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THIRD GENERATION STEAM: FACING THE ENERGY CRISIS

I PART: A TECHNICAL PROPOSAL
II PART: OWNERSHIP AND OPERATING ASPECTS OF MODERN STEAM POWER

SECTION	:	General
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SUMMARY OF PART I

Technical aspects of an advanced "Third-Generation-Steam" proposal achieving an all-year-round thermal efficiency (mine pit to drawbar) of 15,4% $\pm$ are described. Detailed thermodynamical calculations prove the accuracy of the above figure, which is 20,1% when expressed in the costumary "constant speed-level" form excluding various running losses.

This was achieved assuming a second, perhaps a third grade low calorific coal, which can be successfully burned "thanks to the author's Gas Producer Combustion System. (Ref. 2).

The coming energy crisis enhance the importance of thermal efficiency, which "in the past never sold a single locomotive". A minimum research program is detailed, and the author expects that any engineer current with steam locomotive technology will recognize the feasibility of the scheme.

Part II will analize what railways can expect from the technical proposal described in Part I.

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APPENDICES

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1. INTRODUCTION

The general context within which the present paper is written and also the considerations concerning the application of the present proposal will be dealt by the co-author Mr. D. Wardale in Part II. Part I will describe and support by adequate calculations the technical aspects of a proposed steam locomotive which still keeps the traditional Stephensonian form in a non condensing version. If we call "First Generation Steam" what was achieved by the '50s, "Second GEneration Steam" what can be done now incorporating those developments occuring during the last 25 years (Porta 1)*,"Third Generation Steam" (TGS) represents what could be achieved once a discrete - or advanced - research effort was carried out. This proposal of course does not dismiss any other alternatives deserving all the author's respect; it represents a logical development of his own work (2), and rests upon Chapelon's one, who, by the '30s, succeeded in improving the classical locomotive to make a better traffic tool out of it as compared to other not too happy unorthodox developments.

It is important to realize that the present non condensing proposal is not an upper limit, and in due time a condensing version showing some 25% better thermal efficiency is expected to follow. Besides, it has been inscribed in a rather conservative technical scheme sacrificing some points in efficiency so as to show better and nearer chances to become materialized at short notice.

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* Numbers in parenthesis refer to references listed at the end of the paper.

The energy crisis demands that the "all-year-round" thermal efficiency be given paramount importance, and the days in which it was believed that "thermal efficiency never sold a locomotive" are now bygone. This explains the complicated (in the traditional sense) cycle, which nevertheless is entirely and naturally automatic without any special apparatus other than steam traps.

Although all the calculations have been referred to a 6000 CV_e engine (equivalent to a nominal 9000 HP diesel) suited for the Mexicans, USA and Canada Railways, a second figure column corresponds to a smaller size unit suitable to 30 kg m^{-1} rail (60 lb $yard^{-1}$ = 13 tonne axleload) and metre gauge (including the Colombian 981 mm) prevalent in various latinamerican countries.

The reader is particularly sent to Ref. (2) which depict the important development work carried out in Argentine until 1969.

2. PROPOSED NON CONDENSING CYCLE

The range of conditions to be covered by locomotive power is exceedingly wide, so that no universal proposal can be expected to be valid. However, one should not speak about an entelechy; therefore, this description of the thermodynamical cycle will take the form of a sizable example for the USA Railroads bearing in mind that there is a definite trend for smaller train loads, faster speeds and more frequent shipments (le Massena, (3)). If a drawbar thermal efficiency of 21% can be obtained under favourable circumstances, the rated continuous figure of 6000 CV_e (*) at the drawbar requires a heat input of:

$$\frac{5000\ CV_e.\ \ 632,5\ kcal\ CV_e^{-1} h^{-1}}{0.21} = 15.10^6\ kcal\ h^{-1}$$

This is equivalent to some 2000 kg h^{-1} of good 7500 kcal kg^{-1} coal. Which is a quite small bulk to carry, stock, handle, buy and pay; this in sharp contrast with the huge 8000 kg h^{-1} demanded by the famous Niagara, an obvious result related to the fundamental equation of the steam locomotive (1).

The upper bound of the cycle calls for higher steam pressure and temperature: the choice was for 60 kgf cm^{-2}/550C. The former figure is thought consistent with the normal Stephensonian boiler. It may be designed with a Brotan water tube firebox, successfully adopted by Lawford Fry (4) and recommended by Chapelon (5). An alternative to the latter is to keep the flat stayed firebox construction with all inherent virtues as proposed by the author (6).

A steam temperature of 550°C should present no lubrication problems if author's technique is followed (7) and its associated heat transfer knowledge (Porta, (8) (9) (10)). However, special (yet current) steels will be required for superheated piping and part of the superheated tubing. As it is well known, there is no obstacle in obtaining high superheats in the normal locomotive boiler. Such temperature of 550°C dispenses with re-superheating to avoid wet steam conditions at the LP end.

The reciprocating engine is adhered to for the reasons outlined in Section 1. The adoption of double or triple expansion depends on reasons beyond the scope of the present paper; a three cylinder triple expansion system has been chosen for the present exercise.

(*) 1 metric housepower = 1 CV = 0,986 HP. Suffix e refers to power (or effort) at the drawbar at constant speed and on the level.

The reader should know that the author is fully aware that every assertion, estimation, assumption, guess or calculation should require a complete paper in itself to prove what is said. This, of course, is not possible in the present circumstances (although it could provide matter for discussion) while it must be admitted that some of the attacked questions do require research computations, trials and errors, mistakes and troubles, etc.

Referring to Fig. 1, in order to obtain some 6000 CV_e at the drawbar at rated power, the steam generator 1 produces 24,000 Kg of perfectly dry steam per hour at 57 kgf cm^{-2} pressure (*). The maximum nominal working pressure is 60 kgf cm^{-2} gauge (853 psig). After passing through a normal firetube superheater 2, a steam temperature of 550 °C is reached, whilst at full load the High Pressure (HP) steamchest steam pressure is 51 kgf cm^{-2}; the 6 kgf cm^{-2} pressure drops is to be expended in the high temperature region of the superheater so as to get high heat transfer coefficients leading to the lowest possible metal temperatures. The engine is of the three cylinder, triple expansion type with no single expansion working provided. The various receiver pressures are chosen so that approximately equal power is developed in each cylinder, while cylinder diameters are selected so as to give approximately equal piston thrusts; yet the HP (left hand) cylinder develops some 15% more power and thrust to equalize frame stresses and axlebox loads.

Steam is exhausted from the Low Pressure (LP) cylinder (on the right hand side) at a time average back pressure of 1,4 kgf cm^{-2} through the author's Kylpor ejector 4 (Porta and Taladriz (11)). However, the LP piston valve, being according to author's design, allows most of the draught to be realized by the energy available in the incomplete expansion toe of the indicator diagram : this leads to an actual back pressure of 1,17 kgf cm^{-2} (plus passage resistances) during the exhaust phase on the indicator diagram. The atmospheric pressure is 0,97 kgf cm^{-2} = 715 mm Hz (corresponding to a height of 500 m (1500 ft) above sea level) which is considered more representative than sea level pressure.

General engine steam leaks 5 and steam for various unimportant uses have been assumed to be superheated and about 240 kg h^{-1} (internal piston and valve leakage is to be considered when dealing with the internal isentropic efficiency of the engine). This small - but important because of its continuity - figure implies a serious effort to correct small leaks that have been a plague in steam locomotive operation.

95% of the combustion air is passed successively through four steam air heaters 9.8.7 and 6 located on the firebox sides, while the remaining 5% passes through the bottom ash pan afterburning grate 55. Heater 9 is fed by exhaust steam piped directly to the LP valve chest cover, a non return valve 19 being interposed on the pipe so that only steam at a higher pressure obtaining during the exhaust beat can pass. Therefore, the pressure on the feed line 18 is higher than the time average pressure obtaining in the exhaust chamber, even if by a small amount, which increases when working heavily in full gear. Since steam feeding the various heaters is taken at various pressure levels along the expansive pressure drop obtaining along the engine, hence resulting in a regenerative cycle. The final temperature attained by the 95% air passing through the steam heaters is 200°C, while that of the reamining 5% burning the live coals falling from the main grate 43 is not over 450°C. The "95% air" is divided into (i) secondary air 11 (65% to 45% of the total) entering above the firebed through air thimbles 10 as required by the Gas Producer Combustion System (GPCS), and (ii) primary air 12. Combustion gases are subjected to a swirling motion in the gas space of the cyclonic combustion chamber 13 (2). Some steam 14 is piped to help the swirling action, the coming from the receiver II between the Mid Pressure (MP) cylinder 15 and the Low Pressure (LP) cylinder 3.

(*) Unless otherwise stated, all pressures are absolute.

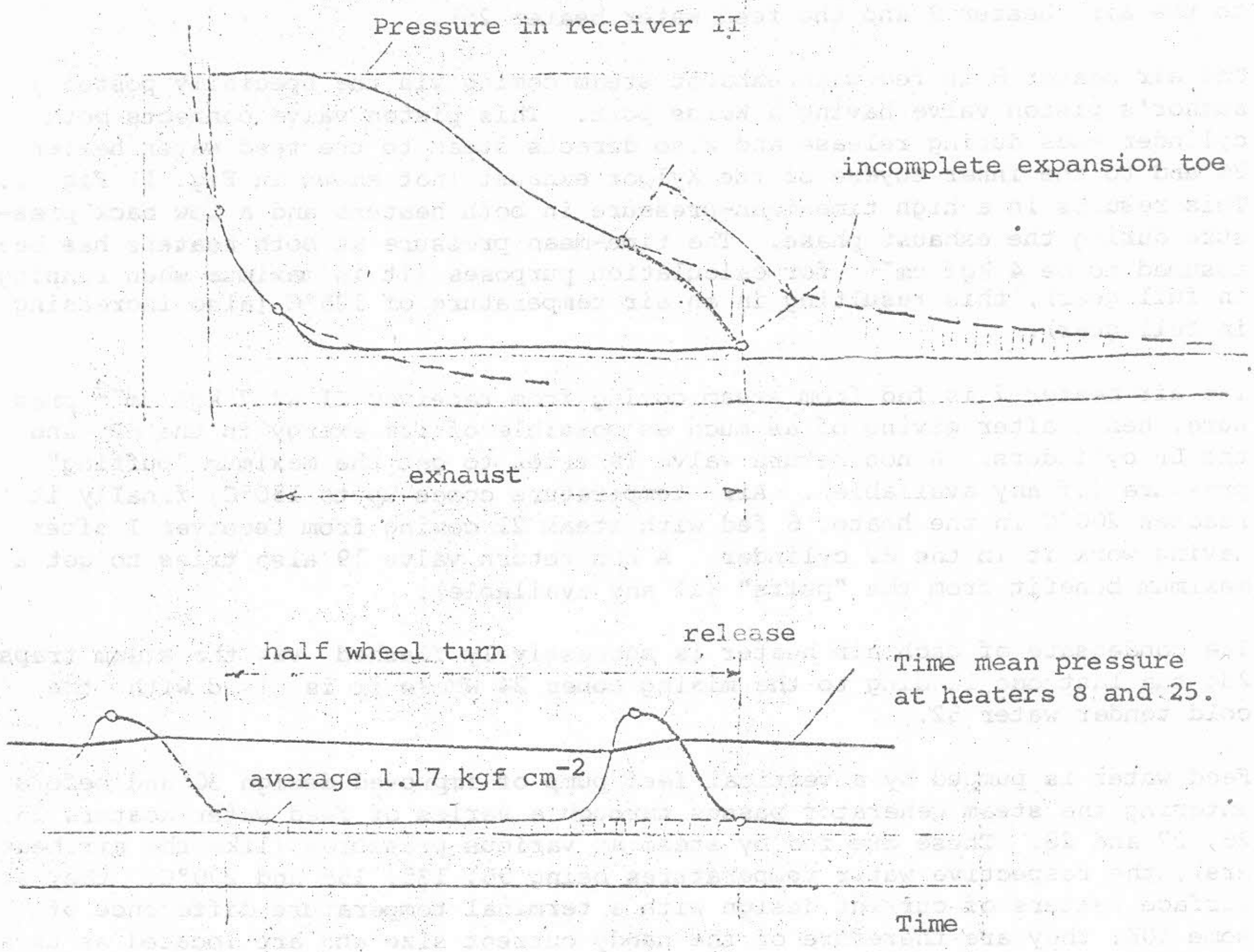

Fig. 2 LP INDICATOR DIAGRAM

The Weiss port communicates both cylinder ends during release and also the heaters 8 and 26, and the inner tuyére of the Kylpor blast pipe. Non return valves are inserted in the heaters feed so that the time average steam pressure in them approach that of the peak at release. Most of draught work is produced by the otherwise lost energy showing as incomplete expansion toe. Therefore, the back pressure during the exhaust phase can be lower than that obtaining with the normal exhaust passage arrangement.

LP exhaust steam is also piped from the LP valve cover to the ashpan for the GPCS where it is distributed by the perforated piping 17 (same steam as branched to the air heater 9 and the feed water heater 25).

