Abstract. Increasing the efficiency of internal combustion engines and reducing their pollution is the key issue nowadays. As a result, researchers have focused their attention on energy recovery either thermal or kinetic. In terms of thermal energy recovery, this can be done with promising results, especially as most of the potential energy of the fuel is lost as heat. Energy recovery can be done either using the Clausius-Rankine cycle or the Organic Rankine Cycle, the use of thermoelectric generators that would convert thermal energy directly into electricity, Sterling engines or even steam engines. In this paper, we propose a steam engine in which water vaporization takes place directly in the cylinder. The engine, unlike the classic locomotive engines is working in a closed cycle. The condensate is preheated and re-injected into the engine. Such an engine has the advantage of a known, cheap manufacturing technology and simple maintenance. A steam engine efficiency of only 10% of the energy dissipated on the exhaust would provide a global increase in engine efficiency by about 3%, taking into account that about 30% of the potential energy of the fuel is dissipated on the exhaust.

1. Introduction
In our days researcher try by any way to increase the internal combustion engines efficiency, to reduce the fuel consumption and pollution.

In addition to optimizing engine operation, there is an increasing interest focused on recovering thermal energy dissipated by the engine in the atmosphere. Thus, this energy can be recovered with a higher or lower efficiency using thermogenerators, Stirling engines, engines based on the Clausius-Rankine cycle or cooling cycles.

R. Zhao et al. [1] presents an energy recovery solution for turbo-charged internal combustion engines where the exhaust gas energy is recovered with an additional turbine in which the exhaust gases mixt with water vapor expand. The water vapor is formed by injecting liquid water into the exhaust stream. The study shows that the adopted solution can reduce specific consumption by 6.0-11.2% in various operating regimes when the steam injection also occurs at the inlet of the supercharger’s turbine. Gonca [2] presents the advantages of steam injection in the cylinder of an internal combustion engine with spark ignition. The results obtained from thermodynamic calculations reveal an increase in power and thermal efficiency by up to 5%. Experimental trials [3] with saturation water injection in the admission valve gate, in the case of a transformed engine to run after the Miller cycle, reveal a reduction of effective power by 6.5% and thermal efficiency by 10%, but which is offset by supercharging, power and efficiency increasing by 18 and 12%, respectively. However, it is not
specified whether this overcharging is assured by the engine or an external source. G. Kokkülünk et al [4], have made both numerical and experimental attempts [5] to see what happens in the case of steam injection in recirculated flue gas (EGR). The results highlight the drastic reduction of NOx emissions but they are accompanied by a slight increase in specific consumption.

Experimental attempts are, however, limited because a classic engine with mechanical control is used without the possibility to change parameters such as injected doses and the moment of diesel fuel or saturation water injection. The saturation water injection and injection system is presented briefly without making any reference to the components used or to its operating algorithm.

Attempts to inject water into the valve gate were also carried out by Daniel Busuttil [6]. The experimental stand used is limited in the possibility of controlling different parameters such as water injection timing, which is done using an electronic timing circuit. However, an increase in engine torque of up to 16% was observed when the air at the entrance of the engine was at ambient temperature.

Bosch, in collaboration with one of the cooperating company, presented at the Vienna Motor Symposium in 2015 [7] the water injection solution they are working on. They want to inject water directly into the engine cylinder in order to reduce both the maximum cycle temperature and the exhaust temperature thus reducing the emissions of nitrogen oxides. Besides, this approach reduces the risk of detonation without the need to reduce the timing advance, consequently increasing the efficiency of the engine at high load and speed.

Also, BMW tests the water injection solution and achieves a fuel saving of 8% [8].

The energy recovery solution by generating steam, by means of exhaust gases, which is introduced into the cylinder during engine operation is presented in the STIPE (Steam Injected Piston Engine) report. The report shows that in the case of an internal combustion engine with spark ignition, it is possible to increase the thermal efficiency from 37% to 48%. This is possible, on the one hand, by increasing the compression ratio, the detonation phenomenon being avoided precisely by steam injection, and on the other hand by reintroducing a part of the exhausted into the engine cycle [9].

In the present work, we propose another water based solution for internal combustion engine exhaust gases heat recovery. More specifically, at the presented steam engine the boiling takes place directly in the cylinder since overheated and pressurized water is injected onto the surface of a hot wall. So, a separate boiler is no longer needed. The piston steam engine becomes much safer in operation. The manufacturing technology is known being similar to that of two stroke internal combustion engines. It works on a water basis and therefore does not pose any risk to the environment.

