Steam and steam engines Part 3

Ok then, so far we have touched on some of the basic fundamentals of water steam and it's generation; and have covered such things as temperature, pressure, heat of formation etc and also met with one of the more vague terms 'enthalpy'.
There is quite a lot more to cover on the generation side, boiler types, heat transfer, fuel, burners etc, however, we shall return to these at a later time.

For this part we shall take a look at what happens inside a steam engine cylinder and also take a look at how it is applied to one or two of the different types of steam engine. I will also introduce you to one or a few more bits of terminology as apply to this element. I don't think you will find it that boring, or I hope not…!!!

Steam in the cylinder

Very early in the history of the steam engine the need arose to know what exactly happened with expanding steam within the cylinder and the problem was resolved, albeit rather crudely, by James Watt who made a special recording device for the purpose, but he naturally kept it a closely guarded secret.
Basically his recorder consisted of a pencil attached to the top of a spring-controlled piston, which rose and fell as the pressure in the engine cylinder varied and at the same time a sliding board was swept from side to side in time with the strokes of the engine.
By attaching a sheet of paper to the sliding board, and allowing the pencil to press upon it, it could draw a graph drawn representing cylinder pressure plotted against piston stroke.
Measurements from this graph, together with the dimensions of the engine and its speed, enabled the power being developed on the piston to be calculated.
This apparatus was known as an engine indicator and the recorded graph it drew was known as the indicator diagram.
Over the next 100 or so years that followed, once the secret had got out, the indicator equipment was refined more and more until finally it became as refined and accurate as any mechanical device (friction and inertia are always present) can possibly become.

In more recent times it was found to be more convenient to change the scale of the 'piston stroke' to that of 'piston swept volume'. The area enclosed by the indicator diagram is then a measure of the work done on the piston.

Ok, so what does this indicator diagram look like? And what can it tell us?

Hypothetical indicator Diagram.

A hypothetical indicator diagram is that which is assumed for an imaginary engine where steam can flow without restriction or leakage (more on this later).

Such a diagram is considered as the first stage in the design of any engine, which has to meet specific requirements of power, speed and steam conditions.

Such a diagram is depicted in **FIG.4**

Please note…for clarity this diagram, and the 2 similar types which follow, represents a single complete cycle of a single acting (steam being admitted and exhausted at the same end of the cylinder) piston within a cylinder. A double acting engine cylinder (one which has identical steam inlet and exhaust ports at both ends) would have an identical, but mirror image, of this diagram superimposed for the piston return stroke.

It will be observed that 4 (four) cardinal points are indicated which correspond to the opening and closing of the cylinder to inlet and exhaust respectively.
These points are: ‘A’ **admission** and ‘B’ **cut-off** for the steam side, ‘C’ **release** and ‘D’ **compression** on the exhaust side. We will meet these terms many times in the texts/chapters which follow on from this one.

Point ‘A’ is the instant at which the steam inlet opens allowing steam in to the cylinder.

Point ‘B’ is the instant at which the steam inlet closes again and you will see that this happens before the piston has travelled the full length of the cylinder, this allows extra work to be gained from a given volume of steam by allowing it to expand within the remaining swept volume of the cylinder.

Note…cut-off is often referred to in terms of a percentage of total stroke length E.g. 75%.

Point ‘C’ is the instant the exhaust port opens which allows the steam to escape from the cylinder.

Point ‘D’ is the instant the exhaust port closes again and as you can see this again occurs before the end of the pistons return stroke.
The advantage of compression is of value thermodynamically, in that the small amount of steam remaining in the cylinder has its pressure raised nearer to that of the inlet pressure before fresh steam is admitted, and it also helps mechanically in that the pressure on the piston slows it down as it nears the end of its stroke.
This reduces the cyclical loading on the bearings and is termed ‘Cushioning’.

The small amount of work done in the compression process can be largely restored during the subsequent re-expansion, in the next stroke, with little net loss.

‘Clearance volume’ is the volume of the space between the piston and the cylinder head + the volume of the inlet and exhaust ports, at the end of the stroke. Ideally this should be such that the pressure at the end of the stroke is very near to that of the steam supply, without to large a compression period being required. The fresh steam can then be admitted without too much turbulence.

