AC 2012-4611: THERMODYNAMIC MODELING OF 18TH CENTURY STEAM ENGINES

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Thermodynamic Modeling of a Newcomen Steam Engine

abstract

The steam engine developed by Thomas Newcomen was the first successful reciprocating engine and celebrates its 300th anniversary this year. Newcomen's first engine was built in 1712 and more than 1400 were built during the 18th Century. Newcomen's design condensed steam inside a piston and cylinder through a water spray injection process. The vacuum formed in the cylinder, in combination with atmospheric pressure on the top of the piston, actuated a reciprocating pump via an overhead "walking beam." This first engine served to pump water from a coal mine in England, but the power technology thus created enabled the Industrial Revolution and sees its legacy in the steam-powered utility power plants of today.

In commemoration of the tercentenary of Newcomen's engine, a group of Mechanical Engineering students at the United States Naval Academy designed and built an instrumented operating model of a Newcomen engine. A significant aspect of this project was to develop an understanding of how the original engines worked. This paper provides authentic design and operating data collected from historical documents for an actual Newcomen engine and illustrates the thermo-fluids analyses of this reverse-engineering portion of the design project.

Today's students should be aware of significant historical developments as part of their engineering education. The types of analyses included in this paper allow for relatively easy integration into existing thermo-fluids courses and, at the same time, allow for development of an appreciation for the history of steam engineering and its contribution to the engineering profession.

introduction

The year 2012 marks the tercentenary of a significant engineering milestone. Thomas Newcomen (1663-1729) invented the first successful reciprocating steam engine with an inaugural installation in 1712. Newcomen's engine used a piston and cylinder configuration and falls into the category of an "atmospheric" engine: the cylinder was open at the top with the piston exposed to atmospheric pressure. When a vacuum was formed under the piston by condensing steam, the "weight" of the atmosphere exerted a net force on the piston, thereby causing the piston to move through its power stroke.

Between 1712 and 1800, more that 1400 of these engines were built.[1] Even while eclipsed by the more famous Boulton & Watt steam engines that were developed in the last three decades of the 18th Century, Newcomen engines continued to be built and operated well into the 19th Century and a few operated into the 20th Century. The purposes of this article are to describe the operation of and perform modern thermodynamic analyses on a Newcomen pumping engine.

A note about units. The Newcomen engine was developed in a time of British Imperial Units from which the United States Customary System (USCS) of units were derived. Out of deference to these original units and the fact that the vast majority of engineers practicing engineering in the United States continue to use "English" units, the analyses of this paper continues their use. Brackets {} are used to highlight conversion factors used in the equations.

background

By the late-17th Century, mining in northern Europe had progressed below the land surface to the point that groundwater impeded access to the coal, mineral, and metal veins. Both reciprocating suction and reciprocating plunger pumps already existed and were used for dewatering. This basic technology had existed for millennia, but to this point in time, these pumps were driven by human, animal, wind, an even water power. The situation called for more effective and less-costly means of dewatering mines.

Thomas Newcomen invented the first practical piston-and-cylinder engine by connecting a piston to one end of a rocking beam that was connected, on its other end, to a separate reciprocating pump. Newcomen started the design of his pump around 1705 and built the first commercial unit in 1712. Fig. 1 shows the basic elements of Newcomen's engine. Subsequent engines also largely followed the general configuration shown in this figure.

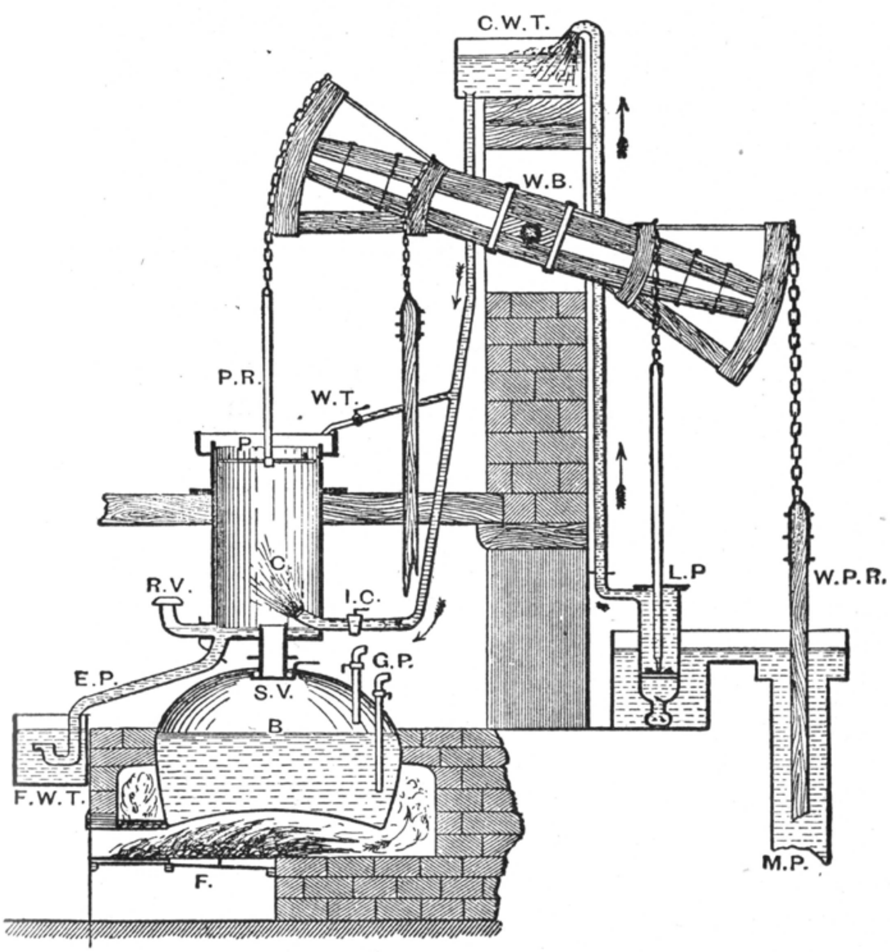

Figure 1 - Newcomen's Pumping Engine[2]
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Newcomen's design raised the piston (P) via gravity on a weighted pump rod (WPR) in a "non-work" reset stroke while admitting steam at low pressure to the cylinder. The main pump (MP), walking beam (WB), piston and associated chains and plug rods that operated auxiliary pumps

(LP) and controlled the positions of various valves (IC and SV) were all balanced such that the piston would reset to the top of its stroke, the position shown in Fig. 1, with steam at or only slightly above atmospheric pressure.

