MEASURING STEAM ENGINE PERFORMANCE

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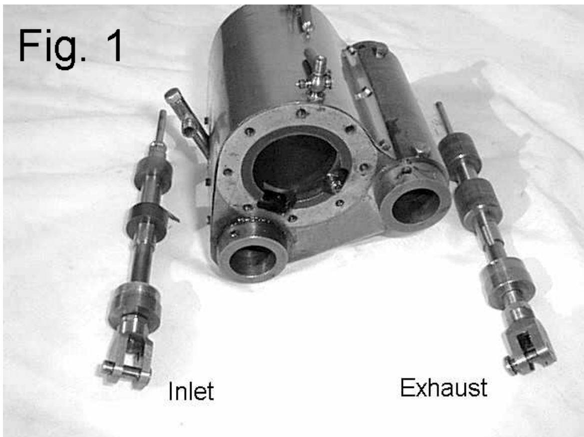
Round about the time the SM&EE was formed there was a determined effort on the part of the Engineering profession to provide a sound basis for steam engine development. In 1896 the Institution of Mechanical Engineers established a Steam Engine Research Committee, and its First Report was submitted in 1905 [1]. Cylinder condensation, and attempts to reduce it by steam-jacketing the cylinder, was one of the major concerns of the Committee and indeed this topic exercised many of the leading engineers of the time. One Fellow of the Royal Society (John Perry) accused another (Osborne Reynolds) of devoting himself to “isolated problems” (like inventing the Reynolds Number!) “having only an indirect bearing on steam engine practice” [2]. Earlier measurements had shown that much more steam was consumed than could be accounted for in terms of the swept volume (to cut-off) and engine speed; but heat losses from even an un-jacketted cylinder could account for no more than a small fraction of the latent heat of condensation of this additional steam. Perry and others realised that the initial condensation was followed by re-evaporation as the pressure in the cylinder fell, but the details of this process remained a matter of contention. Superheat eventually solved the practical problem by preventing condensation; however, no satisfactory description of the detailed mechanism of cylinder condensation was achieved.

Those of us who make and operate small steam engines are well aware that condensation becomes progressively worse as the engine gets smaller - particularly if the steam is not superheated. Further investigation of the circumstances and possible remedies therefore seemed appropriate. The following account describes the equipment used and some of the early results. The effectiveness of two specific remedies was tested - superheat, and the use of separate inlet and exhaust valves.

The Engine and Boiler

Two quantities, not usually measured directly on small engines, formed the basis of the experiments: firstly, a direct measurement of ***steam flow rate*,** and secondly ***Indicated Power*** (in addition to Brake Power). In the early stages I measured steam flow rate with a sonic nozzle (which requires only a single pressure measurement upstream of the nozzle), but in order to achieve sonic flow this meant that the boiler pressure had to be about twice the pressure at the engine, and although it was effective it was ditched in favour of a larger (subsonic) nozzle and a mercury manometer. (It is quite fascinating to observe the continuous change in steam flow at constant load as the engine warms up, or as superheat is increased - something that is not possible if one has to collect and weigh messy condensate to determine the steam flow rate!). The importance of Indicated Power is that it bears directly on the thermodynamic performance of the engine and is not affected by that very variable quantity - friction. It is determined from an indicator diagram which I record with a digital indicator.[3] The measurement of Indicated and Brake power allows the friction to be evaluated.



Prior to the use of superheat, the common methods of reducing cylinder condensation were steam jackets and the use of separate inlet and exhaust valves. Steam jackets did not prove particularly effective, and are decidedly difficult in small size engines. I decided to modify an old 5” gauge locomotive cylinder so that either separate valves or a conventional piston valve could be used. Quite apart from the effect on condensation, separate valves allow one to change inlet and exhaust events independently. Thus as one notches up one can avoid the deleterious effects of premature exhaust and too much compression. I use a ‘half link’ version of Walschaerts gear (only half an expansion link, since I don’t need to reverse the engine) to drive the inlet valve and a simple eccentric to drive the exhaust valve. It is possible to vary the angle of the exhaust eccentric with respect to the crank, although I have found one setting to be satisfactory over a wide range of running conditions. The crankshaft carries a conventional brake at one end and the slotted disc of the digital indicator at the other.

The bronze cylinder has a cast iron liner, 1.27” bore and 2” stroke, and the piston is fitted with two rings. The conventional piston valve (inside admission) and the separate exhaust valve (inside exhaust) are shown in Fig.1 . The two arrangements are:

(i) Single valve with conventional bobbins for inlet and exhaust (inside admission). In this configuration the separate exhaust valve is disconnected from its eccentric and positioned so that the additional exhaust ports are blocked.

