Article type: Original Article

Corresponding Author info

Corresponding Author:

Gerald Müller, Faculty of Engineering and the Environment, University of Southampton, Highfield, Southampton SO17 1BJ, UK. Tel.: +44 2380 592465

Email: g.muller@soton.ac.uk

The condensing engine – a heat engine for operating temperatures of 100°C and below

Gerald Müller [1]

Associate Professor, University of Southampton, Southampton, UK

Alexander Gibby[1], Chun Ho Chan[1], Muhammad Zubair Nazir[1], James Paterson[1], Joshua Seetanah[1], Matthew Telfer[1], Toru Tsuzaki[1], Caleb Walker[1] and Faris Yusof[1]

[1] University of Southampton, Southampton, UK

Abstract:

The cost-effective utilisation of low-grade thermal energy with temperatures below 150°C for electricity generation still constitutes an engineering challenge. Existing technology, e.g. the Organic Rankine Cycle machines, are complex and only economical for larger power outputs. At Southampton University, the steam condensation cycle for a working temperature of 100°C was analysed theoretically. The cycle uses water as working fluid, which has the advantages of being cheap, readily available, non-toxic, non-inflammable and non-corrosive, and works at and below atmospheric pressure, so that leakage and sealing are not problematic. Steam expansion will increase the theoretical efficiency of the cycle from 6.4% (no expansion) to 17.8% (expansion ratio 1:8). In this article, the theoretical development of the cycle is presented. A 40 Watt experimental engine was built and tested. Efficiencies ranged from 0.02 (no expansion) to 0.055 (expansion ratio 1:4). The difference between theoretical and experimental efficiencies was attributed to significant pressure loss in valves, and to difficulties with heat rejection. It was concluded that the Condensing Engine has potential for further development.

Key words: low grade thermal energy, renewable energy, condensation,

NOMENCLATURE

c_p	[J/kg·K]	specific heat of water
c_{St}	[J/kg·K]	specific heat of steam
m	[kg]	mass of steam volume
n	[-]	Expansion ratio L_0 / L_1
p_0	[Pa]	boiler pressure
p_1	[Pa]	steam pressure (adiabatic expansion)
p_2	[Pa]	condenser pressure
A_P	[m^2]	Area of piston (assumed to be equal to 1)
E_{iso}	[J]	Thermal energy required during isothermal expansion of steam
E_{th}	[J]	Thermal energy required for steam generation
F_0	[N]	Force generated by pressure p_0
H_L	[J/kg]	Latent heat of water
ΔL	[m]	Length of expansion
L_0	[m]	Length of cylinder filled with steam of pressure p_0
L_1	[m]	Length of cylinder
P_{Ap}	[W]	Power required for air pump
P_{Fr}	[W]	Friction power
P_{out}	[W]	Power out calculated from pressure difference at the piston
P_P	[W]	Power out. P_{out} - P_{Ap}
P_{sh}	[W]	Shaft power
T_0	[K]	boiler temperature
T_1	[K]	Steam temperature (adiabatic expansion)

T_2	[K]	condenser temperature
T_{ev}	[K]	evaporation temperature of water
W_0	[J]	Work conducted at pressure p_0
W_1	[J]	Expansion work during adiabatic expansion (‘ denotes isothermal)
W_2	[J]	Counteracting work generated by the condenser pressure p_2
W_{tot}	[J]	Total work generated during an adiabatic expansion stroke (‘ denotes isothermal)
γ	[-]	Adiabatic coefficient
η	[-]	Adiabatic efficiency
η'	[-]	Isothermal efficiency
ρ_S	[kg/m^3]	Density of steam

1 Introduction and literature review

1.1 Overview

Thermal energy with temperatures between 50 and 150°C is available from many areas of industry as waste heat, and from renewable energy sources such as geothermal or solar thermal energy and e.g. from biomass. There is a substantial resource of 10 to 20 TWh of low-grade waste heat in the UK with temperatures between 60 and 150°C [1]. This thermal energy corresponds to an average constant heat generation of 3.2 GW, or a theoretically recoverable potential of 326 MW, assuming the maximum realistic efficiency for an operating temperature of 100°C of 10.2% (Novikov and Curzon-Ahlborn efficiency). Waste heat recovery in the UK industry is however still low, due to the low efficiency and high costs of the available technology. The cost effective utilisation of this resource however still constitutes an engineering challenge.

1.2 Existing technology

Currently, the most advanced available technology for the utilisation of low-grade thermal energy are Organic Rankine Cycle (ORC) systems, which can employ working fluids with evaporation temperatures below 100°C. ORC engines operate with pressures of 6 to 20 bar and require an evaporator, turbine, condenser, regenerator and expander as well as pumps and heat rejection, and have to be completely sealed. Such systems are complex, expensive and usually employed for larger power ratings of more than 250 kW with temperatures > 250°C [2]. Their application to lower temperatures between 75 and 250°C is still under development. An experimental ORC with a thermal efficiency of 0.06 for system with an operating temperature of 72°C was described in [3]. Tests on an ORC installation with an operating temperature of 78.1 to 93.7°C gave energetic (waste heat to electricity) efficiencies of 0.066 to 0.076 [4]. Assuming a generator efficiency of 0.9, the actual mechanical efficiency can then be estimated to range from 0.074 to 0.084.

Initial costs for ORC systems range from €612k for a 200 kW and €9.2m for a 3 MW plant (appr. £2,600/kW) [5]. The complex technology of ORC systems means that power ratings of several hundred kW and more are required for cost-effective operation. Currently there are no cost-effective heat engines available for lower power ratings and temperatures below 150°C, despite the very considerable resource of thermal energy from industrial waste heat. There are some technologies commercially available to re-use process and waste steam. Examples are condenser / heat exchanger units, which utilize the waste steam to heat water. Others are low-pressure turbines, which require waste steam pressures of 0.5 bar [6], or steam compressors, which compress the waste steam from near atmospheric to higher pressures e.g. [7].

More exotic concepts to generate mechanical or electrical energy such as fluid piston machines, thermoelectric conversion or thermal expansion machines are still in the experimental stage.

2 The Condensing engine

2.1 Overview

The historic Condensing Engine (CE) is a simple machine, which operates with a temperature of 100°C, and at atmospheric pressure. It uses water as a cheap, non-toxic, and not inflammable working fluid and therefore avoids several of the disadvantages of ORC systems. Its main disadvantage is the comparatively low efficiency. Recent theoretical and experimental work at the University of Southampton demonstrated that the efficiencies of the atmospheric cycle can be increased substantially [8]. This opened up the possibility for further development.

