Experimental investigation of the atmospheric steam engine with forced expansion

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Abstract: Low and medium temperature thermal energy with temperatures of 100° to 150°C is available from renewable energy sources such as solar thermal or geothermal energy. Recent progress in flat plate solar thermal collector technology indicates that economical solutions for this temperature range are now becoming possible. Current technologies to generate mechanical energy from this temperature bracket such as Organic Rankin Cycle machines are however complex, and therefore only economical for larger units. There is a need for a simple, cost-effective medium temperature thermal engine for small scale applications. Recently, the atmospheric steam engine was re-evaluated for this application. The theory was extended to include a forced expansion strike. This can increase the theoretical efficiency of the ideal engine from 6.5% to 20%. In order to assess this theory, a series of experiments was conducted at Southampton University. It was found that the isothermal expansion of steam, and its subsequent condensation, is possible. The experiments showed a maximum efficiency of 10.2% for an expansion ratio of 1:4, indicating the validity of the theory. A further increase of efficiency to approximately 17% appears possible. It was concluded that the atmospheric engine with forced expansion has development potential.

Keywords:

Thermal engine, low and medium temperature, solar thermal energy, steam engine

1. Introduction

1.1 Overview

Low and medium temperature thermal energy is generated in many areas of renewable energy, such as biomass or solar energy, as well as in industrial processes. Solar thermal energy here probably constitutes the most abundant resource, which is also expected to grow in many areas of Europe with the effects of climate change becoming more pronounced [1]. Currently it is mostly employed for domestic heating (low temperature, < 80°C), and for energy generation at large scale installations (high temperature, often > 400°C). Cost-effective, medium temperature (100-200°C), medium scale systems e.g. for applications in industry or commercial companies however still require development. The availability of a simple, efficient and economical thermal engine for this temperature range and for power ratings between 5 and 100 kW would widen the potential area of application of solar thermal energy significantly. Ongoing research at Southampton University aims at the development of a cost-effective solar thermal system for low- and medium temperatures of 100° to 180° C. The system comprises a collector, and a thermal engine to generate mechanical from thermal energy. In this article, recent developments of the thermal engine are described.

1.2 Solar thermal energy

There is a large variety of technologies available for the harvesting of solar thermal energy available, see e.g. the overview in [2] or [3]. Flat plate or non-concentrating solar thermal collectors are probably the most economical collector types for solar thermal energy. Their main disadvantage is the comparatively low operating temperature (usually below 80°C), which makes them not suitable for many processes such as power generation.

Commercially available collectors are mostly designed for operating temperatures below 100 °C, high performance collectors can reach this temperature with 37% efficiency (assuming a solar energy of G = 800 W/m^{2}K), e.g. [4].

Recent developments of higher efficiency flat-plate, solar thermal collectors for low and medium temperatures of 120 to 200 C are however promising and may have the potential to open up this field of solar energy:

ISFH/Germany developed a double glazed flat plate collector with an Argon-filled cavity between glass plates, low-e glass, absorbing paint and with increased insulation [5]. From their data, an efficiency of 24% could be calculated for a temperature difference of 126K with a solar radiation of G = 800 W/m^{2}K. This would correspond to an operating temperature of 144°C, assuming an ambient temperature of 20°C. Losses amounted to 3.5 W/m^{2}K, indicating the potential of flat plate collectors.

Recent development work at Southampton University focussed on a low-cost solar thermal collector built from standard building materials and low-iron glass. The collector employed a large air gap of 150 mm, double glazing and passive convection control. Losses were measured as 2.3 W/m^2K at ΔT = 126 K [6]. With a solar radiation intensity of G = 800 W/m^2, the collector could reach an operating temperature of 144 °C with an efficiency of 45%.

The development of a cost-effective and efficient solar thermal energy supply therefore has reached a stage where the next step, the development of a low temperature thermal engine for decentralised small-scale application, is required.

