**Appendix 1**

# The fire burns much better …

**200 years of steam locomotive exhaust research**

**1804-2004**

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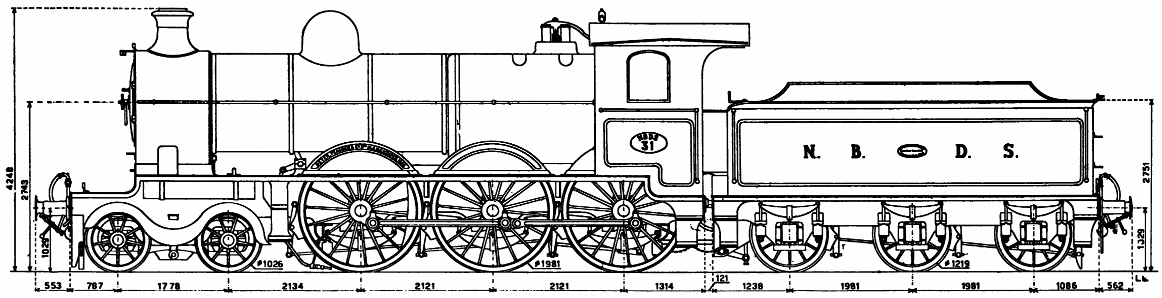
**“The preliminary design of a replica Immingham class locomotive”**

### 1.1.1 The preliminary design of a replica Immingham class locomotive.




*NBDS Nos. 30-35, 2 cylinder 4-6-0 steam locomotive* [1]

A heritage organisation has the ambition to build a replica Immingham class locomotive. This was a design by Robinson for a 4-6-0 type locomotive with two outside cylinders for the Great Central Railway. The locomotives were built by Beyer-Peacock in 1906 and delivered in two batches with slight differences. An inside-cylindered version was built in 1908 for the Dutch Noord-Brabant Duitsche Spoorweg Maatschappy (NBDS)[1] and used on its trains to and from Germany before the First World War.

The main dimensions were:

| | | |
|---|---|---|
| Heated surface firebox | 13.5 | $m^2$ |
| Heated surface tubes and flues | 95 | $m^2$ |
| Superheater | 34 | $m^2$ |
| Grate area | 2.6 | $m^2$ |
| Boiler pressure | 13.4 | bar |
| Cylinders (2) diameter | 510 | mm |
| stroke | 660 | mm |
| Steam distribution | Walschaerts (NL), Stephenson (UK) | |
| Wheel diameter | 1981 | mm |
| Tender water space | 15.9 | $m^3$ |
| Tender coal | 5 | t (of 1000 kg) |
| Locomotive weight incl. supplies | 62 | t |
| Tender weight incl. supplies | 38 | t |
| Total length | 17881 | mm |
| Maximum velocity | 100 | km/h |
| Tractive effort | 8130 | kgf |

All engines later received superheaters and had their cylinder diameters increased. The Dutch locomotives were produced with as many weight savings as possible due to the light track of the railway. Over the life of the locomotive this resulted in cracks in the frames, which led to them being taken out of service from 1938 onwards. The British locomotives lasted until 1949 when they were replaced by the new Thompson designed B1 class.

The heritage organisation would like to build a locomotive that corresponds to the original design and looks like it. However, modern times demand a higher velocity of 120 km/h, while the original velocity was 100 km/h with a train of 400 tonnes.

For the preliminary design the classical approach, as described by Eckhardt[2], was chosen in an attempt to verify a British design using a German method.

**Step 1, determination of the rolling and wind resistance of the locomotive**

The formula of Strahl is used: $W_L = 2G_1 + 7.3G_r + 0.6F\left(\frac{V+8}{10}\right)^2$ in kgf

$G_1$ is the weight of the locomotive in tonnes including the tender with 2/3 of the supplies, $G_r$ is the friction weight, $F$ the equivalent cross sectional area, $V$ is the velocity.

