ASTT Newsletter No. 15

JANUARY 2021

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Chairman's Piece John Hind

Since I wrote my last column back in August, the world has moved on again! We were able to enjoy a short respite in the Autumn, then another lockdown, the Tier System, the release of the vaccine, the emergence of a new variant and now Tier5 (someone must be grateful that they did not make Tier 1 the worst case). Having been through all that, one of our members and his wife have had their first inoculation, so there is hope for 2021.

Despite the setbacks of this year, we have been looking at 2021 and at present don't have confidence to book any venues for our AGM and Conference, so like many other organisations we are planning to hold our 2021 AGM using video conferencing software, with Zoom being the favourite, which most will be familiar with as it seems to have captured the market for families keeping in touch.

Following our established pattern, the AGM will be on first Saturday in March which is 06/3/21. Rather than just have an AGM meeting we have always added in a couple of talks, so if anyone has a talk they can deliver on-line please let me know by e-mailing me at john.hind@advanced-steam.org

Similarly, the Conference would be on the first weekend in October which is 2 & 3/10/21. As life evolves in 2021 we'll take a decision in April whether to hold it as virtual event or as a true physical event where we can meet up.

We would also like to repeat the day we had at the Stapleford Miniature Railway (see report below). The format was different to the other events we hold as it allowed more time for chatting and socialising without the Chairman haranguing everyone with his eye on the agenda! This would be sometime in the summer/early autumn, date to be agreed with the SMR.

Cedric is willing to continue as Secretary with David Nicholson acting as Deputy. Production of the newsletter is currently a joint effort between David Nicholson, Chris Newman, Mike Horne and myself – if anyone would like to take over as Newsletter Editor please let me know.

Bio-coal Update John Hind

As the prospects for running trials in 2021 looked better, we contacted CSR in October about supplies for 2021 and a price. CSR have met with the University of Minnesota, who are working on pricing trial batches of bio-coal. There has been interest from across the U.S. and globe, and they hope 'to be able to coordinate a multi-continental, multi-party research exercise where we can utilize a set series of research methodologies to try multiple batches of fuel'.

Once we have the price, we will be in touch with the HRA to confirm railways that are interested in taking part in the trials.

Existing stocks of English mined steam coal will be used up in early 2021 and the last Welsh mined coal will be in 2022. The Heritage movement is starting to use Russian supplied coal, with the unintended consequence that, with the CO_2 involved in transporting it, the emissions from the Heritage sector increase! - clearly not joined up thinking!

With the UK's Governments current target of having net zero emissions by 2050 and the continuing tide against burning coal, some form of change will be needed by the UK Heritage Railway Industry, whether that is by burning bio-coals or alternative fuels that are seen to be more environmentally friendly remains to be seen.

Stapleford Miniature Railway John Hind

Saturday 19/9/2020

The weather proved to be kind for us at the Stapleford Miniature Railway. It did not rain, but it was windy. The SMR did us proud by steaming 3 engines for us – the Curwen Atlantic, the New York Central Niagara and the New York Chicago and St Louis Berkshire. Runs round the track were on demand and all 3 engines featured, though the Curwen had the lion's share of the runs. The Curwen was instrumented to continue some testing that we had done earlier in the summer and we were puzzled by the results on the day when we were expecting to see an improvement in drafting, when it in reality it struggled for steam. The mystery was solved when on disposal it was found that the petticoat had displaced during the days running.



No 752 New York, Chicago and St Louis Berkshire



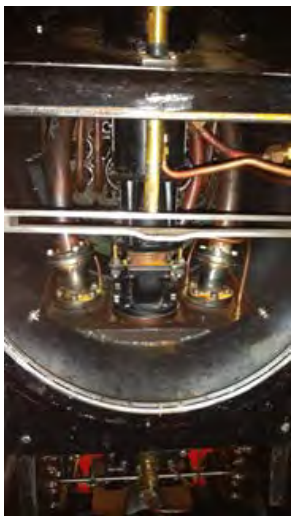
No 6019 New York Central Niagara



Curwen Atlantic

We were also able to look around John Wilks's workshop where his latest project a P2 2-8-2 is nearing completion. The Pictures do not do justice to size of the locomotive or the attention to detail.





P2 - Cock of the North and detail in the Smokebox

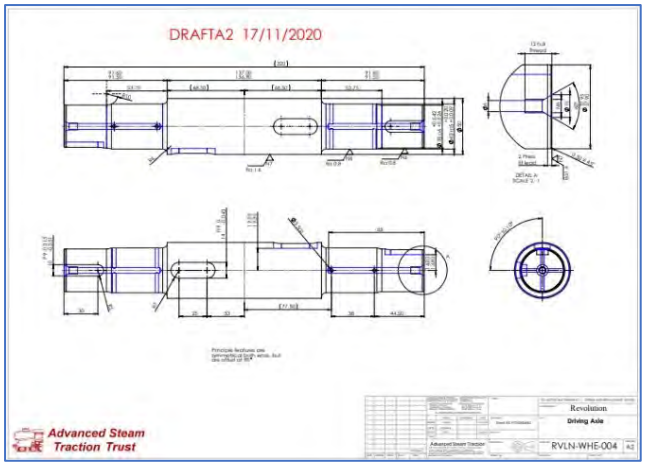
Video Recordings: A 5 minute composite video combining of some of the day's runs can be seen at https://www.youtube.com/watch?v=n83YHCTL-Qc.

Attendees: The gathering was attended by the following members:

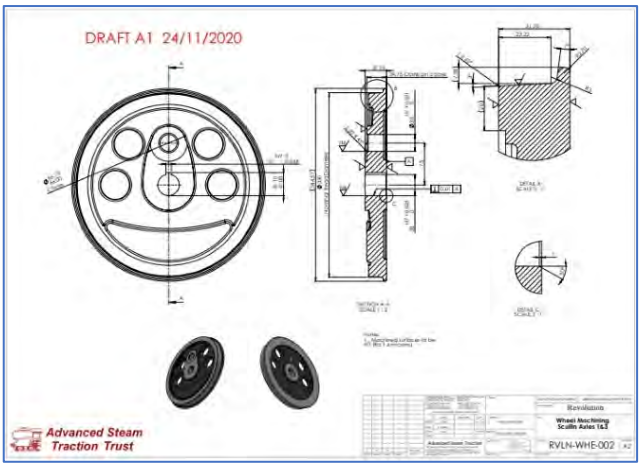
John Hind; Richard Coleby; Jamie Keyte; Grant Soden; Andrew Hartland; Alex Powell; Chali Chaligha; Robin Pennie; Graham Shirley; Owen Jordan; Paul Hibberd; Terry McMenamin and Chris Newman

Revolution Update John Hind / Jamie Keyte

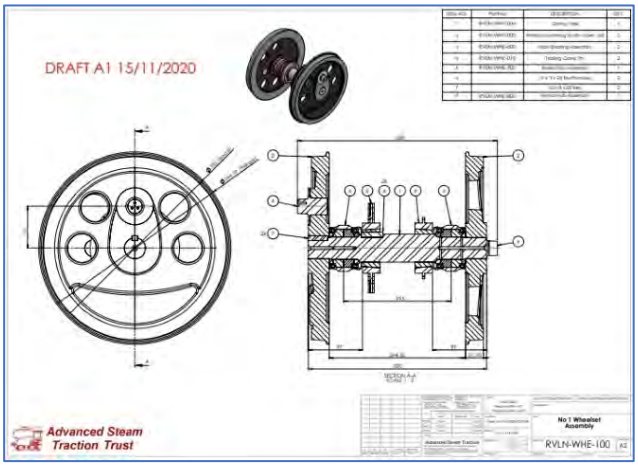
A new phase is starting – with 3D CAD systems, the first phase is to produce a 3D model of the part. The next stage is to produce the traditional drawings, which are referred to as 2D and show the actual dimensions and the tolerances for those parts. This is a slower process than the 3D modelling as more detail goes onto the drawing. 2D's have to be produced for each part and then an assembly drawing. These are some for the driving wheelset and shown are two out of a total of 7 drawings for the assembly. These are now being checked before sign off for manufacture.



Axle

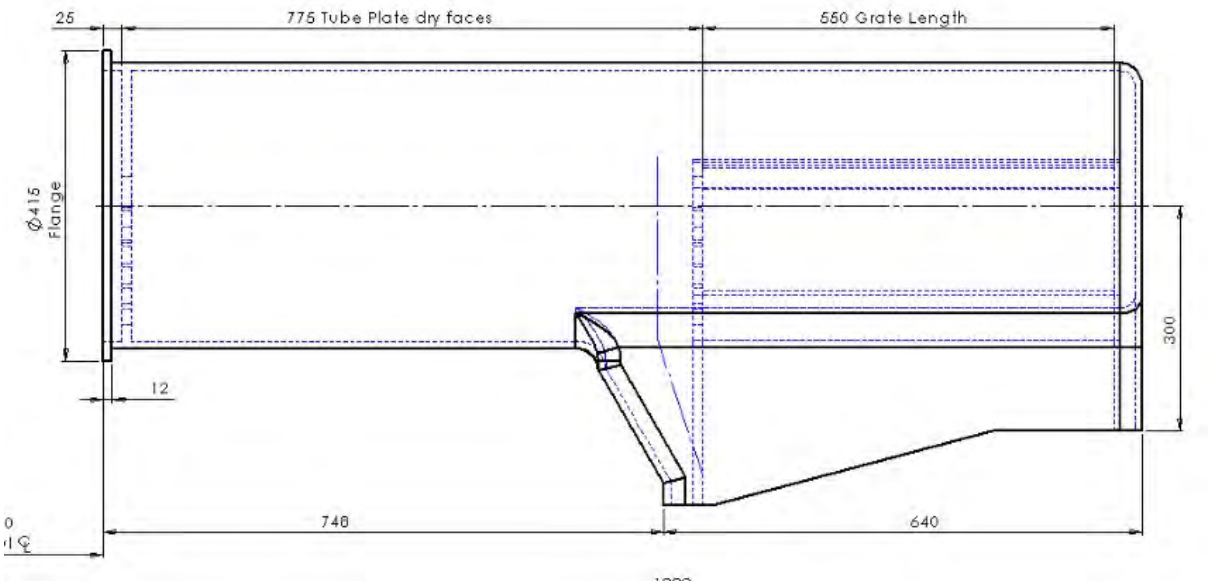


Wheel



Driving Axle Assembly

Boiler – there has been some debate about whether there is any benefit in moving the tube plate forward (as shown by the chain dotted line on the diagram below) to increase the firebox volume and the radiant heating surface at the expense of reducing the tube length by circa 55mm.



Once again Martin Johnson's Boiler Programme showed that there is predicted to be a very small decrease in performance, so we can start the detailed design based on the vertical tubeplate.

Tender - our gathering at the SMR gave Grant a chance to look at the ergonomics of the tender by sitting in the Curwen's tender, which has given some food for thought to improve the ergonomics and the functionality are being worked on before we finalise the design.



Grant Sodon sitting in the Curwen's tender during the Stapleford outing.

Design Work Packages - to date, we have not had anyone volunteer to take on the design of the Bolt on Front Framing, Braking, Lubrication, Superheating and Ashpan. If anyone is interested please contact Jamie. It's a great way of getting involved in a unique project.

Membership Chris Newman

Upgrades to Full Membership

One upgrade has been made since September: our webmaster Grant Soden has upgraded to full member of ASTT.

Any Associate Members 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.

Membership Numbers

Three new members have joined since April.

William Roebuck from Carmarthenshire (Wales): a 3½" gauge model builder using Lempors etc. Has volunteered for workshop duties at the Welshpool & Llanfair Railway.

Mitch Kennedy from the USA (Connecticut): a professional engineer, self proclaimed "energy wonk" and Member of the Society of Manufacturing Engineers; "avid mechanically inclined tinkerer, volunteer on local tourist railway, and writer, educator on the energy and environmental benefits of railroad commerce and transport over gasoline powered wheel-transport of goods and people". 31 years experience in manufacturing including machine shops, foundries, aerospace, electroplating, and power plants. As a process optimization engineer, has sought efficiency gains in many production and design processes. A published author on financing energy projects. Project manager for renewable and conventional energy use, reduction and generation, including cogeneration projects.

David Pritchard from Cambridgeshire (England): Propulsion architect with Airbus, prime level architect responsible for a spacecraft propulsion system manufactured by an external supplier. Has been involved in an extensive rebuild to a rolling chassis of the now operational steam locomotive "Russell" dating from 1906; mechanical rebuilds of three coaches from the 1890s, and rebuilding the chassis structure of Metropolitan Railway number 1, which has since operated on the London Underground during anniversary celebrations.

Membership now stands at 85, broken down as follows:

Full Members:	20	UK members:	61
Associate Members:	59	EU:	12
Student Members:	6	USA	5
		Australasia:	6
Total Membership:	85	China:	1

ASTT Facebook Group Iain Jack

We set up our Facebook Group way back in 2013 and currently, we have 380 members (membership is free, but they do not get the newsletter, discounts on publications or invites to members-only events) and all vetted as having a genuine interest in our aims. Amongst these, we have gained some high-profile followers including CEOs, CMEs, etc. However, currently, we are just not capitalising on all this interest.

ASTT rarely posts to the Group apart from Event Notices and the occasional repost from other relevant groups, certainly nothing has been added this year. There are sporadic outbreaks of discussion amongst the members, but it certainly isn't the go-to place for Advanced Steam debate that it could be.

Therefore, over the second lockdown period, myself and Grant Soden took the opportunity to review the status of the ASTT Facebook Group. Some may dislike Facebook, but it is now the default online publicity tool for most successful organisations, heritage steam included, as fewer people now go searching for websites as their first port of call when looking for news and information.

ASTT aims to be an outreaching, evangelical organisation, and so looks to engage with like-minded individuals and potential customers alike. We also need to recruit suitable candidates as ASTT Members and to sell our expertise and publications successfully.

So how do we make this happen? To this end, we researched how other similar organisations are using Facebook and discovered that the more professional ones, such as the A1 Trust, have both a Page and a Group, where we only have the latter.

A Page is an organisation's shop window. It sets out who we are, what we do, our aims and objectives. It contains links to the website (which can be seen as the Library, Reading Room and Archives); a highly visible link to apply for Membership and contact facilities e.g., messaging, email, etc. A Shop facility is also available, which will be developed in the New Year.

Meanwhile, the Group is akin to a club with a strict door policy, where like-minded folk can meet and discuss their common interest in Advanced Steam. We will continue to use the Group to direct people to our Page, and thence to Membership, Sales and conferences, etc.

It is intended that the new Page will be linked to the ASTT website so that news items and event invitations will be posted in a timely fashion and will therefore be visible to our followers in their Facebook feeds. Overall, we trust that these measures will increase both membership applications and book sales.

The new Page can be found here:

https://www.facebook.com/Advanced-Steam-Traction-Trust-ASTT-101809591807279/

Publications Chris Newman

Book Sales

Book sales have picked up a little since August. They are listed as follows :

Volume 1 of Porta's Papers – Tribology and Lubrication (pub. ASTT): 4 sold (total sales 104);

Volume 2 of Porta's Papers – Adhesion, Compounding and the Tornado Proposal (pub. ASTT): 4 sold (total sales 101);

Steam Locomotive Design Specifications and Calculations for New Build Baldwyn 2-4-2T 'Lyn' (pub. ASTT): 5 (total sales 79);

5AT Fundamental Design Calculations (pub. ASTT): 3 sold (total sales 198 since 2015);

The Red Devil & Other Tales by Dave Wardale (pub. Camden): 1 sold (total sales 260 including 233 sold pre-2015 through the 5AT Group);

Here be Dragons by Phil Girdlestone (pub. Camden): 4 sold (total sales 30);

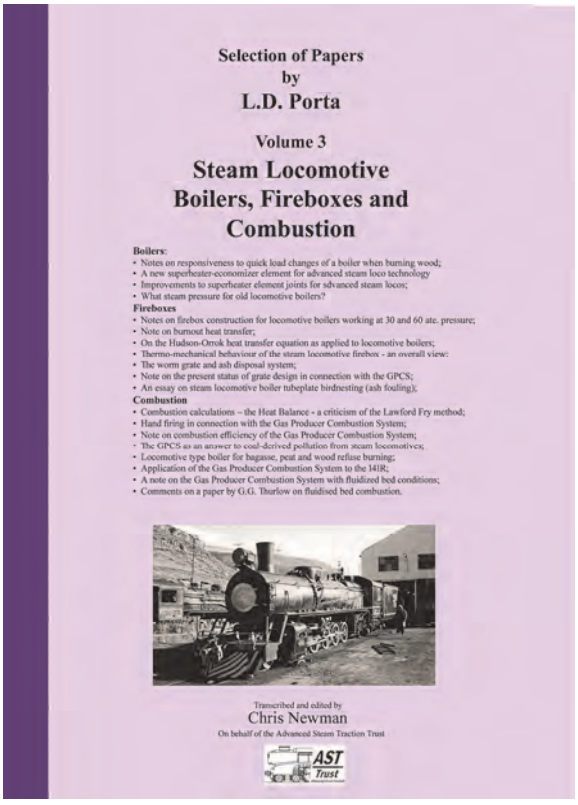
A Defence of the MR/LMS Class 4 0-6-0 by Adrian Tester (pub Crimson Lake): 2 sold (total sales 20)

An Introduction to Large-Lap Valves & Their Use on the LMS by Adrian Tester (pub Crimson Lake): 1 sold (total sales 11)

Important News: Camden report that they have run out of stock of David Wardale's "Red Devil & Other Tales from the Age of Steam". They have no plans for another print run.

New Titles

Porta Papers Volume 3 nearing publication:



The third volume of Porta papers covering boilers, fireboxes and combustion, is nearly ready for publication.

The phrase "Selection of Papers" used in the title would be misleading if it was taken to mean a Darwinian selection – picking the best and rejecting the rest. Unfortunately we don't have that luxury as we can only select from the manuscripts in our possession. So the selection process involves selecting which papers to assemble together within a vaguely defined theme.

Notwithstanding, each of the 19 papers in this volume were of great interest to the transcriber who believes that they all have something to tell us, whether relevant to our endeavours or not.

With 288 pages, the book will be a little larger than Volumes 1 and 2, but it will sell for the same price - £27 for non-members and £21.60 for members.

A brief summary of the 19 papers follows (copied from the Introduction to the volume).