1.3 Thermal machines for low and medium temperatures

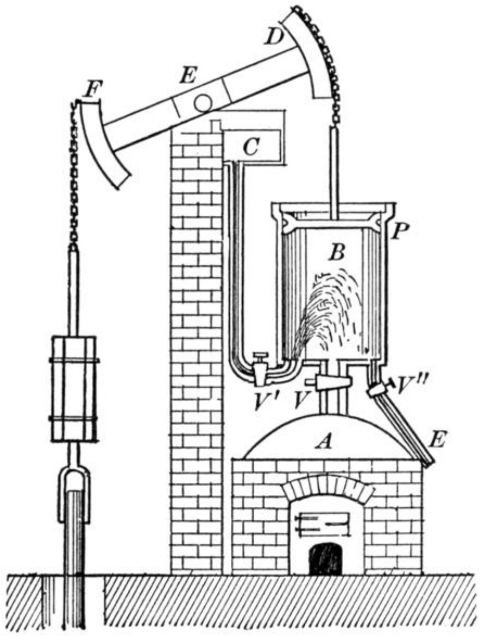
Several technologies for the conversion of thermal energy in this temperature range into mechanical and electrical energy exist. The most common principles for energy conversion are hot air engines (Stirling engines), and Organic Rankin Cycle (ORC) engines. Hot air engines employ the expansion of air when heated, and contraction when cooled. Their conversion efficiency for medium temperature situations is however quite low. Tests with a low temperature Stirling engine resulted in an efficiency of 0.44% for a heater temperature T_{Ev} of 166°C, [7]. ORC engines utilise working fluids with evaporation temperatures well below 100°C. The fluid is evaporated under pressures of 6 to 20 bar with temperatures of 80 to 180°C. The steam drives a turbine, and is then condensed to be evaporated again. Theoretical efficiencies are a function of the boiler temperature and the type of fluid used. Simulations for different fluids and evaporation temperatures gave efficiencies of 5.6% for T_{Ev} = 86°C, 7.7% for T_{Ev} = 109°C, and 13.1% for T_{Ev} = 169°C, [8]. In experimental investigations, an efficiency of 7.98% was reported for an operating temperature of 120°C and a pressure of 9 bar [9]. The system is however quite complex and comprises an evaporator, turbine, scroll condenser, pumps and a regenerator. This complexity, combined with the design requirements for a pressurized, expensive fluid means that smaller units (P < 150 kW) are difficult to produce cost-effectively. Today, ORC thermal machines are mostly used in the fields of biomass and geothermal energy, and waste heat recovery. A promising area for application is seen in small scale solar thermal systems with Fresnel concentrators which deliver lower temperatures than e.g. parabolic trough systems, but require lower investment costs [10].

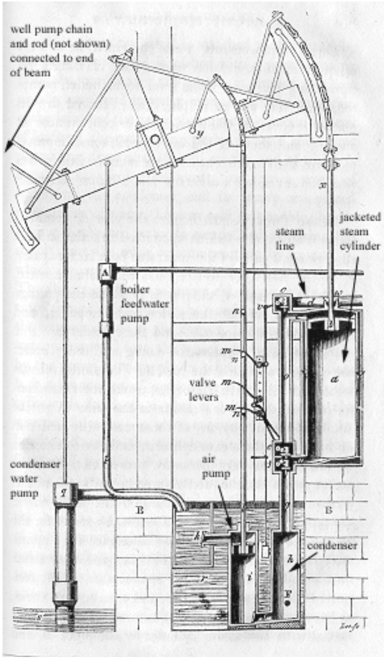
2. The atmospheric steam engine

2.1 Historical development

The atmospheric steam engine is the oldest type of practical steam engine. It was initially developed by Thomas Newcomen in 1712, and significantly improved by James Watt with the introduction of the external condenser in 1776. The ASE operates at atmospheric pressure, and employs a vacuum generated by the condensation of steam as driving force.

In the simplest version, the machine consists of a boiler, a cylinder with an inlet for cold water, and a piston, Fig. 1a. During the upwards motion, steam is drawn into the cylinder. When the uppermost position is reached, the boiler valve is close and cold water injected into the cylinder. The steam condenses, a near vacuum forms and the atmospheric pressure drives the piston downwards.





a. Newcomen's atmospheric engine [11]   b. Watt's engine with external condenser [12]

Fig. 1: Historic machines

Newcomen's engine had very low efficiencies of approximately 0.5%, [13], since with every injection of cold water the cylinder cooled down, and steam had to be employed to heat it up again. James Watt introduced the external condenser in 1776. An additional small vacuum cylinder was added to the machine, Fig. 1b. During the upward motion of the piston, steam was drawn into the main cylinder. In the condenser, the piston was also moved upwards to create a near vacuum. When the working piston reached the uppermost position, the boiler valve was closed and the condenser valve opened. The vacuum drew steam into the condenser where it condensed into water, maintaining the vacuum and drawing more steam. This machine had the great advantage that the working piston remained hot, and the condenser cold so that efficiencies were increased to 3.5%, [13].

The theory shows that the atmospheric engine can only recover the displacement work of the water as it evaporates and displaces 1.69 m^3 of atmosphere for 1 kg (or 0.001 m^3) of water. This work is, in the ideal case, 169 kJ/kg. In the same time, the thermal energy required to heat 1 kg of water to 100°C, and to evaporate it, amounts to 2601.5 kJ/kg, so that the maximum theoretical efficiency is only 6.5%.

With the advent of high pressure machines, the atmospheric engine disappeared. The main reason was the limited efficiency of the atmospheric engine. The atmospheric engine does however have several advantages:

1. Simplicity,
2. It uses a cheap. non-toxic, not inflammable working fluid,
3. It operates at very low temperatures compared with other thermal engines,
4. It operates under atmospheric or sub-atmospheric pressures so that there is no danger of boiler explosions. This reduces manufacturing and maintenance / certification costs substantially.