(ii) Extended bobbins are fitted to block the exhaust ports of the conventional piston valve. The separate exhaust valve operates over the additional exhaust ports (inside exhaust)

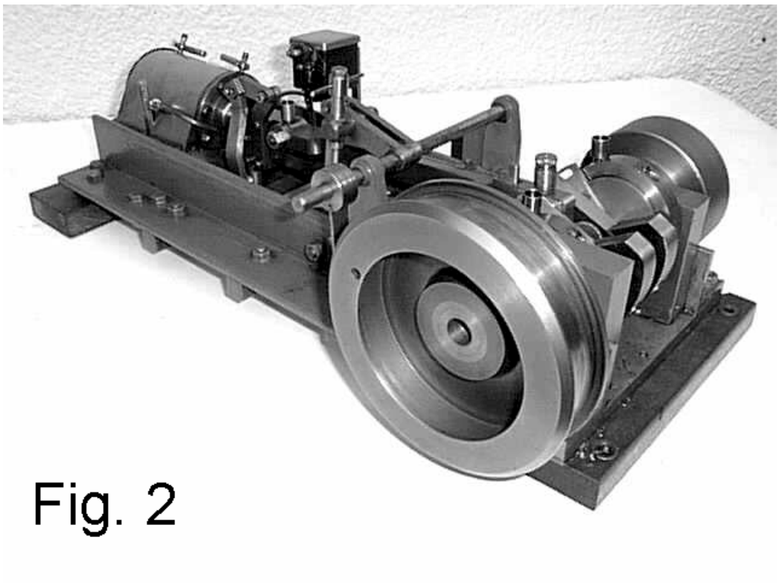


Fig. 2

These arrangements have proved very satisfactory for demonstrating the effect of separate ports. The valves are cast iron, lapped into cast iron sleeves. ‘Floursint’ valves have also been tested, with marginal reductions in leakage. These were in the form of thin sleeves mounted on stainless bobbins fitted with ‘O’ rings to load the Fluorsint on to the port sleeve. Perhaps I overdid the load applied by the ‘O’ rings, because the friction in operation is greater than that with the cast iron bobbins! The engine has proved effective and reliable in spite of its rather inelegant appearance - Fig.2.

The boiler is propane fired and has an extract fan mounted outside the workshop. I can usually complete around 20 runs before low water forces me to stop and use the injector. Superheat is by means of a coil of copper tube inside which a radiant electric heater is placed. Steam flows are usually in the range of 0 to 20 lb/hr, although the boiler would probably cope with 30 - 40 lb/hr. So far I have limited the boiler pressure to around 70 psig during tests, but its maximum working pressure is 120 psig. The complete setup is shown in Fig.3.