Porta Papers Volume 3 – Description of Contents

Section 1 - Boilers:

1 - Notes on the Responsiveness to Quick Load Changes of a certain Well-Known Type of Boiler when Burning Wood (1978): Porta claimed that this paper was written as a challenge to three Swedish boiler engineers to see if they could identify a wood-burning boiler that could: "reach full capacity for steaming in some 10 seconds, dropping instantaneously, letting some time pass, and then facing varying steam demands etc without significant drop in the steam pressure, and all with reasonable efficiency". Perhaps it was more likely written to illustrate what ordinary locomotive boilers could deliver in terms of flexibility and responsiveness as compared to the stationary boilers that the three engineers were familiar with. It includes a transcription of the British Transport Commission's "Handbook for Railway Steam Locomotive Enginemen" Section 4: 'The Boiler'.

2 - Improvements to Superheater Element Joints for Advanced Steam Locomotive Technology (1982): A study into the mechanism leading to leakage of superheater header-element joints and a proposed solution using displaceable washers and long bolts. It includes an appendix on "On Superheater Bolt Stretching due to Temperature Effects".

3 - A New Superheater–Economizer Element for Advanced Steam Locomotive Technology (1980): Porta speculates on the possibilities of "a fin-tube detachable smokebox economizer which can easily be lifted up for repairs or access to the superheater", comparing it to Chapelon's integral economizer. Porta then goes on to consider how Chapelon's economizer might be improved by incorporating finned superheater elements in the boiler section and internally finned flues ("Serve" tubes) in the economizer section. An appendix has been added to help explain Porta's calculations to readers.

4 - What Steam Pressure for Old Locomotive Boilers? (1999): Written towards the end of Porta's life, this paper was published by the Coalition for Sustainable Rail (www.csrail.org) in 2015 as part of their White Paper programme. This transcription includes sections of the paper missed out in CSR's version, including the two Summaries and Section 5 "Legal Aspects". Some of the wording has also been changed back to (or closer to) the original.

Section 2 - Fireboxes:

5 – Notes on the Flat Plate Stayed Firebox Construction for Locomotive Boilers working at 30 and 60 atm. Steam Pressure (1976): This paper investigates the possibilities of increasing boiler pressures to 30 and 60 atm (440 and 880 psi) while retaining the traditional firebox form with water legs and flat stayed surfaces.

6 – On the Hudson-Orrok Heat Transfer Equation as applied to Locomotive Boiler (1977): A 2-page rather academic paper written for David Wardale's benefit, describing a formula for heat transfer in boiler furnaces, which Porta describes as "good for first preliminary calculations".

7 – Note on Burnout Heat Transfer (1980): Burnout occurs when the heat flux on the inside of a firebox exceeds the rate at which the coolant (water) can extract heat, resulting in an uncontrolled rise in plate temperature. From an empirical theory published by Bernard in a 1960 Chemical Engineers journal, Porta calculates that burnout should not occur at the maximum local heat flux likely to occur in a locomotive firebox (~1 $Gcal.m^{-2}.h^{-1}$).

8 – The Thermo-Mechanical Behaviour of the Steam Locomotive Boiler Firebox – an Overview (1984): The paper reviews the development of firebox design with particular reference to innovations, particularly in the form of flexible stays, introduced by Tross in Germany in 1937 and parallel developments carried out in France during the early 1940s, necessitated by the substitution of steel for copper in the inner firebox. The necessity of eliminating scale and corrosion through effective water treatment is also emphasized. Two appendices are provided, spelling out methods of repair to damaged fireboxes and on the design of thin (5/16") tube plates.

9 – The Worm Grate and Ash Disposal System (c.1976): The paper presents Porta's bold imaginings of what could be done to improve the environmental impact of steam locomotives by eliminating the ashpan by removing ashes from below the firebed and conveying them to a sealed container fitted between the bogies of the tender. His approach offers means of cleaning up the design of the grate/ashpan area and perhaps even lowering the level of the grate to allow a deeper firebed.

10 - Note on the present status of Grate Design in connection with the Gas Producer Combustion System (1993): This paper describes various proprietary US grate systems together with some of Porta's own designs, both tried and tested and conceptual, with particular emphasis on clinker break-up and disposal. It includes an innovative firebox and grate arrangement for his large MALLET proposal for the Rio Turbio 750mm gauge line in Argentina.

11 - An essay on steam locomotive boiler tubeplate birdnesting (ash fouling) (1983): The paper offers useful insights into the factors that contribute to ash fouling, and offers qualitative solutions for overcoming or minimising it.

Section 3 - Combustion:

12 - Locomotive Boiler Combustion Calculations – The Heat Balance - A Criticism of the Lawford Fry Method (1974): This was the most challenging of papers for the transcriber. The unusually rough nature of the manuscript suggests that Porta wrote it in a hurry and failed to proof-check it properly - or perhaps it was just a rough draft. However in addition to identifying and correcting errors, it demanded an understanding of Fry's theories and of Porta's criticism of it, for which the transcriber was gratified to be able to call on the invaluable assistance of ASTT members Dr. Louis Wibberley, John Boutwood and especially Martin Johnson. Porta's criticism derived from Fry's assumption that composition of coal particles lost through the chimney are the same as that of the coal as fired, whereas Porta believed that only char (unburned carbon) is lost, all hydrogen and volatiles in the coal being combusted. An appendix has been added explaining the derivation (by Martin Johnson) and purpose of what Porta termed "the Combustion Triangle".

13 – Hand Firing in connection with the Gas Producer Combustion System (1976): In fact this is three papers in one: Part 1 was written to advise David Wardale on converting the firebox of SAR 19D No. 2644 for GPCS operation; Part 2 contains more generalised advice on the hand-firing of GPCS fireboxes, while Part 3 offers advice on the design of the fire door for 2644. The three parts combine to form one of Porta's best illustrated papers, demonstrating his not-inconsiderable artistic skills.

14 - Note on combustion efficiency of the Gas Producer Combustion System (1980): Some of this short paper may not have been authored by Porta even though the second half is transcribed from Porta's hand-written manuscript. The first part – tabulated analyses of coal and ash - appears to derive from the UK National Coal Board's Coal Research establishment at Stoke Orchard, who presumably analysed samples that were carried back from Argentina by John Click following his 1977 visit to the Rio Turbio Railway on a United Nations Industrial Development Organisation (UNIDO) mission. Quite possibly the descriptive text following the tabulations was written by Click to emphasise the effectiveness of Porta's Gas Producer Combustion System when burning the high ash, high volatile Rio Turbio coal and the very low carry-over achieved despite its very small particle size.

15 - The Gas Producer Combustion System as an answer to coal-derived pollution from steam locomotives (1990): Perhaps best described as an update on Porta's earlier papers, it focuses on the reduction in chimney emissions achieved by the adoption of his Gas Producer Combustion System. However it includes descriptions of environmental control of ash disposal and reducing lubricant losses. It includes a 2½ page appendix listing "Items Included in a Typical Locomotive Modernisation".

16 – Locomotive-type Boiler for Bagasse, Peat, and Wood Refuse Burning (1992): The title reveals that this paper is about locomotive-type boilers rather than locomotive boilers *per-se*. It is nevertheless interesting in that it offers insights into drying and combustion of biomass fuels, and speculates on how a locomotive-type boiler might improve the performance and/or reduce the cost of normal stationary biofuel boilers.

17 - Application of the Gas Producer Combustion System to the 141R - An Exercise (1998): A most interesting paper describing how a (US-built) French 141R might be converted to incorporate a Gas Producer Combustion firebox. It was written immediately following the publication of David Wardale's "Red Devil and Other Tales from the Age of Steam" in which Wardale described the difficulties he experienced achieving success with the GPCS firebox in his Class 26 4-8-4 "The Red Devil". Clearly Porta wanted to demonstrate how such a retrofit might be more successfully undertaken. Appendix 2 is titled "Combustion of Biomass by the GPCS".

18 - Comments on a paper by G.G. Thurlow on fluidised bed combustion (c.1978):
Two pages of comments and calculations expressing Porta's skepticism about the potential use of fluidized beds for combustion, saying that the evaporations quoted in Thurlow's 1978 paper were "same as for GOSS's ninety years old 55 lb/ft^2/h of firebox heating surface of 1890s locomotives".

19 - A note on the Gas Producer Combustion System with fluidized bed conditions (1980):
Written two years after Thurlow's paper, this appears to have been an intellectual exercise in which Porta speculated about what experimentation might be tried (as opposed to a serious research proposal). The concluding paragraph contains a strong hint that he was responding to pressure from a Fluid Bed Combustion lobby, and that he considered the exercise one of futility – even at "chicken-feed" expense.

The paper offers a glimpse of Porta's mindset when undertaking work for INTI that had no direct connection to locomotives. Notwithstanding, he couldn't resist putting forward the suggestion that, if a trial was to be conducted, it should (somehow) be undertaken by adapting a (presumably stationary) locomotive to handle an amalgam of GPCS and Fluid Bed Combustion technologies.

Series on Mechanical Lubrication Part IV John Duncan

Editor's Note:

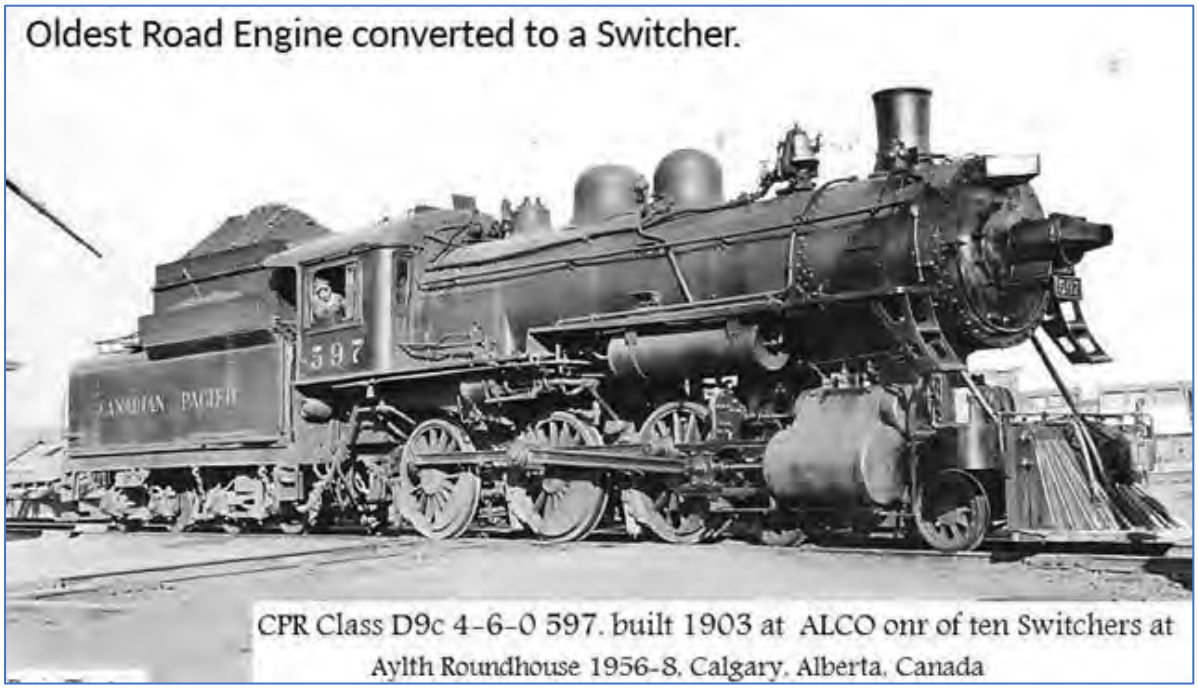
John Duncan's article describes lubrication practices in the UK and North America in the 1950's. L.D. Porta's "Tribology & Lubrication" papers published by ASTT in "Selection of Papers by L.D. Porta Volume 1 – Tribology and Lubrication" describes current thinking and is available from our website.

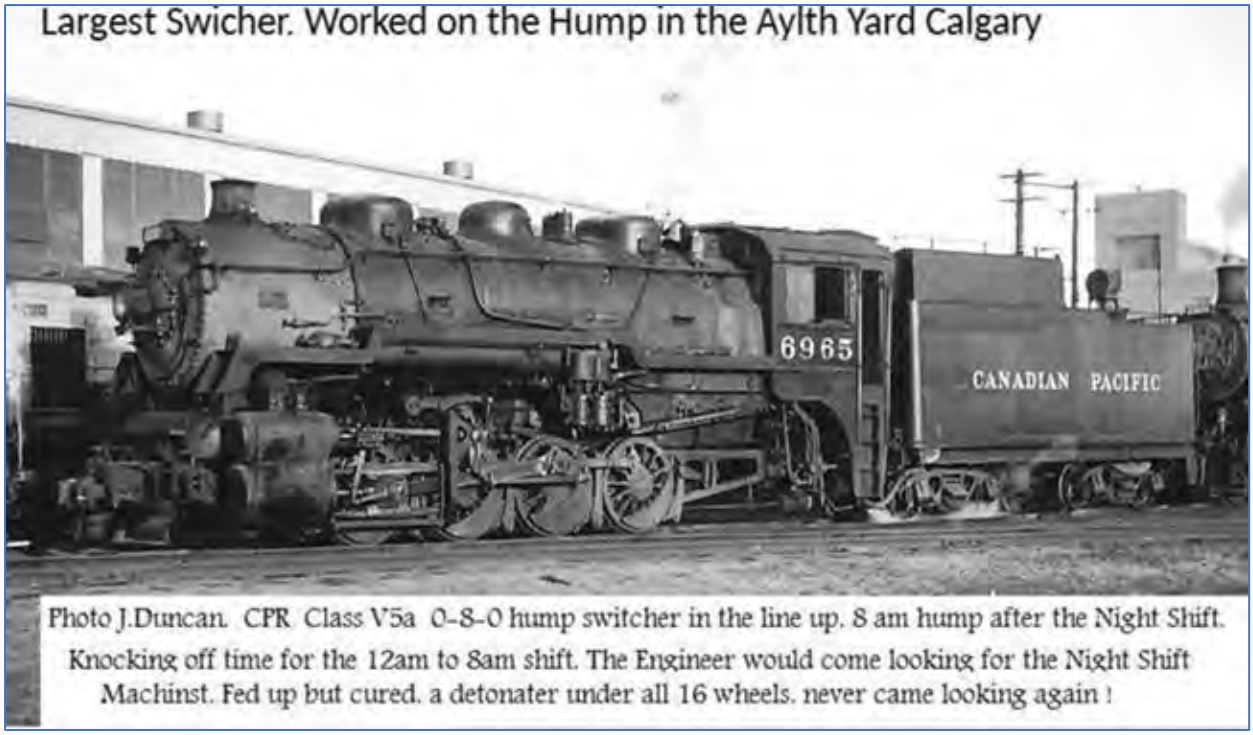
C.P.R. Cylinder & Valve Mechanical Lubrication

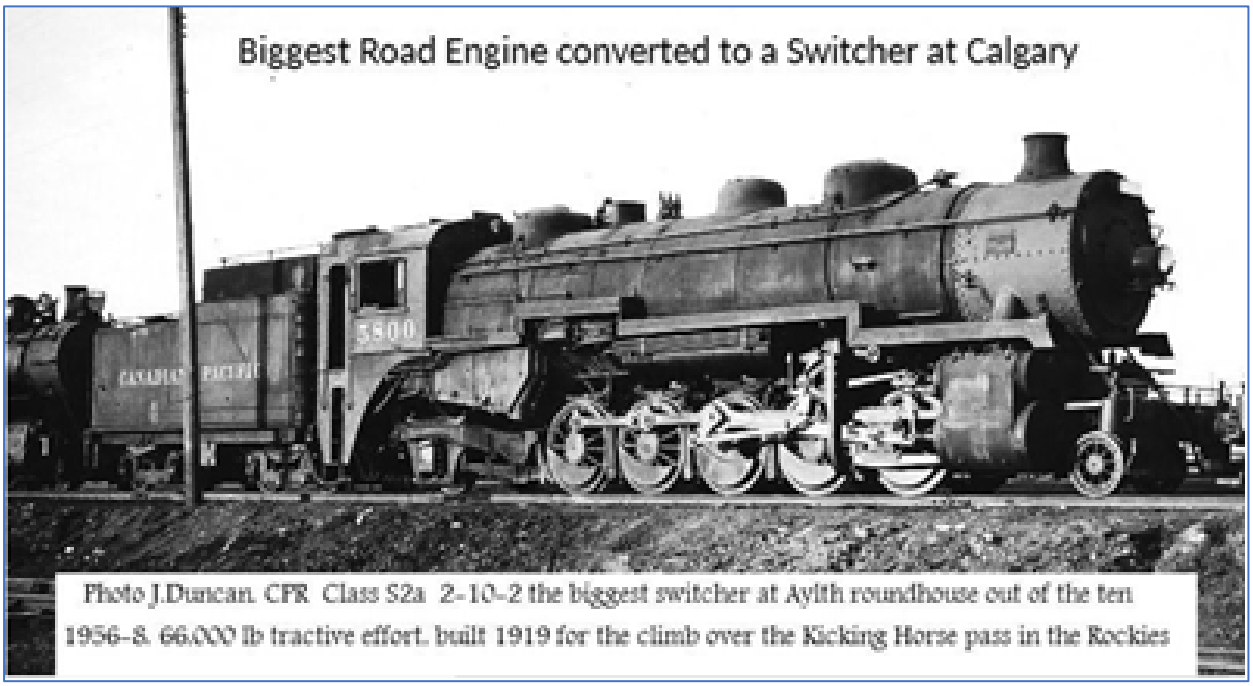
In 1956 to 1959 I worked as a Machinist/Fitter on the Canadian Pacific Railway, in Calgary, Alberta, at Aylth Roundhouse and in the last year in the Diesel Shop. Steam traction finished in 1958 in the Calgary Division.

The first 3 months were a trial period on road locomotives working on superheater elements and axlebox pedestal wedges on T1c 2-10-4s, H1c 4-6-4s, and G3g 4-6-2s. Piston were fitted with segmental 'Kopper' rings and mechanical lubrication was with Nathan valveless mechanical lubricators, one mechanical lubricator for each cylinder, very similar to the S160 2-8-0's in the UK.

After the 3 months, there was a Westinghouse air brake examination to pass. I was then taken on as a machinist on permanent nights, 12 am to 8 am 5 days a week, in charge of 10 switchers stabled for maintenance - locomotives from an S1 class 2-10-2 to a U3 class 0-6-0.







