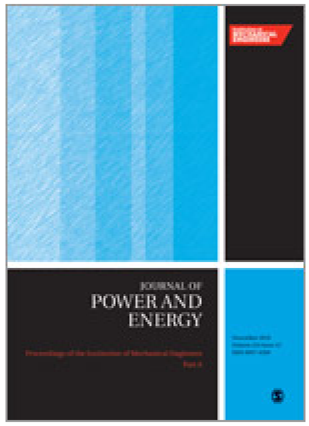

Assessment of the Newcomen Engine's development potential as heat engine for low temperature waste heat

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Assessment of the Newcomen Engine's development potential as heat engine for low temperature waste heat

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1 Introduction

Low-temperature waste heat (LTWH) with temperatures of 80 to 150°C is generated by many industrial processes such as food processing or pulp production. A study reported in [1] found a waste heat potential in Europe with a temperature below 200 °C of around 100 TWh/yr. Worldwide, waste steam emissions account for more than 40,000 GWh per year of energy loss, [2]. Low grade heat is also produced by biomass plants and geothermal sources. The most effective way to utilize this thermal energy would be a direct use for heating buildings, or as process heat. This is however in many cases not possible because there are either no buildings nearby, or there are no low temperature processes required. In addition, waste heat is produced all year whilst buildings only need heating for several months per year so that the larger part of the energy is still wasted. The power generation from LTWH resources is however difficult, since there is currently no cost-effective technology for the generation of mechanical power from waste heat with temperatures of 80 to 150 °C.

Organic Rankine Cycle (ORC) systems are today used to power generation from low temperature heat. In these systems, a working fluid with an evaporation temperature well below 100°C is employed. The fluid is evaporated, to drive a turbine or another expander to generate mechanical power. The steam is condensed, and then pumped back into the high-pressure cycle. ORC systems use refrigerants as working fluids with working pressures of 6 to 20 bar. The systems are comparatively complex. The working fluids often have a high Global Warming Potential, so that leakage and recycling become important environmental aspects. The efficiencies in the working temperature range of 80 to 110 °C range from 4.2 to 6.8 % [3].

The complexity of the ORC systems, and connected costs, have so far prevented a larger scale exploitation of LTWH. In order to make this resource available, a new low-temperature heat engine is required which is simpler than the ORC systems, and which employs a working fluid which is easier to handle.

2 The Condensing Engines

In the historic engineering literature, there are two engines described which employ the condensation of steam as the driving force, namely the Newcomen Engine (NE) and James Watt's Condensing Engine (CE). These engines are simple, and they have an operational temperature of 100 °C, so they could be employed for power generation from LTWH. In addition, they use water as working fluid and operate at atmospheric pressure so that safety requirements and complexity are significantly reduced when compared with ORC systems. Such engines may be of interest again today, and will be described in the following.

The first steam engine which employed condensation, the NE, was built in 1712. The engine uses steam at atmospheric pressure. The steam is first drawn into the cylinder by the piston, then condensed by the injection of cold water. The arising near vacuum – or rather the external atmospheric pressure – constitutes the driving force. The condensing cycle has a maximum theoretical efficiency of 6.4 %. The NE's efficiency however reached only 0.5 % [4], this is attributed to the continuous heating and cooling of the cylinder. Here it must be noted that the efficiencies given in the literature are system efficiencies which include the boiler efficiency (up to 65%), engine losses due to pressure loss and friction (73%), and the pump performance (89.1%) [5]. The first Newcomen Engine on the continent was built in 1722 in Nová Baňa / Slovakia known then as Königsberg) [6]. It will be used later as an example for analysis.

James Watt improved the NE significantly in 1765 to create the CE by adding an external condenser so that the continuous cooling and re-heating of the cylinder and subsequent losses were avoided [4]. In 1776, Watt had also described the possibility of steam expansion as a means to increase the engine's efficiency [4]. Because of the mechanical complexity of the expansion arrangement, this effect was however not employed often. With the advent of high pressure steam engines, the CEs disappeared. The last commercial CE was built by Davey, Hathorn and Co (Leeds) between approximately 1894 and 1898. Detailed measurements of this engine, a double-acting machine without steam expansion, were conducted in 1885, resulting in an engine efficiency of 3.7%, see [7] and the overview and analysis in [3].

Recently, the theory of the CE was re-assessed and improved [8]. Experiments with a bench scale engine resulted in an efficiency of 5.5 % for an expansion ratio of 1:4 [9], These efficiencies were however limited by high pressure losses in valves and comparatively high heat sink temperatures, so that there is still room for improvements. The comparative analysis of ORC and CE efficiencies [3] showed that for operating temperatures of 80 to 120 °C the ORC systems have efficiencies of 4.2 to 6.8 %. The reported CE efficiencies range from 3.7 to 5.5 %, making their performance comparable – in particular since the heat sink temperatures in the CE experiments were considerably higher than those reported for ORC tests.

For commercial applications, cost-effectiveness is of course paramount. The NE is the simplest of all condensing engines, which should be reflected in low costs. Since costs are of prime importance in renewable energy applications, it was decided to investigate the NE to assess whether it still has development potential. A thermodynamic model was developed to determine and quantify those parts of the process which contribute to the efficiency losses. The energy losses from cooling and re-heating of the metal cylinder could be minimised by using plastic for cylinder and piston. The results from the model are then used to postulate a new engine.

3 A thermodynamic model of the Newcomen Engine

3.1 The Engine

So far, and to the author's knowledge, no thermodynamic model of the NE exists. This is very probably caused by the very low reported efficiencies which imply that the NE does not have any further development potential. The low efficiency of the NE is attributed to the continuous cooling and re-heating of the metal cylinder. The use of new materials with low heat transfer coefficients such plastic as may however allow to improve the engine efficiency. The aim of this simple thermodynamic model is therefore to conceptualise the problem, and to approximately quantify its most important aspects.