The cycle description commences with the piston at the top of its stroke and the cylinder full of steam. Water from the cold water tank (CWT) was sprayed under gravity directly into the cylinder (C) to condense the steam. The atmospheric pressure on the top of the piston being greater than the vacuum below, the piston was pushed down in the work stroke. Steam was then admitted to the cylinder from the boiler to reset the cycle. Liquid water was drained and non-condensable gases that had made their way into the cylinder were bubbled out via the feedwater tank (FWT). The piston incorporated a water-aided seal, hence the pipe to supply sealing water via the water tap (WT) to the top of the piston.

Fig. 2 shows a simplified schematic of a Newcomen engine and pump.

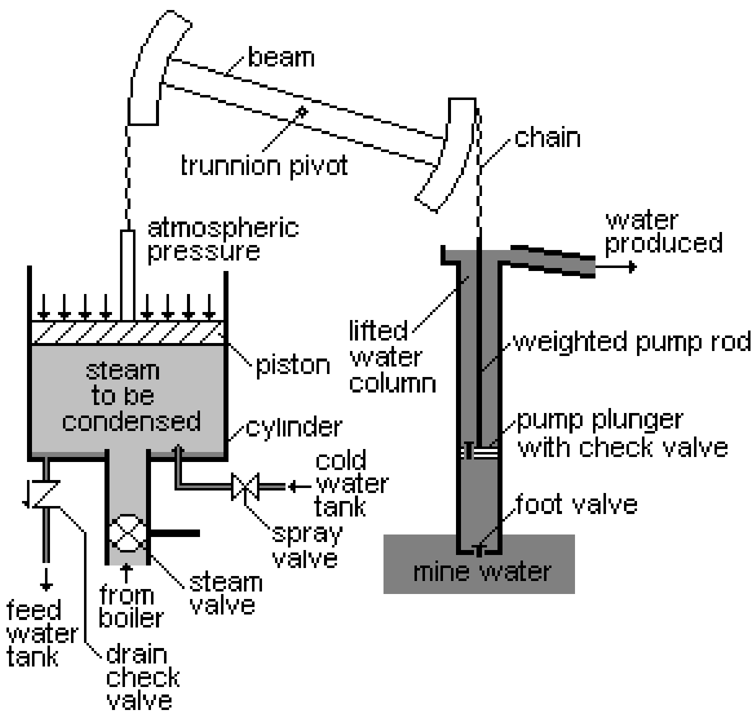

Figure 2 – Simplified Schematic of a Newcomen Engine

The pumps used with Newcomen engines typically employed a series of lifts. This reduced the back-pressure on the pump plunger seals and check valves and would have served to reduce back-leakage. However, leakage was a practical issue, so the lower lifts required additional capacity to make up for it. Each of the pumps was connected via a yoke to the walking beam and the beam pivoted around a central trunnion: each pump had the same stroke as the steam piston. Fig. 3 shows a simplified schematic of a double lift pump.

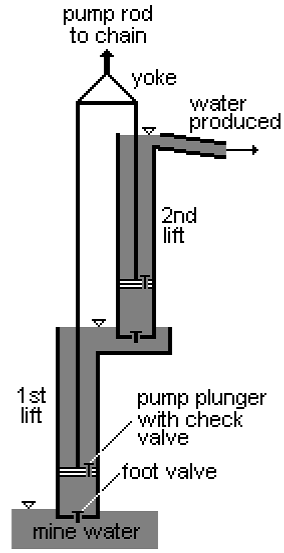

Figure 3 – Simplified Schematic of a Double Lift Pump

The Steam Engine of Thomas Newcomen, by L.T.C. Rolt and J.S. Allen,[1] is considered the consummate book in documenting Newcomen engines. However, historical thermo-fluids engineering data is often only the cylinder diameter and perhaps total pump depth and water pumping rate (often, that is in the old unit of "hogsheads"). Sometimes additional details regarding the pumps, piston and a cylinder are provided, but the book and journal article data that allows calculation of the thermodynamics of Newcomen engines is somewhat scarce.

Newcomen engine and pump general characteristics

The following discusses the general characteristics of Newcomen engines.[1,3,4,5] As represented in Fig. 1, the early cylinders were flat bottomed and had a flared portion at the top to prevent piston seal water from spilling out. Later engines had conical or hemispherical heads to better collect and channel the significant volume of liquid water sprayed and formed inside the cylinder. Early cylinders were of brass. After 1743, cast iron began to be used and essentially all cylinders were made from this material after a brief transitional period. The wall thickness of the cast iron cylinders was typically about 1 inch. In addition to the steam pipe that projected up into the cylinder, the head was fitted with a condensate drain called the "eduction pipe" that drained into the feedwater tank (FWT in Fig. 1). A "non-return valve" (often a leather flap attached at the pipe's opening) was fitted on the submerged end to prevent water from being drawn back up and into the cylinder when the vacuum was formed. The FWT (also later referred to as the "hot well") was able to be drained to the boiler to re-use this heated water.