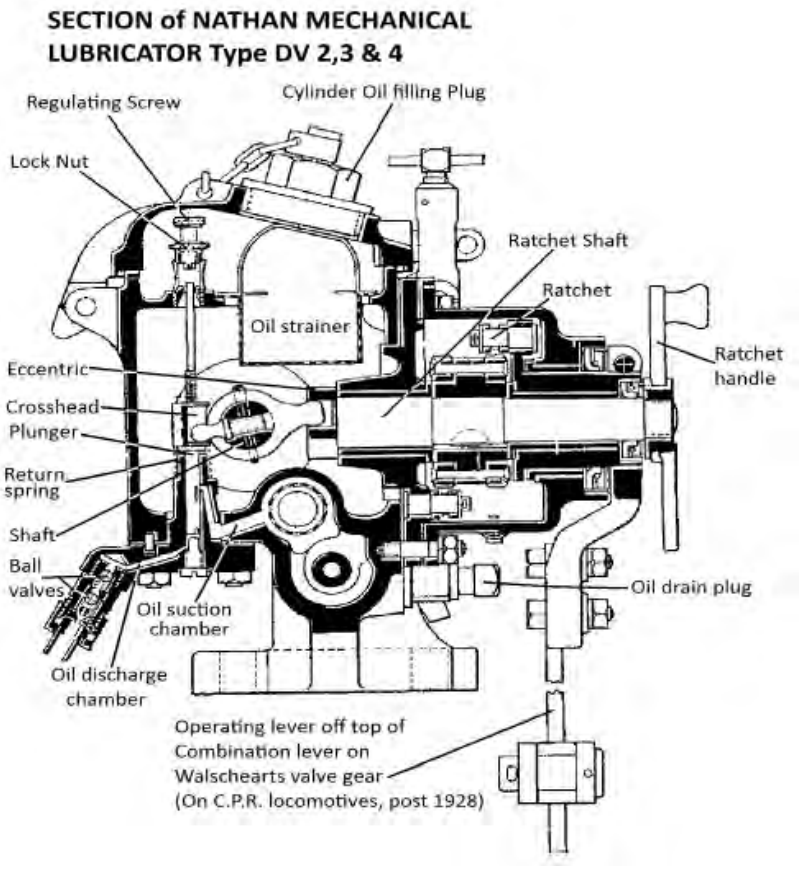
Cylinder lubrication was from a 'Detroit' hydrostatic lubricator in the cab with four pipes, two to each cylinder. One delivered lubricant through a choke to an atomiser 'spoon' (known as a 'spud) in each main steam pipe just above the steam chest and piston valves (similar to ex. LNER steam locomotives). The other delivered lubricant direct to the top of each cylinder. The oil pipes from the lubricator ran under the boiler cladding due to the minus 20 degrees F. in winter.

The following year there was a vacancy on the afternoon shift, 4 pm to 12 am, on road (main line) locomotives. The main job was inspection and testing locomotive from classes, T1c, H1c, G3g and G5d. All were fitted with 'Nathan' valveless mechanical lubricators, fitted just behind the steam chest above the combination lever.

The following paragraphs describe US and Canadian practice in detail.

From :- "Locomotive Cyclopedia" 1930 - the Nathan DV series Mechanical Lubricators description of working, page 753:

"The Nathan Type D.V. valveless mechanical lubricator used principally for oiling parts under pressure, is the result of many years experimentation, development and service tests. It is made in any number of feeds up to 14 and from 8 pints to 24 pints reservoir capacity. In road service, it reduces oil consumption on freight power by 50% and on passenger power by 25% and at the same time is doubling piston packing mileage.



The cross section shows the oil reservoir which is filled through the strainer. On the left is the valveless plunger, one for each feed. The number of plungers depends on the number of feeds required.

The pumps are actuated by the engine through a ratchet operating lever. Each pump consists of a plunger which, in its highest and lowest positions, automatically opens and closes the oil inlet opening and oil outlet opening through a turning movement of the plunger. The plunger then replaces the suction and discharge valves.

The eccentric at the end of the ratchet shaft contains a hole which receives the ball end of the rocker arm attached to the shaft.

The centre of the circular hole is eccentric to the centre of the ratchet shaft so that as the ratchet shaft rotates the vertical throw of the eccentric causes the shaft to oscillate while the horizontal throw of the eccentric causes the shaft to slide back and forth. These oscillating and reciprocating motions are defined since the arm is rigidly attached to the shaft.[1] These two motions are utilized to operate the plungers for pumping and for the control of admission and discharge of oil.

Each plunger is operated by a small arm rigidly attached to the shaft, the ball end of the arm moving in the elongated slot of the enlarged top of the plunger, to be termed the crosshead. The slot in the crosshead is long enough so that on the upward stroke of the plunger the oscillating motion of the small arm will leave the plunger free to be pushed upward by the spring below the crosshead. On the downward or pressure stroke, the ball end of the small arm contacts with the bottom of the slot pressing the plunger down, as the elongated slot is situated on the periphery of the crosshead.[2] The

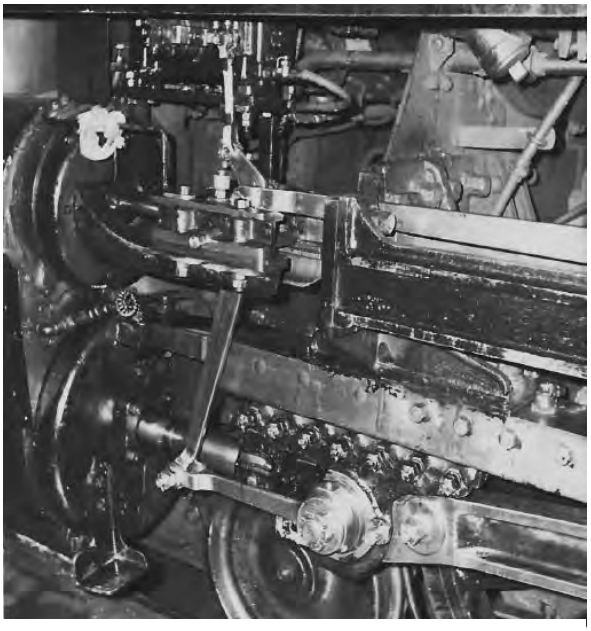
[1] The words are as written in the Nathan article of 1930. 'Defined' refers to the eccentric follower being taper-pinned to shaft. The mechanisms follow the action of the ratchet shaft and the eccentric. The action goes right to the plungers via the follower, shaft and plungers. Hence pumping action with no suction or discharge valves.

[2] See images on Page 26.

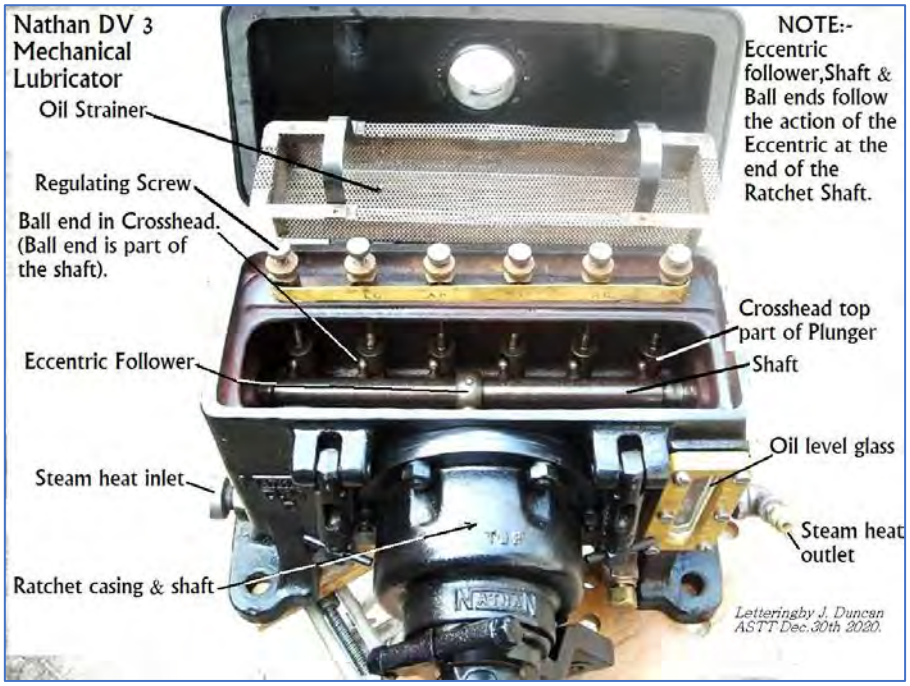
reciprocating motion of the shaft causes the plunger to oscillate and the turning motion thus produced causes a groove in the plunger to make connection to the suction chamber on the up stroke and with the discharge chamber on the down-stroke. The amount of upward motion of the plunger is limited by the contact of the regulating screw and the extension rod of the plunger, the action of which can be visually inspected. The amount of motion can be varied from full stroke to zero by the position of the regulating screw. The ratchet shaft is rotated by a ratchet arranged with three pawls and the ratchet is prevented from a return by a second ratchet also arranged with three pawls. The driving ratchet is moved by the arms attached to some point of the valve gear. The handle permits operating the pumps by hand for priming, testing and inspection.

The lubricator is provided with suitable strainers for refilling and a steam heating arrangement.

The photograph on the right was taken after filling the lubricator with cylinder oil, priming the pipes, test and set the pressure at the Terminal Regulator valves 30 to 40 psi above boiler pressure. The same procedure applied to the right hand cylinder.



CPR Class 3Gg 4-6-2 No 2385
back of left-hand cylinder



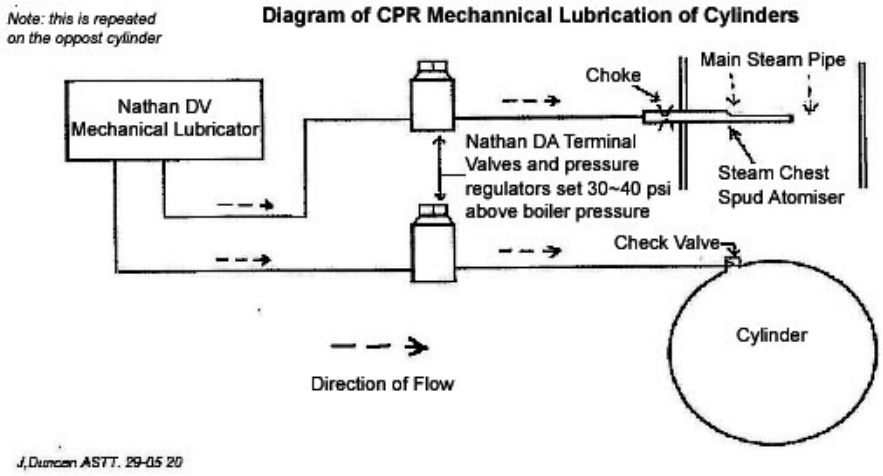
Internal View of a Nathan Lubricator showing parts described above



Calgary allocated. Photo Ernie Plant

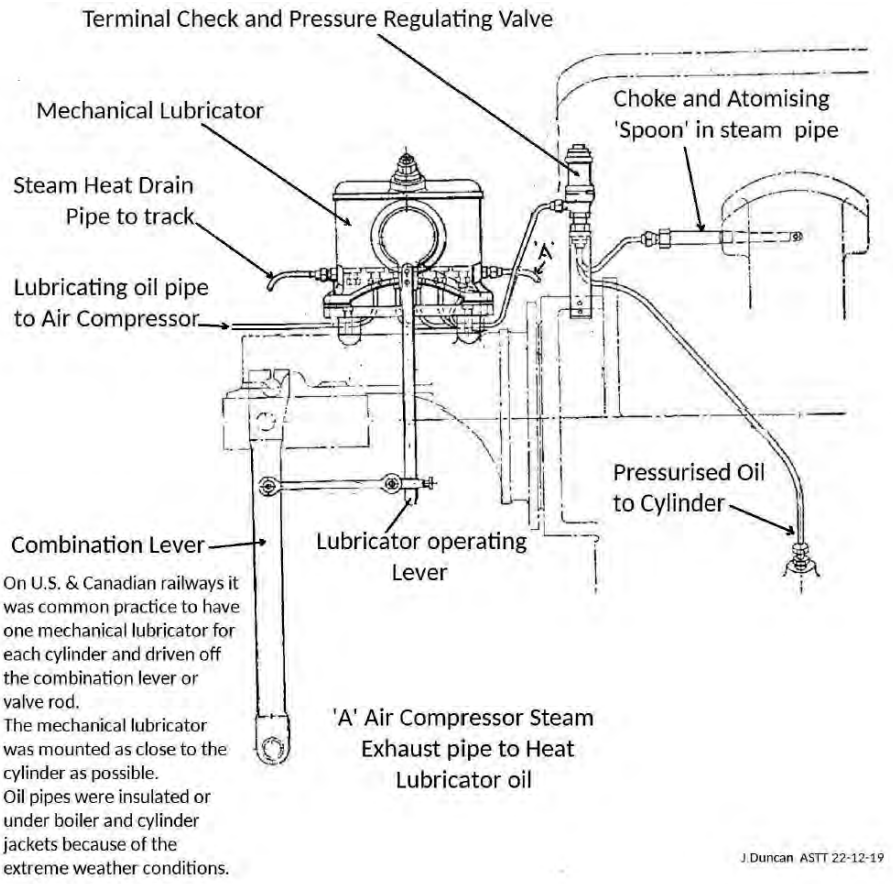
CPR, class G3g,4-6-2, Built 1941, Boiler pressure 275 psi, 2 Cylinders 22" X 30", Driving wheel diameter 6' 3", Tractive Effort 46,000 lbs at 85% of B.P.

One Nathan No. 4 mechanical lubricator for each cylinder and two terminal pressure regulating valves, one to the main steam pipe just above the steam chest through a choke and on to an atomising 'spoon'. The other supply is through the second terminal pressure regulating valve into the cylinder at 'ten to' or 'ten past' at the mid-point of the stroke (not atomised on the CPR). This compact layout was adopted because of the sub-zero conditions.

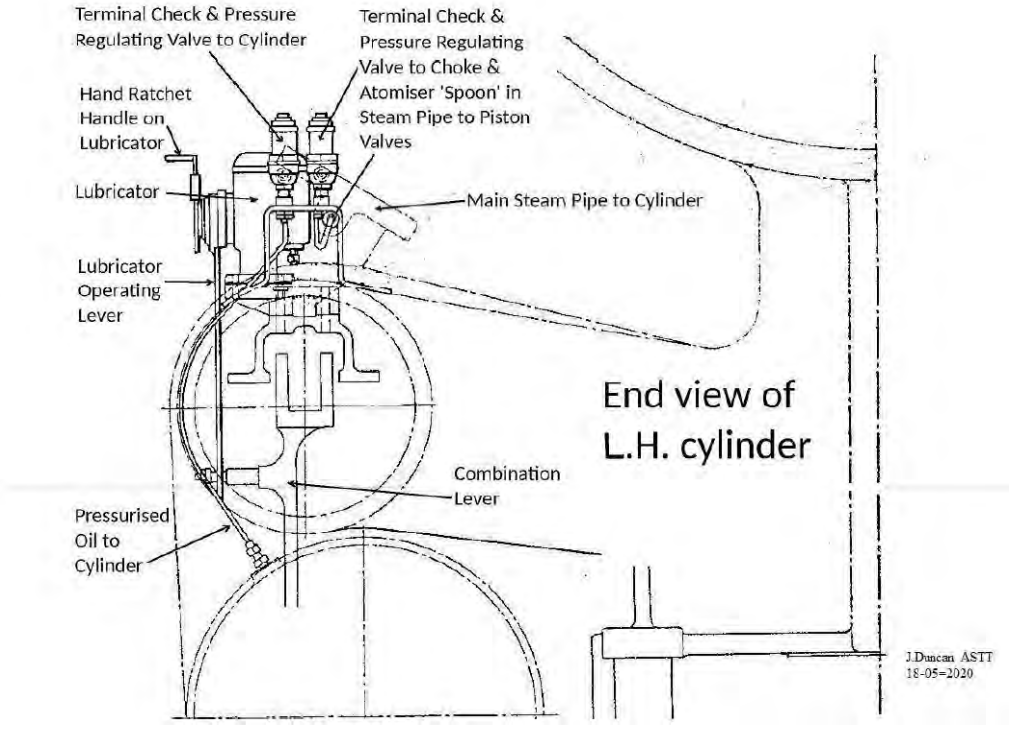


Schematic diagram of CPR mechanical lubrication of cylinders

Two drawings of the cylinder lubrication layout the CPR adopted from 1928 onwards for mechanical cylinder lubrication. This was similar to the previous hydrostatic layout with but with the mechanical lubricator replacing the hydrostatic lubricator and the addition of terminal check valves.



From "Locomotives Cyclpedia of American Practice" 9th Edition 1930: showing side view of mechanical cylinder lubricator driven off combination lever . (Feed to two terminal check-valves, one to the steam pipe 'spoon' atomiser and one direct to the piston.)



From "Locomotives Cyclpedia of American Practice" 9th Edition 1930: showing end view of mechanical lubricator and the two terminal check-valves, the combination lever and the lubricator drive. Note: the lubricator stroke reduces as the cut-off is reduced - an American practice on passenger locomotives. Note: no atomisation on feed to cylinders.

The CPR locomotives on the main lines built after 1928 all had the throttle in multiple valve form after the superheater with the operating rod running through the right hand rail along the side of the boiler.

The Valve and Cylinder was mechanically lubricated with the 'Nathan' system, shown above. Although the steam tightness was not so good with 'Kopper' segmental valve and piston rings, they soon bedded in and gave good wear rates for rings and liners. The rings could go 60,000 miles plus to the yearly statutory inspection of boilers, pistons, valves and motion.

The drive for the mechanical lubricators is taken from top or just below the valve rod on the combination lever at the forward end of the radius rod. The stroke is then varied with the 'cut- off' - more oil in full travel, less oil in short travel. (UK practice was to operate the lubricator from the pivot point on the expansion link.)

