NEWSLETTER No. 16 - MAY 2021

THE END OF THE COLLETT ERA IN 1941.

A PAINTING OF A TYPICAL GREAT WESTERN FOOTPLATE WITH THE ILLUMINATION BEHIND THE SIGHT GLASSES ON THE LUBRICATOR. SEE PAGE 44.

CONTENTS

CHAIRMAN'S PIECE
John Hind

Since I wrote my last column back in January, the world has moved on again! Apart from changing August to January, this is the same opening sentence I used in my last piece and it still applies!!!

The first change is that this is Iain McCall's first newsletter and with it comes a new layout produced using publishing software. This has come as a great relief to those of us who have struggled with producing the newsletter in Microsoft Word with all its idiosyncrasies in integrating pictures with words and getting a consistent format.

The second change is that we held our AGM using Zoom and had a record attendance, though some familiar faces whose company we enjoy were missing.

The third change is that the format for the annual conference is different again. We are going back to the 2-day format, but with one day of talk and then the following day at the Stapleford Miniature Railway (SMR), which we will have exclusive access to. Following last year's event at the SMR a number of people asked if we were going to repeat it, because they could not attend for one reason or another. So, in response to demand we are putting it on again.

Chris has already sent out advance notice of what we are planning, so to help us get down to detail planning, please contact Chris if you are interested in coming along.

FROM THE EDITORIAL KEYBOARD
Iain McCall

This is just a quick note from me to thank everybody for their kind comments since I took over the Editor's role, they have been much appreciated. The format and layout of the Newsletter will evolve over time - I am learning a new piece of software which is better suited to the production of an electronic publication than my usual hard copy outputs. Which is my way of saying that suggestions and feedback are welcome!

Contributions to future Newsletters would also be welcome!!! Please can you send them to me at advancedsteamtrust@gmail.com, the copy deadline for the next Newsletter will be September 18th. Thanks in advance!

Call for Papers

ASTT is planning to hold a

One Day Conference

to be held in the meeting room at the

Great Central Railway

Loughborough

on Saturday 18th September 2021

We expect to offer

six 45 minute speaking slots

which our members are invited to fill.

Please send a brief summary of your proposed topic to

conf@advanced-steam.org

before 12th June 2021

Note: Conference attendees will be invited to an

Informal get-together at the nearby 10¼" gauge

Stapleford Miniature Railway

On Sunday 19th September

A separate notice with programme, prices, lunch menu etc. will be circulated to members in June.

MEMBERSHIP MATTERS
Chris Newman

Upgrades to Full Membership

Six associate members have upgraded to full membership so far in 2021. They are (in alphabetical order:

Adrian Tester; Andrew Taylor; Charlie Chaligha; Nigel Thornley; Owen Jordan and Robin Pennie.

Note: Associate Members who are interested in taking a more active role in the Trust are encouraged to apply for Full Membership by notifying the Membership Secretary at memsec@advanced-steam.org. The main advantage of upgrading is that Full Members are entitled to a vote at general meetings, giving them the opportunity to participate in decision-making.

New Members

We welcome nine new members who have joined so far in 2021 (in date order):

Bob Gilbert from the USA.

Martin Anschütz from Germany. He has over 10 years experience on CNC turning machines.

Steve Rapley from the Staffordshire. Steve has a BSc in Mathematics, an MSc in Theoretical and Applied Fluid Mechanics, and a PhD on windage power loss in Rolls Royce aero engines. He is a member of I.Mech.E, an aerodynamics engineer and experienced CFD practitioner. He recently took on the role of CME with 35011 General Steam Navigation Society.

Chris Parmenter from Devon. Chris spent 26 years as marine engineer with the Royal Navy, 12 years on steam ships and 15 years as a railway maintenance technician at Laira depot. He volunteered for 16 years on the Ffestiniog as a fireman and on restoration and maintenance work and is a shareholder in ex-GWR 2-6-2T No 4160. He has also built an ultra-low-budget vertical boiler loco that runs on the 597mm gauge Launceston Steam Railway.

Iain McCall from Chippenham, who has been kind enough to take on the role of Newsletter editor for ASTT.

Jon Banfield of Templecombe, Somerest. John is a retired IT Consultant and builder of 7¼" gauge locos. He is a license holder of Solidworks which he uses to develop CAD models of locomotives. He also owns his own CNC milling machine and specialises in the manufacture of expansion links. Volunteers as a driver on the Bath & West Rly.

Dan Jones from Derby. Dan is a member of IMechE and works as a mechanical engineer for a rolling stock consultancy. He volunteers as a fireman on the Severn Valley Railway and is the engineering manager for the Stanier Mogul Fund. He is building a 5" scale model of the L&M's "Lion".

David Packer from Buckinghamshire. He is a chartered engineer and Fellow of the Permanent Way Institution. His career has been in railway infrastructure engineering, with significant business experience. He was latterly CEO Permanent Way Institution and is currently Chair of RHDR.

Robin Gibbons from Nottinghamshire. He is the author of a magnificent series of books documenting Chinese steam locomotive history – see https://www.tynedale-publishing.com/.

Non-Renewing Members

We regret that six members failed to renew their membership this year. Three declined to renew, and the other three failed to respond to renewal reminders. Of the latter, it is possible that two may yet renew.

Membership Numbers

Membership now stands at 85, broken down as follows:

Full Members:	28	UK members:	63
Associate Members:	51	EU:	13
Student Members:	7	USA	6
		Australasia:	5
Total Membership:	88	China:	1

PUBLICATIONS PAGE

Chris Newman

81 books have been sold so far in 2021. Of these, 43 were of the latest volume of Porta's papers. Sales are listed as follows:

Publisher	Author	Title	2021 Sales	Total Sales
ASTT	L.D. Porta	Porta's Papers Vol 1	10	114
	L.D. Porta	Porta's Papers Vol 2	8	109
	L.D. Porta	Porta's Papers Vol 3	43	43
	Ian Gaylor	Lyn Design Calculations	12	91
	David Wardale	5AT FDCs	3	200
	Alan Fozard	5AT Feasibility Study	0	38
Camden	David Wardale	Red Devil and Other Tales ..	0	260
	Phil Girdlestone	Here be Dragons	1	31
	Jos Koopmans	The Fire Burns Better …	1	4
	L.D. Porta	Advanced Steam Design	1	3
Crimson Lake	Adrian Tester	A Defence of the MR/LMS 4F 0-6-0	1	22
	Adrian Tester	Introduction to Large Lap Valves	1	12

BIOCOAL UPDATE
John Hind

Since the beginning of the year, a small group of us; Ian Gaylor, Richard Coleby, Chris Newman and David Pawson have been exchanging e-mails, having Zoom calls and planning practical work on BioCoal. As part of that we have been keeping Steve Oates of the Heritage Railway Association up to date and supporting him with information.

Our first step was to contact CSR to discuss reactivating trials of the University of Minnesota's bio-coal. Several railways worldwide are interested in trialling the University's product and CSR are hoping to get sufficient interest to get orders together for a large batch to bring the cost down. The big problem here is that samples are in effect produced in a laboratory albeit on large plant and the cost of a couple of tons for use on a standard gauge locomotive are beyond our means and for that matter any heritage railway. The costs are way above anticipated costs of product produced on an industrial basis. We have asked for costs for a smaller quantity that we can try out at the Stapleford Miniature Railway.

Through Ian Gaylor's connection with the Bure Valley Railway (15") we heard that they are planning to trial a number Coal Products fuels including 'Ecoal50'. Quoting from the Coal Products website:

'renewable materials make up half of the fuel are taken from plants that absorb much of the CO_2 released when burnt during their own life cycle. As the net release of CO_2 into the atmosphere is significantly reduced, Ecoal50 can claim to be one of the world's most eco-friendly coal based solid fuels. Ecoal50 burns 38% hotter and 15% longer than traditional house coal, and also produces 40% less CO_2, and 80% less smoke.'

'Ecoal50' was tried at the Stapleford Miniature Railway ($10^1/_4$") in 2019 and was not successful - it burns well, but its density is less than coal and it disappears up the chimney under heavy draught conditions!

We have an invite to attend the BVR trials and are keen to see if the result is the same as at Stapleford. If they are the same, that will validate the method of testing at Stapleford to prove a fuel before trying out on larger locomotives. The trials are going to take place in June on either a Monday or a Friday, when the railway is not open to the public. Once the summer timetable starts in July then the railway becomes so busy that there are no suitable days for testing until the autumn when it reverts back to 3 day running in mid-week.

To support any tests that we do, we have developed a test protocol, so that tests are carried out in a standardised way and we collect observations on how the fuels perform under different conditions, for instance – lighting up, standing in a station, pulling away, working on a gradient, drifting and disposal.

Chris Newman found a low carbon fuel that is burned at Uskmouth Power Station. Its carbon content is much less than coal, in addition to which it's very cost-competitive and it reduces landfill by being made of waste material. It is made by a Dutch Company called N+P, and is called Subcoal. Its calorific value is 19gJ/tonne, which compares with 30gJ/tonne for Ffos-y-Fran Welsh steam coal. N+P are interested in the use of it in the heritage sector and are looking to develop the fuel to have a design calorific value of 22gJ/tonne, which may go up to 25 gJ/tonne. As part of our discussions, we had a video meeting with N&P's UK Environmental Consultant who is making some pro bono assessments of the environmental and legislative aspects of burning this product in steam locomotives.

N&P are currently commissioning a plant to produce their product on Teesside in the UK.

They have sent us a small sample of the product (*pictured below*).

On Friday 07/05/21 Richard Coleby and Ian Gaylor trialled it on a large 5” gauge locomotive with a marine firebox owned by John Scott. These have confirmed that the product can burn in a locomotive firebox, though it does not perform as well as coal and the ash content seems high. We are expecting to see an improvement when we try the higher CV fuel.

Based on the trials we have asked N&P to supply a larger quantity to try out at Stapleford.

We have supported the Heritage Railway Association (HRA) in two ways: -

- Writing part of a paper that the HRA submitted to the Department for Business, Energy and Industrial Strategy seeking funding to support Heritage Railways now that UK coal is coming to an end.

- Took part in what was meant to be a 30 minute video call, that turned into a 50 minute one, with Steve Oates of the HRA, the Chief Environmentalist and Sustainability Office of Network Rail, the Executive Assistant to the Chair of Network Rail, the Director of the National Railway Museum and the Science Director of the Science Museum Group. The group look to have been meeting informally for some time to discuss the coal situation and how it affects Heritage Railways. Network Rail and the Science Museum Group's brief appears to be offer support and guidance to the HRA but not spend money! We were also joined by a representative from Coal Products Limited and the Director of the Energy Technologies Research Institute (ETRI) at the University of Nottingham. It was the first time I and the gentlemen from Coal Products and the University of Nottingham had joined the group. It soon became apparent that Steve Oates and I were the only ones with experience of the footplate and myself the only one with practical knowledge of how bio-coals behave in a firebox. Ian Gaylor had been scheduled to join us but was suffering from the after effects from his second jab. The outcome was that we would let them know of the dates of the Bure Valley trial and will extend our invite them, though only three of the group expressed an interest.

- We have also suggested to the HRA a conference/seminar on alternative fuels. HRA generally hold two meetings a year that are devoted to business/operational matters and alternatives to coal are discussed but very briefly, so a full day devoted to the subject would help get everyone 'singing off the same song sheet'.

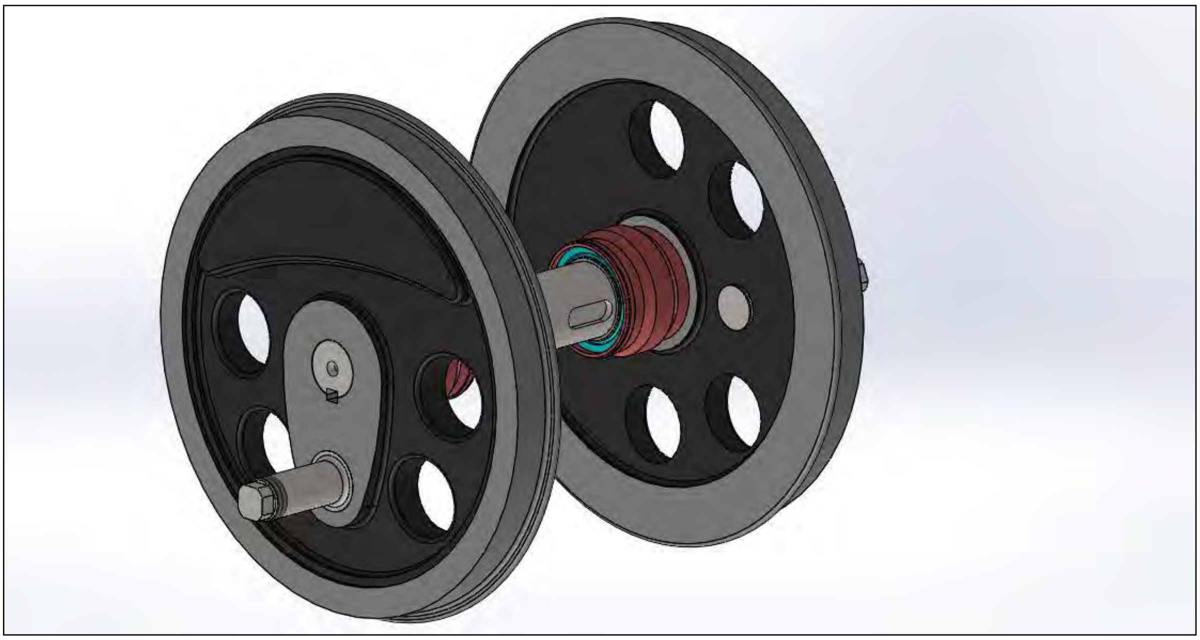
This is proving to be an interesting topic, which we are able to progress because of the varied and complementary skills and knowledge in our group.

REVOLUTION PROGRESS

Jamie Keyte

Driving Wheel Castings

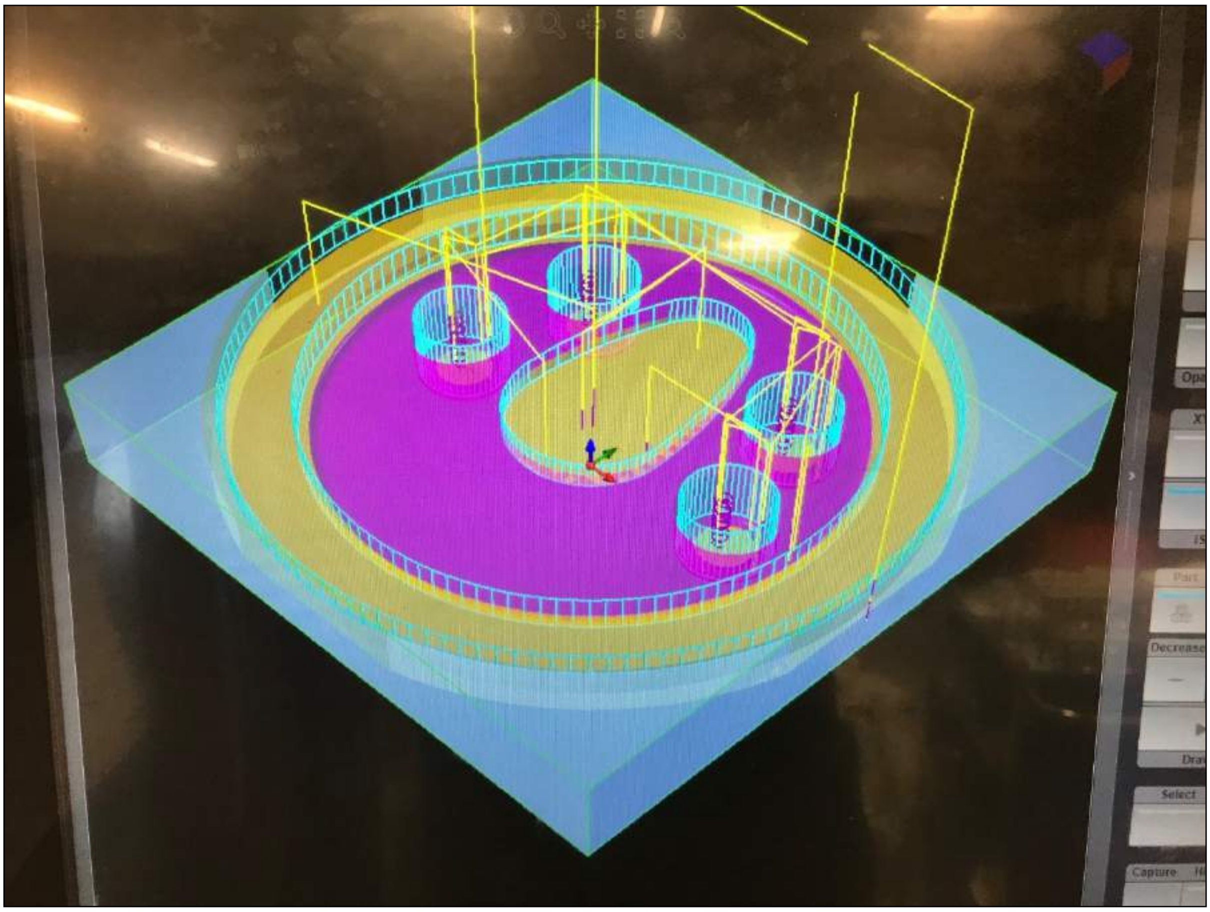
The design is based on the "Scullin" type wheel used in the USA with some revisions. Chosen to give a more modern feel to the locomotive. In full size the casting was hollow and the full disc of the inner and outer faces gave excellent support to the wheel rim. For the 10.25" Gauge "Revolution" the wheel is solid since ensprung mass is less of an issue at the normal operating speed, whereas lack of adhesive weight *is* a problem. Based on experience at the SMR, Spheroidal Graphite (or Ductile) cast iron has been chosen as the material, grade 400/15, which stands for Ultimate Tensile Strength of 400 N/mm², elongation of 15%. This is similar to a typical mild steel but is much easier to cast.



The starting point – CAD model of the driving wheelset.

A note on balancing. In this instance the balance weights don't really contribute much to the wheel mass, they are mostly cosmetic. Fine tuning of the wheel balance will be done by machining pockets in the rear face of the wheel based on a CAD weight analysis. The position of the "balance" weight relative to the crank is based on the method of reciprocating balance used on the BR 9F locos – visually this puts the balance weight on the opposite side of the wheel centreline to the more traditional method where the swaying couple is also balanced. If you want to know more there is an interesting paper by Ron Jarvis on the 9F balancing.

The pattern was cut from Acetal Copolymer on a CNC milling machine. This is nice and easy to machine and we happened to have a spare piece lying about… The pattern was designed so that the balance weights could be attached afterwards; this permitted two different styles of balance weight (leading/ trailing wheels and main crank wheels) to be fitted to a single wheel pattern.



CNC milling machine programme.

Machining the pattern.

The casting was undertaken by Canlin Castings of Langley Mill. The two centre drivers were cast initially to confirm that a sound casting would be produced. The pattern balance weight was then modified for the remaining four wheels to be cast.

The mould

(photo courtesy of Canlin Castings)

After the iron has been poured

On the milling machine – probing the part to check the position and set datums.

Casting and machined wheel.
Note that the tyre profile will be finished once the complete wheelset is assembled.
Axle and crank pin bores are finished to press-fit tolerance.

ENGINEERING OPPORTUNITY

Terry McMenamin

The LNWR George the Fifth Class, like most LNWR locomotives built from the Cauliflower onwards, had Joy's valve motion. This had the advantage of not needing eccentrics, but unfortunately was driven by a pin located near the mid-span of the connecting rod. The pin hole itself was a weak point in the rod, and at speed the motion would imposed heavy but unquantified loads on the connecting rod. In 1922 & 23 there were serious accidents with broken connection rods penetrating the boiler. Redesigned connecting rods were said to have cleared the problem, but this must have involved significant guesswork as it wasn't possible to calculate the motion forces at the time.

LNWR motion parts were far from optimised for minimum mass, which is essential to reduce the forces on the connecting rod. What we require is a form of calculation which provides the loads at each pivot point given the mass, centroid and rotational inertia of each link. This can then be used to design the links whilst trimming off all excess material. Ideally this would be an add-on to our existing Excel spreadsheet which gives link positions at 2.5° increments of crank rotation. Calculation of link velocities and acceleration can be derived and thus acceleration forces. Verification of a test case by CAD motion analysis would be helpful. Sensitivity plots for each link's mass and inertia would be desirable to help us concentrate our efforts in the most effective way.

It will be some time before we will be designing the motion, so there is no immediate time pressure for the work. It is a self-contained package and except for the verification by motion analysis requires no specialised software. The spreadsheet will require documenting to demonstrate its methods to our Acceptance Body.

Anyone interested in helping please contact me via paulhibberd@gmail.com

Thanks in advance!

COMMENT ON SOME COMMENTS

Jos Koopmans

Comment on "Some comments…" by Mr M. Johnson

2.1. Lengthened chimney

Why is the idea that a multiple orifice system can be modelled by a single one an "assertion"?

Since 1864 it was known that a multiple orifice system could give identical performance to one with a single orifice. The explanation was given by Buckingham in 1914, 50 years later and neglected by the steam locomotive trade. I.m.h.o. it is not an assertion but a fact.

2.3 Multiple orifice single Chimney

The problem described about the taper reduction was treated during the ASTT lecture with slide 21. The "chimneys" of the fourfold orifice system of the King would have a 2-degree taper resulting in an almost identical C_{pm}. My remark was that the chimney itself was far more forgiving so that the C_{pm} should be higher as the system worked properly. Since there is a video of the lecture this can be verified.

As an aside, since it is asked, I used the 4-degree taper formula in the 6023 calculations, after all, it is a physical 4 degree diffuser, but it has an internal device that takes care of a flattened exit velocity profile that equals that of the double length chimney.

3. Diffusers

My diffuser angles are based on Ells work, 1:14 taper. Any larger taper gives problems with single orifices. The CFD figure 2 was derived by using the student version of ACRI, the only one affordable back in 2004. It was thought as a section through a long channel, since axisymmetric modelling was not possible. In the case of the King, the original 1:7 chimney would be perfect for multiple orifices since the result could be thought of 4 chimneys with 1:14 taper.

3.1 Diffuser inlet conditions

During the early days of my calculations, some 20 years ago, I had come to the conclusion that the chimney entry conditions really were not that horrible. Young (figure 5) already showed reasonable velocities near the chimney wall at the throat. Figure 4 in the text under consideration supports that view, there is not much difference between the K_ds of nearly uniform and developed inlet flows.

(Note that K_d of 0.6 probably has a drawing error compared to figure 1, as also the C_{pm} of 0.45 in the graph of Macdonald.)

Since multiple orifices only diminish the peaks and valleys of the exit velocity distribution I regarded the diffuser data usable.

Given the small tapers I use I have a continuing debate with Nigel Day about his orifice inclinations. As I regard a multiple system as a bundle of chimneys the logical approach is aiming at the centres of the inscribed circles (as in figure 8).

3.2 Diffuser size

Since Mr Johnson also appears to be a model engineer, I would like to remark that I have built a live steam engine with a LBSC "Titch" boiler and also a Martin Evans "Rob Roy" which has the reputation of being a bad steamer with its original front end. Both have now a 4-orifice blast cap with holes of around 1 mm, some 50% more orifice area, and are perfect steamers. The effect is from small to large!

3.3. Excess diffusion

Indeed, the exit velocity needs to be high enough to cope with the low pressure on top of the smokebox. The SR Pacifics with their Bulleid Lemaitre exhaust are bad examples in this context.

4 Improved front end theory

Since the momentum calculation only regards entry and exit momentum, everything between those planes is not really that interesting. The point is to take care of a uniform exit velocity profile which is better arranged for with multiple orifices.