**Step 2, determination of the resistance of the carriages of the train**

Strahl used the formula: $W_w = 2.5 + \frac{1}{40}\left(\frac{V}{10}\right)^2$ in kgf/tonne of train mass

**Step 3, the wheel diameter**

This is a check that limits of revolutions are not surpassed. The locomotive will again use wheels of 1981 mm diameter (*D*). At a velocity (*V*) of 120 km/h the number of revolutions is : $n = \frac{V}{0.1885D}$ or 321/min, 5.36/sec. These are values within the normal range.

**Step 4, the piston stroke and velocity limit**

The original piston stroke (*s*) of 660 mm is to be used. The average piston velocity is calculated from: $c_m = \frac{sV}{5.64D}$ giving a value of 7.09 m/sec, which is below the limit of 8 – 9.5 m/sec. The values of *s* and *D* are introduced in m, *V* in km/h.

As the wheel diameter and the piston stroke are known, the cylinder diameter can be determined now. In a normal procedure this would be dependent on the friction between rails and wheels. In the present case the cylinder diameter *d* remains at 510 mm.

**Step 5, the friction between wheel and rails**

Eckhardt uses the formula: $\mu = 93 + \frac{7200}{V+42}$ in kgf/tonne.

At a velocity of 0 km/h this would give 265 kgf/tonne.

**Step 6, the determination of the maximum tractive effort**

This is generally calculated at 85% of the boiler pressure by the formula:

$$Z_{\max} = \frac{d^2 s\, i\, 0.85p}{2D} \quad \text{kgf}$$

The linear dimensions are to be used in cm, the pressure *p* in atm, that is kgf/cm$^2$.

The cylinder stroke is *s*, the number of cylinders is *i*, *d* and *D* are used as before.

**Step 7, the amount of steam.**

Data from other historic designs are used as reference values. Eckhardt uses:

$$D_i = \frac{26.9y}{\frac{z(p-1)}{N_i} - \frac{270}{TV}}$$

indicated steam used in kg/h. $V$ and $p$ are pressure and velocity as used before. $T$ is a coefficient given by: $T = \frac{d^2 s}{D}$ for two-cylinder locomotives. $N_i$ would be the indicated (horse) power given by: $N_i = \frac{Z_i V}{270}$ in $Ps_i$/h, if $Z_i$ is the required tractive effort in kgf. The variable $y$ is the specific weight in kg/m$^3$ of steam. The value of $z$, a function of swept cylindervolume/valve diameter and velocity, is read from a graph in Eckhardt's book.

The amount of steam calculated has to be increased by the 1% loss and the steam consumption of other uses, like the 100 kg/hr for the compressed air pump.

**Step 8, the size of the evaporation area**

The evaporation area is calculated from the historical data of existing designs. Eckhardt suggests using 40 – 60 kg/h/m$^2$ of steam. However, this topic was investigated by the Austrian Sanzin[3]who changed the coefficients of the formula of Köchy[4] into:

$$D_n = \frac{5525}{35 + \frac{H_v}{R}} H_v$$ in kg/h.

In this formula, $H_v$ is the evaporation area, measured on the fireside, and $R$ the grate area in m$^2$. Formally, the formula is valid only for coal of 26 MJ/kg. The grate area will remain very close to the original value. The ratio of evaporation area to grate area $H_v/R$ is determined by the type of service for which the locomotive is to be used and the calorific value of the coal used. The replica would be used as a passenger locomotive and should consume the best coal available. Consequently the ratio should be between 40 and 50.

**Step 9, the determination of the superheater area.**

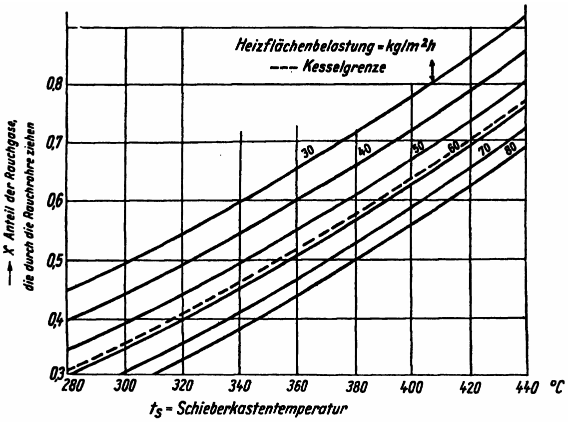



*Figure 2 Superheat and free flue cross sectional area [2]*

In normal practice the superheater area would be chosen at about 33 to 40% of the normal evaporation area. However, superheat should be very high as a design feature; 400- 420 °C is the goal. Eckhardt supplies Figure 2 for its determination. About 66% of the gases should flow through the flues and immerse the superheater elements if the graph is entered for 420 °C and 70 kg/m$^2$/hr of steam. The remainder should go through the tubes.