2.2 Background to the project: the steam condensation cycle

Description: The cycle was developed in the late 18th Century and employs steam at atmospheric pressure to generate mechanical energy. An example is the atmospheric steam engine, invented by *Thomas Newcomen* in 1712. In this machine, steam is drawn into a cylinder. The steam is then condensed, e.g. by injection of cold water. The arising vacuum – or rather the atmospheric pressure acting from outside the system – drives the machine. *James Watt* improved the cycle by introducing an external condenser, which led to the condensing engine in 1772 [9]. The external condenser allowed the cylinder to remain hot, reducing energy losses and improving efficiency. Fig. 1a shows a typical CE with the working cylinder, condenser and air pump on the left and the drive shaft and flywheel on the right.

The CE's efficiency is limited to a theoretical maximum of 0.064, since only the atmospheric displacement work of the evaporation process can be recovered. Actual efficiencies were estimated as 0.03 [9]. In order to smoothen the power generation, *James Watt* introduced the double acting expansion engine in 1782, e.g. [10]. The last atmospheric engine to be produced appears to be the engine of Hathorn, Davey & Cie / Leeds from 1885. This was a single cylinder engine of 0.80 kW (1.09 bhp) power output at 125 rpm. The cylinder had a diameter of approximately 190 mm, with a stroke of 150mm. It was targeted for small businesses. The main advantages were simplicity, low cost, and the fact that the engine ran at atmospheric and lower pressures, so that (a) no safety issues arose, and (b) the engine could be set up practically everywhere since there was no danger of explosion. Detailed tests of the engine are reported in [11]. This information, combined with the drawings given in [12], allowed to determine the overall engine efficiency as 0.036. The engine therefore delivered 56% of the theoretically possible power. No further technical application of the condensing cycle appears to exist apart from an interesting variation of the Newcomen Engine, which was developed to power micro-electronic devices [13].

2.3 The expansion engine

James Watt realised that, due to the pressure difference at the end of the stroke, the steam still contained energy which could be employed to drive the engine. He therefore suggested to cut off the steam supply during the stroke, and let the steam expand to extract most its energy. In this way, a higher efficiency could be achieved. In practice, the resulting variability of pressure and power output was highly undesirable since e.g. pumping required a constant force output. The steam expansion was therefore limited to a factor of 1:1.3 to 1:2. Higher expansion ratios were deemed unworkable due to the pressure variation. *James Watt* and *John Southern* also introduced the pressure-volume diagram to analyse the power generated during every stroke, and to determine the efficiency, Fig. 1b. In this analysis, isothermal conditions were implicitly applied. This was however kept as a trade secret, and only published in 1822, [14]. The expansion from atmospheric to pressures below atmospheric remained a little known aspect of condensing engines, disappeared with them and is today considered of historic interest only. In the early 19th century, when the Condensing Engines disappeared, thermodynamic theory was just at its beginning. To the authors' knowledge, the sub-atmospheric expansion of the steam was therefore never analysed using 'modern' thermodynamics and assessing e.g. the effect of adiabatic and isothermal conditions.

3 Related past and current work

3.1 The expansion engine

At Southampton University, the theory of the condensing engine was improved and expanded. Adiabatic and isothermal expansion strokes were analysed [15]. Theoretical efficiencies range from $\eta = 0.064$ for an expansion ratio of $n = 1$ (i.e. no expansion) to 0.143 for $n = 4$ and 0.176 for $n = 8$ (condenser temperature $T_{cond} = 30°C$. Fundamental tests demonstrated that the theoretical efficiency is a function of the expansion ratio [8].

3.2 The condensation and expansion cycle

The condensing engine consists of a boiler, double acting cylinder, condenser and control valves, Fig. 2. The boiler temperature T_0 is assumed to be 100°C or slightly higher, so that the boiler pressure p_0 is equal to the atmospheric pressure. In the initial stage, Fig. 2 (i), the piston P is at the top position "1", with the boiler pressure p_0 acting on its top. With a piston area $A_P = 1$, the pressure results in a force $F_0 = A_P \cdot p_0$ on the piston.

The space below the piston had been filled with steam, which was condensed as the piston reached the topmost position. After condensation, the pressure p_2 in the cylinder below the piston is close to zero absolute pressure.

No expansion (n = 1): the steam pressure p_0 remains constant throughout the stroke, and acts on the piston. The pressure difference drives the piston downwards from point "1" to "3", and steam fills the upper part of the cylinder. The steam is then condensed and the cycle repeats itself in the opposite direction. A large pressure difference still exists at the end of the stroke, and a substantial part of the steam's energy remains unused.

Steam expansion: to maximize the energy production, it is necessary to let the steam expand, Fig. 2 (i). The piston initially travels a distance L_0, with steam filling the upper cylinder. At point "2", Fig. 2(ii)), the boiler valve is closed. The steam then expands, driving the piston against the vacuum through a distance ΔL from point "2" to point "3". The pressure drops from p_0 at "2" to $p_1 > p_2$ at "3", Fig. 2(iii). Once the piston has arrived at "3" with a pressure p_1, the condenser valve is opened, the expanded steam rushes into the condenser, a vacuum forms and the cycle is repeated in the opposite direction, Fig. 2 (iv).

3.3 Thermal energy required:

The initial thermal energy E_{th} required to generate the steam can be calculated using the tabulated values for density of steam ρ_S, specific heat of water c_p. and the latent heat of evaporation H_L. With a temperature difference dT, a piston area of $A_P = 1$, a mass $m = \rho_s \cdot A_P \cdot L_0$ and a thermal capacity of the water of c_p =4.2 kJ/kg·K this energy becomes

$$E_{th} = m \cdot \left(dT \cdot c_p + H_L\right) \tag{1}$$

With a density of steam of ρ_S = 0.59 kg / m^3, and an initial water temperature of 40°C (condenser temperature), the thermal energy required to heat the water up to the evaporation temperature, and to evaporate it amounts to 1480.31 kJ/m^3 (steam) for T_0 = 100°C and p_0 = 1 bar.