The last application of the atmospheric steam engine known to the author is the machine designed by Davey in 1884, [14]. Davey advocated the design on the grounds mentioned above plus the fact that, since there is no danger of boiler explosions, the machine could be situated anywhere, even in residential areas.

2.2 Recent developments

Recently, the concept of the atmospheric engine was revisited in order to assess its potential for the utilisation of low temperature thermal energy, [15]. The classic atmospheric cycle described in the previous section was modified to include a forced expansion of the steam. The theoretical work indicated that the efficiency of the atmospheric steam engine could be increased from 6.5% to 20%.

In a forced expansion cycle, initially a certain volume of steam is drawn into the cylinder. The boiler valve is closed. The piston is then drawn upwards in order to expand the steam. The mechanical work required for the expansion is the integral of the external force applied over the expansion length. This force is zero at the beginning of the expansion, and reaches a maximum at the end of the expansion. The maximum expansion force is therefore always significantly smaller than the atmospheric force acting on the piston from the outside. Once the prescribed expansion ratio is reached, condensation is initiated. The atmospheric force now conducts work over the full length of the stroke (initial steam volume plus expansion length).

In [15], the theory of the ASE with forced expansion was presented for an adiabatic expansion of the steam. In a real engine however, the cylinder will remain hot so that the expansion there will be isothermal: For an initial volume v_1, and a given expansion ratio n =

v_2/v_1, the expanded pressure is $p_2 = p_1/n$. For a cylinder cross sectional area of $A_{Cyl} = 1\ m^2$, the expansion work W_{exp} becomes:

$$W_{\exp} = p_1 \cdot (v_2 - v_1) - p_1 \cdot v_1 \cdot \ln \frac{p_2}{p_1} \quad (1)$$

Where $p_1 = p_{atm}$.

The total work of the system W_{tot} after condensation of the steam is:

$$W_{tot} = p_{atm} \cdot v_2 - W_{\exp} \quad (2)$$

The thermal energy W_{isoth} which has to be supplied to the expanded steam in order to maintain its temperature is small, due to the low adiabatic coefficient κ of wet steam (κ = 1.035 to 1.08, [16]). The thermal energy required ranges from 0 (n =1) to 5% (n = 12) of the isothermal expansion work W_{exp}, and can be calculated from the temperature drop in the adiabatic expansion. It is included in Fig. 2. The thermal energy input E_{th} required for a given volume of steam $v_1 = 1\ m^3$ can be calculated as follows (all units in m, J, K and kg):

$$E_{th} = (2{,}256{,}500 + 4{,}200 \cdot 70) \cdot \frac{v_1}{1.69} + W_{isoth} \quad (3)$$

2,256.5 kJ/kg is the latent heat of water, the specific heat capacity of water is 4,200 J/kgK, an initial temperature of 30°C is assumed for the water and 1 kg of water amounts to 1.69 m^3 of steam. For the calculation of the thermal efficiency it is assumed that the expansion work is provided by the work generated by the machine, and therefore has to be subtracted from the condensation work. The thermal efficiency η then becomes:

$$\eta = \frac{W_{tot}}{E_{th}} \quad (4)$$

Fig. 2 shows the theoretical efficiency from a forced expansion stroke as a function of the expansion ratio n for both adiabatic and isothermal expansion. The efficiency ranges from 0.065 for n = 1 to 0.198 for n = 10. Isothermal expansion gives in marginally higher efficiencies, and approximately 3% more power output per unit volume compared with adiabatic expansion.

It should be noted that the Carnot efficiency limit does not apply directly here since there is an additional energy input – the expansion force – into the system. A more detailed discussion of this aspect is given in [15].

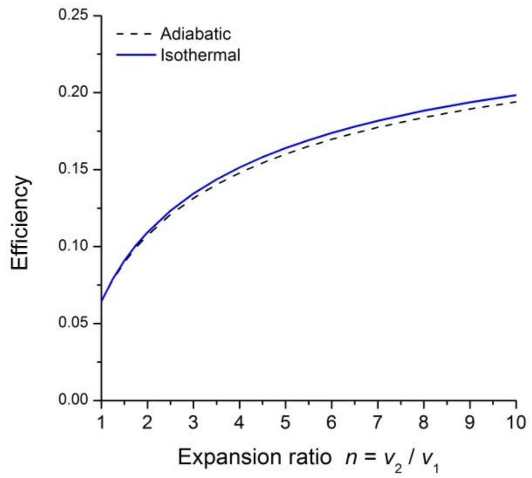


Fig. 2: Theoretical efficiency as function of the expansion ratio *n.*

Forced expansion appears to open the possibility to create a thermal engine for temperatures of 100 °C with efficiencies exceeding those from ORC engines, whilst avoiding complex pressurized systems with expensive working fluids.