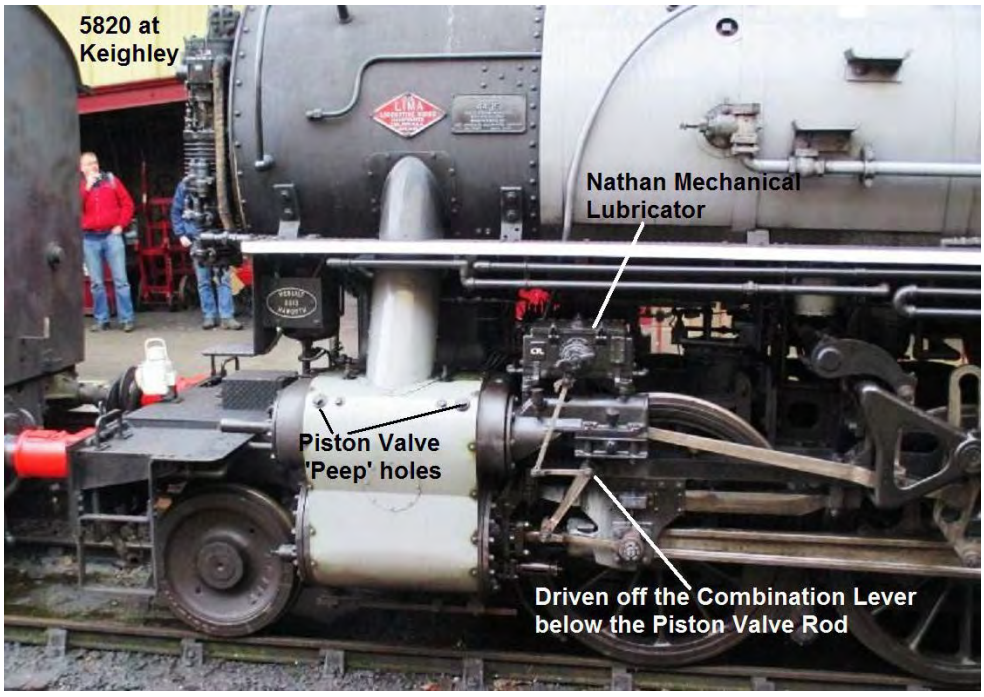
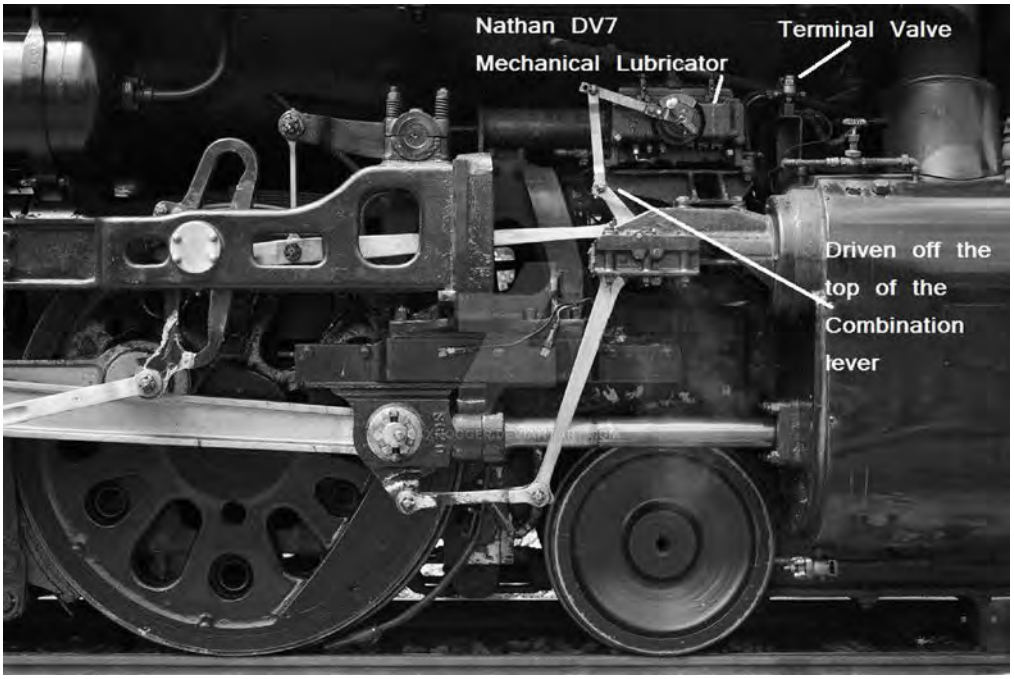


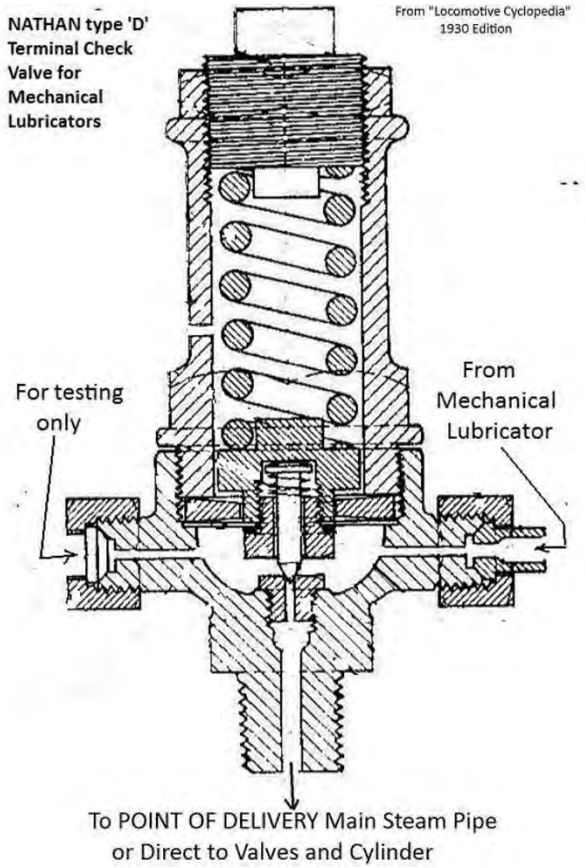
Illustration to show US practice for positioning lubricator and drive from the Piston Valve Rod

The position of the lubricator is important. Positioning it just behind the steam chest and above the valve rod gives the shortest distance between the pumping action and the point of delivery as necessitated by the below zero conditions in Canada and the USA. (Not as in the UK.)

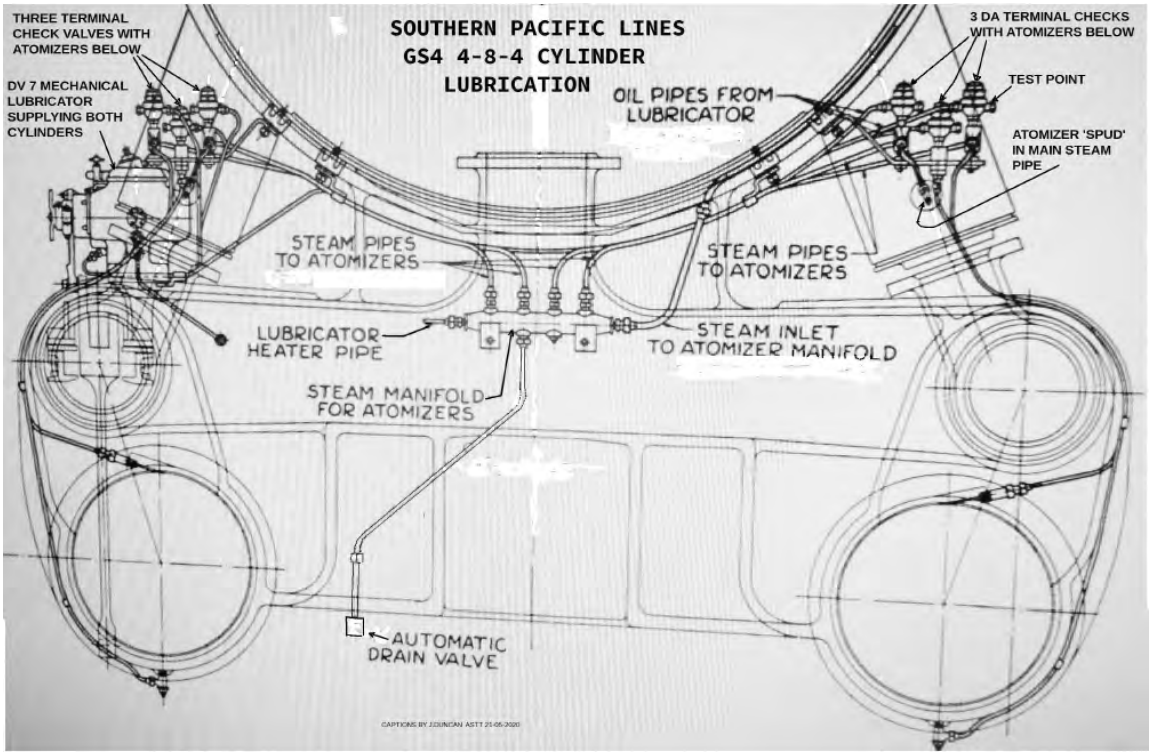
Terminal pressure regulating valves are a very important part between the mechanical lubricator and the point of delivery. They make certain that oil is there at the point of delivery at above boiler pressure.

Terminal pressure regulating valves made by Gresham and Craven were fitted to the early BR Class 7s 'Britannias'. They were set at 250 psi, but without a non return valve in the delivery lines, there was nothing to stop neat un atomised oil being ejected out of the chimney and spattering the front spectacle plates as witnessed by Bill Harvey, Shed Master at Norwich MPD. If it had been set at 30 to 40 psi above boiler pressure and a non-return valve fitted in the atomiser steam supply line, then the system would have worked.

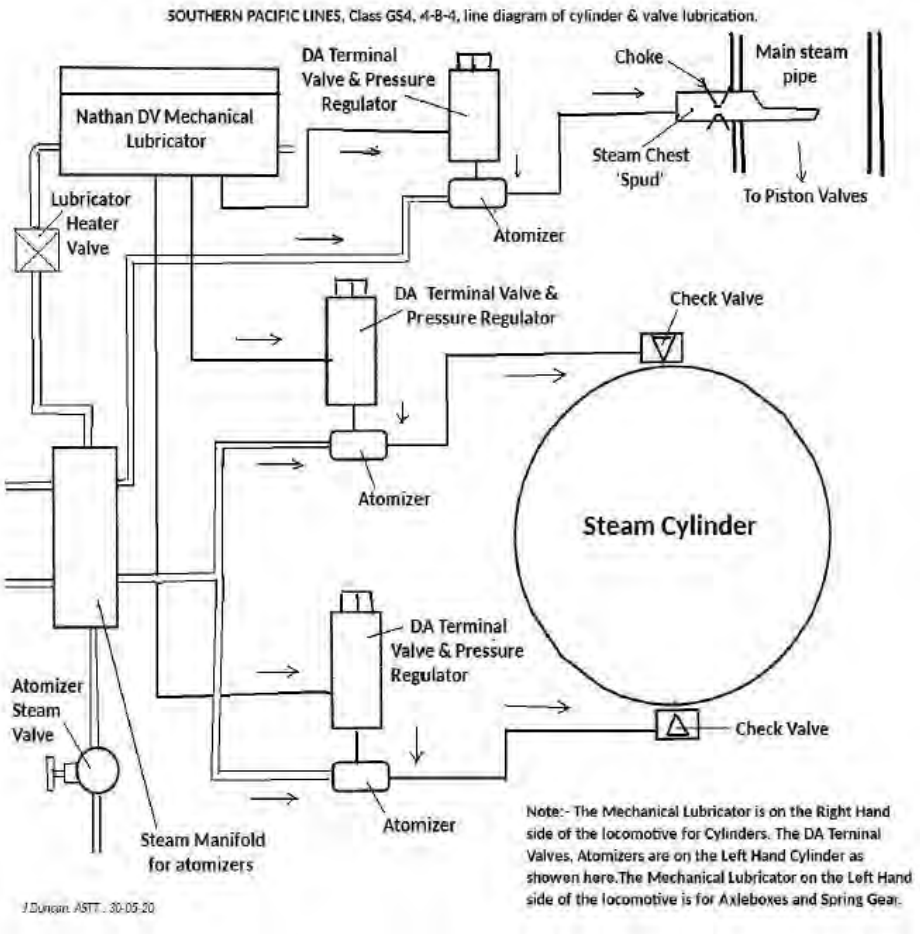

Nathan DV7
Mechanical Lubricator
Terminal Valve
Driven off the top of the Combination lever


NATHAN type 'D' Terminal Check Valve for Mechanical Lubricators
From "Locomotive Cyclopedia" 1930 Edition
For testing only
From Mechanical Lubricator
To POINT OF DELIVERY Main Steam Pipe or Direct to Valves and Cylinder

In the U.S.A. two railroads used atomisers : New York Central, J3 class, 4-6-4's and Southern Pacific Lines, GS4 class 4-8-4's, see diagram below:-

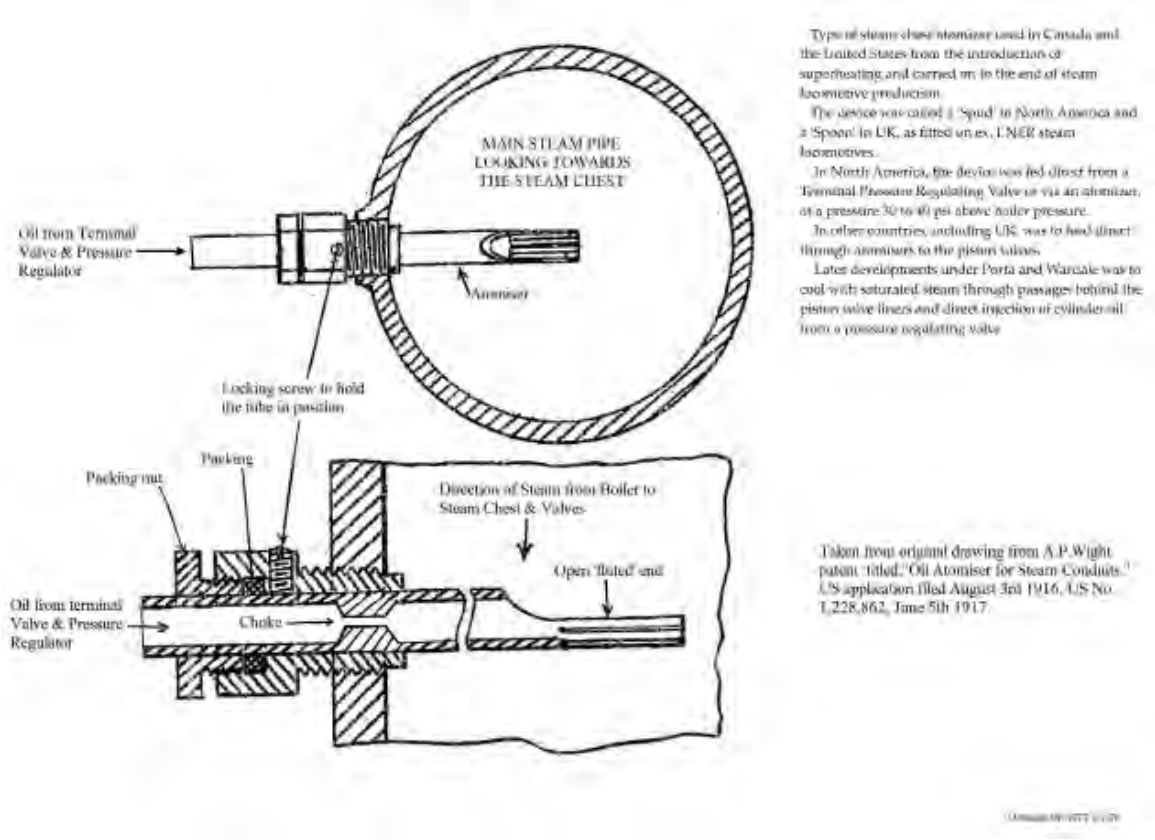


Whether fitted with atomisers or not, the layout with the diagram below looks to be the same across Canada and the United States.

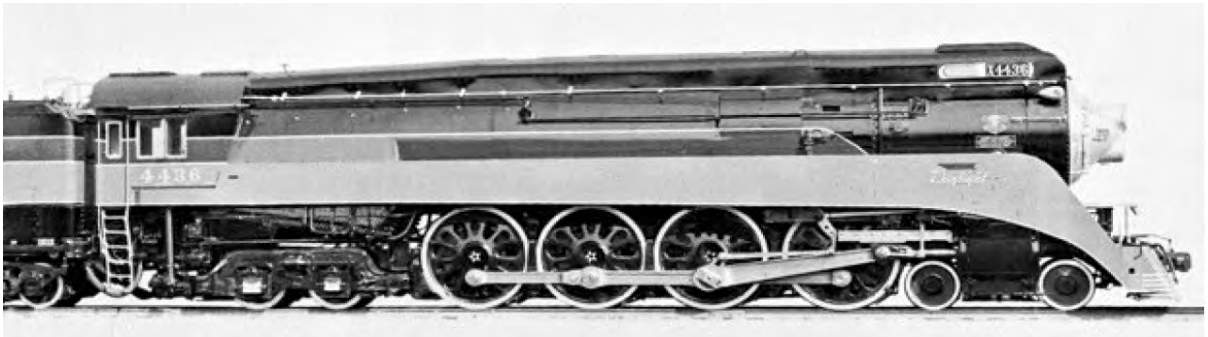


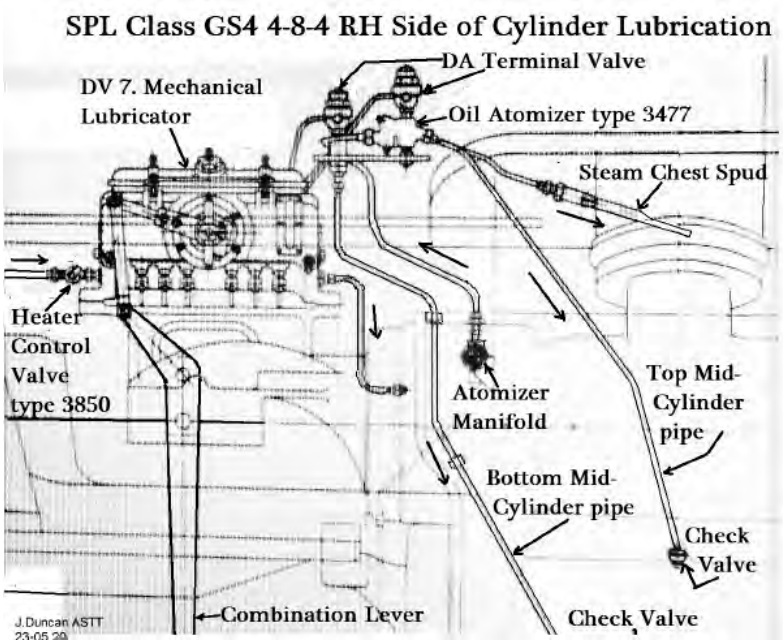
A mechanical lubricator close to each cylinder; if only one it is mounted by the RH cylinder feeding both cylinders. A terminal oil pressure regulator for each line is set at 30 to 40 psi above boiler pressure.