The air heater 8 is fed with exhaust steam coming via the specially posted author's piston valve having a Weiss port. This piston valve connects both cylinder ends during release and also directs steam to the feed water heater 26 and to the inner tuyére of the Kylpor exhaust (not shown in Fig. 1) Fig. 2. This results in a high time-mean-pressure in both heaters and a low back-pressure during the exhaust phase. The time-mean pressure at both heaters has been assumed to be 4 kgf cm^{-2} for calculation purposes (it is maximum when running in full gear), this resulting in an air temperature of 135°C (also increasing in full gear).

The air heater 7 is fed from steam coming from receiver II at 7 kgf cm^{-2} pressure, hence after giving of as much as possible of its exergy in the HP and the LP cylinders. A non return valve 19 tries to get the maximum "puffing" pressure (if any available). Air temperature comes up to 150°C; finally it reaches 200°C in the heater 6 fed with steam 21 coming from receiver I after having work it in the HP cylinder. A non return valve 19 also tries to get a maximum benefit from the "puffs" (if any available).

The condensate of each air heater is successively flashed via the steam traps 23, the last one leading to the mixing cones 24 where it is mixed with the cold tender water 52.

Feed water is pumped by a vertical feed pump of improved design 30 and before entering the steam generator passes through a series of feed water heaters 25, 26, 27 and 28. These are fed by steam at various pressures (like the air heaters), the respective water temperatures being 96, 135, 155 and 200°C; they are surface heaters of current design with a terminal temperature difference of some 10K: they are therefore of the handy current size and are located as usual on the top of the smokebox where they cannot be flooded. Their respective condensators are flashed back via the traps 29 as in the case of the air heaters. The last condensate is piped to the mixing cones 24.

The feedwater pump 30 is fed by steam coming from the receiver II, hence at rather low pressure and after having made up a good use of its available exergy in the HP and MP cylinder. In spite of the various improvements resulting from applying thermodynamic techniques as per the main engine, resort is made to compounding so as to get the maximum efficiency because the feedwater pumping work is important. The pump exhaust 32 discharges into the exhaust steam line 20. The pump aspirates cold tender water from the tender tank 37, and a set of mixing cones 24 (resembling those of an exhaust steam injector), serves to heat up the aspirated water with the condensates coming from the heaters 9 and 25. It is also desired to condense as much as possible of exhaust steam coming with the condensate so as to get the maximum temperature of the water entering the heater 25. This has a limit because the pump 30 begin to do false strokes if the water is too hot at the aspiration. However, if the cones are designed like those of an exhaust steam injector (though not obligatorily so), some pressure gain will be obtained thereby reducing the pumping work.

It has been firmly established (confirming a secular practice) that the coal should be thoroughly wetted before firing. The normal practice is to use a water hose or sprinkler, but this leads to the loss inherent to evaporate the added water. If exhaust steam 38 is sentto the bottom of the coal space 39 it will become condensed yet there will be practically no heat loss associated to wetting, the difference amounting to a saving of some 0,7%. Since the coal is kept hot, freezing problems are solved.

After leaving the last heater 28, the feed water passes through a smokebox economizer 40; before entering the steam generator its temperature reaches 270°C. Outlet gas temperature is 380°C while it is some 570°C at the outlet of the steam generator, thereby facilitating the design of the superheater.

The stoker consists of a normal screw conveyor 41 but the distributing heat 42 is of mechanical type to avoid the heavy consumption of the usual spreading jets.

Energy economics demand a better treatment of the efficiency of the various auxiliaries. There is no case in making efforts in increasing the efficiency of the main engine by a tiny increment later lost in a wasteful air brake pump. Hence, the following means.

- A high speed, automobile type stoker engine 44 fed by receiver II steam 45.
- A crosshead driven air brake boster pump 49.
- A small steam engine 47 driving the electric generator.

All auxiliaries are driven with 7 kgf cm^{-2} steam from receiver II, which is also moderately superheated. The descriptions of means to work them when the regulator is closed is not given here nor detailed in Fig. 1

Some other details of the cycle layout are given in Fig. 1, which also reports design pressures, temperatures and various calculated quantities.

Since any locomotive, unlike a power station, is an intermitly operating machine involving starting and topping losses, means are provided to reduce the latter to a minimum:

* Perfectly airtight automatic dampers closing when the regulator shuts.
* A minimum of steam leakage everywhere.
* Experience has shown that when the regulator shuts, steam from various auxiliares exhausts rapidly cools down the temperature of the firebed so as to reduce volatile matter driving off.
* Most important of all, the guaranteed ability of the engine to steam properly when pulling.

2.1 General Considerations

In Fig. 1 it is very apparent that while the engine is fed by 24,000 kg h^{-1} of steam, cold tender feed water is but 15,539 kg h^{-1}, hence in a ratio of 1.54. This results from a high proportion of condensate amounting to 6,206 kg h^{-1} from the feedwater heaters and 2,255 kg h^{-1} from the air heaters, the latter condensing some 14,5% relative to tender feed. In fact, a generator evaporating 16,000 kg h^{-1} of water is a small one and still within the capabilities of hand firing (which for other reasons is not proposed). That smallness results directly from the fundamental equation of the steam locomotive (Porta (1)):

$$\text{Power} = \frac{\text{Steam produced per hour}}{\text{Specific steam consumption}}$$

Obviously, the efficiency of the cycle reflects on the denominator. Another result is that the generator can be confortably located within the space and weight restrictions inherent to railway motive power; this applies particularly to the steam generator main drum (boiler barrel) whose weight is only 6,000 kgf even if made in ordinary steel and its thickness 43 mm only.

The whole cycle maintains the unsurpassable inherent automaticity of the traditional Stephensonian locomotive, and no special apparatus are required for control. The usual couple of men is still adhered to, yet further development allows to foresee one manning and radio control.

The GPCS has shown, on tests, a large verstility in burning the most unusual fuels:

a. Wood of any quality with any degree of moisture, ash content and characteristics in size from the usual 0,15 m x 0,60 m logs down to otherwise discarded small 50 mm three branches. This increases the ultimate thermal efficiency of wood plantations;
b. ..... which can be of rapid growing species thereby making possible to run railways on solar energy.
c. Sawmill rejects, including sawdust (16).
d. Zero to 5 mm charcoal fines loosely mixed up with fuel oil, pitch, parafinic fuel oil (solid at room temperature), fuel oil semisolid tank bottoms, etc.
e. Ditto with 0 x 4 mm slack coal of any rank.
f. Slack coal alone, run of mine, etc., of any size and rank and coking qualities and firability, ash up to 30%, initial softening point R.A. - 1050 °C.
g. Oil refinery (ash less) coal to any specification, either alone or in combination of the above fuels.
h. cannel coal.

Wood pieces up to 0.15 x 0.15 m. can be fired mechanically.

No experience was made with antracite, lignites or peat, but everything points to the inclusion on the list.

All the above fuels can be burned alone or mixed in various proportions (which may be varied even at full steaming) and without changes in the burning equipment. Steam generator efficiency, thermal efficiency and rated power are not significantly affected, nor antipollution qualities or engine responsiveness.

The author adheres to the classical reciprocating engine for various reasons:

(i) It is a technique which he knows in full detail, by means of which he can reach the proposed goal.
(ii) Everything points to show that reciprocating machinery keeps a heatly actuality in various fields like automotive, marine, compressors, etc., and more recently in non-internal-combustion automobile power plants where they are called "RANKINE expanders".
(iii) Energy conversion efficiency is higher than that obtaining in the best turbine even when considering mechanical losses (x 80% over a wide range of powers and speeds).
(iv) A 150-year-old experience proves its suitability to railway work.

2.2 Environmental pollution

The proposed cycle represents a large progress as compared to the traditional locomotive because various concurrent factors multiply their favourable effects: in spite of keeping to the external form and shape, the proposed locomotive has very little to do with the familiar black-smoking-spark-trailing image we all have.

(i) Since the all-year-round thermal efficiency related to brute useful work passes from 6% to 16%, pollution is divided by a factor of 2.7.

(ii) Author's Gas Producer Combustion System (GPCS) reduces smoke to nil (even with the most offending fuel) which probably means a reduction of other contaminants (CO, hydrocarbours, etc.) by a factor of 5 to 10.

(iii) It shares the capacity of external combustion engines to produce a very low nitrogen oxide contamination. Since its energy consumption per unit of transportation work is not too different from that of a diesel engine, the advantage on this aspect is on the large side.

(iv) It requires at most one quarter of the energy demanded by road transport for same transportation duty.

(v) The GPCS also provides a nearly smoke-free combustion at engine terminals, and this is to be related to the low frequency of lighting up and the size of power demanded by a given transportation duty.

(vi) Char ejection through the chimney, which paved the right ofway of American RRs, is reduced by a factor of 20 to 100 because of the GPCS, the cyclonic flame path and the large grate, even when employing fuable slack coal or lignite. The 2,7 greater thermal efficiency increases the former figure to 60 - 300 times for equal transportation work performed.

(vii) While no data is available, it is possible that the Gas Producer Combustion System reduces the amount of sulphur in combustion gases, at least, it is 2,7 times smaller than that obtaining in normal locomotives for the same transportation duty on account to the higher thermal efficiency. On the pure combustion side, the chances are for the good because the greatest part of the fuel bed is at a temperature between 1000°C and 700/800°C at the top, while a rather cool, highly turbulent, swirling flame burning with low excess air provide favourable conditions for low Nitrogen oxide emission. The GPCS is also a two stage combustion which has been developed for low pollution purposes, while the door is open for further research work because of the possibility of making a three stage combustion by introducing the secondary air at two separate levels along the flame path.

(viii) Since the all-year-round energy consumption per unit transportation work is not too different from that of electric power, heat pollution, besides being 1/4 of that of road transport, is same as that of thermal origin, yet with the advantage that it not causes difficulties in warming up river and lake water.

(ix) All the above can be expected to have an improvement factor of = 1,3 when the condensing version of the proposed cycle will come to light.

3. CYCLE CALCULATIONS

Appendices A-1 and A-2 detail full cycle calculations corresponding to a 6000 CV_e engine whose general arrangement is described in Section 4. These are considered necessary to prove that a thermal efficiency of 21% (based on the lower heating value) is possible and not a matter of guess, and the reader is invited to make a check on himself by the way of a thermodynamic exercise (*). The case corresponds to the maximum design rated power at a speed of 100 Km h^{-1} Figures fall slightly on the pessimistic side, and no optimization process has been done other than a first approach based on author's experience and his more or less forturate wisdom. Appendix A-3 details the most important calculations leading to tentative dimensions of the engine spoken about in Section 4.

It will be noted that no use is made of the theorem stating that the final efficiency is the product of the various partial efficiencies of the various elements, while of course the classical "locomotive ratios" are meaningless as yardsticks of locomotive proportions and performance, something which Chapelon showed to be inconsistent (12).

(*) The author will be pleased to answer any question which the reader wishes to formulate Correspondance should be sent to L.D. Porta, Jefe Departamento de Termodinámica, CC157, San Martín, Buenos Aires, Argentina.

Cold tender water consumption comes to be the uncanny figure of 2,565 kg CV_e^{-1} h^{-1} (drawbar power). Besides considering the adding up favourable influence of high pressure and temperature and all other means disposed to achieve a high theoretical and practical cycle efficiency one must realize that in fact a large amount of water recirculates enside the engine or, in power station language, the make up is sizably below 100%, namely, 34,6%.

The thermodynamical key of the high resulting cycle efficiency is that the irreversibility inherent to the entiopy increase of the system associated to heat transfer accross the various heating surfaces without producing mechanical work, has been reduced as much as possible by making that transfer done with the least possible temperature differences across such surfaces. This is a straight consequence of the Second Principle, and of course well known since more than a century ago; yet locomotive engineers made little concient use of it (unlike their power station brothers) because the shining simplicity of the steam locomotive logic of working dazzled them

4. HOW DOES LOOK A LOCOMOTIVE DESIGNED ACCORDING TO THE PROPOSED CYCLE?

Everytime a new locomotive cycle is proposed, much concern arises about the shape of the hardware traducing thermodynamical figures into practical beings running on rails within the severe restriction of size, loading gauge, weight per unit length, axleload, curving, accessibility, manning ability, etc.

Fig. 3 shows the general layout of the exercise whose calculations have been worked out in Appendices A-1, A-2 and A-3. It is the American version of a rated-continuous 6000 drawbar horsepower engine supposed to be designed for what is today called "fast freight traffic" with a maximum speed of 145 km h^{-1} (90 mile h^{-1}). All the mechanical part can comply with AAR rules (13), while the steam generator can be built to the latest ASME Boiler and Pressure Vessel Code. The reader will recognize many familiar features in Fig. 3. Perhaps the most apparent characteristic is that the whole design is an airy one, in which everything finds plenty of space to be accommodated to the greatest convenience and utmost accessibility for inspection and repair: this is to be explained by the fundamental equation of the steam locomotive. All the important parts are quite modest per American Standards, namely:

a. Piston thrust is 50470 kgf = 111200 lbf, hence nearly one half of the latest and highest ever built.

b. Axleload need not to be greater than 26400 kgf = 58200 lbf, far below from the much harder to the track now usual with diesels.

c. Engine weight is = 11000 kgf m^{-1}, hence light on bridges.