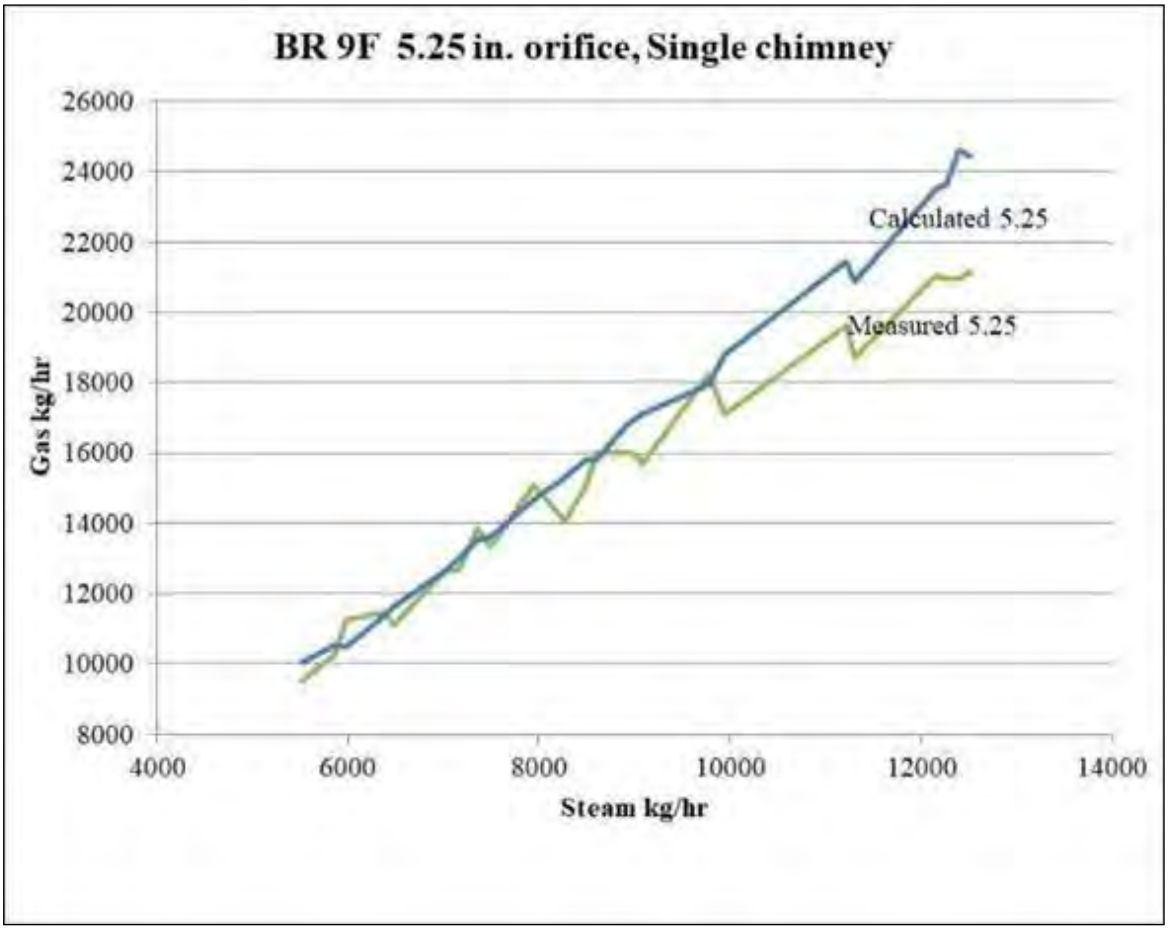
5. The Giesl

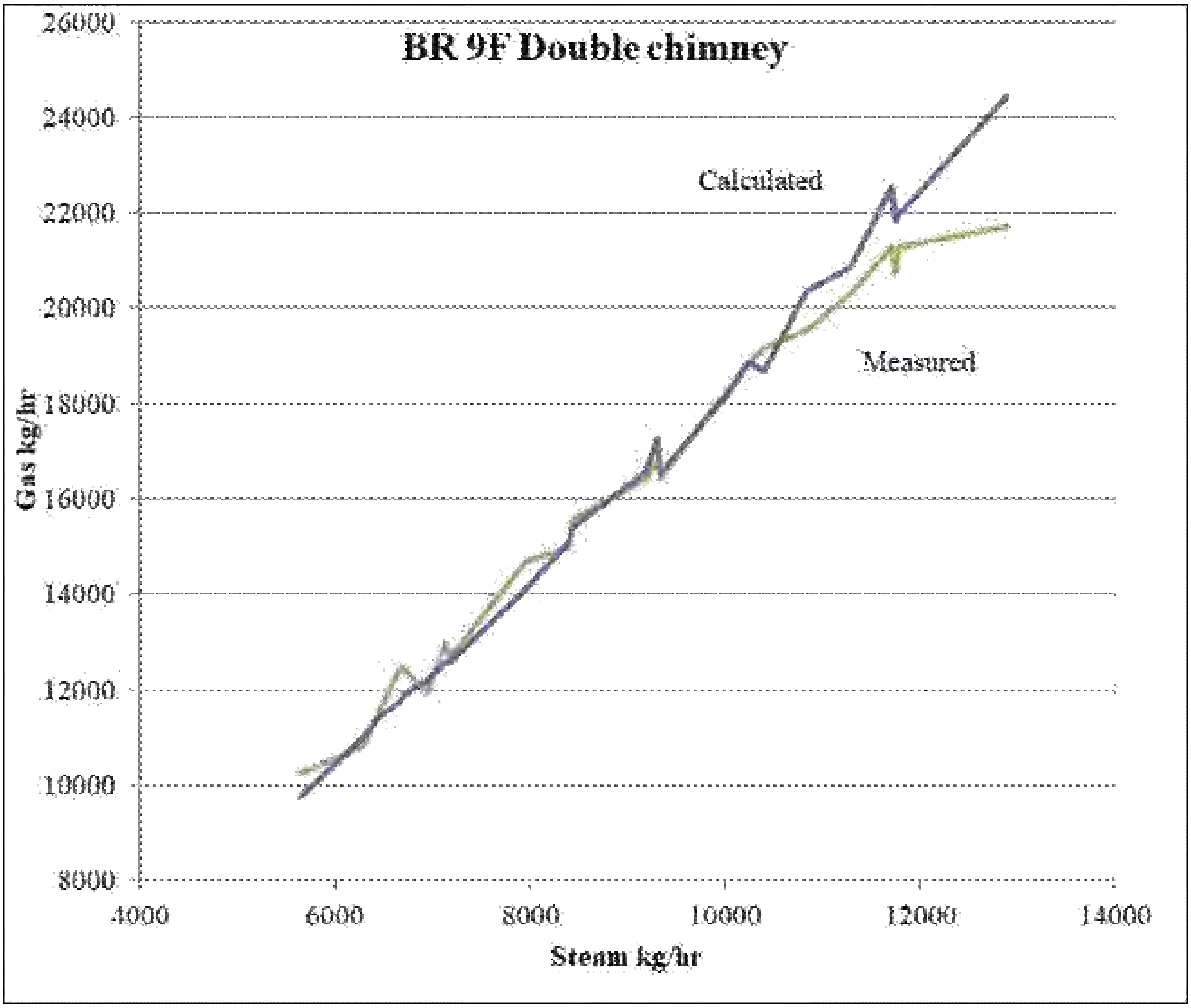
Regarding the larger nozzle at each end of the Giesl Blast cap, Giesl himself noted in one of his texts that that was because of the longer exhaust channel below the orifice. His orifices are individually adjustable so that each one supplies the same flow. If the original reference is needed I shall have to reread all Giesl's texts.

6. Conclusion

Since my calculations show so little difference between the calculated and the measured results, I wonder why this is so if they are regarded as "flawed", misinterpreted and unsound reasoning.

I gladly admit my approach is of a KISS type, but in the recalculation of the 550 or so test results a systemic error would have emerged. Of those tests almost all result in a better performance than the test results show. That is of course the effect of the simplification as all velocities are defined as uniform and also density and temperature at the exit are calculated as uniform.





The test results of the 9F are given as an example. What is calculated is only whether, with given steam flow and vacuum, the chimney dimensions would support the needed gas flow.

I would like to stress the point that most mathematical models of certain phenomena are simplified versions of reality. Even the CFD methods proposed by Mr Johnson are mathematical simplifications of the real world, as such the improved theoretical considerations of Mr Johnson will probably give little reward in accuracy of the results.

SOME COMMENTS ON LOCOMOTIVE FRONT END DESIGN PART 2

Martin Johnson

1 Matters Arising

Before starting this second part, I thought it would be useful to deal with a few things that have been brought to my attention.

1.1 Diffuser Data

Thank you to John Hind for drawing my attention to the following paper:

> Ishikawa, K & Nakamura, I. "Performance Cart and Optimum Geometries of Conical Diffusers with Uniform Inlet Flow and Free Discharge" Publ. JSME Paper No. 87-0973, Series II, Vol. 32, No. 4,pp 559-567, 1989

A free PDF file is available here:

https://www.jstage.jst.go.jp/article/jsmeb1988/32/4/32_4_559/_article
The values of losses quoted are somewhat higher than those in quoted Miller.

1.2 Diffuser shape

Robin Pennie asked whether a trumpet shaped diffuser (or indeed a tulip shape) diffuser would improve matters. The data included in Miller (Ref. 10 of part 1) suggests it would not. To summarise a couple of pages, a trumpet shape would improve things over a straight wall for diffusers in the upper region of Fig. 1, but it would be better still to reduce the area ratio and use a straight wall diffuser. It also says "the use of wall curvature is not recommended without experimental data". As I argue in the article, there is really no advantage to operating above the optimum design line of Fig. 1, with the caveat that "splayed" blast nozzles might help to suppress the flow separation shown in Fig. 2 & 3, but I would want to see test data looking at the flow pattern in the diffuser to really convince me.

However, Chris Corney drew my attention to some work being done for jet pumps in refrigeration systems. These work at high Mach numbers and Ian Eames of Nottingham University shows that by using "Constant Rate of Momentum Change" assumption a trumpet shape diffuser and mixing section results; the high losses associated with the shock transition from super to sub-sonic conditions can thus be reduced, hence improving the pump performance. Mach numbers are not quoted in the paper but from a Schlieren photograph of a similar jet pump in other material sent by Chris, the supersonic flow persists until well along the diffuser. Conversely, in a locomotive, supersonic flow will occur in the blast pipe only at steam back pressures above 8 p.s.i. and would not persist for any significant length downstream of the blast nozzle. Therefore, I would question whether the design methodology adopted by Ian Eames for very high Mach numbers is really applicable to locomotive front ends. Nevertheless, I am always open to persuasion based on evidence.

I include a figure from a literature survey that Chris forwarded to illustrate what is happening within a supersonic jet pump (Figure 1), and a comparison of conventional profile and optimised supersonic profile from Ian Eames' paper (Figure 2).

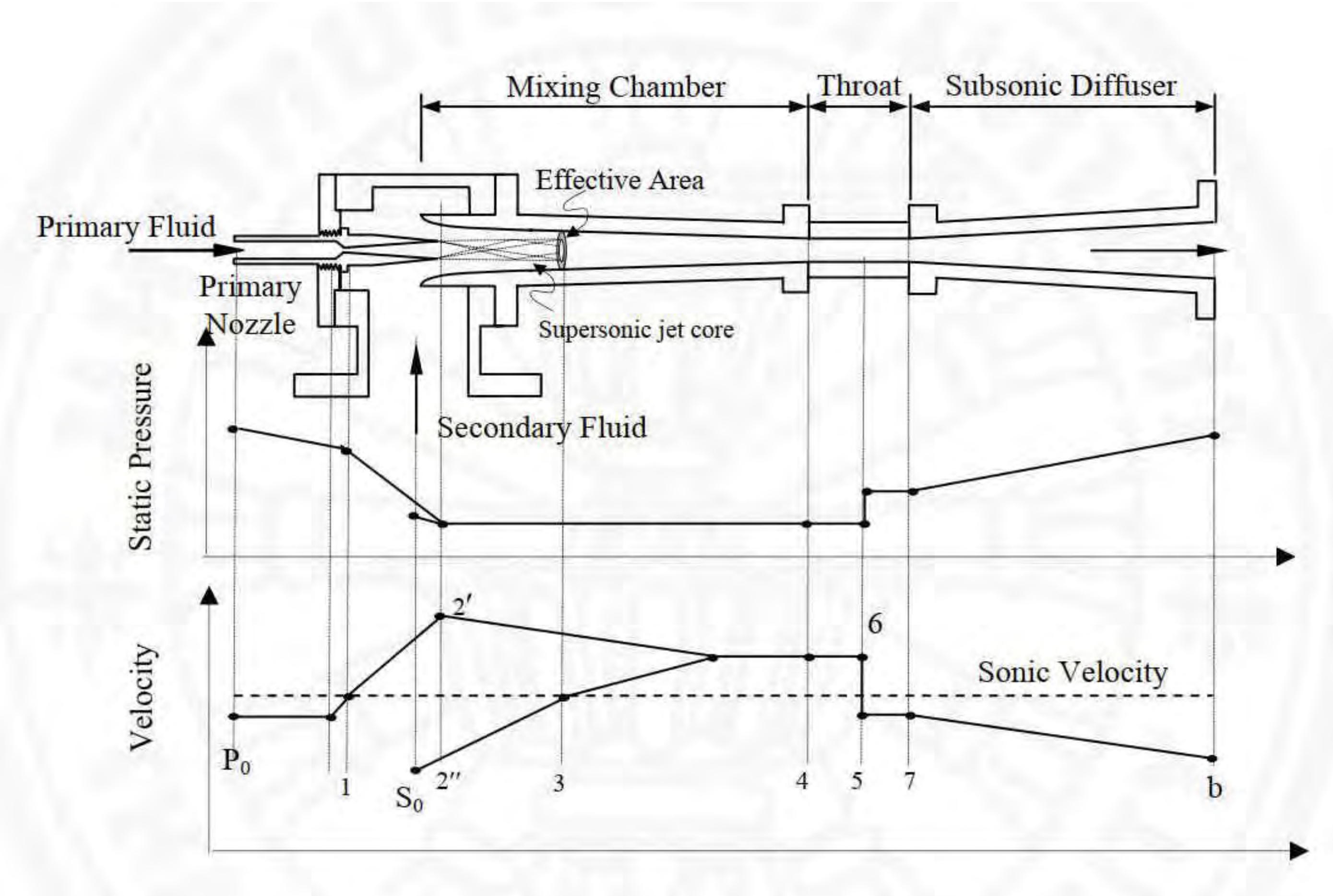


Figure 1: Profile and pressure / velocity plot through supersonic jet pump. Note sonic conditions persist up to plane 5

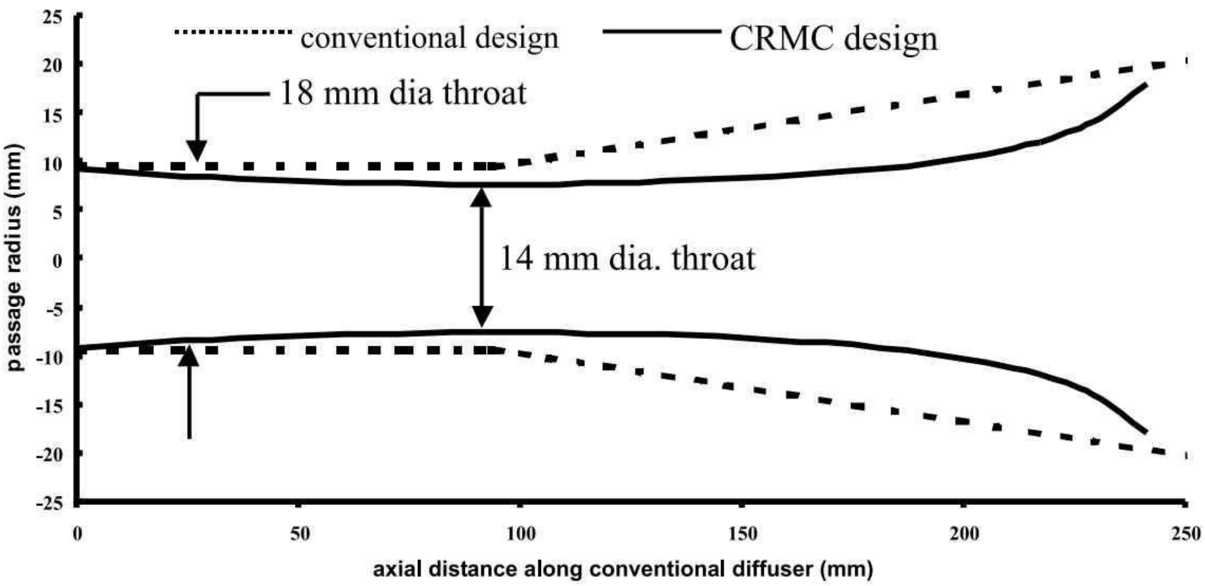


Figure 2: Comparison of conventional jet pump profile and a CRMC design.

1.3 Free Jet Shape

Chris Corney pointed out that the free jet issuing from the blast nozzle is not actually free as there is a radially inward flow of entrained flue gas which has momentum. The effect of this is to "squeeze" the jet. He also conjectured that this effect might explain the performance of front ends with "splayed" blast nozzles (See section 3.1 of Part 1 for a brief discussion of this).

To check this, I have taken a few figures from Rugby test 959 on a B.R. Class 5, as enumerated by Dr. Koopmans in his paper to the SLS in November 2017. I have assumed the radially inward velocity for the gas is based on the throat circumference (362 x Pi mm) x height from blast pipe to chimney skirt (854mm)

Blast nozzle vel.	455 m/s	Gas at throat vel.	8 m/s
Mass flow steam	2.75 kg/s	Mass flow gas	5.21 kg/s
Steam momentum	1251	Gas momentum (radial)	42

The resulting momentum vector would be at an angle of 1.9 degrees to axial. With shorter distances from blast nozzle to throat, the calculated angle would increase, so the effect would become significant in Lempor style front ends.

1.4 Koopmans Correspondence

I had a substantial e-mail from Dr. Koopmans regarding my article and submitted a 1000 word reply. We agreed on the need to keep diffuser angles in the stable zone, that many model designs show very bad draughting practice and that the whole subject is more complex than it appears.

We disagreed on his application of diffuser data (although I think the English - Dutch interface is a problem in our discussion), the application of similarity principles to model chimneys and whether his calculations provide sufficient proof of his methods.

Given the volume and detail of the correspondence, I feel it would be unfair to say much more here, until we can resolve what we do and do not agree on.

2 Some Other Detail Design Issues

2.1 Standoff Ratio & Free Jets

Bill Hall (Ref. 3.3) deduced that free jet theory would suggest that there is an optimum standoff ratio (distance from blast nozzle to skirt or throat / throat or blast nozzle diameter - depending on definition) to suit a required ratio of entrained gas to steam. Hall derived the following relationship:

The rate of increase of the volume flowrate, $V_{steam+gas}$ in terms of the volume flowrate through the nozzle, V_{steam}, is given by the equation: $V_{steam+gas} / V_{steam} = 0.44 (x /d)$

Where:

x = Distance downstream of nozzle

d = Diameter of nozzle

So this would imply that there is an optimum nozzle to chimney skirt distance if we wish all the gas entrainment to take place in the smokebox. The above equation is not valid close to the nozzle outlet, but is applicable for values of x greater than about 5d. The reason for the above equation only being valid some distance downstream of the nozzle is illustrated in following depiction of real jets is taken from Ref. 3.5 .

There are several differences between a real jet and an ideal jet, as follows:

- A real jet with a nozzle of finite diameter has it's apparent origin some distance away from the plane of the nozzle. Upstream in the case of circular nozzles, but downstream in the case of rectangular slots.
- Zone 1 in Figure 3, extends about two to six nozzle diameters (for compact and radial jets) or slot widths (for linear jets) from the diffuser face. In this zone, the peak velocity at the centreline of the jet remains nearly equal to the nozzle velocity throughout its length. (Free jet theory suggests velocity is inversely proportional to distance)
- Zone 2 is a transition zone, and its length depends upon the nozzle type. The transition zone typically extends to eight or ten diameters from the outlet. Within this zone, the maximum velocity may vary inversely with the square root of the distance from the outlet. (Free jet theory suggests velocity is inversely proportional to distance)
- Zone 3 is the zone of fully established turbulent flow. In this zone, the jet does behave generally as the free jet model. The length over which this zone persists does depend on the nozzle design, and the turbulence of the entrained gas.
- The measured divergence angle for a real circular jet is 24 degrees, compared with 20 degrees as calculated by Hall Ref. 3.3 from free jet theory. The often quoted 1 in 3 rule equates to 19 degrees included angle.
- There can be significant differences in flow pattern depending on the nozzle design, which free jet theory would not anticipate.
- A real jet issuing from a rectangular slot has a wider divergence angle than from a circular orifice.

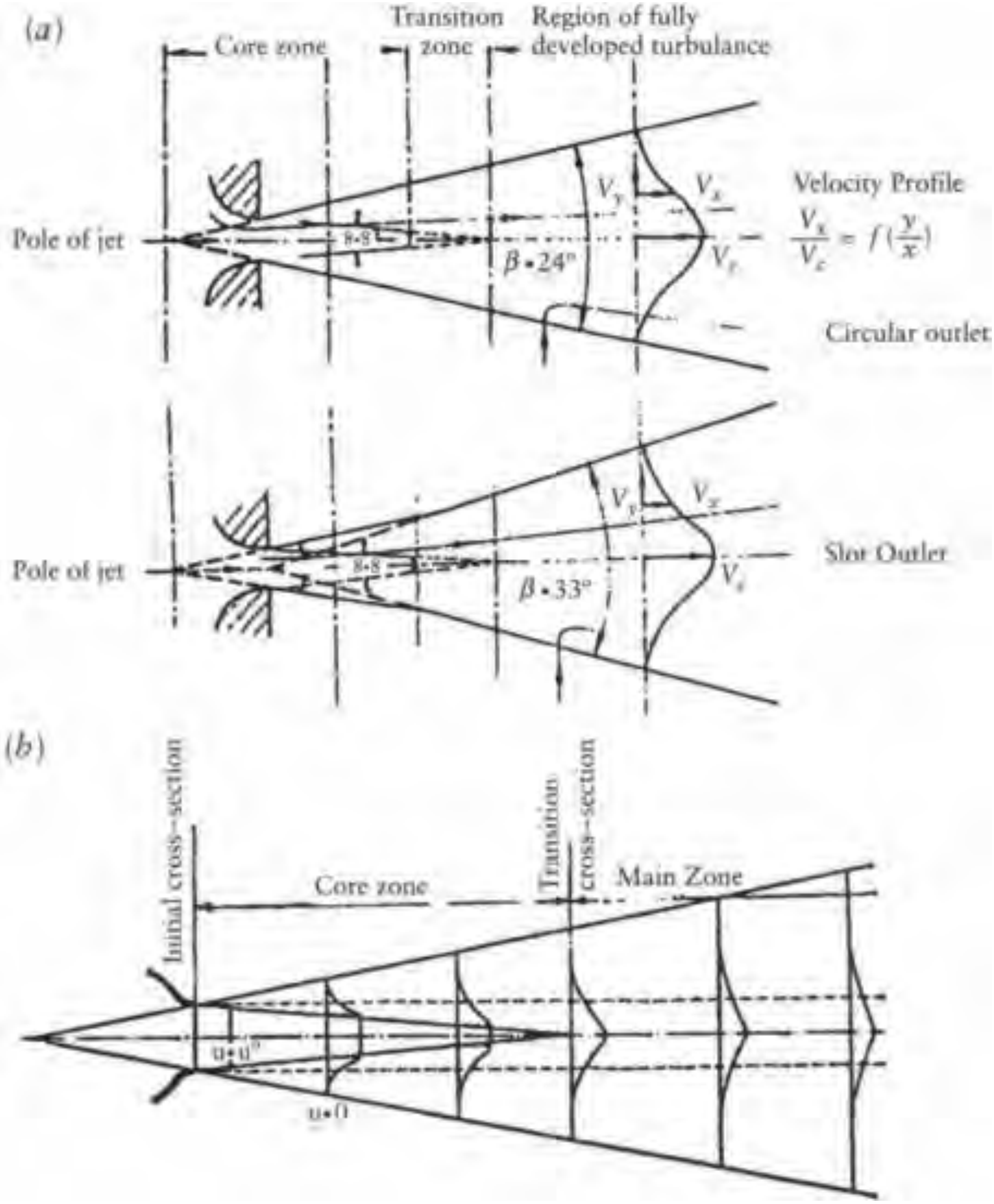


Figure 3: Diagram of flow pattern in real jets (a & b) and a simplified model (c)

The wider divergence angle for rectangular jets shown in Fig. 3b is interesting; rectangular jets diverge at 33 degrees included angle against 24 degrees for a circular jet. So if one accepts that there should be a fixed ratio of blast nozzle to chimney for a given entrainment ratio, as Hall demonstrated, this ratio would also depend on the shape of nozzle. This would imply a locomotive front end with rectangular blast nozzles would need a smaller standoff ratio than an equivalent with circular blast nozzles. That would allow more mixing and diffusing length within a given height restriction, perhaps giving greater efficiency. Young (Ref. 3.4) tested two rectangular nozzles, but due to simultaneous changes in other experimental parameters, and failure to shorten the nozzle to chimney throat distance in line with observations given above, they may not have been fairly compared to other designs. This really underlines the complexity of optimising front end design - too many variables.

There is a further consideration in real locomotive practice in that total height of the blast pipe + standoff + mixing chamber + diffuser assembly is limited by the loading gauge. When performance against the <u>total</u> height was investigated by Young (Ref. 3.4 See Fig. 27) a very ill defined optimum was observed, and for larger total heights a zero value of standoff appeared the optimum which reflects the approach adopted by Porta (Ref. 3.2), E.S.D.U (Ref. 3.1) and others. Note that Young's results go up to a standoff ratio of 6.67, so do not really cover the regime where the chimney skirt is failing to receive all the flow from the jet. Experience shows that an excessive standoff ratio (above around 5 to 6) can give very poor draughting.

Another practical consideration is access into the smokebox and tube cleaning. Traditionally, the chimney skirt was at around the level of the top row of tubes or flues, and the blast pipe arranged to suit, but usually only blocking a few tubes. If we look at the front end of the "Red Devil" for example, the chimney diffusers occupy a significant area of the tube bank. This is of course excellent for providing good diffusion and draughting performance. However, the opinion of shed staff tasked with tube cleaning etc. on such a set up seldom gets recorded - but I think they might prefer the "traditional approach".

