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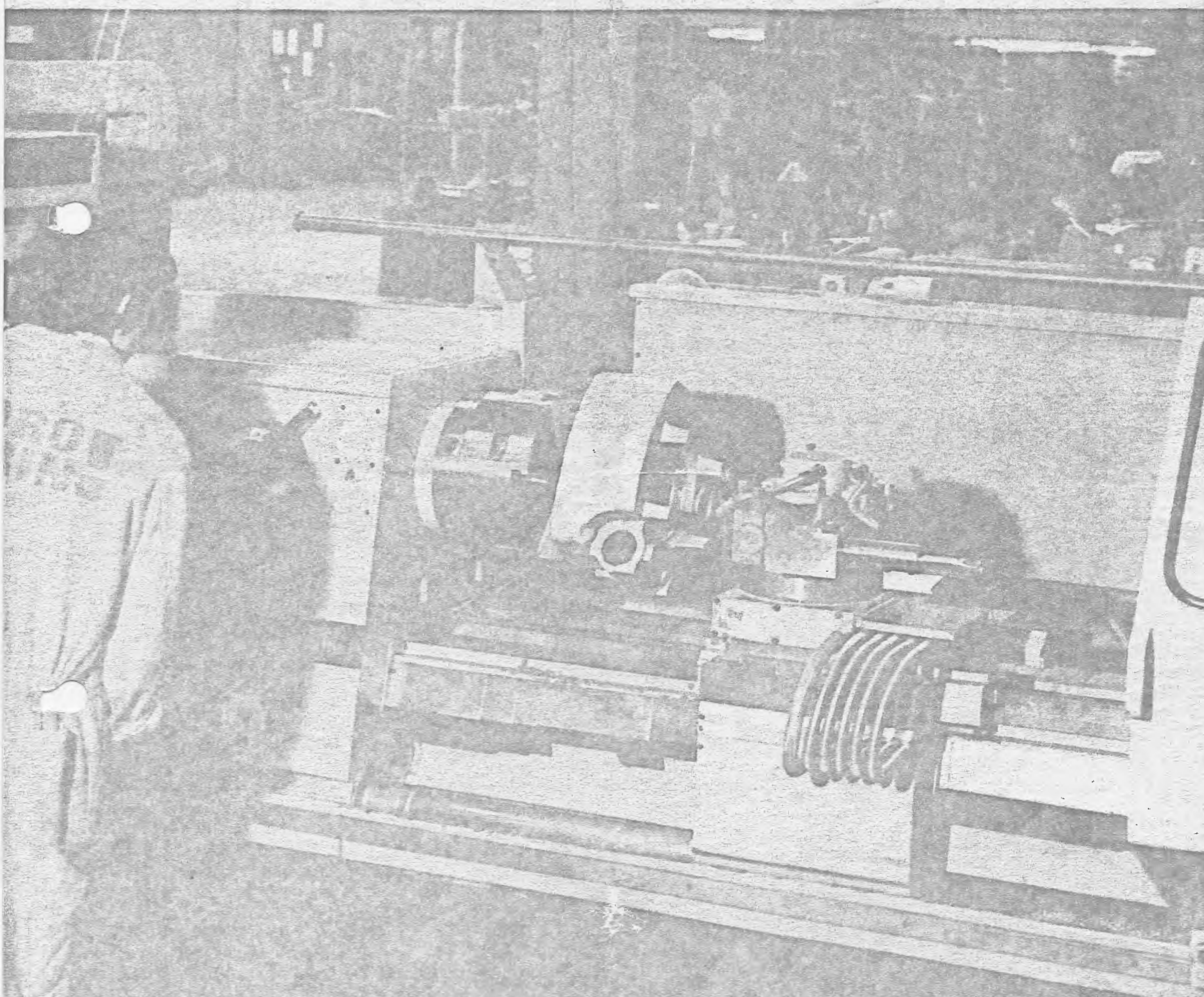


THIS MONTH'S FEATURES

Computer control systems for machine tools ■
An appreciation of the triple expansion engine ■
Plain thrust bearings – their selection and design ■