The steam generator has a very large grate of 6,7 m^2. At rated output the burning rate is 350 kg m^{-2} h^{-1} (72 lb ft^{-2} h^{-1}) when expressed in terms of "good bituminous coal"; however, since only some 30 to 40% of the combustion air passes through the grate, the scrubbing action is equivalent to that of a burning rate of 350 . 0.40 = 140 kg m^{-2} h^{-1} (29 lb ft^{-2} h^{-1}) which is exceedingly small. This explains why the GPCS has such a wide versatility concerning poor fuels.

The tube bundle has the current tube diameters, yet since the later are purposedly artificially roughned to increase heat transfer (a well known technique of today) they can be extremely short; this is enhanced by the high design outlet gas temperature (571°C, Fig. 1) and the low gas temperature at the tubeplate resulting from a very large firebox heating surface. This reflects into a small drum (barrel) weight (3% of total engine weight as compared to the typical figure) in spite of the high pressure.

Ashes are sent to a large pan on the tender where combustion is completed and later sold for industrial uses (cement, ballast, pavements, etc.).

The economizer is made up of fined tube sections which are easily detachable for inspection and repair; water and gas flow countercurrent. There is no char going to the smokebox nor obstructing the tubes because of the cyclonic separation obtaining in the flame space, hence a naturally selfcleaning smokebox.

The low piston thrust of 50,000 kgf (110,000 lbf) do not require a cast steel locomotive bed (although it is much commended) hence removing the inconvenience inherent to a lost art.

Three separate sets of Southern valve gears have been arranged on the outside so that the inside motion is kept to the minimum essentials. The regulator is of the well-known Hulburd system enforcing a correct driving practice in which the sole possibility for power control is by means of the valve gear, hence avoiding thermodinamical irreversibilities.

The whole locomotive takes the classical form of an engine running chimney first, of course with full bi-directionality regarding speed and power, and the alternative of cab running first is of course possible. Tender capacity depends on the required authonomy: representative figures have been chosen as 5h continuous running at full rater power for the water content and 12h for the 5,000 keal kg^{-1} fuel supply. This results in a total length of 24 m, well within the 90 ft turntable.

It is felt that engine hardware will fall short of the total weight required for adhesion, hence water ballast tanks have been arranged at the boiler sides in the usual way (Porta, REf. (1)).

Fig. 4 shows drawbar power characteristics; power curves are very flat as consequence of the joint application of compounding and utmost internal streamlining. Constant thermal efficiency curves cover a wide area because of the flat boiler efficiency characteristic (GPCS, cyclone, large grate), high superheat at low steaming rates and reduced internal resistance inherent to "fat" indicator diagrams (high mechanical efficiency).

Fig. 5 depicts tractive effort characteristics (drawbar) and the option of a Lewty booster increasesdragging ability at low speeds to the point that pulling performance equals that of the famous UP BIG BOY class, yet within a half sized ironmongery and 1/3 fuel consumption. This will condense 30 years of work after the '50s. A maximum starting tractive effort of 60,000 kgf (130,000 lbf) will allow, according to authors experience, the starting of 10,000 tonne trains in cold weather on the level. It is assumed that all what is known about adhesion utilization technique has been incorporated (Porta (14)).

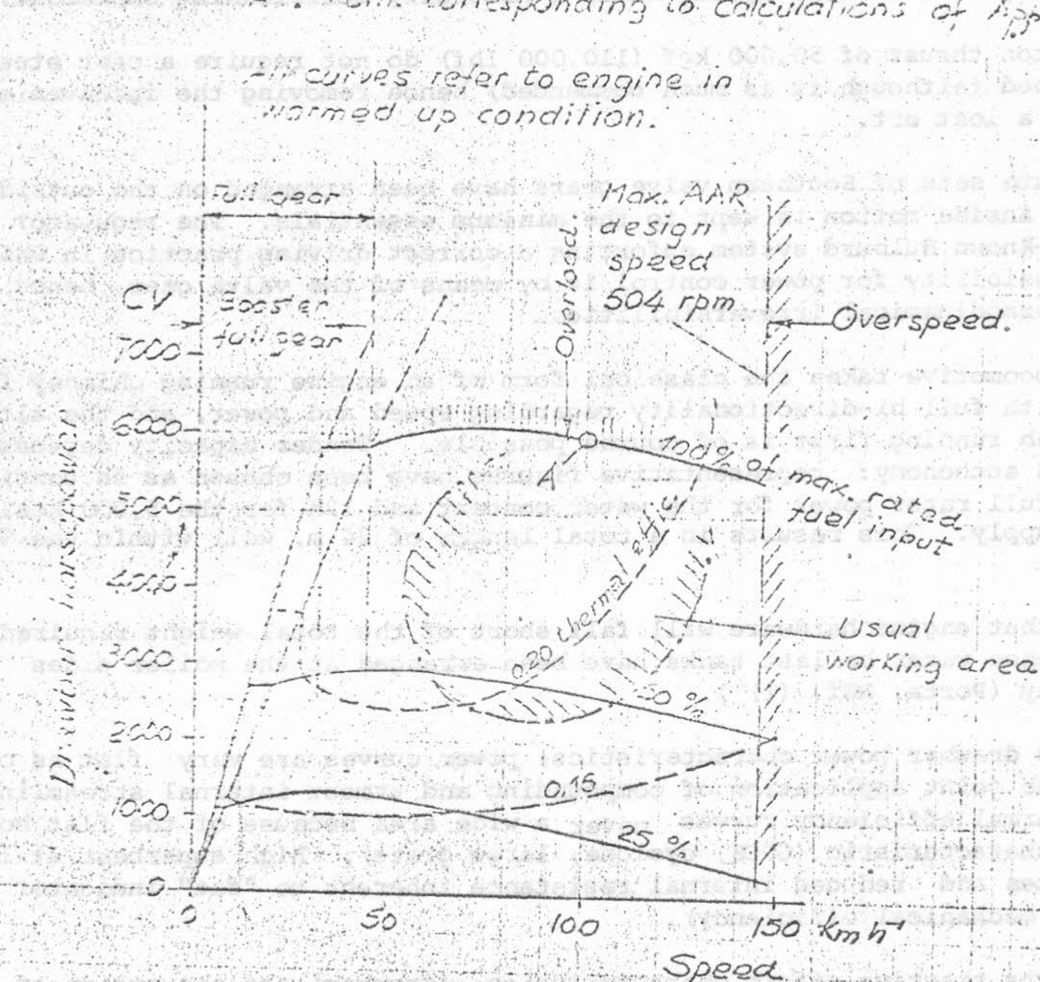

Fig. 4 DRAWBAR POWER CHARACTERISTICS

The flat power curves are a consequence of compounding and utmost internal streamlining. The wide area covered by thermal efficiency curves is the result of the GPCS, cyclonic, large grate area generator and high superheat at low steaming rates.

Overload power corresponds to that obtained by forcing the steam generator and able driving technique. It can perhaps go up to 8000 CV_e corresponding to the American concept of "capacity" power, which in author's philosophy was to be only exceptionallly used.

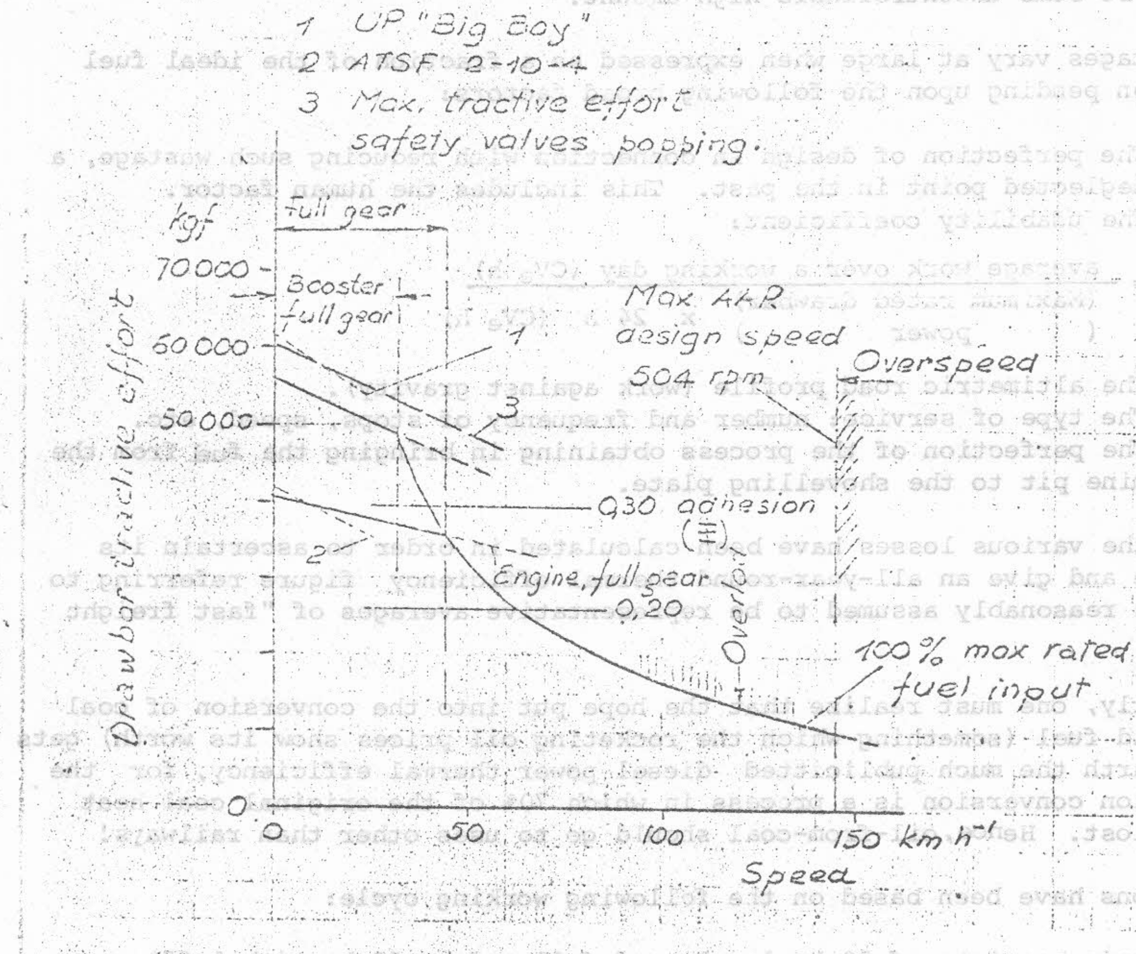

Fig. 5 DRAWBAR TRACTIVE EFFORT CHARACTERISTICS

The low falling off of full-gear lines is characteristic of utmost internal streamlining. Dragging capacity (i.e., the ability to develop high power at low speeds) is enhanced by the optional Lewty booster, in which case author's experience shows that 10,000-tonne trains can be started in cold weather on the level.

5. THE ALL-YEAR-ROUND THERMAL EFFICIENCY

Performance graphs like those of Figs. 4 and 5, obtained either by calculation or test, report thermal efficiency corresponding to the instantaneous working condition. This is a most important value, yet what really count is the ratio between the energy input at the mine pit and the brute mechanical work at the drawbar, which is what is ultimately utilized to produce transportation work. Any form of traction has improductive energy wastages occurring out of the propelling unit itself or even in it when not pulling trains. A typical example is fuel consummed to keep the fire alighted, which the layman always believes to be some uncontrollable high amount.

These wastages vary at large when expressed as a fraction of the ideal fuel consumption pending upon the following broad factors:

- The perfection of design in connection with reducing such wastage, a neglected point in the past. This includes the human factor.
- The usability coefficient:

$$= \frac{\text{average work over a working day } (CV_e\ h)}{\left(\begin{array}{c}\text{Maximum rated drawbar}\\ \text{power}\end{array}\right) \times 24\ h\ \ (CV_e\ h)}$$

- The altimetric road profile (work against gravity).
- The type of service: number and frequency of stops, speed, etc.
- The perfection of the process obtaining in bringing the fuel from the mine pit to the shovelling plate.

However, the various losses have been calculated in order to ascertain its importance and give an all-year-round thermal efficiency figure referring to conditions reasonably assumed to be representative averages of "fast freight services".

Incidentally, one must realize that the hope put into the conversion of coal into liquid fuel (something which the rocketing oil prices show its worth) gets down to earth the much publicitted diesel power thermal efficiency, for the liquifaction conversion is a process in which 70% of the original coal heat value is lost. Hence, oil-from-coal should go to uses other than railways!

Calculations have been based on the following working cycle:

One trip consists of 50 km level road followed by 50 km with 1.25% climbing grade, followed by 50 km with 1.25% downhill. (Total average speed 62.5 km h^{1}), followed by 1/2 h shunting on the road. Four 150 km trips per day (600km day $^{-1}$). 300 working days per year. Yearly kilometrage 180,000 km year^{-1}; traffic availability = 90%; traffic usability roughly 50%. Lighting ups: one every two months; idle kilometrage 5,000 km year $^{-1}$. Starts from standby under steam condition: for per day. Standby hours per day 12, hence 12.300 = 3,600 h year $^{-1}$. Average power on level and uphill sections: 70% of maximum rated drawbar power. Shunting on the road 2 h day^{-1} = 600 h year^{-1}.