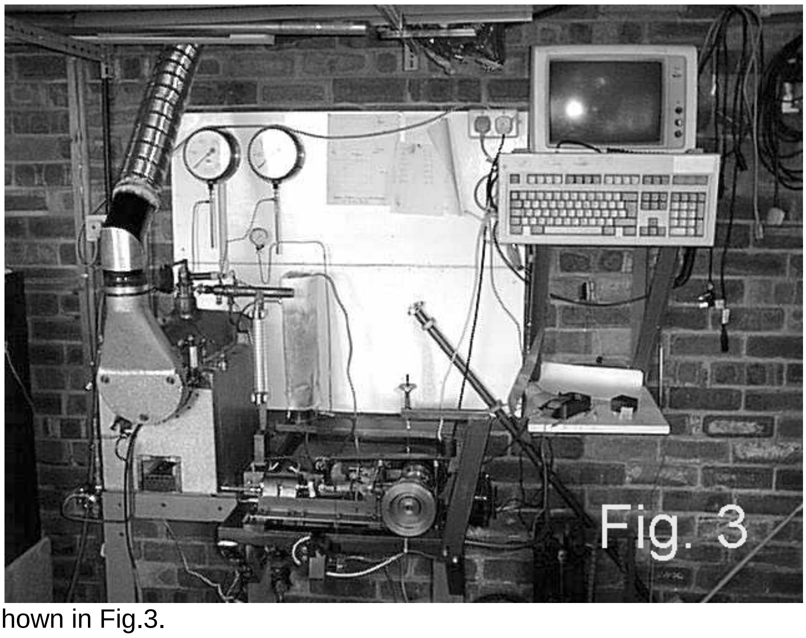


Fig. 3

Instrumentation

During the academic half of my career I had the benefit of help from research students who became much more proficient in electronics and computing than I; here was an opportunity to catch up! The problem of maintaining steady operating conditions, taking readings of steam temperatures and pressures, manometer readings, brake load, engine speed, and oiling round and keeping an eye on the water gauge clearly demanded some automation. I have now reached the state where the computer (an old PC AT286) does everything except read the steam flow manometer and the pressure gauge! I can complete runs at the rate of 30 or so in an hour, although the rate is usually determined by the rather slow response of the superheater and the need to allow the engine to settle down at a new setting. (Back in the 1890s Professor Capper, who carried out the Steam Engine Research Committee's experiments, found that he could manage only two runs a day with the help of eight students; but of course he had no computer!).

Before starting a run the computer continuously displays readings of engine speed, brake load, and steam temperature. Given the signal to start a run it waits for the next Top Dead Centre, and then measures the cylinder pressure every 2° for one revolution of the crank and stores these data in memory. It also averages the brake load over one revolution, and measures engine speed and steam inlet and exhaust temperatures. All these readings are digitised and fed into the computer using a device [4] which plugs into the printer port. Timing pulses from the digital indicator are also fed into the printer port, and a digital output is used by the computer to enable the gate which detects the next TDC event when starting a run. When a key on the computer is pressed it records and displays the indicator diagram and requests input of the steam pressure and the manometer reading. The workings of the digital indicator have already been described [3]; the current version has been adapted for use with the ubiquitous PC rather than the BBC-B computer.

The availability of a computer that can 'talk' to the hardware has revolutionised the process of analyzing experimental results! Gone are the hours of sweat with sliderule, paper and pencil. The computer has the data, so let it do the analysis before it reports back! My workshop computer, slow though it is, has finished working out the results and has stored them on disc before I am ready to start the next set of measurements. All that remains is to transfer the disc to a more modern computer and print out a fancy diagrams and graphs such as those shown in Figs.4, 5, 6 & 7. In all fairness I must admit that a good deal of midnight oil went into the writing of the computer programme!

The original brake consisted of a rope wound round the brake drum with a weight hung on the end. This has now been replaced by a device in which the load is measured by a beam (actually a 3" length of an old steel rule) fitted with strain gauges. The output of the strain gauge bridge is amplified and fed into the computer via the same device[4] that records temperature and the indicator diagram; all that is necessary is to apply a load (via a knurled wheel) so as to regulate speed, and leave the computer to measure and record the load.

A word about the mercury manometer might be in order. I started off with a U-tube design, using suitably strong glass capillary tube, but even when I managed to get rid of the air bubbles any slight heavy-handedness with the steam control valve blew the mercury out. Fortunately my friend Tom Jones suggested a single tube dipping into a mercury reservoir - a splendid solution in that one can now get rid of air or water bubbles by blowing the lot down into the mercury reservoir, whereupon the air or water floats to the top of the reservoir. I calibrated the flow nozzle by building a condenser and weighing the condensate; as a matter of interest I found the discharge coefficient of the nozzle, which was 1/8" dia. and which had a well rounded inlet, to be 1.0 ± 0.05.

Data Analysis

Several hundred sets of data have been recorded. Many of these were in the 'shakedown' period during which bugs in the computer programme or hardware failures required modifications. However, the setup is now pretty reliable, and I am satisfied with the first performance data. These comprise tests in which the performance with a conventional valve is compared with that for separate inlet and exhaust valves, and tests to determine the effect of various degrees of superheat.

Two main indicators of engine performance were investigated: firstly the ratio (referred to below as the 'Steam Ratio') of the actual steam flow into the engine to the steam flow one would calculate from engine speed, swept volume to cut-off, and the specific volume of the steam); secondly, the indicated power from the engine, calculated from the mean effective pressure, the engine dimensions and the speed. The mean effective pressure was determined by using the computer to calculate the area of the indicator diagram. The power delivered to the brake was also calculated, but since this includes friction in the engine it is a rather less direct measurement of the thermodynamic performance. These values of power were then divided by the enthalpy of the steam supplied per second to yield the Indicated Efficiency and the Brake Efficiency of the engine.

