The Design of the Proposed 5AT High-speed 4-6-0 Locomotive

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1. The aim of this paper is to set down the rationale behind the concept and principal details of the 5AT.
2. Origin. The genesis of the 5AT was an example given in the present writer's 1998 book *The Red Devil and Other Tales from the Age of Steam* showing to what degree the performance of 1950's steam traction could have surpassed that of contemporary BR locomotives by better design. The Class 5 4-6-0 was taken as the quintessential latter-day British type, and performance curves estimated from results achieved on the South African Railways with rebuilt 4-8-4 No. 3450 were given showing that this relatively small locomotive could have delivered some 25% more drawbar power than larger 4-6-2's if its detail design had been improved according to then-available knowledge. The first important factor is thus established, i.e. significantly better performance can be obtained from a given mass of hardware than hitherto thought possible, this corresponding to the general trend in power generation engineering and allowing smaller locomotives for a given duty. From the power capacity viewpoint, such a medium-size 4-6-0 would have been more than adequate for any former 4-6-2 duty.
3. This idea was aired in the article *Whither Steam Now?* (*Steam Railway* magazine, April 1998) when it was suggested that steam would have to improve if it was to continue indefinitely in mainline excursion service on a railway where operating standards are, of necessity, becoming ever more stringent. Specifically it would have to be faster, more reliable, and have much greater range between supplies replenishment than today's heritage locomotives. And as the reason for steam's continued use is largely its aesthetic appeal, it must retain its classical appearance and aesthetically attractive operating characteristics, for example those associated with a free-exhausting engine. These factors are foundation stones shaping the 5AT design.
4. The opportunity to flesh out this idea came in the September – October 2000 issue of the magazine *Locomotives International*, in which a tentative description of what the proposal's main features might be was given, together with a line drawing showing what it might look like. From this an artist's impression prepared by Robin Barnes was given in the following issue of *Locomotives International*, the first of a number of such illustrations.
5. After the article in *Locomotives International* a team was gradually assembled to develop a business plan for the project, with the present writer responsible for engineering design. One of the tasks has been to prove the engineering fundamentals of the proposal, which has resulted in completion of the Fundamental Design Calculations (FDC's). These cover thermo-mechanical engineering criteria that the locomotive must satisfy in order to perform as intended, i.e. that are critical to the proposal's feasibility. The present state of the proposal, which is the subject of this paper, accords with the results of these calculations, which have shown that on a purely engineering basis the locomotive should be able to deliver the stated performance. They do not deal with any criteria that the railway and/or safety authorities might insist that the locomotive satisfies.
6. Before making the FDC's a suitable designation for the proposal was agreed on, i.e. 5AT. '5' reflects the design's genealogy, descended from the BR 5MT 4-6-0 of 1951, and 'AT' stands for 'Advanced Technology'. The extent to which the technology is 'advanced' is to some extent limited by the overriding need to retain the steam locomotive's aesthetic appeal and therefore its classical layout, this in turn being mandated by the locomotive's intended use in the leisure train industry. The 5AT is not intended to be a 'high tech.' competitor to diesel and electric traction in normal commercial service, for which it would almost inevitably have to sacrifice its traditional form and therefore all-important aesthetic qualities. Also engineering progress is generally made a step at a time, especially when development resources are limited and successful performance 'first time' has to be as far as possible guaranteed, both of which apply in the present case. This dictates that the technology in the 5AT is a combination of (i) the best of former steam practice that has already been service-proven, (ii) thoroughly thought-out improvements to (i), and (iii) new technology for which there is a very high degree of confidence that it will work as intended and will not interfere with the working of any other part of the locomotive. The design conforms to these requirements. There is nothing contributing to its all-round performance that is beyond the engineering knowledge of the 1950's, with one exception, i.e. what must be a 'state of the art' oil burner for reasons of combustion efficiency. Given that the locomotive's power capacity could have been realised with a 1950's oil burner or with the gas producer combustion system, which is 1950's coal firing technology, albeit at lower efficiency than with today's combustion equipment, the original contention, that steam power such as the 5AT could have been running in the 1950's, has been substantiated by the FDC's. Also, with two non-essential exceptions, the FDC's have been made by methods available to 1950's engineers, i.e. without computers.
7. Several parameters had to be fixed at the outset of the FDC's, as follows.

7.1. To facilitate acceptance by Network Rail and give a wide route availability, the following parameters of the engine (excluding the tender) conform to those of the BR 5MT:

7.1.1. Overall length.

7.1.2. Axle spacing and lateral movement.

7.1.3. Maximum axle load (20 metric tons). This is to apply to each coupled axle giving an adhesive mass of 60 metric tons, with 20 metric tons for the leading bogie giving a total engine mass of 80 metric tons (BR5 = 77.2 metric tons).

7.1.4. Coupled wheel diameter (1 880 mm / 74 in.).

7.1.5. All overall dimensions constrained by the moving structure gauge are not greater than for the BR5, and may be smaller where necessary to comply with the present gauge.

7.2. Maximum continuous operating speed is taken from the AAR design rule for high-speed motion design, which gives a design speed in mph of {1.5 x the driving wheel diameter in inches}, in this case 1.5 x 74 = 111 mph or 178 km/h. This is rounded up to 180 km/h / 112.5 mph. Acceptance may require testing for proof of ability to run safely at 10% overspeed, giving 198 km/h, say 200 km/h or 125 mph. This being the maximum authenticated speed level achieved on test by steam traction in the past, and also Network Rail's current maximum line speed, on certain lines only, it is likely the very highest speed at which a steam locomotive might be allowed to run, and it was taken as the speed for which all parts of the locomotive affected by speed are designed. Whilst it may seem optimistically high, (i) one aim has been to show in principle that a locomotive of BR5 format can be designed for such a speed, (ii) the authorised speed can be anything up to the design speed (but not in excess of it) therefore setting the latter at 200 km/h gives the greatest scope for a high top speed being authorised, and (iii) if the maximum permitted speed is < 200 km/h the locomotive will operate with a high safety factor.

7.3. Two simple expansion cylinders are to be used, as for the BR5. Consideration was given to both 3-cylinder simple and 3-cylinder compound engines, but neither gave a clear enough advantage over two cylinders to justify its extra complexity. In particular, a crank axle for 125 mph running would prove a major design and acceptance difficulty. The suitability of two cylinders for high speed depends entirely on the resultant forces on the locomotive and track: if these can be kept to acceptable levels two cylinders can be used, with all their advantages of simplicity and accessibility. A significant part of the FDC's was to prove this point.

7.4 The preferred fuel from both the combustion and availability aspects is diesel fuel / gas oil, and the FDC's have been based on this being used. Supplementary calculations estimated that burning coal with the gas producer combustion system using mechanical firing is technically feasible (required heat release rate ≈ 21.3 GJ/m^2 grate-h giving a burning rate with good-quality coal ≈ 700 kg/m^2-h) but would result in a drop of at least 10% in combustion efficiency at full boiler load.

7.5. A double-bogie tender is to be used of high capacity (≈ 7 metric tons of oil fuel and 46 metric tons of water). The tender is subject to the conflicting requirements of minimising its mass (of particular importance for high-speed running) and carrying sufficient supplies for long range. The latter has priority, but tare mass is to be minimised by monocoque construction, the total being 80 metric tons with full supplies (axle load = 20 tons). With this tender, total locomotive wheelbase ≈ 18.9 metres and overall length over buffers ≈ 22.1 metres, slightly less than that of the largest former British steam locomotives. Such a tender would have to be designed by specialist vehicle designers, and apart from establishing its overall dimensions and brake system features it has not been considered further in the FDC's.

7.6. The locomotive and tender are to be semi-streamlined, as shown in the various artist's impressions. At 200 km/h streamlining is not a luxury: the degree of streamlining is a compromise between the requirements to retain an aesthetically pleasing appearance and minimise air resistance.

7.7. Target maximum cylinder and equivalent drawbar power – speed characteristics were derived from the cylinder power – speed characteristic achieved by SAR locomotive No. 3450, with due allowance for differences between the two designs (generally favourable to the 5AT, being a new design and 3450 a rebuild). These gave the following target figures for the 5AT at average mass (142.2 metric tons at 2/3 supplies):

7.7.1. Maximum sustainable drawbar power at constant speed on level tangent track = 1 890 kW.

7.7.2. Speed at this power = 113 km/h (71 mph).

7.7.3. Required sustainable cylinder (indicated) power at [7.7.2] to give [7.7.1.] = 2 380 kW.

7.7.4. Maximum cylinder (indicated) power = 2 580 kW.

7.7.5. Speed at this power = 167 km/h (104 mph).

7.8. The normal maximum working boiler pressure is set at 2100 kPa, for acceptance reasons just below the maximum used on conventional locomotive boilers in the past. However for operational convenience (i.e. ease of keeping boiler pressure at the maximum working value without the safety valves lifting) the pressure at which the first safety valve lifts should be slightly higher, and the boiler is designed for 2130 kPa.