Oil is fed into the steam pipe just above the steam chest via a choke valve and atomizer 'spud', a U.S. patent. Sometimes cylinders have one feed at the top on the mid-point. Sometimes an additional feed point is added below the cylinder. All cylinder feeds point have check valves as per the drawing below. This shows a plan section of the main steam pipe with a 'spud' atomizer looking down into the steam chest below. The lower drawing shows the choke and 'spud' atomizer in section. Each CPR yearly inspection, the 'spud' atomizer was always removed for cleaning any build-up of carbon and refitted with new packing. The 'Kopper' piston rings could last over 60,000 miles. Because the system worked, it was carried on even when boiler pressures rose to 300 psi.

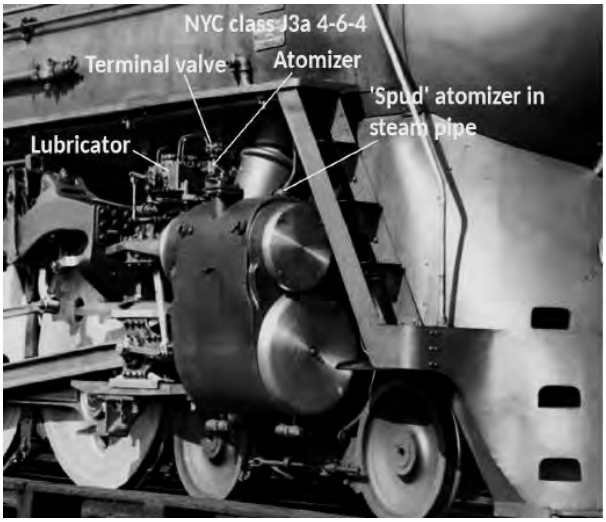


The drawing overleaf is of the piston valve and piston mechanical lubrication as on Southern Pacific, GS4 class 4-8-4 (pictured below).

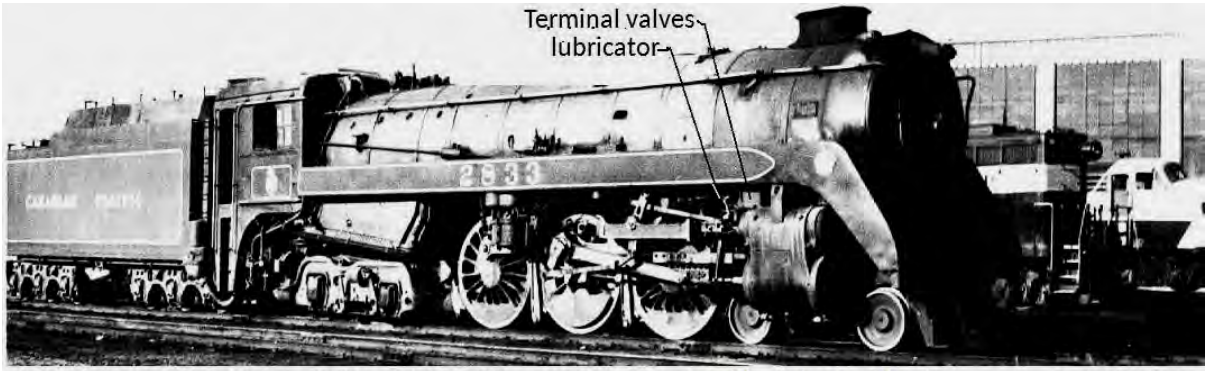




The New York Central class J3 4-6-4 had a similar cylinder lubrication system and other railroads. The photo at right is of the RH front side of one of New York Central's Henry Dryfuss streamlined class J3a, 4-6-4 locomotive showing the close proximity of the lubricator, terminal valve, atomizer, steam pipe and cylinder. Since their introduction in June 1938, the 13 car "20th Century Limited" did the 925 mile run from Harmon, New York to Chicago in 16 hours, clocking up to 130,000 miles per year. Piston and valve ring change at 65,000 miles per year. 'Kopper' segmental piston rings are still manufactured in the U.S.A.



I remember fitting 'Kopper' segmental' cast iron and bronze piston rings to CPR 2833 (Royal Hudson 4-6-4) at Aylth Roundhouse, in 1957 – photo below:



CPR, 2833 Class H1c 4-6-4,off the night mail from Edmonton at Aylth, Calgary, Alberta photo. John Duncan

I'd never seen them before so the other machinists helped. They named them the 'Chinese puzzle'. They came fastened to a plywood board. A special split steel band with a slot in the top to fit the under-springs then the springs that fit through the segments, then the ring segments the back first. They were fitted to the larger main-line locomotives. Once bedded-in, I found them steam tight on testing after washouts with cocks open against the brake, in front or back position of the crosshead. They lasted more than 60,000 miles. The piston valves were similar to our GWR piston valves.

ANDRE CHAPELON quoted in his book:

"For several years the Americans have composite piston rings (Fig. 52 below) in place of the standard cast iron type. Each ring group generally comprises two or more crowns. Each made up of several pieces, joined together, which are pressed into contact with the cylinder walls by a spring steel ring. There are two rows of elements placed side by side to each group, with overlapping joints, one being in special cast iron and the other in special bronze and both are of great hardness.

The successive contact of these two different metals against the cylinder walls gives them a beautiful polish with minimal wear. Contrary to what was thought initially, it is immaterial whether the bronze or the cast iron segments are located at the inner or outer position at the piston head.

Nevertheless bronze segments housed in the outer position produce less wear in the cylinder bore to the end of the stroke were only the bronze ring head is in frictional contact and for this reason it may be preferred to place the bronze ring in the outer position, but this is not for reasons of steam tightness."

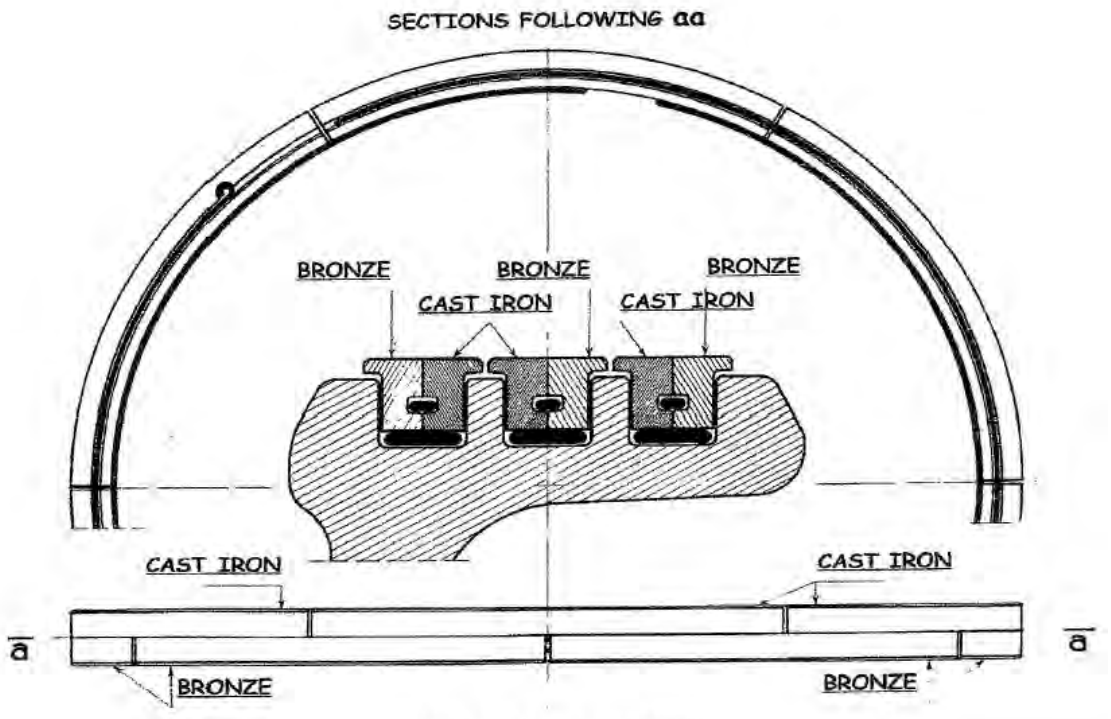
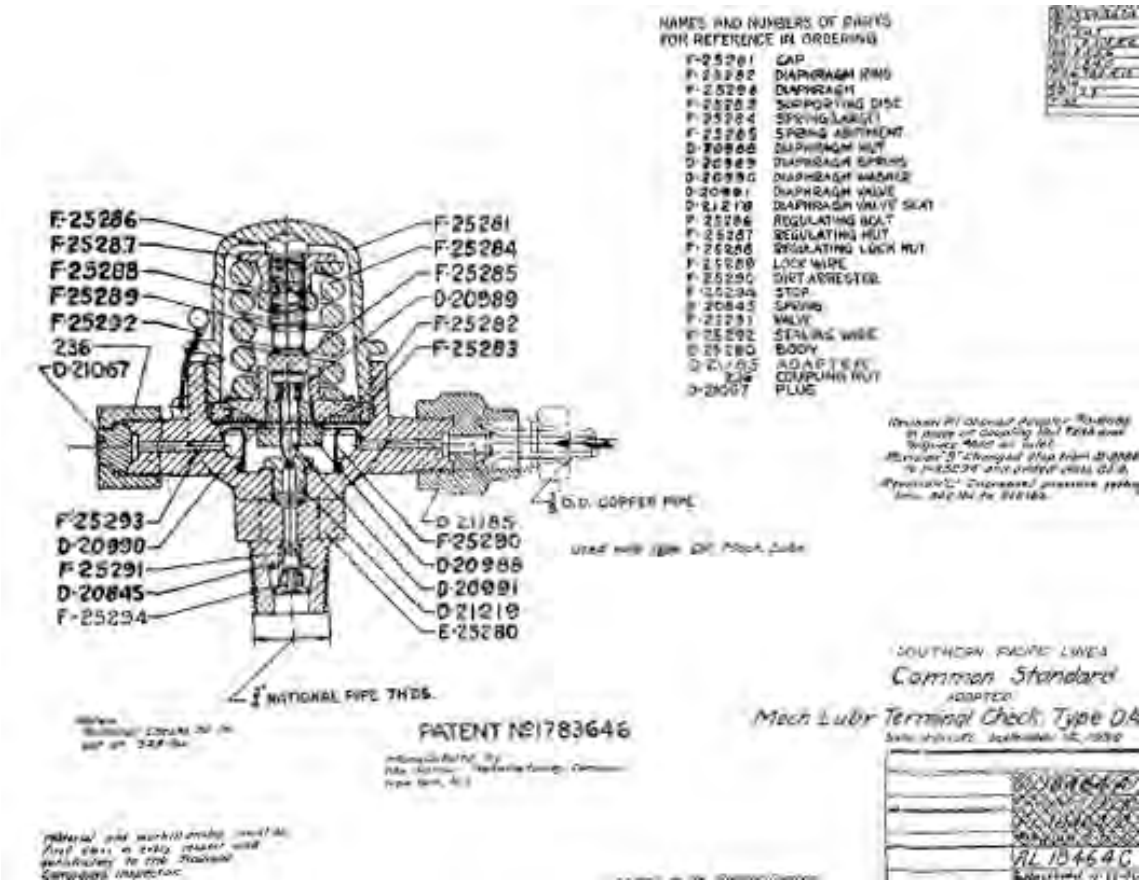


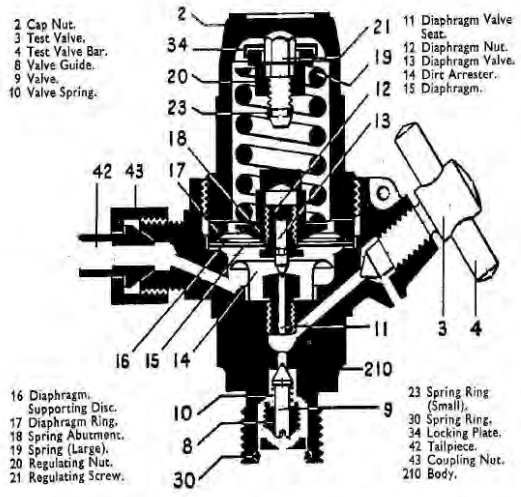
FIG. 52. – KOPPERS PISTON RINGS

Terminal Valves & Atomizers. In Canada & USA, with the introduction of higher steam pressures in the late 20's and 30's, Nathans altered their terminal valves and atomizers, as did 'Olva', which were used with the FSA (Friedmann Schlote Austria) valveless mechanical lubricator.

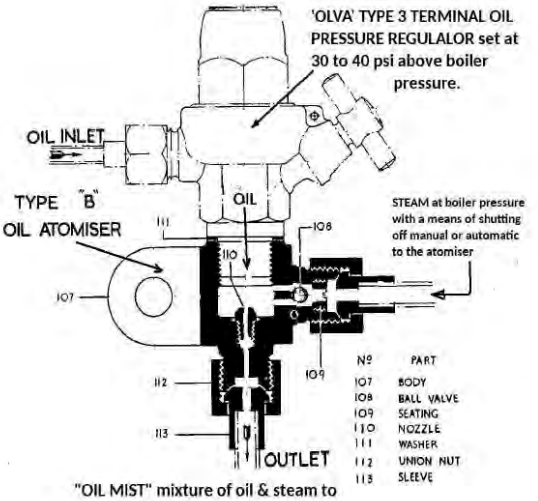
Below: Nathan Terminal Check Valve Type DA and set up on a test rig and sealed. Note the heavier spring. Drawing issued by Southern Pacific Lines on the 12th September 1939 specifying that the oil pressure on the check valves to the steam chests and cylinders be set 320 psi on the Class GS4 with boiler pressure of 300 psi.



Below: the 'Olva' Type 3 Terminal Check Valve. The test valve (3) is removed and test gauge and adapter fitted when setting the pressure. The Oil Atomizer, Type 'B' is on the right.



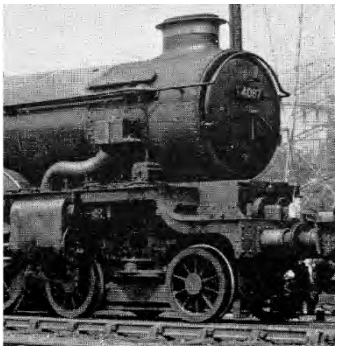
"Olva" Type '3' Terminal Check and Pressure Regulating Valve D & M



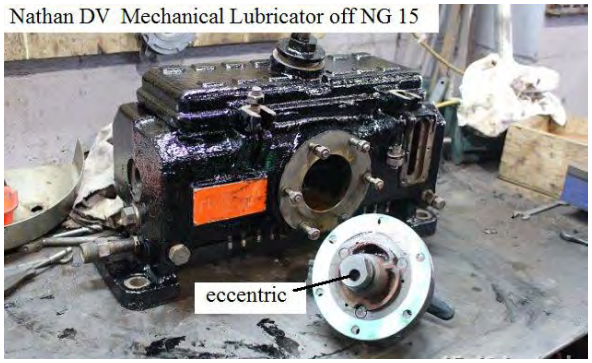
"OIL MIST" mixture of oil & steam to cylinder or steam chest.

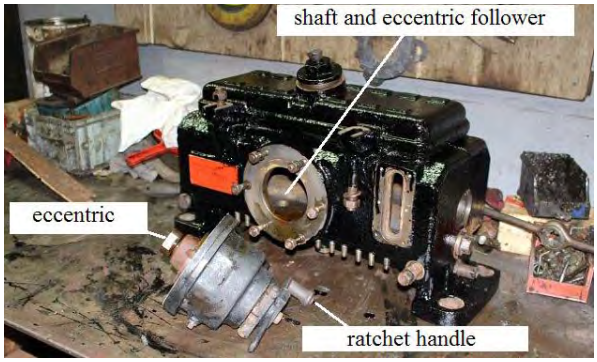
Both the above were manufactured by Davies and Metcalfe Ltd., Romiley, Cheshire. They also manufactured the D.V sliding shaft valveless mechanical lubricator, DA terminal check valves and Type 3477 atomizers, all under license from Alex. Friedmann KG of Vienna, Austria.

The Nathan Manufacturing Co. of New York had license rights for DV mechanical lubricators and accessories.



In 1955, Davies and Metcalfe supplied lubrication systems for a number of 'Castle' class engines on the BR (Western Region) when they were fitted with higher superheat boilers and double chimneys. They supplied a FSA valveless mechanical lubricator with an oil reservoir above, manufactured under licence by Davies & Metcalfe. Three feeds to each cylinder, one to the mid-point of the cylinder one to each piston valve head. 12 GW/LMS type atomisers in all.





Above and below: A Nathan type valveless mechanical lubricator being overhauled at Dinas on the Welsh Highland Railway off an SAR class NG 15, 2-8-2 (below). It may have been made at Friemanns KG in Vienna or Davies & Metcalfe.



SOME COMMENTS ON LOCOMOTIVE FRONT END DESIGN

By M. Johnson

1 INTRODUCTION

My career was in fluid dynamics and it's many applications over some 35 years. With that background, I find the treatment of locomotive front end design to be "insular", using only data that was produced within the railway industry. In this article, I hope to show that a sound understanding of basic fluid flow concepts, taken from all areas of industry, can improve the current state of the art, and more importantly our understanding of the issues involved.

2 KOOPMANS' METHOD OF ANALYSING MULTIPLE ORIFICE SYSTEMS

In his lecture to the ASTT delivered at the ASTT Autumn conference in 2019 (Ref. 7.1), Dr. Koopmans put forward his theory on why multiple exhausts perform better than single exhausts. In brief, Dr. Koopmans' proposal for analysing multi orifice exhaust systems is to apply the normal momentum equations and experimental diffuser loss coefficients BUT the diffuser L/D ratio should be uplifted by √Number of orifices. This section will show the method is fundamentally flawed.

2.1 *A lengthened single chimney*

In Slide 14 Dr. Koopmans quotes data from Young to support his assertion that multiple orifice systems can be modelled by considering them to be a multiple versions of a scale model exhaust.