3.4 Adiabatic cycle

The initial work W_0 conducted by the steam as the piston moves from position "1" to "2" under constant pressure with a piston area of A_p = 1 is

$$W_0 = p_0 \cdot L_0 \tag{2}$$

At point "2", the boiler valve is closed and the piston moves from position "2" to "3", driven by the expanding steam. Assuming an adiabatic expansion with an adiabatic coefficient of γ = 1.08 for wet steam [12], the end pressure p_1 becomes:

$$p_1 = p_0 \left(\frac{L_0}{L_1}\right)^{1/\gamma} \tag{3}$$

During the expansion from L_0 to $L_1 = n \cdot L_0$, the pressure of the steam drops from p_0 to p_1 and the steam conducts the expansion work W_1:

$$W_1 = \frac{1}{\gamma - 1}(p_0 \cdot L_o - p_1 \cdot L_1) \tag{4}$$

In order to determine the actual work, the counteracting pressure p_2 needs to be considered. This creates a work $W_2 = p_2 \cdot L_1$. The total work W_{tot} conducted during a stroke of length L_1 then becomes:

$$W_{tot} = W_0 + W_1 - W_2 = p_0 \cdot (L_1 - L_0) + \frac{1}{\gamma - 1}(p_0 \cdot L_o - p_1 \cdot L_1) - L_1 \cdot p_2 \tag{5}$$

After expansion, the condenser valve at the top of the cylinder is opened, the steam condenses with a pressure p_2 = 7.4 kPa, corresponding to the condensation temperature of 40°C, and the cycle repeats itself in the opposite direction. With the thermal energy E_{th} required to produce the steam known from Eq. (1), the efficiency η can be calculated:

$$\eta = \frac{W_{tot}}{E_{th}} \tag{6}$$

The adiabatic coefficient γ is assumed as γ = 1.08 [16].

3.5 Isothermal expansion

Isothermal expansion implies, that the cylinder temperature is constant throughout the expansion, so that additional thermal energy has to be supplied in order to maintain the temperature. From the adiabatic expansion, the end temperature T_1 of the expansion cycle can be determined with Eq. (2). The mass of the steam m, and the difference between operating and end temperature $\Delta T = T_0 - T_1$ with the specific heat for steam c_{St} then allows to calculate the thermal energy E_{iso} required to maintain steam temperature as a function of the initial steam volume L_0, and the expanded volume L_1.

$$E_{iso} = m \cdot c_{St} \cdot (T_0 - T_1) \tag{7}$$

For a cross sectional area of $A_P = 1$, the work conducted by the steam at constant pressure W_0, and during the expansion W_1' are determined as follows:

Constant pressure:

$$W_0 = p_0 \cdot L_0 \tag{8}$$

Expansion:

$$p_1' = p_0 \left(\frac{L_0}{L_1}\right) \tag{9}$$

$$W_1' = p_0 \cdot L_0 \cdot ln \frac{p_0}{p_1'} \tag{10}$$

The total work W_{tot}' can be calculated, whereby the counteracting pressure p_2 in the lower section of the cylinder needs to be considered:

$$W_{tot}' = W_0 \ + W_1' - W_2 \tag{11}$$

The efficiency η' then becomes the ratio of external work produced W_{tot}, and the total thermal energy required E_{th} $+E_{iso}$.

$$\eta' = \frac{W_{tot}'}{E_{th}+E_{iso}} \tag{12}$$

Fig. 3a shows the theoretical efficiency for adiabatic and isothermal expansion for the classic expansion engine with an operational temperature of 100°C and a condenser temperature of 40°C. The theoretical efficiency increases from η = 0.063 for an expansion ratio of n = 1 (i.e. no expansion), to 0.16 for an expansion ration of n = 8.

3.6 Condensation

For the condensation, a simple model was employed assuming direct heat transfer from the steam to the heat exchanger. The latent heat, and the sensible heat for the temperature difference between steam temperature and condenser temperature was absorbed by the increase in temperature of the cooling fluid. For the theoretical analysis, the condensation temperature was assumed as 40°C throughout.

4 Model tests

4.1 Overview

A model engine of approximately 1:5 scale with a nominal power output of 40 Watt was designed, built and tested at Southampton University within a 4th year student project. The engine lay-out was based on a historic engine from 1782 shown in Farey (1827), Fig. 1a. This offered several advantages:

1. The relative dimensions of cylinder and components can be based on the historic engine.

2. The engine is large enough for all components to be easily accessible.

4.2 System

Fig. 3b shows the engine system. Steam is produced in the boiler **B**, and admitted into the cylinder **C** through the solenoid valves S1 and S2. Steam in- and outflow pipes enter /exit the cylinder **C** through the top and bottom cap. This reduces the dead volume in the cylinder. The condenser **Co** is connected with the cylinder **C** through S3 and S4. Once one half of the cylinder is filled with steam and the piston has reached its upper- or lower most point, the respective valve is opened. The steam rushes into **Co** and condenses, maintaining the vacuum. The air pump **AP** is connected with the beam. It maintains a vacuum in the condenser, and evacuates condensed water and residual air from the steam out of the condenser **Co**. Two check valves connect the condenser with the air pump, and the air pump with the outside world.

4.3 The model engine

The engine had overall dimensions of 900 × 600 mm, with a height of 800 mm. The main components are shown in Fig. 4a, Fig. 4b shows the real engine:

1. Cylinder (a): diameter D = 50 mm, stroke length l = 120 mm. Material: brass
2. Condenser (b), diameter D_c = 21 mm, height L_c = 50 mm. Material: brass
3. Air pump (c), diameter D_{ap} = 25 mm, height L_{ap} = 50 mm. Material: brass

4. The parallelogram (d) serves to compensate for the radial motion of the beam, to keep the piston rod movement vertical.
5. The beam (e) of length 500 mm connects piston rod and crank shaft (f).
6. A 1:3 transmission (g) connects shaft and flywheel (h), which has a diameter of 320 mm and a mass of 3.2 kg.
7. The control hardware is located in a small plastic box (i).
8. The power take-off (j) consists of a torque transducer connected to a 50 Watt permanent magnet motor with a 1:50 gearbox. The energy is dissipated with a 100 Ohm potentiometer.
9. Four Solenoid Valves with 3.5 mm inner diameter to regulate steam in- and outflow.

The cylinder and the valves were insulated with 5mm neoprene to minimize thermal losses. The boiler was a commercial, 1.2 kW steam generator (wallpaper stripper).