Later engines were also typically fitted with a small pipe off the eduction pipe that ran into a shallow water-filled basin. This basin was within the view of the operator. As the steam entered the cylinder, non-condensable gases (air that came out of solution in the boiling process and cylinder air in-leakage when at vacuum conditions) would bubble out of the basin, but steam would condense. This basin typically overflowed to the hot well. This system was called the "snifter." The operator would know that the air had been flushed out by steam when the bubbles in the snifter stopped, since overflow of steam would condense and not produce bubbles in the basin.

The condensing jet sprayed up into the cylinder. The cold water tank was filled by an auxiliary pump operated by the rocking beam and was installed high in the pump house (Fig. 1). When the injection cock (valve) opened, the head of water in the elevated tank caused the spray to start. (Recall that the pressure in the cylinder at the top of the reset stroke was close to atmospheric pressure.) As the vacuum formed due to the spray, this would increase the injection rate.

The cylinder at the end of the power stroke was relatively cold from the copious injection water spray and, to a lesser extent, the drop in saturation temperature with the drop in pressure. To reset the piston to the top of its stroke, steam was admitted via the steam valve and the pressure increased back to atmospheric or just slightly above; the steam not only had to fill the void of the piston moving back to top of stroke, but it also had to heat the piston and cylinder, blow out the condensed water, and blow out the non-condensable gases from air in-leakage.

The pistons were typically flange-shaped, similar to railroad wheels, and constructed from cast iron of slightly smaller diameter than the cylinder bore. The flange shape of the piston is represented in Fig. 1. The annulus was typically sealed with a hemp rope to keep the piston centered and topped with a leather flap that slid along the cylinder walls and formed the seal. Water was added to the top of the cylinder to keep the seals moist and pliable. If the seals wore excessively, the seal water and air would enter the cylinder.

The typical boiler operating pressure was 1½ to 2½ psig. A weighted safety valve protected the boiler and limited the steam pressure available to fill the cylinder on the reset stroke. Steam admission was via a pipe in or near the bottom head of the cylinder. The boiler was built directly below the cylinder on early engines as shown in Fig. 1. Later engines and reconditioned engines incorporated improved boilers that were built off to the side. The steam was then piped to the cylinder.

The walking beam was equally portioned so that the stroke of the piston equaled the stroke of the pump rods. The stroke of these reciprocating machines was not controlled by a crankshaft and, consequently, was variable. To prevent bottoming the piston, or pulling it out of the cylinder at the top, wooden “tupping” springs (also known as “spring beams”) were typically installed as part of the support structure. Each wood arch on the rocking beam had metal rods that projected out from the sides known as “catch wings.” The catch wings were positioned to hit the tupping springs and thereby limit the piston stroke. If impacted by the catch wings, the stiffness of the tupping springs tended to transmit a large mechanical shock through the entire structure. Consequently, the engines were typically operated with a reduced stroke by the engineman to prevent the tupping springs from being impacted.

Chains, most commonly plate-and-pin type (bicycle chains are plate-and-pin type) connected piston-to-beam and pump-to-beam. Chains only work well in tension, so the steam piston was not really pushed up by the steam. Weights on the pump-side balanced the operation and pulled the piston back to its top of stroke position. This tension statically produced about 1 psi vacuum on the piston. In early, low capacity boilers, the piston reportedly could reset without a full charge of steam and, because a significant volume of air was drawn into the cylinder, the condensing jet would not form a sufficient vacuum to lift the pumps.

The pumps were generally divided into multiple lifts operating in series off the same beam arch.

The deeper pump discharged into the suction cistern of the next lift. The deeper pumps often were larger diameter to ensure excess water was pumped to the next level. This ensured the next level's pump suction did not run dry. Excess water drained back down to the lower level. If a pump lost suction or drew in air, the resulting loss of counter weighting could cause the steam piston to drop too fast and too far and the tupping springs would be impacted and cause a large mechanical shock to the machine, as discussed above.

The number of pump cycles per minute tended to be around 10 to 12, but some engines cycled at up to around 17 strokes per minute and others as low as 5 or 6 strokes per minute. The reset stroke was comparatively slow, while the power stroke was quicker. This uneven reciprocating speed challenges current thinking, which is conditioned by high speed crankshaft engines.

An engine indicator was a device that connected both to the piston's mechanism and to a pressure tap in the cylinder. The device produced a tracing of the pressure versus the piston position which could be scaled to volume inside the piston/cylinder. An indicator card was recorded in 1895 on a Newcomen engine with a cylinder of 66 inches diameter and a stroke of 6 feet. The result is shown in Fig. 4. The engine went through 10 strokes per minute and the boiler pressure was 2.3 psi. Note that the cylinder pressure recorded above the atmospheric line was less than 1 psig, so there were some flow-related pressure losses in the piping and steam valve between the boiler and the cylinder.

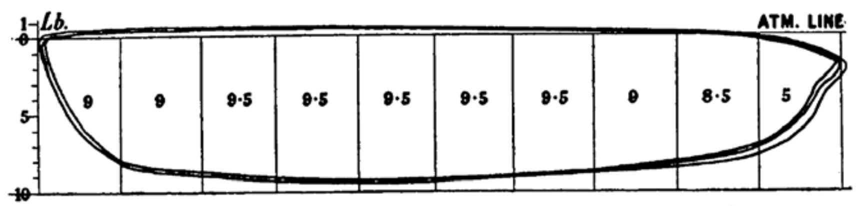

Figure 4 - Newcomen Engine Indicator Diagram[1]
{used with permission}

Fig. 4 shows what appears to be three cycles. The tracing above atmospheric is the reset portion of the cycle (with the piston rising) and the tracing below the atmospheric line is the work portion of the cycle (with the piston lowering). The numbers in the various segments indicate the average pressure in that portion of the cycle. There are ten segments. The first segment on the left clearly has a typographic error and the value is closer to 6. Using that value and determining the overall average pressure below the atmospheric line, the result is 8.5 psiv.