5.1 Improductive Fuel Consumptions

a. Warming up period

Chapelon has mentioned a rough figure of 10% increase each time the locomotive starts from cold on a 100 km run (which includes a correlative loss of maximum power). The author's system keeping permanently warmed up the whole cylinder block at boiler saturation temperature reduces this loss to negligible proportions, which is enhanced by an utmost heat inscilation of the whole front end. A figure of 20 kg of fuel per start from cold has been set rather arbitrarily (Porta (15)). Hence:

$20 \text{ kg start}^{-1} . 4 \text{ start day}^{-1} . 300 \text{ day year}^{-1} = 24,000 \text{ kg year}^{-1}$

b. Lighting up

The corresponding consumption is calculated on the basis of the heat required to rise the steam generator metal, the water and the cylinder block to saturation temperature (270C):

$$35,000 \text{ kg} . 0,02 \text{ kcal kg}^{-1} \text{K}^{-1} . (270 - 10) \text{ K} = 1,092,000 \text{ kcal}$$
$$8,000 \text{ kg} . 270 \text{ kcal kg}^{-1} = \underline{2,160,000} \text{ kcal}$$
$$3,252,000 \text{ kcal}$$

The efficiency during lighting up is very high if made under (obligatory) non smoking, forced draught conditions, say 0.90. Hence, 723 kg of 5,000 kcal kg^{-1} fuel, which extremely modest for a 6,000 CV_e locomotive and roughly equivalent to running 12 min. at full rated power. Per year = 6 . 723 = 4,338 kg year $^{-1}$.

c. Keeping the engine under steam when stationay at sheds and stations

So far, a perfectly insulated boiler is a heat accummulator, while keeping the fire reduced to embers consumes but little heat. In current practice, a large amount of heat was wasted to make up for the various steam leaks and also to heat up air passing through not perfectly air tight dampers. The required heat is made up as follows:

- Losses through boiler piping and cylinder circulation (Only during standby)	= kcal h^{-1}	40,000
- Steam leaks, 50 kg h^{-1}	= "	30,000
	"	70,000

Assuming a boiler efficiency of 90%, this goes to 16 kg of 5,000 kcal kg^{-1} fuel per hour. This figure checks with experimental results obtained at Rio Turbio (Argentina) on smaller and rather imperfect locomotives. Hence per year = 3,600 h year^{-1} . 16 kg h^{-1} = 57,600 kg year $^{-1}$.

d. Running idle to and from trains

The internal resistance of the engine running light can be taken 600 kgf, which is quite low because of roller bearings, etc. Running at 50 km h^{-1} this requires a power of 110 CV plus that incurred in accelerating. Obviously the specific consumption is quite high because of throttling. A tentative figure of 15 kg km^{-1} (including kinetic energies) is set. Hence per year:

$15 \text{ kg km}^{-1} . 5,000 \text{ km year}^{-1} = 75,000 \text{ km year}^{-1}$

e. Kinetic energy dissipated when braking to a stop

The running mass of the locomotive in working order with 2/3 supplies 387 = 293,000 kg (including the inertia of rotating parts). Hence the kinetic energy dissipated at each stop is:

Speed	80	100	120	140	km h^{-1}
Kinetic energy	27	43	61	84	CV h
Coal consumption	17	27	39	53	kg

The coal consumption has been calculated taking 0,2 as thermal efficiency to the wheeltreads during starting. Assuming that stops are made from a speed of 100 km h^{-1}, it is, per year:

4 stoy day^{-1} . 300 day $year^{-1}$. 27 kg $stop^{-1}$ = 32,400 kg $year^{-1}$

f. Work against gravity running uphill

For a total locomotive weight of 276,000 kgf = 386, and a daily climb of 2,500 m day, the mechanical work is 2,556 CV h. This corresponds to a consumption of 1,538 kg day $^{-1}$, hence 461,457 kg year $^{-1}$.
(Thermal efficiency to wheelhead assumed 0,21; locomotive weight with 2/3 supplies).

g. Shunting work during shed movements

This was usually a heavy consumption because engines were moved in a not warmed up condition and no expensive working was sought. Thanks to precautions involved in design, this consumption can be set to 1/5th of what would be a usual figure: we take 200 kg/day; hence per year = 200 kg day^{-1} . 300 day $year^{-1}$ = 60,000 kg $year^{-1}$.

h. Shunting work at stations during train services

This is most difficult to estimate but based on the experience of normal engines worked under economic rules a tentative figure of 120 kg h^{-1} is assumed. Hence per year:

600 h $year^{-1}$. 120 kg h^{-1} = 72,000 kg year $^{-1}$.

i. Energy lost in curving

It has been included in the average internal resistance.

j. Energy expended in transporting fuel from the supply point to the point of consumption

It can be assumed that the whole railway works on a steam technology as the here proposed one. A rough figure based on YS coal consumption is calculated ($120 \frac{\text{lb}}{\text{1000 USton mile}}$ = 37 kg/1000 t km), taking in consideration the improved thermal efficiency (0,20 against 0,06), the tare factor (=2), the heat value of the fuel (7,000 kcal kg^{-1}/5,000 kcal kg $^{-1}$).

It is:

$$37 \frac{0,06}{0,20} \; 2 \; . \; \frac{7000}{5000} = 31 \text{ kg/1000 t km}$$ (calculated on the large side)

For a distance of 1000 km, this makes a consumption of 3,1% of the fuel burned on the engine, incidentally a lower percentage than that obtaining in transporting electricity over the same distance.

k. Penalty incurred because working in somewhat run down condition

This has been a heavy expenditure in the past because of (i) the poor tightness of piston rings and (ii) the ability of the steam locomotive to support abuse. The former problem has been successfully attacked by the author by the simple expedient of adopting "diesel quality" piston rings while the second stems to keeping a tight maintenance discipline and improved detail design.

A factor of 1.025 is adopted as average, which means that the worst engine of the fleet will perform to within 5% than the best one.

l. Energy lost in mining and fuel washing and loading on the tender

This has been assumed to be 4% of that in fuel utilized. (Incidentally in the oil industry some 10% is utilized at the refinery)

m. Loss due to improper work of feed water heating apparatus

Besides that these apparatus cannot be worked in any other for than that intended, the loss is reduced to a minimum because of the need to feed the boiler when the engine is not steaming has been reduced to a minimum. The whole feed water system benefits from the idea involved in the Italian Franco economizer, that is that cold water fed during standby is not immediately fed to the boiler but stored in the economizer, therefy offering an additional possibility of increased temperature drop to the flue gases immediately after normal steaming is reassumed. A pessimistic 2% increase is assumed.

n. Losses due to the human factor

So far, the quoted figures suppose that driving practices are at their best. It is known that men other than the best incur in extra fuel and water consumptions which were very large in past times, the more the motive power was away from the best possible standard condition. In good traditional practice the difference could be up to 10%. The author thinks that more sophisticated engines reduce this margin provided that adequate means are inbuilt into the design so that the chances for incorrect operation are least. In which case the above figure should be no more than 5%, hence to a factor of $1+\frac{0.05}{2} = 1{,}025$ for the average.

However, losses obtaining because of pulling trains doing a mechanical work higher than the minimum actually possible are far greater, but we decide not to put them into the bill; although either electric or diesel traction, being "push button" machines, are more prove to such waste.

Assuming that the locomotive works 1.6 h at 70% of maximum rated power during each cycle (i.e. = 0.7 5800 CV_e = 4060 CV_e) the work performed per day is 4060 CV_e. 1,6 h . 4 day^{-1} = 25,984 CV_e h day^{-1}. This is the drawbar work neglecting the various inproductive consumption as listed above. If a thermal effiency of 0,205 is taken for the usual working range (Fig 4), the corresponding consumption per CV_e is:

$$\frac{632{,}5 \text{ kcal } CV^{-1}h^{-1}}{5000 \text{ kcal } kg^{-1} \; 0{,}205} = 0{,}617 \text{ kg } CV^{-1} \; h^{-1}$$

The yearly consumption will be

$25{,}984 \; CV_e \; h \; day^{-1} \; 0{,}617 \; kg \; CV^{-1} \; h^{-1} \; . \; 300 \; day \; year^{-1} =$
$4{,}809{,}638 \; kg \; year^{-1}$

Hence the various losses proportional to this consumption as orderly represented by the following factors:

- run down condition	1,025
- improper functioning of feed water heating apparatus	1,020
- improper driving practices	1,025

It is 1,025 . 1,020 . 1,025 = 1,072

Hence 4,809,638 kg year 1,072 = 5,155,931 kg $year^{-1}$ which added to the other losses of 786,795 kg $year^{-1}$ make 5,942,726 kg $year^{-1}$.

Other improductive losses factors are:

- Energy for transport at 100 km distance	1,031
- Fuel mining and preparation	1.04

Hence the grand total is

1,031 1.04 5,942,726 kg year = <u>6,372,028 kg $year^{-1}$</u>

The percentage increase incurred in improductive consumptions is:

$$\frac{6{,}372{,}028 - 4{,}809{,}638}{4{,}809{,}638} = 32{,}4\%$$

Accepting that the no loss efficiency is 0,205, the ultimate efficiency between the mine pit and the drawbar is 0,154 (based on the lower heating value). According to whether the operating conditions are more or less favourable, it can vary between $\pm$ 20% limits, namely 0,184 and 0,123.

So far, the purpose of the above calculation is to show that the various non ideal conditions obtaining in the transformation of the energy from the coal mine down to the drawbar hook (excluding washery and mining losses) is some 20 to 40% lower than what can be obtained with the locomotive in its best test, fully warmed up, condition. This includes mining and coal preparation energy, coal transport and distribution at 1000 km distance from the mine, the various losses classically associated to steam locomotion (imperfections, moving itself, braking, work against gravity, human factor, etc.).

6. WATER TREATMENT

It is assumed that the reader is current with the fact that with pressures up to 21 kgfcm^{-2} water treatment was a terminated question for non-condensing locomotives, so that the interest of condensation because of the need to get rid of sealing, corrosion, caustic embrittlement and steam contamination no longer exists and a nearly zero maintenance situation was achieved, particularly with the TIA in France. Internal carbonate treatment was found to be the answer, while the author and his INTI - FCGB proceeded further on developing for the Belgrano Ry. (Argentina) a much ruder treatment allowing a çoarser control by illiterate people, uncanny alka-linites, high silica waters containing suspended clay, etc., to be perfectly tolerated. This was was carried out in 1973, and of course horrified the traditional chemist. Although stationary boiler practice shows increased problems on the water side at increasing pressures, it is the author's convictions that the same approach can still provide an answer at 61 kgf cm^{-2}, essentially because the Sthepensonian steam generator is infinitely more tolerant than any power station boiler; besides, the reciprocating engine does not demand the ultra-high steam purity of its turbine power station counterpart, while railway service include traffic halts during which - should the case be - maintenance work can be carried out quite easily. This conviction is supported by the new knowledge available in water treatment matters concerning the particular steam locomotive field (Porta (17)) in which full exploit is made of the above spoken peculiarities inherent to this machine; some past research work also points to the possibilities of the internal carbonate treatment (Thurstia (18)), an opinion also shared by Richardson (19). Of course some research work will be required: (i) on the life and ability preservation of organic antiframing compounds at 270°C; (ii) on the use of CO_2 to keep under control the carbonate dissociation in boiler water, etc.

On the whole, one must remind that direct water treatment costs have never been a large expense. However, since the tender water consumption will be some three times smaller per unit transportation work as compared to first generation steam, water treatment costs will be reduced in the same proportion. If no recourse is made to phosphates, a point is gained in favour since these tend to become critical materials.

7. WHAT ABOUT THE DIESEL AND THE ELECTRICS?

This is the timely question. Diesel power, besides its recognized dragging ability, put the accent on a considerably higher thermal efficiency as compared to (old) steam for the same all-year-round traffic work. It seems ligetimate to explore the present case calculating the all-year-round thermal efficiency on the basis of the traffic model depicted in Section 5. Same for electric traction.

7.1 Diesel traction

P. Kiefer (22) brilliantly demonstrated in a large scale experiment the practical one-to-one equivalence between diesel and steam power (we refer to the famous ALCO-NYC NIAGARAS). Hence the application of the above spoken model is valid. The working cycle is same, namely: One trip consists of 50 km level road followed by 50 km with 1,25% climbing grade, followed by 50 km with a 1.25% downgrade (Total average speed 62.5 km h^{-1}), followed by 1/2 h shunting on the road. Four 150 km trips per day (600 km day^{-1}), 300 working days per year^{-1} ; Yearly kilometrage 180,000 km year^{-1}; traffic availability = 90%; traffic usability, roughly 50%; idle kilometrage 5,000 km year^{-1}. Standby hours per day, 12, of which 4 with the engine running idle, hence 4 h. 300 day year^{-1} = 2,800 h year^{-1} .