7.9. The piston stroke is made long for good cylinder efficiency (800 mm), which at a maximum continuous operating speed of 180 km/h gives a mean piston speed of 13.5 m/s, within the maximum achieved during the steam era.

7.10. The nominal starting wheel rim tractive effort is initially set for calculation purposes at a realistic 25% of the adhesive weight, = 147 kN.

7.11. From known factors the cylinder bore diameter is now determined from the usual tractive effort equation, with an allowance for 90 mm diameter piston rods and tail rods, giving a cylinder diameter of 452 mm, rounded down to 450 mm, which in turn fixes the nominal tractive effort at 146 kN.

7.12. From [7.7.2] and [7.7.3] the indicated work required per piston stroke at maximum drawbar power is determined, and from this and the known cylinder dimensions an estimated indicator diagram is determined predicting 25.5% cut-off at these conditions.

7.13. The inlet steam temperature at rated maximum evaporation is fixed at 450 ^{O}C, a realistic maximum from both superheater design and cylinder lubrication aspects.

7.14. From [7.12] and [7.13] a cylinder steam flow at maximum drawbar power of 16 200 kg/h is predicted, requiring a total (rated maximum) evaporation of some 17 000 kg/h.

8. The cardinal factors for the locomotive's design are thus determined as above, either fixed as realistic optima based on experience or calculated to achieve the target maximum drawbar power. The more detailed FDC's follow from this and the findings of these will now be described item by item. The predicting of indicator diagrams being a laborious process, no attempt has been made to use this technique to check the overall validity of the target power – speed characteristic, which however has the advantage of being based on what has actually been achieved with steam traction (3450) by applying the kind of technology specified for the 5AT. Instead the maximum cylinder power – speed curve has been predicted by the *Perwal* program of the late Prof. W. Hall using the 5AT data established by the FDC's. This gave close agreement with the target curve at speeds higher than 100 km/h (≈ 60 mph) and higher power at lower speeds. The cylinder power at maximum drawbar power [7.7.3] from *Perwal* was within 0.4% of the target figure used for the FDC's.

8.1. The target power – speed characteristic is sufficient to allow the haulage of a 400 metric ton trailing load (11 coaches) at 160 km/h (100 mph) on level tangent track, and 250 metric tons (7 coaches) at some 180 km/h (112 mph).

9. Reciprocating components of the engine and drive gear (pistons and rings, piston rods and tail rods, crossheads, and reciprocating part of connecting rods). Whilst these components must satisfy functional criteria in respect of strength, resistance to steam leakage, minimum wear, etc., the criterion of minimum mass is of critical importance to a high-speed 2-cylinder locomotive because of the balancing problem. Therefore the components are designed to be as light as possible consistent with their functional requirements, a target of 250 kg per cylinder being aimed for. Their design was worked out in sufficient detail to give an acceptably accurate estimate of their mass, resulting in the following features.

9.1. Pistons, piston rods and tail rods. Each piston head is machined from a single forging to the form of two thin discs joined by a hollowed-out hub and by ten 4mm thick welded-on radial ribs, and having a separate welded-on crown with six diesel-quality narrow rings. Not only does this construction combine strength with lightness but it minimises internal cylinder heat transfer across the piston (the hollow formed by the piston faces and crown may be evacuated and possibly bright chrome (brush) plated to almost eliminate heat transfer across the piston). It is this kind of refinement that negates any advantage of compounding. Material is specified as SAE 4340 C-Mn-Si-Cr-Ni-Mo steel, 'the standard to which other ultra high strength steels are compared' and all-round the best possible material, having the important advantage of weldability. It is specified for all components where high strength : mass ratio and good fatigue resistance are important, e.g. for the piston rods, crosshead bodies, connecting and coupling rods, main crankpins, etc. The pistons are welded to hollow piston rods and supported at the front by hollow tail rods running in split spring-loaded bearings, which prevent tail rod sag and piston head – cylinder liner contact. Fully-floating multiple-element piston rod and tail rod packings are specified for long-term steam tightness.

9.2. Crosshead assemblies and slidebars. The crossheads are based on the successful SAR 25 Class split Timken-type design having a hollow gudgeon pin pressed into the connecting rod small end and 'full complement' taper roller bearings between the pin and the forged crosshead side plates. 'Alligator' slidebars are specified so that the principle dynamic load on the slippers (the 'crosshead reaction') is always compressive and does not therefore cause fatigue, and taking advantage of this the slippers can be in aluminium alloy, such as Alcoa 70-ST alloy (heat-treated condition), for lightness**.** Each pair of slidebars is supported from the mainframe about the position of maximum crosshead reaction, giving stiffness to the arrangement where it is most needed.

9.3. Connecting rods. Each I-section rod tapers vertically over the front part of the rod shank to a maximum depth at the rod centre, which depth is retained in the back half of the shank. The I-section flanges also taper outwards from either end to a maximum width over the middle 70% of the shank's length. In this way the best possible resistance to the complex loading on the rods is achieved with the minimum mass. Despite their arduous duty, each connecting rod has a calculated mass of only some 146 kg, of which some 69 kg is reciprocating. Each rod is fitted with a spherical roller bearing at the big-end, these bearings being used throughout on the crankpins and axles, being the best type for this application on account of their self-aligning function, which accommodates any misalignment and cumulative crankpin/axle/rod deflections.

9.4. The calculated total mass of the reciprocating components of the engine and drive gear at this stage came to 251.3 kg, acceptably close to the target figure of 250 kg (see also par. [11.2.4.]. The determination of this was crucial to the suitability for high speed of the 5AT in its proposed form, and was therefore a necessary condition for continuing the FDC's.

10. Crankpins, coupling rods, and driving & coupled axles. These components are critical to the safe operation of the locomotive, but being revolving, they can, where appropriate, be precisely balanced. Therefore the requirement for minimum mass, which applies to reciprocating components, does not necessarily always apply here. This suggests the use of design rules which have stood the test of time. AAR rules are taken as being the best available, components designed according to these rules having proved safe under the extremely arduous North American operating conditions. However the sustained very high (for steam traction) speeds required of the 5AT are an additional factor not necessarily accounted for by the AAR design rules, and extra calculations involving inertia forces are made to allow for this where necessary.

10.1 Crankpins. The leading and trailing crankpins present no problem and solid carbon steel pins may be of 115 mm diameter. The outside diameters of the hollow main crankpins by the AAR design rules, based on nominal piston thrust and plain carbon steel of 552 – 586 N/mm^2 UTS, are 190 mm at the coupling rod bearing and 180 mm at the connecting rod bearing. However calculations of main crankpin loading due to inertia forces at 200 km/h with worn tyres (coupled wheel rotational speed of 9.8 Hz) found it to be over twice the maximum piston thrust due to steam pressure. There are three items which can be acted on to address the crankpin stress situation: (i) crankpin diameter, (ii) crankpin material and (iii) the extent of cushioning provided to the reciprocating masses by steam compressed in the cylinder as the piston approaches dead centre. The crankpin diameters given above were not altered as any increase would itself increase the mass of the crankpin, connecting rod big-end, coupling rod eye, crankpin roller bearings, withdrawal sleeves and seals, and would therefore give additional inertia load. The crankpin bore (φ75 mm) could be eliminated but would result in only a small increase in section modulus, i.e. small reduction in maximum bending stress. If the diameters remain unchanged alloy steel must therefore be used because of the high stress levels occurring at maximum speed, the AAR design rule allowing for an increase in actual stress proportionate to the increase in material UTS, and the main crankpin material is therefore to be SAE 4340 steel. Even so, the crankpin bending stress due to inertia load at maximum speed would be greater than permissible, which mandates cushioning of the reciprocating masses by compressed steam to relieve the inertia load on the main crankpin. A pressure of approximately 1100 kPa at dead centre is required, which is consistent with mid-gear drifting and the use of a small amount of drifting steam.

