WARDALE ENGINEERING & ASSOCIATES

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CLASS 5AT 4-6-0: FUNDAMENTAL DESIGN CALCULATIONS

1. GENERAL CALCULATIONS.

1.3. PRELIMINARY BASIC CALCULATIONS.

Notes.

1. These calculations refer to a specific performance level, defined by item nos. [1] and [2]. ***Any individual performance figures given in the calculations do not necessarily give the maximum which will be achieved.***
2. The SI system is mostly used, with Imperial units given for some items for the convenience of those not familiar with SI units. Unless otherwise stated "ton" refers to metric ton of 1000 kg. $N*m^3 = m^3$ at NTP.
3. Numbers in square brackets [ ] in column 2 refer to calculation item numbers in the Fundamental Design Calculations (FDC's): firstly the number identifying the calculations concerned, followed by the item number within those calculations, given in round brackets ( ), e.g. [1.1.(16)] refers to calculations 1.1. item no. (16). Where only a single number is given within square brackets, it refers to an item number within these calculations.
4. ***To save space, unit conversion factors for numerical consistency, where used, are not shown in the calculations. Any apparent small numerical discrepancies are due to giving data to limited places of decimals but to taking the full figure for any calculations involving that data.***
5. References are shown in superscript square brackets [ ] and are given in full at the end of the calculations.
6. Fundamental data is in **bold** type.

Item No.	Item	Unit	Amount
1	Maximum sustainable drawbar power at constant speed on level tangent track, trailing a high capacity tender [Calculations 1.1. Fig. 1.1.1]:	kW hp	**1 890** 2 535
2	Speed at the above power:	km/h mph m/s	**113** 71 31,4
3	Equivalent drawbar tractive effort at [1] and [2]	kN	**60,2**
4	Maximum axle load (same as BR Class 5MT 4-6-0)	ton	**20,0**
5	Preliminary estimate of Class 5AT axle loads, at full supplies: leading bogie (2 axles combined) (minimum value given: may be increased if greater centring force required for lateral stability reasons) leading coupled axle driving axle trailing coupled axle tender axles (each of 4 axles in 2 bogies)	 ton ton ton ton ton	 **20,0** **20,0** **20,0** **20,0** **20,0**
6	Total mass of engine (full boiler): (BR Class 5MT = 77,2 tons)	ton	**80,0**
7	Total mass of tender, full supplies: (Note: the large high-capacity tender is an operational requirement due to the absence of convenient watering facilities: heaviest former British tender = A4 type = 66 tons gross mass.)	ton	**80,0**
8	Total mass of engine and tender, full supplies	ton	**160,0**
9	Adhesive mass	ton	**60,0**
10	Approximate gross : tare mass ratio for rectangular section tender of monocoque construction (c.f. best Bulleid tender = 2,67): Note: minimizing tender mass for a given quantity of supplies is of particular importance for high-speed operation, and it would be hoped to increase the gross : tare mass ratio beyond the conservative figure given.	-	3,0
11	Tender tare mass = [7] ÷ [10]	ton	26,7
12	Total supplies (fuel + water) = [7] – [11] (highest former British figure (largest MN tender) = 32,3 tons)	ton	53,3
13	Average supplies during operation as a fraction of total supplies, assumed	-	0,67
14	Average tender supplies = [12] x [13]	ton	35,5
15	Average tender gross mass in service = [11] + [14]	ton	62,2
16	Average mass of locomotive in service = [6] + [15]: (note: further performance figures are calculated on the basis of this average mass).	ton	142,2

Item No.	Item	Unit	Amount
17	Required sustainable cylinder (indicated) power at [2], from [calculations 1.1. (Fig. 1.1.1)]:	kW hp	**2 380** 3 192
18	Cylinder (indicated) tractive effort at [2] and [17] = [17] ÷ [2]	kN	**75,8**
19	Maximum cylinder (indicated) power from [1.1. (Fig. 1.1.1)]:	kW hp	**2 580** 3 460
20	Maximum indicated power per ton of engine mass = [19] ÷ [6] (BR Class 5MT max. indicated power/ton of engine mass = 17,0 kW/ton)	kW/ton	32,25
21	Indicated power per ton of engine mass for other high power locos, for comparison: SAR 26 Class No. 3450 (peak of power curve) SNCF 240P Class Therefore the Class 5AT figure is realistic, given its superior technology.	kW/ton kW/ton	30,5 28,7
22	Coupled wheel diameter (same as BR Class 5MT 4-6-0)	mm in	**1 880** 74
23	Coupled wheel rotational speed at [2] = [2] ÷ (π x [22])	Hz	5,3
24	Indicated tractive effort per unit adhesive mass at [23] = [18] ÷ [9]	kN/ton	1,3
25	Max. indicated t. e. per unit adhesive mass at [23] for SAR 26 Class No. 3450: (›› Class 5AT figure, which is therefore considered to be realistic).	kN/ton	1,9
26	Nominal maximum continuous operating speed in mph is taken as the '1,5 x diameter' speed (AAR std. for motion design), = 1,5 x [22(inches)] = This is rounded up to:	mph km/h km/h	111 178 **180**
27	The locomotive will be tested at 10% over the maximum operating speed, hence maximum design speed = 1,10 x [26] This is rounded up to: All relevant detail design work shall be based on this speed	km/h km/h mph	198 **200** **125**
28	Coupled wheel rotational speed at [26] = [26] ÷ (π x [22])	Hz	8,5
29	Boiler pressure: (the given figure is the normal maximum working (gauge) pressure: the boiler may be designed for and the safety valves set to a slightly higher figure for ease of keeping the working pressure in service without the safety valves lifting). (cf. A T & S Fe 2-10-4 b. p. = 310 psi)	kPa psi	**2 100** 305
30	Engine unit. The preferred choice of engine, considering ***all*** relevant parameters, is 2-cyl. simple. The calculations are made for a 2-cyl. simple and show that the desired performance can be realized with this simplest type of engine, having perhaps a lower level of cylinder performance than more complex and expensive multi-cylinder types. (Note: mass of reciprocating parts per side not to exceed 250 kg.) Hence no. of cylinders:	-	**2**
31	Piston stroke (made long for optimum cylinder efficiency)	mm in	**800** 31,5
32	Mean piston speed at [2] = [23] x [31] x 2	m/s ft/min	8,5 1 674
33	Mean piston speed at [26] = [28] x [31] x 2	m/s ft/min	13,5 2 666
34	Various comparisons of mean piston speed: SAR 26 Class no. 3450 at speed for maximum indicated power (122 km/h by differentiating equation [1.1.(11)]: NYC 'Niagara' 4-8-4 at 160 km/h: N & W J Class 4-8-4 at reported maximum speed of 176 km/h LNER A4 Class at 202 km/h BR 9F Class at 144 km/h Given the 5AT 's superior front end compared to these locomotives, these figures are considered to confirm the acceptability of items [32] & [33].	m/s m/s m/s m/s m/s	10,2 11,5 14,2 11,7 11,9
35	Sample starting coefficients of adhesion for 2 cylinder and 4 cylinder (opposed crank) 6-coupled tender engines: BR Standard Class 5MT: All BR 2-cylinder standard classes (average value for seven classes): Representative pre-nationalization British locos: Representative German standard locos: Representative modern American-built locos:	- - - - -	0,20 0,21 0,24 0,30 0,23