3. Experiments

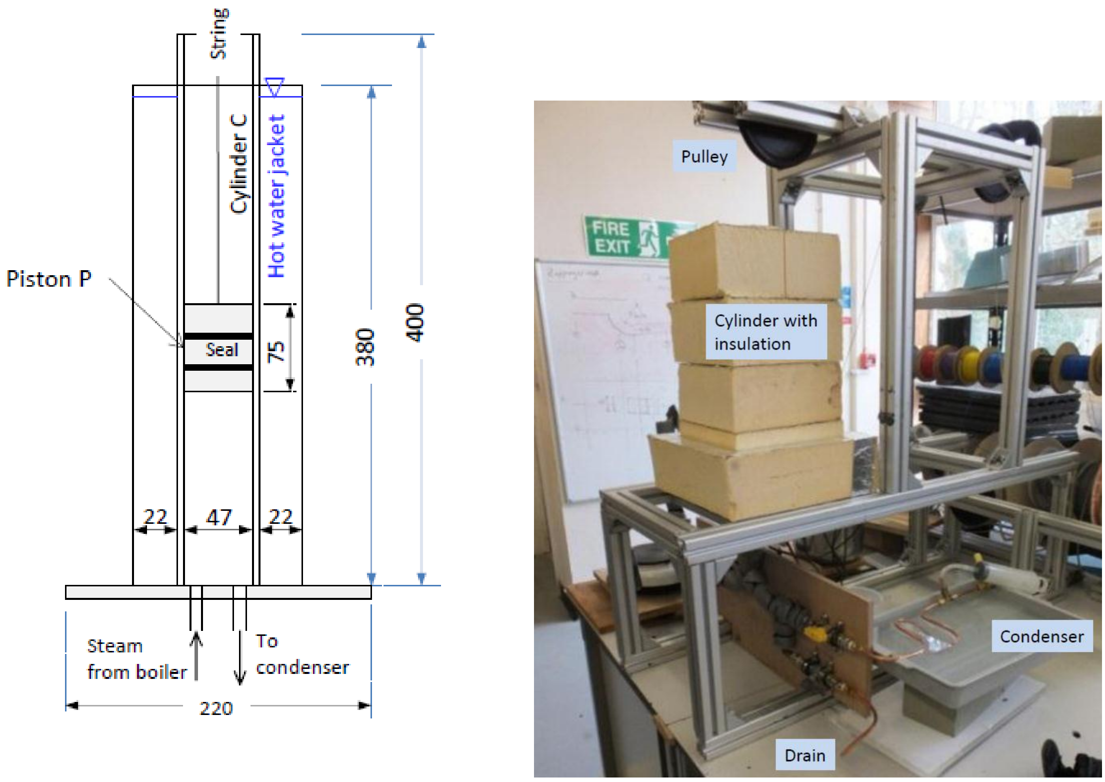
3.1 Experimental set-up

The experiment was designed in order to assess the theory of isothermal forced expansion. The specific aims were:

1. To establish whether or not the isothermal expansion of steam is possible, and
2. To determine the efficiencies of a working stroke without and with forced expansion.

In order to reach these two aims, a simple one-stroke bench model was designed. It consists of a vertical cylinder, a piston, a boiler, a condenser and a load rig which allows to lift the piston. The forces acting on the cylinder are measured with a scale attached to the lifting rope. The piston movement was controlled with a winch.

Fig, 3a shows the cylinder itself. It consists of a brass base plate 220×220 mm, t = 10 mm, a brass inner cylinder of 400 mm height with an outer diameter of 56 and an inner diameter of 47 mm, and an outer cylinder of 100 mm inner diameter. The space between outer and inner cylinder is filled with boiling water in order to maintain the working temperature inside. Additional insulation material (polyurethane foam) of approximately 50 mm thickness was added round the cylinder to prevent further heat losses. The piston was made of stainless steel, with a diameter of 46.5 mm, a length of 75 mm and a mass of 0.95 kg. Two O-rings were used to seal the piston. The cylinder was mounted on a frame made from aluminium profiles. The copper pipes and taps / switches required for operation were fixed to a wooden panel. The complete apparatus is shown in Fig. 3b. Fig. 4 shows a schematic of the whole system.



a. Cylinder and piston b. Complete apparatus

Fig. 3: Experimental set-up

A 205 mm diameter kettle with a maximum content of 5l was used as boiler. During the experiments it was found that a volume of 0.825 l was evaporated within 30 minutes, corresponding to a delivery of 0.78 l of steam per second.