10.2. Coupling rods. Being rotating and therefore capable of precise balancing, coupling rod design is not critical to the feasibility of the 5AT concept, which is the field of the FDC's. However coupling rods are subject to extremely high vertical bending at high speed due to their own mass, which should therefore be kept to a minimum, and exploratory stress calculations were therefore made for the coupling rod shanks by AAR design rules. These showed that the coupling rods would have to be of SAE 4340 steel to keep their maximum vertical depth to a practical level (≈ 190 mm, 10 mm more than the corresponding connecting rod dimension). To minimise their mass the coupling rods shanks must taper down towards their ends both vertically and horizontally from a maximum section at the centre where bending is a maximum.

10.3. Driving & coupled axles. Whilst the axles are most important components for the safe working of the locomotive they are not critical for the purpose of the FDC's, as no limiting clearances prevent them from being made to effectively whatever diameter is required. High speed is also not a limiting factor in the operation of the axles themselves, although it determines the inertia forces of other components which have a bearing on axle loads. For the purpose of the FDC's the main driving axle diameter at the bearing was determined by the very simple (though service-proven) AAR design rule, rounded up to the next highest bore diameter for a standard spherical roller bearing, 240 mm. The AAR rule gives no specific recommendation for coupled axles, which are generally more lightly loaded than the main driving axle and usually of smaller diameter. However the BR 5MT main driving and coupled axles have the same diameter, and this feature is retained for the 5AT as it standardises the roller bearing axleboxes etc., with obvious cost benefits for small batch production (the coupled axles can be hollow if a smaller section modulus suffices).

11. Piston valves and valve gear. Steam distribution by piston valves driven by Walschaerts valve gear is to be used. Piston valve design was developed by the late Ing. Porta to the point where it was in a different class to that hitherto used in all respects (steam distribution, lubrication, suitability for high temperature steam, wear, resistance to steam leakage and minimum mass). Such piston valves have shown no inferiority to poppet valves in the matter of steam distribution (e.g. indicator diagrams of SAR 3450 v's BR 71000) and also remarkably low wear and minimal long-term steam leakage. The 'weak link' in Walschaerts valve gear, i.e. fairly high dieblock – expansion link wear, has been overcome by mechanical lubrication of the affected surfaces (e.g. as on the SAR 25 Class 4-8-4's) and the whole package is relatively simple engineering and much cheaper than a camshaft-driven poppet valve alternative. The requirement for piston cushioning steam when drifting (see [10.1] above) can also be met, unlike with British Caprotti gear as finally developed. There is no hesitation therefore in preferring 'state of the art' piston valves, and with Walschaerts valve gear comes the added bonus of being aesthetically superior to rotary cam drive.

11.1. Piston valves. For maximum 'internal streamlining' with minimum valve mass and cylinder clearance volume, twin 175 mm diameter valves mounted side-by-side are used per cylinder. Steam lap is 65 mm, exhaust lap 18 mm, lead 7 mm at all cut-offs, and maximum fore gear cut-off is 75% for good starting. Each valve head has 13 narrow rings, 7 in high-grade cast iron and 6 in high-tin bronze, and various features, such as insulated heads and exhaust diffusers, are incorporated to improve performance.

11.2. The valve gear is a conventional Walschaerts gear except that the usual valve crosshead is substituted by a double hanger (along the lines used in the past for driving slide valves from Walschaerts gear) to transfer motion from the combination lever to the two piston valves for each cylinder, which lie in different transverse planes. By this means the potentially damaging skewing moments resulting from the lateral force transmission to the inner valve are transformed into forces which are easily taken by the double hanger shaft's roller bearings and a torque which can be transmitted with negligible twist by the torsionally-stiff hanger. For simplicity, and to avoid any tendency for the valve drive to 'jack-knife', the articulated valve spindles (which are pin-jointed to the valves) are coupled directly to the combination lever (outer valve) and double hanger arm (inner valve) without any intermediate links. As these joints swing in an arc centred on the double hanger shaft, their motion has a small vertical component, in contrast to that of a valve crosshead which is constrained by its slidebars to move parallel to the steam chest centre line. This vertical motion is permissible because the valve spindle packings are fully floating and specifically designed to accommodate it.

11.2.1. Priority is given in the valve gear design to obtaining good valve events in the probable most frequent running cut-offs, which for a high-speed passenger locomotive can be taken as from mid-gear to 30% in forward gear. Of the various valve events priority is given to cut-off as being the factor most responsible for obtaining the desired ideal of equal work output from the piston outstroke and instroke. Combining these two priorities gives the design aim of obtaining as near as possible equal cut-offs at the front and back ports over the anticipated normal working range of cut-off from mid-gear to 30% in forward gear, to be achieved by suitable design of the suspension arrangements for the back of the radius rod. The front port and back port cut-offs are arranged to be equal at 25% cut-off in forward gear, i.e. the cut-off for peak drawbar power, and the maximum front port – back port difference over the mid gear – 30% cut-off range is 0.8%.

11.2.2. The valve gear is entirely mounted on roller bearings, of spherical roller and full complement cylindrical roller & needle roller types, as appropriate.

11.2.3. The expansion links are of lightweight 'box' type construction for minimum inertia and optimum link – dieblock lubrication.

11.2.4. The combined mass of the crosshead arm, union link and part of the combination lever (all of SAE 4340 steel) reciprocating with the piston stroke was calculated as 12.7 kg per cylinder, which added to item [9.4] gives 264 kg. However due to the use of a crosshead arm to drive the union link (not originally anticipated) each crosshead assembly will be some 4 kg lighter than originally calculated, giving a total reciprocating mass to be balanced of 260 kg per cylinder (v's 375 kg on the BR5, a 31% reduction).

12. Cylinders. The steel cylinder block, comprising left and right hand cylinders and steam chests, etc., frame stretcher and smokebox saddle, shall preferably be a single casting, or alternatively may be fabricated (welded) from steel plate or of part cast / part fabricated construction, i. e. separate castings welded together. It is to be welded to the mainframe longitudinals immediately behind the cylinders as, for example, on the DB 10 Class 4-6-2's, and bolted to the front frame extensions immediately ahead of the cylinders by shear bolts (which do not take cylinder forces) to satisfy the requirement for an energy-absorbing 'crumple zone' at the front of the locomotive. Good internal streamlining and low clearance volume are desirable features that tend to be mutually exclusive: throughout the design priority is always given to the former, which is achieved in the cylinders with a still-acceptable clearance volume of 10.6%.

The steam chest volume per cylinder ≈ 99% of the piston swept volume, a very high figure which accords with Chapelon's recommendation.

12.1. Cylinder insulation. Heat loss from steam to the cylinders occurs due to three reasons. i) Heat passing through the cylinder block metal and then lost from the exterior cylinder surfaces to the atmosphere. This is minimised by *external* cylinder insulation under welded-on clothings, to include where practical exposed surfaces of the smokebox saddle and those parts of the mainframe close to the cylinders. Wherever clearances allow, insulation is to be ≥ 75 mm thick. (ii) Heat required to raise the cylinder block metal, adjacent part of the mainframes, etc., up to their running temperature and (iii) cyclical heat transfer *within* the cylinders, *from* the incoming high temperature steam and then *to* the 'low' temperature exhaust steam and thus lost to exhaust. These are minimised by *internal* cylinder insulation, or so-called thermal barrier coatings, e.g. of ceramic material, applied to non-rubbing cylinder surfaces exposed to high-temperature steam, which inhibit heat transfer from steam to the cylinder walls. Examples which can be treated are the front and back cylinder covers and possibly the steam chest barrel and steam chest – cylinder passages.

12.2. Cylinder liners are to be of pearlitic chromium cast iron (diesel quality), and where annular spaces exist between liners and cylinder barrels they are to be filled with insulation or a thermal barrier coating.

13. Valve liners. Apart from various design features to improve performance (in respect of steam flow, lubrication and wear resistance) of the eight pearlitic chromium cast iron liners, a special feature is the cooling of the liner rubbing surfaces at the inlet side of each liner by saturated steam to obtain a rubbing surface temperature which is acceptable for lubrication of the valves at high inlet steam temperature. 300 ^{O}C liner surface temperature at 450 ^{O}C inlet steam temperature is designed for, being equivalent to a non-cooled liner operating at a modest inlet steam temperature of some 350 ^{O}C. The system is as successfully used on SAR locomotive No. 3450 and comprises helical cooling passages machined into each liner and fed with saturated steam directly from the boiler whenever the throttle is opened (requiring the multiple-valve throttle to be on the saturated side of the superheater header), the ratio of cooling : superheated steam (approximately 1 : 29) being set by the respective pipe flow resistances and remaining sensibly constant at all evaporations. Valve oil is fed directly to a lubrication inlet in each liner.