3.2 Assumptions for analysis

A simple thermodynamic model was developed for the analysis for the analysis off the heat exchange process es in a NE, for which several assumptions are necessary. The heat exchanges in the cylinder take place when (a) the cold water is injected into the cylinder and the steam condenses, (b) when the water cools down a part of the cylinder and following this when (c) new steam is drawn into the cylinder, heating up the cylinder and the injected water. These interconnected thermal processes will be analysed in the model. For simplicity, the different stages are separated. The following assumptions are made:

1) The steam has a temperature of T_S = 100°C. Condensation leads to a temperature of T_{cond} = 40°C and a pressure of p_{cond} = 0.074 bar (abs,). The injection water has a temperature of T_{in} = 10°C.

2) Convection losses on the out- and inside of the cylinder are small (calculations how that they amount to less than 1 % of the thermal energy of the steam) and will be neglected. The outside temperature of the cylinder is assumed to remain constant.

3) The injected water condenses the steam, and the injected and condensed water fall back down onto the bottom of the cylinder. There is no heat flow from the top part of the cylinder into the water.

4) Only the bottom section of the cylinder, which consists of the base plate and a section of the cylinder wall of height '*d*', is in contact with the water. Here the heat exchange with the injected water and the steam takes place and only this section of the metal cylinder cools down. The cylinder volume has a final temperature of 40 °C. Because of the continuous cooling and re-heating it is assumed that the cylinder itself cools down to an average temperature of T_{av} = (100 + 40)/2 = 70°C.

5) When the piston is at the bottom position, the steam valve is opened to balance the pressure inside of the cylinder so that the water can flow out. The injected water, which has collected at the bottom of the cylinder, is assumed to be heated up to 90°C through the contact with the steam. (Farey in [4] states that "*This water is near boiling hot, or at least at its surface; for whilst it remains at the bottom of the cylinder, it will condense steam, till it acquires this temperature, and it cannot run down, till its surface will condense no more*"). The bottom part of the cylinder is in direct contact with the steam as the water exits, and again heats up from 70 to 100°C.

3.3 The cycle

The analysis focuses on the different heat exchange processes taking place during one cycle. In the idealised thermal exchanges, the water condensation, cooling and re-heating are separated in order to determine the relevant heat flows. In reality, all these processes would of course be continuous and merge into each other. The cycle is divided into six steps (1) to (6). Fig. 1 shows steps (1) to (3).

Insert Fig. 1

(a) (b) (c)

Fig. 1: The Newcomen cycle, (a) Steam inflow, (b) Water injection, (c) Condensation

Step 1: The steam valve SV is open. The piston moves from position '1' to the top position '2', drawing steam at atmospheric pressure from the boiler. The cylinder is filled with steam of volume V_{cyl}. The temperature of the brass cylinder walls and bottom is at T_b = 100 °C.

Step 2: SV is closed. Cold water of volume V_{in} at T_{in} = 10 °C is injected through the Injection Valve IV. As the steam volume V_{cyl} condenses to a water volume V_{con}, the injected water volume V_{in} absorbs the latent and sensible heat Q_s from V_s and heats up.

Step 3: The pressure in the cylinder C drops to p_{cond} = 0.074 bar (abs.), which corresponds to a condensation temperature of T_{cond} = 40 °C. The atmospheric force F acts on the piston P. All the injected water of volume V_{in} and the condensed steam of volume V_{cond} collect at the bottom to a height '*d*'.

Insert Fig, 2

(a) (b) (c)

Fig. 2: The Newcomen cycle, (d) Working stroke, (e) Steam admission, (f) Water ejection

In Fig. 2, the steps (4) to (6) of the working stroke are shown.

Step 4: The piston P moves downward through the stroke length s, performing a work $W = F \times s$. The metal of the base plate and the cylinder wall up to the height 'd', which are in contact with the water, cool down to an average temperature of T_{av} = 70 °C. This heat volume from the cylinder Q_{cyl} is also absorbed by the inflow water volume V_{in}, which now also reaches its target condensation temperature of T_{cond} = 40 °C.

Step 5: When the piston is at the bottom position '1', the steam valve SV is opened. Steam rushes in to balance the pressure inside of the cylinder C so that the water can exit. As the steam comes into contact with the water, a steam volume V_{S1} condenses and heats the water up to the exit temperature of T_{exit} = 90 °C. The cylinder bottom and the wall section of height d are assumed to remain at 70 °C.

Step 6: The ejection valve EV is opened. Further steam flows into the cylinder so that the total water volume V_{tot} can flow out, with a temperature near the boiling point at 90°C. This assumption is confirmed by observation: "*This water, though icy when injected into the cylinder and flowing out again in a moment, becomes almost boiling*" [10]. When the steam comes in contact with the cylinder bottom section of height 'd', it condenses and heats the cylinder's bottom section again up to 100 °C.

Once the water evacuation is complete, the ejection valve is closed, and the cycle can be repeated.

4 Example

4.1 The Engine

For this analysis, the dimensions and reported performance figures of a typical engine as described in [6] will be used. The engine was built in 1722 in Nová Baňa and was used to pump water from a mine shaft. The brass cylinder has a diameter of D = 845.1 mm, a wall thickness t = 21.128 mm, and a stroke length of the piston s = 2.0256 m. The cylinder volume then becomes V_{cyl} = 1.136 m^3 and the area A_{cyl} = 0.561 m^2. The density of brass is ρ_b = 8,720 kg/m^3, the specific heat of brass is c_b = 0.377 kJ/kg K.

4.2 Work and theoretical efficiency

Assuming a steam temperature T_{s0} = 100 °C, an atmospheric pressure of p_{atm} = 101.3 kPa (abs.), a condensation pressure of p_{cond} = 7.4 kPa (abs.) corresponding to a condensation temperature of T_{cond} =40 °C, and with a stroke length *s*, the work *W* conducted per stroke becomes:

$$W = A_{cyl}(p_{atm} - p_{cond})s = 0.561 \times (101.3 - 7.4) \times 2.0256 = 106.7\ kJ \tag{1}$$

The temperature of the water that is fed back into the boiler is assumed to be at 90 °C, giving a heat difference ΔT_w = 10 K. With a steam density ρ_S = 0.597 kg/m^3, a specific heat of water of c_w = 4.2 kJ/kg K and a latent heat of vaporisation of ΔH_v = 2257 kJ/kg the heat needed to produce the steam Q_s becomes

$$Q_s = \rho_s V_{cyl}(c_w \Delta T_w + \Delta H_{vap}) = 0.597 \times 1.136 \times [4.2 \times (100 - 90) + 2257] = 1560.5\ kJ \tag{2}$$

The cycle efficiency η is the ratio of work produced *W* and thermal energy Q_s required

$$\eta = \frac{106.7}{1560.5} = 0.068 \tag{3}$$

This is the theoretical maximum efficiency the engine can achieve.