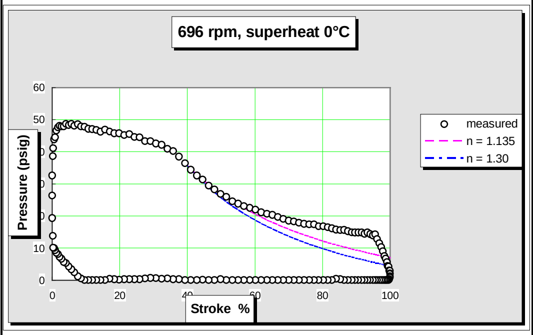
All the above quantities are calculated by the computer in a few seconds following each run and are then displayed on the screen and stored on a floppy disc. In the course of the analysis a number of intermediate quantities were calculated, such as steam flow rate, mean effective pressure, steam properties, and frictional torque.

The inlet steam temperature was measured immediately before entry to the steam chest. The pressure at this point was of course slightly higher than the maximum pressure in the cylinder because of wiredrawing. I decided to define superheat, which I believe to be important mainly because of its effect on cylinder condensation, as the measured inlet steam temperature minus the saturation temperature corresponding to the maximum cylinder pressure. I made an attempt to measure the steam temperature in the steam port just before the steam emerges into the cylinder, but I am not yet satisfied that the thermocouple used has an adequate frequency response. (I used the computer to sample the temperature just at the end of admission when it has been exposed to inlet steam for the longest period; however, this period may not have been long enough).

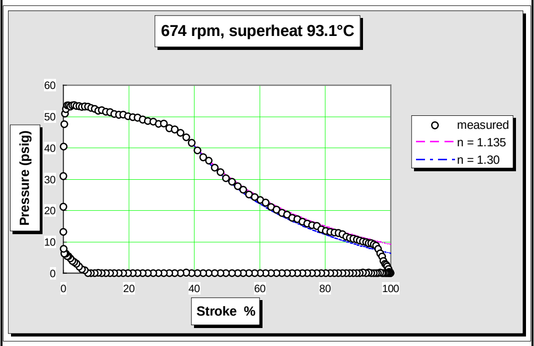
Results

A. The Effect of Superheat

One of the most striking characteristics observed was the large value of the Steam Ratio for those tests in which the steam was not superheated. Often two or three times as much steam was consumed than could be accounted for by the cylinder volume at the end of admission and the engine speed. Separate tests with air did not show this discrepancy, suggesting that it could not be accounted for by leakage. If one assumes that there is sufficient heat removed to condense half the steam, then there is no way that the cylinder block could dissipate that heat to the surroundings (several kilowatts in some of the tests). What happens is that steam initially condenses during the inlet stage of the cycle and is re-evaporated later on during expansion and exhaust - most of it too late unfortunately to significantly increase the work produced. This is clearly shown in Fig.4 in which indicator diagrams are shown for tests without and with superheat. Curves for perfect (thermodynamically reversible) expansion have been added as broken lines - the upper one for wet steam and the lower for superheated steam. In the test without superheat the expansion curve diverges upwards from the theoretical curve. This is caused by re-evaporation of the water produced by condensation during the inlet phase. In the test with 93°C of superheat the expansion follows closely the theoretical line for superheated steam, thus indicating that there was no re-evaporation and therefore no significant condensation during the admission phase. (These phenomena raise interesting heat transfer implications which I have discussed elsewhere[5].).



Steam mass flow 15.1 lb/hr, Indicated power 188.7 watts (0.253 HP)



Steam mass flow 9.3 lb/hr, Indicated power 194.8 watts (0.261 HP)

Fig.4 The Effect of Superheat

The data on which Fig.4 is based were for the separate valve configuration; however, the effect of superheat was much the same for the conventional valve and for cut-off points from 40% to 80%. The trend of Steam Ratio and Indicated Efficiency with superheat is illustrated in Fig.5 and Fig.7 - figures that also show the effect of the different valve arrangement and of the engine speed. It appears that in the case of this engine some 100°C of superheat is necessary to gain the full advantage, whereafter there is little further gain. Before leaving Fig.4 it is worth drawing attention to the steam flow in the two cases; slightly more power is produced with superheat for only 61% of the amount of steam !

B. Separate Inlet and Exhaust valves

Tests were made with each valve arrangement at cut-off values of 40% and 60%. The results were similar, and only those for 40% are shown here (Fig.5). The separate valve arrangement appears advantageous as indicated by the values of Steam Ratio and Indicated Efficiency. The reduced condensation is presumably a consequence of avoiding the cooling of the valve and inlet ports by discharging cold exhaust steam through them; as expected this effect reduces with increasing superheat. The sustained advantage in Efficiency at high superheat with the separate valve is probably due to the fact that in this case the release and compression events are not affected as the engine is 'notched-up'. This is illustrated in Fig.6 in which indicator 'cards' for the two arrangements are compared. Superheats are similar, but the separate valve configuration produces rather more power for 78% of the amount of steam.