2.2 Blast Nozzle Performance

Front end theory as usually presented says very little about blast nozzles. So which of the tapered or orifice versions as seen in Figure 4 would be better? The tapered version on the left is obviously more streamlined but a deeper investigation is warranted.

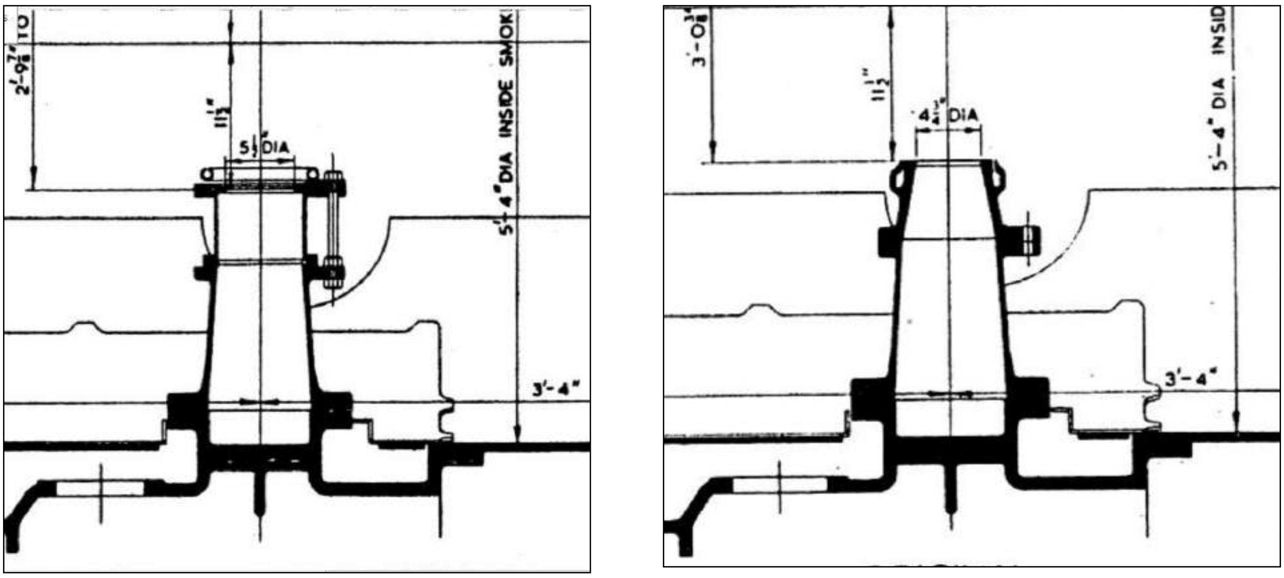


Figure 4: Tapered blast nozzle (left) and orifice blast nozzle (right)

Nozzle performance (and often valves as well) is usually described in terms of a C_d value, which is defined as:

Actual discharge / Theoretical discharge at a given pressure (or enthalpy) drop across the nozzle.

A gently tapered nozzle ("streamlined") would usually have a lower C_d than a short nozzle and one might expect a better performance as a blast nozzle, but this is not necessarily so. To understand why, we need to look at the detailed losses in a nozzle.

The C_d value is the product of two corrections:

- An area correction for the area of the *Vena contracta* compared to the actual throat area of the nozzle. (C_c)
- A velocity correction for the actual velocity in the *Vena contracta* compared to the ideal velocity in the *Vena contracta* (C_v)

What is a Vena contracta?

A nozzle is usually tapered and as the flow accelerates, the outer streamlines must follow that taper. When the flow leaves the nozzle, the streamlines cannot change direction abruptly so the flow continues to reduce in area after leaving the nozzle. The Vena contacta is the point at which the flow occupies a minimum cross section - and hence has maximum velocity.

A sharp edged orifice with a large taper angle will have a vena contracta significantly smaller than the orifice - as low as 62.5 %. A long slowly tapering nozzle would have a vena contracta very nearly equal to the nozzle size.

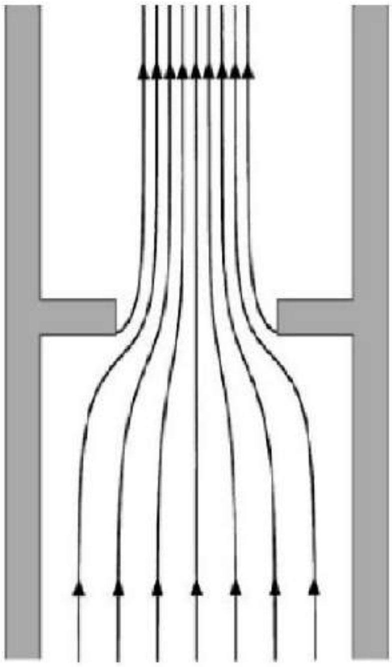


Diagram showing a sharp edge orifice. Note the flow area at outlet is less than the orifice size.

It is rare for the above two corrections to be separately quoted, but the distinction is very important in relation to locomotive blast pipes. The velocity correction C_v is a measure of energy lost as low grade heat from friction. In typical short nozzles, this will have a value in the range 0.97 to 0.99, so 1% to 3% of the incoming energy is completely lost to the system and serves only to increase temperature and enthalpy while decreasing the density and hence increasing the velocity of the outgoing flow.

The C_c correction determines the effective size of the nozzle. Thus, if C_c is 0.9 (a fairly typical value), the flow will be passing through an area only 90% of the orifice opening, and hence will have a velocity 1/0.9 ≈ 1.11 times what it would otherwise be at a given pressure.

The important point in relation to blast pipes is that the "extra" velocity energy is not a loss, as it will be used in exchanging momentum with the entrained flue gases. So the only worthwhile design is one that maximises C_v (minimum surface area for skin friction), even if C_c is reduced. Nozzles with a low C_c factor may need to be slightly larger than a nozzle with a high C_c factor, but that is an easy design modification to make.

There are also practical points in favour of an orifice nozzle:

- The orifice plate nozzle is a very simple component - readily made or modified at low cost, so more likely to be optimised for performance.
- The orifice plate nozzle can be made shorter, allowing more height for mixing and diffusion stages.

3 References

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3.3 Hall, W. - "Front End Design" Private Communication (3/8/1999)

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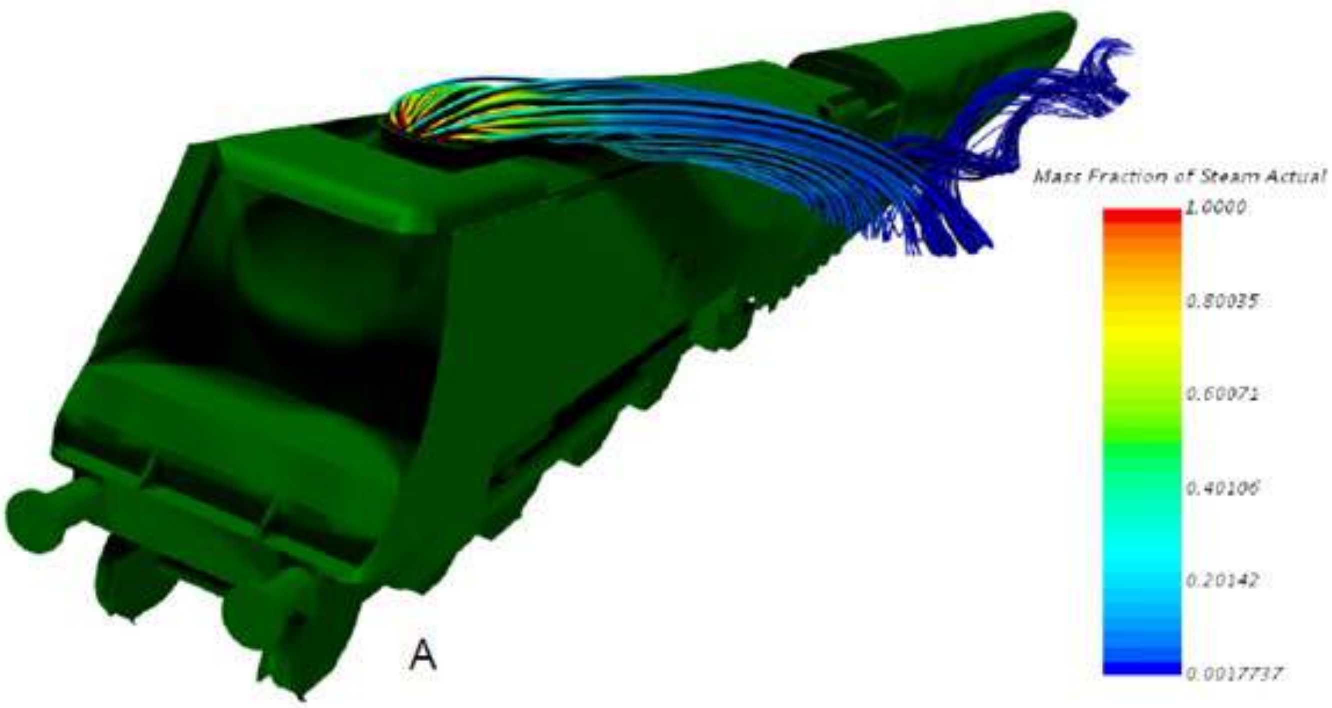
EXTERNAL AERODYNAMICS OF A MERCHANT NAVY PACIFIC A CFD STUDY

Steve Rapley



1. Background & Introduction

The General Steam Navigation CIC are custodians of BR Rebuilt Merchant Navy Pacific locomotive 35011 *General Steam Navigation*. 21c11 "General Steam Navigation" was the first of the second batch of ten members of the Merchant Navy class, entering service on 30th December 1944 to a design developed from the first batch, with improvements to smoke clearing added, extending to the modification of the cab from the 'flat front' to the 'wedge' style to give improved forward visibility for the crew in September 1950.

All members of the Merchant Navy class locomotives underwent rebuilding with no 35011 being one of the last 6 locomotives in original condition. This process included removing air smoothed casing and the oil sump, replacing the Bulleid chain-driven valve gear with three sets of Walschaerts gear, manufacturing a new middle cylinder, reverser, smokebox, ashpan and grate (the boiler, frames, outside cylinders, and wheels were all retained). 35011 re-entered service on 4th July 1959 having run 670,782 miles in original form, running a further 398,346 miles before withdrawal from service in February 1966. After a brief spell in store at Stewarts Lane it was moved to Eastleigh Works where the centre crank axle was swapped for a plain axle, the crank axle being fitted to 35026, and our locomotive was sold to Woodhams Brothers in Barry in March 1967. 35011 was purchased for preservation and left Barry in March 1983, after many years and moves around the country ownership of 21c11 / 35011 was transferred to the current General Steam Navigation CIC in 2016, and the loco was moved to Blunsdon station on the Swindon & Cricklade Railway in 2019. Due to the expected costs of replacing the missing crank axle and valve gear being similar whether the Walschaerts or Bulleid chain-driven valve gear was fitted, the project to recreate an original Merchant Navy complete with the original air smoothed casing, chain driven valve gear and a replacement middle cylinder was started.

To introduce myself, professionally, I am a Mechanical Engineer, specialising in Thermal-hydraulics and Aerodynamics. I have a life-long interest in railways, especially the British Railways Southern Region, nurtured and encouraged in that respect by my late father. I was appointed Chief Mechanical Engineer for the General Steam Navigation CIC in September 2020, so as a way of drumming up interest in our project, I approached contacts I have at Loughborough University about conducting some CFD studies of the external aerodynamics of a Merchant Navy, to see if we could improve the issue of drifting smoke with the design. After conversations with Professor Versteeg, this idea grew into two projects, looking at the internal and external aerodynamics of a Merchant Navy, using Computational Fluid Dynamics (CFD) to place our locomotive in a virtual wind tunnel. Our goal with these projects was to explore (1) conditions in the smokebox and (2) the external exhaust clearance, with the possibility of making improvements on the design. In this article I'll give a condensed version of the report from the external aerodynamics study and hope to give details on the internal aerodynamics project in the future.

2. Literature review

The known issue of exhaust drift on the MN class locomotive was also common on other classes of locomotive. Smoke deflectors were therefore introduced, with the function of "promoting a flow of clean air along the boiler sides or lifting the smoke by a vertical air current at the chimney" (Peacock, 1951). The concept of streamlining the locomotive body proved beneficial in "preventing the spread of the smoke" (Holcroft, 1941), using "a pair of longitudinal plates to form an open duct on top of the boiler casing". Another design strategy used involved adding a "flume" at the front of the locomotive to channel free air through the front and out through a "passage" behind the chimney, subsequently causing a flow of high pressure air underneath the exhaust chimney which raised the plume above the cab. These modifications have been proven to improve the visibility of the driver but have adversely affected the original aesthetics of the locomotive. From studying videos of locomotives, cold conditions (below 3°C) and banking on either side of the track (where flow is funnelled) cause an increase in probability of obscuration. However, despite clear, colour videos of the MN locomotive, particularly when exhaust smoke caused an issue not being available, valid comparisons can still be made.

3. Group Work

a. Group Problem Definition and Objectives

From the information received from GSNLRS, a series of objectives were defined based on the requirements of the company and potential project outcomes. These are shown below:

• Based on data provided by GSNLRS, develop a geometry model of the MN locomotive, making suitable approximations to enable high-quality CFD predictions.

• Develop boundary conditions for different engine running speeds and varying exhaust production rates.

• Develop a CFD simulation of the air flow over the locomotive, with the capability to model simple buoyant plumes.

• Decide a validation strategy to build up the scientific credibility of the simulations, helping characterise driver obscuration.

• Develop post-processing tools for high-quality result visualisations.

b. CFD Group Simulation Design

i. General Description of CFD Problem and Validation Test Case

Investigation into the condition(s) that cause obscuration of the driver's view were carried out, with obscuration being validated using the mass flow rate of steam present on a volume in front of the cab, along with sufficient reduction of residuals and the surface average on the driver's view plane. The computational domain used is shown in Figure 1.

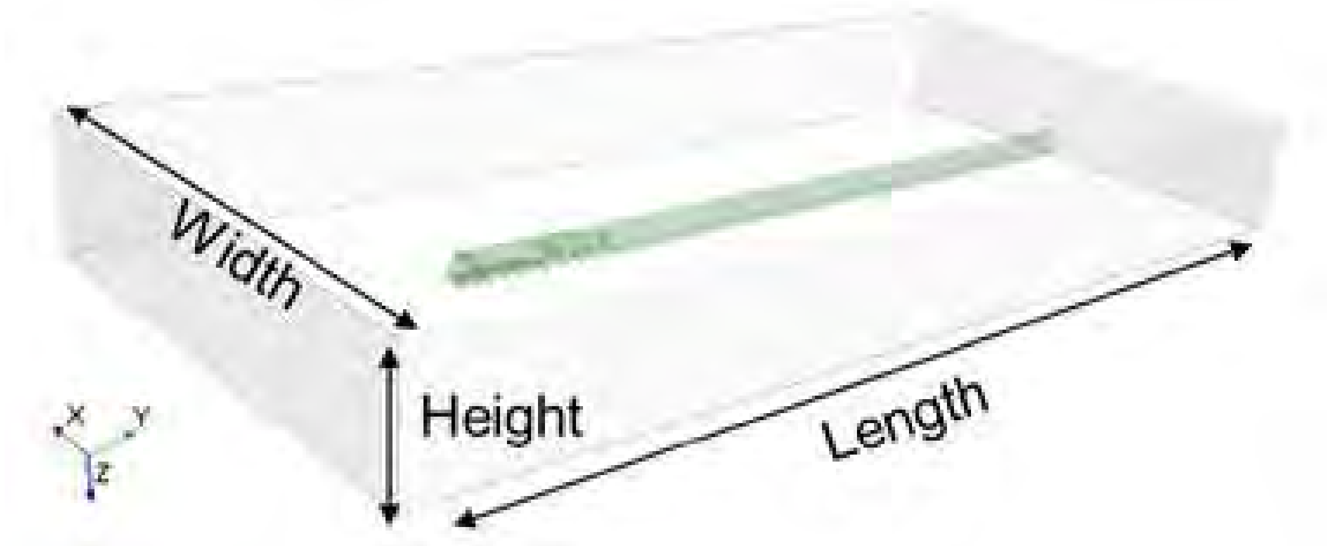


Figure 1: Domain used for to capture the full external flow of the locomotive

ii. Modelling Assumptions

The simulation model investigated the obscuration of the drivers view. In order to optimise total computational time, several assumptions can be made:

- Effect of rotating wheels and coupling rods beneath the locomotive are assumed to have a negligible effect on obscurant flow. To include these moving components, an overset mesh could be used however, with this simulation it was not used.
- Constant Mass Flow Rate of Steam, 7.5 lb/s (3.4 kg/s).
- Uniform and Constant Crosswind and Forward Velocity.
- Constant Ambient Temperature: 53.33 °F (285 K) - observations from videos of Bullied Locomotives in the winter showed increased obscuration.
- Constant Steam Outlet Temperature: 252 °F (395 K).
- Low Turbulent Energy of Velocity Inlet flow: Due to the Front and Left Boundary faces of the control volume being velocity inlets, the turbulent energy of the inlet flow is low. In reality the flow would be turbulent and would be a result of the effects of the surrounding environment.

iii. Physics Models

Due to the cyclic nature of a steam engine, the obscurant flow rate from the chimney is inherently unsteady; With this simulation model however, a steady state model is used to reduce the total solver time and to generate solutions within a tight time scale. Furthermore, in videos of the phenomenon, by the time the steam-air mixture from the chimney has reached the driver's window and is obscuring the view, the flowrate of the obscurant appears steady. The coupled flow physics model was selected over segregated flow as the flow physics model, due to preliminary simulations giving more steady convergence. Multi-component gas and gravity were selected to allow the steam-air obscurant mixture from the chimney to be tracked and analysed as a single species; gravity allowed buoyancy to be modelled – a critical phenomenon when dealing with the turbulent gas flow from a steam locomotive chimney.

iv. Mesh Design

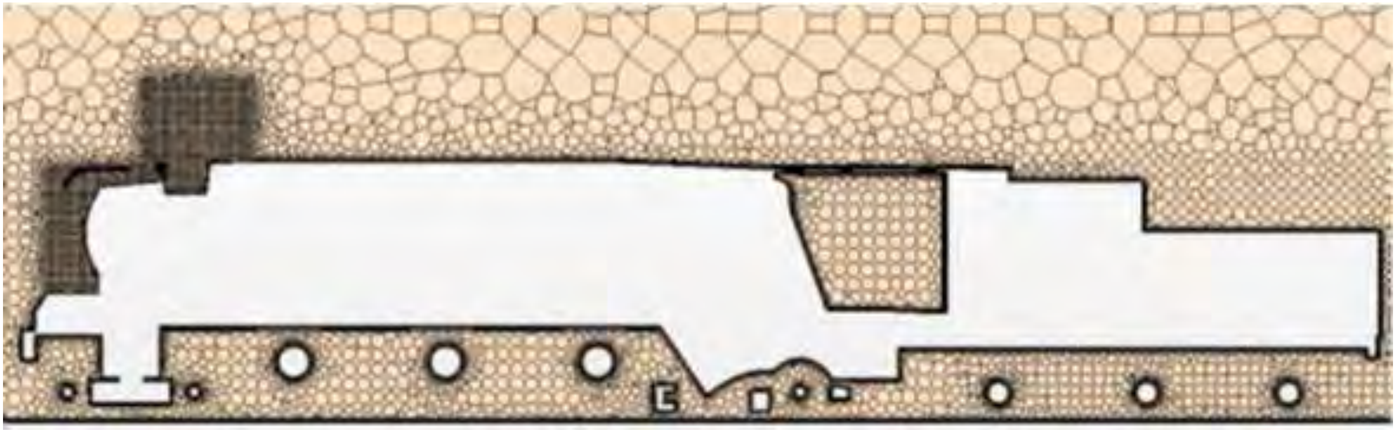


Figure 2: A 2D mesh scene of the final base simulation mesh, generated on the mid plane of the locomotive

After several major redesigns and parameter changes, the final mesh was decided to be viable for the given CFD problem. A 2D mesh depiction is shown in Figure 2 and contains 2,329,939 cells, 11,212,796 faces, and 7,549,117 vertices. Wall Y+ values (an important metric for determining the resolution of the near-wall flows) can be seen in Figure 3.

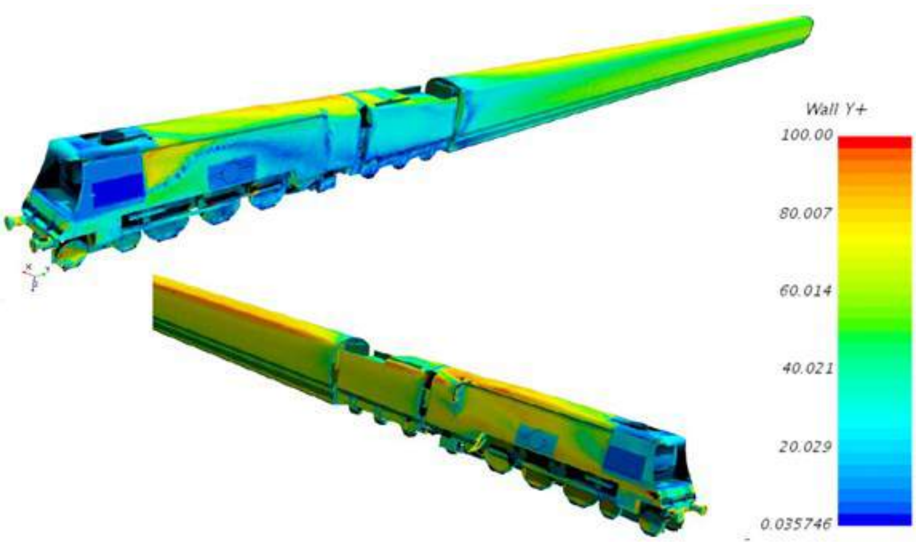


Figure 3: Y+ scene for the final group base simulation with a scale up to a maximum value of 100

The Y+ values seen are mostly below 100. There are some significant areas above 100 on the windward side of the locomotive; though convergence may be affected by these high wall Y+, values for the areas around the exhaust and obscuration volume on the leeward side of the locomotive the wall Y+ values are well within the required limits. This mesh therefore satisfies the criteria defined.

c. Obscuration Condition

To define whether the driver's view is obscured or not, a derived part placed on the leeward side of the locomotive was used to represent the view of the driver and quantitatively monitor the exhaust flow passing through it. The monitor was created from a small, rectangular volume (8.51 m length, 1.95 m width, 1.97 m height) in order to capture the unsteady nature of the view under working conditions. Assuming the models will be underestimates of the obscuration caused by the flow, any steam captured within the volume will be considered as causing an obscurity.

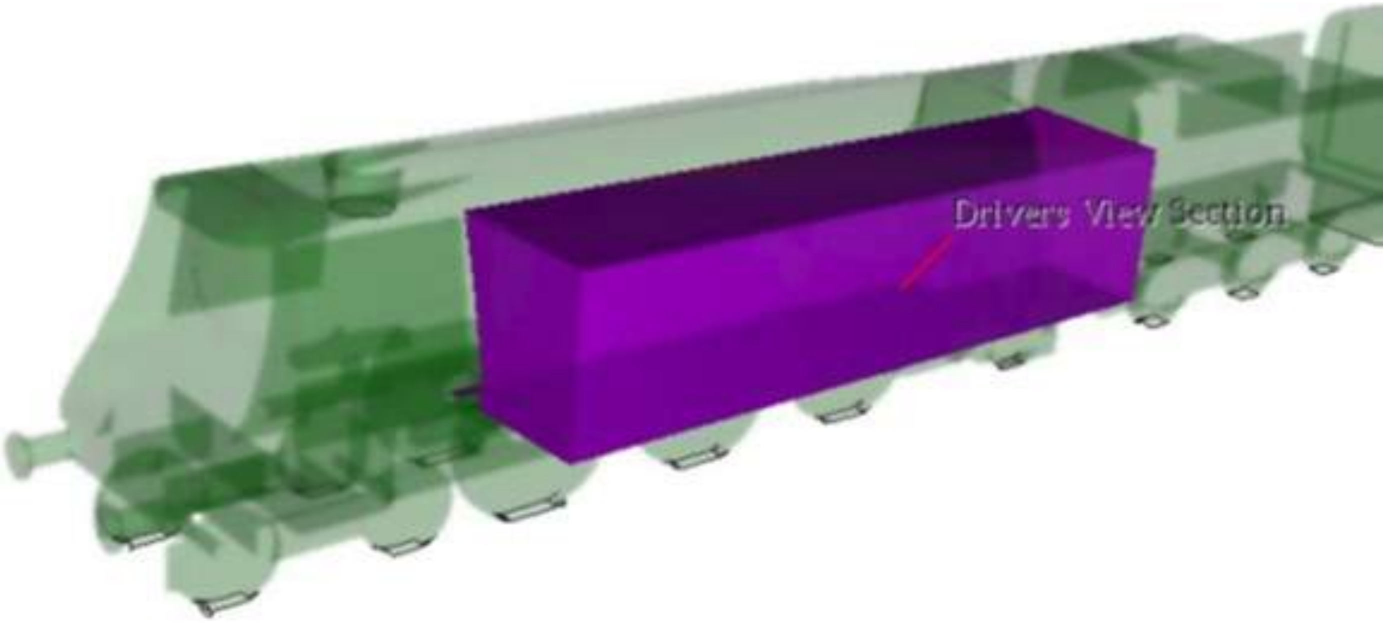


Figure 4: Obscuration block shown in relation to the locomotive

d. Results and Discussion

i. Observed Flow Behaviour

Aside from the high number of iterations to achieve convergence, the exhaust flow was well captured in the base simulation, agreeing with the expected flow from literature and observations of locomotives in action. Scalar fields of mass fraction of steam or velocity were deemed most appropriate to visualise the flow, as these parameters determine the behaviour of the steam flow most significantly on exit of the exhaust.