The calculations above are made for velocities from 90 to 120 km/h. The values of the other variables used are:

| | | |
|---|---|---|
| Locomotive weight $G_l$ | 71 | t |
| Adhesion weight $G_r$ | 53 | t |
| Area value *F* | 10 | $m^2$ |
| Train weight | 400 | t |
| Piston stroke *s* | 660 | mm |
| Cylinder diameter *d* | 510 | mm |
| Boiler pressure *p* | 15 | bar |
| Number of cylinders | 2 | |

The following table could be produced:

| Step | Velocity | 90 | 100 | 110 | 120 | km/h |
|---|---|---|---|---|---|---|
| 1 | Resistance of locomotive | 1032 | 1156 | 1291 | 1439 | kgf |
| 2 | Resistance of train | 1810 | 2000 | 2210 | 2440 | kgf |
| | Total resistance | 2842 | 3156 | 3501 | 3879 | kgf |
| 3 | Wheel diameter | 1981 | 1981 | 1981 | 1981 | mm |
| | Revolutions *n* | 241.02 | 267.80 | 294.58 | 321.36 | *n*/min |
| 4 | Stroke | 660 | 660 | 660 | 660 | mm |
| | Piston velocity *Cm* | 5.32 | 5.91 | 6.50 | 7.09 | m/s |
| 5 | Friction | 147.55 | 143.70 | 140.37 | 137.44 | kgf/tonne |
| | At *V*=0 | 264.43 | 264.43 | 264.43 | 264.43 | kgf/tonne |
| | Resistance / Friction ratio | 19.26 | 21.96 | 24.94 | 28.22 | |
| 6 | Maximum tractive effort | 11049 | 11049 | 11049 | 11049 | kgf |
| | | | | | | |
| 7 | Amount of steam, calculation of the dimensions of heating area: | | | | | |
| | Cylinder swept volume *J* | 134.83 | 134.83 | 134.83 | 134.83 | $dm^3$ |
| | Valve diameter $d_s$ | 250 | 250 | 250 | 250 | mm |
| | Ratio $J/d_s$, to read *z* value from graph | 0.539 | 0.539 | 0.539 | 0.539 | $dm^3/mm$ |
| | *z* value, supplied by Eckhardt | 1.85 | 1.85 | 1.85 | 1.85 | |
| | Indicated power $N_i$ | 947 | 1169 | 1426 | 1724 | $Ps_i$ |
| | Steam request | 4926 | 6078 | 7419 | 8968 | kg/h |
| | Total steam incl. loss, pump | 5075 | 6239 | 7594 | 9158 | kg/h |
| | Steam generated | 67 | 67 | 67 | 67 | $kg/m^2/h$ |
| 8 | Heating area $H_v$ needed | 76 | 93 | 113 | 137 | $m^2$ |
| | Grate area | 1.52 | 1.86 | 2.27 | 2.73 | $m^2$ |
| 9 | Superheater area | 29 | 35 | 43 | 52 | $m^2$ |

In order to verify some of the numbers, a paper by Franco[5] can by used. He delivered a lecture discussing the comparative results of the largest Dutch locomotives in 1915. He concluded that the boiler of the NBDS locomotive supplied 7445 kg of steam/h for a train of 400 tonnes travelling at 90 km/h, for which a tractive effort of 2100 kgf was needed plus 1091 kgf for the locomotive. Franco used a heating surface which was $10m^2$ too large.

The table above shows about the same resistances but a smaller amount of steam, which is understandable, keeping in mind that Eckhardt's calculation is from 1948.