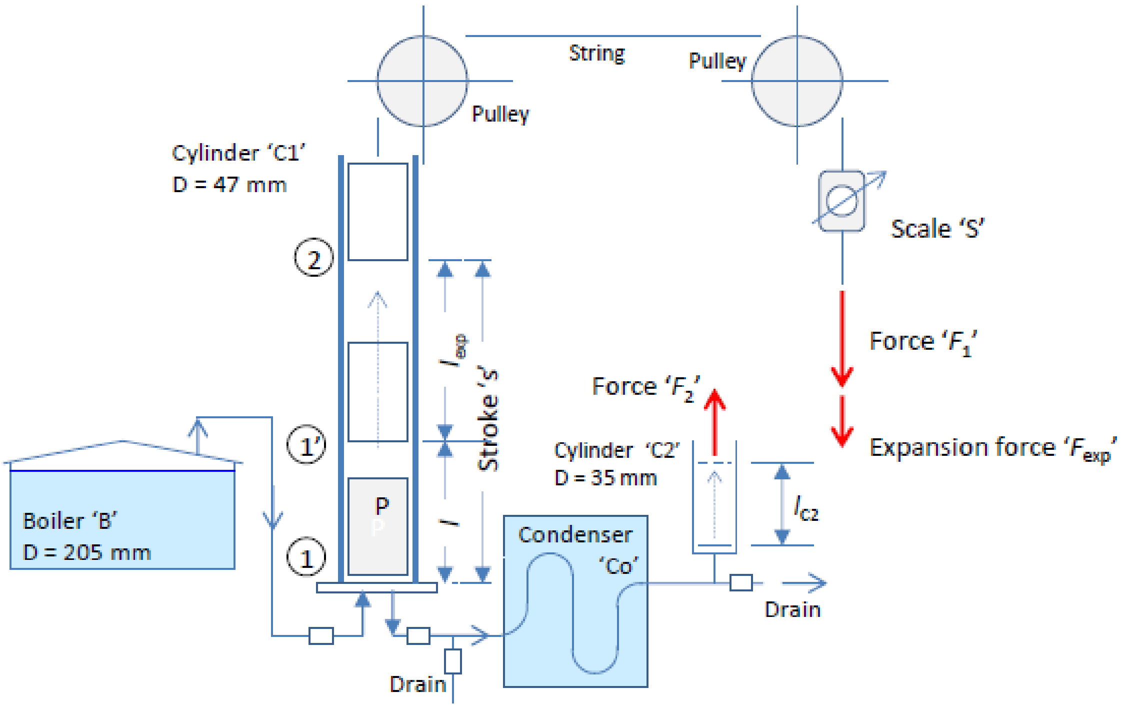


Fig. 4: System with external forces

The condenser was built from an 800 mm long copper tube with an external diameter of 6 mm, and an internal diameter of 3 mm. The condenser is connected to the working cylinder C1 with a tap which allows to open or close the connection. A drain tap is also attached so that after every stroke the condenser can be cleared of condensation water. In the condensation section, the copper tube was bent into a W-shape which in turn was set into a basin with cold water. A 100 ml medical syringe with an internal diameter of 35 mm was used as cylinder C2 to create a low pressure inside the condenser initially, and then to evacuate the air which leaked into the cylinder C1 during the working stroke. The boiler B is connected to the working cylinder C1. Inside C1 runs the piston P. attached to P is a string which runs over two pulleys and is connected to a scale. The external load is applied at this point.

3.2 Tests

3.2.1 Overview

Two series of tests were conducted:

(1) Series 1 with condensation only,
(2) and series 2 with forced expansion.

The working stroke length *s* was constant for all tests with *s* = 200 mm. Initial steam volumes varied with *l* = 50 to 200 mm, and expansion lengths of l_{exp} = 50 to 150 mm. Before the tests, the rig was heated up by filling the jacket with boiling water, and by drawing and expelling steam several times. The condensation which formed initially was thereby drained as well. For every expansion ratio, a four strokes were measured. Friction forces were determined as 28 N upwards, and 6 N downwards. These forces were added to (downwards) or subtracted from (upwards) the force measurements in order to obtain the actual forces acting on the piston.

3.2.2 Series 1: condensation only

The piston P is lifted from the starting position at point ‘1’ by a distance ‘s’, filling C1 with steam. When point ‘2’ is reached, the boiler valve is closed, and a force is applied to the cylinder C2 to create a low pressure in the condenser Co. Then the condenser valve is opened, condensation occurs, the pressure in the cylinder drops, and the piston P, which is initially held in position, is released slowly back to position ‘1’. The force F_1 acting on P is measured with scale S at the beginning (F_{12}. pos. 2) and the end of the working stroke (F_{11}, pos. 1). While the piston P moves, the plunger in Cylinder C2 is lifted through a distance l_{C2}, which was constant in this test series at l_{C2} = 80 mm, by applying a force F_2 in order to extract the air from C1, and thus to allow P to return to Pos. 1. The work is then calculated as follows:

Condensation work W_{cond}:

$$W_{cond} = F_1 \cdot s = \frac{F_{11} + F_{12}}{2} \cdot s \qquad (5)$$

The work W_{C2} conducted at C2 is calculated by reducing the force F_1 with the area ratio of cylinders C1 and C2 (assuming that the pressure in the system is the same everywhere):

$$W_{C2} = F_1 \cdot \frac{D_2^2}{D_1^2} \cdot l_{C2} \qquad (6)$$

Total work W_{tot}:

$$W_{tot} = W_{cond} - W_{C2} \qquad (7)$$

The tests showed that the seal was not perfect, and therefore a complete vacuum could not be achieved. Cylinder pressures at condensation only reached an average value of p_{cond} = 46 kPa (abs.). The maximum theoretical efficiency of an atmospheric cycle with a perfect vacuum (i.e. a pressure difference of 100 kPa) is 6.5%. With a residual pressure 46 kPa (abs.), the maximum theoretical efficiency η_{theor}, becomes

$$\eta_{theor} = \frac{100 - 46}{100} \cdot 6.5 = 3.5\% \qquad (8)$$

In addition, thermal energy is required to heat the air drawn first into C1, and then into C2. With a specific heat capacity of air of 717 J/kgK and a density of air at atmospheric pressure of ρ_{air} = 1.25 kg/m^3 the energy E_{Air} required becomes:

$$E_{air} = 717 \cdot 1.25 \cdot \frac{0.035^2}{4} \cdot \pi \cdot l_{C2} \cdot \frac{p_{atm} - p_{cond}}{p_{atm}} \tag{9}$$

With a specific heat capacity of the water of 4.2 kJ/kgK, the required thermal energy E_{th} (assuming an initial temperature of the water of 30°C) then is:

$$E_{th} = A_{C1} \cdot l \cdot (70 \cdot 4200 + 2256{,}500)/1.69 + E_{air} \tag{10}$$

The total energy W_{tot} becomes

$$W_{tot} = W_{cond} - W_{C2} \tag{11}$$

With this input energy, the efficiency η can be calculated:

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

3.2.3 Series 2 (with forced expansion):

Steam is drawn in from the boiler for the initial length ‘*l*’ from pos. 1 to 1’. Then the boiler valve is closed. The piston is lifted further through the expansion stroke length ‘l_{exp}’ to pos. 2 by applying a force F_{exp}, which varies from zero (pos. 1’) to a maximum value at pos. 2. A force is applied at the cylinder C2 to create a low pressure in the condenser Co. Then, the condenser valve is opened. Condensation takes place, resulting in a sudden increase in F_{12}. The piston P is still held in position ‘2’. A force F_2 is applied at C2 to draw the air from cylinder C1 in until the piston P reaches pos. 1 again, with a reduced end force F_{11} acting now on the piston. The work is then calculated as follows:

Condensation work W_{cond}:

$$W_{cond} = F_1 \cdot s = \frac{F_{11} + F_{12}}{2} \cdot s$$

(13)

The expansion pressures measured in test series 1 did not correspond well with the theoretical values for the measured expansion lengths. This led to the conclusion that air was drawn into C1 during the expansion stroke. Using the theory of isothermal expansion in order to calculate the expansion work was considered as inadequate. In order to determine the expansion work, a linear variation from zero to F_{exp} was therefore assumed.

$$W_{\exp} = \frac{F_{\exp}}{2} \cdot l_{\exp} \tag{14}$$

Work in cylinder C2:

With condensation, a force F_2 acts on the plunger in C2. The plunger is moved upwards by a distance l_{C2}, which varied from 80 to 94 mm, in order to remove air from C1. Work conducted at C2:

$$W_{C2} = F_1 \cdot \frac{D_2^2}{D_1^2} \cdot l_{C2} \tag{15}$$

Total external work W_{tot}:

$$W_{tot} = W_{cond} - W_{C2} - W_{\exp} \tag{16}$$

Due to the isothermal expansion, thermal energy equal to the expansion work has to be added to the required thermal energy E_{th} . which becomes:

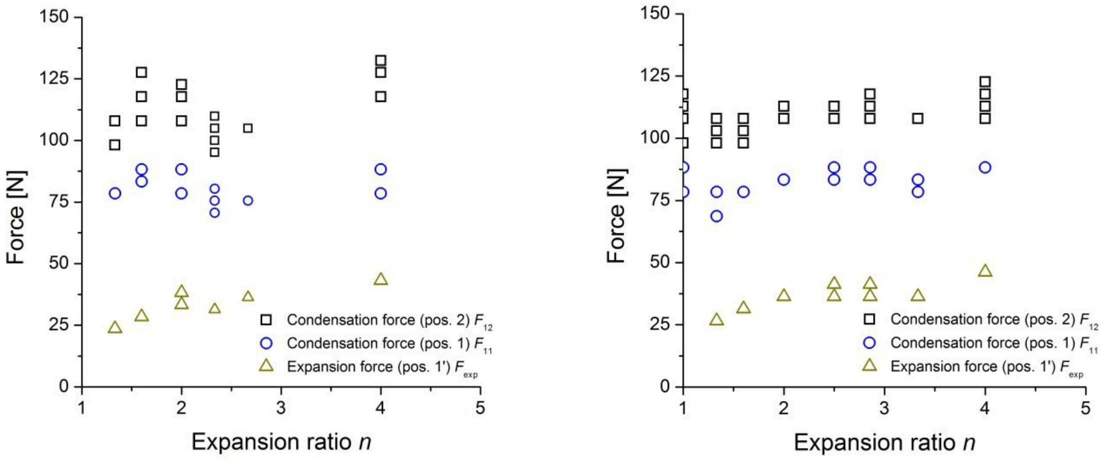
$$E_{th} = A_{C1} \cdot l \cdot (70 \cdot 4200 + 2260000)/1.69 + W_{\exp} + E_{air} \tag{17}$$