14. Wheel balancing. Optimum balancing of the engine's revolving and reciprocating masses is critical, for the following reasons: (i) the locomotive's high potential maximum speed of 200 km/h; (ii) the necessity for acceptably low wheel – rail dynamic forces; (iii) acceptable riding of the locomotive longitudinally ('fore-and-aft'), vertically, and laterally (although it must be noted that lateral stability is very largely dependent on chassis design rather than on the 'nosing couple' due to unbalanced reciprocating masses); and (iv) no transference of vibrations to the train due to imperfect balancing that are of sufficient magnitude to be detected by passengers. In short, successful balancing is one of the vital keys to enable the 5AT to achieve its target performance.

14.1. All revolving masses are to be precisely dynamically balanced ('cross-balanced') on the same wheelset as they belong to, and are then perfectly balanced and result in zero dynamic augment. As balancing of revolving masses is therefore not critical to the 5AT feasibility it is not considered further in the FDC's, apart from laying down the method involved – it can only be done numerically once the mass of each revolving component is known from detail design.

14.2. Balancing reciprocating components. Research on this subject towards the end of the steam era in the USA concluded that static balance only should be used, consequently a given fraction of the reciprocating masses is to be statically balanced. The division of the total required reciprocating balance between the driving and coupled wheels is to be such as to give the same dynamic augment ('hammer blow') on each wheel, i.e. to make the maximum individual wheel dynamic augment as low as possible. This criterion results in less reciprocating balance being allocated to the main driving wheels than to the coupled wheels because the former are subject to a separate dynamic augment resulting from connecting rod angularity. This research also concluded that the maximum unbalanced mass of reciprocating parts for one side of a locomotive should be 1/250 of the total locomotive mass in working order. The latter can include the tender mass by using 'solid' (unsprung) engine-tender drawgear and an engine – tender radial buffer such as the Franklin type, which is mandatory for the 5AT. The above fraction is reduced to 1/340 to allow for the maximum coupled wheel rotational speed of the 5AT being higher than the '1.5 x diameter speed' used in the USA. Even so, by these criteria, it was found that balancing of the reciprocating masses on the 5AT could be entirely dispensed with, even for operation at 200 km/h with worn tyres, completely eliminating wheel – rail dynamic augment due to this cause. Riding quality would be improved by adding some balance however, and as the dynamic augment of the BR 5MT has proved to be acceptable at the present maximum permitted speed of that class on Network Rail track (75 mph, 120 km/h), the maximum reciprocating balance of the 5AT is taken such that at 200 km/h its dynamic augment due to reciprocating balance shall not exceed that of the BR 5MT at 120 km/h, by whichever criterion dynamic augment is measured. This applies to both types with new tyres, as on both a given amount of tyre wear will increase dynamic augment by the same % amount, the nominal coupled wheel diameters being the same. By this criterion, a maximum of 19.7% of the 5AT's reciprocating masses may be balanced, leaving an unbalanced mass per cylinder of 1/511 of the engine + tare tender mass (i.e. minimum 'worst case' mass). The dynamic augment per axle and for the whole locomotive at 200 km/h would then be less than allowed by the 1928 Bridge Stress Committee at a coupled wheel speed of 5 Hz (≡ 106 km/h, 66 mph, for the 5AT).

14.3. Vibration analysis gave a total amplitude of fore-and-aft oscillations for non-resonant vibrations of from 2.2 ~ 2.6 mm for maximum ~ minimum reciprocating balance respectively (v's 3 ~ 5 mm given as an acceptable value), and a linear amplitude of nosing oscillation at the extreme front and rear of the engine of ± 1.05 ~ 1.2 mm for maximum ~ minimum reciprocating balance respectively.

14.4. The critical speed of the coupled wheels, i.e. the speed at which wheel-rail reaction may drop to zero due to the dynamic augment when the reciprocating balance weight is at top quarter, is of importance, particular for the leading coupled wheels if they retain any lateral guiding function. It was found to be not less than 285 km/h (178 mph), safely above the design speed.

14.5. The conclusion of the balancing calculations – generally carried out for 'worst case' conditions – is that the 2-cylinder 5AT locomotive can be satisfactorily balanced for 200 km/h operation: the key to this is the utmost lightness of the reciprocating parts and the special engine-tender connection. There is, of course, a price which has to be paid for this achievement, which is the very sophisticated – and probably expensive – engineering of the reciprocating components.

15. Feedwater heating. The overall feedwater heating system on the 5AT comprises three elements. (i) A 'hot well' and auxiliary heater in the tender tank, i.e. a compartment in free communication with the rest of the tank and from which the feed pipe to the boiler feedwater pump is run. Exhaust steam condensate from the feedwater heater proper is piped back to the tender and mixes with tender water in the auxiliary heater at the entrance to the hot well. By this means condensate is effectively mixed with the boiler feed when the feed pump is working and the hot well is continually replenished by tender water passing through the auxiliary heater, but if the engine steams for short periods without the pump operating and uncondensed exhaust steam reaches the tender it avoids undue rise in the water temperature in the hot well. (ii) A 'closed' or 'surface' type (shell & tube) heat exchanger is situated between the feedwater pump and the boiler: feedwater under pressure is pumped through the tubes and is heated by exhaust steam condensing on the outside of the tubes in the heater shell. (iii) The front of the boiler barrel is used as a Chapelon-type economiser. The boiler clack valves are situated on this economiser section, all incoming feedwater entering the boiler at the front where the combustion gasses are at their lowest temperature, giving the highest gas – water temperature difference, and consequently highest possible heat transfer in this part of the boiler, improving its absorption efficiency.

15.1 The principal source of heat recovery and feedwater heating is the surface type heater and the following factors are critical to its design for the 5AT: (i) size: the heat transfer capacity of the heater is dependant on its size, which is to be as large as possible: (ii) its position must not impair exhaust steam lifting from the chimneys: (iii) its position must not impair forward vision ahead from the cab: (iv) the condensate outlet from the heater must be above the maximum water level in the tender as condensate from the heater flows back to the tender tank by gravity, and the higher the heater the higher can be the tender tank, reducing tender length for a given water capacity. The best solution for compliance with these criteria is to split the heat exhanger into two separate cylindrical units, each mounted above the leading boiler barrel course immediately behind the chimney as on latter-day French compounds. By this means a large total heat transfer area (water side) of some 18.3 m^2 can be achieved, giving an estimated feedwater temperature at maximum evaporation of ≈ 110 ^{O}C, say 105 ^{O}C with average tube scale deposits. Some 16% of the cylinder exhaust steam passes through the feedwater heater. The preferred feedpump is the German KP4-250 compound reciprocating pump, available from Meiningen Locomotive Works.

16. Combustion air preheating. One of the few departures from normal steam practice, a combustion air preheater is not a luxury but an essential part of the locomotive. It is a crossflow finned-tube heat exchanger, similar in principle to a motor car radiator, located under the front of the firebox foundation ring. Exhaust steam flowing through the heater tubes condenses and transfers heat to combustion air passing through the heater matrix, after which this air is ducted to the oil burner air inlets in the case of oil firing. For coal firing separate heaters would be required for primary and secondary air. The heater for oil firing faces partly forwards, benefiting from the 'ram effect' by which the locomotive's forward motion assists to force air through the heater (as on a front-mounted car radiator), thereby reducing draught requirements at high speed. The nominal frontal air flow area of the heater positioned for oil firing is 1.0 m^2 and its depth in the direction of air flow some 180 ~ 220 mm depending on the type of matrix used, which is sufficient to raise the temperature of all combustion air at rated maximum evaporation from an assumed year-round-average ambient value of 15 ^{O}C to the minimum target figure of 100 ^{O}C, using 6% of the cylinder exhaust steam. This acts to reduce the sensible heat loss in the smokebox gas (one example

of 'loss control', which is the general engineering philosophy used to enable the 5AT to reach its target performance), and is one (important) factor in lifting boiler absorption efficiency at maximum rated evaporation from a typical good value of 80% to an estimated 86.3% for the 5AT.