Item No.	Item	Unit	Amount
36	From [35], a realistic starting coefficient of adhesion for the Class 5AT, all possible adhesion improvements being incorporated, is (see [49]):	-	0,25
37	For a 2-cylinder engine, typical ratio of peak : mean tractive effort[23]	-	1,25
38	Peak coefficient of adhesion required to prevent 'quarter slip' = [36] x [37]	-	0,31
39	The maximum available starting coefficient of adhesion may be taken as [Calculations 1.1. Fig. 1.1.2]: Dry rail: Wet rail: A figure for sanded wet rail is deduced as:[24] As the figures for dry rail and sanded wet rail are › [38] the Class 5AT's full starting tractive effort should be useable with minimal slipping, provided good (air) sanding is fitted.	 - - -	 0,34 0,26 0,375
40	Nominal wheel rim tractive effort based on adhesion = [9] x [36]	ton kN	15,0 147
41	With the usual notation, the tractive effort for a 2-cyl. locomotive is: T.E. = $(k \times P \times n \times (d^2 - d_1^2) \times s) \div (2 \times D)$ T.E. = [40] = 147 kN, P = [29] = 2 100 kPa, n = [30] = 2, s = [31] = 800 mm, D = [22] = 1 880 mm		
42	The factor k allows for less than 100% cut off being available and for frictional losses from the pistons to coupled wheels. For the Class 5AT (fully roller bearing equipped and with state of the art tribological design and lubrication) the starting transmission efficiency (= wheel rim work / indicated cylinder work) is taken as:[1]	-	0,93
43	The ratio of mean effective pressure (m.e.p.) : boiler pressure at starting depends largely on the maximum cut-off. For easy starting of the 2-cyl. Class 5AT this is made (cf. BR 5MT = 78%)	%	**75**
44	At a cut off = [43] the ratio of m.e.p. : boiler pressure at starting is deduced from SAR 25NC and 26 Class starting indicator diagrams made at 80% and 65% cut off respectively:[2]	-	0,90
45	Factor k in equation [41] = [42] x [44] (generally accepted value with 80% cut-off = 0,85)	-	0,84
46	d_1 = piston rod and piston tail rod outside diameter (BR 5MT = 3½" = 88,9 mm)	mm	90
47	Substituting known data into equation [41], cylinder diameter, d: This is rounded down to:	mm mm in	452 **450** 17,7
48	Based on [47] nominal wheel rim tractive effort from equation [41]: (BR Class 5MT = 116 kN). See also item [169]	kN lbf	**146** 32 830
49	Based on [48] nominal coefficient of adhesion = [48] ÷ [9]	-	**0,248**
50	Net piston face area (front and back) = $\pi/4 \times ([47]^2 - [46]^2)$	m^2	0,153
51	Nominal maximum piston thrust, front and back = [29] x [50] (BR Class 5MT = 283,7 kN)	kN lbf	**320,6** 72 098
52	Stroke : diameter ratio = [31] ÷ [47]: (high for good cylinder efficiency) (BR Class 5MT = 1,47)	-	1,78
53	Starting indicated tractive effort = [48] ÷ [42]	kN	157
54	Ratio of indicated tractive effort at [1] and [2] : starting indicated tractive effort = [18] ÷ [53]	-	0,48
55	Corresponding ratio at maximum power for SAR 26 Class No. 3450 at [23]: Actual figure from test data: Estimated figure if 3450 had a maximum cut-off = [43]: As these figures are › [54] the Class 5AT figure is considered to be realistic.	 - -	 0,63 0,59
56	Approximate initial estimation of cut-off required at [17] and [2] is made by deduction from data on SAR 26 Class No. 3450. If maximum cut-off of 3450 = [43] its starting indicated t. e. would have been approximately:	kN	245,0
57	[54] x [56]	kN	117,6
58	Speed of 3450 at [23] with coupled wheel tyre diameter = [1.1. (3)]:	km/h	89,9
59	Indicated power at [57] and [58] = [57] x [58]	kW	2 937