An appreciation of the triple expansion engine

by G E Tubman

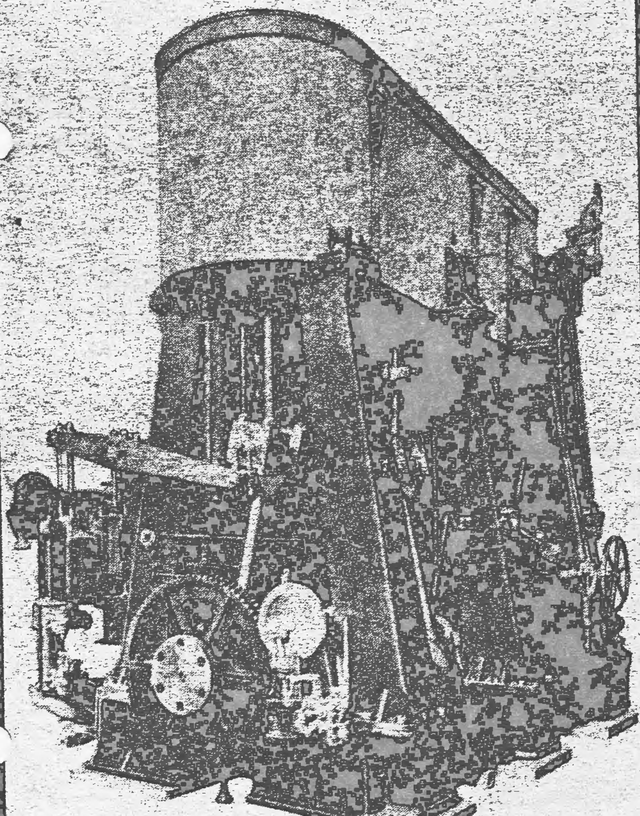


Fig 1. This fine example of a marine triple expansion engine is one of hundreds built for the US Maritime Commission during World War 2 for powering the war-time Fort and Liberty ships. This model EC2/S/C1 developed 2500 ihp and propelled ships of 10 000 ton displacement at 12 knots. The cross-section of Fig 2 and the indicator diagrams of Fig 4 are of the same engine. Engines of this type continued to give excellent service for many years after the war.

An engine, the basic design of which qualifies it for a place in engineering history as a major prime mover on land and sea for over 70 years, must have possessed some remarkable qualities. This article, while evoking memories in older engineers may also interest the younger generation who may be curious to know just what the 'triple's' virtues were.

The article does not indulge in nostalgia but contains a more practical purpose—it suggests that a fresh look at this outstanding and reliable engine—against a background of high maintenance costs, fuel shortage and pollution problems —might prove beneficial within the concept of total energy. The author wonders whether a re-design of those technological limitations which caused its demise may in the light of present-day technological advances now be justified.

They ran like sewing machines

Most young engineers have probably never seen a triple expansion steam engine operating, although older engineers will probably have fond memories of the 'old up and downers' as they were called. The historical background of the triple expansion engine can be traced back to Watt's double-acting condensing steam engine of the late 1790's—the thermal efficiency of which was about 2 per cent. Very low compared with present-day standards but in those days, great difficulties were experienced in boring cylinders true and maintaining pistons steam-tight.

Technological advance during the Industrial Revolution demanded intensive development of steam power and by the 1890's the steam engine had developed into the triple expansion engine and was firmly established as the major prime mover on both land and sea. During the next 50 years of operational service steam pressures increased to 250 lbf/in² (1723 kN/m²), superheat was introduced, triple developed into quadruple expansion (four cylinders) and engines were designed to exhaust into a low-pressure turbine, hydraulically coupled to the output shaft.

For many years the up and downer propelled the merchant shipping fleets of the world, at a steady 10 to 12 knots, running as marine engineers used to say "like sewing machines". Typical thermal efficiencies of around 20 per cent were attained on engines operating on superheated steam at 315°C (600°F).