the Westfield engine

An article written by mining engineer, Mr. G. T. Newbould, and published in the *Transactions* of the Newcomen Society in 1934, documented a Newcomen engine with sufficient details of the engine and pump dimensions and operating characteristics to allow more advanced analyses of a Newcomen engine. Built in 1823, the engine at Westfield, Parkgate, Yorkshire, was one of the last Newcomen-type engines constructed. Since this was at a coal mine with abundant waste coal, a Newcomen engine was built rather than the more expensive and more efficient Watt engine. The cylinder bore was 54 inches in diameter and was 10 feet long. The maximum stroke, presumably limited by the tupping springs, was 7 ½ ft, but the average actual stroke was 5 feet 9 inches. The pump operated at 10 strokes per minute and pumped from a depth of about

60 yards in two lifts. The lower lift of 22 yards used a 16 inch diameter pump, and the upper lift of 37 yards used a 15 inch diameter pump. Newbould reported that the average water produced was 48 gallons per stroke. The coal consumption reported by Newbould was 18 lb per whp. This engine was dismantled in 1934, after more than 100 years of service.[5] While the engine was removed, the engine house has been preserved and may be visited to this day.

Of particular interest to the current study, Mr. Newbould indicated this engine and annotated the indicator diagram with valve timing. The timing of the steam admission and condensing spray processes is significant to any more-thorough thermodynamic analyses. Fig. 5 shows the indicator diagram as reported in the Newbould article. The direction of the tracing is counterclockwise.

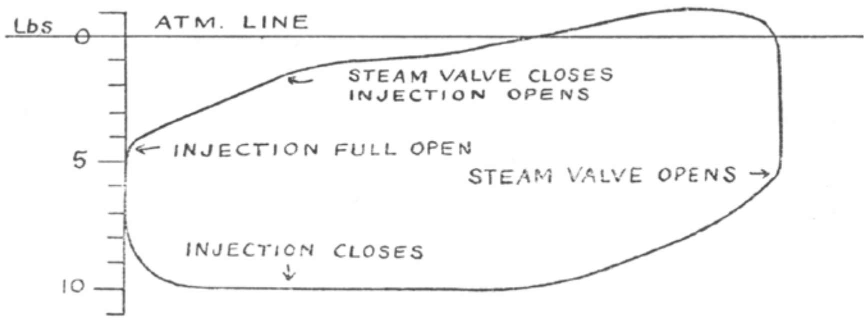

Figure 5 – Indicator Graph of the Newcomen Engine at Westfield[5]
{used with permission}

Newbould did not provide data on the boiler performance. However, the 1861 *Treatise on the Steam-Engine*…,[6] by John Bourne provides some useful operating data from John Smeaton. Between 1765 and 1772 Smeaton (1724-1792) made a systematic study of Newcomen engines and is credited with optimizing the performance. Erected in 1774, had a cylinder bore of 52 inches and a stroke of 7 feet, Smeaton's Long Benton engine was similar in size to the Westfield engine.[10] The boiler used in the Long Benton engine evaporated 90 ft^3/hr of water and consumed 8½ bushels of coal per hour.[6] Bourne also reported that the injection water drained from the cylinder tended to overflow the FWT and that the temperature in the FWT of one of Smeaton's engine was 134^{o}F.

methodology

The general model that applies to a heat engine operating a pump is shown in Fig. 6.

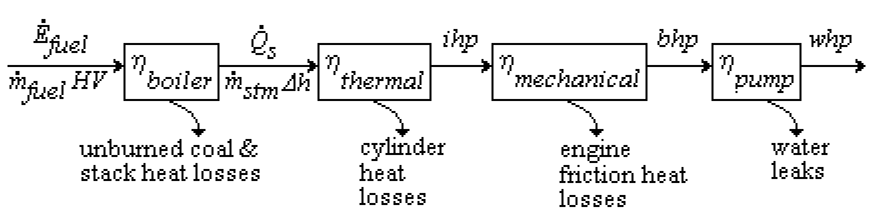

Figure 6 - Pumping Engine Efficiency Relationships

The notation convention used herein is that "dotted" terms indicate the time rate of the extensive variable. The fuel energy rate ($\dot{E}$) is the product of the mass flow rate of the fuel ($\dot{m}$) and its heating value (*HV*). The thermal power ($\dot{Q}_s$) supplied to the cylinder is the product of the mass flow rate of the steam ($\dot{m}_{stm}$) and the change in enthalpy (Δh) of the water in the boiler. The engine has a *thermodynamic efficiency* ($\eta_{thermal}$) that satisfies the Second Law of Thermodynamics and accounts for non-adiabatic heat and non-isentropic work processes in the cycle of the working fluid. The thermodynamic output of an engine on a power basis is called the indicated power, with indicated horsepower being the power units of this paper (*ihp*). The *mechanical efficiency* ($\eta_{mechanical}$) more directly accounts for the friction in the engine, beam, and pump. The mechanical power input to the pump here adopts the rotational engine analog of brake horsepower (*bhp*). For the purposes of this analysis, the pump efficiency focuses on water produced as compared to the power necessary to lift the water in the pumps. The water delivery rate allows calculation of water horsepower (*whp*). All of the efficiency blocks multiplied together produce the system overall efficiency ($\eta_{overall}$).

analysis

<u>boiler analysis</u>

Boiler efficiency is the thermal power output supplied to the engine ($\dot{Q}_s = \dot{m}_{steam}\Delta h$) vs. the chemical energy input of the fuel released through combustion ($\dot{E}_{fuel} = \dot{m}_{fuel}HV$). This is represented graphically below.