With this input energy, the efficiency η can be calculated:

$$\eta = \frac{W_{tot}}{E_{th}} \tag{18}$$

4. Results and analysis

Two test series were conducted, the first on 19.03.2014, and the second on 24.03.2104. Fig. 5a and b show the forces measured at the piston P for expansion ratios of n = 1 (condensation only, no expansion) to n = 4. The forces measured on 24.03.2014 (Fig. 5b) are slightly smaller than those measured on 19.03., this was thought to be caused by abrasion of the sealing rings, and subsequent increased air ingression. Fig. 5 indicates that the tests are fairly repeatable.



a. Tests 19.03.2014 b. Tests 24.03.2014

Fig. 5: Measured forces

Table 1 shows the mechanical power and the thermal input power generated in the experiments. Column 2 shows the minimum and maximum power measured during the working stroke, column 3 the total mechanical output from Eq. 7 (n = 1:1) and Eq. 16 (n = 2:1, 4:1). In column 5 finally the thermal energy required is shown. It can be seen from columns 2 and 3 the work generated during the down stroke (working stroke) does not differ very much for the different expansion ratios. The total work (col. 3) for n = 1:4 is slightly less than the work from the tests with n = 1:1. However the thermal energy input for the expansion ratio n = 4:1 is only a quarter of the energy input for the fully atmospheric cycle (n = 1:1).

1	2		3		4		5	6
	Work downstroke [J]		Expansion Work [J]		Total work per stroke [J]		Steam vol. [cm^3]	Thermal work [J]
Expansion ratio	Min.	Max	Min.	Max	Min.	Max.		
1:1	17.7	21.6	0	0	14.0	17.1	346.8	523
1:2	19.6	21.1	1.7	1.9	13.7	14.0	173.4	265
1:4	19.6	22.1	3.2	3.2	12.8	13.9	86.7	136

Table 1: Work measured during the experiments (Tests 19.03.2014)

Fig. 6 shows the efficiency calculated from the measurement values as a function of the expansion ratio n. The tests without expansion (n = 1) resulted in efficiencies of up to 0.032. The average condensation pressure acting on the piston P after condensation for all tests was 55 kPa. This means that the maximum theoretical conversion efficiency for a condensation stroke (pressure in C1 = 45 kPa abs.) is 0.036, slightly higher than the experimental values. The highest measured efficiency for n = 4 was 0.102, exceeding even the theoretical maximum efficiency of the ideal atmospheric cycle (0.065) by 60%.

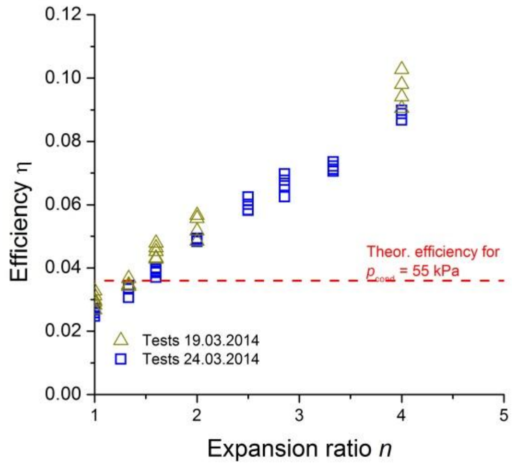


Fig. 6: Efficiencies

The working cycle was assumed to be isothermal, since the cylinder temperature was kept at 100°C by the hot water filled external jacket tube in which the cylinder was located. Internal temperatures were not measured, so that it is difficult to ascertain the actual degree of isothermal expansion. Theory as well as the low magnitude of the expansion work determined from the tests (only up to 16% of the work from the condensation stroke) indicate that temperature differences during expansion were small. This implies that near isothermal conditions prevailed.

The test rig was designed for single stroke operation. It may however be of interest to estimate what power output can be expected from a rotating engine. Assuming continuous operation, and a rotational speed of 120 rpm, the power output of the experimental rig would range from 24.6 to 30W for a cylinder volume of 0.347 l. The engine's power density per unit swept volume can then be determined as 0.083 kW/l cylinder volume. With improved sealing, a condensation pressure of 4 kPa (abs.) should be possible. This would increase the efficiency to approximately 14%, and the power density to 0.115 kW/l.

A real machine would however need to be significantly larger than the experimental rig. Also, a higher expansion ratio of n = 7 to 10 would probably be chosen to increase engine efficiency. This comes however at the cost of power density. Assuming a cylinder diameter of 400 mm, a stroke of 800 mm, a speed of 90 rpm and a twin cylinder machine for smoother running the power output for an expansion ratio of n = 1:8 would reach 13.7 kW for a thermal input of 96 kJ. The machine would require a steam volume of 37.7 l/s.