Item No.	Item	Unit	Amount
60	At [58] and [59] cut-off is:[3] See item [74] for a more accurate assessment of the required cut-off on the Class 5AT at [2] and [17].	%	25
61	m.e.p. at [17] and [2] = [17] ÷ ([50] x [31] x 2 x [30] x [23]):	kPa	917
62	[61] ÷ [29]	-	0,44
63	Tentative diameter of piston valves: This dimension may depend on clearance with the moving structure gauge, and the requirement for minimum cylinder clearance volume (item [67]) will probably necessitate the use of two piston valves per cylinder (as for the inside cylinder of the SNCF 242A-1 4-8-4), each ≈ **175** mm diameter.	mm in	**350** 13,8
64	Tentative piston valve steam lap	mm in	**65** 2,56
65	Ratio of valve diameter x lap : cylinder diameter^2 = [63] x [64] ÷ [47]^2 (cf.: value for SAR 26 Class No. 3450 = 0,046 for BR Class 5MT = 0,051)	-	0,112
66	Tentative piston valve exhaust lap	mm	**10**
67	Target *maximum* cylinder clearance volume as % of piston swept volume: Single valves: Double valves:	 % %	 **9** **8**
68	Indicated work done per piston stroke at [17] and [2] = [17] ÷ (4 x [23])	kJ	112,3
69	Piston swept volume per cylinder end = [31] x [50]	m^3	0,122
70	Estimated boiler - steam chest pressure drop at [17] and [2]: (4,8% of rated boiler pressure)	kPa	100
71	Estimated steam chest (gauge) pressure at [17], [2] and [29] = [29] – [70]	kPa	2 000
72	Estimated cylinder (gauge) back pressure at [17] and [2]	kPa	50
73	Required inlet steam temperature at steam chest at [17] and [2]	^{o}C	**450**
74	From the estimated indicator diagram at [17] and [2] (see items [225] – [244] and Fig. 1.3.1) the cut-off required to give work per stroke = [68] at a speed = [2] is (≈ item [60]): This is a good figure, well in the zone of high cylinder efficiency, and confirms the suitability of the cylinder dimensions for the required power output at speed = [2].	%	**26**
75	Adiabatic steam flow to the cylinders per stroke (see items [249] – [253])	kg	0,204
76	Adiabatic heat drop of steam in cylinders = [68] ÷ [75]	kJ/kg	550
77	Inlet steam enthalpy at [71] and [73] from h – s chart	kJ/kg	3 356
78	Exhaust steam enthalpy = [77] – [76]	kJ/kg	2 806
79	Exhaust steam temperature at [72] and [78] from h – s chart	^{o}C	167
80	To allow for heat transfer to the cylinder walls during steam admission (i.e. add the ‘missing quantity’) item [75] is increased by 5% to: The low value of the ‘missing quantity’ is a result of using all practical features to reduce it, such as very high superheat, long stroke : diameter ratio, optimum cylinder insulation, high rotational speed at normal train speed, low clearance volume, special engine component design, etc.	kg	0,214
81	Cylinder steam flow = (4 x [23]) x [80] This is rounded up to:	kg/s kg/h lb/h kg/h	4,55 16 393 36 146 **16 400**
82	Actual specific work done by steam in cylinders = [68] ÷ [80]	kJ/kg	525
83	Isentropic heat drop from [71] and [73] to [72] from h – s chart	kJ/kg	650
84	Cylinder isentropic efficiency at [2] and [74] = [82] ÷ [83]: This is not the maximum figure, which will occur at shorter cut-off than item [74]. (cf. BR 8P Class 4-6-2 No. 71000 = 86% at minimum s.s.c.)	%	81
85	Indicated s.s.c. (based on cylinder steam flow) at [17], [2] & [74] = 1 / [82] This very low figure for such a high power is a consequence of the high-efficiency front end and high superheat (cf. minimum indicated s.s.c. are: BR 8P Class 3-cyl. simple 4-6-2 No. 71000 = 12,2 lb/hp-h, SNCF 141P Class 4-cyl. compound 2-8-2 = 11,2 lb/hp-h).	kg/MJ lb/hp-h	1,90 11,2

Item No.	Item	Unit	Amount
86	Leakage steam upstream of the cylinders: experience with SAR 26 Class no. 3450[4] gives total leakage past the piston valve rings = 0,5% of [81]	kg/h	82
87	Superheated steam flow = [81] + [86]	kg/h	**16 482**
88	In addition to the cylinder steam, steam is (typically) required for some/all of the following when the locomotive is under power (* shows that exhaust steam from these auxiliaries may be piped back to the tender tank, totalling ≈ 40% of the total normal auxiliary steam consumption with oil firing): (a) Air compressor* (for brakes, air sanding, air-controlled auxiliaries) or vacuum brake ejector. (b) Mechanical stoker motor* and distributing jets (coal firing) or oil heating* (if required) and atomising (oil firing). (c) Boiler feed pump(s).* (d) Turbo generator.* (e) Cylinder oil heating and (optional) atomising. (f) Cab heating.* (g) Steam heating of coaching stock. (h) Whistle. (+ blower and drifting steam when not under power) For the purpose of these calculations it is assumed the locomotive is fired with gas oil (no oil heating required) and works electrically heated/air con. stock: the sum of this auxiliary steam as a percentage of [87] is taken as:	%	4
89	Auxiliary steam at [1] and [2] = [87] x [88]:	kg/h	**659**
90	Total steam generated by the boiler at [1] and [2] = [87] + [89]: (See item [99] for equivalent evaporation.) This is rounded up to:	kg/h lb/h kg/h	17 141 37 796 **17 150**
91	The saturated/superheated fractions of the auxiliary steam will be decided at the detail design stage. For the present calculation purposes all auxiliary steam is assumed to be dry saturated at pressure = [29]. Its enthalpy is:	kJ/kg	2 801
92	Enthalpy of superheated steam leaving superheater ≈ [77]	kJ/kg	3 356
93	Saturation temperature corresponding to [72] (note: exhaust steam still has some superheat, see item [79]).	^{o}C	112
94	Feedwater temperature at inlet to boiler, after preheating in a surface type exhaust steam feedwater heater (with average (small) fouling deposits on h.t. surfaces): (Note: a considerably higher temperature could be achieved by passing the feedwater through an economizer formed from the front section of the boiler barrel, this being an optional refinement at this stage).	^{o}C	**105**
95	Feedwater enthalpy at [94]	kJ/kg	440
96	Total heat transferred to the steam leaving the boiler at [1] and [2] = [87] x ([92] – [95]) + [89] x ([91] – [95])	GJ/h	49,6
97	Heat given to cylinder steam by fuel = [77] – [95]	kJ/kg	2 916
98	Cylinder thermal efficiency based on [97] = [82] ÷ [97]	%	18,0
99	Equivalent evaporation at [1] and [2] = [96] ÷ 2 256,7 kJ/kg	kg/h	21 980
100	The combustion air is to be preheated by exhaust steam: required air temperature ≈ (see item [186])	^{o}C	**100**
101	Probable boiler absorption efficiency at [90] without combustion air pre-heating[5]	%	80
102	Due to item [100] and other factors (low excess air, optimum tube bundle/superheater design, good insulation, (preheater at front of boiler barrel)) boiler absorption efficiency may be increased to: (see item [186])	%	85
103	Heat release rate in firebox at [1] and [2] = [96] ÷ [102]	GJ/h	58,4
104	Firebox volume (same as BR Class 5MT for the purpose of this calculation)	m^3	4,8
105	Heat release rate per unit firebox volume = [103] ÷ [104]: cf. BR Class 5MT at maximum evaporation ≈ 9,4 GJ/m^3-h LMR Class 2 2-6-0 at maximum evaporation ≈ 11,0 GJ/m^3-h SAR 3450 at maximum measured firing rate = 12,6 GJ/m^3-h From Chapelon[25] it is deduced that a high oil burning rate = 15 GJ/m^3-h.	GJ/m^3-h	12,2