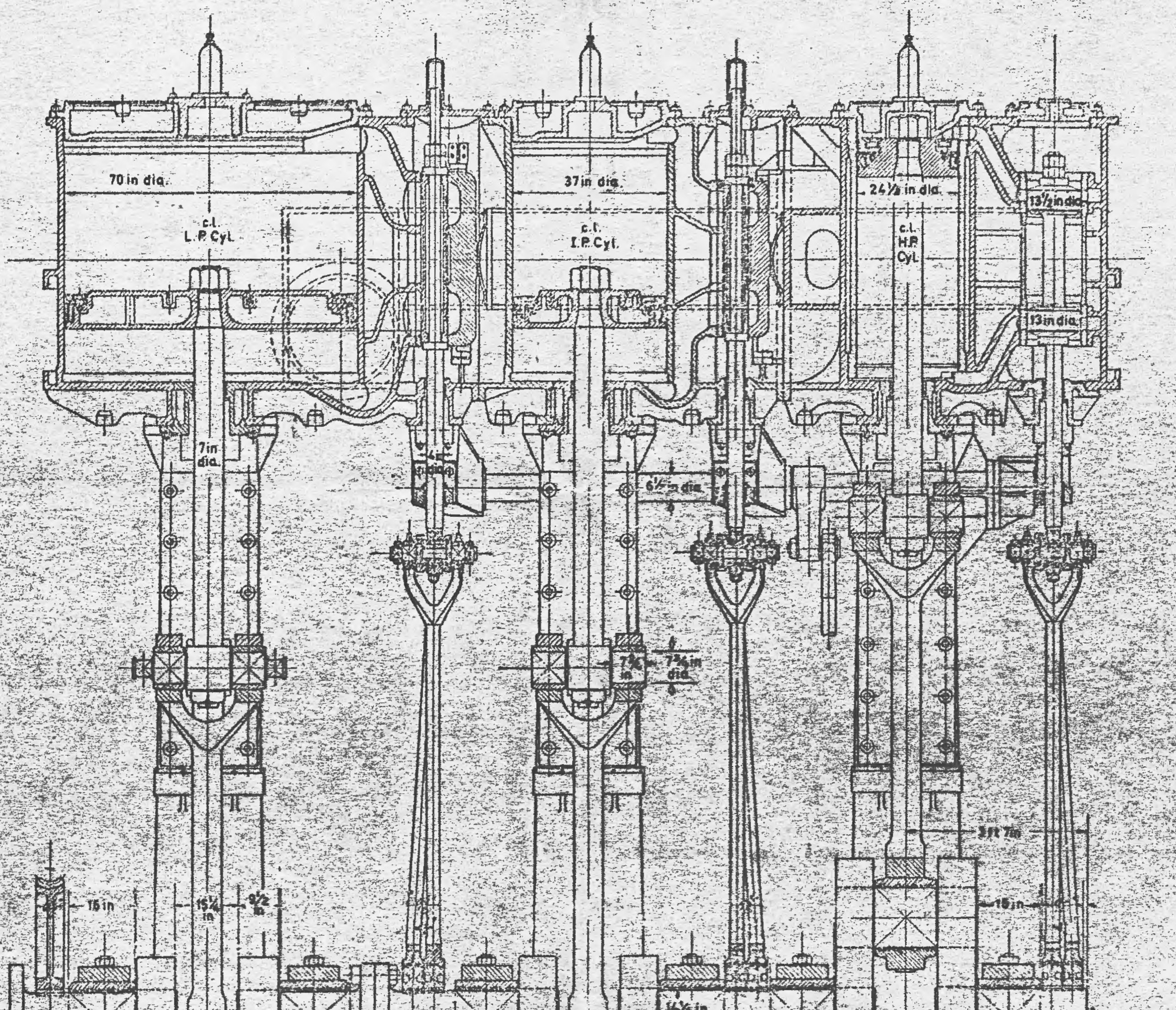
However, although at the end of World War 2 a large proportion of the Allied merchant fleets were propelled by the up and downer, in the face of post-war competition from the steam turbine and diesel engine, this fine engine faded away.

A cross-section of a typical wartime marine engine is shown in Fig 2. This engine had a stroke of 48 in (1·219 m) with cylinder dia of 25 in (0·635 m), 37 in (0·94 m) and 70 in (1·778 m) respectively.

Construction

Cast iron was used extensively in its construction, each cylinder being individually cast and bolted together to form a cylinder block, supported from the cast bedplate by box section columns. The three back columns were provided with water-cooled crosshead guides to take ahead and astern piston and connecting rod thrusts. To cope with alternating stresses piston and connecting rods were machined from 0·2 per cent

Fig 2. Cross-section of a typical triple expansion steam engine fitted with Stephenson's link motion reversing gear.



carbon steel, and the crankshaft from forged steel with a tensile strength of 35 tonf/in² (540 MN/m²). Main and bottom end bearings were of Babbitt white metal (83·3 per cent Sn, 11·1 per cent Sb, 5·6 per cent Cu) often run in after scraping, on pure castor oil to obtain a hard surface skin. Usually the crosshead bearings were machined from gun metal (90 per cent Cu, 10 per cent Sn) running on mild steel pins.