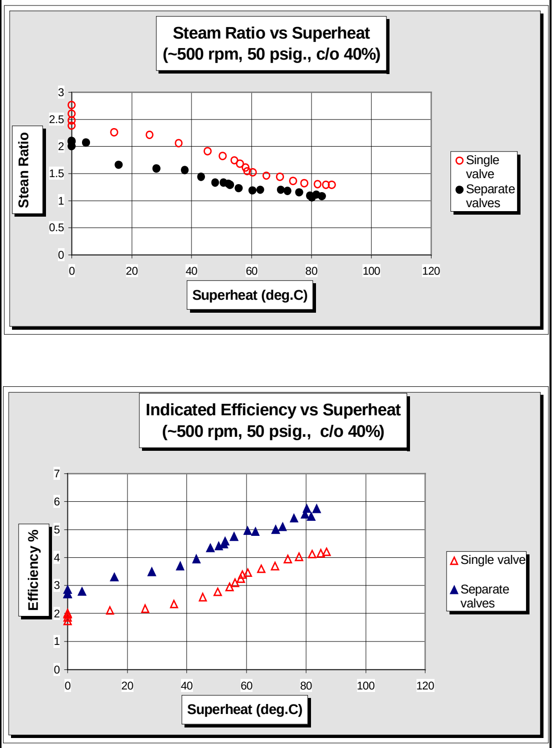


Fig.5 The effect of separate valves

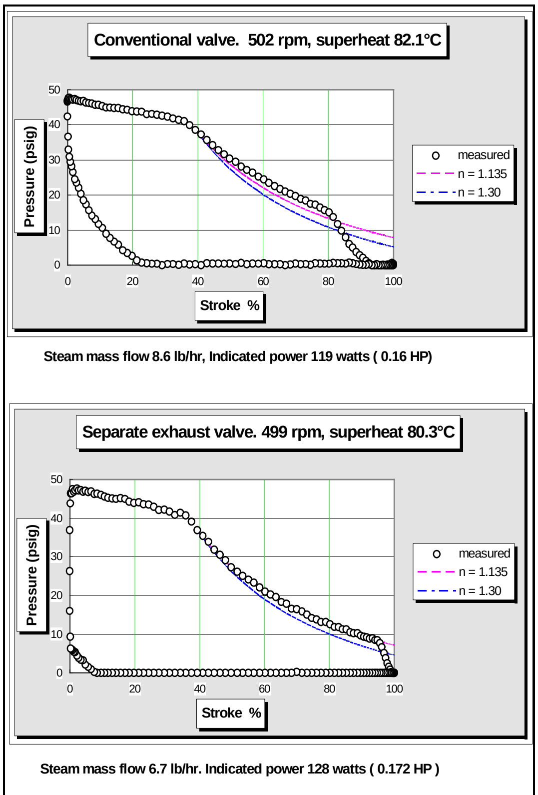


Fig. 6 Comparison of Single with Double valves

C. The effect of engine speed

It was very evident when conducting tests with unsuperheated steam that an increase in engine speed produced a decrease in condensation and an increase in efficiency. This is illustrated in Fig.7 in which results for two engine speeds are displayed.The first point to note is that the increase in efficiency is almost entirely attributable to the reduction in Steam Ratio (i.e. condensation). An increase in superheat also inhibits condensation so that there is less scope for improvement by increased speed, and the curves for the two speeds then approach eachother.The reason for the beneficial effect of speed (when using unsuperheated steam) is because there is less time for the heat of condensation to be (temporarily) stored in the cylinder wall. Fig.7 refers to tests at a cut-off of 60%, and tests at 40% cut-off follow a very similar pattern.

D. The effect of Cut-off

I have not yet had time to repeat the above measurements systematically for a wide range of values of cut-off. A limited comparison for cut-offs varying from 40% to 80% show almost no effect on Steam Ratio, and very little on Indicated Efficiency. This is rather surprising in that there would appear to be more scope for condensation when the inlet phase is longer. However, the rather large clearance volume of the engine (about 15% of swept volume) will have a disproportionate effect on efficiency at short cut-off, and it may be that this masks the effect of condensation. The results confirm the widely held view that cut-off affects efficiency rather less than one would expect from a simple thermodynamic analysis. Nevertheless, shortening cut-off to reduce load is ***still*** likely to be more economical in steam than closing the regulator. The above measurements were made at almost constant steam chest pressure and speed. Following load variations by changing cut-off retains the advantage of using the full boiler pressure, whereas throttling with the regulator in full gear does not!