4.3 The Newcomen cycle

The model developed here is not intended to simulate the time-dependent process, but to assess the principal stages and to analyse the energy losses. The cycle is separated into four different steps, it starts with the cylinder full of steam and the piston at the uppermost position '2', Fig. 1a.

1. Water injection, Fig. 1b: When the piston has reached the top position '2', cold water is injected into the cylinder. The water absorbs the latent heat plus a part of the sensible heat of the steam. The heat Q_w which needs to be absorbed by the injected water is slightly higher than Q_s, since the condensate has to be cooled down to the condensation temperature of T_{con} = 40 °C:

$$Q_w = \rho_s V_{cyl}(c_w \Delta T_w + \Delta H_v) = 0.597 \times 1.136 \times [4.2 \times (100 - 40) + 2257] = 1703.1\ kJ \tag{4}$$

2. Heat exchange with cylinder: The cold water is assumed to fall down onto the bottom of the cylinder, Fig, 1c, filling up a lower section of the cylinder of height d. This section of the cylinder will be cooled down due to the direct contact with the injected water. The mass being cooled down consists of the base plate and a ring of the cylinder of height d, Fig. 1c. The heat Q_{cyl} transferred from the cylinder to the water is a function of that part of the cylinder bottom volume V_{bot} which changes temperature, the specific heat c_b of brass, and the temperature change. Assuming that the average temperature of the cylinder after cooling down is 70 °C, the temperature difference becomes ΔT_{cyl} = 70 - 40 = 30 K.

$$V_{bot} = t\left(\frac{(D+t)^2}{4}\pi + d\pi(D+t)\right) = t\left(\frac{(0.845+0.021)^2}{4}\pi + d \times (0.845+0.021) \times \pi\right)$$

$$= 0.5890 + 2.721d \quad (5)$$

$$Q_{cyl} = c_b \Delta T_{cyl} \rho_b V_{bot} \quad (6)$$

$$Q_{cyl} = 0.377(100-70) \times 8720 \times 0.021[0.5890 + 2.721d] \quad (7)$$

The total heat flow Q_{tot} which needs to be absorbed by the injected water is the sum of the heat from the condensation process Q_w and the heat flow from the hot bottom part of the cylinder Q_{cyl}:

$$Q_{tot} = Q_w + Q_{cyl} = 1703.1 + 1228.0 + 5670.2d \quad (8)$$

The volume of injected water V_{in} required to absorb this heat and bring the temperature of the condensate to 40 °C can be calculated as follows:

$$V_{in} = \frac{Q_{tot}}{4.2\,\rho_w(40-10)} = \frac{1703.073 + 1227.951 + 5670.243d}{4.2\,\rho_w(40-10)} \quad (9)$$

3. Total volume of injection water, and determination of depth *d*: The total volume of water V_{tot} is the sum of the injected water V_{in} and the volume of the condensed steam $V_{s\,w}$, with $V_{s\,w} = V_{cyl}\,\rho_s / \rho_w$. The injected water also has to absorb the heat Q_{cyl} from the cylinder. In step (2) it is assumed that the heat exchange between cylinder and water only takes place in that section of the cylinder where the water is in direct contact with the cylinder wall, i.e. up to a height 'd' above the cylinder base. This imposes a second condition for V_w which must be fulfilled:

$$V_{tot} = V_{in} + V_{s\,w} = d\frac{0.845^2}{4}\pi \quad (10)$$

With these two equations, the depth *d* can be determined as *d* = 0.047 m and the total volume of water becomes V_{tot} = 26.564 litres.

4. Re-heating of cylinder bottom section and injected water: When the piston reaches the bottom position '1', steam is admitted, Fig. 2b, so that the pressure is balanced and the water can exit. The steam condenses and heats the water up to T_{exit} = 90°C. As the water exits, the cylinder wall and bottom come in direct contact with the steam and are heated up to 100°C, for which again a heat of Q_{cyl} is required. The total heat for one cycle Q_{tot} can then be determined as

$$Q_{tot} = Q_s + Q_{cyl} + c_w \rho_w V_{tot}(90 - 40) = 1560.5 + 1557.1 + 5534.6 = 8652.2\ kJ \qquad (11)$$

Since the steam has now replaced the water in the bottom section, the additional energy Q_{add} required to generate this volume of steam needs to be added to the thermal energy Q_{tot}.

$$Q_{add} = \rho_s V_{tot}(c_w \Delta T_w + \Delta H_v) = 0.59 \times 0.0256 \times (4.2 \times 10 + 2257) = 34.5\ kJ \qquad (12)$$

The theoretical engine efficiency then becomes

$$\eta = \frac{W}{Q} = \frac{106{,}7}{8652.2 + 34.5} = 0.012 \qquad (13)$$

5. Validation: The efficiencies for the Newcomen engine of 0.5 % given in the literature include a boiler efficiency of 65 %, the engine efficiency considering pressure changes and friction as 73 % and the pump efficiency of 89 % [5]. With these values, and the result from Eq. (13), the theoretical engine efficiency becomes η_E = 0.65 × 0.73 × 0.89 × 0.012 × 100 = 0.51%. This is quite close to the reported value of 0.5 %, indicating the validity of the model and in particular the assumption of an average temperature of the bottom section of T_{av} = 70°C.