A lever motion connected to each side of the low-pressure cylinder crosshead bearings actuated a reciprocating motion to drive the air and bilge pumps. It was possible, with a little ingenuity, to construct a 'Heath Robinson' washing machine driven from this motion, to pound one's overalls at a steady 75 rev/min. Engine lubrication by gravity feed oil boxes was augmented by a regular application of the hand oilcan.

This type of engine would weigh about 150 ton (152·4 t) and develop some 2500 ihp (1858 kW) at 75 rev/min from steam at 220 lbf/in² (1517 kN/m²) at 232°C.

Fig 3. Mollier chart for an engine taking steam in the superheated region to exhaust in the wet condition, ie below the saturation line. The chart is plotted on enthalpy and entropy coordinates—the former being total heat per kg, the latter, the ratio of heat taken up to its absolute temperature.

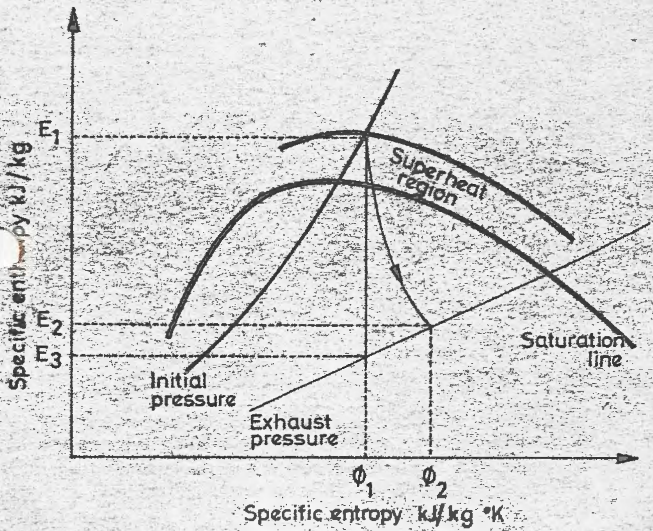
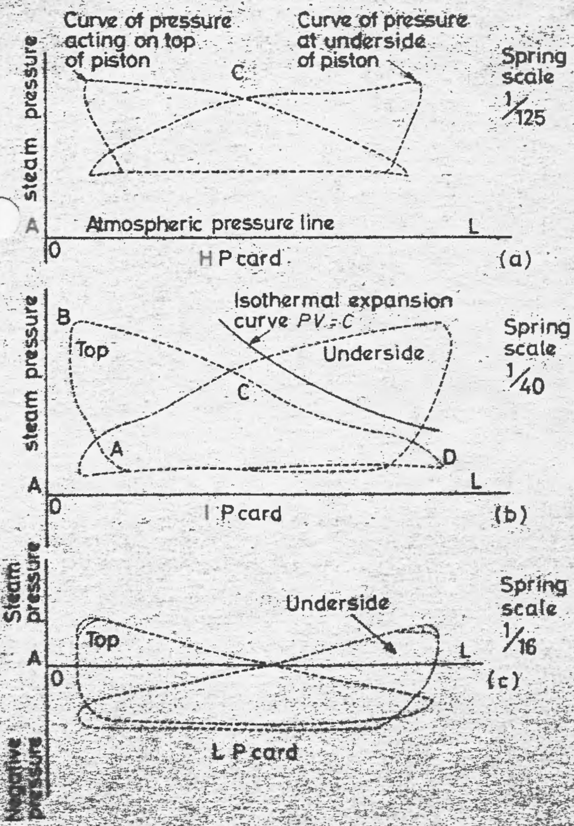


Fig 4. Indicator diagrams as produced by the Dobbie McInnes steam engine indicator. (a) hp cylinder, (b) ip cylinder, (c) lp cylinder.



A true heat engine

Was the up-and-downer a true heat engine? The answer is yes, provided you refer to the engine only. Certainly it fulfills the criteria of the heat engine in that it receives and rejects heat, converting a proportion to mechanical work in the process. Reference to the Mollier chart (Fig 3) will show that expansion takes place with an increase in entropy, and that this increase is less in the superheated region. The ideal expansion —never reached in practice—would be adiabatic, ie without the loss or addition of heat along the line of constant entropy.