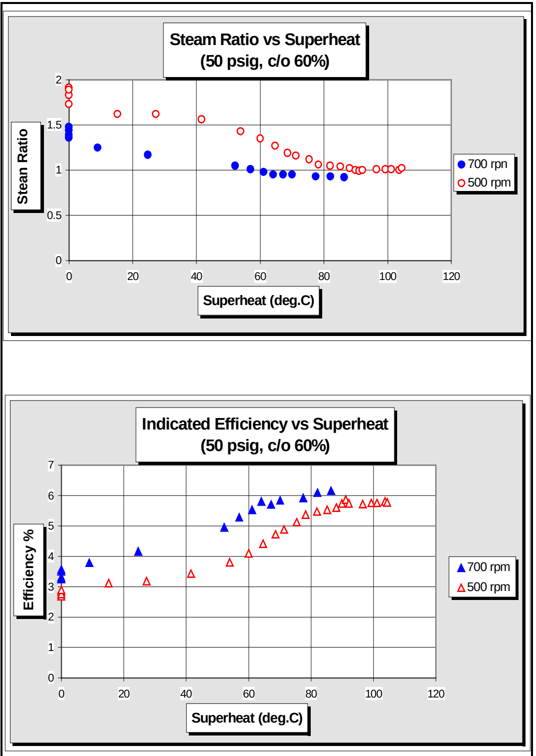


Fig. 7 The effect of Engine Speed

Proposed Future Experiments

It has often been suggested that the use of materials of low thermal conductivity for cylinder and piston might be advantageous in reducing cylinder condensation. There is perhaps less incentive when a similar effect can be achieved by using superheat, although there are situations where it would be attractive to dispense with the problems of lubrication and temperature control that are introduced with superheat. It would be very interesting to accurately measure superheat on a number of locomotives during operation. My ancient engine (a LBSC 'Speedy') refuses to show any superheat even when driven hard. It probably suffers from too many firetubes and hence a low pressure drop across the superheater flues, and would obviously benefit from a radiant superheater. One wonders how many other engines suffer from the same complaint!

However, before I get involved in radiant superheaters, I intend to rebuild the experimental engine with slide valves at inlet and a separate exhaust valve. Slide valve become more attractive when they do not have to handle exhaust as well as inlet steam, and can be located close to the ends of the cylinder, thus reducing port length. I might use insulating material on the faces of the piston and end covers, but I am not convinced that it would be either worthwhile or feasible to do likewise on the cylinder bore! There is also a problem recognised but not resolved by Professors Perry and Reynolds. This is the effect on condensation of air in the steam. On the instrumentation front, it would be nice to indicate the steam chest, and maybe I will automate the steam pressure and manometer readings so that I can leave the whole job to the computer! Further work is required on the problem of measuring the temperature of the steam where it enters the cylinder.

Acknowledgement

Many thanks to my friends Tom Jones and Duncan Webster for practical help and advice. For checks on my calculations, for sorting out my electronics and for constant reminders about how much more there is to do!

References

1.Capper D.S, ‘First Report to the Steam-Engine Research Committee’. Proc.I.Mech.E.(1905)

2. Perry John, ‘The Steam Engine and Gas and Oil Engines’, Macmillan, 1909. (Perry’s [anonymous] remarks about Reynolds occur in the Preface; that they do in fact refer to Reynolds is made clear by an entry in the Index! The footnote on p.587 gives a rather more considered opinion of Osborne Reynolds’ work.

3. Hall W B. ‘A Digital Indicator’, SM&EE Monograph No.2, 1993.

4. Pico ADC-11. This is a 10-bit resolution, 11 channel Analogue to Digital Converter which can be connected to the printer port of an IBM-compatible PC. It also has a digital output channel but no digital input channel (apart from those serving the ADC). However it is possible to use a spare input line of the printer port for this purpose. The ADC-11 is manufactured by:

Pico Technology Ltd.
Broadway House,
149-151 St Neots Road,
Hardwick,
Cambridge, CB3 7QJ

5. Hall W B. SM&EE Journal, Vol.8 No.7, Jan 1996, p.32.