17. Boiler. The overall size of the all-steel all-welded Belpaire-firebox boiler is kept nominally the same as that of the BR 5MT boiler (if a size adjustment to the overall dimensions were possible it would be downwards, to increase the possible thickness of insulation). This is because (i) it gives an appropriate boiler size for the required evaporation rate [7.14], (ii) it gives a boiler of suitable mass bearing in mind the locomotive's permitted total mass [7.1.3] and (iii) it allows good forward visibility from the cab, a very important feature for high-speed operation.

17.1. The question of which boiler code the strength of a new locomotive boiler should be calculated from appears problematical, as former locomotive boiler codes are obsolete and current codes, either explicitly or implicitly, do not apply to the boilers of locomotives, for which operating conditions are particularly arduous. An obvious solution is to design any part of the boiler according to either (i) a former locomotive boiler code or (ii) the current code of choice, *whichever is the more stringent*. In this way current industrial boiler design rules *and* the former design rules particularly applicable to locomotive boilers are both satisfied, which is the best guarantee of meeting all current safety and certification regulations and insurance requirements and of safe construction and operation of a new locomotive boiler. For the purposes of the FDC's, a code to which all-welded locomotive boilers have been successfully designed in the steam era has been used, i.e. the 1953 edition of the German code, *Werkstoff – und Bauvorschriften für Dampfkessel*, including amendments. Although this is a general code covering all land and marine boilers, it was formulated at a time when locomotive boilers were still in regular production and included them within its scope. The last all-welded high-capacity locomotive boilers built for the German Federal Railways were designed according to this code, *with the reservation that deviations from it were (only) allowed where justified by the particular conditions of railway service*. FDC's made accordingly specify all principal dimensions of importance to the boiler's structural strength, and show that despite the higher design pressure of the 5AT compared to the BR5 (2130 kPa v's 1551 kPa) the in-service boiler mass would be some 1.4 metric tons less. This reduction is due primarily to the welded construction eliminating butt straps and all lap joints, a smaller volume of water in the barrel due to a larger tube bundle, thinner inner firebox plates due to the use of steel rather than copper, and to the use of a U-section foundation ring instead of the solid type.

17.2. Combustion equipment. The chosen fuel is gas oil / diesel fuel (which are chemically and thermodynamically equivalent) and the combustion equipment has to be specifically tailored to this. To generate the 5AT's rated maximum evaporation in a BR5-size firebox requires a heat release rate of 11.8 GJ/m^3-h. Whilst this is not abnormal by the standards of past steam practice, and has been exceeded with both oil and coal firing, it has generally only been achieved with quite low combustion efficiency, which is not acceptable now. As an example of current standards, the Swiss 'Sonvico' oil burning system has a claimed combustion efficiency of 99.5% at a relatively low specific heat release rate, and in view of this the target full-load combustion efficiency for the 5AT has been set at a seemingly conservative 95% - if this is exceeded, so much the better.

17.2.1. The design of oil firing combustion equipment is a highly specialised field, with a large accumulation of theoretical and practical knowledge and expertise, which goes far beyond that of the present author. Given this, it is clear that the design of the combustion equipment must be undertaken by specialists in this field. On the other hand, those working in other fields of engineering are generally unfamiliar with steam locomotive design and operation, and particularly the thermal load placed on the locomotive boiler, which is size-for-size much greater than that in fixed boiler installations (for example compare the 5AT figure given above with a 'typical' figure given for "high heat release rate" in an industrial boiler of 3,2 GJ/m^3-h). This phenomenon is inherent in locomotive operation, the need to generate large quantities of steam from a restricted size of boiler to get the required tractive power from a relatively inefficient engine unit necessitating 'forcing' the boiler. As typical locomotive specific heat release rates and evaporations, together with the frequent, rapid and extreme load changes inherent in locomotive operation, tend to be viewed almost with disbelief by non-locomotive engineers, close liaison would be required between those familiar with locomotive design and working conditions and those familiar with combustion equipment to obtain an optimum 'state of the art' design for the 5AT. Given all this, the FDC's are mostly confined to the critical factor of ascertaining the basic feasibility of achieving the required heat release rate at acceptable combustion efficiency, which is essentially an oil atomisation problem. The interesting parameter is the Sauter Mean Diameter (SMD), the diameter of a liquid drop whose surface / volume ratio is equal to that of the entire atomised spray. Calculations showed that at maximum evaporation a fuel drop evaporation time of 0.1 second would be needed to complete combustion in the firebox, and that this required an atomised oil SMD of 680 µm. The SMD that may be expected from a well-designed atomiser of the type proposed for the 5AT (internal-mixing twin-fluid (steam-oil) type) is 75 µm for an oil viscosity of 100 SSU. The viscosity of gas oil (at 40^{O} C) is approximately 30 – 45 SSU, which implies a SMD of < 75 μm can be achieved with this fuel. A lower SMD gives a higher rate of fuel-air mixing and fuel evaporation, and higher volumetric heat release rate and a wider burning range. As 75 μm is almost an order of magnitude < 680 μm, the conclusion is that, despite the preliminary nature of the analysis, atomisation quality, as expressed by SMD, should be well within that necessary to complete vaporisation of the oil within the required time for complete combustion within the firebox at maximum boiler load. Therefore not only is the 5AT's rated maximum evaporation achievable from the combustion viewpoint, but the combustion efficiency may well be higher than the 95% figure assumed. Other factors determined during the combustion analysis were: (i) an internal-mixing twin-fluid atomiser uses entrainment as the steam-oil mixing mechanism, a factor that can be used together with gravity feed to determine oil flow to the burner(s) and dispense with an oil pump (as, for example, successfully used on the SMR), (ii) atomiser steam must be superheated: with the throttle on the saturated side of the superheater header this requires a separate atomiser steam superheater element in the firebox (but not directly in the flame path), and (iii) with a modern oil burner giving a fuel spray pattern and flame dimensions precisely tailored to the firebox size and shape, no flame impingement should occur on any firebox wall, eliminating the need for refractory lining in the firebox and consequently increasing firebox volume and radiant heating surface area.

17.3. Boiler tube bundle and superheater design. As the superheater elements are placed inside the boiler flue tubes these items are interlinked, the competition of boiler and superheater for the gas-carried heat passing through the tube bundle necessitating one of the critical compromises in steam locomotive design, i.e. the selection of superheater element and flue tube diameters (a problem which becomes more acute on compound locomotives with re-superheating between high and low pressure cylinders, the limited tube bundle gas flow area having to accommodate two superheaters through both of which all cylinder steam must pass). The design requirement is to obtain the rated evaporation and steam temperature with the highest possible heat transfer efficiency, lowest combustion gas flow resistance through the complete tube bundle, and lowest steam pressure drop through the superheater elements. This involves iterative fluid flow and heat transfer calculations that are laborious by conventional means, and were in fact the most lengthy of the entire FDC's. Two types of superheater, each requiring different flue tube diameters and totally different tube bundle layouts, were considered: firstly with type A elements, which have 4 element passes within a single flue tube, and secondly with type E elements which have 2 element passes within a single flue tube, each element being contained in 2 flues, giving a total of 4 passes. Each type of element has advantages and disadvantages, but the overriding factor is the ability of the boiler and superheater as a whole to deliver the required amount of steam at the required temperature. The rated inlet steam temperature at maximum evaporation of 450 ^{O}C [7.13] refers to the mixture of superheated and saturated valve liner cooling steam actually entering the cylinders, requiring the superheated steam to be at a temperature of 462 ^{O}C. Calculations for finned type A elements showed that although the required steam generation could be achieved the maximum superheat temperature that could be expected was of the order of 440 ^{O}C instead of 462 ^{O}C, and that the overall boiler and superheater absorption efficiency would be slightly less than the (then) target figure, 83,7% v's 85%. The calculations were repeated for finned type E elements with better results, principally because these elements (i) give a higher steam temperature, the most important consideration (455 ^{O}C to within some ±12 deg C at maximum rated evaporation was the best calculated figure) (ii) allow a higher boiler tube evaporative heating surface area, which in turn allows the target evaporation to be reached with a greater margin, and (iii) give a much higher total steam flow area and therefore a corresponding reduction in steam flow pressure drop through the superheater. The type E superheater and corresponding tube bundle are therefore specified for the 5AT, with the following principal details.

17.3.1. Number of large boiler tubes (flues) of 88.9 mm o/d x 3.6 mm wall = 96.

17.3.2. Number of small boiler tubes of 44.5 mm o/d x 2.6 mm wall = 70.

17.3.3. Number of Type E superheater elements of 33.7 mm o/d x 2.6 mm wall tube = 48. Each element pass has 4 welded-on fins to increase gas-side heat transfer, the length of these fins being 3496, 3000, 3000, and 1870 mm over passes 1 to 4 respectively.