The exhausting flow appears to dictate the behaviour of the immediate surrounding air flow. Away from the exhausting fluid, the streamlines of air follow a path that converges to the RHS of the domain (from the view looking down the length of the locomotive). This is expected from the velocity boundary conditions applied. The air streamlines immediately around the exhausting plume however are pulled into the vortex formed. It is further observed the exhausting plume influencing flow further than in the immediate surroundings as an air streamline is pulled in the opposing direction of expected flow into the vortex. The main characteristic identified from the exhaust flow, in the area of interest, was that it closely follows the locomotive body, and the recess around the cab window was a hotspot for entraining exhaust steam, suggesting this region is at a lower pressure compared to the upstream flow. Aside from the momentum provided by the high-speed flow of crosswind air, this low exhaust exit momentum may be explained by the circular geometry of the original exhaust, which results in exhaust steam exiting the exhaust in a disorganised and uneven fashion, without much in the way to route the flow either higher above the locomotive or outboard.

In Figure 6 the mass fraction of steam which travels alongside the upper edge of the locomotive is around 0.3, and this trail of steam fails to mix significantly with the air. Regions with high negative gradients in mass fraction of steam also mean that any steam mixing with air has a greater condensation effect in the surroundings, which would contribute to the production of obscurant and subsequently worsen vision from within the cab as more heat energy is transferred from the hot steam to the surroundings, condensing the steam into water vapour due to the enthalpy loss at the lower environmental temperature and developing a denser, wider cloud of obscurant which then continues to disperse around the locomotive.

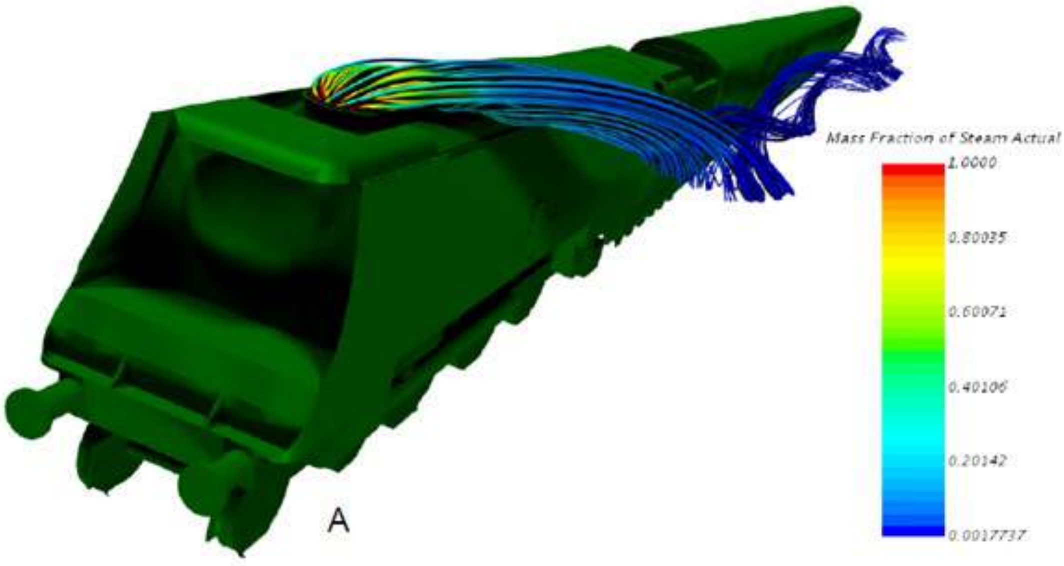


Figure 5: Isometric view of the mass fraction of steam present in the exhaust plume that runs parallel to the locomotive. From the base simulation for the 80-20 condition.

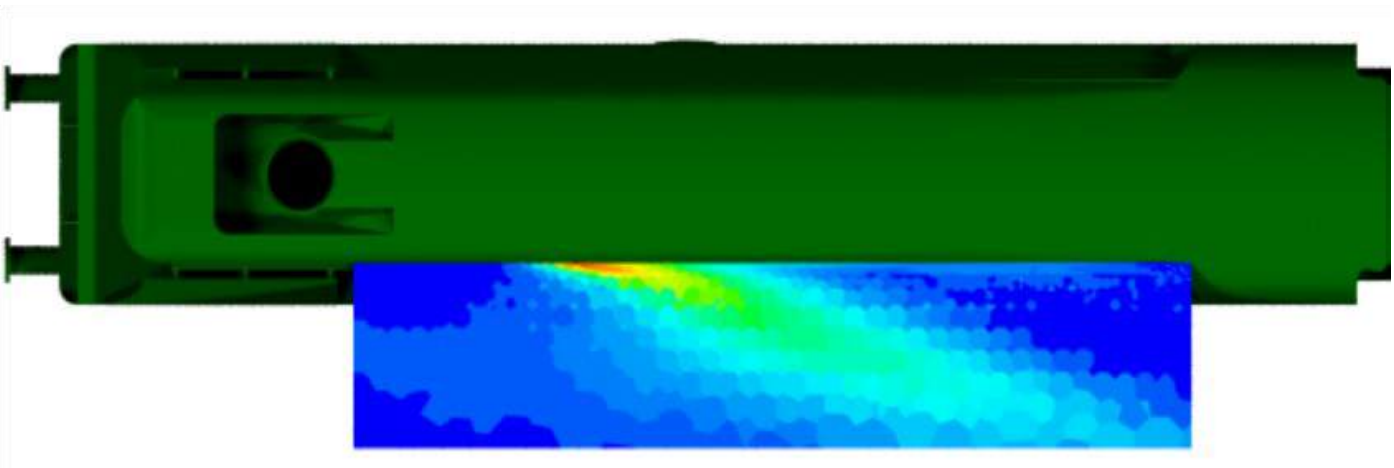


Figure 6: Top-down view of the locomotive, showing the mass fraction of steam present on the obscuration block (scale from 0 to 0.3).

The simulation was run at 11 other predefined forward speed and crosswind conditions, where the flow of each simulation was examined in a similar manner to look for obscuration. It was found that two conditions appeared to present obvious obscuration and could be used as the basis for the individual studies. The results indicate obscuration was present around the cab window of the locomotive most predominantly at the forward speed of 80 mph and at crosswind speeds of 10 and 20 mph. Some obscuration was also present at 50 mph forward speed and a crosswind speed of 40 mph. However, compared to the higher forward speed conditions, the level of obscuration was negligible as a larger fraction of steam was diverted away from the locomotive and therefore this condition was discounted. For the individual tasks, the 80_20 condition (80mph forward speed, 20mph crosswind) was selected, as this condition gave the greatest level of steam obscuration and therefore any differences in flow because of any modifications could be distinguished with ease.

4. Individual tasks

The individual tasks aimed to investigate several different modifications to the base simulation. With positive obscuration measured at a locomotive speed of 80mph and a crosswind of 20mph the individual simulations will focus on the modifications impacts at these speeds. Individual tasks that made modifications to the locomotive geometry are as follows:

- Analyse the effects of modified ducts on obscuration and exhaust flow characteristics.
- Analyse the effects of the following exhaust types on exhaust flow and obscuration:

◊ 5, 6 and 7° Conical diffuser,

◊ 6° Oblong diffuser,

- Analyse the impact of deflectors on reducing obscuration.

The geometry modifications were suggested due to their feasibility to be implemented upon the locomotive during its restoration. To validate the steady-state assumption from this study, the base simulation was also modified to:

- Analyse transient flows from the exhaust.

The individual simulations were run with identical physics settings and values to the original geometry base simulations to increase the validity of comparison between simulations. Solver settings were altered for each simulation and the changes will be outlined in the relevant section.

a. Modified ducting

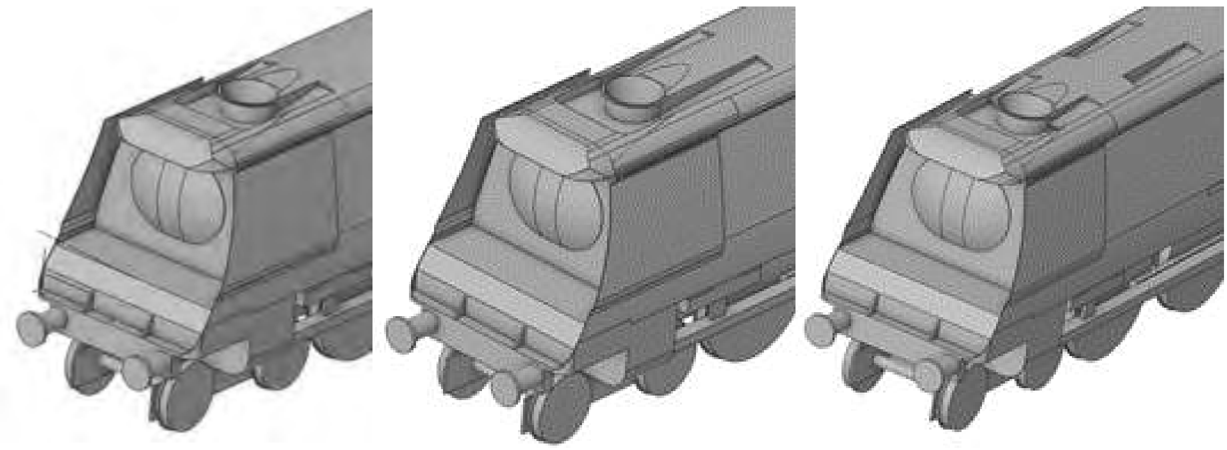


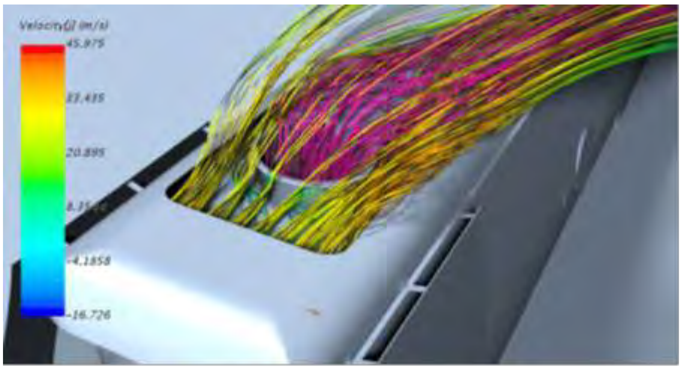
Figure 7: Original, extended and divided duct geometries

The first feature that was modified on the locomotive was the ducts. The ducts are located at the front of the locomotive and help to channel and accelerate the incoming air towards the top of the locomotive. This in theory should create an upwards flow after the exhaust, reducing the chance of the plume collapsing over the side of the locomotive into the driver's view. The flow observed in Figure 8 is from the 80_20 speed condition, where positive obscuration was determined. The flow is seen to initially stagnate around the inlet to the duct and then accelerates back to the free stream value. The flow is then channelled through the duct with a slight incline to produce vertical velocity. Extending the ducts rearward, as shown in Figure 9, creates a relatively condensed exhaust plume, with little diffusion down the side of the locomotive when compared to the original and divided ducts. In addition, the extended ducts simulation clearly shows greater separation between the plume and the top surface of the locomotive. This indicates that the extended ducts are more effective in 'lifting' the exhaust flow at this speed/crosswind condition.

The streamlines in Figure 8, Figure 9, and Figure 10 shows the flow from the ducts and its interaction with the exhaust plume. While all three designs show very similar 'upward' or 'z-component' velocities at the outlet of the duct (ramped portion), the position of the extended duct outlet causes the exhaust plume and duct flow to intercept providing a reasonable explanation for the greater effectiveness for the extended ducts at this condition. The flow through the divided ducts appears similar to the original design, except with a relatively small amount of flow reaching the second outlet. This creates a small wake behind the second outlet, drawing the exhaust plume down and appearing to cause a more violent vortex in the plume. This disturbance to the flow across the top of the locomotive could also explain the wider and more diffused exhaust flow.

While the simulations indicate a minor improvement to obscuration (-10.5 %) when using an extended duct at this condition, obscuration is still largely present. The divided ducts cause a potentially worse obscuration condition (+4 %) due to the limited flow through the second duct. In conclusion, it is certain that changes to the ducting have measurable impact on the flow direction and concentration.

Figure 8: Original Duct Geometry Streamlines



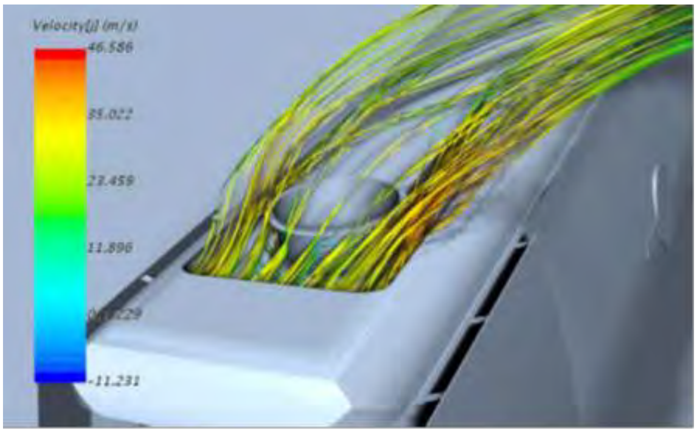


Figure 9: Extended Duct Geometry Streamlines

Figure 10: Divided Duct Geometry Streamlines

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b. Exhaust Modifications: 5° Diffuser

The second of the individual tasks looked at the impact of introducing an extended diffuser, of a shallower angle to the original Bulleid-Lemaître, as would be introduced if something akin to a Lempor was introduced. I have some disagreement with some parts of the flow analysis the student produced, as I think they have misunderstood what is going on in the simulation in this case, so I present my own for comment, in italics. Text in this section that is non-italicised is from the original report, and I am in agreement with the interpretation.

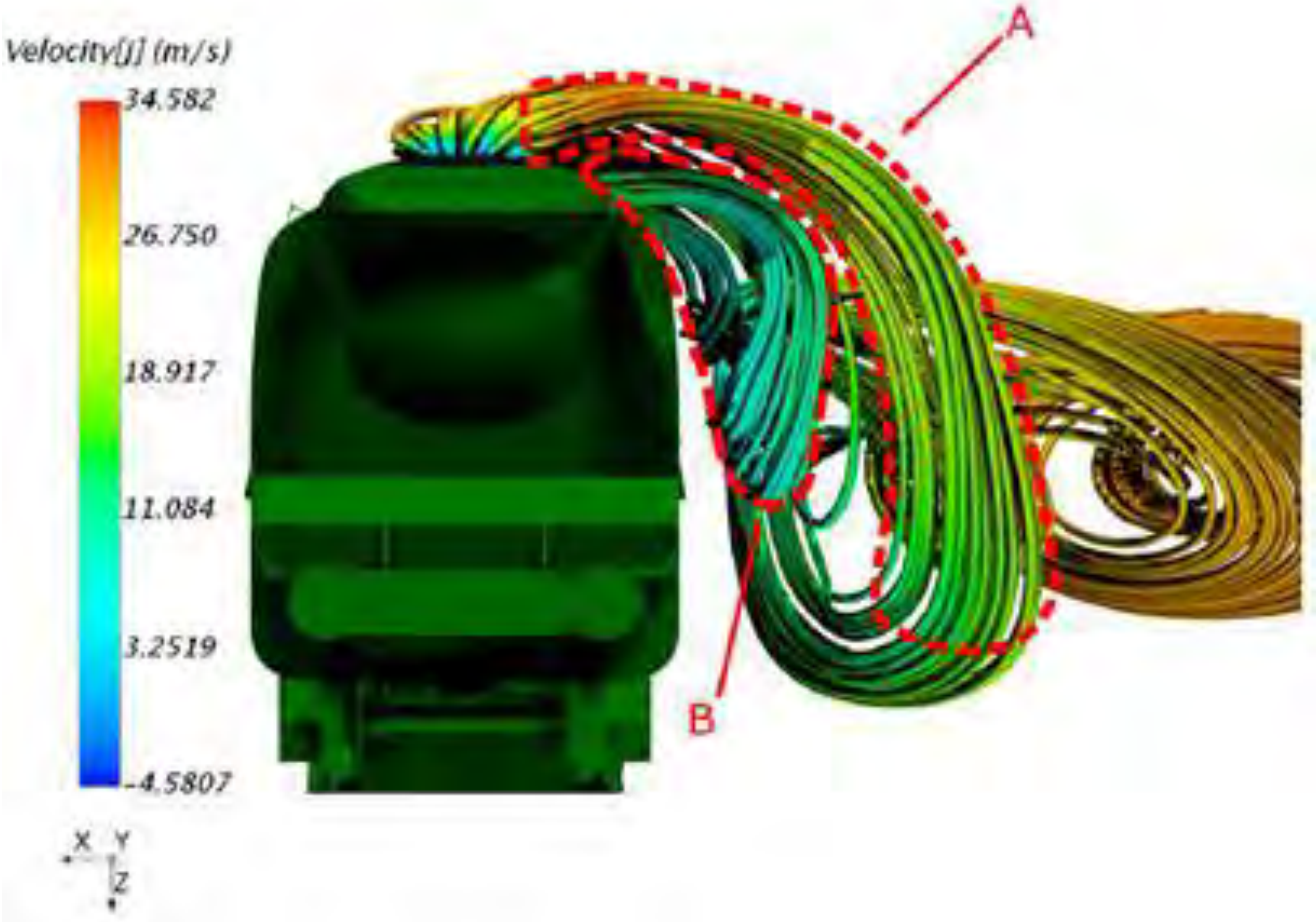


Figure 11: Diagram showing the streamlines shaded as a function of velocity in the y direction. The separation of flows (A and B) are outlined in red.

Figure 11 shows the flow of the mixture from the front of the locomotive with the streamlines coloured as a function of velocity in the y direction. The flow gets dragged down by the wake of the crosswind coming over the locomotive from the left-hand side, this results in a y velocity gradient away from the locomotive. As the obscurant moves down the length of the locomotive, it appears to separate. The lower velocity streamlines appear to be drawn down the side of the locomotive and carriages, causing potential obscuration. The higher velocity streamlines, on the other hand, are separated from the boundary of the locomotive and appear to arch over the driver's field of view.

Figure 12 *shows plots of turbulent kinetic energy and vertical velocity on the centreline of the locomotive for the original and modified design. These show significant turbulent mixing downstream of the exhaust in the original design, and flow separation in the diffuser, suggesting the diffuser angle is too great. By shifting to the 5° diffuser, the turbulent mixing downstream of the exhaust is reduced, and the flow separation in the diffuser is reduced. The first of these is likely the indicator of how this change in diffuser reduces obscuration by 49%, there is less turbulent mixing of the plume with the boundary layer on top of the casing, giving more opportunity for the exhaust to travel away from the locomotive body. The improved performance of the diffuser is the mechanism that drives this; turbulence begets turbulence, so suppressing its' generation in the diffuser gives less opportunity for it to seed turbulence as the exhaust mixes with the external air.*

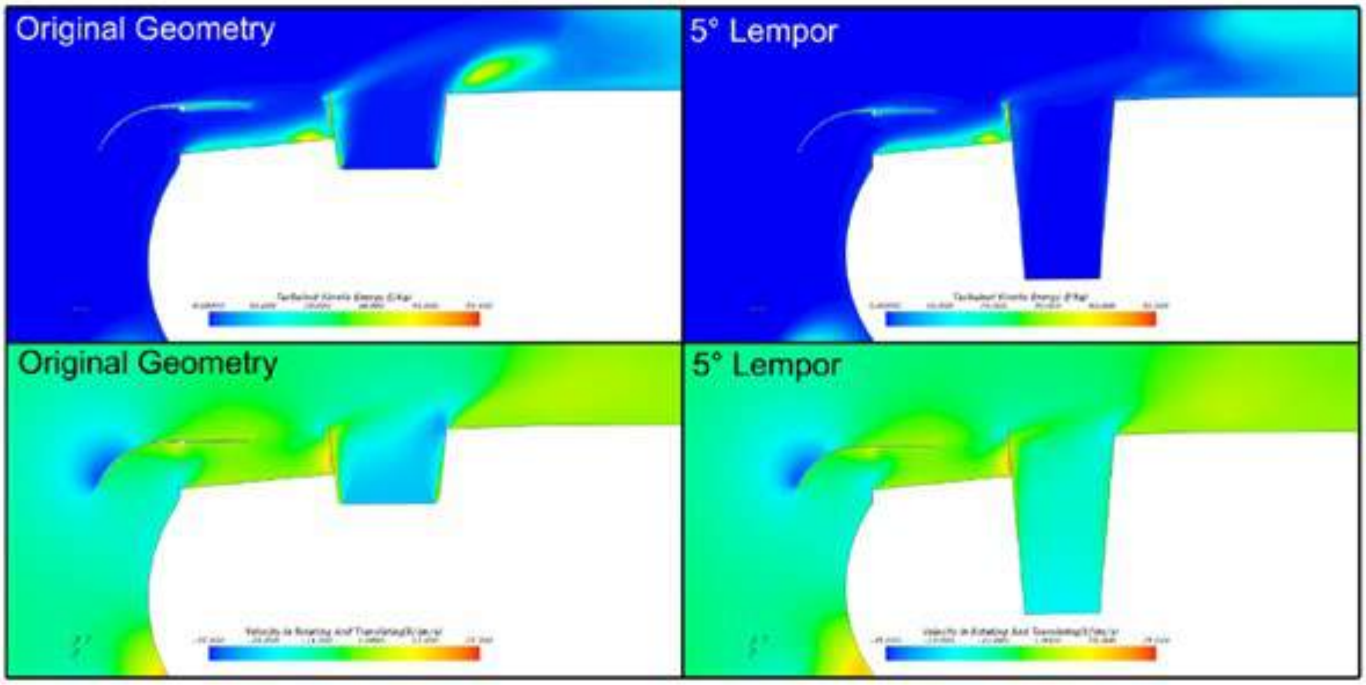


Figure 12: Comparison of the turbulent kinetic energy (top) and z component of velocity (bottom).

c. Exhaust Modifications: 6° Giesl, 6° and 7° Lempor

The third of the individual tasks looked at the impact of introducing an extended conical diffuser of 6 and 7 degrees, as would be introduced if something akin to a Lempor was introduced, as well as an oblong diffuser, similar to a Giesl. I have more confidence in this students work, it is also useful to compare the results of the previous section with this, as it shows the significant impact the angle of the diffuser can have on smoke obscuration.

The 6° Giesl and 6° and 7° Lempor exhausts were investigated at the 80_20 base simulation condition, whereby the main objective was to determine the effects of each exhaust on obscuration, the geometries being shown in Figure 13 and Figure 14. It was hypothesised that the Giesl exhaust should show the greatest improvement, as the oval cross-section meant that this type of exhaust was able to propel the exhaust steam higher and provide improved aerodynamic performance. A decrease in angle on the Lempor exhausts was also believed to reduce obscuration, as the reduced angle would allow for a more consistent and laminar flow of exhaust steam from the chimney.