Item No.	Item	Unit	Amount
106	Combustion efficiency with "state of the art" oil firing: (99,5% at relatively low value of heat release rate per unit firebox volume is claimed for the Swiss 'Sonvico' system.)	%	95
107	Heat in fuel fired = [103] ÷ [106]	GJ/h	61,5
108	Boiler efficiency at [90] = [96] ÷ [107] = [102] x [106]: This high efficiency at such a high boiler load is primarily due to the high combustion efficiency possible with modern oil firing technology.	%	81
109	Fuel: for various technical, practical and environmental reasons, oil firing is preferred (see item [192] etc. for coal firing). Its ready availability makes diesel fuel / gas oil the most practical fuel, of lower calorific value[28]:	MJ/kg kcal/kg	42,9 10 240
110	Firing rate at [1] and [2] = [107] ÷ [109]:	kg/h	**1 434**
111	Assuming burner is of the high pressure atomising type, (superheated) atomising steam required per unit of fuel fired: From Kempe's Engineers Year-book, 1985, p. F2-35: Claimed figure for 'Sonvico' system: Assuming average of two figures, it is:	kg/kg " "	0,3 0,1 0,2
112	Atomising steam flow = [110] x [111]: Note: this is 43% of the estimated auxiliary steam production item [89].	kg/h	287
113	Overall thermal efficiency of locomotive referred to the indicated output at maximum drawbar power = [17] ÷ [107]: (Maximum figure for SAR 26 Class No. 3450 = 13,1%[6])	%	13,9
114	Overall thermal efficiency of locomotive referred to maximum drawbar power = [1] ÷ [107]: This is a very high figure when generating a specific power as high as item [20] and trailing a large tender of the same nominal weight as the engine itself. By comparison with the best level achieved with simple expansion locomotives in former times, the BR Class 7MT 4-6-2's at maximum evaporation, generating 17,3 indicated kW per ton of engine weight, gave a drawbar thermal efficiency of 7,7%, and the BR Class 5MT, at its maximum of 17,0 indicated kW per ton of engine weight, gave 6,8%.	%	11,1
115	The feedwater heater heat balance is, with the usual notation: m_s x Δh_s = m_w x Δh_w : presuming steam leaves heater as saturated water at pressure = [72], condensate enthalpy is:	kJ/kg	467
116	Average tender water temperature, assumed (with an allowance for warming by the condensate and auxiliary exhausts fed back to the tender): (This temperature will be higher if the tender tank is partitioned to create a 'hot well'.)	^{o}C	20
117	Tender water enthalpy at [116]	kJ/kg	83,9
118	Substituting known data into equation [115]: m_s x ([78] – [115]) = [90] x ([95] – [117]) from which m_s =	kg/h	2 611
119	Fraction of cylinder exhaust steam going to feedwater heater = [118] ÷ [81]	%	16,0
120	Heat balance for the combustion air preheater is: m_s x ([78] – [115]) = m_a x Δh_a : assume average ambient air temperature:	^{o}C	15
121	Temperature rise of the air passing through the heater = [100] – [120]	deg. C	85
122	Specific heat at constant pressure (c_p) for air	kJ/kg deg.K	1,005
123	Δh_a = [121] x [122]	kJ/kg	85,4
124	Stoichiometric air:fuel ratio by weight, diesel fuel/gas oil (see item [255])	kg/kg	14,5 : 1
125	Excess air coefficient at [90] and [110], assumed: Note: this is a 'safe' value, and the combustion equipment must be designed to allow adequately complete combustion with the minimum of excess air.	-	1,3
126	Combustion air supply, based on fuel fired = [110] x [124] x [125]	kg/h	27 031
127	Substituting known data into equation [120]: m_s x ([78] – [115]) = [126] x [123] from which m_s =	kg/h	987
128	Fraction of cylinder exhaust steam going to combustion air preheater = [127] ÷ [81]	%	6,0
129	Total exhaust steam to the feedwater & combustion air heaters = [118] + [127]	kg/h	3 598