17.3.4. Distance from smokebox tubeplate to Chapelon economiser tubeplate = 1 400 mm.

17.3.5. Total gas flow pressure drop (vacuum) through the boiler at rated maximum evaporation (through the combustion air preheater, firepan air inlets, firebox, boiler tube bundle and smokebox) estimated as 5200 Pa (530 mm H_2O).

17.3.6. Total steam flow pressure drop from boiler to steam chests at rated maximum evaporation (through the dry pipe, throttle, superheater header, superheater elements, and branch steam pipes) estimated as 61 kPa (2.9% of the normal maximum working boiler pressure).

17.3.7. Boiler and superheater absorption efficiency at maximum rated evaporation = 86.3%. At an assumed 95% combustion efficiency the corresponding overall boiler efficiency = 82%.

17.3.8. Increase in superheater mass, 5AT v's BR 5MT = 1250 kg.

17.4. Of the various instructions given in the FDC's concerning boiler and superheater construction, of note are (i) boiler insulation should be as thick as possible, up to a 150 mm maximum wherever clearances and forward vision considerations allow, (ii) as it is important with roller bearing coupling rods that mainframe longitudinal expansion between the main driving and trailing coupled axles is minimised, thin insulation or a thermal barrier coating needs to be applied between the outer firebox sides (but not blocking any stay tell-tale holes) and the mainframe legs, and (iii) in the case of bolted superheater element – header joints (which are preferred to welding) high elasticity must be built into the joints to avoid loosening, due for example to fretting corrosion, the best method being to use conical disc springs (Belleville washers) under the nuts.

18. Exhaust system. The exhaust system, dynamically connecting the boiler and cylinders, is thermodynamically the heart of a steam locomotive and must therefore be as good as possible, within practical limitations. That the exhaust entrains sufficient combustion air to sustain the combustion rate necessary to match the steam demand throughout the boiler's evaporative range is a cardinal point for good performance from any steam locomotive, and that it does this with the minimum of exhaust steam energy is the key to optimum performance. This point is especially important on the 5AT as the locomotive is to operate mostly at high speed with full throttle and short cut-off, giving high heat conversion to mechanical work in the cylinders and therefore limiting the amount of energy available for draughting work in the exhaust steam (it is common for locomotives to steam adequately at long to medium cut-offs but not at short, for this reason), and this is compounded on the 5AT by the use of piston valves with exhaust lap, delaying release.

18.1. Whilst the exhaust is a critical component of the 5AT, the available theory virtually guarantees a satisfactory design, which follows that made for the Chinese Railway's modified QJ locomotive, in turn a development of that for SAR 26 Class No. 3450. Together the theories used constitute the best practice known to the present author. Firstly a Lempor (Lemaitre-Porta) exhaust was designed using conservative (safe) values of various factors, then an alternative exhaust was designed according to *The Prediction of the Optimum Performance of Ejectors* by Kentfield & Barnes (Proceedings of the Institution of Mechanical Engineers, Volume 186, London. 1972), taking optimum values of the various factors used: this may be regarded as giving the very best possible exhaust proportions. The design is made only for the conditions existing at rated maximum evaporation: this avoids a premature 'front-end limit' and experience is that an exhaust properly designed for these conditions will function correctly at all lower evaporations. The exhaust is designed for oil firing: the different combustion conditions with coal firing plus the necessity for a spark arrestor and perhaps self-cleaning plates in the smokebox would give different exhaust parameters and dictate a different design if the 5AT were coal fired.

18.2. A double chimney arrangement appears optimum. It will give significantly larger blast nozzle tip area (the important criterion) than a single chimney, whilst a triple chimney gives only 2% increase in blast nozzle tip area in the case of a Lempor, which is insufficient to justify the extra complexity.

18.3. Of the two designs, both consisting essentially of blast nozzles, bell-mouth chimney gas entry section, mixing chamber and diffuser, and structurally identical, the most significant difference is in the chimney mixing chamber flow area, that according to *The Prediction of the Optimum Performance of Ejectors* being smaller than given by the Lempor theory. As a result the former gives a blast nozzle tip area some 3% larger than the Lempor (165.4 cm^2) and is thus preferred: it has been termed a 'modified Lempor'. The calculation of an exhaust being less than 100% precise (as, for example, complete accuracy is not possible for data such as total boiler vacuum used in exhaust calculations), it is usual to 'tune up' an exhaust system in service, and given this and the similarity of the Lempor and modified Lempor designs, a final decision on which to use would only be made after testing both.

18.3.1. Some criticism of the 5AT exhaust has been made in *The fire burns much better ...* by J.J.G. Koopmans. However it is believed this criticism is unfounded: it has been dealt with in the present writer's comments on this publication, which are to be obtained from the 5AT website at www.5at.co.uk.

18.4. Constructional details generally follow those already established in the Chinese Railway's modified QJ design. Of particular note is the proposed design of the Kordina junction of the exhaust passages from right and left hand cylinders under the blast pipe. It is proposed to take advantage of the entirely new cylinder block necessary for the 5AT to make this junction co-axial and thereby eliminate any tendency towards non-coaxial flow of exhaust steam relative to chimneys, which seems to occur with the usual junction of the exhaust passages from right and left cylinders. This is, however, a development that must first be investigated by laboratory testing on a suitable rig

18.5 Good exhaust lifting is vital for forward visibility, especially for high-speed operation. The appreciable height of the top of the chimneys above the smokebox (≈ 440 mm), the streamlined front of the locomotive, and the effective German-type exhaust deflectors, are all essential factors favouring good exhaust lifting. The first of these is only possible with a relatively small boiler and is a very important factor favouring the 5AT over large-boilered locomotives for high-speed work.

19. Mainframe. This is the 'backbone' of the locomotive and the correct functioning of all the parts attached to it depends on its integrity. The placing of the firebox between the frames requires the maximum width between the mainframe longitudinals: this mandates a plate frame with the longitudinals spaced as far apart as allowed by clearances with the coupled wheel tyres. This in turn means that the axlebox thrusts and spring rigging loads are applied eccentrically to the frame longitudinals, an unavoidable disadvantage leading to higher stress. In these respects the 5AT has a typical 4-6-0 mainframe. A welded plate frame is to be used. The advantages of a welded fabrication are similar to those given for the cast steel bed, i.e. (i) overall structure can be designed such that the heaviest sections are opposite the highest loads, (ii) great reduction in the number of lap joints, bolts and rivets, with a corresponding saving in weight (up to 2 tons compared to a comparable bar frame) and machining, and removal of many points of stress concentration (i.e. at fastening holes), (iii) elimination of cylinder – mainframe bolted assembly (which can work loose) by incorporating a welded-on cylinder block in the frames, (iv) can generally design for high lateral and vertical rigidity (although high lateral rigidity is not possible where the firebox lies between the frame legs this part of the frame is not so heavily loaded), and (v) damaged sections can be replaced relatively easily be welding in new ones. Frame details are to be based on the post-war development of welded plate frames in France and Germany, all new Deutsche Bundesbahn steam classes having such frames when steam development resumed in 1950. Numerically, the FDC's are mainly confined to quantifying the principal forces acting on the mainframe. In addition various recommendations for frame design details are given in the FDC's, the following being of note.

19.1. The thickness of the 5AT mainframe longitudinals should be between 32 & 38 mm depending on weight restrictions.

19.2. Tying the mainframe adequately to the boiler, which has a much higher mean section modulus than the frame, is important for frame stiffness. Apart from the smokebox saddle connection (a 'high saddle' type with integral steam passages from smokebox to steam chests is to be used) and firebox-frame expansion plate, a single slide bearing, centred on the locomotive's transverse centre line and mid-way between the rear of the smokebox saddle and the firebox expansion plate, is to be fitted to the 5AT, with suitable connections to the underside of the boiler barrel and top of the mainframe longitudinals. This transverse brace will be just ahead of the main driving axle, i.e. at the critical part of the frame with regard to stress.

19.3. To minimise the risk of cracking near the cylinder – mainframe welded joint, the large-radius cylinder – frame transition found to be necessary on American cast steel beds is to be replicated in the cylinder block as far as possible.