So what is the correct clearance volume required for an engine? Well this depends much on the exhaust pressure and an engine working at say 75psi – 100psi with an atmospheric exhaust could use a clearance volume of around 1/7th or 1/8th of the total piston swept volume, whereas and engine finally exhausting into a low-pressure condenser, such as a compound engine (double, triple or even quadruple expansion), would need a much smaller clearance, say 1/20th if it is to make effective use of compression. This is mainly due to the exhaust pressure being below atmospheric in these types of engines.

The expansion ratio of an engine is defined as: -

$$\frac{\text{Piston swept volume + clearance volume}}{\text{Volume at cut-off + clearance volume}}$$

Diagram factor.

Steam valves/ports do not open and close instantaneously and steam is also subject to inertia and friction, therefore, the hypothetical diagram can never be attained.
The real diagram actually lies within the hypothetical diagram and has all its corners rounded off. The area of the real diagram, expressed as a fraction of the hypothetical one, is termed 'the diagram factor'. A realistic diagram factor would be around 0.7.

Mean effective pressure. (M.E.P)

This is the difference between the average pressure on the piston during the power stroke, (this may, given suitable diagram scales, be calculated from the graph, however, for a good starting point the average pressure on the power stroke is usually taken as 75% of the boiler pressure in the UK (85% in the USA) but this is influenced by the cut-off position), and that on the return stroke.

Engine power.

The mean effective pressure combined with engine dimensions and speed gives a means of calculating the power being developed on the piston of an engine.
Such powers are known as 'Indicated Horse-Powers' (**I.H.P**).
Often this was the only power measurement made for most large steam engines.

The power delivered by an engine to its shaft can be measured. Since such powers were almost universally measured using a brake mechanism they became known as 'Brake Horse-Power' (**B.H.P**) or 'Shaft Horse-Power' (**S.H.P**) (The more common term used for marine engines).

The difference between I.H.P and B.H.P is that which is lost by friction within the engine, and the power needed for its essential auxiliaries (feed pumps, extraction pumps etc) and the ratio of B.H.P to I.H.P is the mechanical efficiency.

The mechanical efficiency of any engine is very much determined by the loading placed on the engine, and its speed. Measured values can lie between 0.9 for a low-pressure non-condensing steam engine running at low speed on full load and as low as 0.2 for a high-speed infernal combustion engine, also running on full load.

STEAM WINS EVERY TIME.

Model Engines.

The issue of engine efficiency is an important one, especially for model engines where, inevitably, higher frictional losses, steam leakage and other thermal losses can have a great influence.
I say 'inevitable' since it is quite normal for such small engines to have much larger bearing surfaces (non-scale) than a full size version and often things like piston rod glands are omitted (due to impossibly small size) which inevitably results in higher bearing friction and steam losses.
I am, of course, referring here to fully working engines (those used for our model boats etc), rather than true scale models, These latter types, whilst very much a true measure of the immense skills and talents of their constructors, would not necessarily have the mechanical rigidity to stand up to the rigors of working life in a model.
Take for example a true scale model of, lets say a triple expansion engine as was installed in the 'RMS Titanic'
These engines were over 30 ft high, so to build a model of one of these at say 4" tall would mean 1/90th scale. Now take, for example, one of the link pins for the top of the eccentric straps on the Stephenson reversing gear…on the full size engine these were approx 6" or so in dia…..at 1/90th scale this would become 0.066", which, although a manageable size to manufacture, would not necessarily be large enough to withstand extensive use in a fully working engine….3/32" or even 1/8" would be more likely in this instance.

There are, of course, many other examples, which could illustrate this, but I will not be going in to these. Suffice to say, that for those with the undeniable, and enviable, talent to build such exquisite examples of model engineering, then may the gods of superb engineering be with you…. For those of us who require an engine for the rigors of a hard life, then we will have to accept the inevitable additional losses.

Calculations of power.

Indicated power.

If P = Mean effective pressure
(measure from indicator diagram or 75% of boiler pressure)

L = Piston stroke.

A = Piston area.

N = Number of working strokes in unit time.
(not necessarily the same as revolutions)

Then the indicated power **I.H.P** is given by: - **P L A N**

For standard International units: -

P is in Newton/metres^2 (N/m^2)

L is in metres

A is in metres^2 (m^2)

N is in working strokes/second

And the power is in **Watts (W)**

In old money these are: -

P is in lbf/in^2

L is in feet

A is in in^2

N is in working strokes /min.