From the chart the maximum heat drop available for work (ΔH_1) is $E_1 - E_3$ but the actual heat drop converted to work (ΔH_2) is $E_1 - E_2$; thus the thermal efficiency $\eta = \frac{\Delta H_2}{\Delta H_1}$

As 3600 kJ = 1 kWh, the engine steam rate is $\frac{3600}{(E_1 - E_3)\eta}$ kg/kWh. From a well maintained engine one could expect a steam rate of 8·52 kg/kWh (14 lb/hp/h).

The chart shows that expansion approaches nearer the ideal in the superheat region. The engines' upper limitation was its inability to use steam with a high degree of superheat ie at pressures above 250 lbf/in² (1723 kN/m²).

If we look at present-day engineering achievements—bearing in mind that steam engine development over the past 20 years has been negligible compared with the diesel engine—an elevation of steam pressure and superheat temperature would now be possible. However it would be necessary to provide six cylinders to develop the complete range of expansion, a not insuperable task in the light of present-day technology.

The expression 'triple expansion' is really a misnomer, for although steam expansion took place in 3 cylinders the actual number of expansions depended upon the steam inlet pressure divided by the terminal pressure, both expressed in absolute units.

Although not strictly true, expansion is usually assumed to be isothermal ie pressure × volume = constant. It follows that in the case of this engine, there is a large increase in specific volume when terminal pressure is reached in the low-pressure (lp) cylinder. The difficulty of providing lp cylinder volume and exhaust port area sufficient to pass high specific volume steam to the condenser, imposes an inherent limitation on the lp side of the cycle.

In order to avoid undercooling of the condensate, the back pressure carried by the up and downer never exceeded 26 in Hg.

Engine tuning

An important diagnostic instrument used by those who tended the health of the up and downer was the Dobbie McInnes steam engine indicator. The indicator diagram plotted by the instrument showed the variation of steam pressure in the cylinder throughout the stroke cycle. Fig 4 shows a set of indicator cards taken from the triple expansion engine propelling a vessel on a voyage from South America to London in 1947. The total indicated hp was 1735 and with the 16·5 ft pitch propeller turning at 63·5 rpm the 7800 ton vessel was steaming at 11·4 knots.

Let us look at a card in more detail. As the engine is double-acting each diagram shows the pressure curve for both above, and on the underside of the piston. Taking the curve for pressure above the cylinder on the lp card in Fig 4 as an example, the first part of the diagram to be drawn is the atmospheric line A-L and the variations in pressure that follow are then traced as follows: at A pressure rises with compression of the trapped cushioning steam, to point B where steam from the hp cylinder exhaust is admitted. Admission continues to cut-off point C where expansion with the consequent drop in pressure (and temperature) continues to point D—the point of release. The cycle is completed with the exhausting of the steam to the lp cylinder inlet ie from D to A.

The mean pressure throughout the stroke was calculated by planimetering the area of the curves for above and below the piston: the total mean area for each curve was then divided by the diagram's length in order to establish the mean height: this figure multiplied by the spring scale, gave the mep (mean effective pressure) in lbf/in². The ihp (indicated horse power) for the cylinder was calculated from the well-known formula:

$$\text{ihp} = \frac{P_m LAN}{33\,000}$$

where: P_m = mean effective pressure in lbf/in²
L = length of stroke in ft
A = area of piston in in²
N = number of power strokes per minute

An interesting feature of the hp diagram of Fig 4 illustrates one of the diagnostic features of the indicator card. A study of the diagram will reveal a rise in the expansion curve after cut-off at point C. This rise of pressure is due to a leak in the hp piston valve occurring after cut-off has taken place.

Balancing the engine consisted of checking that all cylinders developed approximately equal power (and therefore equal torque). This was achieved by adjusting the expansion link attached to the Stephenson's link motion gear (Fig 5) in an operation known as 'linking-up'. The power of the whole engine was regulated by the adjustment of the hp valve expansion link. The action of opening out the link brought the ahead eccentric rod more to the vertical position in relation to the valve operating rod; thus valve travel was increased resulting in a later cut-off and allowing more steam to flow through the engine. Conversely, closing in the link reduced power output.