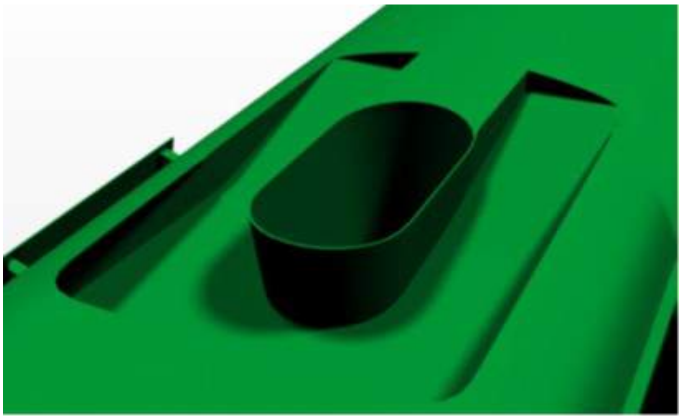


Figure 13: Exhaust geometry of 6° Giesl

Figure 14: Exhaust geometry of 6° Lempor

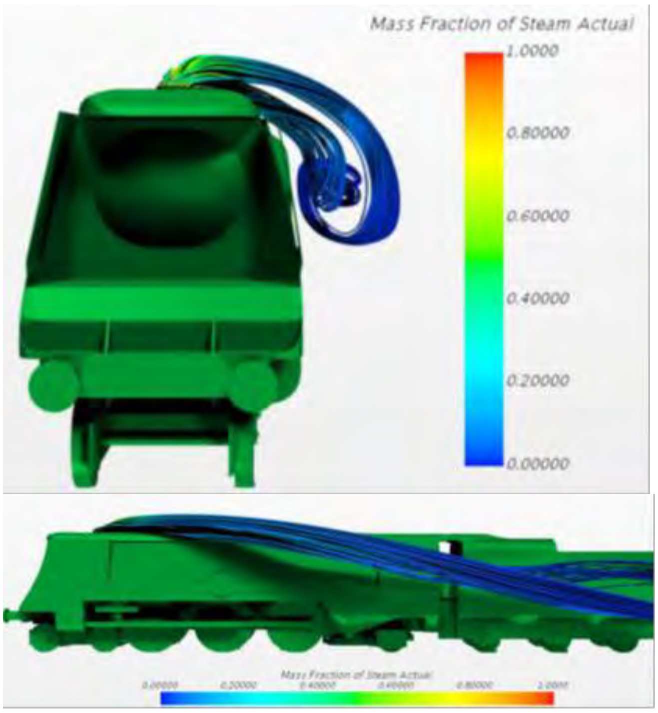


Figure 15: The exhaust flow from the 6° Giesl exhaust variant from the front and side

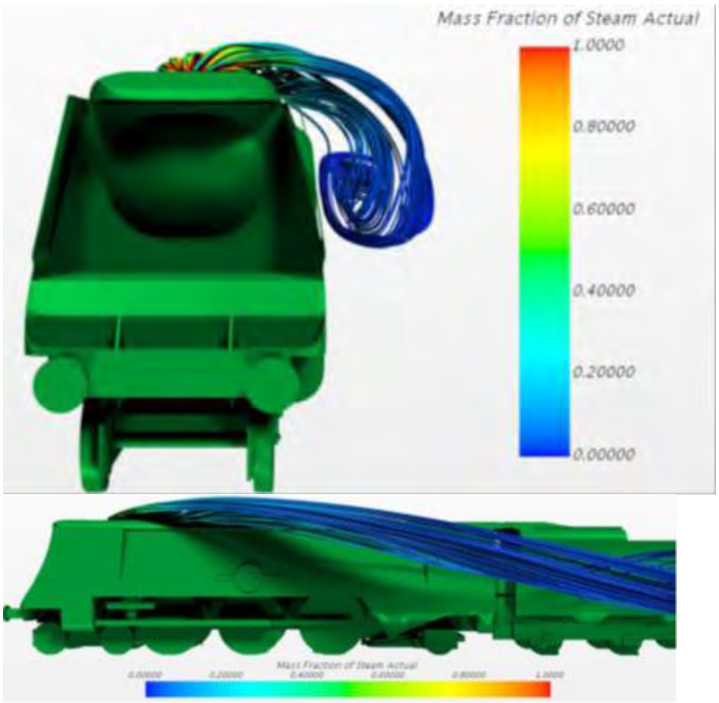


Figure 16: The exhaust flow from the 7° Lempor exhaust variant from the front and side

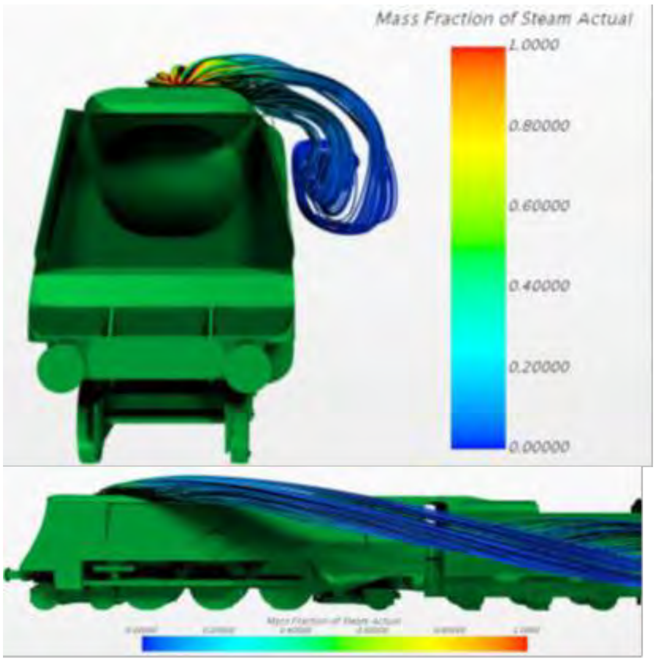


Figure 17: The exhaust flow from the 6° Lempor exhaust variant from the front and side

Figure 18: The exhaust flow from the 6° Giesl exhaust variant from the front.

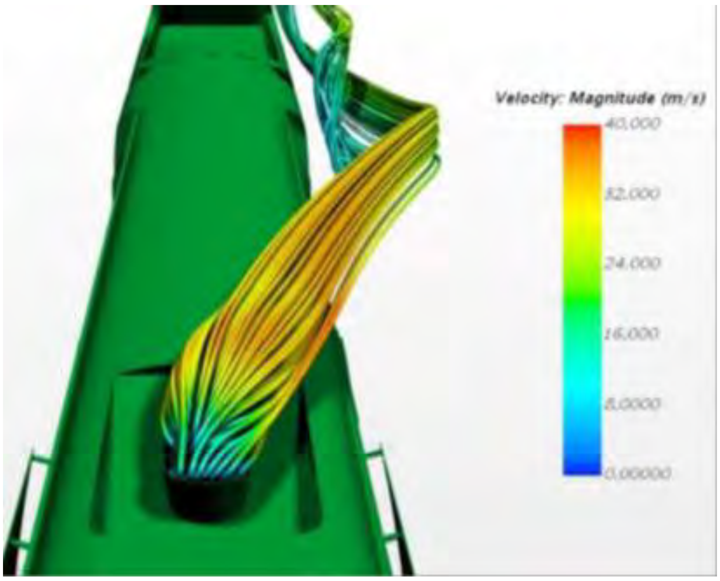


Figure 19: The exhaust flow from the 6° Giesl exhaust variant from above; note the uniform flow of streamlines

By visualising the flow characteristics of each simulation, it appeared that the 6° Giesl exhaust type was the most appropriate at reducing obscuration and channelling flow away from the locomotive, though the visual difference is subtle (Figure 15, Figure 16, and Figure 17). The elongated profile of the exhaust inlet compared to the base simulation exhaust geometry causes the x-velocity along the width of the exhaust to remain low until it exits the exhaust. As the steam outlet flow rate is constant in all simulations and as per the conservation of momentum, the main momentum components which therefore make up the flow of the exhaust steam are predominantly in the y and z axes (where x is parallel to the locomotive width, y is parallel to the locomotive length and z is parallel to the height of the locomotive).

Both Lempor exhaust variants continued to show obscuration, as some steam streamlines appeared to wrap closely around the cab window. Compared to the Giesl exhaust, the larger circular profile of the Lempor designs suggested that exhaust gases would dissipate and mix with the surrounding air more randomly on exit of the exhaust and over a greater distance of x, which may have been a factor in the noticed obscuration as the exhaust steam dissipates more sporadically. Visibly, the 6° Lempor design appeared to perform better in directing exhaust flow outboard from the locomotive, albeit some exhaust streamlines continued to flow alongside the body of the locomotive until nearer the carriages.

All simulations appeared to show two distinct separated flows which generate turbulent vortices soon after leaving the chimney (see Figure 18). This separation in flow may be a cause of the obscuration experienced, as one “branch” of the separated steam flow appears to pass significantly closer to the cab window than the other. Similarly, both streams of separated flow for the Giesl locomotive were relatively low velocity, at around 10 and 20 m/s for the nearer and further stream respectively, and the streamlines were evenly grouped together, suggesting the flow was neatly guided away from the locomotive and did not become turbulent prior to the generation of vortices. This higher steam velocity around the cab on the Lempor configurations indicates a lower pressure region which accelerates the flow and may play a part in the observed obscuration, as the higher velocity presents less opportunity for the steam to dissipate and is instead rapidly entrained towards the cab.

Next, the surface averages and mass fraction of steam measurements of each simulation were studied, to validate and reflect on the flow characteristics through studying the effects of obscuration and overall aerodynamic performance of each exhaust. These surface average values can be used in conjunction with the mass fraction of steam plots in Figure 34 and Appendix iv to observe how the steam-air mixture develops and where and how far the exhaust steam travels in this time as it leaves the exhaust. The 6° Giesl, 6° and 7° Lempor respectively change the obscuration by -19%, +3% and +7% compared to the original design. *Unfortunately, the analysis presented does not give plots that would allow an understanding of whether or how much flow separation is occurring in the conical diffuser in these designs; my expectation is as there is a significant change in behaviour relative to the 5° diffuser, the diffuser is no longer working in the flow conditions simulated.*

The Giesl exhaust geometry performs best in minimising obscuration from the cab, as the reduction of 18.9% in the surface average value compared to the base simulation means a lower mass fraction of steam is within the viewing perspective of the driver through the cab window. This is possibly due to the high crosswind velocity and the large length-to-width ratio of the exhaust profile, which causes a large percentage of the steam mass fraction to be drafted towards the back of the chimney from the interfering crosswind before exiting the exhaust in the z axis, unlike the Lempor designs and the base simulation where the mass fraction of steam is distributed randomly over the chimney exit.

The high negative gradient in mass fraction of steam on the Giesl exhaust outlet also suggests that much of the enthalpy the high temperature steam possesses has already been transferred to the surroundings soon after it leaves the exhaust. This resultant gradient indicates a large proportion of steam condenses above the exhaust, meaning it does not obstruct the driver's view as the densest region of water vapour which forms above the exhaust is free to disperse before it is entrained along the sides of the locomotive, leaving a low mass fraction of steam no greater than ~0.15 within the driver's view arbitrary block.

d. Deflectors Investigation

An investigation into the role of the smoke deflectors was carried out to validate the current use of them on the locomotive. Literature states smoke deflectors have a function of "promoting a flow of clean air along the boiler sides or lifting the smoke by a vertical air current at the chimney" (Peacock, 1951). Therefore, it was predicted that using smoke deflectors would improve the driver's visibility compared to removing them. To evaluate this a CFD simulation has been conducted with no deflectors at the 80_20 and 50_40 conditions. A simulation where the deflectors have been extended back by 500 mm was also performed to investigate whether lengthening the deflectors could lift the exhaust flow further away from the locomotive's body, thus improving the driver's visibility.

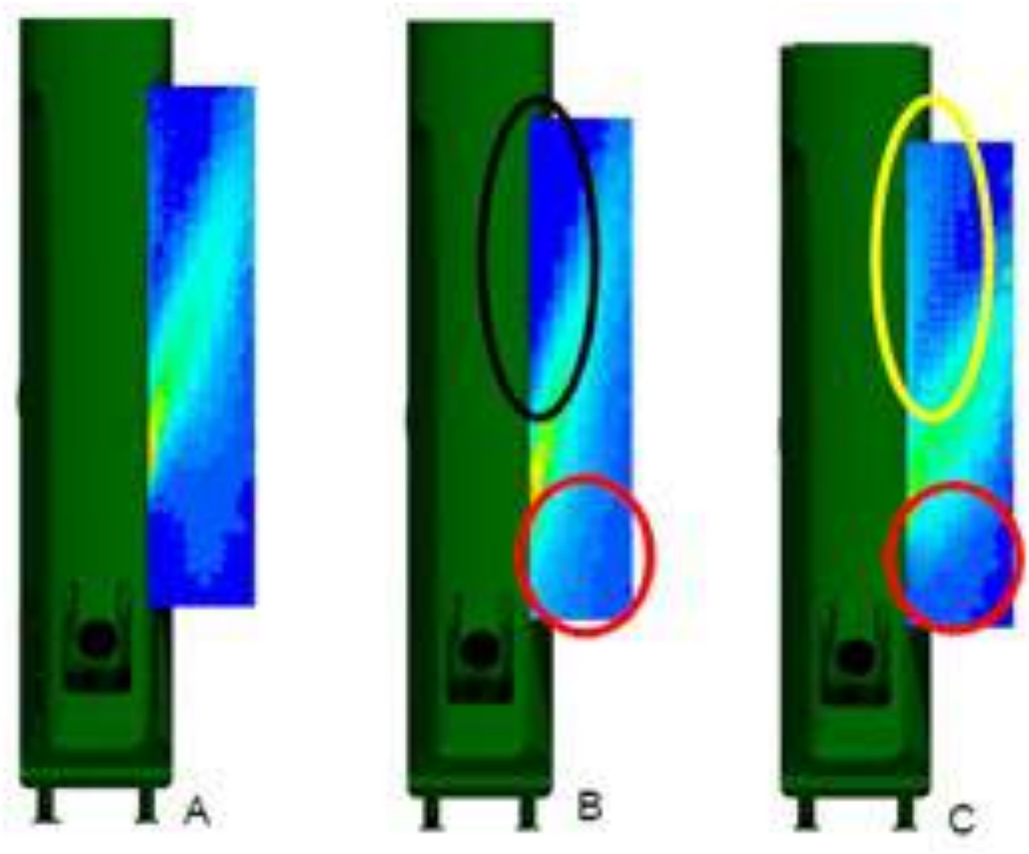


Figure 20: Top-down view comparing the mass fraction of steam on the obscuration volume. A-: base simulation, B- No deflectors 80_20 condition, C- No Deflectors 50_40 condition.

Analysis of the results give a positive indication that removing the deflectors has a negative impact on the driver's view. Figure 20 is a top-down view of the mass fraction of steam present on the obscuration volume for the base condition and the no deflectors at the 80_20 and 50_40 conditions. Comparing the three images shown, it is clear that more steam is present at the front of the block for the no deflectors, shown by the red circles. This suggests that the steam is not being lifted up and away from the locomotive effectively, adding to the obscuration. B also shows the steam 'sticks' less to the side of the locomotive (black circle. However, it translates the obscuration further upstream. Whereas C has the plume sticking to the locomotive for longer (yellow circle), at a greater concentration, and dispersed over a greater area, suggesting why the surface average percentage difference is much greater. The simulations performed show that deflectors have a positive impact (at least a 5.27% improvement) on the driver's visibility, backing up the theory of deflectors and concurs with experiments carried out when the locomotive was first being built.

e. Unsteady Exhaust Flow Study

A transient study was conducted to see the effect of unsteady exhausting flow in order to validate the use of a steady model in the simulations conducted. From videos of working locomotives there are noticeable peaks and troughs of mass flow rate as the cylinders exhaust, which the steady simulation does not consider. This study could become significant if there is an increase in the obscuration as result of unsteady flow as this will effectively mean all the studies up to this point are underestimates.

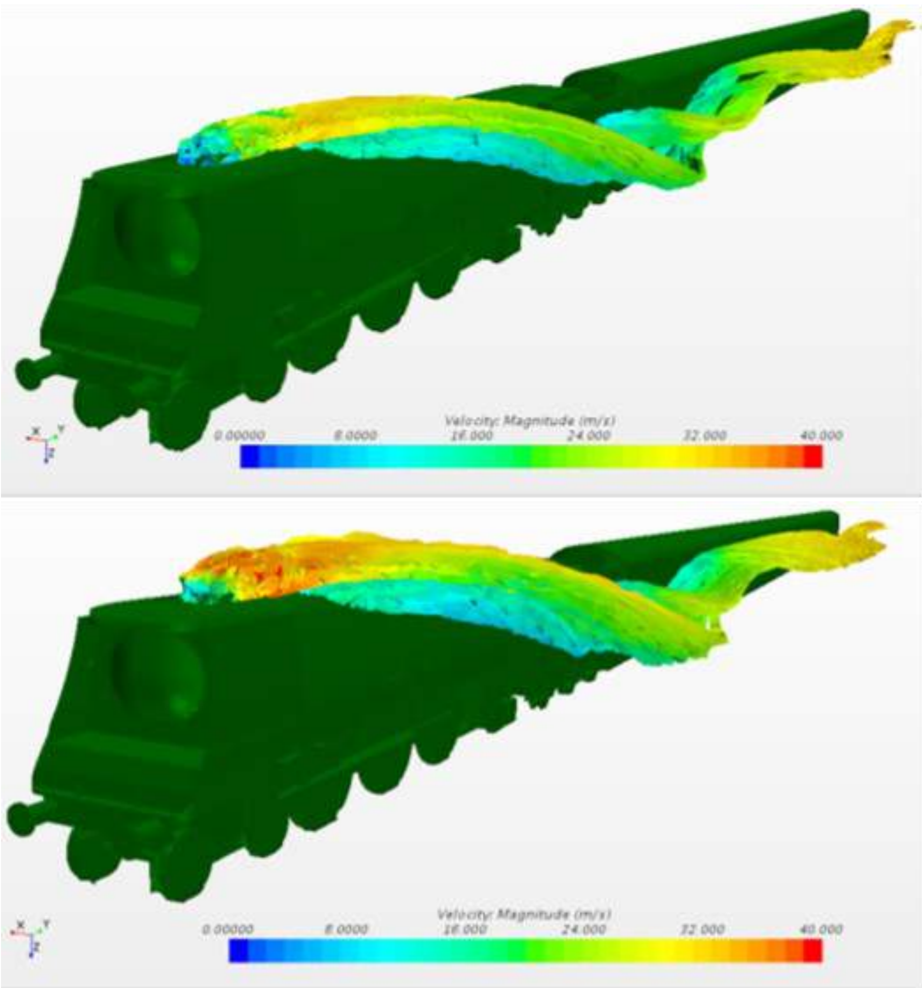


Figure 21: Streamlines of the exhaust from transient simulation, showing peak and trough behaviour

A predicted mass flow rate for the transient case has been derived using the assumed exhaust cycle and flow rate behaviour. The peak exhaust flow has been modelled as 43.5 % higher than the mass flow rate of the steady case and the minimum exhaust flow 56.2 % lower. It is assumed the decrease in flow follows an exponential relationship and has been modelled as a halving every tenth of an exhaust cycle. It is predicted, the effect of the increased peak exhaust flow will cause an increase in obscuration in the drivers view as there is a larger amount of steam present and potentially the unsteady exhaust behaviour could result in an accumulation of steam in front of the driver's view. The unsteady simulation has been monitored at two different points so comparisons can be made between the minimum (trough) and maximum (peak) exhaust flow cases, as shown in Figure 21.

In the trough case the flow is exhausting at a quicker rate, with a max velocity of around 40m/s compared to the peak case which is exhausting at around 20m/s less, this is caused by the travel time for the peak flow to pass from the inlet of the model to the top of the exhaust. Furthermore, there is a notable lack in y momentum of the exhausting steam in the peak case. The differences between the flow continue downstream. This is where the effect of unsteady flow becomes most apparent with the flow forming a much tighter vortex in the trough case. Increased separation of flow can be observed in the peak case just behind the cab where it appears four separate paths have been formed.

The unsteady mass flow cases show increasing difference the further downstream the flow is observed from the exhaust boundary, with little difference in the driver's view developing into large differences at the back of the domain.

5. Conclusions and further work

From the base simulation, it was clear to conclude that, as expected, higher forward speeds of the locomotive gave rise to a visibly more severe obscuration effect. The forward speed of most significant obscuration effects was at 80mph, where for the crosswind speeds of 10 and 20 mph, the visibility from the driver's cab was affected detrimentally by the exhaust drift and the development and condensation of the steam-air mixture around the body of the locomotive. Henceforth, the forward speed of 80 mph and crosswind speed of 20 mph was selected as the speed conditions for the proceeding individual tasks, as these precedents displayed the most extreme levels of obscuration from the 12 group simulations conducted (mentioned in Section 2.5), and so any modifications to the locomotive which affected the exhaust flow pattern would be easier to distinguish between when examined at these defined conditions.

The results of the individual simulations also presented some significant findings and conclusions. Of all modifications investigated, the 5° Lempor design proved the most beneficial in limiting the obscuration. Analysis of the surface average of the arbitrary section found there was almost a 50 % reduction in obscuration at the 80_20 condition. The 6° Giesl design also proved highly effective in drafting exhaust steam away from the driver's view in a laminar manner, reducing the obscuration by 19% from the base simulation. It was found that the changes in ducting have a minor but noticeable impact, the flow direction and concentration with the extended duct reducing the amount of obscuration by 10.4%. All other modifications produced an increase in obscurity. This is a significant finding as it has been shown a small change in geometry can be result in a large difference in flow.

The expectation going forwards is that in academic year 2021/2 we will run a continuation of the internal aerodynamics study, depending on the findings of that we may return for a continuation of the external aerodynamics study in 2022/3.

6. References

Holcroft, H. (1941). Smoke Deflectors for Locomotives. *Journal of the Institution of Locomotive Engineers*, 462-509.

Peacock, D. (1951). Railway Wind Tunnel Work. *Institution of Locomotive Engineers*, 606-661.

MECHANICAL CYLINDER LUBRICATION, PART V: G.W.R. HYDROSTATIC LUBRICATION TO VALVES & CYLINDERS

John Duncan

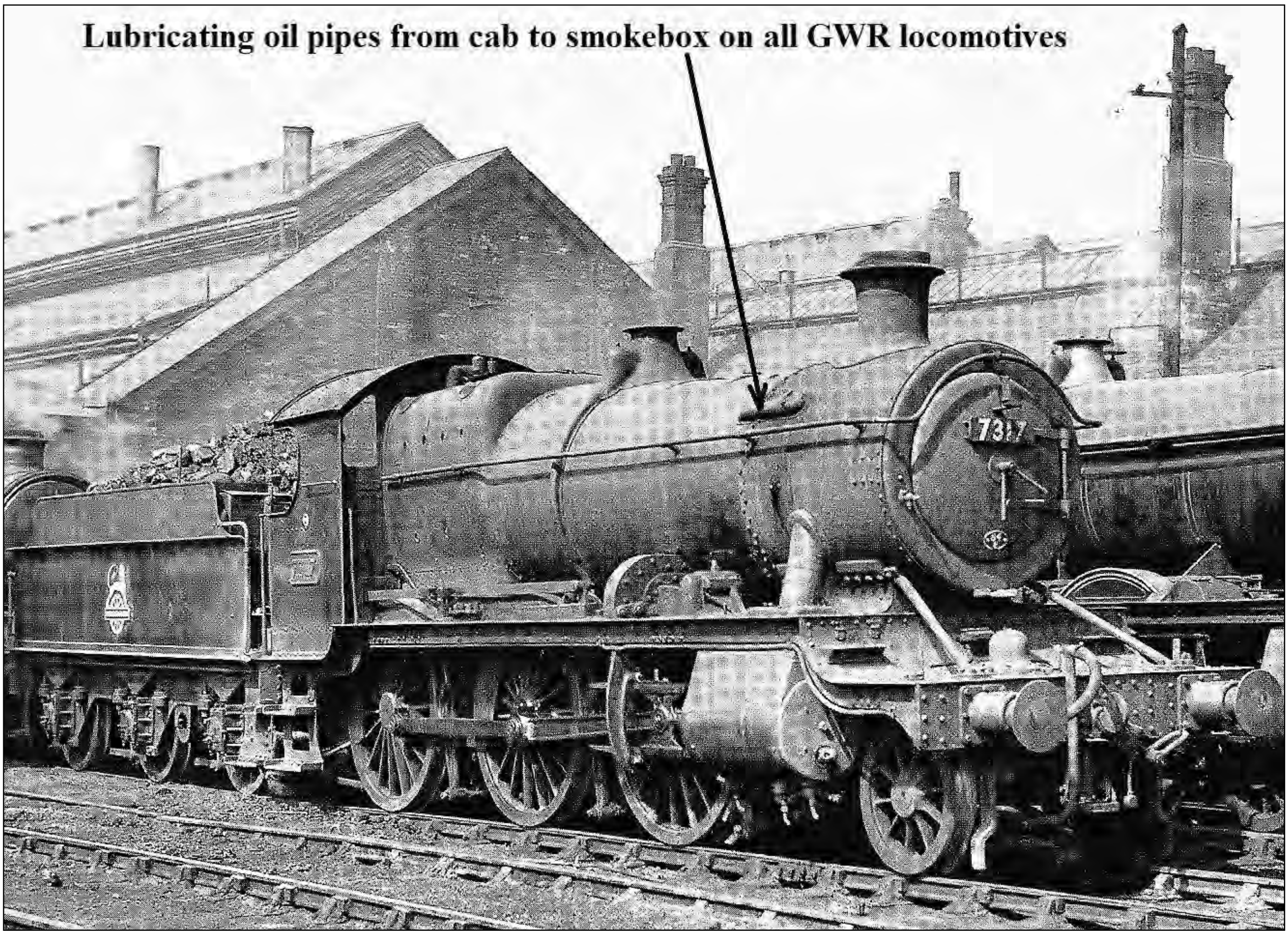
My experience and knowledge of GWR steam locomotives is from April to August 1960. On April 11th 1960, Chester Engine Shed, 84K (CHR) closed and all the allocation of locomotives moved to Chester 6A. They sent an instructor from Wolverhampton, (Stafford Road), 84A.