6. Energy loss components: The last question is, how much of the energy loss can be attributed to the cylinder cooling and re-heating, and how much is caused by the re-heating of the injected water and condensation. The total energy required for one cycle is Q_{tot} = 8652.2 kJ (Eq. 11), of which Q_S = 1560.5 kJ are required for the steam. The energy loss E_L is $Q_{tot} – Q_S$ = 8652.2 – 1560.5 = 7091.7 kJ. The energy loss E_{cyl} due to re-heating of the cylinder then becomes

$$E_{cyl} = \frac{1557.1}{7091.7 + 34.5} 100 = 21.8\ \% \qquad (14)$$

The energy loss due to re-heating of the water E_w is

$$E_w = \frac{5534.6}{7091.7 + 34.5} 100 = 78.2\ \% \qquad (15)$$

The current assumption that the energy losses are predominantly caused by the cooling and re-heating of the cylinder can therefore be dismissed.

5 Analysis and discussion

5.1 Energy losses

The thermodynamic model gives an approximation of the two different factors which determine the NE's efficiency. The really interesting aspect is the realisation that the cooling and re-heating of the cylinder causes only 21.8 % of the energy loss, whilst the re-heating of the injected water and condensed steam accounts for 78.2 %, negating the assumption often stated in the literature that the cooling and reheating of the cylinder is the prime factor which determines the low efficiency. The model implies that cylinder materials with a lower heat conductivity than brass would increase the efficiency, but only to a limited extent.

5.2 The Internal Condensation Engine

Today, it is however not only the analysis of a historic technology which is of interest, but also the assessment of its potential for a possible renaissance. The previous analysis showed that a new engine based on the internal condensation as the driving force would require two different characteristics:

1. The cylinder and piston must be made of a material with a very low specific heat transfer coefficient such as heat resistant plastic.
2. The injected water must be ejected before the new steam enters the cylinder. This could be done e.g. by using the piston for the ejection. If there is only a very small gap between the piston at the lowermost position '1' and the cylinder bottom, then the piston in the downstroke can eject the water through a check valve.

The CE described in [3] and [7] had a mechanical efficiency of 3.7 %. An engine with internal condensation, and with the cylinder and piston made as precision plastic components would be simpler than the CE, whilst having a similar performance. To illustrate this, Fig, 3a shows the schematics of a CE with external condenser, Fig. 3b that of an internal condensation Engine ICE.

Insert Fig. 3

(a) (b)

Fig. 3: Functional schematics of Condensing Engine (a) and Internal Condensing Engine (b)

The reduction of complexity is clearly visible. Fig. 3b indicates that the ICE has only three main components, namely cylinder and piston, the injection water container and the heat exchanger plus three valves whilst the CE (Fig. 3a) has four main components and four valves.

Simplicity of course means reduced costs. In [6] a case is mentioned where a coalmine, which had an abundance of waste coal, built a NE in 1822 despite the fact that higher efficiency engines were already available. The reason for the decision was that whilst the fuel was more or less free, the initial investment for the NE was much lower. This confirms the potential construction cost advantage of the NE.

5.3 Efficiency improvements

The efficiency of the ICE can be further improved by introducing steam expansion. Here, the steam valve is closed before the piston reaches the uppermost position. An additional force e.g. from a flywheel or from a second cylinder then expands the steam, see e.g. [9]. The efficiency becomes a function of the expansion ratio n. For n = 1, i.e. no expansion, the theoretical efficiency is 6.4 %. This increases to 9.7 % for n = 2, and 13.9 % for n = 4. For steam expansion, a controllable steam valve SV is required. This is possible today using e.g. rotary valves.

Further improvements are possible by increasing the operating temperature to 110 °C, with a steam pressure of 1.5 bar (abs.). In many countries, steam boilers up to this pressure are not regulated and do not need certification so that the advantages are maintained. This would allow to increase the theoretical efficiency further from 13.9 % for an expansion ratio of 4 to 14.8 %.

5.4 Condensation

Both CE and ICE employ spray condensation, where the cooling fluid comes in direct contact with the gaseous medium. This is considered as the most effective condensation process. Its main disadvantage is seen in the fact that the steam comes in direct contact with the cooling fluid, which in our case however is not important since both are the same.

The CEs uses a spray condenser CO (Fig. 3a) – a tube with a volume of around 1/8th of the cylinder volume which is submerged in a cooling water basin. Cold water is drawn through a small opening into the condenser by the vacuum inside. The water jet disintegrates into a spray, condensing the steam by direct contact. The vacuum was maintained by a piston-type air pump AP (actually a vacuum pump) located beside the condenser in the water basin and as large as the condenser. It was actuated by the main driveshaft. The pump evacuated the condensate as well as residual air from the condenser, maintaining its vacuum. The complete arrangement was larger than the cylinder itself. In the last commercial CE, the spray condenser was replaced with a plate heat exchanger. The CE described in [7] has a cylinder diameter of approximately 190 mm with a stroke of 150 mm whilst the heat exchanger had a width of around 600 mm, a depth of 150 mm and a height of 1200 mm. It was the largest single component of the engine.

In the ICE, the spray condensation is driven by the head difference between the injection water container IWC (Fig. 3b) and the cylinder. Combined with the developing vacuum in the cylinder, this provides sufficient pressure to generate a spray inside of the cylinder. The condensate, injection water and residual air is then expelled by the piston. This means that neither condenser, water basin or air pump are required. The ICE therefore has a very effective condensation process where the cylinder have a double function, thereby reducing the number of components required.

5.5 Performance and comparison

A particular property of the Condensing Engines is the fact that because of the comparatively low operating pressures, large volume engines will be required. Table 1 shows the main dimensions and estimated performances (power output and efficiency) of historic engines as well as the projected values for an ICE. With the implementation of steam expansion its actual efficiency could probably be increased to 9 %.

Insert Table 1

The Westfield Engine described in [5] was one of the last NEs ever built. Still, the power output is with 23.5 kW quite small when compared with the dimensions. A modern ICE could have a higher speed and an improved efficiency. With an expansion ratio of n = 4, and with four single acting cylinders of 0.5 m diameter and 0.5 m stroke the engine would produce 19.7 kW from a thermal power of 220 kW. Clearly, efficiency and operating safety here are bought with larger dimensions.