7.1.1. Diesel Improductive fuel consumptions

English Electric Deltic was selected for companion, 2,7 units considered equivalent to the sketched out 6000 CV_e steam proposal. At 100km h^{-1}, the Deltic produced 2530 CV_e on the last diesel notch, but a conservative rating equivalent to the above steam figure (which is not "capacity" power) is 2200 CV_e : hence the 2,7 units. The performance is discuted in Ref(23).

a. Keeping the engine idle 2,800 h $year^{-1}$

Idle fuel consumption has been taken 100 kg h^{-1}. Hence, 100 kg h^{-1} . 2800 h $year^{1}$ = kg $year^{-1}$ 280,000 (fuel of 10,100 kcal kg^{-1},l.h.v.).

b. Running idle to and from trains

A tentative figure of 3 kg km^{-1} has been taken. Hence 3 . 5000 = kg $year^{-1}$ 15,000.

c. Kinetic energy dissipated when braking to stops

The running mass of 2.7 Deltic units is 292 tonnes. Hence, by comparison to the steam figure, assuming a thermal efficiency of 0,26 during (head) starting (including traction motor losses at low speed) it is:

$$32{,}400\ \text{kg year}^{-1} \cdot \frac{292\ \text{tf}}{276\ \text{tf}} \cdot \frac{5000\ \text{kcal kg}^{-1}}{10100\ \text{kcal kg}^{1}} \cdot \frac{0{,}205}{0{,}25} = 13911\ \text{kg year}^{-1}$$

d. Work against gravity when running uphill

By comparison with steam, taking a diesel efficiency figure of 0,275 at the wheeltread, it is·

$$461{,}457\ \text{kg year}^{-1} \cdot \frac{5000\ \text{kcal kg}^{-1}}{10100\ \text{kcal kg}} \cdot \frac{292\ \text{tf}}{276\ \text{tf}} \cdot \frac{0{,}205}{0{,}275} =$$

$$180{,}163\ \text{kg year}^{-1}$$

e. Shunting work during shed movements

An arbitrary figure of 100 kg day is assumed, hence 100.300 = 30,000 kg $year^{-1}$.

f. Energy expended in transporting fuel from the supply point to the point of consumption

It is assumed that the whole railway operates on diesel power. A tentative figure is 17 kg/1000 tkm, which is better than the general railway figure because of the various favourable factors.

g. Penalty incurred because of run-down condition

Experience tells that the difference in performance between the best and the worst unit of a diesel fleet is at least 10%. Hence a multiplier of 1.05.

h. Energy lost in mining and preparation

Refining operation consumes at least 8% of the oil input. A conservative figure of 12% should include oil field consumption.

i. Loss due to the human factor

The same reasoning as for steam is valid, although here, being the diesel a "push-buttom" machine, gives no possibility of incorrect operation. However, the tendency for use of more energy than what is strictly necessary is more sensible as proven by a daily experience: diesel trains run faster on the same timetable. Hence a comparative factor of 1.05 is adopted.

The various consumptions listed above are:

- engine running idle	kg year^{-1}	280,000
- idle to and from trains	"	15,000
- kinetic energy	"	13,911
- work against gravity	"	180,163
- shunting on sheds	"	30,000
Total	kg year^{-1}	519,074

As for steam, the drawbar mechanical work is 25984 CV_e day^{-1}. From graph 4, Ref. (23), the thermal efficiency (measured on test) of the Deltic can be taken as 0,274 over the range of interest. Hence the fuel consumption per CV_e is:

$$\frac{632{,}5 \text{ kcal } CV^{-1} \text{ h}^{-1}}{10{,}100 \text{ kcal kg}^{-1} \cdot 0{,}274} = 0{,}229 \text{ kg } CV_e \text{ h}^{-1}$$

Hence per year is:

$$25{,}984 \; CV_e \text{ day}^{-1} \cdot 0{,}229 \text{ kg } CV_e^{-1} \text{ h}^{-1} \cdot 300 = 1{,}785{,}101 \text{ kg year}^{-1}$$

The various sensibly losses proportional to the later quantity are:

- fuel transport over 1000 km distance 1.7%	kg year^{-1}	30,347
- run down condition, 5%	"	89,225
- fuel preparation, 12%	"	214,212
- improper driving practicy penalty, 5%	"	89,255

All added up the various losses amount to 942,143 kg year^{-1}, which is 52,7% of the above calculated figure of 1,785,101 kg year^{-1}. Therefore, the brute thermal efficiency for the Deltic is 0,274 : 1,527 = 0.18. One may assume that, under the oil crisis shortage, the above value (referring to a now 20-year old engine) will be improved. Hence we adopt 0,20 as diesel power thermal efficiency between the oil field and the drawbar hook.

7.2 Electric traction

Some tentative values will suffice for the purpose of the paper. It is assumed that the same 5000 kcal kg^{-1} is burned on a power station situated at the mine pit and transported 1000 km to the railway substation.

Transportation and kilometrage quantities are same as for diesel traction.

7.2.1 Improductive energy losses

a. Mining and preparation energy

As for steam, 4%.

b. Running idle to and from trains

A tentative figure of 5 kwh km^{-1} is assumed, hence 5000 km $year^{-1}$. 5 kwh km^{-1} = 25,000 kwh $year^{-1}$

c. Kinetic energy dissipated when braking to stops

The equivalent mass of the locomotive has been taken to be 169 tonnes. Assuming stops from a speed of 100 km h^{-1}, four tonnes per day, the dissipated energy is about 24,6 CV_e h. 4.300 day year = 29,530 CV_e h $year^{-1}$ = 21,735 kwh $year^{-1}$.

d. Work against the gravity

Since the cycle requires four 625 m climbs/day = 750,000 m year, for a locomotive weight of 130 tf the mechanical work is 266,000 kwh $year^{-1}$.

e. Shunting work during shed movements

Tentatively it is taken 50 kwh day^{-1}, hence 15,000 kwh $year^{-1}$.

f. Shunting work at stations during train services

A tentative figure is taken as = 120,000 kwh $year^{-1}$.

g. Power station efficiency to bus bars

Assumed to be 0,37.

h. Transmission line efficiency

Along 1000 km, down to substation, is assumed 0,95.

i. Substation, catenary and return line efficiency

Assumed 90%.

j. Locomotive efficiency between catenary and drawbar

Assumed 77%.

The total efficiency between the mine and the drawbar, excluding improductive current consumption, therefore is:

$$0,96 \cdot 0,37 \cdot 0,95 \cdot 0,90 \cdot 0,77 = 0,234$$

The total mechanical work to be developed is 25,984 CV_e h day^{-1}.

This corresponds to a fuel consumption at the mine pit of

$$\frac{632{,}5 \text{ kcal } CV_e^{-1} h^{-1}}{5000 \text{ kcal kg}^{-1} \; . \; 0{,}234} = 0{,}541 \text{ kg } CV_e^{-1} \; h^{-1}$$

Hence per year it is:

$$25{,}984 \; CV_e \; h \; day^{-1} \; . \; 0{,}541 \text{ kg } CV_e^{-1} \; h^{-1} \; . \; 300 \text{ day year}^{-1} = 4{,}217{,}203 \text{ kg year}^{-1}$$

Various losses are to be added:

- Running idle

$$\frac{25{,}000 \text{ kwh year}^{-1} \; . \; 860 \text{ dcal kwh}^{-1}}{5{,}000 \text{ kcal kg}^{-1} \; . \; 0{,}234} = 18{,}376 \text{ kg year}^{-1}$$

- improper driving penalty, 5%	= 210,860 "
- kinetic energy losses 21,735 kwh^{-1}, equivalent to	15,975 "
- work against gravity 266,000 kwh year^{-1} equivalent to	195,521 "
- shunting work at the shed	603 "
- shunting on the road	60,000 "
Total	519,711 "

which added to the previous figures make a grand total of 4,736,914 kg year^{-1} Hence, the total all-year-round thermal efficiency between the mine pit and the drawbar is:

$$\frac{25{,}984 \; CV_e \; h \; day^{-1} \; . \; 300 \text{ day year} \; . \; 632{,}5 \text{ kcal } CV_e^{-1} \; h^{-1}}{4{,}736{,}914 \text{ kg year}^{-1} \; . \; 5000 \text{ kcal kg}^{-1}} = 0{,}208$$

Assuming further progress in the coming years, we take 0,23 for electric traction.

7.1.3 Summary of thermal efficiencies

These are to be understood between the mine pit (or the oil field) and the drawbar traffic-producing mechanical work on "fast freight service":

a. future electric traction of thermal origin	0,23 ± 5%
b. future diesel power (oil burning)	0,20 ± 5%
c. third generation steam condensing below atmospheric pressure, probably	0,19 ± 20%
d. presently proposed non condensing third generation steam	0,154 ± 20%

8. CONCLUSION TO PART I

So far, thermal efficiency - now becoming of utmost importance - has been the prime consideration here to prove that better figures were possible within the frame of a practical, traffic producing and revenue earning proposal. Triplicating the all-year-round thermal efficiency of the best steam power known in America by the '50s is something that cannot be dropped unless counterproving its falsehood. Yet the story do not terminates here, for a locomotive is something more than mere thermodynamics: this will be shown in Part II.

It is felt that there is room for further cycle simplification and improvement if a fully regenerative cycle is adopted when rising the pressure of the working medium; which can be along the lines initiated by Anderson and by Holcroft (20) under the name of "Condensing by Compression". The importance of the thing was not seen at that time, nor its thermodynamics studied, until, like many times happened along locomotive history, teething troubles and lack of understanding shelved the experiment. Basically, its worth is understood when looking at the entropy-temperature diagram (Fig. 6).

Another coming progress is using the atmosphere as a heat sink at a temperature below 100°C by condensing at sub-atmospheric pressure. This will allow a net gain of about 25% in the efficiency, hence leading to a further smaller heat generation hardware for the same power. Recent compact heat exchanger technique (Kays and London, (21)) coupled to a heat dissipation requirement divided by four will reduce the otherwise large condensing tender which, for example, shows the SAR 25 class.

In Section 7 it was proven that the thermal efficiency of the proposed scheme was on all-year-round basis, not too different from that of diesel or electric power. Yet that result was obtained without recourse to rocketing, hysteria producing, oil prices, nor demanding the huge investments of electrification and burning, if necesary, the same fuel of the latter.

There remains, of course, the problem of the unknowns. These were the terror in the past, hence hindering the quota of credibility required to support the materialization of any proposal. So far, as it has already been said, the unknowns have been reduced to a minimum, but are still to be accounted for, otherwise the present proposal would belong to "Second Generation" Steam. Hence a required research program which will be greater the more ambitious the proposal is.

It can be expected that the following technical areas will demand attention:

(i) Complete furnace, heat transfer theory so as to disclose local uneven heat flow, expansions, water circulation, tube plate behaviour, etc.

(ii) As usual, the usual amount of time consumming, "desperating" teething troubles.

(iii) Development of a mass production, component replacement maintenance scheme as an alternative to the old lost art.

(iv) Water treatment special problems.

(v) Automation, one manning, slave locomotives, etc.

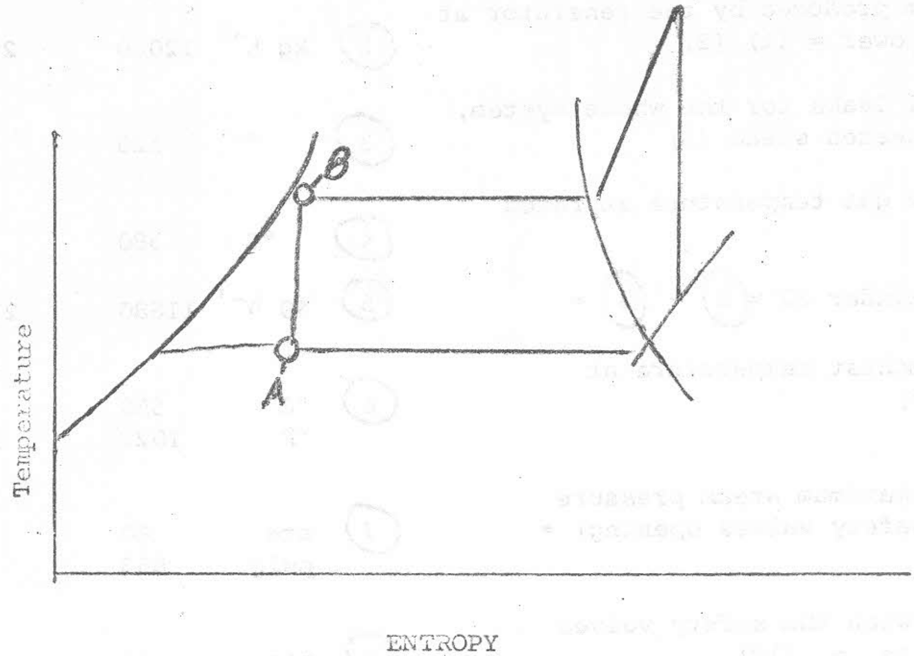

Fig. 6 Regenerative cycle known as "Condensing by Compression"

Anderson patents, applied by Holcroft to a S.Ry locomotive (20) disclosed that very little mechanical work was required to increase the working fluid pressure from A to B.