2. Proposal and numerical approach
The proposed steam engine arrangement is presented in Figure 1. The exhaust gases from the internal combustion engine passes through the main heat exchanger 1 for steam super heating, followed by a secondary heat exchanger 2 for liquid water heating. The hot liquid water is injected with the injector 4 into the engine cylinder 3 where vaporizes and expands pushing the piston 7 from the top dead center to the bottom dead center. At the bottom dead center, the piston opens the exhaust port 10, and the steam is evacuated. The evacuated steam condenses in the condenser 5, and the obtained liquid water is pumped towards the injector with the pump 6, and the circuit is restarted. The linear movement of the piston is transformed into a circular one, with the connecting rod 8 and the crankshaft 9.

The thermodynamic analysis is done taking in account some simplifications and hypothesis based on the model depicted in Figure 2. In the cylinder, two independent subsystems are considered, liquid (l) and vapor (v), in between which exist a heat and mass transfer noted as \( Q_{lv} \) and \( m_{lv} \). It is also considered that: the vapor phase behaves like an ideal gas, the heat capacities are constant, and the volume of the liquid phase is neglected. The mixing of new-come fluid is done in the bulk and instantaneously and the processes are taking place in time but not in space.
Figure 1. Proposed steam engine for heat recovery

For estimating the performances of the engine, the liquid and steam evolution through the engine is computed considering a constant angular speed of the crank-shaft.

Figure 2. Schematic of the heat and mass fluxes through the engine

For computing the boiling temperature at different pressures the Clausius-Clapeiron equation is used:

\[ \ln \left( \frac{P_v}{P_0} \right) = -\frac{\Delta H_v}{R} \left( \frac{1}{T_v} - \frac{1}{T_0} \right) \]  

(1)
where: $p_v$ is the pressure of the vapor within the cylinder, $p_o$ is a reference temperature, $\Delta H_v$ is the latent heat of vaporization, $R$ is the gas constant and $T_o$ is the boiling temperature at pressure $p_o$.

The mass of liquid that vaporizes $m_{lv}$ is estimated based on the Schrage theory, which after mathematical approximations and simplifications can be written as:

$$
\dot{m}_{lv} = \frac{1}{R_{lv}} \left[ T_{vi} - T_{v} \cdot \left( 1 + \frac{p_v - p_l}{\Delta H_v} \right) \right]
$$

(2)

where $\dot{q}_i$ is the heat flux at the interface, $R_{lv}$ is the interface thermal resistance, $T_{vi}$ is the temperature of the vapor at the interface, $T_v$ is the vapor temperature in the bulk, $p_v$ is the vapor pressure $p_l$ is the liquid pressure and $\rho_l$ is the liquid density.

Considering the vapor temperature at the interface equal with the liquid temperature, and the vapor pressure equal with the liquid pressure, then it can be written:

$$
\dot{q}_i = m_{lv} \cdot \Delta H_v = \frac{1}{2} \cdot (T_i - T_v)
$$

(3)

The liquid-wall, liquid-gas and gas-wall heat transfer is calculated based on the convective heat transfer equation:

$$
Q_{lv} = \alpha_{lv} \cdot A \cdot (T_i - T_w)
$$

(4)

The temperature and pressure of the vapor within cylinder is computed based on the first principle of thermodynamic for open systems, the equation of state and the mass conservation:

$$
\dot{U} = Q - W + \sum \dot{m}_{in} \cdot h_{in} - \sum \dot{m}_{out} \cdot h_{out}
$$

(5)

where the internal energy is:

$$
\dot{U} = \frac{dU}{dt} = \frac{d(m \cdot h)}{dt},
$$

(6)

the equation of state:

$$
p \cdot V = m \cdot R \cdot T,
$$

(7)

and the mass conservation:

$$
m_i = m_{in} - m_{lv}
$$

(8)

$$
m_v = m_{lv} - m_{vout}.
$$

(9)

The produced work $L$ and heat inputs $Q$ are computed by integration over one cycle, one crank shaft rotation respectively.

$$
L = \int_{0}^{2\pi} p_v dV
$$

(10)

$$
Q = \int_{0}^{2\pi} Q_{lv} \cdot \omega \cdot d\alpha
$$

(11)

The thermal efficiency is:

$$
\eta = \frac{L}{Q}
$$

(12)
The exhaust flow rate is computed considering the exhaust port a convergent nozzle:

\[
m_{\text{out}} = A \cdot \rho_{\text{out}} \cdot \sqrt{2 \cdot \frac{\gamma}{\gamma-1} \cdot \frac{P_v}{P_t} \left[ 1 - \left( \frac{P_{\text{out}}}{P_t} \right)^{\frac{\gamma-1}{\gamma}} \right]}
\]

(13)

where \( \gamma \) is the adiabatic coefficient.