However, there are also test results from the BR tests at Rugby. Both the LNER B1[6] and the BR Class 4[7] locomotives were tested and their performances were within the limits given. From the reports the following data can be extracted:

| Locomotive: | B1 | BR 4 | |
|---|---|---|---|
| Maximum test velocity | 112.5 | 120.5 | km/h |
| Tractive effort at maximum velocity | 2268 | 1814 | kgf |
| Steam generated | 9072 | 8890 | kg/h |
| Train weight | 400 | 400 | imperial ton |
| Tractive effort at 85% boiler pressure | 11956 | 11053 | kgf |
| Vacuum needed | 1246 | 1744 | Pa |

These numbers compare favourably with the Eckhardt calculation. Apparently, the resistance of a 400-ton train was lower than determined by the Strahl formulae, probably because of improved bearings and the difference in the German loading gauge allowing a larger cross-section. As the tractive effort at 85% boiler pressure is about the same as that of the replica locomotive, the steam engine appears to be capable of the required performance. It should be noted that both the B1 and BR4 had long lap and long travel valves.

**New design features**

If the replica locomotive is to meet the specifications of the heritage organisation, the following changes should be contemplated:

- higher superheat, to give better quality steam to begin with;
- long lap, long travel valves, to improve on the usage of the steam;
- an improved front end, giving a lower backpressure;
- a higher boiler pressure: a new boiler should still meet the weight of the original riveted boiler, 15 bar is suggested.

The thermodynamic performance of both the Class 4 locomotives should be considered next. The tests were made using coal of 32 MJ/kg and, as the steam amounts and temperatures are known, a thermodynamic balance can be made:

| Locomotive: | B1 | BR 4 | |
|---|---|---|---|
| Coal burned | 1270 | 1286 | kg/h |
| Heat released by 32 MJ/kg | 40642 | 41150 | MJ/h |
| Enthalpy of steam raised | 28204 | 27963 | MJ/h |
| Boiler efficiency | 69.4% | 67.9% | |
| Enthalpy of steam ejected at 120 $^{o}$C, 1.2 bar | 24624 | 24132 | MJ/h |
| Heat used for effort | 3580 | 3831 | MJ/h |

Apparently, if the replica locomotive could supply at least 3831 MJ/h, the required performance could be met. Assuming that the boiler delivers the same enthalpy to the steam as that of both locomotives, the required performance should be met and the following data could be calculated for the replica:

| | | |
|---|---|---|
| Requested enthalpy of steam raised | 28500 | MJ/h |
| Enthalpy of 1 kg steam of 15 bar, 420 °C | 3.298 | MJ |
| Steam raised | 8641 | kg/h |
| Enthalpy of the amount ejected, 1.2 bar 120°C | 23455 | MJ/h |
| Heat available for work | 5045 | MJ/h |

This amount appears to be quite sufficient for the required performance. One way to convert the amount of heat to actual steam amounts is shown in the graph by Postupalsky[8] as reproduced by Giesl-Gieslingen[9].

The graphs, Figure 3, show that 7.13 kg of steam at 325 °C are needed for 1 indicated horsepower-hour. At 425 °C this value has dropped to 5.7 kg/h.

Interpolation for 320°C and 15 bar gives 7.17 kg steam and for 420°C it drops to 5.736 kg.

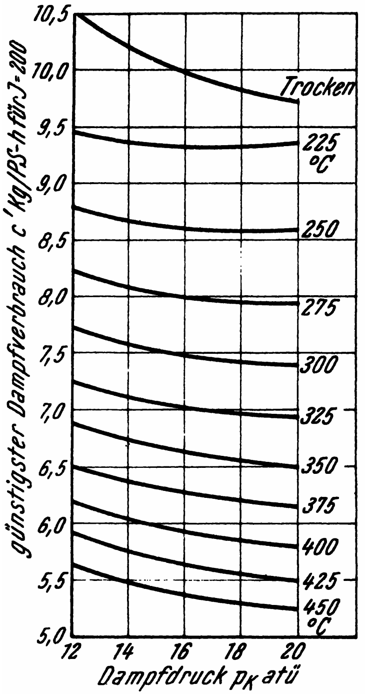



*Figure 3 Optimal steam consumption [9]*

An additional correction for cylinder size is needed as Postupalsky calculated for a cylinder of 200 litre volume. The correction from Figure 4 would be 5 % extra steam for the smaller cylinders of the replica.