APPENDIX A-1

CYCLE CALCULATIONS - NON CONDENSING

(Numbers included in parenthesis refer to comments made in Appendix A-2)
(8) (28) (29) (30)

Approximated rated drawbar power	1	CV_e	3000	6000
Saturated steam produced by the generator at maximum rated power = (1) (2)	2	kg h^{-1}	12000	24000
Assumed general leaks for the whole system, taken as superheated steam (3)	3	"	120	240
Design smokebox gas temperature at rated power (16)	4	°C	380	380
Steam to HP cylinder 22 = 2 - 3 =	5	kg h^{-1}	11880	23760
Design HP steamchest temperature at rated power (17)	6	°C °F	550 1022	550 1022
Nominal design maximum steam pressure (beginning of safety valves opening) =	7	ate psig	60 853	60 853
Steam pressure when the safety valves are fully popping = (18)	8	ate	63	63
Average running steam generator pressure =	9	"	56	56
Pressure drop across the superheater	10	at	6	6
HP steamchest pressure (absolute) 9 - 10 + 1 at =	11	ata	51	51
LP steamchest eutholpy (VD1 steam tables) '25'=	12	kcal kg^{-1}	847,4	847,4
Design feed water temperature at feed heater 28 outlet = (19)	13	C	200	200
Feed water eutholpy corresponding to 13 =	14	kcal kg^{-1}	203,3	203.5
Eutholpy increase of 1 kg of steam 12 - 14 =	15	kcal kg^{-1}	643,9	643,9
Heat to steam = 15 . 2 =	16	10^6kcal h^{-1}	7.727	15.454
Heat radiated by generator, ashpan and piping (estimated) = (4)	17	"	0,038	0,072
Total heat passing through generator heating surfaces = 16 + 17 =	18	"	7,763	15,526
Assumed lower heating valve of the fuel loaded on the tender =	19	kcal kg^{-1}	5000	5000

Design excess air coefficient = (6)	20	kcal kg^{-1}	1,2	1,2
Combustion gas produced per kg of fuel really burned (Rosin '26' '27' diagram D16) =	21	(9) Nm^3 kg^{-1}	7.20	7.20
Minimum air required per kg of fuel burned (function of 21, (Rosin 27, diagram D16) =	22	"	5,60	5,60
Air per kg of fuel burned 20 . 22 =	23	"	6,72	6.72
Steam to coal bunker 38 for wetting purposes, estimated =	24	kg h^{-1}	= 90	= 180
Steam to ashpan 16 for the GPCS, estimated =	25	"	= 600	= 1200
Steam 14 to help swirl in the furnace 13, estimated =	26	"	120	240
Sum 25 + 26 =	27	"	720	1440
Estimated average entholpy of steam 27 =	28	kcal kg^{-1}	650	650
Eutholpy of steam 27 at smokebox temperatura 4 =	29	"	772	772
Heat involved in heating steam 27 = 27 . 29 - 28 =	30	10^6 "	0,088	0,176
Air temperature after heater 6, by design =	31	°C	200	200
Air ambient temperature =	32	"	12	12
Fraction of combustion air passing through heaters 6 to 9 (assumed)	33	-	0,95	0,95
Fraction of air entering through the bottom ashpan grate 19 = 1 - 33 =	34	-	0,05	0,05
Mean air temperature when entering the steam generator system = 31 . 33 + 32 . 34 =	35	c	191	191
Air enthalpy at temperature 35 from the it diagram '27' '28' =	36	kcal Nm^{-3}	59	59
Combustion gas volume, per hour (7)	37	Nm^3 h^{-1}	To be calculated	ditto
Heat input due to heat in combustion air referred to 1 Nm^3 of combustion gas= $\frac{23}{21} \cdot 37 \cdot 36 \cdot \frac{1}{37}$ =	38	kcal Nm^3	55	55
Heat input due to combustion referred to 1 Nm^3 of combustion gas = 19 : 21 =	39	"	694	694

Enthalpy of combustion gases at smokebox temperature 4 , from the it diagram = (7)	40	kcal Nm^3	128	128
Enthalpy drop of 1 Nm^3 of combustion gas accross the generator = 39 + 38 - 40 = 41	41	"	621	621
Heat transpassing the heating surfaces 18 + heat involved in heating steam 30 = 18 + 30 =	42	10^6 kcal h^{-1}	7,850	15,701
Volume of combustion gas per hour = 42 : 41 = 37 =	43	Nm^3 h^{-1}	12,642	25,283
Fuel really burned per hour = 43 : 21 =	44	kg h^{-1}	1,756	3,511
Combustion gas density NTP (Ref 27), (diagram D14), =	45	kg Nm^{-3}	1,315	1,315
Mass of combustion gas per hour = 45 . 43 =	46	kg h^{-1}	16,624	33,247
Mass of steam, 24 + 25 + 26 =	47	"	810	1,620
Total mass of gases = 46 + 47 =	48	"	17,434	34,867
Specific volume of steam 47 at NTP =	49	Nm^3 kg^{-1}	1,245	1,245
Volume of steam 47 = 47 . 49 =	50	Nm^3 h^{-1}	1,008	2,017
Total gas volume NTP = 50 + 43 = (per hour)	51	"	13,650	27,300
Density of total gas, NTP = 48 : 51 =	52	kg Nm^{-3}	1,277	1,277
Assumed combustion efficiency=(10)	53	-	0,97	0,97
Total fuel fired per hour = 44 : 53 =	54	kg h^{-1}	1,809	3,619
HP cylinder inlet steam flow = 5	55	"	11,880	23,760
Design HP exhaust steam pressure (receiver I pressure) (11)	56	ata	20	20
Design MP cyl exhaust steam pressure (receiver II) (11)	57	"	7	7
Saturation temperature corresponding to 56 = (also at heater 6)	58	°C	211	211
Saturation temperature corresponding to 57 = (also at heater 7)	59	"	164	164
Mean exhaust steam pressure during exhaust stroke, assumed; =	60	ata	1.17	1.17
Atmosphere pressure at 500 m height over sea level =	61	"	0,97	0,97
Steam pressure at heater 9 (after the non return valve 19) = (12) =	62	"	1,3	1,3

Saturation temperature at heater 9 =	63	°C	107	107
Steam pressure at heater 8 (after the non return valve 19) = (13)	64	ata	= 4	= 4
Saturation temperature at heater 8 =	65	°C	143	143
Design air temperature after heater 6 = 31 =	66	"	200	200
Design air temperature after heater 7 =	67	"	150	150
On the it diagram (Ref. 26), air enthalpy at temperature 66 =	68	kcal Nm^{-3}	61.5	61.5
Same at temperature 67 =	69	"	46.0	46.0
Enthalpy rise in heater 6. = 68 - 69 =	70	"	15.5	15.5
Total combustion air per hour = 43 . $\frac{23}{21}$ =	71	Nm^3 h^{-1}	11798	23597
Air passing per hour through air heaters = 71 . 33 =	72	"	11209	2241
Heat transmitted to air in heater 6 = 72 . 70 = (14)	73	kcal h^{-1}	173735	34747
Enthalpy of exhaust steam of the HP cylinder (Fig. A1.1) (Receiver I)	74	kcal kg^{-1}	779.0	779.0
Enthalpy of steam 56 after giving up its heat in heater 6, at saturation temperature 58 =	75	"	215.8	215.8
Enthalpy drop in heater 6 = 74 - 75 (32) =	76	"	563.2	563.2
Steam condensed in heater 6 = 73 : 76 =	77	kg h^{-1}	308	617
MP cylinder exhaust steam enthalpy receiver II (Fig A1.1)	78	kcal kg^{-1}	714.6	714.6
Enthalpy of steam 57 after giving up its heat in heater 7, at saturation temperature 59 =	79	"	165.6	165.6
Enthalpy drop in heater 7 = 78 - 79 =	80	"	549.0	549.0
Air temperature after heater 8, by design	81	°C	135	135
On the it diagram (Ref. 26), enthalpy of air at temperature 81	82	kcal Nm^{-3}	41.5	41.5
Enthalpy rise of air in heater 7 = 69 - 82 =	83	"	4.5	4.5
Heat to air in heater 7 = 85 . 72 =	84	kcal h^{-1}	50439	100877
Enthalpy drop of condensate coming from heater 6 = 75 . 79 =	85	kcal kg^{-1}	50.2	50.2

Id per hour = 85 . 77 =	86	kcal h^{-1}	15482	30963
Heat to be furnished by steam in heater 7 = 84 - 86 =	87	"	34957	69914
Steam condensed in heater 7 = 87 : 80 =	88	kg h^{-1}	64	127
Approximate enthalpy of steam to heater 8, at pressure 64 . =	89	kcal kg^{-1}	688	688
Enthalpy of steam condensed in heater 8, at temperature 65 =	90	"	143.7	143.7
Enthalpy drop of steam condensed in heater 8. = 89 - 90 =	91	"	544.3	544.3
Enthalpy of air at temperature 81, after heat 8 (it diagram) =	92	kcal Nm^{-3}	41.5	41.5
Design temperature after heater 9 =	93	°C	80	80
Enthalpy of air at temperature 93 on the it diagram	94	kcal Nm^{-3}	24.5	24.5
Enthalpy rise of air in heater 8 = 92 - 94 =	95	"	17	17
Heat to air in heater 8 = 95 . 72 =	96	kcal h^{-1}	190546	381092
Enthalpy drop of condensate 88 in heater 8 = 79 - 90 =	97	kcal kg^{-1}	21.9	21.9
Heat given up by condensate coming from heater 7 = 88 . 97 =	98	kcal h^{-1}	1393	2786
Heat given up by steam in heater 8 = 96 - 98 =	99	"	189154	378307
Steam condensed in heater 8 = 99 : 91 =	100	kg h^{-1}	347	695
Ambient temperatura 32 =	101	°C	12	12
Air enthalpy at temperature 101 =	102	kcal Nm^{-3}	3.6	3.6
Enthalpy rise of air in heater 9 94 - 102 =	103	"	20,9	20,9
Heat to air in heater 9 = 103 . 72 =	104	kcal h^{-1}	234260	468519
Condensate enthalpy at saturation temperature in heater 9 63 =	105	kcal kg^{-1}	107.1	107.1
Enthalpy drop of condensate coming from heater 8 = 90 - 105 =	106	"	36.6	36.6
Heat given up by condensate coming from heater 8 = 106 . 100 =	107	kcal h^{-1}	12715	25430
Heat given by steam condensed in heater 9 = 104 - 107 =	108	"	221545	443090

Approximate enthalpy of steam at pressure 62 =	109	kcal kg^{-1}	650	650
Enthalpy drop of steam in heater 9 = 109 - 105 =	110	"	542.9	542.9
Steam condensed in heater 9 = 108 : 110 -	111	kg h^{-1}	408	816
Total condensate coming from the various heaters = 77 + 88 + 100 + 111 =	112	"	1127	2255
Design temperature of water after heater 28 = 13 = (20)	113	°C	200	200
Design temperature of water after heater 27 =	114	"	155	155
Enthalpy of water at temp. 113 =	115	kcal kg^{-1}	203.5	203.5
Enthalpy of water at temp. 114 =	116	"	156.1	156.1
Enthalpy rise of water in heater 28 = 115 - 116 =	117	"	47.4	47.4
Heat to water in heater 28 = 117 . 2 = (21)	118	kcal h^{-1}	568800	1137600
Enthalpy of steam entering heater 28 = 74 =	119	kcal kg^{-1}	779.0	779.0
Enthalpy of condensate at saturation temperature 58 =	120	"	215.4	215.4
Enthalpy drop of steam in heater 28 = 119 - 120 =	121	"	563,4	563,4
Steam condensed in heater 28 = 118 : 121 =	122	kg h^{-1}	1009	2018
Design temperature of feed water after heater 26 =	123	°C	135	135
Enthalpy of water at temperature 123 =	124	kcal kg^{-1}	135.5	135.5
Enthalpy rise of water in heater 27 = 116 - 124 =	125	"	20,6	20,6
Heat to water in heater 27 = 125 - 2 =	126	kcal h^{-1}	247200	494400
Enthalpy of condensate at saturation temperature 58 & pressure 57 = 79 =	127	kcal kg^{-1}	165.6	165.6
Enthalpy drop of condensate 122 in the heater 27 = 120 - 127 =	128	"	49.8	49.8
Heat given up by condensate 122 in the heater 27 = 128 - 122 =	129	kcal h^{-1}	50258	100516
Heat to be supplied by steam feed 35 = 126 - 129 =	130	"	196942	393883

Enthalpy of steam entering heater 27 = 78 =	131	kcal kg^{-1}	714.6	714.6
Enthalpy of condensate at saturation temperature in heater 27 = 127 =	132	"	165.6	165.6
Enthalpy drop of steam 35 in heater 27 = 131 - 132 =	133	"	549.0	549.0
Steam condensed in heater 27 = 130 : 133 =	134	kg h^{-1}	359	717
Condensate issuing from heater 27 into heater 26 = 122 + 134 =	135	"	1368	2736
Design temperature of water getting out of heater 25 = temp. entering heater 26 =	136	°C	96	96
Enthalpy of water at temp. 136 =	137	kcal kg^{-1}	96	96
Enthalpy rise of feed water across heater 26 = 124 - 137 =	138	"	39.5	39.5
Heat to feed water in heater 26 = 138 . 2 =	139	kcal h^{-1}	474000	948000
Enthalpy of condensate at pressure 64 =	140	kcal kg^{-1}	143.6	143.6
Enthalpy of condensate entering the heater 26 = 132 =	141	"	165.6	165.6
Enthalpy drop of condensate entering heater 26 = 141 . 140 =	142	"	22.0	22.0
Heat given up by condensate entering heater 26 = 142 . 135 =	143	kcal h^{-1}	30096	60192
Heat to be supplied by steam 33 = 139 - 143 =	144	"	443904	887808
Enthalpy of exhaust steam 33 = 89 =	145	kcal kg^{-1}	688	688
Enthalpy drop of steam 33 in heater 26 = 145 - 140 =	146	"	544.4	544.4
Steam 33 condensed in heater 26 = 144 : 146 =	147	kg h^{-1}	815	1631
Total condensate getting out of heater 26 = 147 + 135 =	148	"	2183	4367
Temperature in tender tank 37, assumed =	149	°C	10	10
Enthalpy of tender water, as per 149 =	150	kcal kg^{-1}	10	10

Applying the various mass and heat balances to heater 25, it results (Sub-Appendix ......)