The ICE would however be a much simpler and more effective low-temperature heat engine than ORC systems in this temperature range. Table 2 gives a comparison of ORC, CE and ICE engines for operating temperatures between 80 and 110 °C and power ratings below 1.5 kW.

Insert Table 2

The CE and the ICE with expansion have efficiencies in the range of, or better than those from ORC systems. Here it must however be mentioned that the heat sink temperature in the CE tests reported in [9] were between 60 and 65 °C, significantly higher than those for the ORC tests which ranged from 14.0 to 36.1°C [3]. The results are therefore not directly comparable. A more detailed analysis is given in [3], where the second law efficiencies, which take the heat sink temperature into account, were determined. This analysis showed clearly that the CEs have a significant efficiency advantage over ORC engines. Further advantages an ICE when compared with ORC systems would be:

- It employs water as a non-corrosive, not inflammable, non-toxic and cheap working fluid,
- It works at atmospheric pressure so that leakage problems are minimised (ORC systems use 6 to 20 bar pressure).
- Low pressure means that there are no additional safety and certification requirements as they apply for pressurized systems. This reduces initial and operating costs significantly.

6 Conclusions

A simple theoretical model for the heat exchange in a Newcomen engine was developed. The model showed good agreement with reported very low efficiency values. The analysis allowed to postulate a new engine concept which promises improved cost-effectiveness. The following conclusions could be drawn:

- The energy losses in the Newcomen engine are mainly caused noy but the heating and colling of the cylinder, but by the re-heating of the cold water from condensation and water injection.
- The re-heating of the cylinder accounts for 21.8 %, the re-heating of the injected water for 78.2 % of the energy losses.
- The resulting theoretical engine efficiency was determined as 1.2 %, similar to values reported in the literature.

The findings from the theoretical model allowed to develop the concept of the Internal Condensation Engine ICE) where energy losses are minimised through the use of plastic as main material, and through the removal of injected and condensed water from the cylinder by the piston. The ICE has several advantages when compared with other technologies such as the ORC systems:

- It has an operating temperature of 100 °C, it employs water as a non-corrosive, non-toxic and not inflammable working fluid.
- Its efficiency can be improved to 9 % using steam expansion, exceeding efficiencies of ORC systems.
- It is a very simple machine with only three main components (cylinder and piston, injection water container and heat rejection) so that manufacturing and maintenance costs should be very competitive.

Declaration of Conflicting Interests

The Authors declare that there is no conflict of interest.

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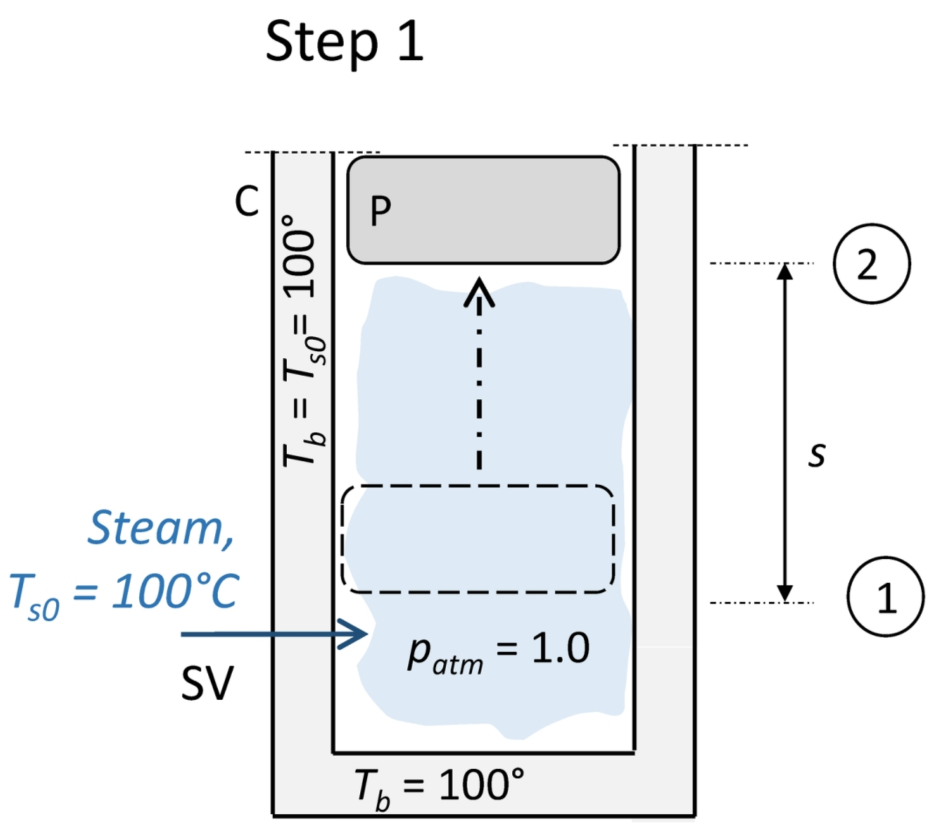