The results quoted in Slide 14 refer to tests on a chimney of constant taper. When the chimney is lengthened, the area ratio (inlet to outlet) of the diffuser changes in proportion because the taper angle remains constant. So according to data presented in Slide 16, there would be a significant improvement in C_p as shown in Table 1 below. For the table, I have had to use data from Figure 1 (Ref. 7.10 , See Section 3 below), as the range of included angle and L/D1 are not included on Slide 16. Virtually all of the improvement in C_p is due to the decreased exit velocity which decreases the kinetic energy loss.

Diffuser Details	Young Fig. 17, Slide 14 Basic Chimney	Young Fig. 17, Slide 14 Basic Chimney + 13" extension
Throat diameter, D_1	4.375"	4.375"
Chimney height, L	12.5"	25.5"
Exit diameter, D_2	4.875"	5.395"
Included angle	2.29 Deg.	2.29 Deg.
L/D_1	2.85	5.83
Area Ratio D_2^2/ D_1^2	1.24	1.52
C_{pi} (Equation on Slide 10)C_{pm} (From Ref. 7.10 , Fig 11.3)	0.35	0.56
C_{pm} (From Ref. 7.10 , Fig 11.3)	0.31	0.48

Table 1: Diffuser Performance for Young's tests.

2.2 *Double Chimney*

A similar situation exists for a typical double chimney configuration. Table 2 presents data for a double chimney as proposed by Ell. In calculating the table, I have assumed dimensions for a GWR King class (single chimney). For calculating the twin chimney diameter, I have divided the single chimney diameter by √2. (Ell's formula takes account of improved efficiency and calculates a larger blast orifice and chimney, which would not then be a fair comparison). I have used Ref. 7.10 again to derive values of Cpm as they also fall beyond the range of Koopmans' Slide 16.

Diffuser Details	S.O. Ell Basic Chimney (60xx King class)	Double Chimney
Throat diameter, D_1	15.25"	10.78"
Chimney height, L	29"	29"
Exit diameter, D_2	17.25"	12.78"
Included angle	3.95 Deg.	3.95 Deg.
L/D_1	1.9	2.7
Area Ratio D_2^2/ D_1^2	1.28	1.41
C_{pi} (Equation on Slide 10)	0.39	0.49
C_{pm} (From Ref. 7.10 , Fig 11.3)	0.35	0.4

Table 2: Diffuser Performance for Ell's recommendations for single and double chimneys.

Table 2 above shows there is a substantial improvement in C_{pm} , despite the very modest taper angle of the GWR chimney. As before, the improvement is due to reduced exit velocity energy loss.

Therefore, in the case of a lengthened single chimney and a typical double chimney arrangement, a significant improvement in C_{pm} would support Koopmans' hypothesis that this is the cause of improved performance.

2.3 *Multiple Orifice Single Chimney*

If we take the example of four blast nozzles discharging into a given chimney, the L/D ratio doubles (according to Dr. Koopmans) but the the Area Ratio for each quarter chimney must remain the same as for the single chimney. That being so, the taper angle must reduce by $1/\sqrt{n}$, where n is the number of orifices. Dr. Koopmans' Slides 23 and 15 imply that the taper angle remains constant.

The method stated in Slide 23 might be an error, which may have crept in since Dr. Koopmans used a different correlation of C_{pm} against Area Ratio and Non dimensional length in his SLS paper (Ref. 7.8), and for that presentation Slide 23 would be correct(but the method as I show would yield no improvement in C_{pm}. It is not stated in either the SLS or ASTT papers precisely what assumptions were made in his multi-orifice calculations to support his correlations of test and predicted results.

Table 3 shows my assumed dimensions for a GWR 60xx (King class) and calculations for a single chimney and it's conversion to a 4 orifice blast nozzle. It will be seen that there is no change in Cpm.

Diffuser Details	S.O. Ell Basic Chimney (60xx King class)	Example chimney 4 orifice blast As slide 23
Throat diameter, D_1	15.25"	7.63
Chimney height, L	29"	29
Exit diameter, D_2	17.25"	8.63
Included angle	3.95 Deg.	2 Deg.
L/D_1	1.9	3.8
Area Ratio D_2^2/ D_1^2	1.28	1.28
C_{pi} (Equation on Slide 10)	0.39	0.39
C_{pm} (From Ref. 7.10 , Fig 11.3)	0.35	0.35

Table 3: Diffuser Performance for an identical chimney with single and quadruple blast orifices.

Referring to Figure 1 reveals the reason. In the lower half of the graph, lines of constant K_d and hence Cpm are virtually flat. Hence, unless we change the area ratio, Cpm remains constant (If the original chimney were designed above the "optimum design line" as shown in Figure 1, there would, according to Dr. Koopmans' hypothesis, be some cases in which an improvement would be shown. The reason this would not work will be explained in Section 3 .)

However, experience suggests that changing to multiple orifices without changing the chimney - as illustrated by Dr. Koopmans' changes on GWR King 6023 and Netherlands SHM5 - DOES improve performance, so there has to be another explanation. Experience also shows that multiple orifices improve the performance of a parallel chimney, but Dr. Koopmans' theory quite obviously does not address that problem.

3 **DIFFUSERS**

Having put some doubt on Dr. Koopmans' approach to the theory of diffusion, this seems an appropriate place to clarify a few points about diffusers.

Dr Koopmans limits his discussion of diffuser design to one slide (Slide 10) quoting data from Fox & McDonald's textbook in Ref. 7.1 , while referring to data from Cockerill quoted in Ref. 7.8 . Both sources are quite old and more up to date information is available in Ref. 7.10 , which includes the design chart shown in Figure 1. All of the sources present a similar pattern of data, although the detail may change slightly. Note that Figure 1 gives values of K_d, which is a coefficient of loss from the diffuser, whereas other sources give values of C_{pm}. To convert, $C_{pm} = 1-K_d$. Figure 1 also defines non dimensional length in terms of Length / **RADIUS**, not Length / Diameter.

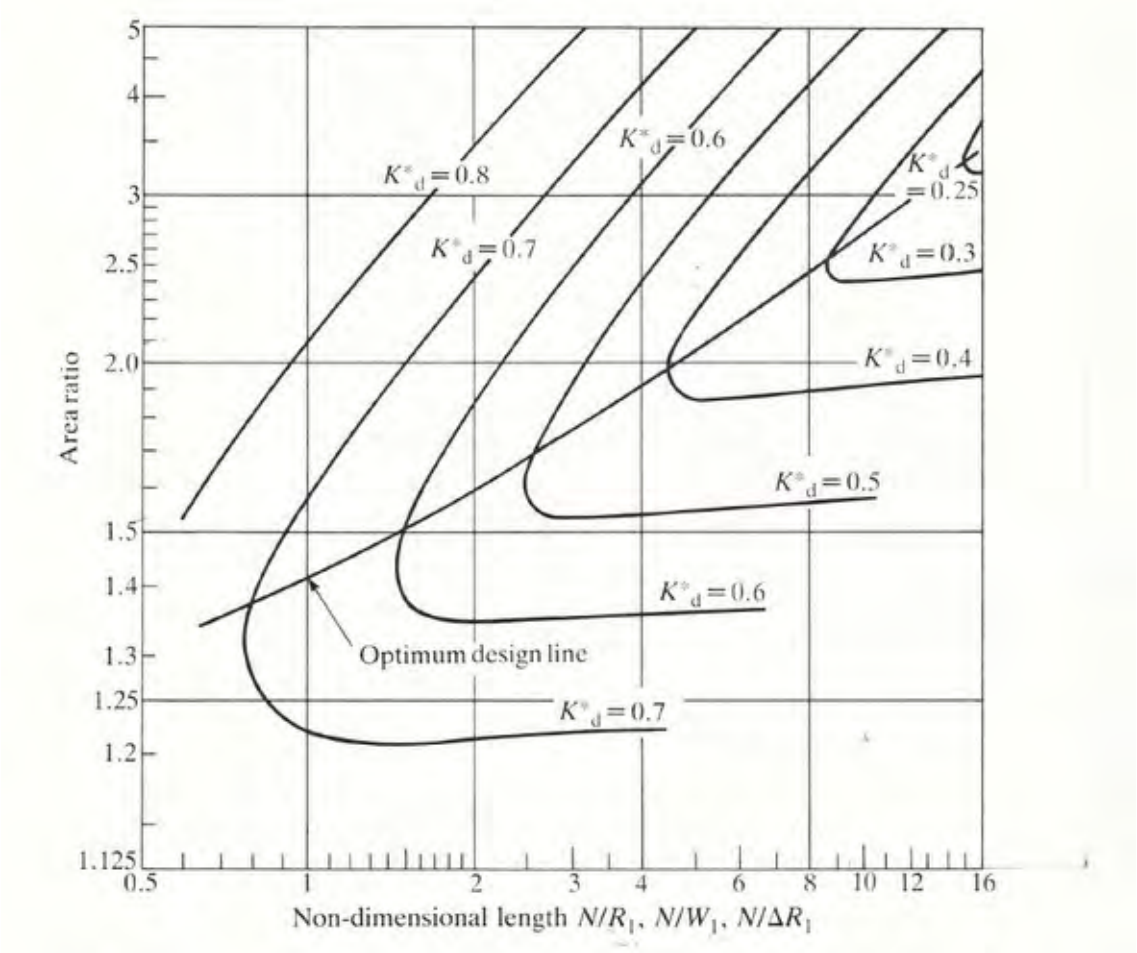


Figure 1: Miller's design chart for diffuser losses with "developed" inlet flow.

Figure 1 shows two separate regimes separated by an "Optimum design line" and for designs below that line, the flow stays attached to the diffuser walls and losses are virtually independent of length, depending only on Area Ratio (AR). That is why Table 3 shows no improvement in Cpm between a single and four nozzle blast pipe and chimney.

However, flow in a diffuser is inherently unstable As an example referring to Figure 1, for an AR (diffuser inlet to outlet) of 2.0, the loss would be approximately (0.39 x inlet kinetic energy) for all one-dimensional lengths (NDL) from 16 down to 5. If the NDL is reduced below 5, however, the Kd value rapidly increases up to nearly 0.8 at an NDL of 1. The sharp change in behaviour below a certain NDL and hence wider taper angle is due to flow separation from the diffuser walls. This is exactly analogous to an aerofoil stalling at high angles of incidence. Therefore, a widely tapering chimney is unlikely to convert much velocity energy back into pressure energy and will be counter productive. This is illustrated well in a CFD study produced by Dr. Koopmans' in support of discussion following one of his his papers and given in Ref. 7.11 , part of which is reproduced below in Figure 2. I also show my sketch of the same data as flow vectors in Figure 3.

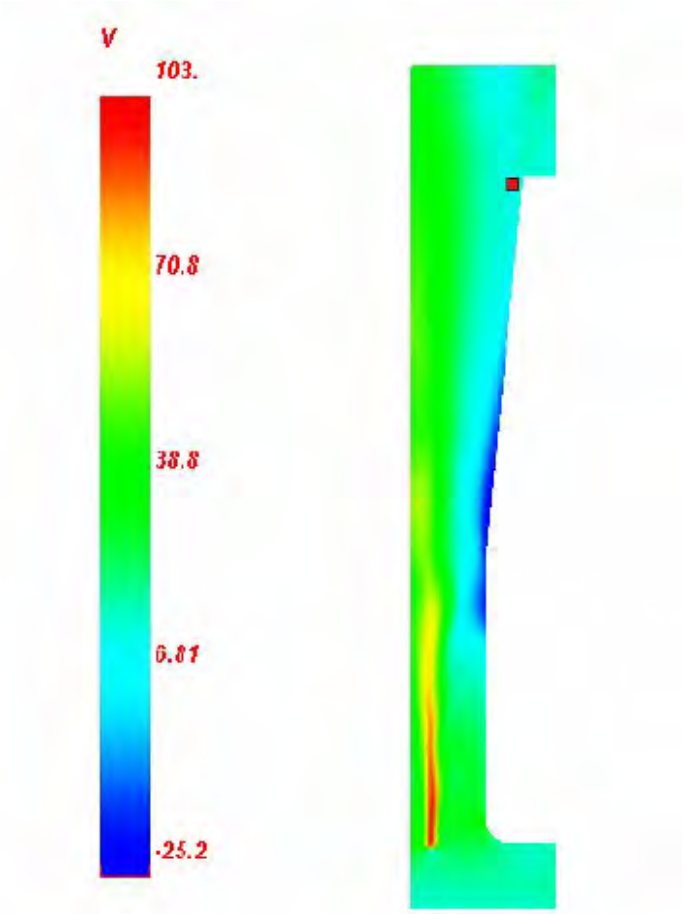


Figure 2: Velocity plot from a CFD simulation of a Lempor exhaust given by Koopmans.

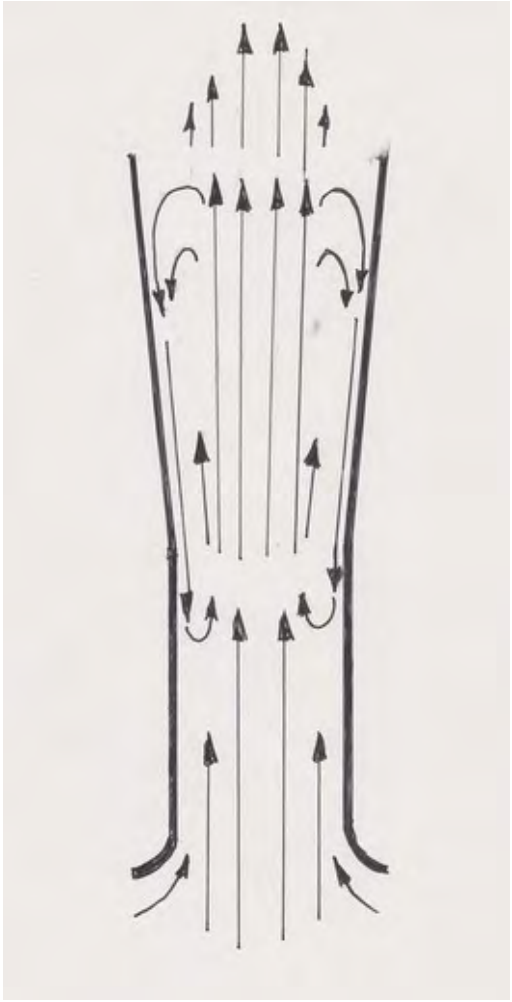


Figure 3: Flow pattern interpretation of figure opposite.

The plot shows velocities in a Lempor style exhaust with a mixing chamber of about 2.3 diameters in length and a diffuser of 2.4 diameters in length with an included angle of about 11 degrees, which implies a diffuser AR of 2.2 and a N/R1 of 4.8. This is significantly above the "optimum design line" in Figure 1. Three things should be noted from the velocity plot:

- Flow after the throat entering the diffuser section is highly disturbed, with velocity of around 50 m/s at the centre with a significant region at around 25 m/s and some reverse flow persisting at the circumference.
- There is serious flow separation at transition from the throat to the diffuser as shown by the dark blue region on the right hand side. Reverse flow (DOWN the chimney) of up to 25 m/s is indicated.
- There is a "core flow" detached from the diffuser walls extending the length of the diffuser, shown by the core of mid green (approx 38 m/s) surrounded by a slower region (magenta blue) of approx. 15 m/s - a very non-uniform outlet profile.

This shows that a diffuser design above the optimum design line will give poor performance as the fluid is not being decelerated, resulting in excess kinetic energy being ejected. This is discussed more fully in Section 4.1 below. Attempting to rectify a chimney with a diffuser above the optimum line by applying multiple blast orifices will not work, because it does not (and cannot) address the fundamental instability problem **unless we put physical walls in the chimney to split the flow into separate streams.** Extra internal walls are an established way of improving diffuser performance in industry, but do not seem to have been applied to the steam locomotive.

In my view the included angles for taper on diffusers advocated by Porta (less than or equal to 12 degrees) and Wardale (10 degrees) are too large and will lead to less than optimum performance. Significantly, Loubser and Cox (Ref. 7.12 , Page 398) state "Theoretically 1 in 6 (10.5 degrees included angle) would be the best value, but this or values near it give an arrangement which is very sensitive to any error in the relative position of cap and chimney". One might add that any irregularity in the incoming flue gas flow will have the same effect.

Therefore, loco front ends should be designed in the region to the right of the "optimum design line". It is not known precisely how conservative a design needs to be - that requires an experimental or CFD analysis programme to study the detailed diffuser performance with typical blast pipe geometries supplying the inlet flow. That is in stark contrast to the usual "Put it on a loco and see what happens..." approach, which gives very little useful information for development.

3.1 *Diffuser Inlet Conditions*

Diffuser performance is affected by the inlet conditions, which as Figure 2 has illustrated will be highly disturbed in a locomotive chimney application.