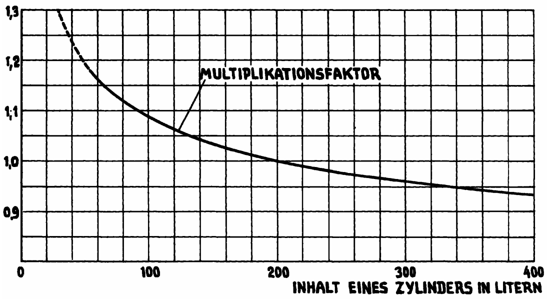



*Figure 4 Cylinder size correction [9]*

If the enthalpy is calculated, the following table can be compiled:

| | | | |
|---|---|---|---|
| Superheated steam temperature | 320 | 420 | °C |
| Enthalpy of steam, | 3081 | 3298 | kJ/kg |
| Enthalpy calculated for | 7.17 | 5.736 | kg |
| Total enthalpy of these amounts | 22088 | 18917 | kJ |
| Enthalpy ejected at 1.2 bar, 120 °C | 19461 | 15569 | kJ |
| Enthalpy ejected at 1.2 bar and 150 °C | | 15915 | kJ |
| Heat available for work from the amounts above | 2627 | 3348 | kJ |
| However, if the exhaust steam is ejected at 150 °C | | 3002 | kJ |

The 5.736 kg of steam at 420 $^oC$ would release some 14% more heat than the 7.17 kg at 320 $^oC$. This is very much in line with the approach of Westendorp[10] who quoted a 1% improvement for every 6 $^oC$ increase in superheat. As a consequence, as 5.737 kg steam at 420 $^oC$ could release 3.35 MJ of heat for work, only 6595 kg of steam/h is needed. If this is increased by the 5 % correction for the cylinder size, the steaming rate of only 67 $kg/m^2$ would need $1.05 \cdot 6695 / 67 = 103.34\ m^2$ of evaporating heating surface. This result could also be calculated from the steam tables in a straightforward fashion.

However, it could very well be that the assumed exhaust steam temperature would not be met. Instead of the 120 $^oC$, a temperature of 150 $^oC$ will be used and the amount of steam will increase to 7356 kg/h. The evaporating heating surface would become 115.77 $m^2$. The locomotive would burn some 1171 kg of coal per hour and eject some 14399 kg of smokegas.

An iteration process should now be used to calculate the amounts of evaporation area, gas throughput area and superheater area. The latter would be subject to the constraints of space since the tubes should not be too close together and should fit within the parallel boiler diameter. Also, the resistance of the tubes should be slightly larger than that of the flues to prevent incorrect preferential draughting. The gases should not prefer the tubes for the easiest passage. Finally, all of this should be designed with standard sizes of boiler tubes, flues and superheater elements.

The iterative boiler design could end with 103 boiler tubes of 40 x 45 mm, 28 flue tubes of 125 mm and superheater elements of 4 tubes of 35 mm. The total evaporating heating surface will become 111,2 $m^2$, including the 13 $m^2$ of the firebox. The superheat area would become 43 $m^2$ and the gas throughput area ratio of the flues to the total area approximately 64.6%. This ratio is needed to safeguard the superheat temperature.

The evaporating heating surface suffices if the Sanzin formula is used, predicting close to 70 $kg/m^2$.

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[7] British Railways. *Performance and Efficiency tests with Live Steam Injector*. British Railways Standard Class 4. 2 Cyl., 4-6-0 mixed Traffic Locomotive. British Railways, Mechanical & Electrical Engineer’s Department. Bulletin No.4. January 1952.

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[10] Westendorp, Prof. Ir. F. Constructie van het Spoorwegmaterieel. Delftsche Uitgevers Maatschappij. Delft 1947. pp. 160. 97 Figs. p. 26