4.4 Data acquisition and engine control:

The measurement system was designed to monitor the main engine parameters and consisted of

1. Four RS 797-4970 pressure transducers with a range of -1 to 9 bar at an accuracy of 0.25% and an operating temperature range of -200 to +135 °C to monitor pressures in the cylinder, the boiler and the condenser.
2. Torque transducer AEP R2A 25 Nm with a hysteresis of 0.2% and a maximum rotational speed of 4000 rpm to measure the torque at the crankshaft.
3. An AEAT-601B rotary magnetic encoder with a resolution of 256 pulses per revolution (quadrature, i.e. 1,024 counts per revolution) and a maximum speed of 12,000 rpm connected to the drive shaft to provide control signals.
4. A National Instrument LabView V2016 based combined data acquisition and control system design virtual instruments specifically custom designed for this application. The data acquisition VI monitored pressures and torque, and controlled the valve operation using the encoder signals to determine the piston position. Four solenoid valves controlled steam access to the cylinder from the top and bottom (S1 and S2), and the steam exit into the condenser (S3 and S4). The control software allowed to close the steam inflow during the cylinder stroke, to allow for steam expansion with expansion ratios of n = 1 to n = 4.

4.5 The condenser / heat exchanger

The condenser is a vital component of the system. It controls the heat sink temperature and the engine efficiency limits. The temperature differences between heat source and sink are very limited, and even small increases in condenser temperature can have significant effects on the engine efficiency. Three different condenser types were evaluated to determine an optimum configuration:

1. A simple 25 mm diameter, 50 mm high brass tube condenser, with spray entry through a 0.5 mm hole in the condenser side wall.
2. A tube condenser with cooling fins, Fig. 5a.
3. A heat exchanger made up of a 3 mm inner diameter copper pipe coil of 45 mm height (8 turns, "hot coil") and a "cold coil" of 6 mm internal diameter copper pie with a height of 200 mm and 20 turns, Fig. 5b. The coils were set inside an 88.9 mm diameter, 300 mm long brass tube. Cooling water was pumped through the pipe coils so that the steam condensed inside the brass tube. The heat exchanger was designed not only to condense the steam, but also to recover part of the thermal energy at a higher temperature level for further use. The "hot coil / cold coil concept" was developed for this purpose. The expected effect did however only occur at a temperature level well below expectations.

Condensers (1) and (2) were located inside a cooling water basin, condenser (3) in the open air. The initial tests showed that condensers (1) and (2) were inferior to the heat exchanger, so that only the results from C3 will be presented here.

5 Test results

5.1 Overview

The tests were conducted in the University of Southampton's Hydraulics Laboratory. Expansion ratios of n = 1 (no expansion) to n = 4 were investigated. In all, 14 tests runs were conducted whereby in five runs the expansion ratios were varied, in nine tests they were kept constant at n = 1. The engine speed ranged from 25 to 126 rpm, and depended on the expansion ratio as well as the test duration.

5.2 Engine performance

The pressures in boiler, cylinder and condenser give a good indication about the performance of the engine. Figures 6a to c shows typical results for cylinder pressures for expansion ratios of n = 1, 2 and 4. The condenser pressures were a function of the expansion ratio and reached 0.342 bar (n = 1), 0.261 for n = 2 and 0.13 bar for n = 4, whilst the boiler pressure remained constant. From the pressure measurements, it can be seen that:

1. During steam admission, the top cylinder pressures remained between 0.15 bar (n = 1) to 0.2 and 0.25 bar (n = 2 and 4) below the boiler pressure.
2. The bottom cylinder pressures during steam admission remained between 0.2 bar (n = 1) to 0.3 and 0.55 bar (n = 2 and 4) below the boiler pressure.
3. The pressure drop during condensation does not occur instantaneously.
4. The lowest pressure in the bottom cylinder nearly reaches the condenser pressure.
5. The lowest pressure in the top cylinder remains approximately 0.1 bar above the condenser pressure (n = 2 and 4).
6. For n = 4, the pressure difference in the upstroke is nearly zero, so that no work is produced in the upstroke.

5.3 Analysis

The input energy was determined with the thermal energy E_{cyl} required to fill the cylinder volume with steam. The steam volume depended on the cylinder volume V_{cyl}, and the engine speed f (rpm). With $m = 2 \cdot V_{cyl} \cdot \rho_{steam} \cdot f/60$

$$E_{cyl} = m \cdot \left(dT \cdot c_p + H_L\right) \tag{13}$$

The output energy P_{out} was then calculated for two different conditions:

1. Piston power: the power generate at the piston P_P was determined from the pressure difference acting on the piston, and the piston displacement for every time step. The power required to drive the air pump P_{ap} was determined from the pressure difference between up- and down stroke and subtracted from the piston power.

The efficiency η was then determined as

$$\eta = \frac{P_{out} - P_{ap}}{E_{cyl}} \tag{14}$$

2. Shaft power: the shaft output power P_{sh} was determined from the engine speed and the torque measured at the crank shaft.

3. Friction: The first trials indicated that significant friction occurred in the system. This was attributed to the bearings, the friction in the piston rod seal and friction in the crank shaft. The friction was quantified by running the hot engine with all valves open using an electric motor fixed to the torque transducer.

Figure 7a shows that there is a significant friction, which is an approximately linear function of the engine speed. The friction power P_{Fr} was then determined as a function of the engine speed f (rpm):

$$P_{Fr} = (f - 40)\frac{10}{(100-40)} \tag{15}$$

In Fig. 7b the calculated piston power $P_P = P_{out} - P_{Ap}$ is plotted against the sum of shaft and friction power. It can be seen that there is reasonable agreement between the two, indicating that the real power output of the engine can be determined in this way.

5.4 Power out

Fig. 8a shows the power output for all expansion ratios. The piston power ranged from 3.64 W (n = 4) to 26.7 W (n = 2). Speed and power output reduce with increasing expansion ratio. The shaft power output, Fig. 8b, ranged from 0.4 to 12 W, with overall efficiencies of 0.02 to 0.058.

5.5 Efficiency

Fig. 9 shows the efficiencies (determined with the piston power and the cylinder steam demand) as a function of the expansion ratios. The highest efficiencies were a function of the expansion ratios and ranged from 0.020 (n = 1) to 0.040 (n = 2) and 0.055 (n = 4). It also shows the theoretical maximum efficiency as the upper limit for the engine. The tests were conducted on five different days. The efficiency increases with increasing expansion ratio. Efficiencies reach around 30 to 35% of the theoretical maximum value.

6 Sub-atmospheric expansion

In some long duration test runs, the steam production dropped below the steam demand, and the boiler pressure fell below atmospheric. The engine still produced power. This scenario had been predicted theoretically [15]. Since operation at sub-atmospheric pressure is (a) novel and (b) potentially interesting for low temperature applications it will be described here in more detail.

