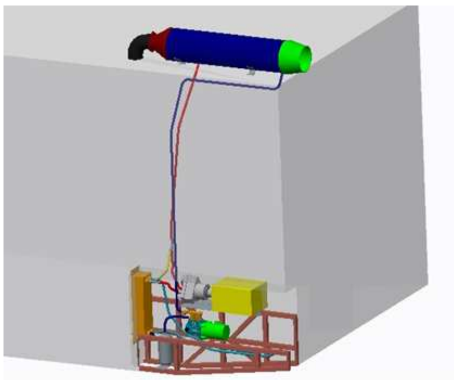

Figure 9: Model of the system mounted in the bus.

However, this assembly simulation is only valid for this particular analysis, and a new assembly design would be required for, e.g., a different engine. It therefore appears that the proposed system should have flexible options for mounting

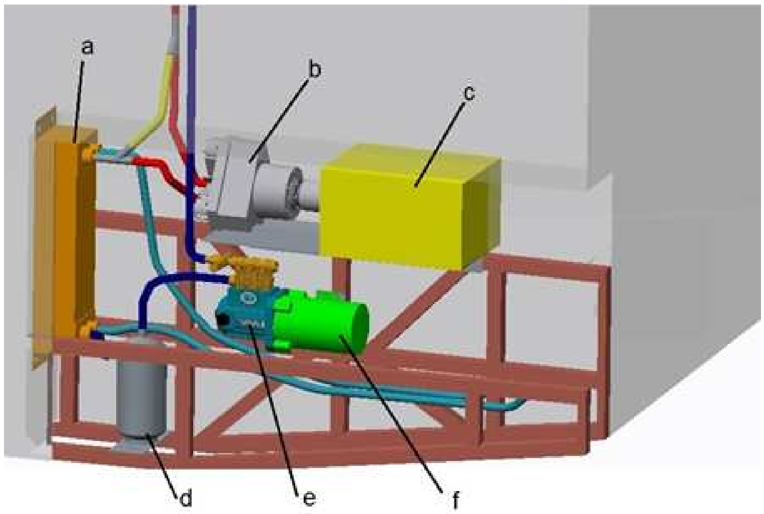

Figure 10: Close up look at the engine bay; a – plate heat exchanger-condenser, b – scroll expander, c – generator, d – buffer tank, e – pump, f – VDS.

and assembly, as do air conditioning systems. Special care should be given to the distribution of vibrations around the system and to the movement of parts. Traditional pipes could be replaced with hoses, similar to those used in air conditioning (AC) systems. Hoses are superior to pipes for this particular solution. They do not carry any surface stress, suppress vibrations and are easier to lead out to the roof. The joins between the trusses and components should also be able to dampen vibrations. Anti-vibration rubber pads could be located between the trusses and the supports, between supports and components, or in both places.

8 Economic considerations for the project

The most common static criterion of economic efficiency is payback time (simple payback period – SPBP, simple payback time, SPBT), measured in years. This is defined as the time required to recoup investment costs incurred during the implementation of a project. It is counted from the start of investment until the sum of benefits derived balance the expenditure,

$$\mathrm{SPBP} = \frac{C_i}{VAB} \,, \tag{11}$$

where: C_i – investment cost, VAB – value of annual benefits.

The cost of installing organic Rankine cycle systems on public transportation may vary, so scope was made for three costs, of 30 000, 40 000 and 50 000 PLN. The assumed the level of the ORC cost were estimated according to the industrial experience of the Institute of Turbomachinery of Lodz University of Technology employee and [15]. There was made a general assumption, that the system will be produced in short series about 50 units to cover initial design and investments costs. On the basis of the simulations discussed in this paper, an ORC system with the proposed parameters would lower fuel consumption by 5 dm^3/100 km. The annual benefit would in this case be the value of the fuel saved.

The recovered waste heat energy could be utilized to power on-board equipment, including the following systems suggested by MPK Łódź and Solaris service employees:

a) external multimedia systems such as television sets,
b) additional heating equipment. There are cold zones in the passenger compartment of the city bus. It is difficult to provide required (comfortable) temperature, e.g., in the driver's cabin and door areas,
c) additional air conditioning system (as above, especially in the driver's cabin and door areas).

Providing power from recovered waste heat could enable the number of mounted alternators to be reduced from four to three. The proposed system would allow the third alternator to be used only periodically, at times of high electric power consumption.

If follows that the simple payback period for this investment would be between 1.8 and 4.5 years. An analysis of the economic benefits of implementing ORC technology on city buses is presented in Tab. 3.

Table 3: Analysis of economic benefits of implementing ORC technology in city buses

Average mileage of city bus per day/per month		220/6800 km	
Average fuel consumption of Urbino 12/Urbino 18		37/55 dm^3/100 km	
Fuel consumption decrease diesel oil per year (U12/U18)		2500/3800 dm^3	
CO_2 emmision reduction per year (U12/U18)		7.8/9.2 × 10^3 kg CO_2	
Assumed ORC system cost	**30 000** PLN	40 000 PLN	**50 000** PLN
Amortization period Urbino 12	**2.6** years	3.5 years	4.5 years
Amortization period Urbino 18	1.8 years	2.4 years	**2.6** years
Estimated profitability of proposed system for quantities for MPK Łódź		28 x U12 and 89 x U18	
Fuel consumption decrease diesel oil per year (U12/U18)		67 000/320 000 dm^3	
CO_2 emmision reduction per year (U12/U18)		220/813 ×10^3 kg CO_2	

From the data, it is clear that investment in ORC would be economically beneficial. To emphasize its economic impact, the analysis was extended to consider not only the Solaris Urbino 18 buses in the MPK Lodz fleet but also for other buses in this class.

Perceived problems:

a) Nonstable input states for the ORC system, especially for the expander. The presented simulation results do not include process dynamics in the condenser and catalyst (engine exhaust system). Given the masses (thermal inertia) of these systems, the actual operating conditions will much better than those presented in this article.
b) Leakiness of the system due to random vibrations, especially in the operating conditions of a large city. In the author's opinion, the design should be adapted for these nonstationary operating conditions.

9 Remarks and conclusions

The tool presented in this paper facilitates analysis of a vehicle's energy consumption, taking into account real life driving patterns registered using a simple, readily available GPS device. The proposed method makes it possible to adjust the model to real life conditions, creating a hybrid model that combines the advantages of simulation with experimental research. The proposed CBD model enables the creation of 3D maps, which in turn make it possible to use the results without the time-consuming need to construct a simulation model for estimating the influence of vehicle energy intensity on the emission of waste heat from the cooling and exhaust systems of a compression ignition engine.

From the data and analysis presented in this paper, the following conclusions can be made:

- it is possible to design and implement an ORC system for city buses;
- the methodology for preliminary estimation of the main parameters of an ORC system presented in this paper seems adequate and useful for such purposes. This method includes considerable experimental feedback to make the estimated parameter more reliable;
- assuming initial installation costs of around 50 000 PLN for a three axle city bus (Solaris Urbino 18 class), the system would break even and become profitable after 2.6 years. If the initial cost were around 30 000 PLN for a two axle city bus (Solaris Urbino 12 class), it would also become profitable after a 2.6 year period;
- in the author's opinion, this work indicates a new area for multidisciplinary

research into applications of ORC products in Poland;
- the proposed project for implementing an ORC system on city buses should include service costs.

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