Horse power is given by: - $$\frac{PLAN}{33000}$$

Brake mean effective pressure.

This is a conceptual idea often used for engines where it is not possible to produce an indicator chart, but which can have their power measured on a brake. An artificial figure is then calculated from the result.
This is the Brake Mean Effective Pressure and is found by equating the Brake power to the expression derived for indicated power.

The Brake Mean Effective Pressure is given by: -

$$\textbf{B.M.E.P} = \frac{\textbf{33000 x Brake Horse Power}}{\textbf{LAN}}$$

Ok then, enough of the horrid number things…He He.

Over Expansion – Negative loop.

Returning to the indicator diagrams, take a look at **FIG 5**.

All engines necessarily work against a **Back-Pressure**, which, in a simple case may be that of the atmosphere (14.7 lb absolute), also the inevitable friction within the engine may be considered as an increment in this back-pressure, therefore the return stroke pressure line on the indicator chart will be inclusive of this.
If an engine, working on low pressure, has its cut-off point set to early in the stroke then expansion may take place to a pressure below that of the effective back-pressure and work has to be drawn from the shaft to pump the steam back out of the cylinder. Such a process is called a **Negative loop**.
It is, therefore, important to minimise these and not try to use too early a cut-off. 50% is probably the minimum cut-off point for most engines.
Negative loops can also occur at the beginning of a power stroke, particularly with engines using Stephenson Link reversing gear, near to the mid gear point.

We will return to this at a later time when I look at valves and valve gear.

For most of us, when considering our model boat engines, these concepts, real as they are, would not be of very great concern, since for most the reversing valve gear would not be used in the ways which could easily introduce such issues.
For Model Rail enthusiasts, they are a very real issue, since these guys make much more use of the techniques for changing the cut-off point, in order to increase efficient running and minimise steam usage. That is not to say that such things could not benefit marine use…they most certainly could, however, it is not quite so simple in a model boat, where you cannot see the position of the reversing gear.
For those of you who own a man carrying steamboat, then you would most certainly be looking to use this…It is a technique known as **Notching up** and I will explain it further when dealing with valves and valve gears.

Compound engines.

Up to now we have been considering what are known as 'Single Expansion' engines, which, for the most part, are usually more than adequate for most purposes.
Having said this, there are occasions/applications where the need to extract as much work from a given amount of steam are deemed desirable, and also to obtain as much power for a given engine size.
Large marine applications, such as the large cruise liners (like RMS Titanic) or warships (such as turn of the $19^{th}/20^{th}$ century battleships) or even large commercial goods carrying ships, which were required to travel over ever increasing distances Were usually fitted with 'Multiple Expansion' engines. AKA 'Compound engines'

These engines used a technique, which divided the expansion pressure drop amongst two (2) or more cylinders, each subsequent stage (cylinder) using the exhaust steam from the preceding cylinder. Thus extracting as much power as possible from each charge of fresh steam. For a 2 cylinder engine these were designated as 1^{st} cylinder = High Pressure cylinder (HP) with the second/final cylinder being the Low Pressure cylinder (LP).
For 3 or more cylinders then the second and or/third cylinders were termed Intermediate Pressure cylinders (IP).
The final cylinder in all cases was exhausting into a low-pressure condenser.

Naturally, there was a limit, expansion could only occur to the lowest back-pressure attainable, which then led to the added complications/load factors etc, of needing to install, and power, what are known as AIR PUMPS (in marine technology) in fact these were Vacuum pumps used to reduce the internal pressure of the final exhaust condenser stage, to a pressure below that of the atmosphere.

Such engines were fitted with 2 or more cylinders, each subsequent stage having a piston of greater area then the preceding stage. The dimensions were calculated such that each stage, theoretically, produced the same power output and also a similar ratio of expansion.
It soon became accepted, by most users, that the increases in improved economy (less coal burnt, less water used etc were of great enough benefit to overcome the many added complexities of such engines. Not the least of which was higher initial price.

So why were these engines seen to have such high levels of economy?
One old theory, the theory of initial condensation was dragged out again, which suggested that the lower temperature difference between the stages would lead to lower condensation. This seemed to fit, up to a point, and to some extent is still quoted today.
However, the main reason for the improvement in overall economy is the fact that compounding leads to a much higher ratio of expansion.

The hypothetical indicator card for a 2 cylinder compound engine is shown in **Fig 6**.