5. Discussion

5.1 Experiments

The experiments conducted at Southampton University confirmed that the theoretically predicted concept of the atmospheric engine with forced expansion is feasible. The possibility to expand steam, and to condense the expanded steam, thereby increasing the efficiency of the atmospheric steam engine, was demonstrated.

For the evaluation of the experimental results presented in the previous section, the theoretical maximum efficiency of the ideal atmospheric engine (without any losses / with perfect vacuum) with 6.5% constitutes one benchmark. The residual pressure observed in the experiments after condensation was 45-46 kPa (abs.), nowhere near a perfect vacuum. The maximum theoretical efficiency of the ideal atmospheric working stroke (without forced expansion) was therefore only 3.5%. With maximum efficiencies of 8.9 to 10.2%, this benchmark was exceeded by a factor of 2.8. Even the efficiency of the ideal atmospheric cycle was exceeded by 60%.

The experiment suffered from an unsatisfactory sealing of the cylinder, which was caused by the use of a tube as cylinder. The brass tube had a deviation of the true diameter of 0.2% or approximately 1 mm, which the O-ring seals could only partially compensate. In consequence, the minimum pressure in the cylinder at condensation did not drop below 45 kPa (abs.), limiting the possible work of the condensation stroke. During the expansion stroke only about 1/3 of the theoretical pressure was reached, and a significant amount of air drawn into the cylinder.

5.2 Performance

The overall performance however was considered promising. The highest measured efficiency of 10.2% substantially exceeds values reported for much more complex ORC thermal engines for higher operating temperatures of 120°C (7.98%), [9]. A lower condensation pressure will be achievable with better sealing and a more accurate cylinder and piston. Condensation at 4 kPa (abs.) should increase the efficiency from 10% to 15%. In a real machine, there would be losses from the boiler and energy losses through the cylinder insulation, so that the actual efficiency from energy in to mechanical energy out would be somewhat lower.

For actual applications, the atmospheric engine does however have limitations:

1. The comparatively low energy density of unpressurized steam means that large volumes for cylinder and boiler are required.
2. The speed of the machine will also be low, it is currently estimated at 90 rpm due to the long stroke.

3. The condenser produces low-grade heat with temperatures approximately 10K above ambient. The condenser fluid will need to be cooled down to ambient temperature, and the thermal energy will need to be released into the atmosphere.

The potential advantages can be listed as follows:

1. With boiler efficiencies of 90%, and further 5% thermal losses in the cylinder, total system efficiencies 14% for n = 8 seem achievable. The atmospheric engine with forced expansion therefore constitutes a significant improvement.
2. The ASE is simple compared e.g. with ORC engine systems, indicating cost effectiveness,
3. Operating temperatures are low compared with other thermal engines, widening the possible area of application.
4. The working fluid is cheap, readily available, non-toxic, not inflammable.

5.3 Solar thermal system

The work on cost-effective, medium temperature flat plate solar thermal collectors described in [6] indicates that for larger collectors (e.g. 3×3 m area) are more efficient. For operating temperatures of 130°C (G = 800 W/m^2), efficiencies of 60% are possible The overall mechanical efficiency (sun to shaft) of a collector combined with an atmospheric steam engine (operating temperature T_{Ev} = 100°C) can then be estimated as 9 to 9.5%. This would probably give a sun-to-wire efficiency of approximately 8%. It should be noted that the ORC engine reported in [9] had an engine-only efficiency of 7.98% for an operating temperature of 120°C.

Overall efficiencies of the solar thermal system would be lower than those of e.g. PV systems. The atmospheric engine is however a simple machine, so that the overall cost-effectiveness needs to be considered in the next development step.

6. Conclusions

Low and medium temperature thermal energy is available from many renewable energy. sources. The cost effective conversion of thermal into mechanical energy however still poses an engineering problem. One solution for thermal energy with temperatures of 100 to 150°C could be the atmospheric steam engine. Its theory was recently revisited in order to increase the machine’s efficiency. The improved theory indicates that the addition of a forced expansion stroke can increase the theoretical efficiencies from 6.5 to 20%. At Southampton University, a series of fundamental model tests was conducted in order to assess these predictions. The following conclusions were drawn.

1. The theoretically postulated atmospheric cycle with forced expansion of steam is possible.
2. Air leakage through the seals limited the performance of the experimental machine.
3. The efficiency without expansion reached 3.2% with a condensation pressure of 46 kPa (abs.).
4. Efficiencies with forced expansion ranged from 4.1% for an expansion ratio of 1.33:1 to 10.2% for an expansion ratio of 4:1.
5. The theoretical maximum efficiency of the simple atmospheric cycle of 3.6% was exceeded by a factor of 2.8%.

The concept of forced expansion was demonstrated successfully. A substantial increase in cycle efficiency was observed. The atmospheric engine with forced expansion has significant further development potential.

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