The water injection rate is calculated based on the following formula:

\[
m_{\text{in}} = m_0 \cdot \left(1 - e^{-\alpha \left(\alpha_{SOI} - \alpha\right)^{m+1}}\right)
\]

(14)

where \( m_0 \) is the total mass that is going to be injected, \( a \) and \( m \) are shape parameters, \( \alpha \) is the current position of the crankshaft, \( \alpha_{SOI} \) start of injection, and \( \Delta \alpha \) is the duration of the injection.

All the above equations have been implemented in a MatLab Simulink model presented in Figure 3.

![MatLab Simulink model](image)

**Figure 3.** MatLab Simulink model

### 3. Results and discussions

Based on the mentioned model it was possible to estimate the fluid evolution within the engine cylinder. The functional and geometrical characteristics used for the calculations are presented in Tables 1, 2 and 3.

The most important diagram for evaluating the engine performances is the p-V diagram presented in Figure 4. From this diagram was calculated the produced work based on the formula 10. Two important points can be identified: the start of injection (SOI), before TDC, is the point where the diagram presents an inflexion point and the pressure starts to increase faster due to the vaporization process, and the opening of the exhaust port, before BDC, where the pressure starts to decrease achieving the ambient pressure considered before the condenser.
In Figure 5, the temperature evolution of the fluids are presented. The temperature of the liquid phase is most of the time equal with the temperature of the heating wall considered constant. This is happening because it was assumed that there is a small quantity of liquid adsorbed by the wall, which will not vaporize.

The liquid temperature decreases during water injection as it can be seen in Figure 5, due to the low temperature of the injected water and to its vaporization. The temperature follows the change of pressure and increases suddenly at the end of the vaporization process. Vapour temperature decreases during injection due to freshly arrived vapours at a lower temperature. The temperature continues to decrease during vapour expansion and the vapours start to condensate when achieving the saturation
temperature. The pressure decreases more when the exhaust port opens. As it is considered that all the condensate is evacuated, only saturated vapour exist within the cylinder when the compression starts. Its temperature will increase during this stroke due to compression.

![Figure 6. Mass fluxes trough the engine](image1)

![Figure 7. Evolution of the liquid and vapour phases](image2)

In Figure 6, the mass fluxes of liquid and vapour are presented. The liquid is injected before TDC and the gas is evacuated during the opening of the exhaust port around BDC. The mass of vapour within cylinder starts to increase after injection due to vaporization as can be seen in Figure 7. After vaporization process finish, when there is no more liquid that can vaporize, the mass of vapour achieves its maximum and stays at this level till its temperature achieves the saturation temperature.
and the condensing process starts. As can be seen, the vapour mass slightly decreases during this process. The evolution of the vapour mass continues with an abrupt decrease due to the opening of the exhaust port.

The estimated efficiency of the presented engine is 19% before any other analysis and optimization. An increase of the efficiency is expected after a numerical optimization.

Table 1. Geometrical characteristics of the engine

| Parameter                  | Value | Units |
|----------------------------|-------|-------|
| Bore                       | 0.24  | m     |
| Crank radius               | 0.025 | m     |
| Connecting rod length      | 0.12  | m     |
| Compression ratio          | 15    | -     |
| Exhaust port height        | 0.01  | m     |
| Exhaust port length        | 0.04  | m     |

Table 2. Water injection characteristics

| Parameter               | Value | Units |
|-------------------------|-------|-------|
| Injected mass           | 0.006 | kg    |
| Start of injection      | 40    | °BTDC |
| Injection duration      | 30    | °     |
| Temperature             | 465   | K     |

Table 3. Functional parameters

| Parameter                          | Value | Units |
|------------------------------------|-------|-------|
| Engine speed                       | 30    | rpm   |
| Temperature of the hot wall        | 673   | K     |
| Heat transfer coefficient wall-liquid | 9000 | W/m²K |
| Heat transfer coefficient wall-vapor | 50   | W/m²K |
| Heat transfer coefficient liquid-vapor | 50   | W/m²K |
| Pressure in front of condenser     | 1     | bar   |

4. Conclusions
Within the present paper a steam engine was proposed as a possible candidate for heat recovery in case of internal combustion engine. The advantage of the presented engine consists in eliminating the boiler and all the problems related to it. The vaporization process is taking place directly in the cylinder. For its performance analysis, a mathematical model is proposed and, some numerical results are presented.

The mathematical model has been developed to analyse the operation of such an engine in different configurations and operating parameters. The final goal is to identify the optimal solution to design and execute it.

For the verification of the model and, for the determination of some heat transfer coefficients, some preliminary experiments must be carried out. So we will propose an experimental stand for the study of heat and mass transfer and, vaporization and condensation processes in enclosed enclosures.

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