Fig. 1a

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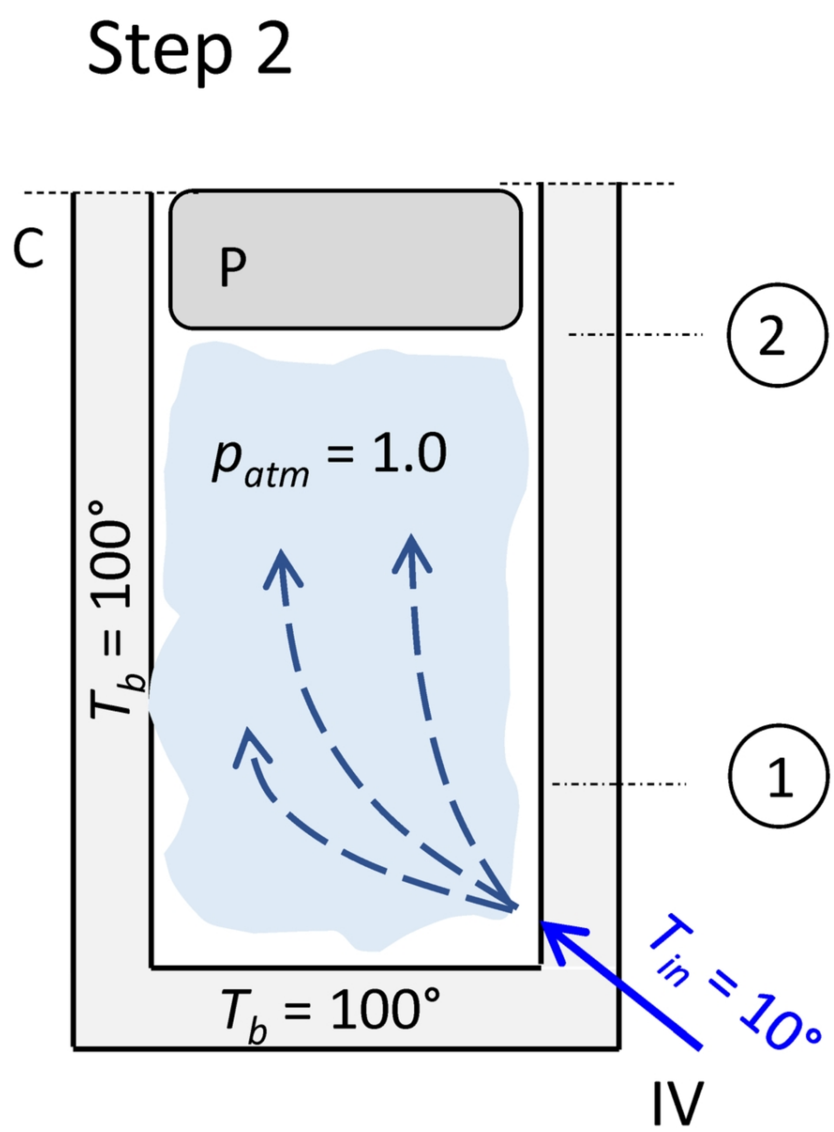

Fig. 1b

100x122mm (300 x 300 DPI)