Item No.	Item	Unit	Amount
130	Total exhaust steam to the feedwater & combustion air heaters as a fraction of the cylinder steam flow = [129] ÷ [81] = [119] + [128]	%	22,0
131	% of cylinder steam flow going to blast nozzles = 100- [130]	%	78,0
132	Steam to blast nozzles = [81] – [129] = [81] x [131]	kg/h	12 802
133	Ratio of combustion gas flow : blast nozzle steam flow = ([110] + [112] + [126]) ÷ [132]	kg/kg	2,25 : 1
134	Total condensate piped to the tender from feedwater heater and auxiliaries as a fraction of the total evaporation ≈ ([118] + 0,4 x [89]) ÷ [90]	%	16,8
135	For every unit of tender water evaporated in the boiler, the amount of raw water is (100 – [134])	%	83,2
136	Split of supplies: item [12] can be split into fuel and water in any ratio to suit operating conditions, but generally it is now at least as easy to take oil fuel as to take water. In UK steam times, maximum coal supply for the longest duties was (for LMR and BR Standard Class 8 4-6-2's):	Imp. ton m. ton	10 10,2
137	Typical lcv of good former British locomotive coal	MJ/kg	32
138	Tender energy capacity = [136] x [137]	GJ	326
139	Corresponding fuel supply of Class 5AT = [138] ÷ [109]: This is rounded down to:	ton ton	7,6 **7**
140	Autonomy at [1] and [2] based on fuel capacity = [139] ÷ [110]:	h	4,88
141	Range at [1] and [2] based on fuel capacity = [2] x [140]: This is well beyond the distance that the loco would be expected to cover at constant maximum drawbar power without refuelling, therefore giving a high fuel capacity safety margin for the expected duty (see also item [151]).	km mile	552 345
142	Allowable capacity of tender water tank = [12] – [139]	ton or m^3	**46,3**
143	Autonomy at [1] and [2] based on water capacity = [142] ÷ ([90] x [135])	h	3,24
144	Range at [1] and [2] based on water capacity = [2] x [143] This is well beyond the distance that the loco would be expected to cover at constant maximum drawbar power, so that in practice the range between water replenishments would normally be greater than as given (an exception is if long periods of high power were required at lower speed going upgrade).	km mile	367 230
145	Increase in range based on water capacity due to returning auxiliary exhausts & condensate from feedwater heater to tender = (100 ÷ [135]) - 1	%	20,2
146	Representative load factor (defined as ratio of (distance) average cylinder power : full rated cylinder power) in normal service	-	0,5
147	Specific fuel and water consumptions will be fairly flat functions of power under typical charter train operating conditions, except for relatively high values during periods of acceleration. Fuel and water consumption rates at load factor = [146], as fractions of the full load consumptions, are therefore conservatively estimated as:	-	0,6
148	Under representative average service conditions, autonomy based on fuel capacity = [140] ÷ [147]	h	8,13
149	Under representative average service conditions, autonomy based on water capacity = [143] ÷ [147]	h	5,40
150	With a maximum operating speed = [26] the average train speed can conservatively be assumed = [2]:	km/h mph	113 71
151	Under representative average service conditions, range based on fuel capacity = [148] x [150]	km mile	919 **(920)** **575**
152	Under representative average service conditions, range based on water capacity = [149] x [150]. If extra range is required, a simple water tank car could be added behind the tender and/or part of any support vehicle (if required for providing electrical power for train heating or air con. etc.) could be fitted with an auxiliary water tank.	km mile	**610** **380**
153	Relative density of diesel fuel / gas oil	-	0,83
154	Volume of tender fuel tank (for gas oil) = [139] ÷ [153]	m^3	8,4
155	Approximate cross sectional area of tender fuel tank	m^2	1,4

Item No.	Item	Unit	Amount
156	Approximate length of tender fuel tank = [154] ÷ [155]	m	6,0
157	Approximate volume of tender water tank well section between bogies	m^3	5,5
158	Approximate cross sectional area of tender water tank, excluding well section	m^2	4,8
159	Approximate length of tender water tank = ([142] – [157]) ÷ [158] To allow for volume occupied by internal tank bulkheads, etc., this is increased to:	m m	8,5 9,0
160	Approximate overall length of engine and tender over buffers: (cf. LNER A1 Class = 22,2 m, LMS 'Coronation' Class = 22,5 m, LMS 'Princess Royal' Class = 22,7 m)	m ft	**22,1** 72,5
161	Ratio of length of engine : length of tender (engine length same as for BR Class 5MT)	-	1,26 : 1
162	Approximate overall wheelbase of engine and tender	m ft	**18,9** 62,0
163	Summary of design maximum axle loads (static, excluding any dynamic augment, and based on 20 ton total leading bogie load): (a) per axle: (b) per metre of engine rigid wheelbase: (c) per metre of total wheelbase (engine and tender): (d) per metre of total length over buffers:	 ton ton/m ton/m ton/m	 **20,0** **12,7** **8,5** **7,2**
164	According to Koffman[7] the specific starting resistance on level tangent track for roller bearing stock is:	kg/ton N/ton	7 69
165	Applying this to the average tender mass in service gives starting resistance of tender = [15] x [164]	kN	4,3
166	Specific starting resistance of engine will be greater than [164] on account of more machinery to set in motion: it is taken as:	N/ton	100
167	Starting resistance of engine = [6] x [166]	kN	8,0
168	Total starting resistance of engine and tender = [165] + [167]	kN	12,3
169	Starting drawbar tractive effort on level tangent track = [48] – [168]: This is rounded up to:	kN kN lbf	133,7 **134** 30 132
170	Starting drawbar efficiency (= e. db. t.e. ÷ wheel rim t.e.) = [169] ÷ [48]: This is rather low for a roller bearing equipped locomotive, probably partly because item [166] may be less than assumed, but also reflecting the large tender mass for the locomotive's nominal tractive effort.	%	92
Supplementary calculations to check the assumed boiler absorption efficiency, item [102]			
171	Boiler absorption efficiency = (heat transferred to water/steam in boiler and superheater ÷ heat released in firebox).		
172	Heat transferred through heat transfer surfaces = (heat transferred to water/steam in boiler and superheater + boiler radiation loss). The radiation loss from a boiler with average quality of insulation as a % of the energy in the fuel burnt at full load (boiler stress ≈ 100 kg/ m^2-h for the boilers concerned)[8] ≈	%	3
173	For a heavily insulated modest-size boiler at very high boiler stress (≈ 112 kg/ m^2-h at [90], assuming for the purpose of these calculations the same total evaporative heating surface area as the BR 5MT (153,3 m^2)) assume this is reduced to:	%	2
174	Heat lost by radiation ≈ [103] x [173]	GJ/h	1,2
175	Heat transferred through boiler and superheater heat transfer surfaces = ([96] + [174]) = (([102] x [103])+ [174])	GJ/h	50,8
176	[175] = {heat entering the firebox + heat generated by combustion – heat lost in smokebox gases}		
177	Heat entering the firebox = {heat in combustion air + heat in atomizing steam + heat in fuel}. The last is negligible and is ignored (this gives a conservative (safe) result to these calculations). From combustion gas enthalpy-temperature (h-t) chart [Fig. 1.3.2.] enthalpy of air at [100]:	kcal/N*m^3 kJ/kg	31 100
178	Heat in combustion air = [126] x [177]	GJ/h	2,72