Water from tender tank =	151	kg h^{-1}	7769	15539

Condensate issuing from heater 25 =	152	kg h^{-1}	3103	6206
Enthalpy of water after the mixing cones 24 =	153	kcal kg^{-1}	44	44
Temperature of water after heating cones 24, from 153 =	154	°C	44.1	44.1
Steam 34 condensed in heater 25 =	155	kg h^{-1}	920	1840
Feed pump calculations (22)				
Design feed pump discharge pressure (33)	160	ata	65	65
Feed pump suction pressure estimated (15)	161	"	5	5
Hydraulic work $\frac{(160 - 161) \cdot 2 \cdot 0{,}001\ m^3\ kg^{-1}}{3600\ s\ h^{-1} \cdot 75\ kgf\ m\ s^{-1}\ CV^{-1}}$ =	162	CV	26.6	53.3
Maximum design pump capacity =	163	kg h^{-1}	12000	24000
Design pump speed, assumed =	163	strokes per min	200	200
Theoretical cylinder capacity $\frac{163 \cdot 1\ dm^3\ kg^{-1}}{60\ min\ h^{-1}\ 200\ stroke\ min^{-1}}$ =	164	$\frac{dm^3}{stroke}$	1.00	2.00
Tentative cylinder dimensions, (theoretical), stroke =	165	m	0,229	0,229
diameter =	166	m	0,006	0,105
Water cylinder area =	167	cm^2	44	88
Hydraulic force = 167 . (160 - 161) =	168	kgf	2626	5251
Nominal mean indicated pressure, estimated	169	at	5	5
Steam piston area = 168 : 169 =	170	cm^2	525	1050
Steam cylinder diameter, from 170 =	171	m	0,257	0,366
Theoretical steam consumption at a feed pressure 169 + 62, including 10% extra for clearance spaces and 5% for hydraulic losses:				
Pump admission pressure = 169 + 62 =	173	ata	6.3	6.3
Steam temperature (Fig. A1.1) =	174	°C	269	269
Steam specific volume from 174 and 173 =	175	m^3 kg^{-1}	0,395	0,395
Steam cylinder theoretical volume = 170 . 165 =	176	m^3	0,024	0,024

Theoretical steam consumption = $\frac{176 \cdot 1.10 \cdot 1.05 \cdot 163 \cdot 60 \text{ min h}^{-1}}{175}$	177	kg h^{-1}	421	842
Steam leakage loss, estimated =	178	"	30	60
Wall effect loss, estimated	179	"	30	60
Total steam consumption = 177 + 178 + 179	180	"	481	962
Making the pump a compound one, the consumption becomes =	181	"	300	600
Estimated consumption of the stoker engine (23) =	182	"	60	120
Estimated consuption of the electric generator 17	183	"	42	84
Estimated consumption of the 48 air brake pump (the crosshead booster 49 on) =	184	"	0 when running	0 when running

Engine performance

(Sec. Fig. A1.1)

Inlet steam pressure, HP cyl, 12 =	201	ata	51	51
Inlet steam temp. HP cyl, 6 =	202	°C	550	550
From 201 and 202, steamchest enthalpy (HP cyl), = 12 =	203	kcal kg^{-1}	847.4	847.4
HP cyl exhaust pressure = receiver I pressure = 56	204	ata	20	20
On the Mollier, steam enthalpy at HP exhaust, adiabatic expansion, from 201 202 and 204 =	205	kcal kg	773.0	773.0
Adiabatic heat drop on the HP cylinder = 203 - 205 =	206	"	74.4	74.4
Assummed internal efficiency of the HP cyl. (24)	207	-	0,915	0,915
Actual heat drop converted into work = 207 . 206 =	208	kcal kg^{-1}	68.1	68.1
HP cyl. exhaust steam enthalpy = 203 . 208 =	209	"	779.0	779.0
On the Mollier, HP cyl. exhaust steam temperature from 209 and 204 =	210	°C	409	409
Steam flow to the HP cyl. 5 =	211	kg h^{-1}	11880	23760

Specific steam consumption of the HP cylinder = $\frac{632.5 \text{ kcal } CV^{-1} h^{-1}}{208}$ =	212	$kg\ CV^{-1}\ h^{-1}$	9.29	9.29
HP cyl. indicated power = 211 : 212 =	213	CV_i	1279	2557
MP cylinder design exhaust steam pressure = 56 =	220	ata	7.0	7.0
From 209 , 56 and 220 , the MP exhaust steam enthalpy, adiabatic expansion is = (Mollier diagram)	221	$kcal\ kg^{-1}$	710.5	710.5
Adiabatic heat drop in the MP cylinder = 209 - 21 =	222	"	68.5	68.5
Assumed adiabatic internal efficiency of the MP cyl. = (24)	223	-	0.94	0.94
Actual heat drop converted into work in the MP cylinder = 222 . 223 =	224	"	64.4	64.4
Exhaust steam enthalpy = 209 - 224 =	225	"	714.6	714.6
MP cyl. exhaust steam temperature, from 225 and 220 (Mollier diagram)	226	°C	269	269
Steam 21 to heater 6 = 77 =	227	$kg\ h^{-1}$	608	617
Steam 36 to heater 28 = 122 =	228	"	1009	2018
Total extraction from HP cyl. exhaust = 227 + 228 =	229	"	1318	2635
Net steam to MP cylinder steamchest = 5 - 229 =	230	"	10562	21125
Indicated specific steam consumption MP cyl. = 632,5* : 224 =	231	$kg\ CV^{-1} h^{-1}$	9.82	9.82
Indicated power, MP cylinder 230 : 231 =	232	CV_i	1075	2151
LP cylinder exhaust pressure (by design) = 60 (25)	240	ata	1.17	1.17
From 220 , 225 and 240 the adiabatic drop LP steam enthalpy is (Mollier diagram)	241	$kcal\ kg^{-1}$	629.5	629.5
Adiabatic heat drop in the LP cyl. 225 - 241 =	242	"	85.1	85.1
Assumed adiabatic internal efficiency of the LP cylinder = (26)	243	-	0,80	0,80

Actual heat drop converted into work in the LP cyl, 242 . 243 =	244 kcal kg^{-1}	68.1	68.1
Exhaust steam enthalpy = 225 - 244 =	245 "	646.5	646.5
LP exhaust steam temperature, from 245 and 240 - (Mollier) =	246 C	116	116
Steam to feedwater pump 181 =	247 kg h^{-1}	300	600
Steam to air heater 7 = 88 =	248 "	64	127
Steam to furnace swirling 26 =	249 "	120	240
Steam to heater 27 134 =	250 "	359	717
Steam to stoker engine 182 =	251 "	60	120
Steam to electric generator 188 =	252 "	42	84
Total extraction from MP steam exhaust Σ 247 through 252 =	253 "	944	1889
Net steam to L.P. cylinder = 230 - 253 =	254 "	9618	19236
Indicated specific steam consumption, LP cylinder = 632,5* : 244 =	255 kg $CV_i^{-1}h^{-1}$	9.29	9.29
Indicated power, L.P. cyl. = 254 : 255 =	256 CV_i	1036	2071
Total indicated power = 213 + 232 + 256 =	260 "	3389	6779
Assummed internal resistance at 100 km h^{-1} including average curves =	262 CV	360	720
Net drawbar power = 260 - 262 =	263 CV_e	3029	6059
Total fuel fired per hour = 54 =	264 kg h^{-1}	1810	3619
Lower heating value of fuel 19 =	265 kcal kg^{-1}	5000	5000
Drawbar thermal efficiency $\frac{263 \cdot 632.5^*}{264 \cdot 265}$ =	266 -	0,212	0,212
L.P. exhaust "puff" : steam 33 to heater 26 147 =	280 kg h^{-1}	815	1631
L.P. exhaust "puff" steam to heater 8, 100 = 281	"	347	695
Total L.P. exhaust "puff" steam extracted by the special valve post before release = 280 + 281 =	282 "	1163	2326
L.P. exhaust steam 16 to ashpan 25 =	283 "	600	1200
L.P.exhaust steam 18 to heater 9 = 111 =	284 "	408	816

L.P. exhaust steam 34 to feed water heater 25 = 155 =	284.1	kg h^{-1}	920	1840
L.P. exhaust steam 38 to wet the coal on the tender 24 =	285	"	90	180
Total L.P. exhaust steam exhausted from the L.P. cylinder through the normal parts = 283 + 284 + 284.1 + 285 =	286	"	2018	4036
Total L.P. exhaust steam extracted from the L.P. inlet steam = 282 + 286 =	287	"	3181	6361
Net steam to blast pipe (from cyl.) 254 - 287 =	288	"	6437	12875
Steam from stoker exhaust 182 =	289	"	60	120
Steam from turbogenerator exhaust 188 =	290	"	42	84
Total steam to blast pipe = 288 + 289 + 290 + 292 =	291	"	6839	13679
Exhaust steam from feed water pump 181 =	292	"	300	600
Steam leaks 3 =	300	"	120	240
Steam to firebox swirling jets 26 =	301	"	120	240
Steam to ashpan 25 =	302	"	600	1200
Steam to wet coal 24 =	303	"	90	180
Steam to blast pipe 291 =	304	"	6839	13679
Total steam getting out from the circuit $\sum$ 300 through 304 =	305	"	7769	15539
This checks well with water entering the circuit 151 =	306	"	7769	15539
Water leaks and for various uses =	307	"	180	360
Water intake to tender tank 37 306 + 307 =	308	"	7949	15899

Economizer performance

Design outlet water temperature =	309	°C	270	270
Enthalpy of feed water at economizer outlet (temp. 309) =	310	kcal kg^{-1}	283.0	283.0
Ditto at economizer inlet = 115 =	311	"	203.5	203.5
Enthalpy gain = 310 - 311 =	312	"	79.5	79.5
Heat to economizer = 2 . 312 =	313	kcal h^{-1}	954000	1908000
Approximate specific heat of gas passing through the boiler	314	kcal $Nm^{-3}K^{-1}$	0,365	0,365

Temperature drop of gas in the economizer $\frac{313}{51 \quad 314} =$	315	K	191	191
Gas temperature before the economizer = 315 + 4 = (27)	316	°C	571	571

Fuel and ash balance:

Fuel fired per hour = 54 =	500	kg h^{-1}	1810	3619
Unburned carbon enaping out of the cyclone, estimated =	501	"	18	36
Steam for coal wetting 24 =	502	"	90	180
Sum, 500 + 502 =	503	"	1900	3799
Air, 71 =	504	Nm^3 h^{-1}	11798	23597
Ditto in kg h^{-1} = 504 . 1,288 kg Nm^{-3} =	505	kg h^{-1}	15196	30393
Steam to ashpan 25 =	506	"	600	1200
Steam to furnace swirl 26 =	507	"	120	240
Sum 502 + 506 + 507 + 505 + 500 - 501 =	508	"	17816	35632
Total gas 48 =	509	"	17434	34867
Ash + unburned carbon to ashpan = 508 - 509 =	510	"	382	764
Fuel burned per hour = 44 =	511	"	1756	3511
Unburned fuel to ashpan = 500 - 501 - 511 =	512	"	36	72
Ashes to ashpan = 510 - 512 =	513	"	346	692
Ash % of fuel burned = 513 : 511 =	514	%	19.7	19.7

APPENDIX A2

NOTES AND COMMENTS ON CYCLE CALCULATIONS (NON CONDENSING)

(1) "Maximum rated power" is defined by the "continuous (i.e. lasting indefinitely not were by traffic stops) steaming rate which is the base for all calculations, component dimensioning, etc. It may be safely assumed that the engine can be worked harder, but this is "overload" its maximum being loosely defined either by indifferent steaming, adhesion, exaggerated fuel and water consumption, component life, etc.

(2) The rated evaporation figure has been chosen so as to result in drawbar power around 1.

(3) A leak of (200 or 240) $kg\ h^{-1}$ is rather on the low side, yet every effort should be done to reduce it as much as possible bearing in mind the high pressure of the steam.

(4) It assumes a most careful insulation.

(5) A figure of 5000 $kcal\ kg^{-1}$ (9000 $btu\ lb^{-1}$) means that lower grade fuels are necessarily to be considered (low h.v.)

(6) A figure as low as 1.2 means that a high turbulence and a low combustion rate in relation to furnace volume must be adopted if non polluting gases are to be ejected from the chimney.