The engine shown in Fig. 2 is a closed system. The evaporation temperature depends on the internal system pressure p_0. If p_0 drops below atmospheric, evaporation occurs at a lower temperature. As long as boiler temperature and boiler pressure remain higher than the condenser pressure / temperature, the system can still produce power. For an expansion ratio of n = 1, the theoretical efficiency reduces from 0.063 (n = 3: 0.124) for T_0 = 100°C, p_0 = 1 bar to 0.041 (n = 3: 0.069) for T_θ = 60°C, p_0 = 0.2 bar, assuming a condenser pressure p_2 = 0.07 bar / condenser temperature of

40°C, Fig. 10a. This means that operational temperatures below 100°C with water as working fluid are possible. Steam expansion is also possible, and increases the efficiency. In the adiabatic case (Fig. 10a), the condensation temperature is reached during the expansion process as indicated by the dashed line. The Carnot efficiency is close to the condensing cycle efficiencies, indicating the effectiveness of the cycle.

Two tests were conducted with runs of up to 800 seconds and an expansion ratio of n =1 to assess the effect of continuous operation. In these tests, the boiler pressure started at atmospheric, and then reduced continuously, down to -0.254 bar or 0.746 bar (abs.), whilst the engine still produced power. Fig. 10b shows the pressures in boiler, top and bottom cylinder and in the condenser. The net power production at 0.746 bar (abs.) boiler pressure with an expansion ratio of n = 1 (no expansion) was 5.78 W, with an efficiency of 0.011. Again, there is a significant difference between boiler and cylinder pressures during steam inflow, and a smaller difference between cylinder and condenser during condensation indicating high flow resistance in the valves.

7 Potential real applications

The CE technology is currently at the beginning of its development. There are however several areas where a direct implementation of the technology appears possible.

7.1 Waste steam:

The most promising area is probably the direct use of waste steam, which is often released at atmospheric pressure directly into the atmosphere. A tyre manufacturing plant emits 70,000 tons of steam per year [18]. This steam could be directly fed into a CE. Assuming an overall efficiency (steam to wire) of 0.09, this would result in a continuous electricity production of 300 kW. The total yearly production would be 2.65 GWh with a value of £332,000.

7.2 Low Temperature District Central Heating (LTDCH):

Usually, district central heating (DCH) systems are run with temperatures of 70 to 120°C. LTDCH uses operating temperatures of 50 to 55°C, thereby reducing energy losses and investment for pipelines considerably. The heat source for DCH systems is usually a combined heat and power plant. The CE could be employed in LTCDH systems as heat exchanger, reducing the system temperature from the input temperature (here the lower temperature range of 120 to 200°C would be of particular interest) to the LTDCH requirements. With condenser temperatures of 50 to 55° C, the condenser pressure would be 0.13 to 0.16 bar, reducing the CE's efficiency from 0.063 (n = 1) and 0.16 (n = 8) for

T_2 = 40°C to 0.06 and 0.13 respectively for T_2 = 50°C. In [19] an example of a LTDCH system is described, which is fed from a 700 kW_{th} CHP and a 200 kW_{th} geothermal plant. Assuming an initial temperature of the heat source of 200°C, and a CE with an efficiency of 0.08, the heat exchange process could generate 48 kW of electric power.

7.3 Low temperature solar thermal combined desalination and power systems

Flat plate solar thermal collectors are the cheapest solar thermal converters and can reach efficiencies of 50% for operating temperatures of 120°C. This allows for the evaporation of seawater. The CE does not require lubrication, so that it can not only produce energy from the steam, but also distilled water as a by-product. Such systems do not exist as yet, but with the further development of the CE all major the components will be available.

8 Discussion

The Condensing Engine as a heat engine for low-grade thermal energy with temperatures of up to 150°C offers interesting advantages when compared with other available technology such as ORC systems:

1. It is a simple engine, which implies cost-effectiveness.
2. It uses water as working fluid, which is non-corrosive, non-toxic, easily available and cheap.
3. It operates at pressure near and below atmospheric, so that there is no danger of a boiler explosion and safety regulations are much less stringent that for pressurized systems, e.g. [20]. Leakages are also less problematic.

The Condensing Engine appears particularly suited for applications with lower power ranges of 2 to 200 kW. Experiments with a CE with steam expansion were conducted in order to prove the concept, to obtain information about its performance, and to tests and compare components.

For the efficiency calculations, the steam demand of the cylinder displacement was employed to determine the thermal power input. This will result in over-optimistic results, since e.g. thermal losses are neglected. The results from tests on a full scale engine given in [11] however allow to compare cylinder and total engine efficiency. The efficiency determined from the cylinder displacement volume was 0.052, whilst the overall efficiency determined from the weight of the evaporated water was 0.036. Using the theoretical steam demand to determine the efficiency therefore overestimates the efficiency by around 30%.

The efficiencies in the model tests reached 30 to 40% of the theoretical maximum, and several causes for this could be identified, in particular friction and pressure losses in the solenoid valves. The operation with boiler pressures below atmospheric / boiler temperatures below 100°C was demonstrated, also confirming earlier theoretical predictions. The efficiencies were calculated as thermal efficiencies with the cylinder steam demand so that other thermal losses were neglected.

The adverse effects observed in the experiments point out the requirements for further development:

1. Internal friction: the losses due to internal friction accounted for approximately 60% of the total power production. This was thought to be caused by unsuitable bearings, friction at the piston rod seal, and possibly friction in the parallelogram. The very high internal friction constitutes not a conceptual problem but a problem of components and detailing, a brief calculation indicated that it should be possible to reduce the internal friction by 90%.
2. The pressure drop from boiler to cylinder, and from cylinder to condenser indicated that the solenoid valves were not suitable for this application. The valves should have a minimum flow resistance, and should made from plastic to reduce thermal losses.
3. There was a problem with condensation blocking steam inflow into the bottom cylinder, and steam exit from the top cylinder. The access points at the top / bottom caps of the cylinder do not allow for gravity outflow of condensate at these two points. Historic drawings show that gravity outflow was ensured for all exit points.
4. The condenser pressures ranged from 0.346 to 0.13 bar for n = 1 to n = 4, the expected pressures below 0.1 bar could not be achieved. The fact that the condenser pressure dropped for increasing expansion ratios / reduced steam volume shows that the principle in itself works, but that heat rejection is incomplete, and that the heat transfer from condenser to the environment must be improved.