19.4. Special wedge arrangements are necessary between the driving and coupled wheel axleboxes and the welded-on frame hornblocks to ensure correct axle spacing in service with roller bearing axles and rods having very small bearing radial clearances, especially in view of the probability of some frame expansion due to heating level with the firebox. Consequently the following wedge arrangement is specified: (i) main driving axle, parallel guide behind, Franklin type wedge in front, (ii) leading coupled axle, manually adjusted wedge behind, Franklin type wedge in front, (iii) trailing coupled axle, Franklin type wedge behind and in front.

19.5. The front buffer beam and dragbox may be bolted to the mainframe by shear bolts as part of the required energy-absorbing crumple zone at the front of the locomotive.

20. Springs and spring rigging. A conventional steam locomotive suspension is specified for the 5AT. Since the steam era many changes have been made to railway vehicle suspensions. However the steam locomotive is different in a number of ways from other railway vehicles, e.g. (i) in its chassis layout, with coupled axles running in the mainframe rather than in bogies, (ii) in its sprung mass being subject to periodically applied vertical loads, e.g. due to connecting rod angularity, and (iii) in its comparatively high centre of gravity and resultant rolling tendency, which requires relatively hard springing, especially with inboard springs. Given this, and the necessity for the 5AT to perform reliably from the start with the minimum of 'tuning' in service, the type of spring and spring rigging that have proved themselves suitable for steam locomotives during millions of km of safe running, including at speeds > 160 km/h (100 mph) are the best approach, unless it can be proved otherwise.

20.1. The 5AT will have a 3-point suspension with compensated springing for the 3 coupled axles. The 3 suspension points are the bogie centre and the two centres of support of the coupled wheel springing, one at each side of the locomotive. The mass transfer to the bogie is at the bogie pivot, i.e. nominally at the locomotive's lateral centre, and the bogie therefore gives nominally zero lateral support to the locomotive. The coupled wheel springs on each side of the locomotive are not cross-compensated with each other, therefore they provide (all) the lateral support. This 3-point compensated suspension system is that almost universally used with steam traction: compared to the uncompensated springing in general use in the UK, to which it is inherently superior, it gives more constant axle loads with track irregularities and consequently better riding and traction qualities and reduced spring peak dynamic loads and stresses. It should be pointed out that the spring system for a 4-6-0 is a very simple example of the 3-point principle, which all compensated spring rigging, even on large multi-wheeled engines, seeks to achieve.

20.2. The FDC's specify the engine springs and spring rigging, as far as it is possible to do so at the FDC stage. *They are concerned only with the suitability of the spring system for the loads to be taken, and do not consider vehicle dynamics and associated ride quality and safety, which are outside the scope of the FDC's*. As individual component masses can only be known as a result of detail design, the FDC's proceed of necessity from the target axle loads, whereas at the detail design stage the axle loads are the *end product* of the individual component masses and centres of gravity.

20.3. Springs to the BR 5MT coupled wheel spring design are suitable for use on the driving and coupled wheels of the 5AT at the locomotive's target sprung loads.

20.4. Engine bogie springs are to be a combination of leaf + coil + disc springs in series (the last between the equalising beams and axleboxes). This arrangement was used on the DB 10 Class 4-6-2's of 1957 and represents the final development of suspensions for high-speed steam locomotive bogies. It gives generally beneficial soft springing, permissible for the bogie which does not have to counter rolling. The BR 5MT type of bogie spring, with from 21 to 24 leaves depending on which gives the best overall characteristic in combination with coil and disc springs, may be used for the leaf springs.

21. Brake gear. The BR 5MT has a simple (almost crude) brake system: a steam brake acts on one (the leading) side of the coupled wheels only through non-compensated brake rigging using a single central brake rod. The bogie is unbraked. It is reasonable to assume that this system would not be acceptable for continuous running at the 5AT's maximum operating speed of 180 km/h. However given the need for the 5AT to perform reliably from the start, the brake gear should be based on what has proved successful on steam traction in the past. The best known option is the Knorr-Bremse Kss air brake system, developed for high-speed steam-hauled services in Germany (maximum speed 175 km/h / 109 mph). The particular features of interest are (i) all wheels are braked, (ii) clasp brakes on the tyre treads are used throughout, except for the trailing wheels of the bogie where limited clearance allows braking at the front of the wheels only (note that the use of brake blocks acting on wheel tyre treads has proved successful on steam locomotives operating in excess of 160 km/h / 100 mph), (iii) special double brake blocks per brake shoe are used on the coupled wheels, (iv) brake force on the coupled wheels is a function of speed (a brake pressure regulator incorporating an axle-driven valve that is speed-sensitive is used, arranged so that the brake force on the coupled wheels is not allowed to exceed 75% of the wheel load for speeds under 60 km/h but allows up to 200% at higher speeds, assuming cast iron brake blocks) and (v) compensated brake rigging is used for the coupled wheel brakes with two brake pull rods. The tender is not covered by the FDC's, but the Kss brake system is also applicable here. It gives the same brake forces as a % of tender weight as for the engine coupled wheels, i.e. 75% for speeds < 60 km/h and up to 200% at speeds > 60 km/h. As the tender weight varies continuously as supplies (principally water) are consumed, the brake force can be regulated by the tender water level and therefore kept approximately a constant % of the actual tender weight as water is used up, a system in use from about 1935 on all German tenders. Modern measuring equipment, monitoring individual tender bogie or axle loads, can produce the same result as the Kss system but with greater accuracy because the diminution of fuel supply would also be accounted for.

21.1. Overall brake force as a % of locomotive weight. It is assumed that the tender brake force is kept proportional to the overall tender weight at any given time. The calculation is made for an assumed average tender weight of 610 kN giving an average locomotive weight of 1 395 kN. The engine axle loads are 20 metric tons for each of the driving and coupled axles, and 10 metric tons for each of the leading bogie axles. Overall brake force on the wheels as a % of average locomotive weight = 73.6% at speeds < 60 km/h (37.5 mph) and 181.0% at speeds > 60 km/h and up to 200 km/h (125 mph). This compares with a constant 60.7% for the BR 5MT, which was obviously considered adequate for the maximum speeds reached by this class in service, i.e. in the 80-90 mph range. Therefore without departing from a well-tried system proven by many years of use on steam traction, the braking of the 5AT will be superior to that of the BR5 at low speeds and overwhelmingly so at high speeds. Assuming cast iron brake blocks, the above wheel brake forces translate into brake forces at the rail of some 5.5% of locomotive weight up to 60 km/h and 13.6% from 60 – 200 km/h, the latter being approximately 40% higher than that of 125 mph modern traction (43, 67 and 91 classes).

22. Engine bogie and engine stability. The leading bogie performs 3 essential functions: (i) it supports that part of the engine weight in excess of that taken by the coupled wheels, (ii) it guides the engine round curves, thereby preventing excessive flange forces at the outside leading coupled wheel, and (iii) it stabilises the motion of the engine, i.e. acts to prevent 'hunting'. This last function is why the bogie and stability calculations must be considered together. As the engine structure can be considered to be cantilevered forward from the generally quite long coupled ('rigid') wheelbase, it overhangs curves, to

negotiate which the bogie must be provided with controlled side play relative to the mainframe: it is the relationship between this lateral movement and the force acting to align the bogie and mainframe that provides the guiding action, item (ii) above. The bogie design is in principle to be according to the last development of cast steel bogies for steam locomotives in the USA, with geared roller centring which can give any desired lateral force – displacement relationship. Whilst bogie design has generally moved on since the steam era, that specifically for steam locomotives has not, and in view of the special functioning of a steam locomotive bogie it cannot be expected that modern bogie design can be simply transplanted onto a steam locomotive and then work as intended. For acceptance reasons the following five bogie parameters are made the same as those of the BR 5MT:

22.1. Bogie wheelbase = 1905 mm.

22.2. Distance from bogie pivot centre to leading coupled axle centre = 2629 mm.

22.3. Bogie wheel diameter = 914 mm.

22.4. Maximum bogie – mainframe lateral movement level with the pivot = ± 101.6 mm.

22.5. Minimum radius of curvature for locomotive without gauge widening = 120.7 m (90.5 m dead slow).

22.6. Stability on tangent track. This was estimated by the method given in *On the Stability of Running of Locomotives* by F. W. Carter (Proceedings of the Royal Society, 1928) which is one of the forerunners of today's computerised methods of predicting stability. Its accuracy is conditioned by the approximate nature of some of the input data, and it can be considered to give only a first estimate of the bogie lateral force – displacement relationship necessary for stable running, to be refined at the detail design stage by computer simulation. Three cases were considered:

22.6.1. With axle loads as given in [7.1.3], i.e. bogie axle load = 10 metric tons.

22.6.2. With bogie axle load increased from 10 to 15 metric tons by ballasting the main truck (= all mass carried by mainframe) + a small increase in bogie mass.