Ignoring clearance volumes, the ratio of expansion is: -

$$\frac{\text{Volume of LP}}{\text{Volume of HP at cut-off}}$$

At HP release there is an irreversible pressure drop down to LP. Inlet pressure. This is not completely lost energy however, since some, but not all, can be recovered by the drying effect (remember un-resisted expansion from part 2).
The total power of these engines can only be influenced by either the initial pressure or by the Cut-off point in the HP cylinder.
Variations in cut-off in the lower pressure cylinders influence the power distribution between the cylinders. A later LP cut-off reduces LP power.

Under normal conditions a typical cylinder/cylinder volume ratio of 4 (diameter ratio of two) is generally assumed to provide approximately equal power per cylinder. Also, since equal work per cylinder is the aim, then it follows that the smaller cylinders must work with the biggest pressure drop. This means that, if we start with dry steam, and if the expansion in the high-pressure cylinder is efficient, then the steam will be wet when it enters the first receiver (this is the inlet area containing the second stage steam valve). As stated above, the un-resisted expansion into this receiver will often assist in drying this steam, however, it is often necessary to provide additional heat by some other means (passing the steam to the receiver via some sort of heat exchanger for example) if the overall increase in expected engine power is to be realised.

A boiler superheater may safely raise the temperature of steam to a point, which is higher than that which can be safely passed into the HP cylinder. If this steam is first passed through heat exchangers (on the isolated side) feeding the second/subsequent stage receivers the extra heat can be used to heat and dry the HP exhaust prior to final input to the LP cylinder/s. This needs very careful design if the HP cylinder is not to be damaged by excessive temperatures.

Another problem here is the fact that an engine so equipped would almost certainly display reduced manoeuvring response due to the increased volume of steam downstream of the stop valve.

Over-expansion.

Compound engines may also exhibit problems, which a simple (single expansion) engine would be able to suppress.
These can arise when a two, or more, crank engine is throttled down to run at low power and speed. In these circumstances the High Pressure cylinder may then take more than its fair share of the load and if an indicator chart could be plotted, a pronounced negative loop would be shown in the low-pressure diagram. In other word, the high-pressure cylinder would have to provide work to overcome the friction in the lower-pressure cylinders and to pump (force) out the over expanded steam.
Typical symptoms would be low-pressure cross head “slop” where the piston rod is pulling on the cross head, rather than pushing it, or vice-versa, erratic running or even complete stalling.
The problem stems from lack of matching between the engine and the load conditions.
Whilst compound engines are excellent for high power, high efficiency at a moderate/high speed/load they are not good for low speed loads.

Compound versus simple.

Whilst a compound engine has many undeniable virtues, the most endearing being its ability to use a large ratio of expansion, which leads to a marked saving in fuel usage, it is an ideal choice when long, uninterrupted voyages are involved.
This is not so when short journeys, often involving many slow manoeuvring stages, where apart from the above mentioned negative loop problem, the starting of a compound engine is also a common issue.
Since the steam for the lower pressure cylinders must first pass through the HP cylinders, it follows that if the engine is stopped at any other position than inlet for the HP cylinder, then the engine will not start.
To overcome such dead centre problems, users of compound engines have to resort to a few sneaky tricks…they need to install bypass steam valves, which will provide the low-pressure cylinders with a whiff of HP steam to get them started. This is fine if you have a fully manned engine room…not so good for the day skipper with a small canal boat or something similar.

Not so simple to arrange in a model ship, where the only control is remote via radio.

I have had the personal experience of this problem, when assisting my good friend, and owner, of the Clyde Puffer 'VIC32', Mr Nick Walker in bringing the puffer through the 'Crinan Canal'. The many locks require many stops and starts and, since the 'VIC32' has a 2 cylinder compound engine, it is more often than not required to crack open the 'WHIFTER VALVE' as we have named it, to get the darned thing to go.

For small river/canal boats, or smaller sea going vessels, a two or three crank single expansion engine has the advantage in being free from manoeuvring problems and generally from expansion problems, especially at the lower speeds generally involved.
Another area where the simple engine wins out is its lack of requirement for added auxiliary equipment, such as vacuum pumps and receiver heat exchangers etc.

Ok then, I think that will do for this espisode..Ha Ha.

Next issue will take a look at some different types of engine, as used for our models, and perhaps make a start on valves and valve gear.