The Examining Fitters were dealt with first. We were told the cylinder lubricators were not our responsibility, it was the Enginemen's only. We did get a lot of help and information from Enginemen.

The GWR cylinder hydrostatic system developed from the Churchward era and his introduction of superheating. 1900s to 1930s and Charles Collett's 'King' class, pressed to 250 psi with the rise in steam temperature. There were improvements right up to the last hydrostatic with the modified 'Halls', the 6959 class and the 'County' class, the '1000' class. With a boiler similar to the LMS 8Fs, built at Swindon during the Second World War. The boiler was pressed to 280psi, later reduced to 250 psi.

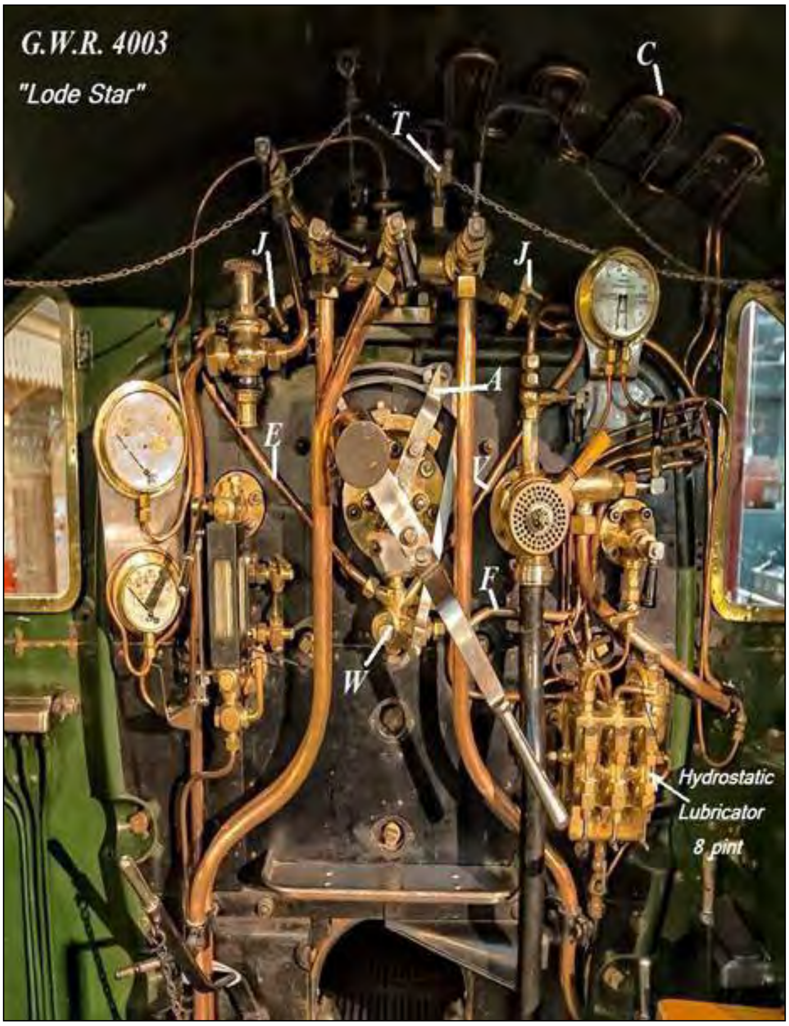
Material was used from experience and publications. A paper, by Mr W.H. Pearce, "Recent Developments in Cylinder Lubrication" at the Swindon Engineering Society, November 22nd 1931. Chaired by Mr W.A. Stanier. M.I.Mech.E.(President). A paper by Mr K.J.Cook, OBE, (Vice President). Paper No.492, "The Late G.J.Chuchward's Locomotive Development on the Great Western Railway". 1950, The Institution of Locomotive Engineers.

The system used by Churchward with a condenser in the roof in the locomotive cab, then to the hydrostatic lubricator with a direct feed to the regulator valve in the smokebox. The other feed is to the Combining valve where boiler steam mixes with the oil from the lubricator. A form of 'Atomiser' opens when the regulator handle is moved three quarters of an inch from 'shut' to admit a small amount of steam to the cylinders. A pin on the regulator moves in a slot connected to a rod that opens a valve to allow the mixture of steam and oil droplets to the 'O' in the smokebox, where it divides to each steam pipe via a choke and perforated nozzle inside the steam pipe. The oil droplets passed on to the steam chests and on to the cylinders. The oil pipes pass through under the boiler cladding and appear at the end of the boiler barrel and smoke box in a raised cover plate on the right hand side by the hand rail where they enter the smoke box.

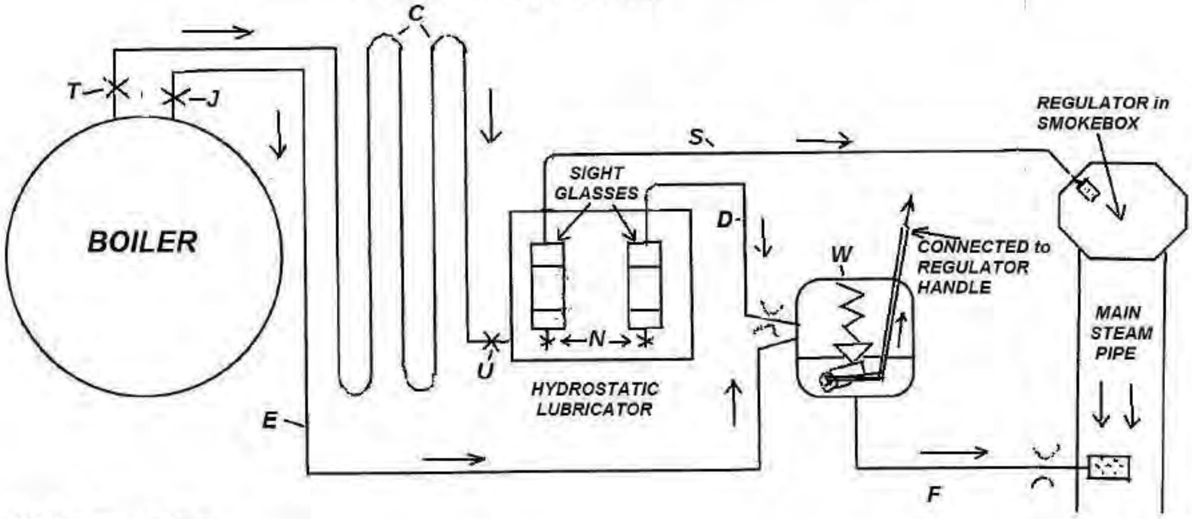


A typical Churchward design I worked on, on Afternoon Shift at Chester, 6A. Engine for the 'Barmouth Mail', ''Knock at front end'', Engineman said. Found bolts between the cylinder and frame loose. Later sent to Wolverhampton Factory .

Different to the 'Detroit' and 'Eureka' enclosed hydrostatic lubricators. The condenser is up in the cab roof, in a 'zig zag' formation. Later under Collett it was two coils. The lubricator was a GW design and the combining valve, 'atomiser' opened when the regulator was just open. The whole system is under the control of the Engineman, (Driver).



G.W.R. SCHEMATIC DIAGRAM of the ORIGINAL HYDROSTATIC LUBRICATING OIL SYSTEM to the ENGINE CYILNDERS



KEY to symbols :-

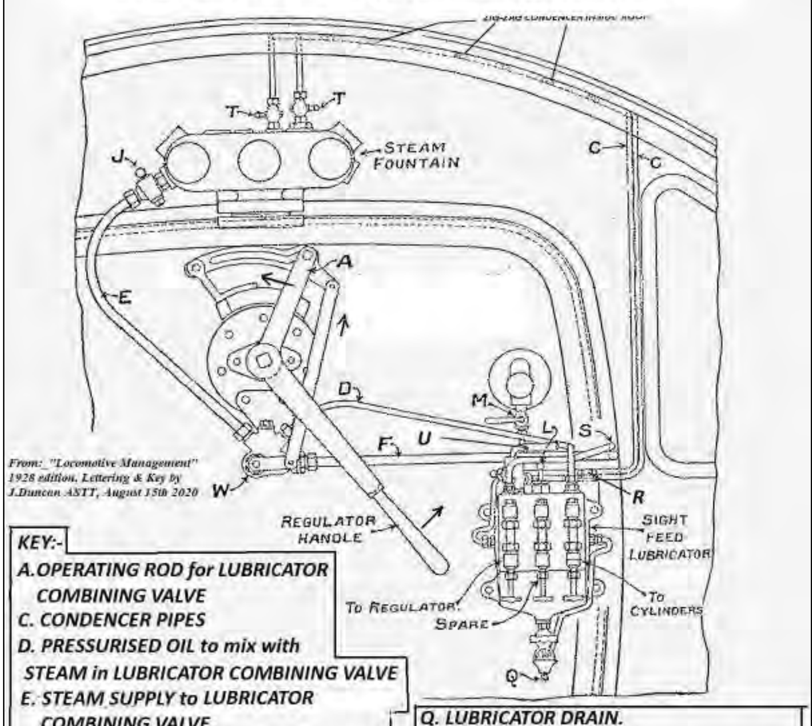
- COCK type VALVE.
- CHOKE in PIPE
- FIXED SPRING
- VALVE
- PERFORATED NOZZLE in MAIN STEAM PIPES and REGULATOR VALVE SPACE

KEY :-

C = CONDENSER PIPES
D = OIL from HYDROSTATIC LUBRICATOR to LUBRICATOR COMBINING VALVE
E = STEAM TO LUBRICATOR COMBINING VALVE
F = ATOMISED OIL to MAIM STEAM PIPE and CYLINDER
N = SIGHT GLASS CONTROL COCKS
S = OIL DELIVERY PIPE to REGULATOR VALVE
T = STEAM VALVE to CONDENSER PIPES
U = FILL COCK CONDESERS to LUBRICATOR
W = LUBRICATOR COMBINING VALVE

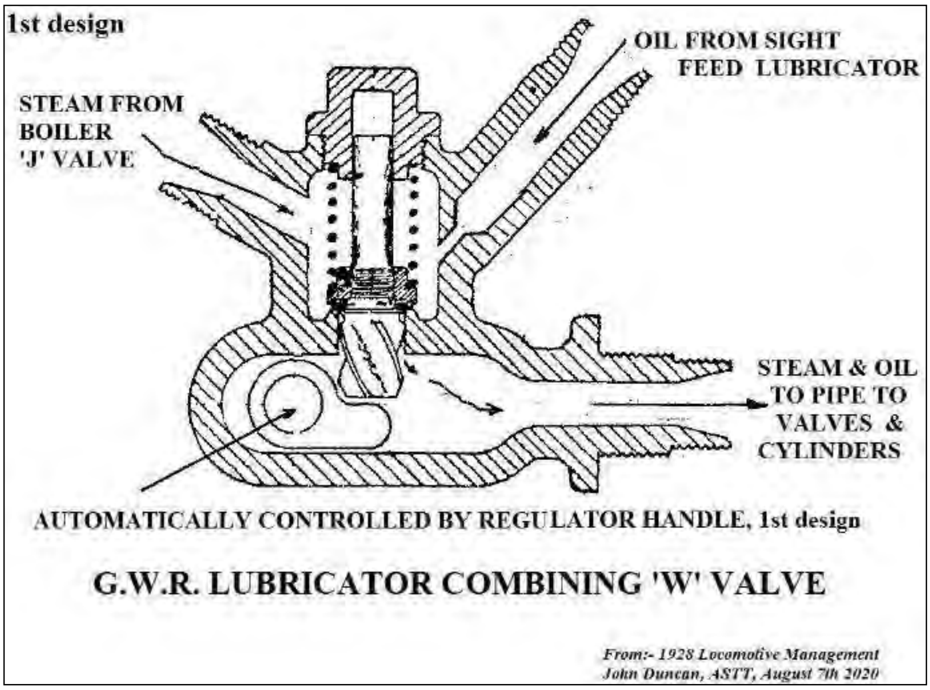
J.Duncan ASTT, August 23rd 2020

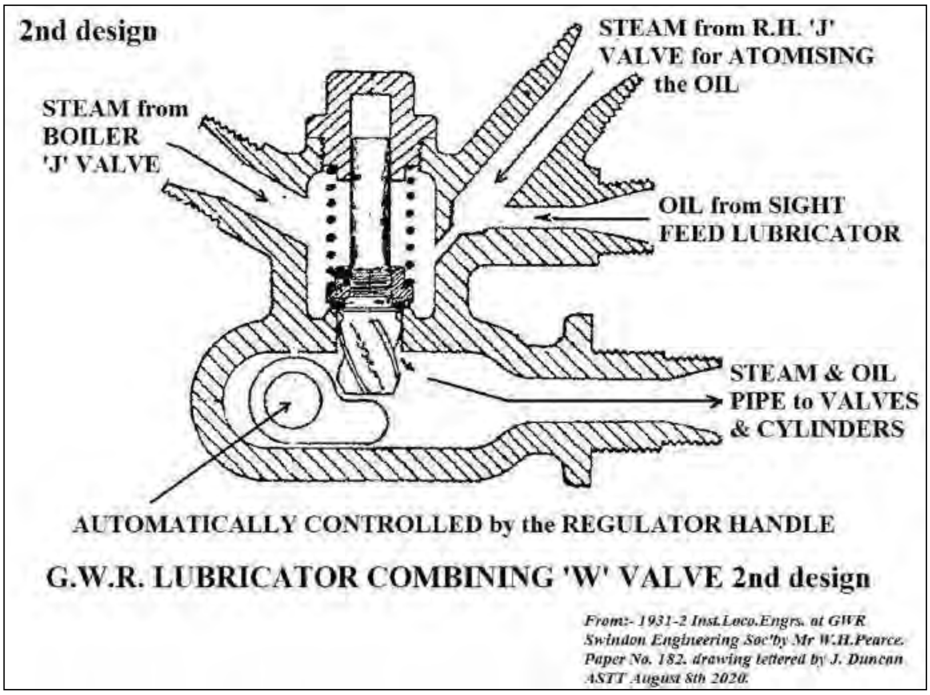
GWR ORIGINAL FOOTPLATE SCHEME for HYDROSTATIC LUBRICATION of REGULATOR VALVE, SLIDE & PISTON VALVES and CYLINDERS.

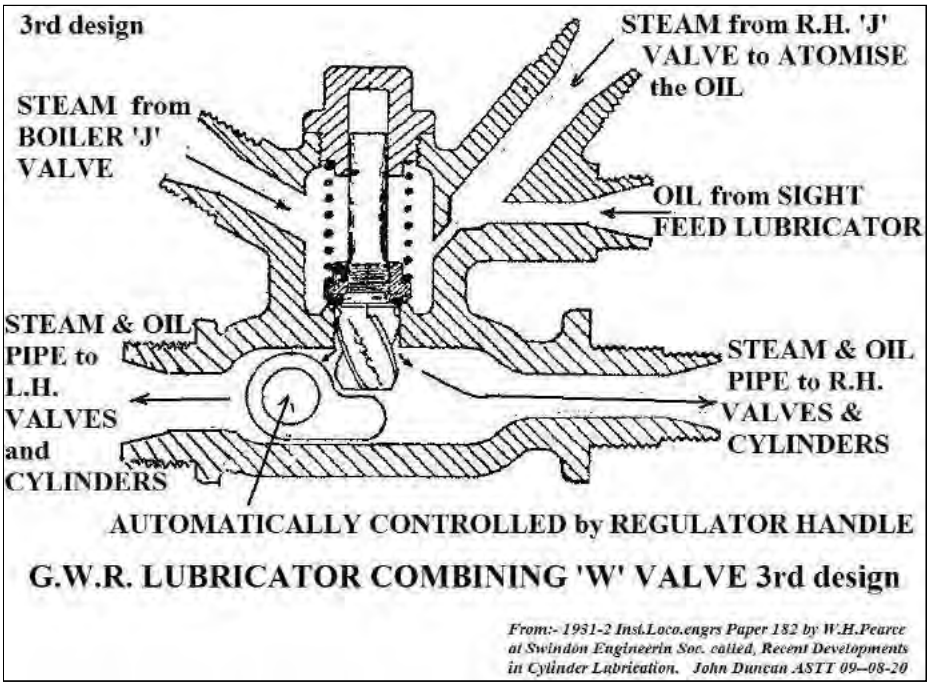


KEY:-

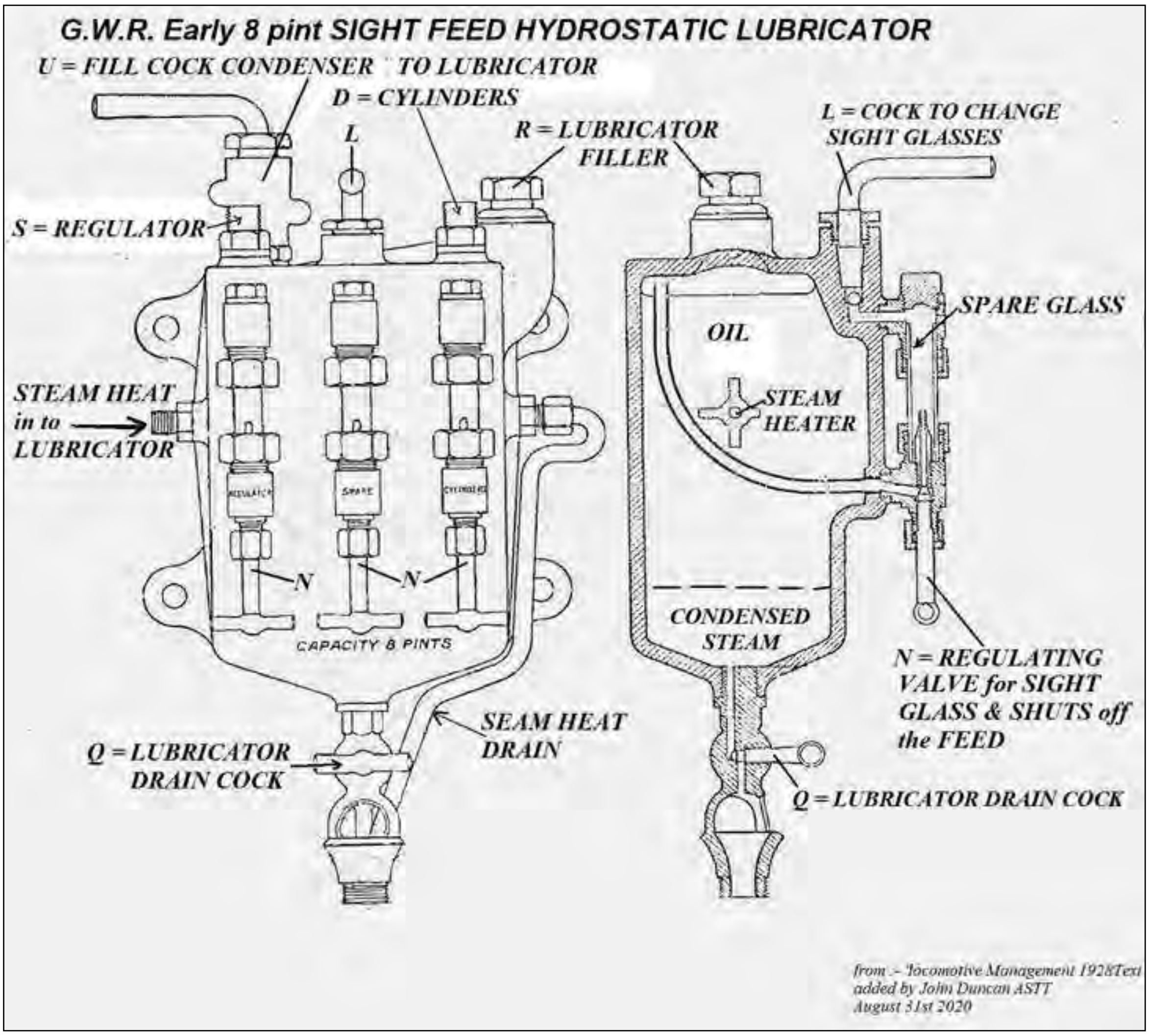
A. OPERATING ROD for LUBRICATOR COMBINING VALVE
C. CONDENCER PIPES
D. PRESSURISED OIL to mix with STEAM in LUBRICATOR COMBINING VALVE
E. STEAM SUPPLY to LUBRICATOR COMBINING VALVE
F. ATOMISED OIL to VALVES & CYLINDERS.
J. COCK to open STEAM to LUBRICATOR COMBINING VALVE
L. COCK to change to SPARE GLASS.
M. COCK for WARMING PIPE
Q. LUBRICATOR DRAIN.
R. LUBRICATOR FILLER.
S. OIL DELIVERY pipe to REGULATOR VALVE.
T. STEAM VALVES to CONDENCERS.
U. FILL COCK CONDENCERS to LUBRICATOR.
W. LUBRICATOR COMBINING VALVE.

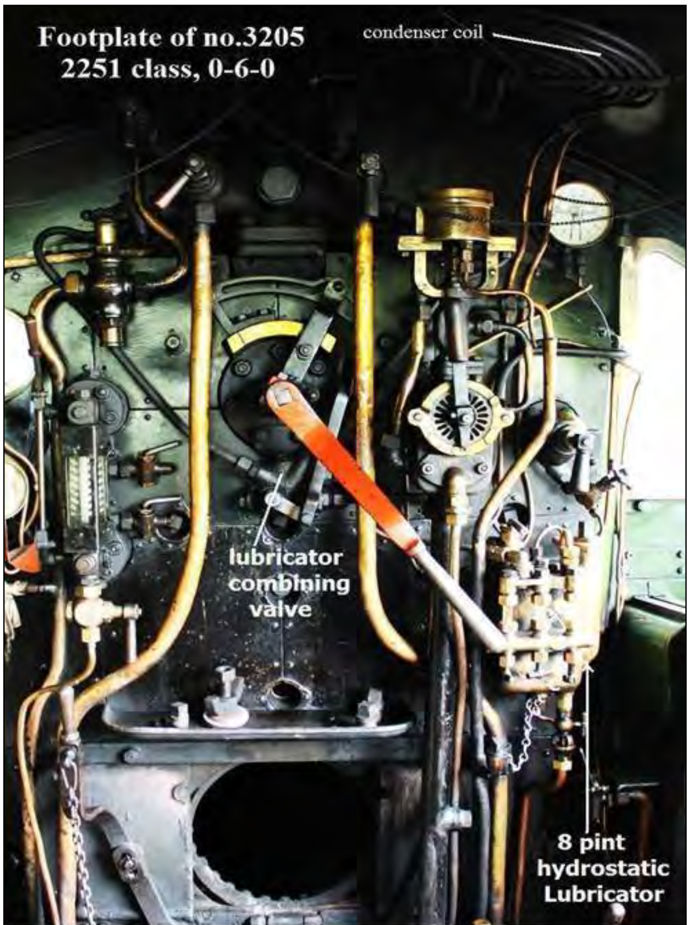




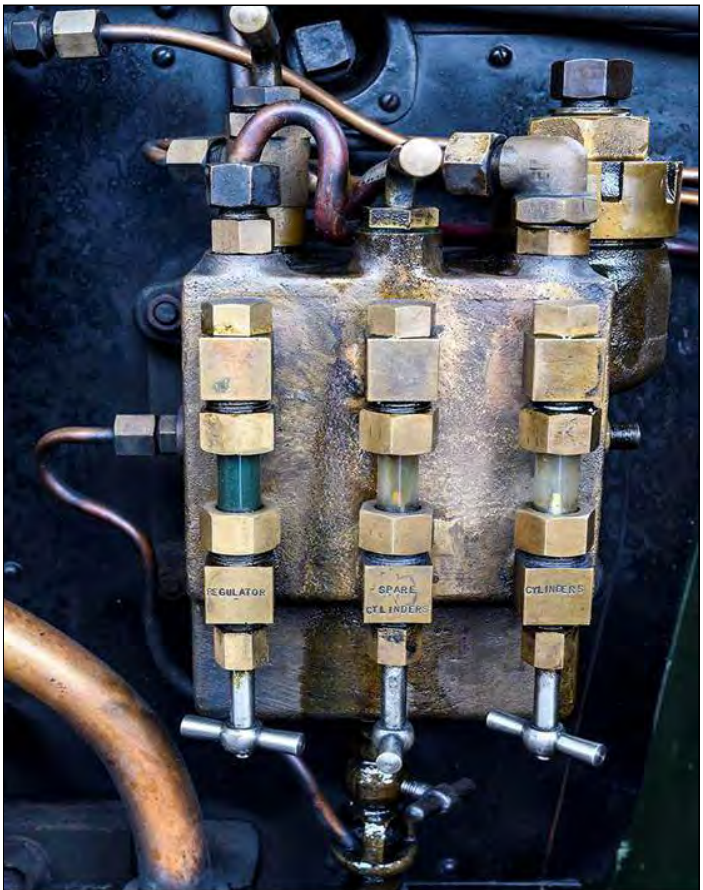


The 3 versions of the 'W' Combining Valve are a progression in the process of 'Atomisation' of the oil, under pressure from the Hydrostatic Lubricator to the main steam pipes in the smokebox. A sectional drawing of the 8 pint lubricator used on larger engines from the 1900s to 1920s is shown. It is still used today on smaller 2 cylinder engines.