22.6.3. With bogie axle load increased from 10 to 15 metric tons by ballasting the bogie only.

22.7. The analysis gave the following maximum speeds for stable running: (i) bogie according to [22.6.1] = 163 km/h with a bogie – mainframe lateral force of 1610 N/mm, (ii) bogie according to [22.6.2] = 155 km/h with a bogie – mainframe lateral force of 1970 N/mm, (iii) bogie according to [22.6.3] = 168 km/h with a bogie – mainframe lateral force of 2170 N/mm. The 5 km/h increase in maximum speed for a ballasted compared to an unballasted bogie is not sufficient to warrant the 10 tons extra mass required, and having 20 tons total on the 2 bogie axles, which conforms to accepted practice, is sufficient from the stability viewpoint.

22.8. The elementary stability analysis (which would not have been carried out on new designs during the steam era, design for stability being empirical and subject to revision depending on service experience) is very approximate. For example all the bogie centring forces given above would give excessive bogie flange forces at high bogie displacement, giving a high risk of flange climbing on small radius curves, therefore their values have practical significance only for small bogie displacements such as occur for tangent track stability, the subject of the calculations. With roller centring the bogie – mainframe lateral force would not be proportional to bogie displacement as assumed above, but would give a fairly high force against initial bogie lateral movement from its central position followed by a constant or slightly increasing force for further displacement. Experience is that such a bogie – mainframe lateral force – displacement relationship improves stability. (Note: with roller centring this force could even be made to decrease for high displacements to prevent excessive bogie flange forces on very small radius curves, which would always be taken at low speed.) Also no allowance is made in the analysis for the steadying effect of the engine – tender connection. Not only is this connection 'tight' longitudinally (see par. 14.2.) but it can incorporate inter-vehicle dampers as used on modern high-speed trains, for example to counter yaw motion, which may be expected to materially improve the yaw stability of the main engine truck and as a consequence reduce any bogie – mainframe lateral displacements on straight track. Therefore the initial stability analysis, which predicts stability only up to 90.6% of the proposed maximum continuous operating speed (180 km/h) and 81.5% of the design speed (200 km/h) almost certainly errs on the 'safe' side, predicting a lower value of stable speed than is to be expected in practice. This is ultimately to be determined by computer simulation of the riding and curving qualities of the entire locomotive, from which the optimum bogie – mainframe lateral force – displacement relationship will be found.

22.9. Miscellaneous stability points: it may be thought that a 4-6-2 rides steadier than a 4-6-0, and a 3 or 4-cylinder locomotive than a 2-cylinder one. Neither is true *per se* (although examples of bad-riding 2-cylinder 4-6-0's certainly existed). A 4-6-0 is in principle more conducive to stable running than a locomotive with a trailing truck, and vehicular stability is not so much a matter of the nosing couple due to unbalanced reciprocating masses (which may be higher with 2 cylinders) but of the ability of the chassis to ride steadily under the influence of all disturbing forces.

23. Auxiliaries and proprietary equipment. The principal auxiliaries are specified in the FDC's, which also determine important data relating to them, such as transverse clearances for the air compressor (German

Tolkien high-speed two-stage type) mounted under the left-hand running board, and boiler feed pump (German KP4-250 compound reciprocating type) mounted under the right-hand running board. Although ideally the design of all such auxiliaries should be uprated to current standards (just as the 5AT is an uprated BR5), the limited design resources for the 5AT and limited extent to which auxiliaries affect the overall performance of the locomotive (assuming of course that they operate with acceptable reliability) mean it is better at this stage to specify good proprietary equipment that can be purchased complete (either new or reconditioned), rather than design afresh.

24. Miscellaneous items.

24.1. Adhesion. Wheel rim tractive effort diagrams (i.e. tractive effort variation over a full coupled wheel revolution) are found at (i) starting, (ii) maximum equivalent drawbar power (113 km/h, 71 mph) and (iii) a hypothetical overload condition combining the indicator diagram of par. [7.12] with maximum design speed (200 km/h, 125 mph). In each case these show tractive effort peaks for part of a wheel revolution which are above what adhesion could transmit on wet rails based on nominal adhesive weight (marginally only for case (ii) and then completely within wet rail adhesion if the 'dynamic' adhesive weight is taken). All peaks are below the adhesion limit with dry rails. Given that (i) is for full throttle + full forward gear at starting, which rarely if ever occur together in passenger service, and (iii) is hypothetical, the slight transgression of the adhesion limit on wet rails is not considered significant, especially as the foot pedal operated air sanding is to be 'state of the art' and an independent engine brake for the coupled wheels may be fitted, which can be used to control slipping. Nevertheless the power is about as high as may be transmitted by a 2-cylinder engine with 60 metric tons adhesive mass. A 4-8-0 would be better from an adhesion viewpoint, but being an entirely new chassis design it would pose considerably more design workload and difficulty with acceptance by Network Rail, and its extra length would reduce tender size and capacity for the same overall locomotive length.

24.2. Driving and coupled wheels should ideally be of the Scullin type, which apart from its conventional advantages virtually eliminates the 'fan effect' of spoked coupled wheels, an unquantified air drag phenomenon that may be significant at high speed. However this type was quasi-experimental and in case of resistance to it by acceptance bodies, and also to reduce design work, the Bulleid BFB type, which to some extent accomplishes the same function, may be preferred.

24.3. Lubrication equipment (such as very extensive mechanical lubrication, using cylinder and machine oil as appropriate, each lubricator being supplied from its own large-capacity oil reservoir) shall be arranged such that lubrication tasks will be required only from servicing staff, footplate staff having no lubrication duties.

24.4. The cab and controls, whilst being essentially to traditional steam locomotive design (which is what volunteer steam locomotive crews want), shall wherever possible be improved within this format in such matters as ergonomic arrangement of controls, vision, comfort, etc., and shall comply with all mandatory health and safety requirements.

24.5. Electrical equipment. A steam turbine driven electrical generator or generators, charging a battery system if necessary, shall be fitted, the whole system being of sufficient capacity to power all electrical equipment on the locomotive. Lights for both directions of running shall comply with the relevant Network Rail *Railway Group Standard.*

24.6. The front of the locomotive, i.e. front frame extensions ahead of the cylinders, streamlined front casings of the running board and smokebox, and the exhaust deflectors, shall, if so required, be designed for energy-absorbing in case of severe impact.

25. Summary.

25.1 The FDC's have established the engineering feasibility of the 5AT for its proposed performance.

25.2 They have enabled the overall thermal efficiency based on cylinder indicated output at maximum drawbar power (par. [7.7.3]) to be estimated as 14.3%, and the corresponding drawbar thermal efficiency as 11.4%. This latter figure is heavily influenced by the large tender used, but is nevertheless very high compared to former steam practice, especially in view of the high specific power of some 30 indicated kW per ton of engine mass. By comparison, the BR 5MT gave 6.8% drawbar thermal efficiency when generating a maximum of 17.0 indicated kW per ton of engine mass. The absolute maximum drawbar thermal efficiency that may be achieved by the 5AT is tentatively estimated as about 14%.

25.3 The FDC's have also allowed the 5AT's potential range (i.e. to empty tank) to be estimated as approximately 615 km (385 miles) on a tenderful of water under representative average main line service conditions (the corresponding range based on fuel supply ≈ 940 km, 590 miles). This is beyond what it would be likely to run in a single trip (i.e. outward or return) in excursion service, indeed many round trips could be comfortably achieved on one tenderful of water. Yet further extended range could be achieved with an auxiliary tender.

25.4 This paper is but a brief and selective survey of the engineering work to date. For a proper understanding of the 5AT it must be read in conjunction with the Fundamental Design Calculations and the Engineering Specification as given in the Business Plan.

25.5 The FDC's have provided a foundation for the 5AT project. The work required to build on this to the point at which the 5AT is actually running in main line excursion service is multi-faceted and very formidable. From the engineering aspect, the main challenges are detail design and satisfying Network Rail's acceptance requirements, these two being interlinked. What this would need is a dedicated team of design engineers, comprising specialists in each field of engineering relevant to the design and acceptance process working together with an engineer or engineers who have intimate knowledge of steam locomotive design.

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