PREDICTING PERFORMANCE --THE PROBLEM OF CONDENSATION

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Introduction

The paper, "Predicting Locomotive Performance", [Ref.1], deals primarily with steam flow in the cylinders. It calculates flow and pressure drop through the ports, backpressure across the blastpipe, and constructs an indicator diagram from which MEP and IHP can be calculated. No account is taken of condensation in the cylinders, and the prediction is likely to be valid only if the steam is sufficiently superheated to prevent it. A reasonably encouraging comparison with BR Class 7 test data is made in the paper, and a more extensive set of comparisons has been made by Pawson [Ref.2].

Condensation in a steam engine cylinder has been recognised as a source of inefficiency for more than a century. In small engines (cylinder diameters of a few inches) it can easily halve the engine efficiency. In most cases the power developed for a given steam chest pressure is not greatly affected, but condensation causes a significant increase in the steam consumption. The reason for this behaviour is that during the admission period a slight reduction in cylinder pressure (as a result of condensation on the cylinder walls) causes more steam to flow through the inlet ports. Later in the cycle the condensate may evaporate as the cylinder pressure falls; this will contribute little to the work output, and indeed may reduce it as a result of the greater backpressure caused by the greater steam flow. The problem largely disappeared with the use of superheated steam and probably for this reason it has not attracted sufficient interest for a quantitative model of the process to be produced The improvement in efficiency caused by superheating simple non-condensing engines may well in fact have been due more to reduced condensation than to improved thermodynamic efficiency.

Whilst there is probably sufficient basic understanding of condensation and evaporation of water films to model these processes, this can only be done if the cylinder temperature is known. The cylinder is subjected to considerable heat input and heat removal if condensation and evaporation occur, and it can also lose heat to the surroundings – a process that is not easily assessed. The current theoretical model and its associated computer program is not able to predict cylinder temperature, but in the absence of a full solution it may be instructive to model condensation on the basis of a range of arbitrarily chosen cylinder temperatures. That is the purpose of this note.

Some physical aspects of condensation and evaporation

The kinetic theory of gases enables one to calculate the rate at which gas or vapour molecules impact upon a solid or liquid surface. Equilibrium between steam and water is a dynamic situation involving 'condensation' of molecules that impact on the water surface, balanced by the 'evaporation' of molecules at an equal rate. In the non-equilibrium state, when the water surface is below saturation (i.e. equilibrium) temperature, the gross condensation rate remains high but the evaporation rate is diminished, resulting in a net condensation rate. On this basis the condensation coefficient (i.e. the net condensation rate divided by the temperature difference between the vapour and the surface of the water layer) would be very large indeed. For example, at atmospheric pressure a temperature difference of 1°C would result in a net condensation rate of about 3.4 kg/sq.m sec (1860 lb/ sq.ft hr), corresponding to a heat transfer rate of 0.76

kW/sq.cm (1.8 10^6 Btu/sq.ft hr). This is to be compared with the heat transfer rate through a 0.025 mm (0.001") thick water layer, with the same (1°C) temperature drop across it, of about 0.0027 kW/sq.cm (6400 Btu/sq.ft). Clearly the resistance of the water layer dominates.

In the case of an engine cylinder, the prime factor that governs these processes is the relationship of the cylinder temperature T_c to the **saturation** temperature T_s of the steam in contact with it. If $T_s > T_c$ condensation will occur whereas if $T_s < T_c$ evaporation will occur provided a water layer exists. In general we expect condensation during admission, followed by evaporation as the steam expands and its saturation temperature falls below the cylinder temperature. The water layers formed are very thin, but nevertheless provide virtually all the resistance to heat transfer. This means we have a relatively simple formulation: the water surface temperature is equal to the saturation temperature of the steam, and the condensation or evaporation rate is governed by the thermal resistance the water layer.

The modified program

The surface of the cylinder in contact with the steam is assumed to be at a uniform and steady temperature. The surface temperature will in fact fluctuate, but the range of the fluctuation is likely to be small compared to the fluctuation in steam saturation temperature. (Some supporting evidence for this assumption will be found in Ref.3.) In the program the temperature is assigned arbitrarily as an input parameter. If it is set above the steam chest saturation temperature no condensation will occur (unless compression takes the steam pressure above the steam chest pressure); it can then be reduced in stages until unreasonably thick water layers are formed. Thick water layers reduce the condensation rate because of their high thermal resistance, but on the other hand they will eventually become unstable and run off the surface. No attempt has been made to define this stability limit. The condensation is based on the surfaces of the covers, piston faces, ports, clearance volumes and swept surface of the cylinder to the point of cut-off..

The program presents a diagram that shows how the water layer thickness changes around a cycle. Initially the layer grows rapidly because the thin layer presents little thermal resistance. It reaches a maximum during admission, and then decreases as the cylinder pressure declines during expansion. With lower cylinder temperatures the layer may persist around the whole cycle, but the layer then thickens rapidly and the results are likely to be unreliable. All the predictions take account of the operating conditions such as speed, steam conditions, cut-off etc. , so one may get an idea of how these factors affect condensation.

The original calculation has been modified by including an additional term in the differential mass balance equation; this accounts for the condensation of steam or the evaporation of the water layer. Condensation will tend to reduce the pressure in the cylinder, and if the inlet valve is open this will cause additional steam to flow in. Evaporation of the water layer later in the stroke will tend to increase the cylinder pressure above the level corresponding to no condensation. In an attempt to model the effect of superheat on condensation, the enthalpy associated with condensation has been increased to account for the higher enthalpy of the superheated steam; no change is made to the enthalpy of evaporation. This is a mechanism that will affect the cylinder temperature, but this is not modelled in the program since the cylinder temperature is assigned arbitrarily. A summary of the changes is given in the Appendix.