The experiments highlighted the importance of the heat sink. The small temperature differences mean, that even a 10 degree increase in heat sink temperature has a strong influence on the engine performance. In the same time, the heat volumes which have to be exchanged are substantial. The available temperature difference is probably limited by the ambient temperature. Therefore, a heat sink temperature of 40°C was assumed throughout in order to obtain realistic results. Condenser temperatures of only to 15 - 30°C, as described e.g. in [4], appear impractical.

The further development will have to go towards a horizontal cylinder arrangement with a crosshead guide. This will allow for gravity driven exit flow from the cylinder, and will also reduce space requirements. Multi-cylinder arrangements would be very beneficial from the point of view of power quality for higher expansion ratios. The use of plastic for main components such as cylinder and piston and valves is possible and would reduce unit costs for larger production runs significantly.

8 Conclusions

The theory of the condensing engine with steam expansion was developed further and expanded. A 40 Watt model engine was designed, built and tested. The following conclusions were drawn from the research programme:

- The Condensing Engine with steam expansion has theoretical maximum efficiencies of 0.063 (no expansion) to 0.16 (expansion ratio 1:8) for an operating temperature of 100°C and a heat sink temperature of 40°C. This compares well with existing ORC technology.
- In the model tests, efficiencies ranged from 0.020 (no expansion) to 0.055 (expansion ratio 1:4) for heat sink temperatures of 55 to 70°C.
- Losses were caused by pressure drops at the valves, and significant inner friction and amounted to approximately 60% of the total power production.
- A heat exchanger condenser was found to perform best as heat sink.
- Operation with power production was demonstrated for boiler pressures down to -0.25 bar / boiler temperature of 90°C. This confirmed earlier theoretical predictions.

Theory and experiments demonstrated the validity, and the development potential of the condensing engine as heat engine for an operating temperature of 100°C.

REFERENCES

[1] McKenna, R. C. and Norman, J. B. Spatial modelling of industrial heat loads and recovery potentials in the UK. *Energy Policy* 2010; 38(10): 5878-5891.

[2] Tchanche, B. F., Lambrinos, G., Frangoudakis, A. and Papadakis, G. Low-grade heat conversion into power using organic Rankine cycles–A review of various applications. *Renewable and Sustainable Energy Reviews* 2010; 15(8): 3963-3979.

[3] Kosmadakis, G., Manolakos, D. and Papadakis, G. Experimental investigation of a low-temperature organic Rankine cycle (ORC) engine under variable heat input. *Appl. Thermal Engineering,* 2016; 92: 1-7.

[4] Minea, V. Power generation with ORC machines using low-grade waste heat or renewable energy. *Applied Thermal Engineering* 2010; 69 (1): 143-154.

[5] Quoilin, S., Van Den Broek, M., Declaye, S., Dewallef, P. and Lemort, V. Techno-economic survey of Organic Rankine Cycle (ORC) systems. *Renewable and Sustainable Energy Reviews* 2013; 22: 168-186.

[6] Enva *Energiemodul.* .http://www.envaenergy.de/energiemodul/ (2016, accessed 30.09.1016)

[7] TLV. 2016. *Steam compressor*. http://www.tlv.com/global/DE/products/210000.html (2016, accessed 30.09.2016)

[8] Müller, G. and Parker, G. Experimental investigation of the atmospheric steam engine with forced expansion. *Renewable Energy* 2015; 75: 348-355.

[9] Dickinson H.W. *A short history of the steam engine*. Cambridge, Cambridge University Press, 1938.

[10] Farey, J. *A Treatise on the Steam Engine Historical, Practical, and Descriptive*. Longman, Rees, Orme, Brown, and Green. By John Farey. Illustrated by Numerous Engravings and Diagrams, 1827. Available at: https://archive.org/details/treatiseonsteame01fareuoft (accessed 25.10.2016)

[11] Revue Industrielle. *Moteur domestique, construit par M Albaret. Paris*, 1885; 10 (3): 93-94. Available at: (http://gallica.bnf.fr/ark:/12148/bpt6k97674548, accessed 26.06.2017).

[12] Dingler (Ed.), Ueber Neuerungen an Kleindampfmaschinen. *Polytechnisches Journal*, Leipzig. 1886; 259: 1–9. Available at: http://dingler.culture.hu-berlin.de/article/pj259/ar259001

[13] Wang, Y., Zhou, Z., Zhou, J., Liu, J., Wang, Z., and Cen, K. Micro Newcomen steam engine using two-phase working fluid. *Energy*, 2011; 36(2): 917-921.

[14] Anonymous. Account of a steam-engine indicator. *Quarterly Journal of Science*, 1822; 13: 95.

[15] Müller G. The atmospheric steam engine as energy converter for low and medium temperature thermal energy. *Renewable Energy*, 2013; 53: 94-100.

[16] Dietzel F. and Wagner W. *Technische Wärmelehre*. 9th Ed. Vogel Verlag, Würzburg, 2006.

[17] Müller G. The condensing cycle for heat engines with operating temperatures of 50 to 100°C. *Proc. HEFAT 2017 Conference*. Portoroz / Slovenia. 2017.

[18] Schirner C. *Cluster-Forum Abwärmenutzung in der Industrie.* Bayern innovativ. 2010, NH Hotel Nürnberg – Germany (http://www.bayern-innovativ.de/abwaermenutzung2010/nachbericht?, accessed 18.08.2017)

[19] Schmidt, D., Kallert, A., Blesl, M., Svendsen, S., Li, H., Nord, N., and Sipilä, K. Low temperature district heating for future energy systems. *Energy Procedia*. 2017; 116, 26-38.

[20] Safety of pressure systems: *Pressure Systems Safety Regulations 2000*. Approved Code of Practice L122 HSE Books 2000 (www.hse.gov.uk/pubns/books/L122.htm, accessed 07.07.2017).