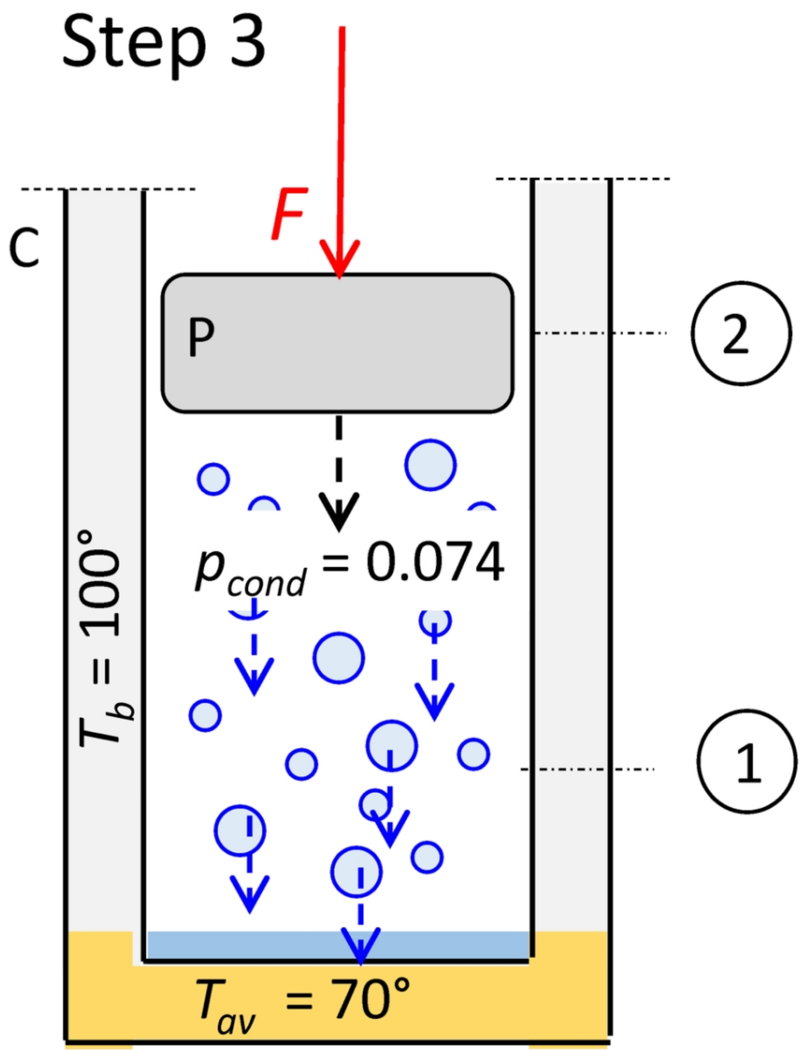

Fig. 1c

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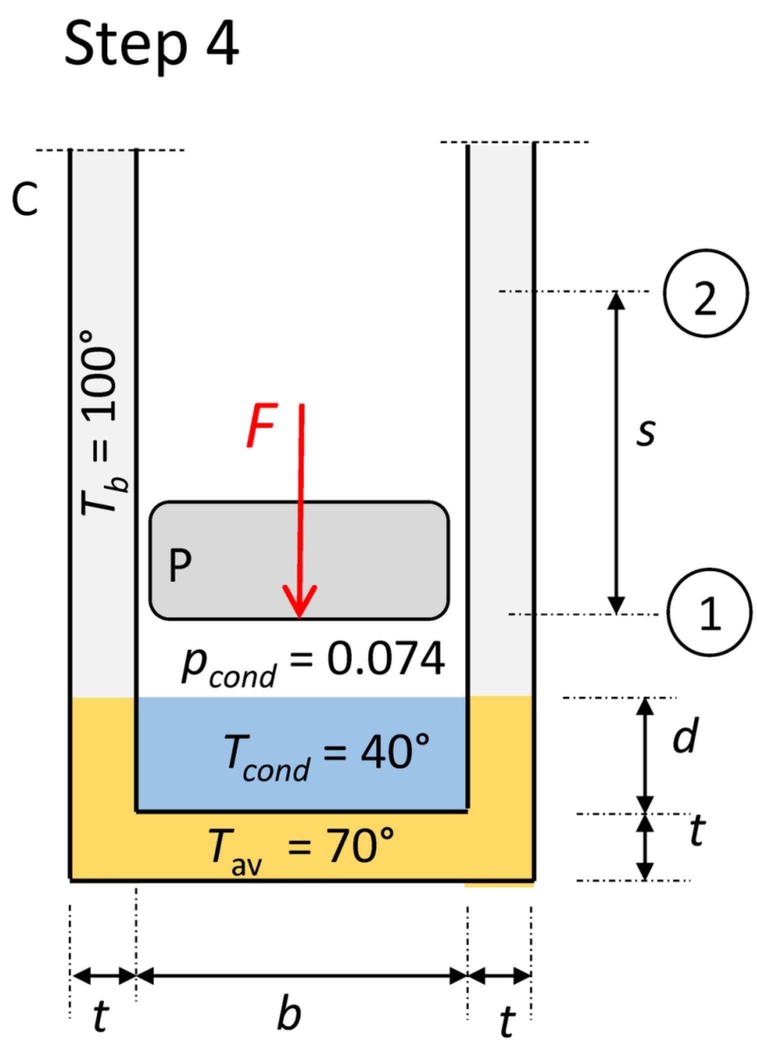

Fig. 2a

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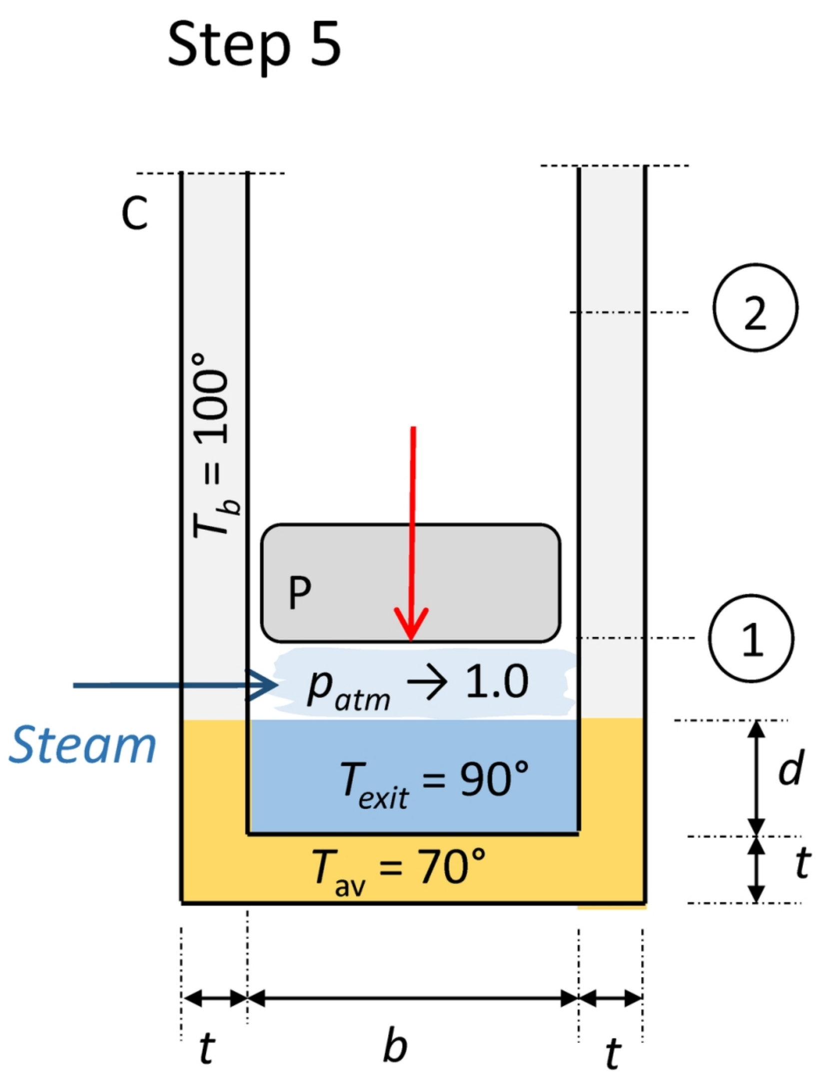

Fig. 2b

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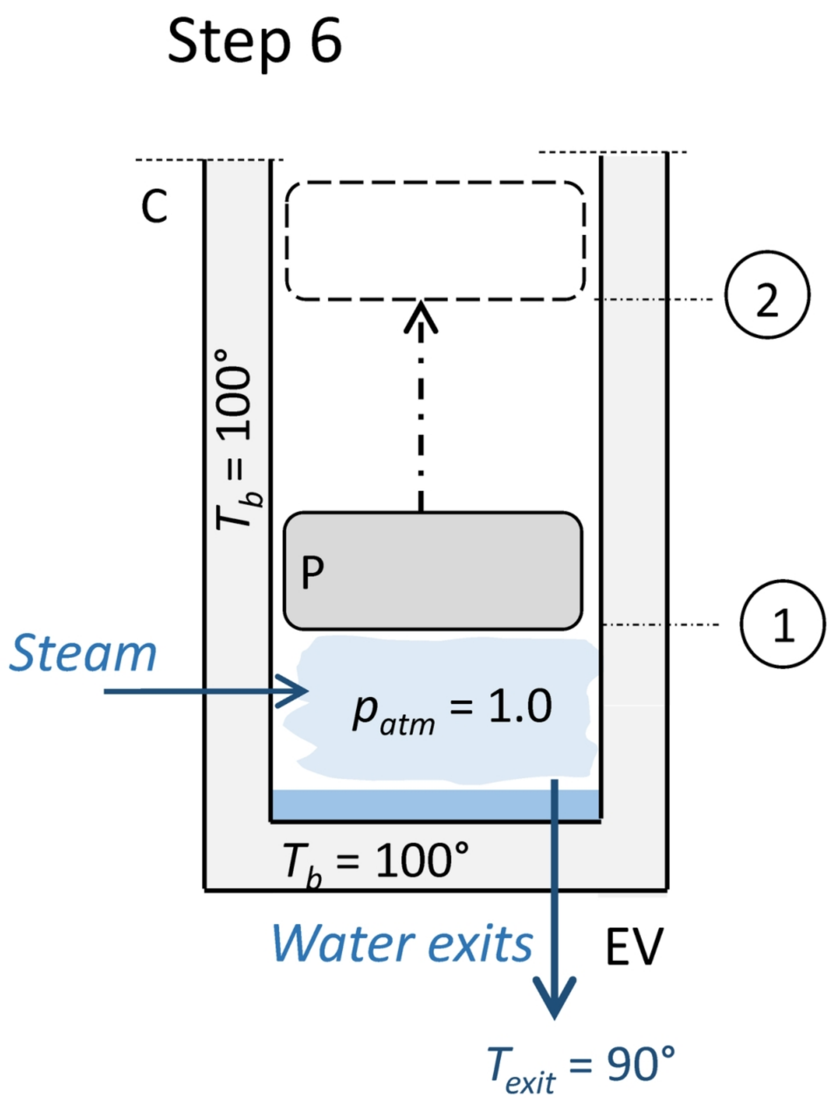