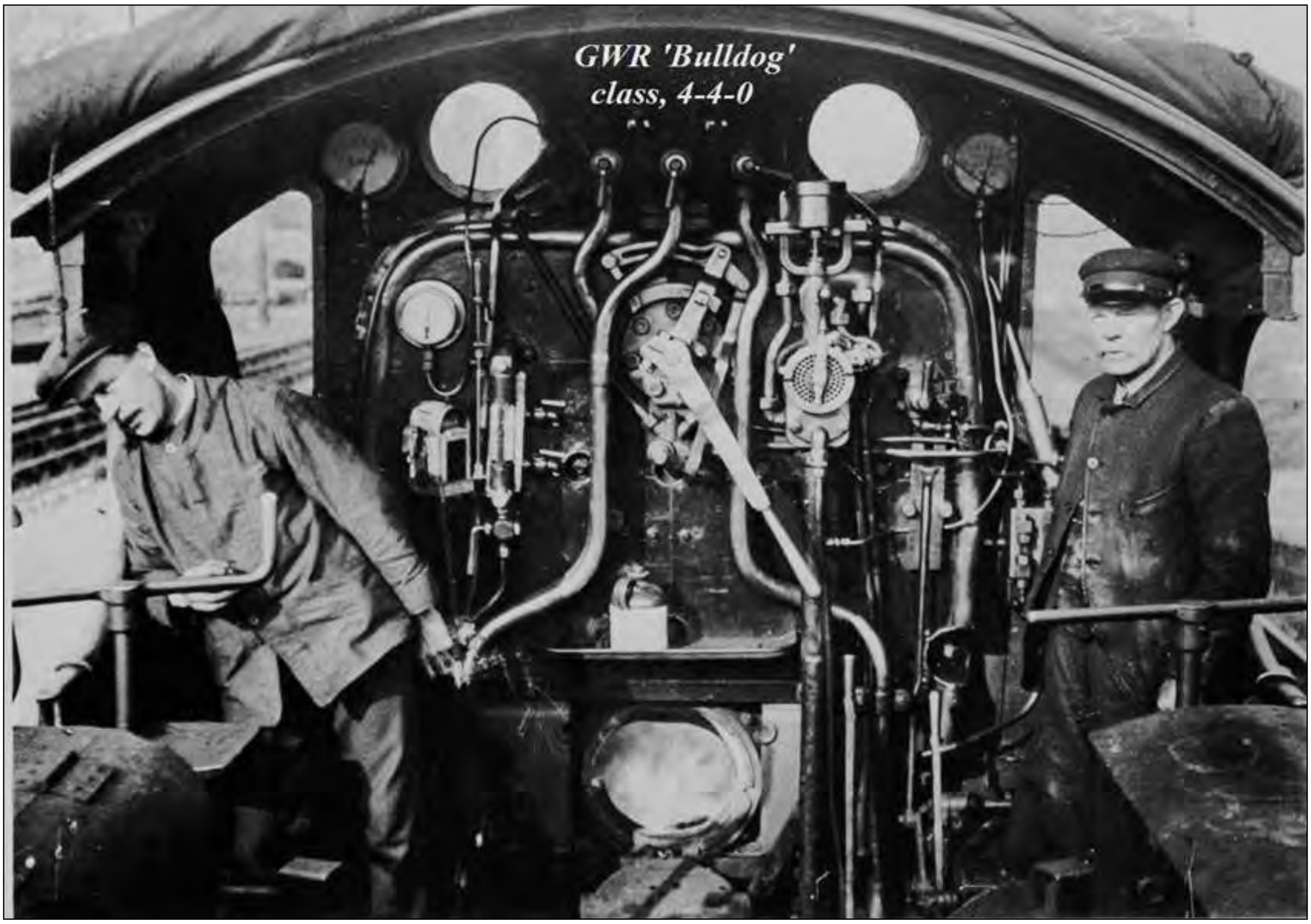


GWR, 2251 class, 0-6-0, built as a replacement for the Dean, 2301 class under C.B.Collett in 1930 onwards.



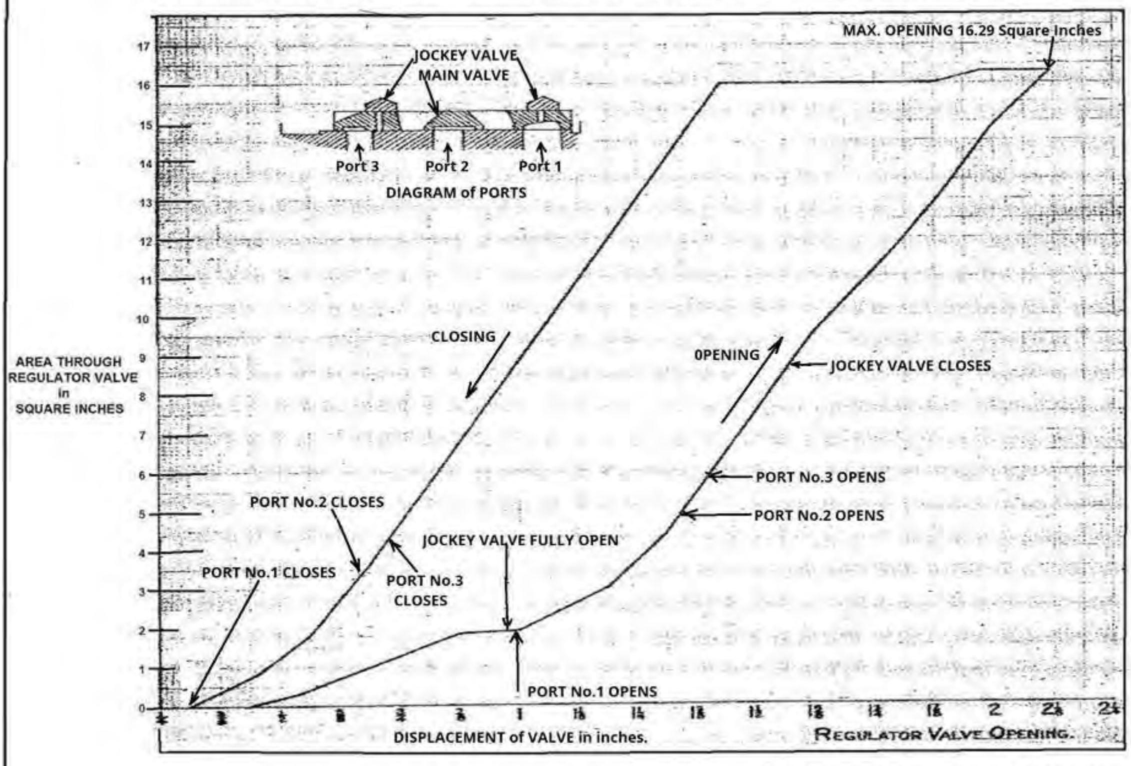
GWR 8 pint hydrostatic lubricatoras on small & tank engines

Churchward, 'County' 3800 class, 4-4-0, on the footplate, the 8 pint lubricator to the left of the Driver, the 'W' combining valve below the regulator gland and the rod to operate it when the regulator is opened. Any GWR footplate is very much the same.



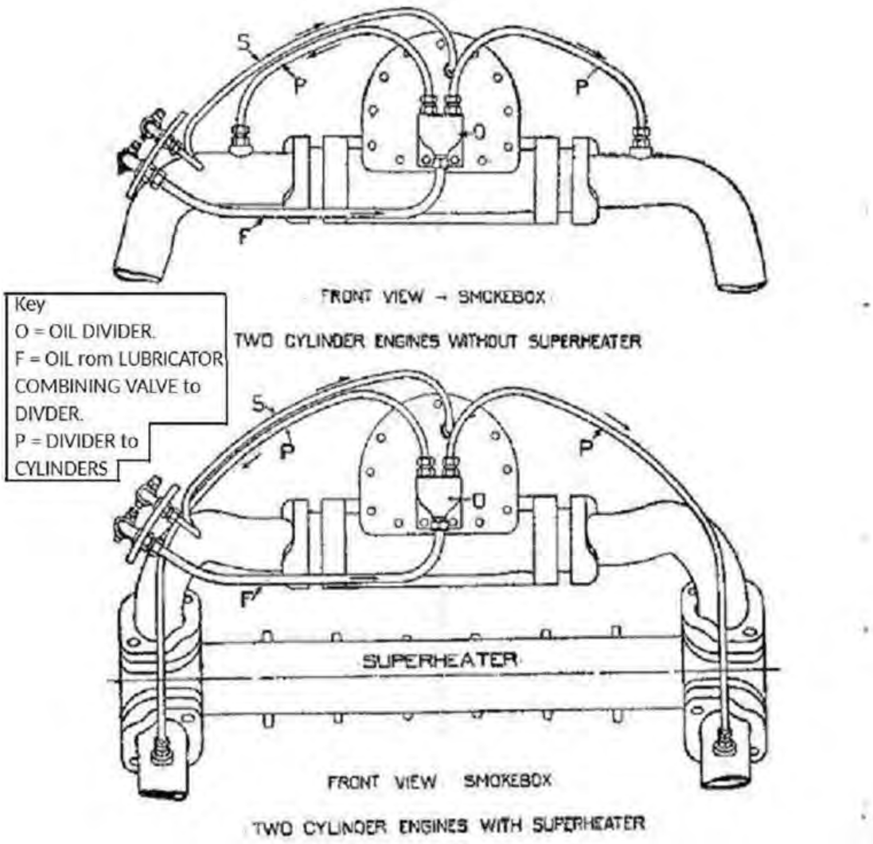
GWR DIAGRAM OF REGULATOR OPENING. Standard on GWR Steam Locomotives.

To quote :-K.J.Cook in His Paper. " A regulator Valve was provided in the smokebox which was of the usual jockey valve type, although there was considerable detail development and its area diagram (below) was carefully plotted to ensure delicate control of steam admission and this particular feature has played a large part in obtaining the freedom of Great Western engines from slipping when starting heavy loads."



This Diagram is from Paper 492 in March/April 1950 by K.J.Cook, O.B.E. to the Institution of Locomotive Engineers. Vol. xi, No. 214 "The Late G.J.Churchward's Locomotive Development on the Great Western Railway." Lettering and enhanced by J.Duncan for ASTT, October 15th, 2020

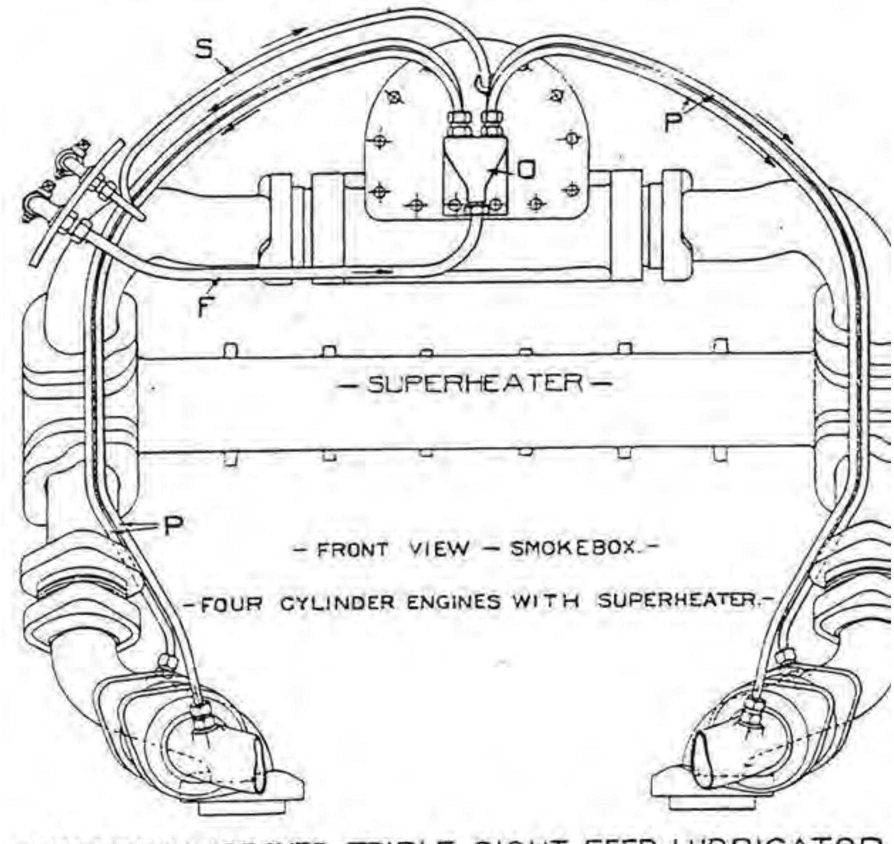
At the smokebox end diagrams show the cylinder oil pipes on saturated, two cylinder and four cylinder arrangements, and the regulator oil pipe, in the Churchward era.



G.W.R. SMOKEBOX DIAGRAM of LUBRICATION PIPES to CYLINDERS

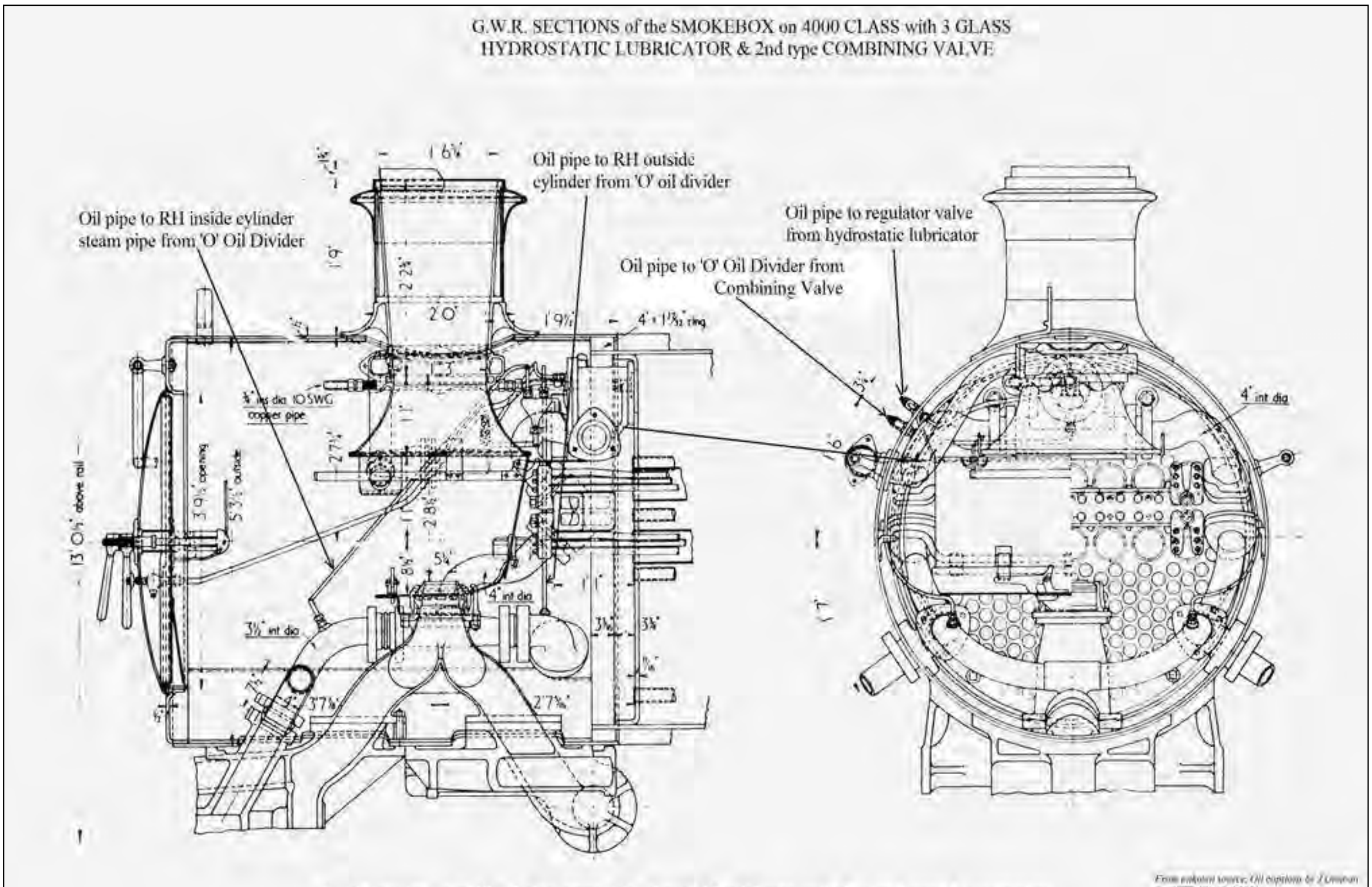
From :-"Locomotive Engine Driving" by A. Oliver Page 106. Key and text J.Duncan ASTT, August 26th 2020

'Star' 4000 class inside steam pipes & 2nd type combining valve.



–SWINDON IMPROVED TRIPLE SIGHT FEED LUBRICATOR.

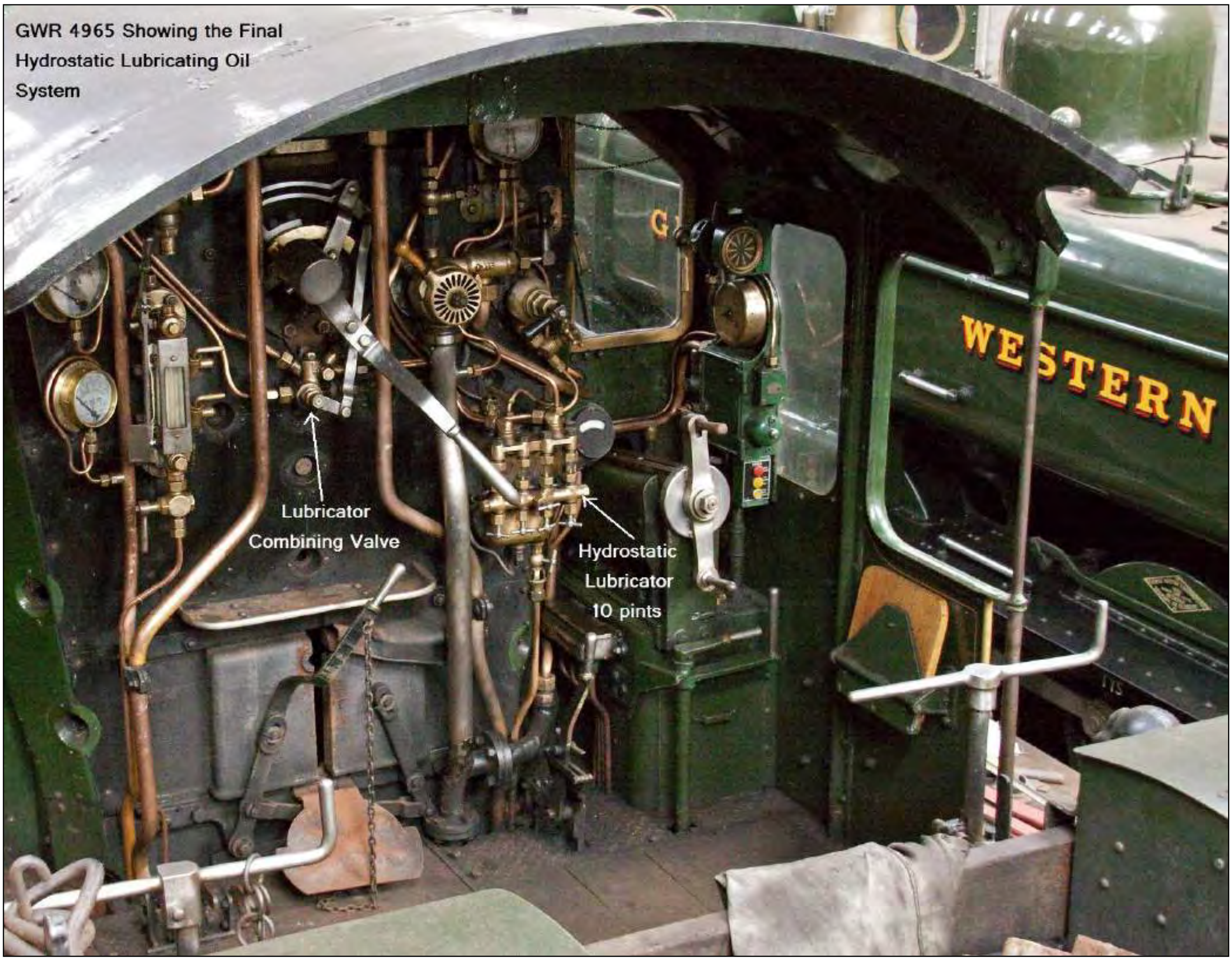
The last sectional drawing is the 1920s and the handover period from the Great Western Railway to the enlarged G.W.R. and Churchward retires in 1922, Collett takes over as C.M.E. From the 'Star' class to the bigger 'Castle' class and later that decade the 'King' class with an increase in boiler pressure from 225 psi to 250 psi and the resulting increase in saturated steam temperature, from 397.4 degrees F to 406.2 degrees F. Problems continued with the hydrostatic system into the 1930s with the temperature as well as the distance the steam and oil atomised mixture had to go to reach the place where the mixture was to do its work, passing through a very hot smokebox. Some very interesting solutions were progressed by Collett's Staff at Swindon, described by W.H.Pearse at Swindon Engineering Society in a meeting of the Institution of Locomotive Engineers Paper No.182, November 1931.

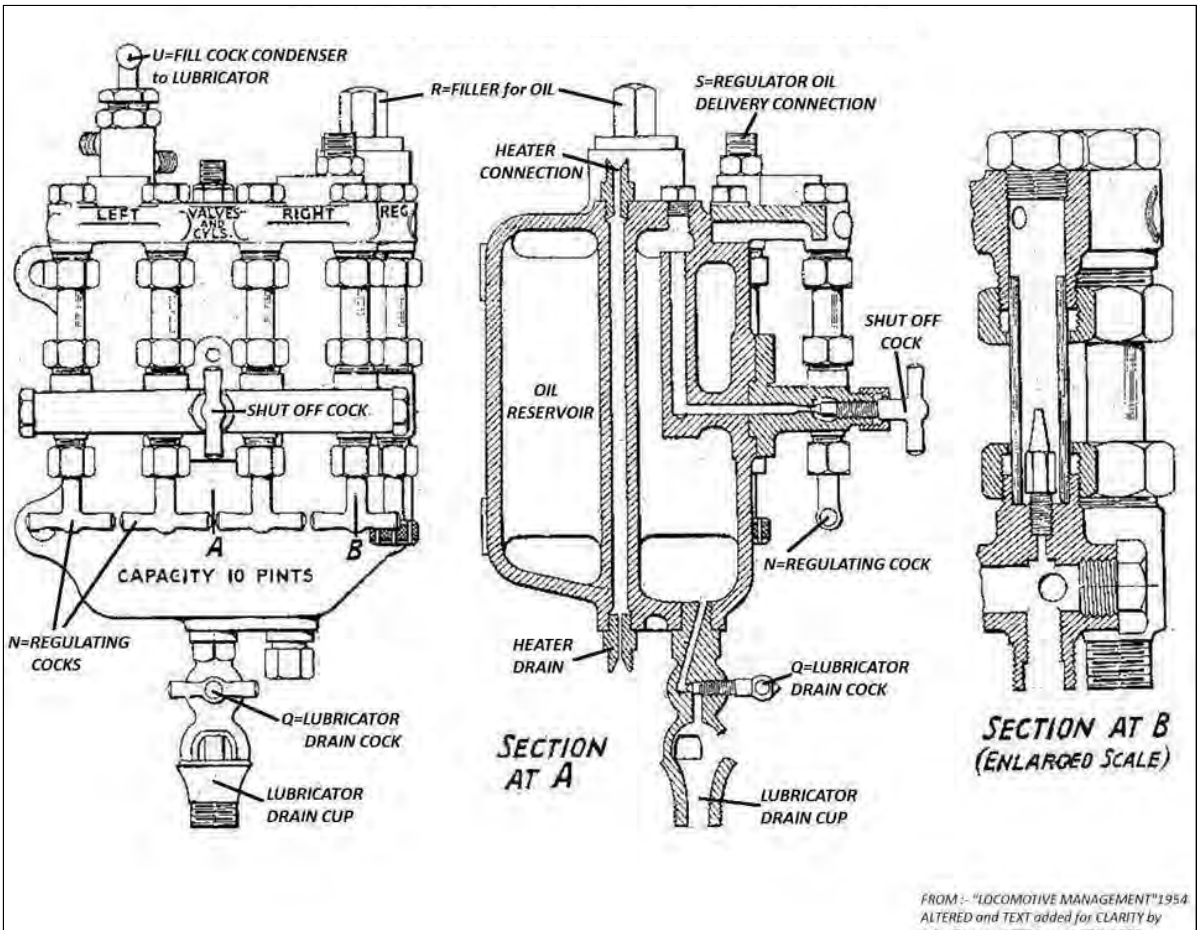


Smokebox layout of 'Star' class engine fitted with 'Castle' type cylinders and outside steampipes

During Collett's time in office the combining valve was altered with an oil feed down the left side of the boiler as well as the right, doing away with the smokebox oil divider where some of the highest temperatures were. The condenser was changed to a double copper coil. The Hydrostatic lubricator was changed to 10 pint capacity with 5 glasses, 2 for cylinders, 2 spare so the Engineman can change over if one breaks, one on RH corner, direct to the regulator valve in the smokebox. These changes will be shown in the next few pages and Collett's instructions of 1937.