Linking-up adjustments to balance the power output of the three individual cylinders required some thought and reasoning. For example, to increase the power output of the intermediate cylinder, the ip link was shut resulting in an earlier steam cut-off thus taking less steam from the hp cylinder exhaust. This had the effect of increasing back pressure on the hp cylinder and reducing its power output; but increased hp back pressure also meant an increase in steam pressure to the ip cylinder resulting in the latter cylinder developing more power.

Before leaving the subject of indicator diagrams it is interesting to consider briefly the ideal—but theoretical—expansion curve. After the cut-off point potential energy is converted into work by the expansion of steam in the cylinder driving the piston to the point of exhaust opening. In theory, expansion is isothermal following Boyle's Law, ie pressure × volume = constant. The expansion curve is hyperbolic in shape. If, for simplicity, the cylinder clearance volume is neglected, the theoretical mep is:

$$P_m = \left[\frac{P}{r}(l + \log_e r) - pb\right]$$

where: P_m = mean effective pressure lbf/in² (N/m²)
P = initial steam pressure lbf/in² absolute (N/m²)
r = ratio of expansion = $\frac{\text{final volume}}{\text{initial volume}}$
pb = back pressure lbf/in² absolute (N/m²)

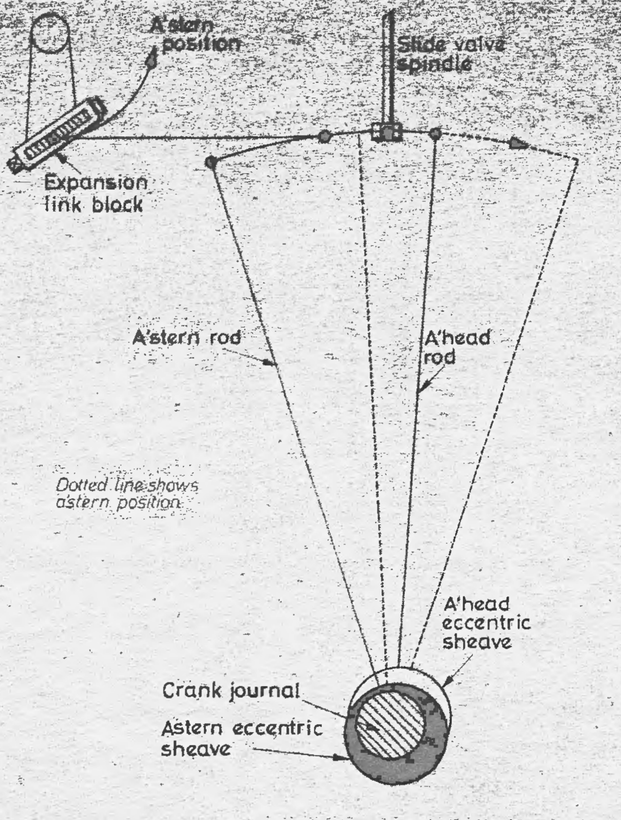


Fig 5. Stephenson's link motion. Engine output was varied by adjusting the expansion link, so increasing or decreasing the piston valve cut-off point. The volume of steam allowed to flow through the engines was therefore controlled.

Isothermal expansion was not obtained in practice, as Fig 4 shows, because steam temperature was not constant throughout the stroke and valve gear motion was not instantaneous. To compensate for these imperfections the theoretical calculations were multiplied by a 'diagram factor' of 0·7 to obtain the actual mep.

Why they ran so smoothly

There is no doubt why the up and downer ran so well; it was the outcome of 100 years of steam engineering evolution, based upon empirical knowledge and backed by sound engineering theory. The engine was robustly constructed, slow running but with high torque output and designed with extravagant factors of safety. It was responsive to load variation without undue vibration or critical speed limitation. Reversing was certain and full power in either direction could be obtained. Maintenance demand and downtime were low because it was possible to run some 20 000 h between major overhauls and there were very few expendable components.

As an example of sound basic engineering design it has had few rivals. The ease with which the three cylinder dia were determined will illustrate this.

At the planning stage the initial and final steam pressures were usually known, as were the ihp to be achieved at the proposed piston speed. An indicator diagram factor was decided upon and the point of steam cut-off in the hp cylinder, defined. The first step then, was the determination of the engines' referred mean pressure (rmp).