Fig. 2c

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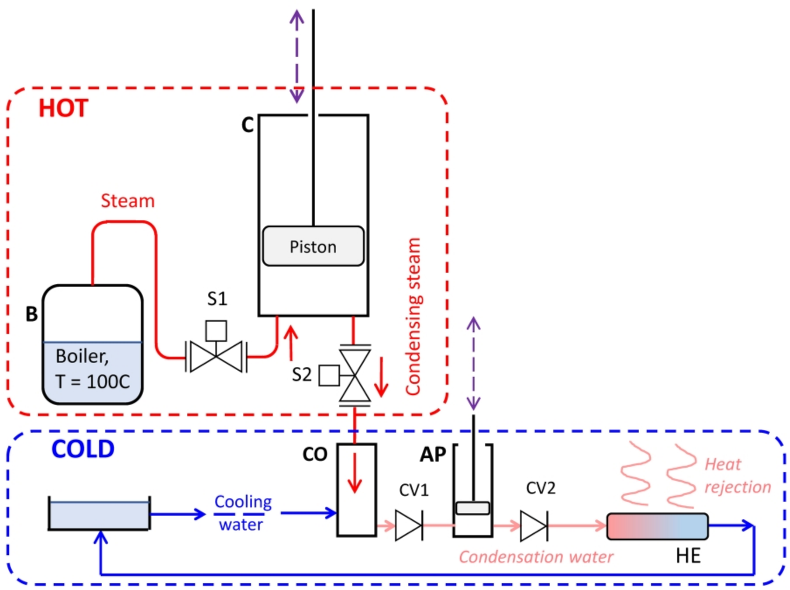

Fig. 3a

297x189mm (300 x 300 DPI)

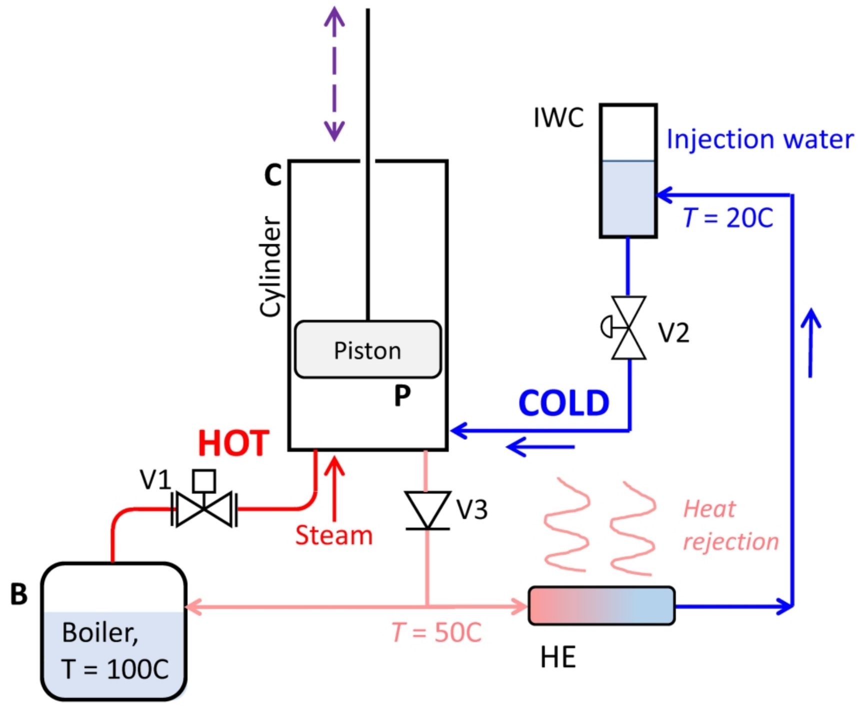

Fig. 3b

206x189mm (300 x 300 DPI)

Table 1: NE performance

Engine	Diam. (m)	Stroke (m)	f (rpm)	P (kW)	Eff. (%)
Nová Baňa [6]	0.85	2.02	12	12.5	1.2
Westfield [5]	1.37	1.73	10	23.5	1.4
ICE, *n* = 1	0.50	0.50	90	8.7	4.0*)
ICE, *n* = 4, 4 cyl.	0.50	0.50	90	19.7	9.0*)

*) Estimated value

Table 2: Comparison of ORC, CE and ICE

Type	**Theoretical max. eff. (%)**	**Measured mech. eff. (%)**	**Comments**
ORC [3]	11.0	4.2 - 6.8	0.46 – 1.43 kW
NE [5]	6.4	1.2 – 1.4	12.5 - 24 kW
CE, without exp. [3]	6.4	3.7	0.84 kW
CE, with exp. 1:4 [9]	14.8	5.5 / *8.4**)	30 W
ICE, without exp.	6.4	*3.5**) – *4.2**)	Theory
ICE, with exp. 1:4	13.9	*9* *)	Theory

*) Projected values