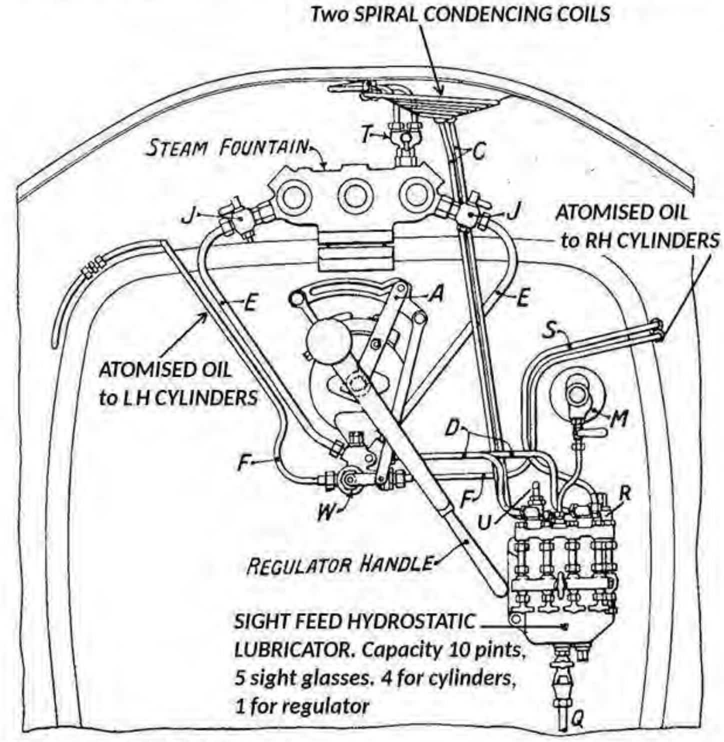
Most GWR steam locomotives in operation today conform to the basic Churchward/Collett hydrostatic cylinder lubrication. Some have been changed by owners to mechanical lubrication.


GWR 4965 Showing the Final Hydrostatic Lubricating Oil System
WESTERN
Lubricator Combining Valve
Hydrostatic Lubricator 10 pints


U=FILL COCK CONDENSER to LUBRICATOR
R=FILLER for OIL
S=REGULATOR OIL DELIVERY CONNECTION
HEATER CONNECTION
LEFT
VALVES AND CYLS.
RIGHT
REG
SHUT OFF COCK
SHUT OFF COCK
OIL RESERVOIR
A
B
CAPACITY 10 PINTS
N=REGULATING COCK
N=REGULATING COCKS
HEATER DRAIN
Q=LUBRICATOR DRAIN COCK
Q=LUBRICATOR DRAIN COCK
SECTION AT A
SECTION AT B (ENLARGED SCALE)
LUBRICATOR DRAIN CUP
LUBRICATOR DRAIN CUP
FROM :- "LOCOMOTIVE MANAGEMENT" 1954
ALTERED and TEXT added for CLARITY by
John Duncan ASTT August 27th 2020

THE FINAL SCHEME OF THE G.W.R. SWINDON HYDROSTATIC LUBRICATOR.

Till the inroduction of the 4 row superheater and Mechanical Lubrication of valves and cylinders in the latter part of the 1940s and 1950s. Before dieselisation and end of steam.

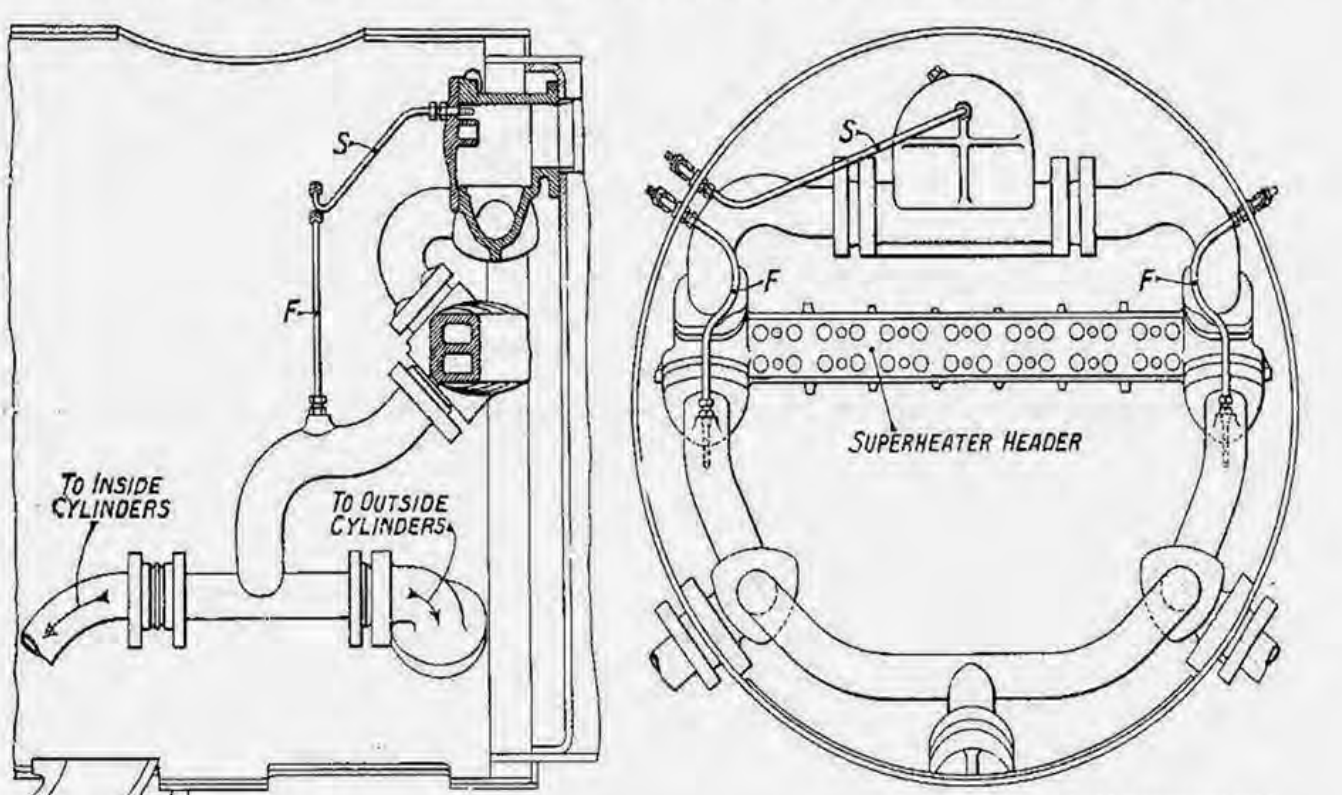


The improvements were :-

1. *Seperate feeds to Left and right hand cylinders.Mainly for 4 cyl.engines and to give a more balanced feed.*
2. *twin coiled condencer to give a better head to the hydrostatic lubricator.*
3. *Larger capacity sight feed hydrostatic lubricator. More reliable and a capacity for long non-stop runs with 4 cylinder engines.*

From :- Locomotive Management, 1954, 10th edition.
Lettering by J.Duncan. ASTT. August 18th 2020

G.W.R. SMOKEBOX DIAGRAM OF LUBRICATION PIPES FOR 4 CYLINDER ENGINES



S = OIL FEED PIPETo REGULATOR. F = OIL FEED PIPES to CYLINDERS

From :- "Locomotive Management" 1954
Remove lines & added text. J.Duncan ASTT
August 26th 2020

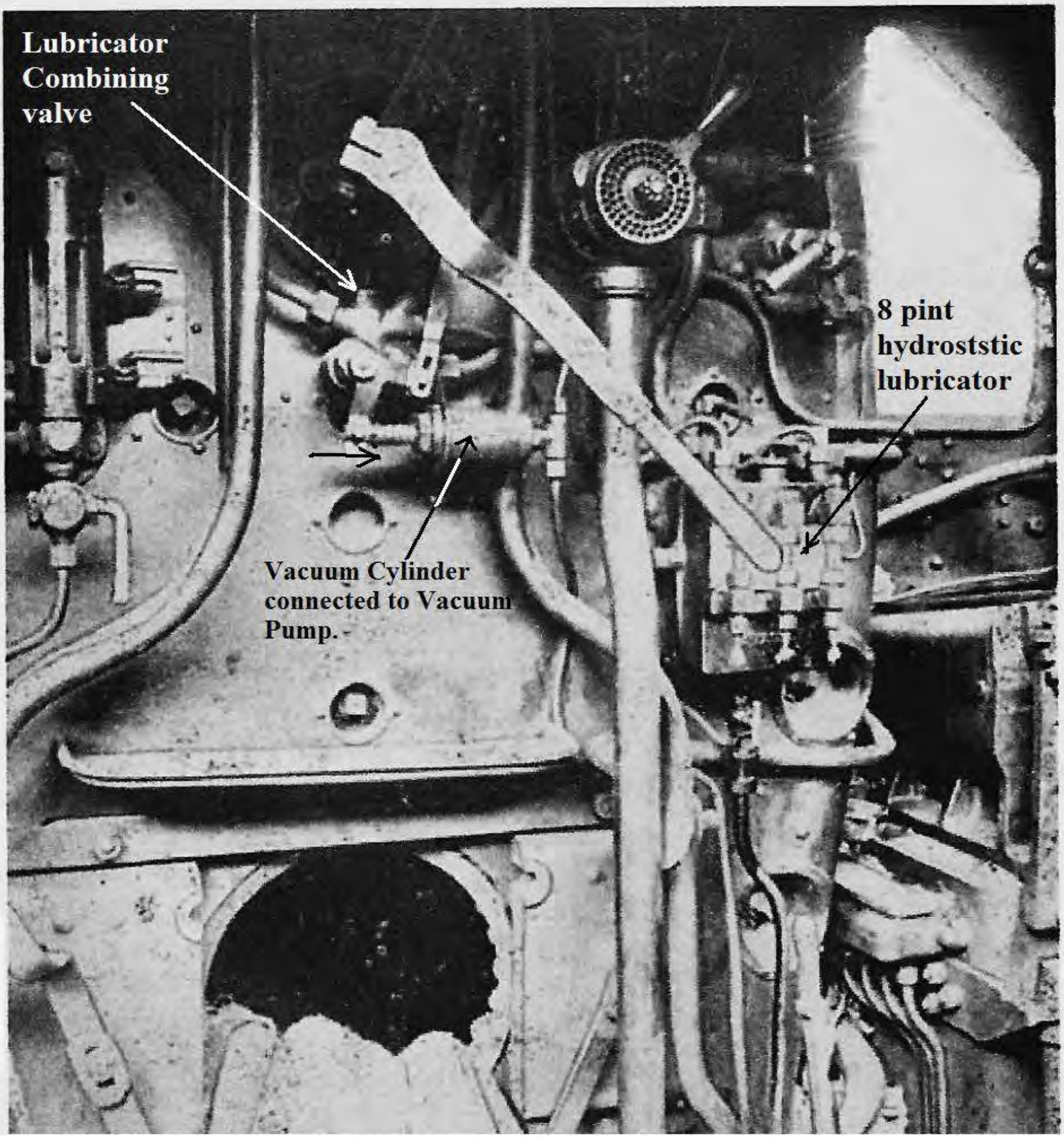
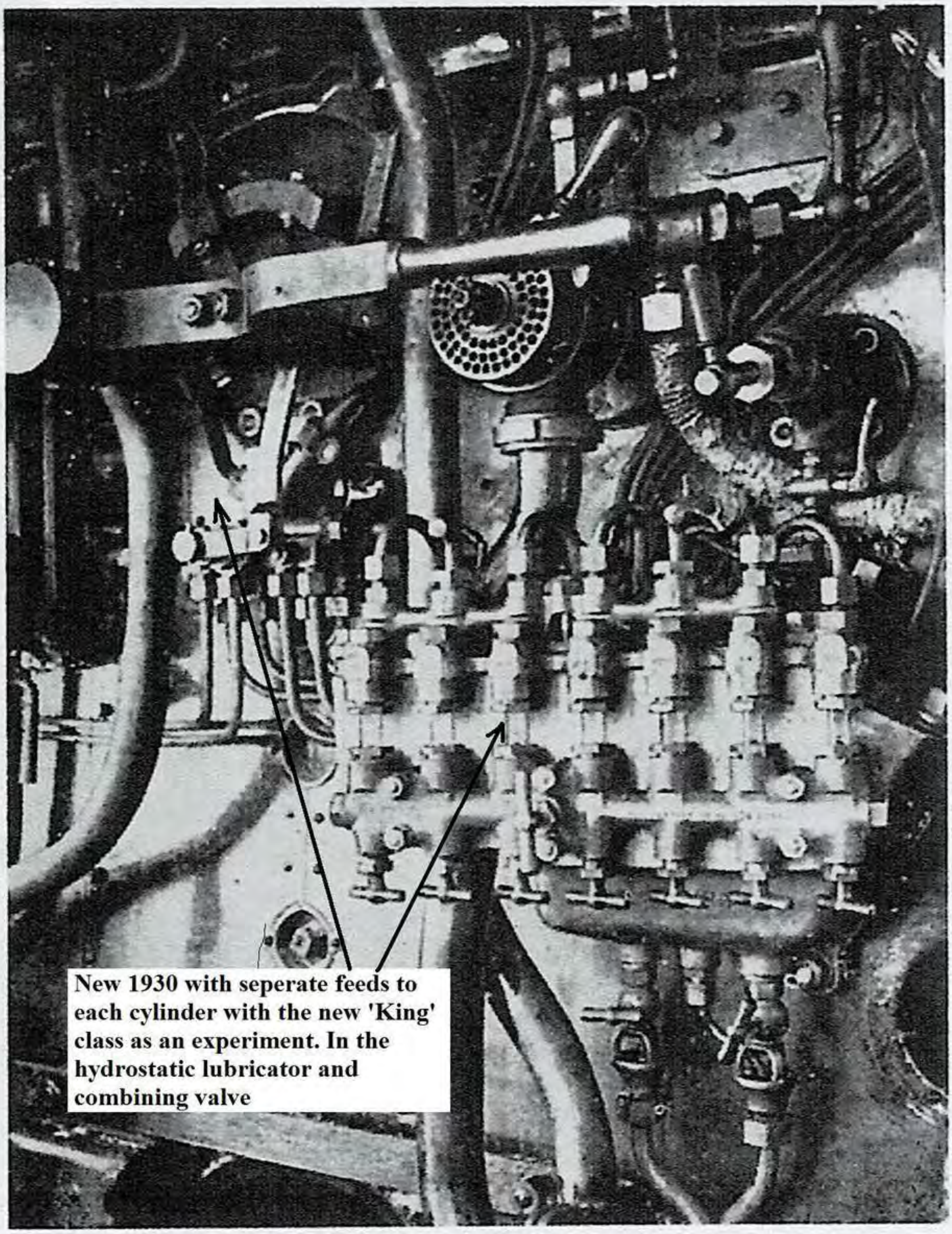


Fig. 11

Experiment to allow oil to flow to the cylinders when the regulator is shut. The slot in the operating rod allows the combining valve to open by partial vacuum from the vacuum pump and stops when the engine is at rest. Maybe it was used on suburban engines.

From a Paper by W.H.Pearce (Member of the Institute of Locomotive Engineers at the GWR Swindon Engineering Society on October 22nd 1931 Paper No.182 titled "Recent Developments in Cylinder Lubriction". Captions by J.Duncan ASTT 1-9-2020.



Described in Paper 182, "Recent Developments in Cyilinder Lubrication" by W.H.Pearce, on pages 18,19 & 20 of a new Hydrostatic Lubricator and Lubricator Combining Valves with individual feeds to left hand and right hand cylinders.

The Hydrostatic Lubricator has an 8 pint chamber for Cylinders and 1.5 pint chamber fot the Regulator Valve on the far right of the lubricator. The Left of the lubricator the 3 LH glasses are are 2 for LH cylinders and 1 spare. the same for the RH cyilinders on the next 3 glasses. Individual control valves fot each feed and a main shut off valve above.

The lubricator Combining Valves have seperarate feeds in and out to the cylinders on each side. See the full description in paper 182.

text above by John Duncan ASTT 17-09-20

GREAT WESTERN RAILWAY.

Circular No. 5801
(This cancels Circular No. 3744 of March, 1922.)

Swindon Improved Triple Sight Feed Lubricator - 8 Pints Capacity
Swindon Modified Triple Sight Feed Lubricator - 4 Pints Capacity
Swindon Modified Triple Sight Feed Lubricator - 10 Pints Capacity
Swindon Modified Five Sight Feed Lubricator - 10 Pints Capacity

By means of these lubricators, oil drops are fed through sight feed glasses and conveyed with steam in small delivery pipes to the main steam pipes, thence to the cylinders, thus lubricating the valves and pistons. Oil drops are also fed from a sight feed glass through a small bore oil pipe, directly to the regulator valve.

The arrangement of the system ensures that a supply of oil reaches the cylinders immediately on starting the engine. Also, by means of a special combining valve, the supply can be maintained while drifting, by placing the pointer attached to the regulator handle in the drifting position. The oil supply ceases shortly after the regulator is fully closed.

PARTICULARS OF LUBRICATORS.

IMPROVED TRIPLE SIGHT FEED—8 PINTS CAPACITY.

2 glasses for oil to cylinders (R.H. glass and centre).
1 glass for oil to regulator (L.H. glass).
SINGLE combining valve "W" under regulator handle.
1 delivery pipe "F" to oil distributor "O" in smokebox.

MODIFIED TRIPLE SIGHT FEED— 4 PINTS CAPACITY.
MODIFIED TRIPLE SIGHT FEED—10 PINTS CAPACITY. See Page 8.

2 glasses for oil to cylinders (L.H. glass and centre).
1 glass for oil to regulator (R.H. glass).
1 central valve "A" to control oil to the 3 nipples (on 4 pints lubricator).
SINGLE combining valve "W" under regulator handle.
1 delivery pipe "F" to oil distributor "O" in smokebox.

MODIFIED FIVE SIGHT FEED—10 PINTS CAPACITY.

2 glasses for oil to R.H. cylinders (R.H. front glasses).
2 glasses for oil to L.H. cylinders (L.H. front glasses).
1 glass for oil to regulator (R.H. corner glass).
1 central valve "A" to control oil to the 5 nipples.
DOUBLE combining valve "W" under regulator handle.
2 delivery pipes "FF" (1 to oil spray nozzle in L.H. Steam pipe).
(1 to oil spray nozzle in R.H. Steam pipe).

NOTE.—Existing Improved Triple Lubricators are being replaced by Modified Lubricators.

WORKING OF LUBRICATORS.

TRIPLE SIGHT FEED.

Oil is fed from the cylinder sight feeds and passes through the oil pipe "D" to single combining valve "W." Steam through the pipes "E" and "V" (or "V" only on small engines) picks up the oil from the valve "W" and carries it through the auxiliary pipe "F" to the oil distributor "O" in the smokebox, and then through pipes "PP" to the oil spray nozzles in the steam pipes.

FIVE SIGHT FEED.

Oil is fed through the pipes "DD" to the double combining valve "W," mixes with the steam from "E" and "V," and passes through the delivery pipes "FF" directly to the oil spray nozzles in the steam pipes.

With each type, the oil feeds are controlled by nipple valves "N." Where a central valve "A" is fitted, this must first be opened before oil can be fed to the nipple valves "N."

WORKING OF COMBINING VALVE.

The regulator valve does not admit steam to the cylinders until the pointer on the regulator handle has been moved about $\frac{3}{4}''$ from its stop, but the combining valve "W" is arranged to open just before this position has been reached. When the regulator handle is placed in this position while the engine is drifting, the oil supply to the cylinders is maintained. On most engines, a pin on the pointer of the regulator handle engages with a catch on the stuffing box when the handle is in the drifting position.

WARMING THE OIL.

To warm the oil in the lubricator in very cold weather, steam may be passed through the lubricator by partially opening cock "M," but it should only be used when the oil is too thick to feed.

INSTRUCTIONS.

FILLING OF LUBRICATOR.

Turn handle of condensing cock "U" on lubricator to the position marked "off" and then open drain cock "Q," remove filling plug "R," allow only water to drain off, close cock "Q," and fill slowly with warmed oil through the filling inlet. If for any reason there is not sufficient oil to fill the lubricator, always make up with water before replacing the filling plug "R." It is very important that the lubricator should be absolutely filled.

To prepare the sight feeds, close each nipple valve "N," remove the top plug over each glass, see that the glasses are clear, fill with clean water and replace the top plugs. See that the handle of the 3-way cock "T" on boiler is pointing vertically downwards, turn the handle of cock "U" on lubricator to the open position, and the lubricator is ready for use.

GENERAL INSTRUCTIONS.

It is important that the "J" valve or valves on fountain be fully open before the engine is moved. If, however, the combining valve "W" is found to be leaking (as shown by emission of steam through the open cylinder cocks) the "J" valves must be closed when the engine is off duty.

It is imperative to start the cylinder feeds before moving the engine. This will ensure lubrication of the valves and pistons when the engine starts.

The pins and rubbing surfaces controlled by the regulator handle should be oiled occasionally.

IN THE CASE OF A BURST CONDENSING PIPE "CC," THE HANDLE OF THE 3-WAY COCK "T" MUST BE MOVED THROUGH AN ANGLE OF 90 DEGREES TOWARD THE OTHER PIPE. This shuts off steam from the defective pipe, but maintains a supply to the unbroken pipe. The handle of condensing cock "U" on the lubricator must now be moved round to the other "on" position stamped on the plug body.

In cases of burst delivery pipes "F" in cab or under cleating, close both "J" valves and the stop cock of pipe "F" on R.H. or L.H. side of smokebox. In case of a burst delivery pipe "F" in smokebox, close the stop cock on smokebox. If accessible, the broken end of pipe "F" may be hammered over. In either case, the engine must be taken to Shed as soon as possible after the failure, and whilst this is being done extra oil must be used through the regulator sight feed. In case of a burst regulator oil delivery pipe "S" in cab or under cleating, close the regulator nipple valve "N" and the top stop cock on the R.H. outside of smokebox.

For periodical and easy cleaning out of lubricators, an additional draining plug is fitted at the bottom of 4 pints and 10 pints capacity lubricators.

Special strainers under filling plug "R" must not be removed except for cleaning purposes.

In frosty weather, run all water out of lubricator by opening cock "Q," before putting engine away.

ADJUSTMENT OF FEEDS.

All these lubricators have two glasses for each independent cylinder feed. Oil may be passed through either (thus keeping one spare) or both of these glasses at the same time, but the rate of feed must be adjusted to suit the number of glasses in use. The total feed to the cylinder should vary according to the type of engine, and engine working conditions :—

Type of Engine.	*Total No. of Drops per Minute through Cylinder Glasses.*
4-6-0 "King" Class only	20 to 36
4-6-0 except "King" Class	15 to 30
4-4-0 2-6-0 2-6-2T 2-8-0 2-8-0T 2-8-2T ...	10 to 20
0-6-0 0-6-0T 0-6-2T	5 to 10
2-4-0 2-4-0T 0-4-0T 0-4-2T	4 to 8

Do not adjust the oil feed more than necessary, as it should vary but little for different openings of the regulator.

Careful adjustment of each needle valve "N" (up to one complete turn) will give the required feed. If it is found necessary to increase the rate of feed, in very cold weather, the right hand "J" valve may be very slightly closed. Overheating the oil to secure the required feed by the warming cock only must be avoided.

The regulator sight feed glass supplies oil alone from the lubricator directly through the pipe "S" to the regulator valves. A small quantity of oil should be fed to these valves occasionally during each run, particularly with 4-6-0 engines.

If engine is required to remain standing for some time, economy of oil may be effected by closing central valve "A," when fitted, or condensing cock "U" on lubricators not fitted with valve "A."

DRIFTING.

When the engine is to be run without steam, the regulator handle should be first shut and then opened to the drifting position. On those engines not provided with the drifting position catch, the pointer must be moved $\frac{3}{4}$" from the shut position stop.

FAILURE OF GLASSES.

Should a cylinder glass fail, the needle valve "N" controlling that glass must be shut, and the spare feed brought into use as follows :—

Improved Triple Sight Feed.

Turn handle "L" over centre feed to the position marked "spare," open needle valve controlling the spare feed.

Modified Triple and Modified Five Sight Feed.

Open the needle valve "N" that controls the spare feed, if not already in use.

Should the regulator glass break or clog, it must be replaced or cleaned, as no spare feed is provided.

CHIEF MECHANICAL ENGINEER'S OFFICE, SWINDON. — C. B. COLLETT.

November, 1937.