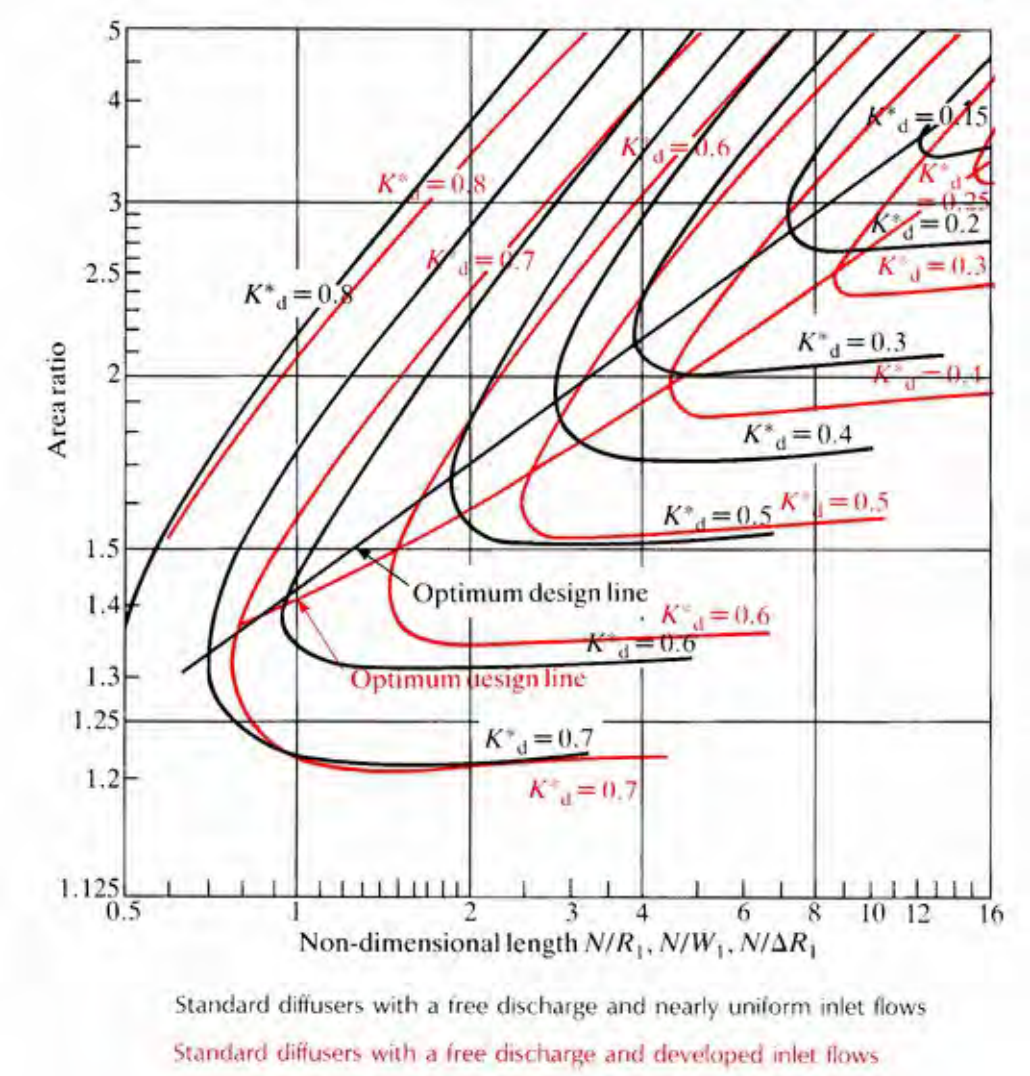


Figure 4: Comparison of diffuser performance for uniform inlet flows (black) and moderately disturbed inlet conditions (red).

Figure 4 shows the performance plot for good inlet flow (black) and moderately disturbed inlet flow (red). For a given combination of AR and N/R, the energy losses increase with disturbed inlet flow. In addition, the "optimum line" drops to lower area ratios at a given length; that is the

taper must be less for effective performance - supporting Cox and Loubser's observations of instability.

Dr Koopmans claims in Slide 21 of Ref. 7.1 that "Exit momentum defines all poor behaviour", following his calculation of diffuser performance. That would be so if the experimentally determined diffuser performance charts were applicable to the highly disturbed inlet profiles associated with a blast pipe; unfortunately the data quoted by Dr. Koopmans in Refs. 7.1 and 7.8 is for 'nearly uniform' inlet flows.

Figure 1, is drawn for "developed inlet flow" which Miller defines as 5 to 10 x pipe diameters of straight upstream pipe after a significant disturbance. Miller goes on to say "Very poor performance is to be expected from diffusers of optimum geometry .....when low energy fluid is concentrated .....at the wall.", which is precisely what happens in a loco front end as low energy gases are entrained by a high energy steam jet. This explains why Monsieur LeMaitre advocated directing the steam around the circumference of the chimney throat. It also provides a more cogent explanation of Nigel Day's practice of directing the blast nozzles outwards by 7.5 degrees, aiming them at the transition from the mixing section to the diffuser, thus injecting high energy fluid into the area of reverse flow identified in Figure 2, in order to suppress the reverse flow.

A significant improvement in diffuser inlet conditions can be achieved in a relatively short space; Miller suggests using 0.5 to 2 diameters of parallel pipe to settle inlet flow, advising this will give better performance than a longer slower taper over the same axial distance. Porta (Ref. 7.5) suggests 2.5 diameters of parallel pipe, while Wardale advocates 2 diameters.

3.2 *Diffuser Size*

Dr Koopmans notes that diffuser data is presented in terms of dimensionless parameters and wonders if absolute size is taken out of the analysis (Slides 10 & 13). The fourth dimensional grouping that he quotes in Slide 13 is Reynolds number, which includes a linear dimension of the system under study. The effect of Reynolds number changes will be small or negligible for large Reynolds numbers (typically > 10^5 for pipe systems) but becomes increasingly important for Reynolds numbers below 10^4. Thus model and miniature engines would be expected to have lower performance than full size main line engines due partly to linear scale and also to lower piston speeds and hence lower steam velocities which both serve to reduce Reynolds number. Curves for a diffuser performance correction factor based on diffuser inlet Reynolds number are given in Ref. 7.10 .

3.3 *Excess Diffusion*

The more that the exhaust flow can be slowed down, the less kinetic energy is wasted and the less work will need to be done by the blast nozzle (and hence exhaust steam), thus taking load off the cylinders. Given sufficient room, a diffuser could be designed to massively slow the exhaust gases.

However, taking diffusion to excess will not leave enough kinetic energy to throw the exhaust clear of the locomotive. I remarked in Section 2.2 on the low taper angle and area ratio for the GWR King class - a class which as far as I know never needed smoke deflectors. So there is a design balance to be struck - reduce blast nozzle back pressure OR eliminate the need for drag inducing smoke deflectors.

4 TOWARD AN IMPROVED FRONT END THEORY

4.1 *Kinetic Energy Correction Factor*

Throughout their theoretical derivation and subsequent numerical analysis of test results Dr. Koopmans, Zeuner, Strahl, Porta and others implicitly assume the velocity at various planes

through the chimney is completely uniform. This is not so, the disturbed Gaussian velocity distribution at entry to the chimney throat will be carried forward and will persist (but attenuated to some degree) at the chimney exit. Young's pitot test results (Ref. 7.9 , Page 135-136) clearly show this effect, as does Figure 2 above.

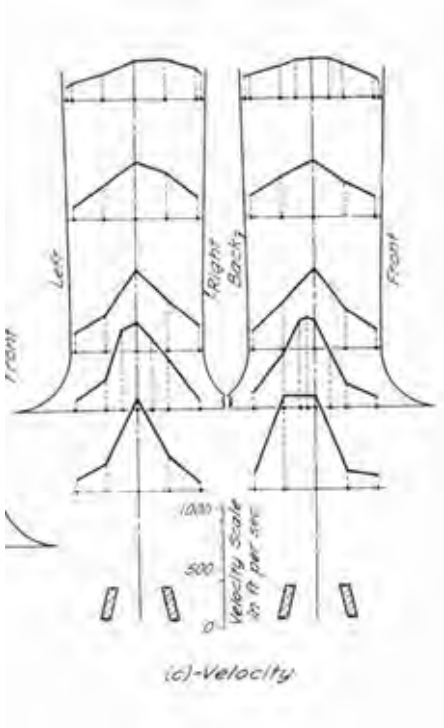


Figure 5: Results from Young showing development of velocity profile through length of chimney

The kinetic energy of fluid passing through a plane is **NOT** the product of total mass flow and mean velocity squared, as Koopmans, Porta and others calculate it. (A similar argument applies to momentum, but I shall deal with kinetic energy as that will be more relevant later in this series of articles.)

The kinetic energy is actually the **integral** of (mass flow x velocity^2) for each element across the plane. The Kinetic Energy Correction Factor (KECF) is defined as the integral of kinetic energy across a plane divided by the total mass flow x average velocity^2.

As an example, consider the following velocity profiles:

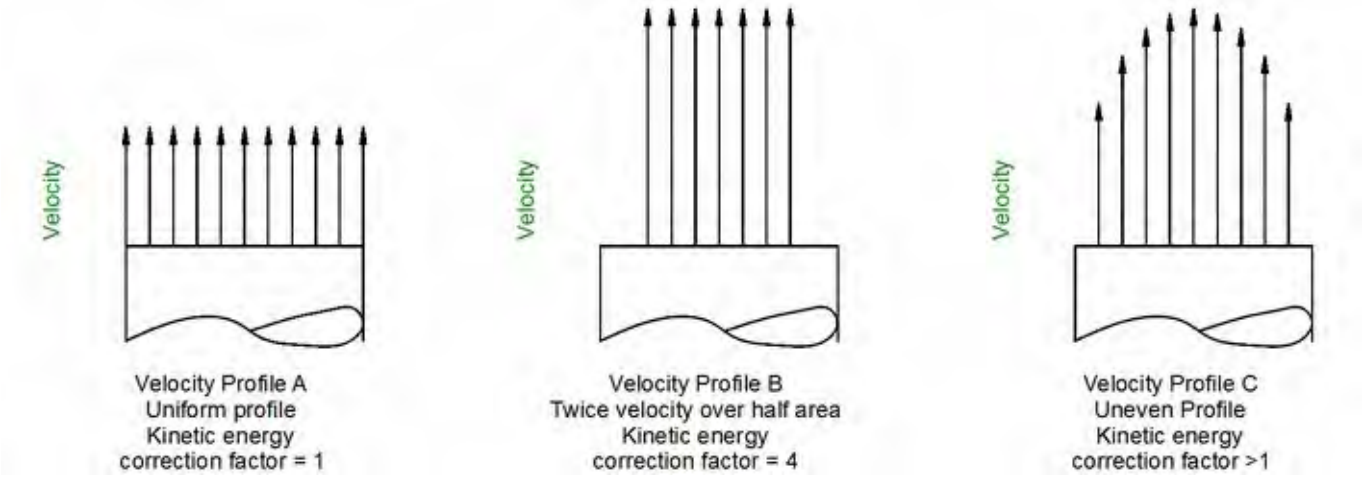


Figure 6: Examples of velocity profiles and associated Kinetic Energy Correction Factor (KECF).

In velocity profile "A", the velocity is completely uniform across the section and the KECF is unity; this is the assumption made by most (all?) of the theories on chimney performance. In profile "B" the flow has been constrained by some arbitrary method to only half the area, it would flow at twice the velocity and hence the kinetic energy is 2^2 times that of case A, hence the KECF is 4. Profile "C" shows a more realistic profile and it will have an KECF greater than 1 - that is, the energy of the flow is greater than the mass flow x mean velocity squared

For settled turbulent pipe flow the velocity distribution is near uniform (except near the pipe wall) and the KECF is about 1.06, so in engineering terms this is usually sufficiently close to unity to be ignored.

The jet issuing from the blast pipe is very far from uniform leading to a momentum correction factor greater than 1. The velocity distribution of a free jet is well known. Bill Hall (Ref. 7.6) gives the equation as:

$$\frac{U}{U_{max}} = \frac{1}{(1+62.5(r/x)^2)^2}$$

Where:

U = Velocity at radius r

U_{max} = Maximum velocity (at pipe centreline)

x = Axial distance from blast nozzle

From the above, the KECF at the chimney throat entrance can be derived. It has a value of 3.4 and that value is constant for all free jets (following from the free jet equation).

Similarly, a numerical integration of the measured chimney outlet velocity profile of Young's test Ref. 7.9 , Page 135-136 gives a KECF of 1.2. This calculation did not include the effects of density variation across the chimney diameter.

So we can see a decay in KECF from 3.4 to 1.2 across a length of 2 throat diameters in Young's tests (Ref. 7.9 Page 136). We also know the decay in KECF will be an exponential decay toward a turbulent flow distribution, so by substitution we can infer that KECF decays approximately to the following formula:

$$KECF = 1.06 + 2.345\, e^{(-1.4\, L/D)}$$

Where

KECF = Momentum correction factor at a given plane in chimney

L = Distance of plane downstream of chimney throat [m]

D = Diameter of chimney throat at entrance [m]

e =Base of natural logarithm

The above formula is applicable to a single blast pipe chimney and cannot be considered as accurate, since it is based on just two data points. However, it shows the probable trend of decay in KECF for a chimney with a very modest taper angle. Given that for experimental work, 50 to 100 straight diameters of pipe my be needed to ensure a settled turbulent flow pattern, the above formula seems to predict too rapid settling of the velocity profile. Much more data from physical or mathematical models would be needed to develop the method to a useable design tool.

4.2 **Multiple Orifice Arrangements**

Inspection of the front end energy equation, as summarised by Porta, for example shows that if the kinetic energy leaving the chimney can be reduced, then the overall performance will be improved. The exit kinetic energy in this context includes the additional energy implied by the KECF. So if the KECF can be reduced, then the chimney performance will be improved. This

section will show how multiple orifices reduce the exit KECF toward unity and thus improves front end performance.

Consider the two velocity profiles shown in Figure 7. They are drawn to scale for a free jet to equations outlined in Ref. 7.6 and can be considered as a velocity profile at entrance to a chimney. In the left case, there is a single nozzle and as shown above it has a KECF of 3.4; the momentum equations also dictate that it must have a certain velocity to give the required chimney draught in a given chimney.

On the right of Figure 7, a two dimensional view of a four nozzle chimney is shown. The peak velocity must be the same as for the single chimney to give an equivalent performance and the KECF is still 3.4, since we are integrating two of the same free jet velocity equations.

However, the shear rate dV/dx for the four nozzle case is twice that for the single nozzle chimney, because we have two complete cycles of the same velocity profile occupying the same chimney diameter. If we consider a simple turbulent viscosity model (a principle used by Dr. Koopmans in his analysis of momentum re-distribution in Slide7 of Ref. 7.1) then the decay of the velocity profile is proportional to dV/dx. So the velocity profiles in the four nozzle chimney decay at a rate twice that for the single nozzle chimney.

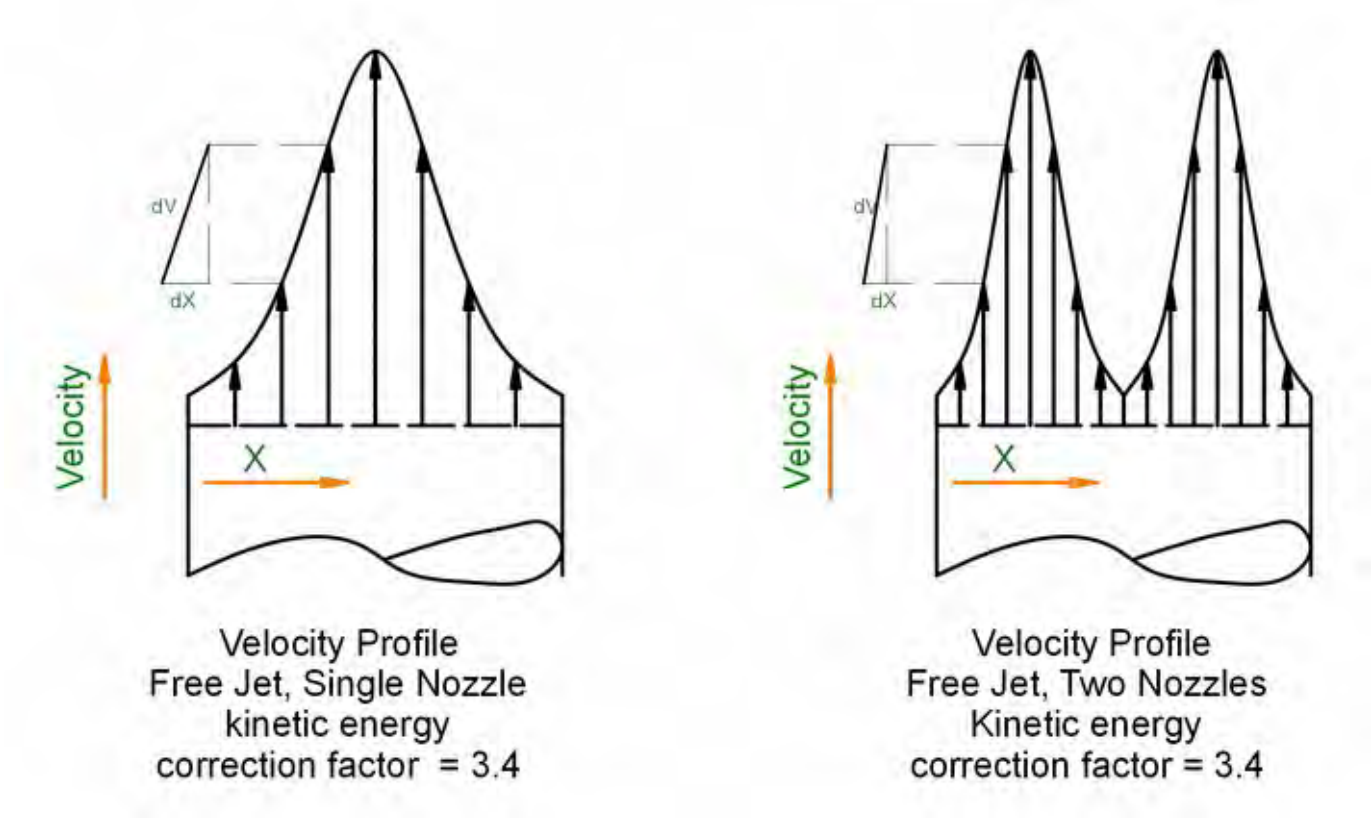


Figure 7: Velocity profile for free jets entering chimney, drawn to scale

The foregoing description of the mechanism responsible for improved performance from multiple nozzles is a simplified quasi 2D description of a more complex 3D flow pattern.

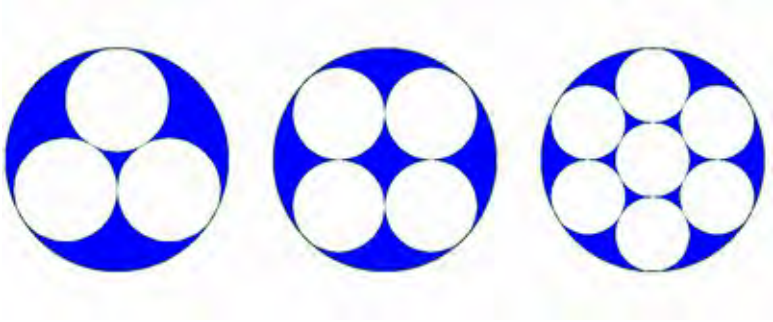


Figure 8: View of how various multi-nozzle arrangements would fill the chimney throat.

Figure 8 shows a view vertically down the chimney, if we designed 3, 4 or 7 blast nozzles such that the jets just filled the chimney throat. The blue areas show areas of "slack" flow, which become less in area as the number of nozzles increases, and hence performance would increase if all other things remained equal. This shows that we are really dealing with quite a complex three dimensional flow field, and the principle outlined in Figure 7, needs some modification for 3D effects. In addition, turbulence modelling is far more complex than the turbulent viscosity model quoted. Nevertheless, the essence of the problem is as shown in Figure 7.