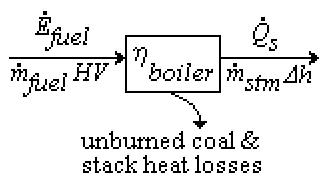

Figure 7 – Boiler Efficiency Graphical Representation

Mass flow rates of the coal and water may be calculated as follows:

$$\dot{m}_{coal} = \rho\dot{V} = \left(84\,{}^{lbm}\!/_{bushel}\right)\left(8.5\,{}^{bushels}\!/_{hr}\right) = 714\,{}^{lbm}\!/_{hr}$$

A note about British coal. The majority of coal produced in Britain was and is bituminous coal which has a heating value ranging from 10,000 to 14,000 Btu/lbm. The average heating value of coal in the UK is about 10,750 Btu/lbm.[7]

We need the density of the water entering the boiler for the mass flow rate calculation.

$$\rho_{feedwater} = \frac{1}{v_f\,(130^\circ F)} = \frac{1}{0.016247\,{}^{ft^3}\!/_{lbm}} = 61.5\,{}^{lbm}\!/_{ft^3}$$

$$\dot{m}_{water} = \rho\dot{V} = \left(61.5\,{}^{lbm}\!/_{ft^3}\right)\left(90\,{}^{ft^3}\!/_{hr}\right) = 5535\,{}^{lbm}\!/_{hr}$$

The heat supplied to the water is the change in enthalpy of the water (Δh). We can look up the enthalpy values using Steam Tables.[8] The boiler is assumed to produce dry saturated steam.

$h_{out} = h_g$ (16.7 psia) = 1152.5 Btu/lbm

$h_{in} = h_f$(130 ^{o}F) = 98.0 Btu/lbm

The boiler efficiency is:

$$\eta_{boiler} = \frac{\dot{Q}_s}{\dot{E}_{fuel}} = \frac{\dot{m}_{steam} q_s}{\dot{m}_{caal} HV} = \frac{\left(5535\,{}^{lbm}\!/\!_{hr}\right)\left(1152.5 - 98.0\,{}^{Btu}\!/\!_{lbm}\right)}{\left(714\,{}^{lbm}\!/\!_{hr}\right)\left(10{,}750\,{}^{Btu}\!/\!_{lbm}\right)} = \frac{5.84E6\,{}^{Btu}\!/\!_{hr}}{7.68E6\,{}^{Btu}\!/\!_{hr}} = 76.0\%$$

$$\boxed{\eta_{boiler} = 76.0\%}$$

We'll use this value in the summary analysis, below.

<u>pump analysis</u>

The pumps were connected to the same beam arch and both had the same stroke as the steam piston. At an average stroke of 69 inches, the 15 inch diameter upper lift pump with no leakage losses would have delivered:

$$V = \frac{\pi D^2}{4} L = \frac{\pi 15^2 in^2}{4} 69\,in = 12{,}193\,in^3 \left\{\frac{1\,ft^3}{1728\,in^3}\right\} = 7.06\ ft^3 \left\{\frac{7.48\,gal}{ft^3}\right\} = 52.8\,gal$$

The 16 inch diameter lower lift pump with no leakage or cistern overflow losses would have delivered:

$$V = \frac{\pi D^2}{4} L = \frac{\pi 16^2 in^2}{4} 69\,in = 13{,}873\,in^3 = 8.0\ ft^3 \left\{\frac{7.48\,gal}{ft^3}\right\} = 60.1\,gal$$

Considering Newbould's reported water production rate of 48 gallons per stroke, the upper lift pump is 91% effective.

The weight of water lifted was quite impressive. Recall that both pumps were connected to the same yoke and operated together. The specific weight (γ) of cold fresh water is 62.4 lbf/ft^3 and has the same numerical value as density (ρ), but the units of γ are lbf/ft^3 as compared to lbm/ft^3 for ρ. Assuming that each pump discharge pipe retained the same diameter throughout its full height, the weight of the water simultaneously lifted in both pumps was:

$$Wt = \gamma V = \left(62.4\frac{lbf}{ft^3}\right)\left[\left(\frac{\pi 16^2 in^2}{4}\right) 22\,yds \left\{\frac{3\,ft}{yd}\right\} + \left(\frac{\pi 15^2 in^2}{4}\right) 37\,yds \left\{\frac{3\,ft}{yd}\right\}\right]\left\{\frac{1\,ft^2}{144\,in^2}\right\}$$

$$Wt = 14{,}250\,lbf$$

The work of lifting the two pumps through one stroke is:

$$W = F \times d = Wt \times \Delta z = (14{,}250\, lbf)(5.75\ ft) = 81{,}939\ ft \cdot lbf$$

The power input to the pump to lift 10 strokes per minute, analogous to brake horsepower (*bhp*) in a rotational engine, is:

$$\dot{W} = W\ N = \left(81{,}939 \frac{ft \cdot lbf}{stroke}\right)\left(10 \frac{strokes}{\min}\right) = \left(819{,}390 \frac{ft \cdot lbf}{\min}\right)\left\{\frac{1\,hp}{33{,}000\ {ft \cdot lbf}/{\min}}\right\}$$

$$\dot{W} = 24.83\,bhp$$

When a pump's performance is measured in the water delivered, the power is referred to as "water horsepower" (*whp*). The power as measured by the water delivered by a reciprocating pump is calculated as follows:

$$\dot{W} = \dot{m}w = \frac{m}{t}\frac{g}{g_c}\Delta z = \left(\rho \frac{V}{t}\frac{g}{g_c}\right)(\Delta z) = \gamma \dot{V} \Delta z$$

$$\dot{W} = \left(62.4 \frac{lbm}{ft^3}\right)\left(\frac{48\,gal}{stroke}\right)\left\{\frac{1\,ft^3}{7.48\,gal}\right\}\left(\frac{10\,strokes}{\min}\right)\left\{\frac{1\,lbf}{1\,lbm}\right\}(59\ yards)\left\{3\frac{ft}{yd}\right\} = 708{,}757 \frac{ft \cdot lbf}{\min}$$

$$\dot{W} = \left(708{,}757 \frac{ft \cdot lbf}{\min}\right)\left\{\frac{1\,hp}{33{,}000\ ft \cdot lbf\ /\min}\right\} = 21.48\ whp$$

Based upon these values, the efficiency of the pump is:

$$\eta_{pump} = \frac{whp}{bhp} = \frac{21.48\,hp}{24.83\,hp} = 86.5\%$$

$$\boxed{\eta_{pump} = 86.5\%}$$

<u>cylinder and piston analysis</u>

Analyzing the indicator diagram allows us to determine the indicated power. The author photocopied and enlarged Newbould's 1918 indicator graph onto gridded paper and interpolated the pressures for 30 different piston positions.