Item No.	Item	Unit	Amount
179	For purposes of this check atomizing steam is assumed to be superheated at temperature = [73] and its enthalpy is taken as [77]. Heat in atomizing steam is then [77] x [112]	GJ/h	0,96
180	From equation [176] heat lost in smokebox gases = ([178] + [179]) + [103] – [175] =	GJ/h	11,28
181	Smokebox gas flow = [110] + [112] + [126]	kg/h	28 752
182	Smokebox gas enthalpy = [180] ÷ [181]	kJ/kg	392
183	Smokebox gas density at [125] ≈	$kg/N*m^3$	1,3
184	Smokebox gas enthalpy = [182] x [183]	$kJ/N*m^3$ $kcal/N*m^3$	510 122
185	From combustion gas enthalpy-temperature chart [Fig. 1.3.2.] at [184], [109] and [267], temperature of gases leaving the boiler tube bundle is:	^oC	357
186	For SAR loco No. 3450 smokebox gas temperature at steam temperature = [73][9] is: This is significantly > [185], which however should be possible with a steam temperature = [73] by careful design of the superheater. If so, the boiler absorption efficiency estimate, item [102], is shown to be acceptable by items [175], [176], [180] & [185]. However the importance of optimizing all items affecting boiler absorption efficiency is indicated (especially the combustion equipment, to minimize excess air, item [125] being considered a maximum figure at full boiler load, and also the need for a combustion air supply temperature of at least 100 ^oC (item [100]).	^oC	405
187	If no air preheater were fitted and combustion air entered at 15 ^oC (item [120]) enthalpy of combustion air entering firebox (from Fig. 1.3.2.) =	$kcal/N*m^3$ kJ/kg	5 16,2
188	Heat in combustion air = [126] x [187]	GJ/h	0,44
189	Heat lost in smokebox gases = ([188] + [179]) + [103] – [175] =	GJ/h	9,0
190	Smokebox gas enthalpy = [189] ÷ [181]	kJ/kg $kcal/N*m^3$	313 97
191	From combustion gas enthalpy-temperature chart [Fig.1.3.2.] at [190], [109] and [267], temperature of gases leaving the boiler tube bundle would be: This is considered unrealistic for a steam temperature = [73], therefore the necessity of a combustion air preheater is confirmed.	^oC	283
	Supplementary calculations for coal firing (GPCS = Gas Producer Combustion System)		
192	L.C.V. of locomotive coal now available, assumed:	MJ/kg	30
193	Coal fully burned ≈ [103] ÷ [192] (note: is approximate as some constituents of coal burn preferentially to others)	kg/h	1 947
194	Ash content of coal of L.C.V. = [192] ≈	%	8
195	Coal gasified during combustion = [193] x (1-[194])	kg/h	1 791
196	Firegrate area (here assumed same as BR Class 5MT, although a larger (longer) grate will be fitted if possible)	ft^2 m^2	28,7 2,67
197	Specific burning rate = [193] ÷ [196]	kg/m^2-h	729
198	Maximum sustained specific burning rate, SAR 26 Class No. 3450[10] ≈	kg/m^2-h	638
199	Burning rate at the apparent grate limit, SAR 26 Class No. 3450[11] ≈	kg/m^2-h	830
200	[198] < [197] < [199]. The burning rate at [90] with coal firing would therefore be near the absolute maximum possible with loco. No. 3450, however the better GPCS conditions on the Class 5AT, allowing a higher 2dy/1ry air ratio, favour the possibility of high specific combustion rates.		
201	Stoichiometric air : fuel ratio by weight for coal of L.C.V. = [192] ≈	kg/kg	10
202	Combustion air flow = [193] x [201] x [125] (c.f. item [126] for oil firing)	kg/h	25 311
203	For the following analysis four levels of primary air flow are considered: (1) 30% of total combustion air as primary air, corresponding to optimum GPCS operation (2) 40% of total combustion air as primary air, corresponding to average GPCS operation (3) 50% of total combustion air as primary air, corresponding to poor GPCS operation (4) 100% of total combustion air as primary air, corresponding to 'classical' combustion		

Item No.	Item	Unit	(1)	(2)	(3)	(4)
204	Primary air as a % of total combustion air	%	30	40	50	100
205	Primary air flow = [202] x [204]	kg/h	7 593	10 124	12 655	25 311