So we can see the equation governing the value of KECF along the chimney would probably be be of the form:

$$KECF = 1.06 + a\,e^{(-b\,L\sqrt{n}/D)}$$

Where:

n = The number of nozzles.

a & b = Experimentally determined constants.

Other terms as previously defined.

While the above does not offer a simple formula for calculation of performance, it should be feasible to develop charts showing the change of momentum correction factor along the length of a chimney by using Computation Fluid Dynamics or physical modelling to provide the data for a range of geometries. Even at this stage, it is clear that multiple nozzles offer considerable advantage for short chimneys, but the benefit decreases rapidly when applied to longer chimneys because the exponent in the above equation is the product of L/D (chimney length) and √n (number of nozzles), so if L/D is already large there is little to gain by making it larger with multiple nozzles.

5 THE GIESL EJECTOR

An examination of the Giesl ejector may help to bring some of the above concepts together.

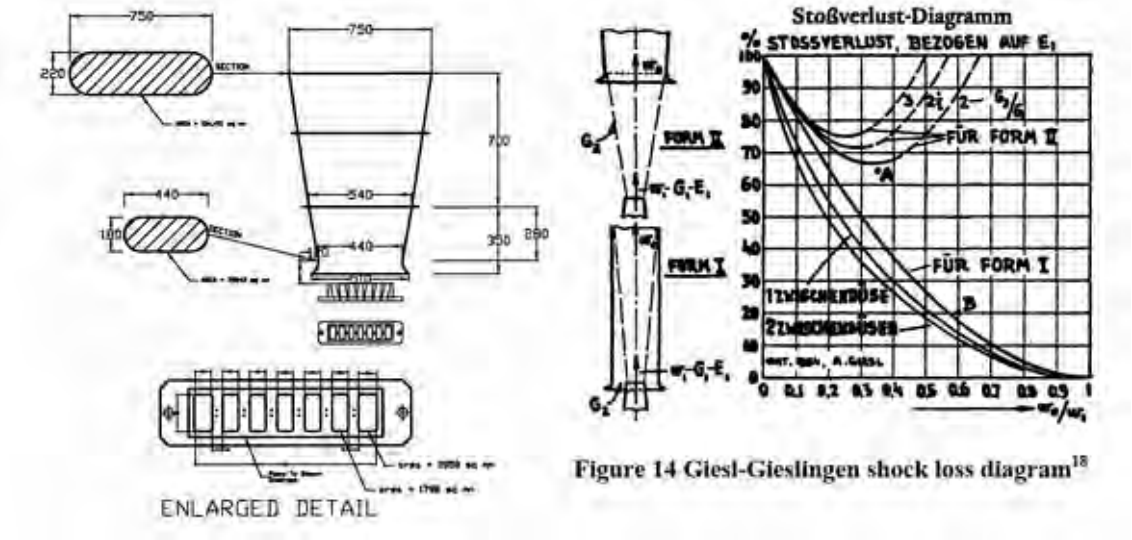


Figure 14 Giesl-Gieslingen shock loss diagram[18]

Figure 9: Giesl ejector from Ref. 7.7

Starting with the diffuser, the above diagram gives details to calculate the area ratio at 2.14. The non-dimensional length is a bit more difficult; Miller (Ref. 7.10) gives data for rectangular diffusers diffusing only in one plane (i.e. 2 sides parallel), which is not applicable to the Giesl as it tapers on four sides. Accordingly, I have estimated a non dimensional length on an equivalent area circle and a hydraulic radius at the throat. That gives N/R1 between 6.5 and 7.4, which puts the diffusion comfortably in the stable region below the optimum line in Figure 1. The blast nozzles include a large nozzle at each end of the 7 nozzles, helping to direct high energy flow into the area of greatest diffusion (See Section 3.1). The nozzles are splayed to fill the side view of the diffuser, helping to establish a diffusing flow pattern and in addition, each nozzle is shaped to assist the resulting jets to completely fill the throat. With seven discrete streams of flow, high shear rates within each jet will give rapid settling of the velocity profile to give a low kinetic energy correction at exit.

6 CONCLUSION TO PART 1

Dr. Koopmans proposed method of calculation for multiple nozzle front ends mis-interprets diffuser performance data and is not based on sound reasoning.

I have summarised some salient points about diffuser design as applied to locomotives, based on recent collations of experimental data and hope this dispels some of the myth surrounding this difficult topic.

I have also put forward a soundly based development of why multiple nozzle systems perform better than single nozzles.

Next time, I shall take a look at detailed blast nozzle design and then show how front end design equations can be modified to account for realistic velocity profiles.

Further down the line, I shall put forward a rational method of selecting the front end design point and take a look at the topic of time dependent behaviour (Chuffing!) and spark throwing.

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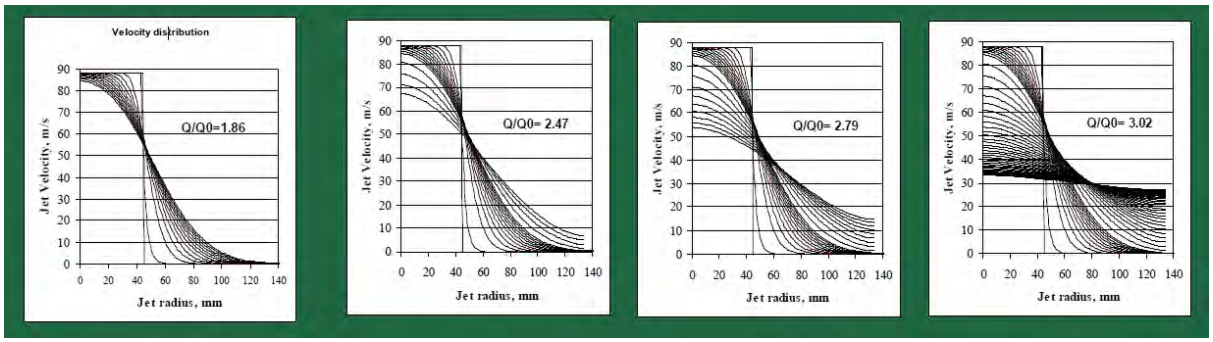
7.14 Wardale, D. "The Red Devil and Other Tales from the Age of Steam", ISBN-10: 1909358010 Publ. Camden Miniature Steam Services; 2nd edition edition (9 May 2013)

Extracts from Jos Koopmans' presentation at ASTT's 2019 conference referred to in Martin Johnson's article.

Go to https://www.advanced-steam.org/wp-content/uploads/2019/10/Jos-Koopmans-on-Multiple-Orifices.pdf to download the whole paper

Slide 7

Momentum redistribution in a confined jet, (kiss) calculation



- The calculation shows a degeneration of the rectangular velocity distribution at the orifice into apparent normally distributed velocities.
- The systematic momentum transfer shows an increase of the entrained mass and a flatter velocity distribution.

Slide 10

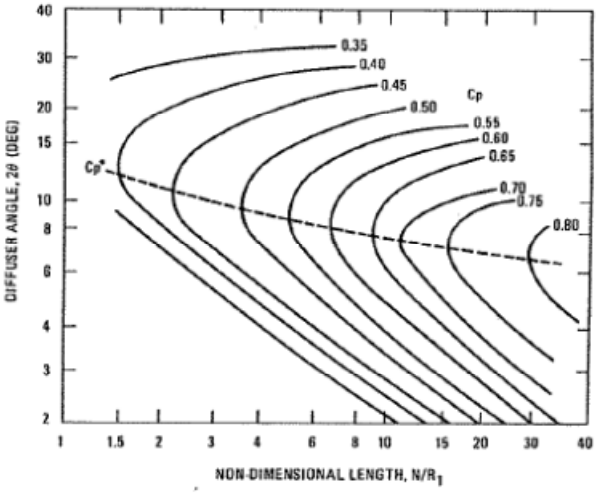
Diffusers, what do textbooks say: Fox McDonald "Introduction to Fluid Mechanics"

Losses in diffusers depend on a number of geometric and flow variables. Diffuser data most commonly are presented in terms of a pressure recovery coefficient, C_p, defined as the ratio of static pressure rise to inlet dynamic pressure,

$$C_p \equiv \frac{p_2 - p_1}{\frac{1}{2}\rho V_1^2} \qquad (8.41)$$

This shows what fraction of the inlet kinetic energy shows up as a pressure rise. It is not difficult to show (using the Bernoulli and continuity equations; see Problem 8.201) that the ideal (frictionless) pressure recovery coefficient is given by

$$C_{p_i} = 1 - \frac{1}{AR^2} \qquad (8.42)$$



- Of course "real" diffusers behave differently. Cpi=0.41 vs 0.35 at L/R=4
- Behaviour is expressed in non-dimensional ratios.
- What about absolute size???

Slide 13

Buckingham and chimneys

The 8 independent variables of a locomotive chimney problem are:

- *d* diameter of orifice
- *W1* velocity of steam from orifice
- *ρ* density of orifice steam
- *μ* viscosity of orifice steam
- *ω* exhaust beat frequency
- *x* distance from orifice to throat
- *D* diameter of chimney throat
- *L* length of chimney, throat to exit

Dependent variables are:

- Δp smokebox vacuum
- *W2 mixture velocity in chimney*

The primary dimensions are: *M, L, T*: Mass, Length and Time.

The repeating parameters would be: *d*, *W1*, *ρ*

Following the recipe of the Buckingham method the final relation for the chimneys results in :

- f (D/d, x/d, L/D, μ / (ρ W1 d), ωd /W1)
- Reduced from 8 to 5 ratios
- Absolute size is taken out, hence a scaled system is “similar” and will give scaled performance.

Slide 14

Young (Illinois) test: performance comparison of single (double length)-four orifices

Proof of Buckingham explanation 1933

A study of the locomotive front end ; including tests of a front-end model; Everett Young
https://www.ideals.illinois.edu/handle/2142/4435

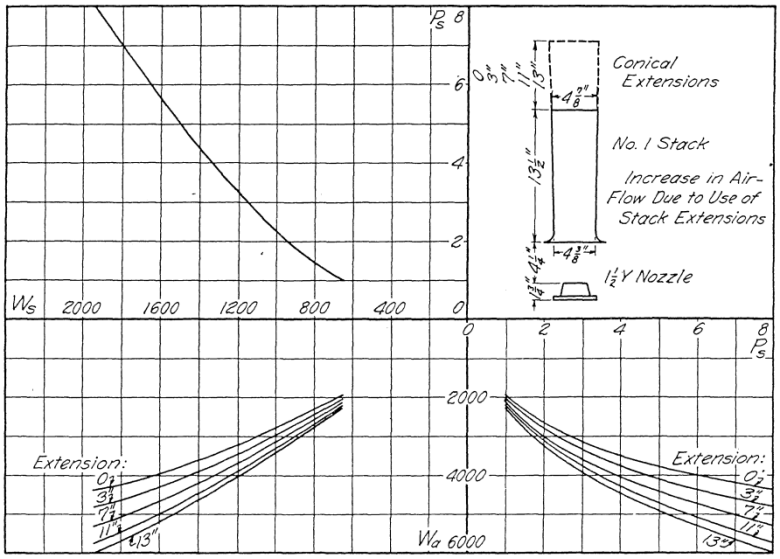


FIG. 17. STACK HEIGHT TESTS, NO. 1 STACK

Conical extensions of four different lengths having the same taper as the No. 1 stack, were added to it; the increase in air flow was remarkable, but any extension greater than 3 in. represents an impracticable condition in actual locomotive service due to clearance limits.

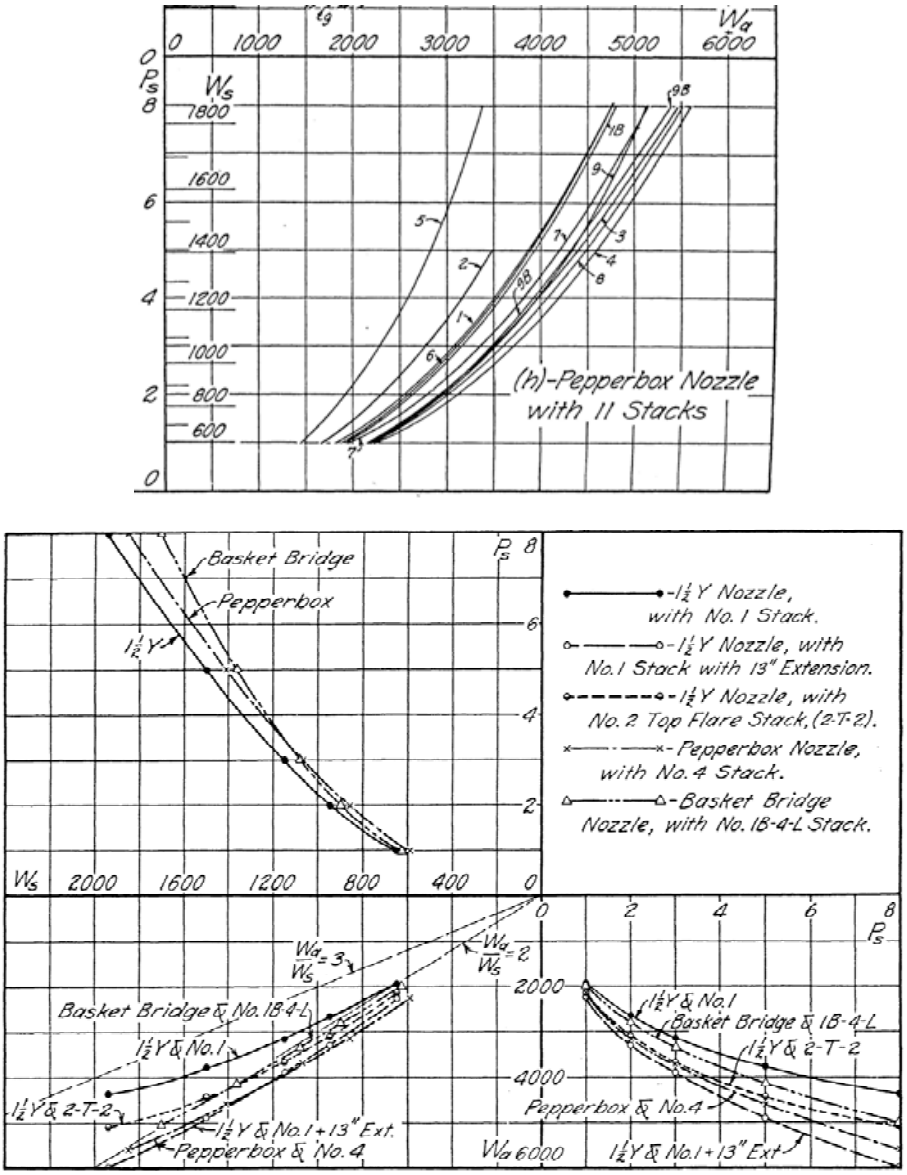


FIG. 28. COMPARISON OF VARIOUS STACK AND NOZZLE ARRANGEMENTS

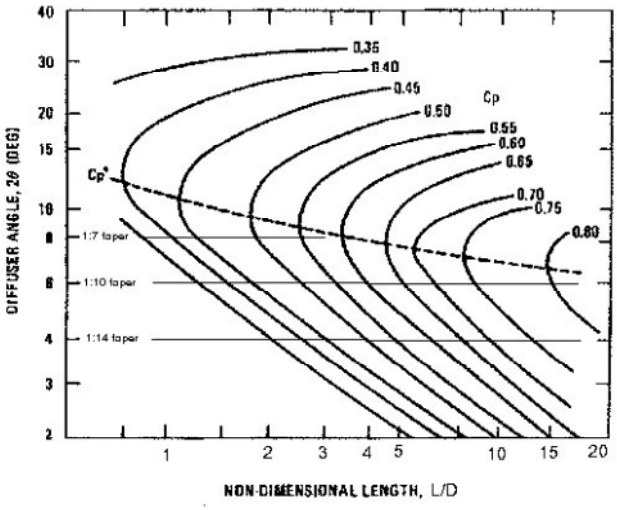
Slide 15

Fact: Any multi-orifice system is a scaled version of a single orifice system with an enlarged chimney and will give equivalent perfo rmance.

Slide 16

Diffuser data reconsidered, taper and L/D

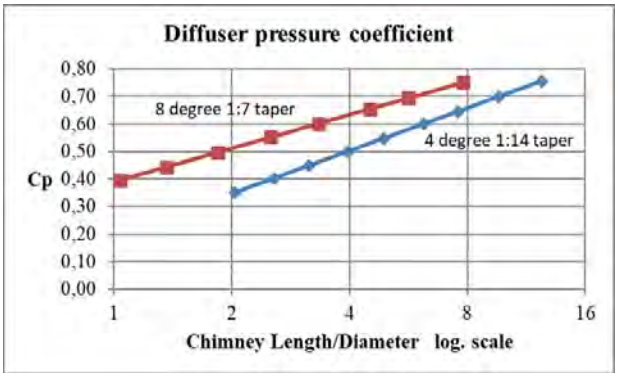
from Blevins "Applied Fluid Dynamics Handbook"



Equations:

4 degree 1:14 taper y = 0,1552x + 0,1919
8 degree 1:7 taper y = 0,1226x + 0,3887

Note x is logarithmic scaled: x= log(L/D)/log(2)



Slide 21

The momentum equation (4)

The exit momentum defines all "poor behaviour"

Since the pressure coefficient is defined as: $C_p = \Delta P / \frac{1}{2} \rho V_i^2$, it can be rearranged as:
$\Delta P = Cp \times \frac{1}{2} \rho V_i^2$.

Insertion into the momentum equation "just for the fun of it" gives:

$$A\, Cp \tfrac{1}{2} \rho V_i^2 = A_i \rho_i V_i^2 - A_e \rho_e V_e^2 \text{ or}$$

$$A \tfrac{1}{2} \rho V_i^2 = (A_i \rho_i V_i^2 - A_e \rho_e V_e^2)/ Cp$$

This is only needed to show the location of Cp in the exit momentum

Hence if the exit momentum is relieved from its ideal C_{pi} and this is replaced by the measured C_{pm} the "useful" momentum equation can be used.

Define efficiency: $\eta_{cp} = C_{pi}/C_{pm}$.

Slide 23

The momentum equation (6)

- For a single orifice use c_{pi}/c_{pm}
- For a double chimney calculate cpm from a L/D ratio sqrt(2) larger, 141%
- Same for 4 orifice unit, L/D is 2 times larger, sqrt(4).
- 5, 6, 7, 8 orifices, use sqrt (no of orifices).