List of Figures

7 Internal friction

a. Friction tests

b. Comparison piston and shaft + friction power

8 Engine performance as function of expansion ratio n

a. Piston power P_{out}

b. Shaft power and speed

9 Efficiency and expansion ratio

10 Sub-atmospheric evaporation

a. Theor. efficiency [17]

b. System pressures, T_0 = 90°C, η = 0.011

Fig. 1a

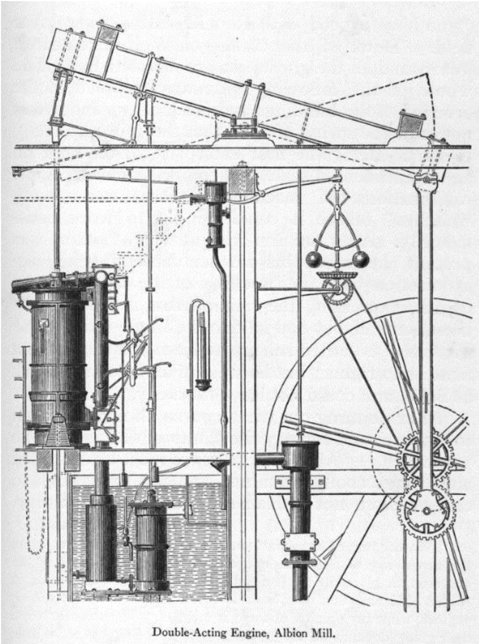


Fig. 1b

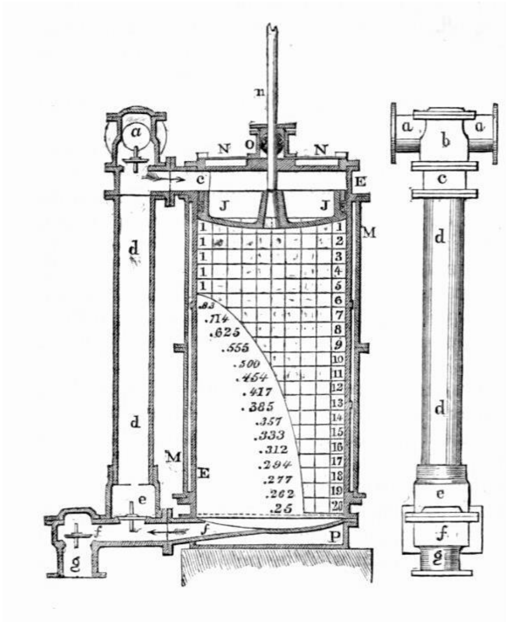


Fig. 2:

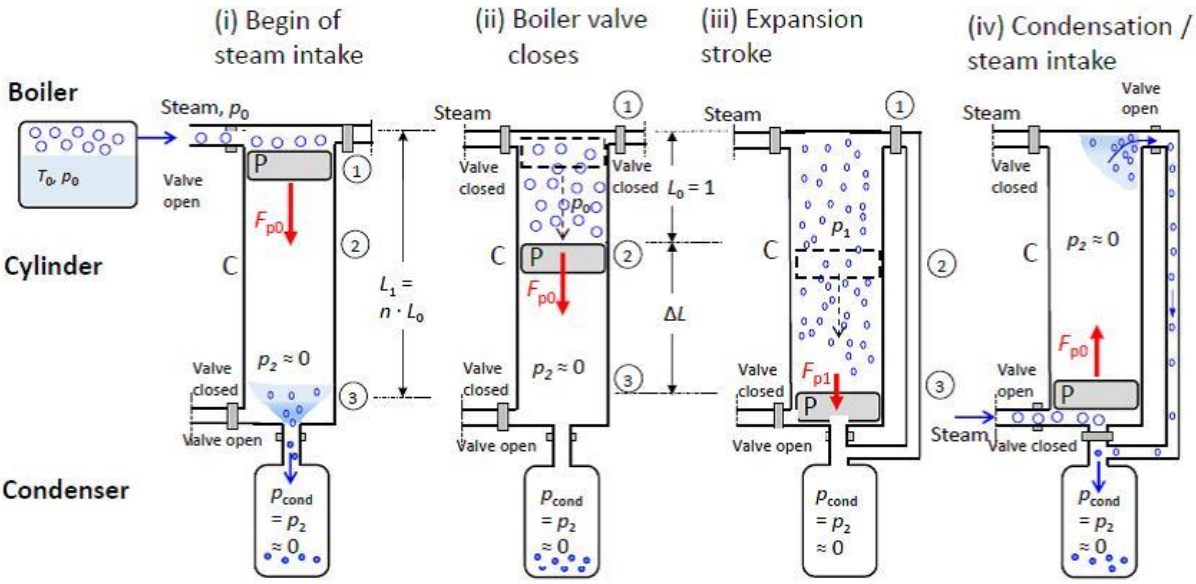


Fig. 3a:

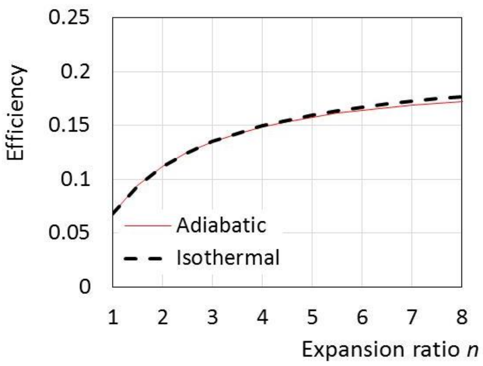


Fig. 3b

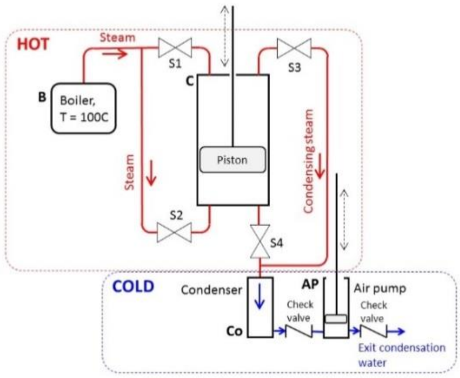


Fig. 4a

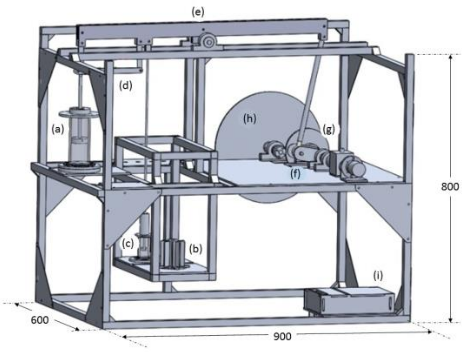


Fig. 4b

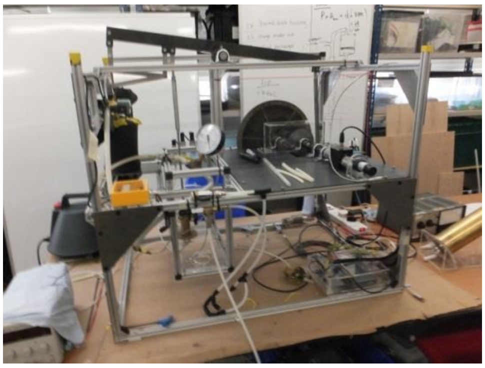


Fig. 5a

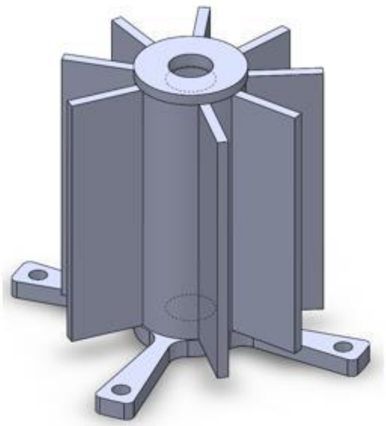


Fig. 5b

Fig. 6a

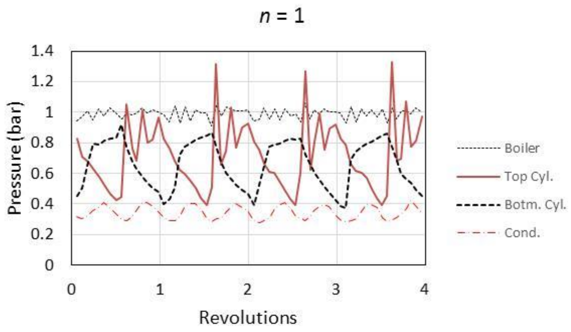


Fig. 6b

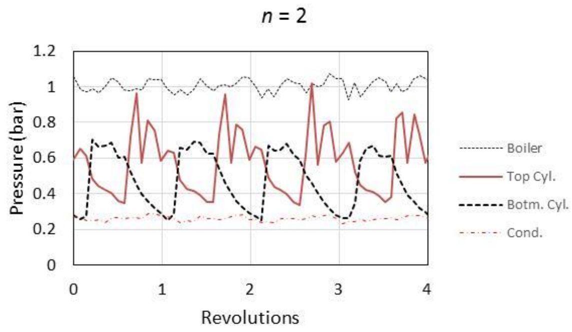


Fig. 6c

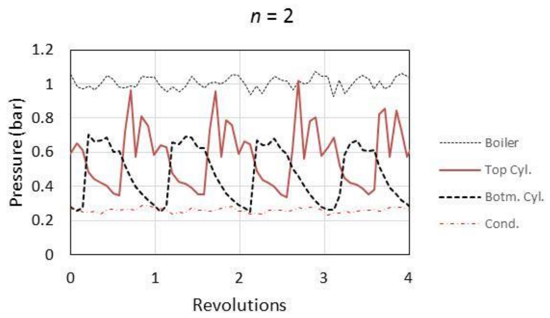


Fig. 7a

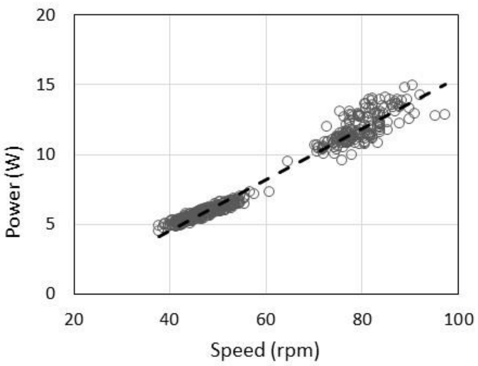


Fig. 7b

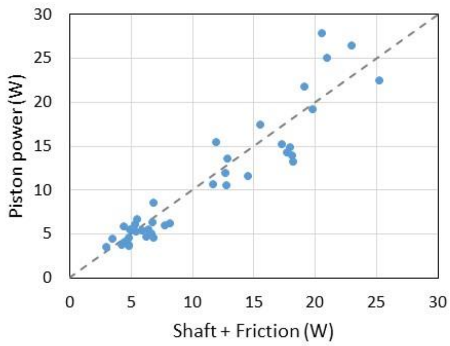
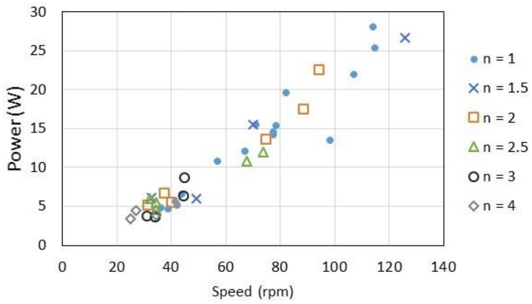


Fig. 8a:



Fig, 8b:

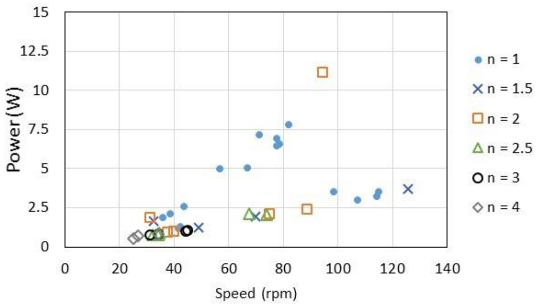


Fig. 9:

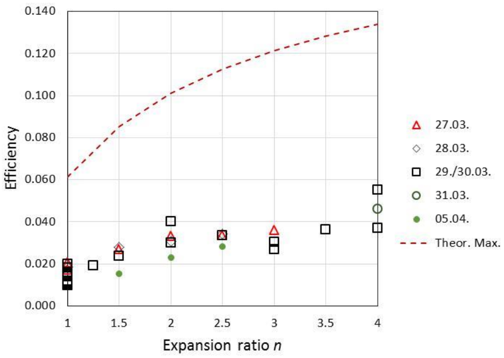


Fig. 10a

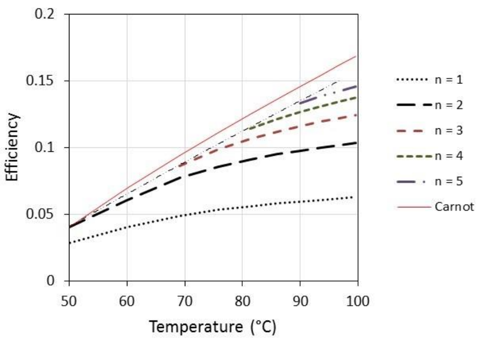


Fig. 10b