Some typical results

The sample of results shown below is for the BR Class7 Britannia running at 70 mph and 30% cut-off. The table and the graphs show the effect of progressively reducing the cylinder temperature (As mentioned above, the surface for condensation has been taken as the area of the cylinder cover, the piston and the clearance space; uniform temperature is assumed)

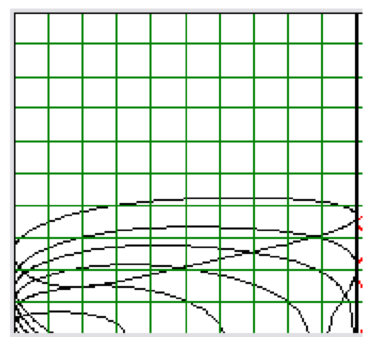
The top line in the table assumes a cylinder temperature that is above the saturation temperature of the steam in the steam chest, so no condensation occurs. The left hand ordinate of the graph represents TDC, and the graph shows the thickness of the condensate layer around one cycle of the engine. For example, at a cylinder temperature of 200°C the layer is formed at TDC and persists to just over 30% stroke. At 170°C the layer grows to about 0.02 mm in thickness and persists for the whole of the working stroke. At 150°C the layer persists around the complete cycle.

The graphs of water layer thickness are for assumed cylinder temperatures from 200°C to 150°C, the thickness increasing as the temperature decreases.

Engine: britannia

Cut-off %	Speed mph	St.cons lb/hr	IHP	BackP psi	ITE lb	MEP psi	Effy %	Cylinder deg C
30.0	***70.0***	***27756***	***2085***	***9.63***	***11170***	***73.8***	***14.3***	***210***
30.0	70.0	28390	2113	10.17	11320	74.8	14.2	200
30.0	70.0	29069	2061	11.04	11039	73.0	13.5	190
30.0	70.0	29575	1995	11.81	10688	70.6	12.9	180
30.0	70.0	29956	1942	12.43	10405	68.8	12.4	170
30.0	70.0	30356	1912	12.99	10241	67.7	12.0	160
30.0	70.0	29206	1878	11.79	10062	66.5	12.3	150

0.1 mm



0.0 mm

There are a number of interesting trends exhibited by the performance figures in the table. Firstly, condensation generally increases steam consumption, as would be expected. The exception in the case of 150° arises because the thermal resistance of the relatively thick water layer reduces the condensation rate. However it would be unwise to place too much weight on this effect because thick layers become unstable. The increase in steam consumption is of course accompanied by a greater backpressure, and this is probably the main factor in reducing IHP. However, there is an effect on IHP that can operate in the opposite direction; the water layer, by evaporating later in the stroke can contribute work. This effect may be responsible for the increase in IHP in the second line of the table.

Conclusions

Whilst the program has not reached the stage when it can predict performance under conditions where condensation occurs, it does offer some indication as to the probable mechanisms that govern the process. It may seem a small step to close the loop by predicting cylinder temperatures, but a great deal more data on the construction of the locomotive would be necessary in order adequately to predict heat losses. The assumptions concerning cylinder temperature – that it is uniform over the condensing surface, and that it is steady over the cycle – also deserve further scrutiny.

References

1. Hall, W B "Predicting Locomotive Performance", S L S Journal, Vol.75 No.797, 1999

2.

3. Hall, W B "The Effect of Superheat on Cylinder Condensation", The Model Engineer, Vol.187, No.4161, Jan. 2002

APPENDIX

Modifications introduced by condensation in Perform.exe

Heat flux through water layer, $q = (T_{sat} - T_m).k/\lambda$ watts/m^2

where

T_{sat} = saturation temperature at cylinder pressure = $104.87 \times 0.94911^{1/p} \times p^{0.23647}$ °C

p = Cylinder pressure (bar)

T_m = Mean wall temp. °C

k = Thermal conductivity (W/mK) [$\cong$ 0.680]

λ = Thickness of water layer (m)

Rate of change of water layer thickness $d\lambda/dt = .(T_{sat} - T_m).\, k/\rho_f h_{fg}\lambda$ (m/sec)

where

t = time

h_{fg}= enthalpy of condensation (J/kg) [$\cong$ 2.47.10^6]

ρ_f = density of water (kg/m^3) [$\cong$ 1000]

[As an approximation assume constant values of ρ_f , k, and h_{fg}.]

Thus $d\lambda/dt = 2.75.10^{-10}.\,(T_{sat} - T_m)/\lambda$ (m/sec)

Rate of change of mass of steam as a consequence of condensation

$$dm/dt = -\rho_f\, d\lambda/dt = -.(T_{sat} - T_m).\, k/h_{fg}\lambda = -2.75.10^{-7}.\,(T_{sat} - T_m)/\lambda \quad \text{(kg/m}^2\text{ sec)}$$

(e.g for $T_{sat} - T_m = 100$ and $\lambda = 10^{-5}m$, q = 6.8 MW/m^2 or 0.68 kW/cm^2 & dm/dt = 2.75 kg/m^2 sec)

Rate of volume change

$$dV/dt = v.(dm/dt).A = v_0\left(\frac{p_0}{p}\right)^{1/n}.(dm/dt).A = -2.75.10^{-7}.A.v_0\left(\frac{p_0}{p}\right)^{1/n}\frac{(T_{sat} - T_m)}{\lambda}$$

The calculation in the original version of the program is modified to include the effect of this volume change on the steam in the cylinder. The equation giving the rate of change of the water layer thickness is then introduced as an additional differential equation to be solved by the Runge-Kutta routine.