Item No.	Item				Unit	Amount
(204)	Primary air as a % of total combustion air	%	30	40	50	100
206	Average value of clinker control steam required per kg of primary air[12]	kg/kg	0,12	0,12	0,12	-
207	Clinker control steam flow =[205] x [206]	kg/h	911	1 215	1 519	0
208	1ry air + clinker control steam = [205]+[207]	kg/h	8 504	11 339	14 174	25 311
209	Specific primary air + clinker control steam flow through firebed = [208] ÷ [196]	kg/m^2-h	3 185	4 247	5 309	9 480
210	At [209] combustion efficiency (deduced from[13])	%	<u>83</u>	75	66	<50
211	% free gas flow area through firebed ≈ (In a truly packed bed the % free gas flow area would be < 10% [14] but the figures here allow for progressive 'unpacking' of the firebed which occurs as the air flow rate increases.)	%	15	20	25	40
212	Combustion gas temperature at firebed top[15] ≈	°C	900	1 000	1 200	1 400
213	Comb. gas spec. vol. at [212] (taken = that of air)	m^3/kg	3,3	3,6	4,2	4,7
214	Combustion gas velocity at top of firebed = [209] x [213] ÷ [211]	m/s	19,5	21,2	24,8	30,9
215	Size of coal particles which will be carried off firebed at [214][16]	mm	4,8	5,3	7,1	10,7
216	Particle mass $\propto$ (linear dimension)^3. Therefore mass of coal particles carried off firebed as % of that for 30% 1ry air = ([215] ÷ 4,8)^3 x 100%	%	100	135	325	1 110
217	This analysis is approximate and assumes even air flow through the fire – channeling could greatly increase the size of coal particles carried off. It shows the steep rise in the mass of coal particles which can be lifted off the firebed as the 1ry air flow increases ([216]) and the corresponding drop in combustion efficiency ([210]) (judicious directing of the GPCS 2dy air streams can be used to return escaping particles to the fire) Item [210] indicates optimum operation of the GPCS would be needed for combustion efficiency with coal firing to be acceptable at maximum evaporation. 'Classical' 100% 1ry air combustion will be unacceptable, in fact it is most probable that the grate limit would prevent it from attaining the required burning rate. Given the deep firebox of the Class 5AT - ideal for the GPCS - near-optimum combustion may be realized in practice with the right kind of (high volatile) coal, so the analysis is continued on the basis of GPCS operation with 30% primary air, giving 83% combustion efficiency at the boiler's maximum rated output.					
218	Specific firing rate = [197] ÷ [210]				kg/m^2-h	878
219	Firing rate = [218] x [196]				kg/h ton/h	2 345 2,35
220	Allowable sustained hand firing rate for a single fireman in UK[17]: [219] is 72% higher than [220], therefore a mechanical stoker is obligatory for obtaining full rated boiler output.				lb/h kg/h	3 000 1 360
221	To give same range as with oil firing, bunker capacity = [140] x [219] = This is 64% > item [139] and would reduce the water supply by approximately 10% for a total supplies weight = [12]. In the case of coal fuel a closer relationship between the ranges based on fuel and on water supplies than is the case with oil firing may be advantageous.				ton	11,5
222	Mechanical stoker steam jet consumption ≈				kg/h	100
223	Total combustion gas flow through the boiler tubes = [195] + [202] + [207] + [222] (c.f. item [181] for oil firing)				kg/h	28 113
224	**Summary**. With coal as fuel the rated boiler output (item [90] should be realizable, but at a lower combustion efficiency than assumed for oil firing, compare items [106] and [210]. The higher fuel consumption will slightly reduce the operating range for a given total quantity of supplies. At a combustion efficiency of 83% (item [210]) the char carry-over will probably be such as to require a self-cleaning and spark –arresting smokebox. However the combustion efficiency may rise by more than it does with oil firing as steam demand decreases, therefore the ratio of fuel consumption at part load to that at full load may be better than for oil firing (item [147]), so that under average service conditions the difference in performance between the two fuels would be expected to be less than indicated by these calculations. Better performance than indicated here will also be possible if coal of higher calorific value than given in item [192] can be supplied.					

Item No.	Item	Unit	Amount
Supplementary calculations for obtaining the estimated indicator diagram at [2] and [17], Fig. 1.3.1.			
225	Known data is; Steam chest pressure (assumed constant during cycle) (item [71]) Exhaust steam pressure (item [72]) Piston swept volume, each end of cylinder (item [69]) Cylinder clearance volume, assuming twin piston valves (item [67])	 kPa kPa m^3 %	 2 000 50 0,122 8
226	Indicated work per piston stroke (item [68])	kJ	112,3
227	The following data required for drawing the estimated indicator diagram is deduced from indicator diagrams made on SAR 26 Class locomotive No. 3450. The speed of this locomotive at coupled wheel rotational speed = [23] is 89,9 km/h (item [58]) and the nearest diagram to this speed and a cut-off = 25% item [60] is at 84 km/h and 28% cut-off [18]. For this diagram, ΔP at point of cut-off, as a % of the peak cylinder pressure, is:	%	16
228	ΔP at point of cut-off is dependent on factors such as the mean inlet port opening relative to the cylinder volume, cylinder wall effects, and particularly the speed of valve closure, which are more optimal on the 5AT. Therefore ΔP for the 5AT is taken as: Note: the 5AT peak cylinder pressure is assumed = steam chest pressure	%	12
229	Cylinder pressure at cut-off = [71] x (1 – [228])	kPa	1 760
230	From the 3450 diagrams, peak cylinder pressure is generally reached after dead centre. For the diagram concerned the piston position at peak pressure as a % of the stroke, ΔS, is:	%	7
231	Due to various beneficial factors on the 5AT (e.g. longer lead, lower clearance volume and reduced wall effects) ΔS is taken as:	%	2
232	For 3450, the maximum pressure reached at dead centre as a % of the peak cylinder pressure (ideally 100%) is:	%	64
233	Due to the various beneficial factors on the 5AT given in item [231], [232] is conservatively increased to:	%	80
234	Maximum pressure at dead centre = [71] x [233]	kPa	1 600
235	Caprotti gives the index of expansion as 1,2[19] and Porta as 'smaller than adiabatic'[1,3][20]. However due to the high superheat [73] and all cylinder design factors aimed at achieving it, the expansion will be close to isentropic and may be assumed to follow the curve ($pv^{1.3} = k$), where p is absolute pressure. This is confirmed by expansion lines of high-speed diagrams taken on 3450.		
236	For the 3450 diagram, % of the piston stroke at which pressure departs from the expansion line at the start of release ≈	%	84
237	Due to longer exhaust lap, [236] is increased for the 5AT to:	%	85
238	For the 3450 diagram, gauge pressure at the end of the stroke as a % of the gauge back pressure:	%	200
239	[238] is retained for the 5AT: pressure at end of stroke = [72] x [238]	kPa	100
240	For the 3450 diagram, % of the return stroke at which pressure falls to the back pressure line (assumed same for the 5AT):	%	7
241	For the 3450 diagram, % of the return stroke at the apparent compression point, i.e. the point at which the valve commences to close to exhaust and where the exhaust pressure starts to rise above the back pressure line:	%	76
242	Due to longer exhaust lap [241] is decreased for the 5AT diagram to:	%	75
243	The compression is effectively isentropic, [19][20] i.e. $pv^{1.3} = k$. Point [242] does not define the true start of the compression line[21] but is assumed to do so for the purposes of these calculations (a 'safe' assumption as it reduces the diagram area).		
244	The foregoing gives all data for drawing the estimated indicator diagram except for the cut-off. Diagrams are drawn, starting with the roughly estimated cut-off, item [60], until the diagram area matches the required indicated work [226]. This diagram is given in Fig. 1.3.1. from which the required cut-off for a cylinder power item [17] at a speed item [2] is:	%	**26**
245	From Fig. 1.3.1. the gauge compression pressure at the moment the valve opens to lead steam ≈	kPa	800
246	Assuming isentropic compression from the back pressure line at [72] and [79], the temperature of the compressed steam at [245], from h – s chart:	^{o}C	395