The graph does not indicate the specific location of the piston, so the bottom of the piston was assumed to be centered in the cylinder at the center of its average stroke. As stated above the total length of the 54 inch bore diameter cylinder was 10 feet. However, the maximum stroke of the piston was cited as 7 feet 6 inches (7.5 feet). It is assumed that the tupping springs were situated such that the maximum stroke was centered in the cylinder length. Recall that the purpose of these springs was to prevent piston ejection from the cylinder and piston impact on the bottom cylinder head. Likewise, it is assumed that the engine was operated such that the listed average stroke was centered within the 7.5 foot maximum. That would result in clearance at the bottom of the average stroke of 2 feet 1 ½ inches (2.125 feet) and at the top of the average stroke of 7 feet 10 ½ inches (7.875 feet). Assuming a flat cylinder head, the piston cross-sectional area and internal volume of the cylinder at the bottom and top of the average stroke is:

$$A = \frac{\pi D^2}{4} = \frac{\pi (54in)^2}{4} = 2290\, in^2 = 15.90\ ft^2$$

$$V_{bottom\ of\ stroke} = \frac{\pi D^2}{4} \cdot L = (15.90\, ft^2)(2.125\, ft) = 33.80\, ft^3$$

$$V_{top\ of\ stroke} = \frac{\pi D^2}{4} \cdot L = (15.90\, ft^2)(7.875\, ft) = 125.25\, ft^3$$

Since the negative gage pressure is related to the net force on the piston, gage pressure was retained. The indicator graph in Fig. 8 was developed. Shifting to the convention of the larger internal volume to the right flips the indictor diagram from what was in the Newbould article.

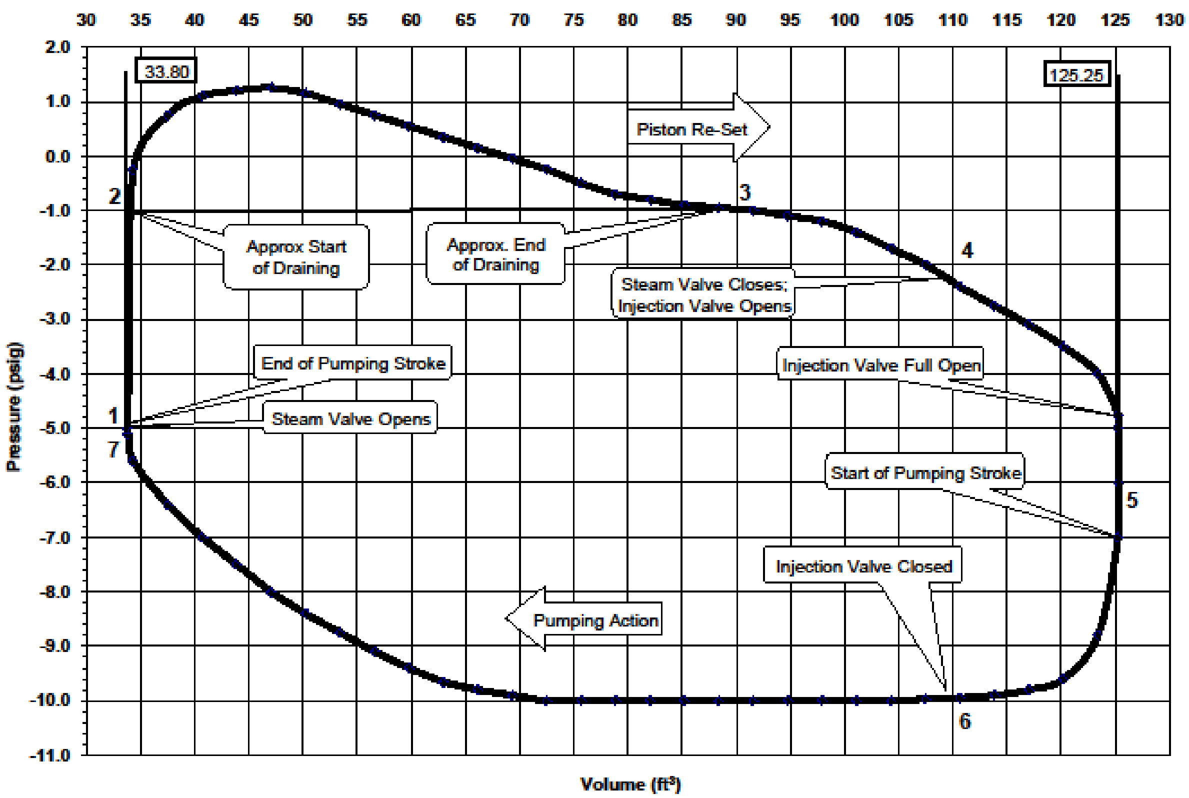

Figure 8 – Annotated Indicator Graph

Fig. 9 shows a simplified schematic series showing the piston location and valve position for each phase of the cycle. The State Points that are annotated on Fig. 8 match with the system schematics of Fig. 9.