The rmp is the steam pressure which, acting alone on the lp piston, would develop the total engine power in the lp cylinder. Rmp's were usually designed to the order of 35 lbf/in² (241 × 10³ N/m²) and it follows that the design of the lp cylinder had to be of a size and strength sufficient to accommodate the volume of steam after expansion.

The lp cylinder dia (D in) is extracted from:

$$\text{ihp} = \frac{RMP.LAN}{33\,000}$$

where: ihp = indicated horse power
RMP = referred mean pressure (lbf/in^2)
LN = piston speed (ft/min)
A = area of low-pressure cylinder (in^2)

$$A = \frac{\pi}{4} D^2$$

As the initial and final steam pressures are known, their specific volumes are known.

The hp cylinder dia (d in) is extracted as follows:

Ratio of expansion = r

$$r = \frac{\text{final volume}}{\text{initial volume}}$$

$$r = \frac{D^2}{d^2 \times \text{cut-off}}$$

where: d^2 = hp cylinder dia (in^2)
D^2 = lp cylinder dia (in^2)

The relationship between cylinder dia is based upon proved empirical knowledge:

$$\frac{d}{x} = \frac{x}{D}$$

where: x = intermediate cylinder dia in inches

The three cylinder dia are now established.

Retrospection

The way of life of a society depends upon its ability to provide power sufficient to support its needs; this is particularly pertinent today.

Looking back, the triple expansion engine fulfilled the needs of the society that produced it. Indeed 'the triple' became a way of life in itself. For example, take the bulk transportation of goods and materials by sea in the 'thirties. Cargo and tramp steamers engined, mainly, by the up and downer, travelled at a steady 10 to 11 knots and the commercial markets of the world were geared to the speed these vessels were able to maintain. Fuel consumption varies as the square of the speed for any distance, so it was possible to propel a relatively large tramp steamer, full of cargo, at 10 knots for a reasonable consumption of fuel. It was a 10-knot world that suited the pace of society.

Present-day society makes speed demands upon sea transport in the order of 20 knots and above, with the attendant large increase in fuel consumption.

Rebirth via modern technology?

As a direct result of closing the Suez canal in 1956 cargo-carrying capacity has increased dramatically so that ships of 200 000 ton deadweight are now commonplace.

During this period, development of the triple expansion engine has been neglected; the full weight of engine R & D has been concentrated in the diesel and steam turbine it being thought—unwisely in my opinion—that the triple was not capable of further development. But today the needs of society are undergoing yet another change resulting from the energy crisis. Surely, in the light of present-day engineering technology, the time is right to consider whether further development of the triple expansion engine is now possible. It would not be the first time that an early, basically sound, engineering concept had been resurrected to meet new-found needs; the Stirling engine, revived after 100 years, is one outstanding case.

Today for example, it is possible that the former upper limitations of pressure and temperature of the triple, could be raised, as a consequence of which six cylinders may be necessary to complete the full range of expansion.

The valve motion could be improved to make cut-off instantaneous; this would prevent wire-drawing of the cylinder steam and so increase thermal efficiency.

An improvement in the weight bhp ratio could result from designing the engine to modern, realistic, safety factors. It could be made totally enclosed and self-lubricated; in this form it may fulfill a need of society in the total energy or standby-generation fields.

Finally it could be argued that as the triple is a true heat engine, it must always be possible to develop the cycle to conform more closely to the Carnot ideal.

It is therefore hoped that the suggestions put forward in this article may at least stimulate thought that may lead to further development of the 'triple', although society's needs will probably be the arbiter as to what is possible.

G E Tubman *CEng FIMarE Member, is Chief Engineer of the West Hertfordshire Main Drainage Authority. He received his technical education at Walton Technical College, Liverpool and at the City of Liverpool College of Technology. After a five-year indentured engineering apprenticeship at Cammell Laird & Co Shipbuilders, he served eight years as an Engineer Officer in the Merchant Navy. He writes with first-hand knowledge of the triple expansion engine, having world-wide Service experience in both steam-reciprocating and turbine-engined vessels. On leaving the MN in 1950 he worked first in the power engineering field and later in port engineering and dredging. He took up his present appointment in 1969, where his interest is concerned with the re-cycling of waste products and the exploitation of the total energy concept.*