Item No.	Item	Unit	Amount
247	[246] < [73], but in practice there will be some heat transfer from the cylinder walls to the exhaust steam, making the temperature at the start of compression higher than [79]; if the temperature at the start of compression = 205 ^{o}C or more, the temperature at [245] ≥ [73].		
248	The indicated m.e.p. at [17] and [2] = [68] ÷ [31] x [50]	kPa	917
	Supplementary calculations for obtaining the cylinder steam flow, item [75].		
249	The method of Porta is used[22]. In the diagram Fig. 1.3.1. the variable inlet pressure is substituted by an equivalent mean inlet pressure giving equal work done by equating the hatched areas. This pressure is:	kPa	1 880
250	At [73] and [249] the steam specific volume (from steam tables) is:	m^3/kg	0,166
251	Volume at point (A) Fig. 1.3.1.	m^3	0,0056
252	Volume at point (B) Fig. 1.3.1.	m^3	0,0395
253	Mass of steam admitted per stroke = ([252] – [251]) ÷ [250]: This is the adiabatic quantity, i.e. assuming zero heat transfer to the cylinder walls. This heat transfer results in a reduction in admission steam temperature and specific volume and hence in a larger amount of steam being admitted (i.e. the so-called ‘missing quantity’), and this is allowed for by item [80].	kg	0,204
	Supplementary combustion calculations for item [124] and use of Fig. 1.3.2.		
254	The following uses the method given in Ref. [26]. First, the calculation of the theoretical air per kg of fuel burnt is made, for diesel fuel / gas oil. The following is per 100 kg of oil and is for combustion only and excludes atomizer steam.		

Item No.	Constituent	kg per 100 kg of oil[27]	÷ mol weight	= kmol	kmol of O_2 required	Theoretical air
255	Carbon	86,3	12	7,19	7,19	= 10,51 x 100/21 = 50,0 kmol
	Hydrogen	13,2	2	6,60	3,30	or 50,0 x 28,9/100
	Sulphur	0,5	32	0,02	0,02	**= 14,5 kg air / kg oil**
					Σ = 10,51	(= item [124])

Item No.	Item	Unit	Amount
256	Actual air = 50 kmol/100 kg oil x [125]	kmol/100 kg oil	65
257	N_2 in combustion air = [256] x 79%	kmol/100 kg oil	51,35
258	O_2 supplied in combustion air = [256] – [257]	kmol/100 kg oil	13,65
259	Constituents of combustion (flue) gas:		
	(i) CO_2 (from [255] column 5) =	kmol/100 kg oil	7,19
	(ii) SO_2 (from [255] column 5) =	kmol/100 kg oil	0,02
	(iii) O_2 = [258] - [255] Σ column 6 =	kmol/100 kg oil	3,14
	(iv) N_2 (item [257]) =	kmol/100 kg oil	51,35
	(v) H_2O (from [255] column 5)	kmol/100 kg oil	6,60
260	Total of item [259]	kmol/100 kg oil	68,30
261	Combustion gas composition by volume = [259] ÷ [260]:		
	(i) CO_2 =	%	10,53
	(ii) SO_2 =	%	0,03
	(iii) O_2 =	%	4,60
	(iv) N_2 =	%	75,18
	(v) H_2O =	%	9,66
262	At excess air = 30% (item [125]), carbon in 100 kmol of dry flue gas = (100 ÷ [260]) x (86,3 ÷ 12) =	kmol	10,53
263	Carbon per 100 kg of oil (from [255] column 5)	kmol	7,19
264	Flue gas produced per 100 kg of oil = 100 x ([263] ÷ [262]) = [260] =	kmol/100 kg oil	68,3
265	Composition of [264] (final numbers in equations are molecular weights):		
	(i) CO_2 = [264] x [261](i) x 44 = [259](i) x 44 =	kg/100 kg oil	316,4
	(ii) SO_2 = [264] x [261](ii) x 64 = [259](ii) x 64 =	kg/100 kg oil	1,0
	(iii) O_2 = [264] x [261](iii) x 32 = [259](iii) x 32 =	kg/100 kg oil	100,5
	(iv) N_2 = [264] x [261](iv) x 28 = [259](iv) x 28 =	kg/100 kg oil	1 437,8
	(v) H_2O = [264] x [261](v) x 18 = [259](v) x 18 =	kg/100 kg oil	118,8
266	Total of item [265]	kg/100 kg oil	1 974,5

Item No.	Item	Unit	Amount
267	Fraction of CO_2 in combustion gas = [265](i) ÷ [266] (for use in Fig. 1.3.2.)	%	16,0
268	Total combustion gas flow, including atomizer steam = (([266] ÷ 100) x [110]) + [112] = This gives good agreement with item [181] (within 0,5%).	kg/h	28 601

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