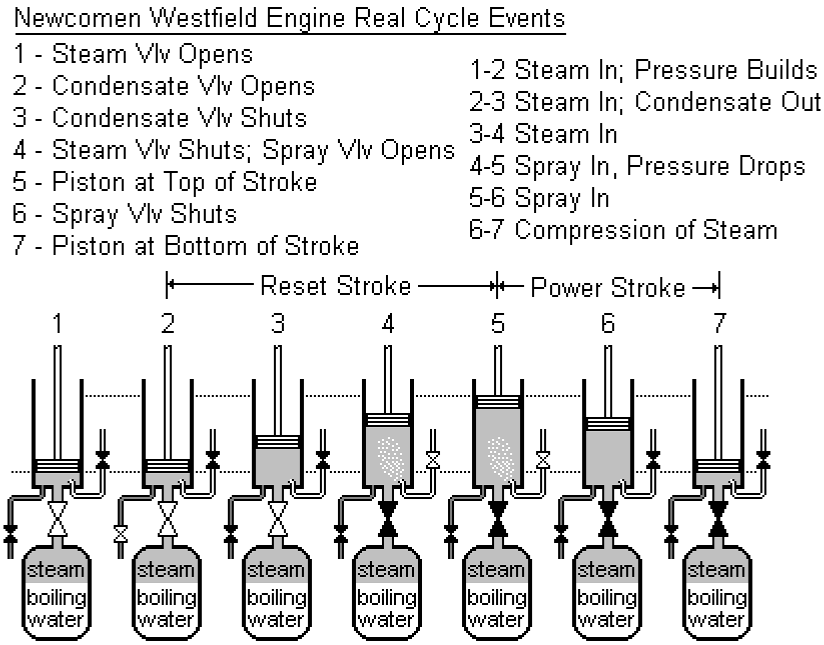

Figure 9 – Simplified System Schematics for the Westfield Engine Real Cycle

Referring back to Fig. 8, the entire enclosed area represents net work. However, it is necessary to consider the nature of the reset stroke versus the pumping stroke. While the reset stroke involves steam admission, the energy to make the stroke comes from gravity pulling on the weighted pump rod. Then, the work stroke has to overcome this additional weight. What we know about this weight balance is that it was sufficient to cause about 1 psiv in the cylinder under static conditions. Since the pump must overcome the additional weight instead of recover energy from the reset stroke, only the working stroke is used in calculating parameters in this analysis. While the enclosed area of this graph is related to the net work of the cycle, the pump work would be just the lower portion of the graph and below the balance point and the slight vacuum created by the additional weights on the pump side of the machine must be accounted for.

Newbould reported that the mean effective pressure (*MEP*) of his graph was 8.6 psig.[5] The authors performed a piecewise integration of the area using a spreadsheet analysis and adjusted the reset static pressure until the spreadsheet calculated a *MEP* of 8.60 psig. Using this calculation, the static reset pressure was determined to be -0.65 psig. This value is in the range of the reported counterweight balancing point for Newcomen engines.

Because the piston is moved up to the top of its stroke, technically work is done. Furthermore, steam is admitted and a significant portion is condensed in heating the piston and cylinder, so energy transfer is occurring during the reset stroke. However, since the energy input is not stored in a flywheel or crank, the reset process can also be left out of the analysis. The indicator graph that represents the pump work is shown in Fig. 10, with the *MEP* of the stroke superimposed.

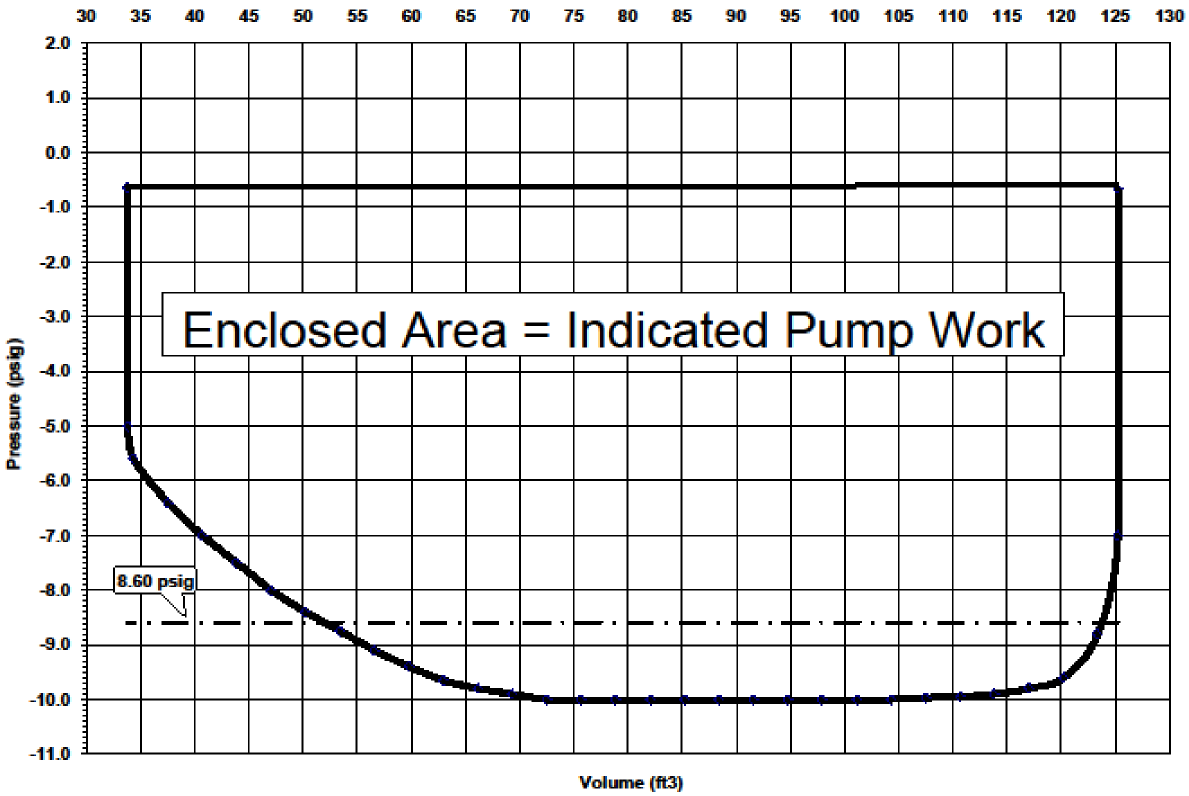

Figure 10 – Indicated Pump Work

Newbould stated that the *MEP* was 8.6 psi. It is thus possible to calculate the indicated work and power. The indicated work is:

$$W = \bar{p}\Delta V = \left(8.6\frac{lbf}{in^2}\right)\left\{\frac{144in^2}{ft^2}\right\}91.45\,ft^3$$

$$W = 113{,}251.7\ ft \cdot lbf / stroke$$

The indicated power is:

$$\dot{W} = \frac{W}{t} = \left(113{,}251.7\frac{ft \cdot lbf}{stroke}\right)\left(\frac{10strokes}{\min}\right)$$

$$\dot{W} = \left(1{,}132{,}517\frac{ft \cdot lbf}{\min}\right)\left\{\frac{1hp}{33{,}000\ ft \cdot lbf / \min}\right\}$$

$$\dot{W} = 34.32\,ihp$$

These results enable the calculation of the engine mechanical efficiency.

$$\eta_{mechanical} = \frac{bhp}{ihp} = \frac{24.83}{34.32} = 72.3\%$$

$$\boxed{\eta_{mechanical} = 72.3\%}$$

overall efficiency analysis

John Smeaton also defined the "great product" of a pumping engine based upon the number of pounds of water (in millions) raised one foot while the boiler consumed one bushel of coal. Great product evolved to the term "duty," but either way the concept facilitated comparisons between different engines.[9] Smeaton's Long Benton colliery engine produced a "duty" of 9.45 million pounds of water raised one foot per bushel of coal consumed.[10]

$$\eta = \frac{W_{out}}{E_{in}} = \frac{9.45E6\ ft \cdot lbf\ /bushel}{(84\,lbm/bushel)(10{,}750\,Btu/lbm)\{778\ ft \cdot lbf\ /Btu\}} = 0.0135 \cong 1.4\%$$

In the case of the Westfield engine, it is possible to calculate the overall efficiency ($\eta_{overall}$) of the engine from the reported hourly coal consumption of 18 lb per *whp*. The hourly coal consumption is:

$$\dot{m}_{coal} = \left(\frac{18lbm}{whp \cdot hr}\right)21.48\,whp = 386.6\frac{lbm}{hr}$$

It is important to note that the mass flow rate is also the weight flow rate in the English and USCS unit systems.

$$\eta_{overall} = \frac{\dot{W}_{out}}{\dot{E}_{in}}$$

$$\eta_{overall} = \frac{708{,}757\ {}^{ft \cdot lbf}\!/_{\min}\ \{60\ {}^{\min}\!/_{hr}\}}{\left(386.6\ {}^{lbf}\!/_{hr}\right)\left(10{,}750\ {}^{Btu}\!/_{lbf}\right)\{778\ ft \cdot lbf\ /Btu\}}$$

$$\boxed{\eta_{overall} = 0.013 = 1.3\%}$$

This calculated result is commensurate with Smeaton's improvements, so the coal consumption time unit of "per hour" is assessed to be correct.

summary analysis

At this point, we know the overall efficiency (1.3%), the boiler efficiency (76.0%), the pump efficiency (86.5%), and the mechanical efficiency of the engine (72.3%). We can now determine the thermal efficiency of the Newcomen engine.

$$\eta_{boiler}\,\eta_{thermal}\,\eta_{mechanical}\,\eta_{pump} = \eta_{overall} \Rightarrow \eta_{thermal} = \eta_{oveall} \big/ \eta_{boiler}\,\eta_{mechanical}\,\eta_{pump}$$

$$\eta_{thermal} = 0.013 \big/ (0.760)(0.723)(0.865) = 0.027 = 2.7\%$$

$$\boxed{\eta_{thermal} = 2.7\%}$$

As can be seen from the above analyses, the boiler, mechanical, and pump efficiencies are all an order of magnitude higher than the thermal efficiency of the Newcomen engine. Contrasting this to the thermal efficiency of a modern Otto Cycle internal combustion engine with a compression ratio of 8 at about 56% really places the thermal efficiency in context. While we used data for a boiler from an earlier time than the Westfield engine, even if the boiler efficiency for the Westfield engine were 100%, the thermal efficiency would then be about 2.1%. The thermal efficiency was the main case of the low overall efficiency of these engines.

conclusions

This article describes the operation and provides construction and archival operating data for the Newcomen steam engine. Analyses using modern calculation techniques are presented. The overall efficiency of the engine is presented. A variety of calculation techniques are illustrated and span the topics of thermodynamics and fluid mechanics. The piston and cylinder engines discussed in this article, by modern standards, used very low pressure steam, and were stationary engines. Improving the component efficiencies all contributed to the dramatic increase in overall efficiency of engine technology over the ensuing years, but the most fruitful increase was made possible by improving the thermal efficiency.

The year 2012 marks the 300th anniversary of the Newcomen pumping engine, the first truly successful steam engine. The Newcomen engines were used for pumping water from mines, pumping water to run mill water wheels, and town water supplies. Engines directly driving a rotating shaft came later in the progression. The total operational period of this technology was about 220 years: an impressive record.

It is the hope of the authors that today's engineering students be enriched by being exposed to the legacy left by our early engineers and that our students develop their engineering analytical skills by practicing the types of calculations illustrated in this article. The author has created homework and in-class exercise problems for use in teaching thermo-fluids courses. These problems can be used at the appropriate point in the syllabus and provide a historical context for the technology while practicing modern engineering analyses, only requiring minimal additional class time to discuss the historical perspectives.

postscript

As inspection of the list of references shows, many of the references used in this paper are very old and may not be on the library shelves. However, they are available through the "Google Books" project. The authors recommend that readers investigate this outstanding source for these and additional resources.

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