(7) "Combustion gases" are defined as those coming exclusively from the combustion of coal plus the excess air. "Total gases" are "combustion gases" plus (steam to ashpan 25 + steam to bunker 24 + swirl steam 26).

(8) Decimal places are not to be interpreted in the sense of accuracy but as a simple result of computations.

(9) Nm^3 means one cubic meter of gas at normal pressure (760 mm Hg) and 0°C point.

(10) The efficiency of 0,97 includes solid unburned fuel loss (which is very low because of the concurrent action of the GPCS and the cyclonic separation in the firebox), the ashpan loss (which is very small because of the afterburning in grate 9) and minor losses in gas phase as CO and hydrocarbons. (lower heating value).

(11) Receiver pressures 56 & 57 are given as tentative figures, later to be slightly corrected so as to realize the desired power distribution between the three cylinders. A slight alteration in the latter has but a minor influence on cycle performance.

(12) This is slightly higher than 60 becuase the non return valve 29 allows to pass steam only at the beginning of the release.

(13) Same as per (12), but in this case the author's specially posted piston valve communicates the cylinder 3 with the air heater 8 and the feed water heater 25 before the release phoper.

(14) Radiation not considered in air heater calculations.

(15) Although cycle calculations have been made on the assumption that the condensate of heater 25 gets out through a trap 29, nothing impedes that a mixture of water and steam goes to the mixing cones 24, hence leading to some increase in the pressure helping the pump 30 to require smaller mechanical pumping work.

(16) Since the feedwater temperatue (13) is 200°C, the terminal difference (4) - (13) = 180 k, which is about that obtaining in normal steam locomotive boilers.

(17) A steam temperature of 550°C = 1022 °F now bears no relation with cylinder lubrication problems (Ref. 7) and is only conditioned by the behaviour of the hottest part of the superheater in connection with creep, corrosion, influence of ashes, etc. Normal cylinder is to be used.

(18) This pressure is higher than the maximum working pressure (7) = 60 ate, and the 5% excess over the latter is useful for developing a higher starting tractive effort under exceptional circumstances. This excess is 10% in the German Boiler Code (24).

(19) This temperature is conditioned by the saturation temperature corresponding to the HP cycl. exhaust.

(20) For all water heaters the terminal temperature difference has been taken as 10 K, which is a normal value in locomotive practice. This supposes that no scale is deposited on them.

(21) As in the case of air heaters, radiation has not been considered for the feed water heaters. They are assumed to be carefully lagged.

(22) Feed pump calculations are supposed to be a first approximation only.

(23) At maximum rated output the fuel consumption is (54) = 3620 kg h^{-1} (which is quite small). The stoker is so designed that its full output is as small as possible (for example 4800 kg h^{-1}) and its engine designed involving steam automobile technique, hence, very economical.

(24) The internal indicated efficiency of the MP cylinder has been taken to the very high figure of 0,94 which is about that obtaining in high pressure cylinders of Chapelon's compounds. This is a rather pessimistic figure because the now available knowledge on heat transfer, leakage, etc, permit to think on a better figure, perhaps 0.96. No allowance has been taken on external radiation because the whole cylinder block is exceedingly well insulated against heat losses. The efficiency of the HP cylinder has been taken slightly lower because of the more important cooling required to keep the rubbing surfaces at temperatures low enough to match lubrication requirements. Such high efficiencies are of course to be associated to an utmost internal streamlining and insulation.

(25) This is the total (Pitot) pressure measured on the outer blast pipe when the valve communicates it to the cylinder during the return stroke (Porta 29) (Mean back pressure 200 gh cm^{-2}).

(26) The efficiency of the LP cylinder is rather low because of the incomplete expansion tow of the indicator diagram. However, the latter can be expected to be quite low even at rated power because the chosen cylinder volume has been selected on the large side, its clearance volume is quite reduced (9%) in spite of very large steam passages and the design of its thermal part is claimed to be highly perfect.

(27) This high temperature favours the design of the superheater and considerably shortens the tube bundle and the size of the barrel, hence its weight. This effect is complemented by a large furnace heating surface leading to a low gas temperature at the tubeplate. The whole firebox is not heavier than the ordinary one because it is built in 3/8" plates (12).

(28) Power is expressed in "metric horsepower" = 75 kgfm s^{-1} CV_e is for drawbar power on the level and at constant speed; CV_e is for indicated power; CV_i is for wheelhead power. 1 CV h = 270,000 kgf m =0,9863 HP; the IT kcal has been adopted; 860 kcal = 1 Kwh; 1 CV h = 632,5 kcal; 1CV h = 632,5 kcal; 1 CV = 0,735 Kw. An asterisk means that the coefficient is dimensional (x. 632,5*).

(29) Pressures have been expressed in "metric atmospheres". 1 at = 1 kgf cm^{-2} Gauge pressures are indicated "ate"; absolute pressures by "ata".

(30) Calculations are to be understood with reference to Fig. 1 and cycle description given in Section 2.

(31) For the concerned combustiongas the following temperature-enthalpy applies (from '27' and '28'):

Temperature °C	0	100	200	300	400	500	600
Enthalpy, kcal Nm^{-3}	0	33	66	100	135	171	209
Ditto, for air Kcal Nm^{-3}	0	31	62	93	125	158	192

(32) It is assumed that no overcooling of condensate occurs both in air and water heaters, which is a slightly pessimistic hypothesis.

(33) An important pressure difference is assumed between feed pump and steam generator so that water velocity, hence the heat transfer coefficient can be increased resulting in small terminal temperature difference at the various heaters.

(34) It cannot be otherwise, all the available knowledge in adhesion is incorporated (Porta 14).

APPENDIX A3

TENTATIVE DIMENSIONS OF ENGINE DESIGN
(Fast freight type) Ref. to Fig. 1

Rated drawbar horsepower (263) =	320	CV	3029	6059
Diameter of driving wheels by design	330	mm	1524	1524
Maximum design speed by the "1.1/2 diam." AAR rule	331	km h^{-1}	145	145
Estimated engine weight without ballast tanks, in working order	332	kgf	67200	134400
Number of coupled axles	333	-	5	5
Maximum axle load	334	kgf	13200	26400 58186 lbr
Adhesion weight = (334) . (333) =	335	"	66000	132000
Idle wheel weight (pony) =	336	"	7200	14400
Sum (335) + (336) =	337	"	73200	146400
Ballast = (337) - (332) =	338	"	12000	24000
Nominal "0.85" tractive effort =	339	"	19800	39600 87282 lbf
Nominal adhesion factor (335) : (339) = (34)	340	"	3.3	3.3
Nominal "100%" tractive effort = (339) : 0.85 =	341	"	23294	46588
Piston stroke (design) =	342	mm	736.6	736.6
Average piston thrust = (341) . $\frac{\pi}{4}$. (330) ÷ 1.5 . (342)	343	kgf	25255	50511
L.P. and M.P. cylinder piston thrusts giving 15% less than the HP piston thrust = (343) . 0,952	344	"	24043	48086
Design pressure for the L.P. cylinder = (220) - (240) =	345	at	5.83	5.83
L.P. cylinder diameter = $\frac{(344)}{(345) . \frac{\pi}{4}}$	345	m in	0,720 28.3	1,024 40.32
MP cylinder design pressure (50) - (220)	346	at	13.0	13.0
MP cyl. diam. = (344) / (346) . $\frac{\pi}{4}$	347	m in	0,482 18.9	0,686 27.0

HP cyl. design piston thrust (344) . 1,15	(348)	kgf	27649	55297
[illegible] cyl. design pressure = (11) - (56) =	(349)	at	31	31
HP cyl. diameter = $(348)/(349) \cdot \frac{\pi}{4}$ =	(350)	m	0,335	0,477
		in	13,2	18,8
Equivalent "good" (7500 kcal kg^{-1}) coal burning rate = $\frac{(44) \cdot (19)}{7500 \text{ kcal kg}^{-1}}$ =	(351)	kg h^{-1}	1171	2341
"Good coal" burning rate (by design) referred to grate at rated power, tentative maximum	(352)	kg m^{-2} h^{-1}	400	400
Grate area (351) : (352) = (Minimum)	(353)	m^2	2,93	5,86
Approximate mean diameter =	(354)	m	1,93	2,73

<u>Other locomotive dimensions are as follows:</u>

Time to use coal capacity at full rated power =	(380)	h	12	12
Ditto, water =	(381)	h	5	5
Water supply =	(382)	m^3	40	79
Fuel supply =	(383)	t	22	43
Tender tare =	(384)	kgf	29000	58000
Total locomotive weight in working order =	(385)	"	151	302
Total locomotive weight with 2/3 supplies =	(386)	"	138	276
Total locomotive mass, including the equivalent inertia of rotating parts (2/3 supplies) =	(387)	t	146	293

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SUMMARY OF PART II

The suitability of "Third Generation Steam Locomotives" for modern railway operations is dicussed in terms of the fundamental effectiveness equation relating a locomotive's work output to its total operating costs. This takes the form of comparing possible operating costs with those of alternative locomotives for the same duties, including coal-burning-turbine proposals.

The means by which the Stephensonian locomotive can be enlarged to operate very long trains, which are necessary for capacity operation on certain railways, are outlined.

INTRODUCTION

Part I has detailed, with adequate supporting calculations, the performance level to be expected from Third Generation Stephensonian steam locomotives, henceforth designated TGS. This is summarized by the locomotive "test" drawbar thermal efficiency = 0,21, its year-round overall drawbar thermal efficiency over a representative section = 0,15 and the drawbar performance curves shown in figs. 4 & 5, Part I. It is important to note that these latter refer to a given example of TGS, i.e. a 2-10-0fast freight locomotive, fig 3a., Part I, whilst the principles of TGS are applicable to any form of the steam locomotive.

Whilst the ability to achieve this performance level is fundamental to the claim for recognition of TGS, it will take more than this for railway managements to reconsider the machine they abandoned but a few years ago. After all, higher (albeit just higher) thermal efficiency is still granted for diesel and electric traction. So the question is, what do railways stand to gain by adopting TGS? The answer to this must be sought within the fundamental locomotive effectiveness equation.

THE LOCOMOTIVE EFFECTIVENESS EQUATION

This states that:

Effectiveness of a locomotive fleet =

$$\frac{(\text{ drawbar work performed by fleet })}{(\text{ total cost of operating fleet })}$$

which is expanded as:

$$\text{Effectiveness} = \frac{N.\int_{O}^{U.L} P_d.dt}{C_c+C_m+C_f+C_{cr}+C_d+C_s+C_l}$$

where: N = number of locomotives in fleet.
U = Locomotive utilization factor.
L = Locomotive life.
P_d = drawbar power at any instant of utilized time dt.
C_c = Capital cost of locomotives including any interest charges, insurances and taxes.
C_m = Maintenance cost of locomotives.
C_f = Fuel cost of locomotives, including water costs.
C_{cr} = Crew labour charges.

C_d = Cost of locomotive stabling depots.
C_s = Cost of locomotive supervisory staff, e.g. engineers.
C_l = Lubrication cost of locomotives.

In this equation P_d refers only to drawbar power resulting from a positive drawbar pull because some equivalence is sought in defining locomotive effectiveness between locomotive drawbar work output and railway traffic output (tonne-km) which latter requires the expenditure of so much positive work. The relationship between the two outputs is readily seen when considering that a traffic increase must be accompanied by increased locomotive output, e.g. by better utilisation, more locomotives, etc. The effect of dynamic braking (which may be considered to produce a "negative" drawbar work) is best included solely in the denominator as a cost advantage (if any) to locomotives so equipped.

Note that the above equation is valid to compare locomotive effectiveness in a given service because any operation requires (i) sufficient t.e. and (ii) sufficient power to maintain schedule with a given train, and both are always obtainable with any form of locomotive if enough are coupled to the train.

WORK OUTPUT OF THE LOCOMOTIVE FLEET

As stated, there is a loose equivalence between the numerator of the effectiveness equation and the tonne-km output of a given railway. If this railway must move a certain amount of traffic in a given time the haulage work output of its locomotives, and hence the numerator of the effectiveness equation is sensibly fixed over that time period and the problem is to select the most economical locomotives.

In service the capacity of a locomotive for haulage work is measured by its power capacity provided that the physical structure of the railway and operating methods allow that power to be exploited. Given this condition TGS will develop fairly high power in relation to its weight of power-producing hardware and its capital cost, this being a consequence of high thermal efficiency reducing the specific steam consumption. For a given weight of locomotive, high power implies high speed operation which reduces the number of locomotives required and offers a more atractive service to shippers.

The utilization factor is a function of operating effectiveness, the availability possible from the locmotives and their degree of flexibility to handle a variety of train services (the last factor giving rise to the so-called "mixed traffic" locomotives). Tests in 1940's (1) proved the ability of the steam locomotive to match the diesel in utilization; just as the diesel has undoubtedly improved in reliability and hence potential utilization, so TGS with advances in design and boiler feed water treatment can parallel this improvement. However with their ability to be operated as single locomotives or combined into consists, diesel and electric locomotives may yet retain a slight advantage in flexibility, and hence potential utilization, over TGS.

It is generally accepted that the steam locomotive has a longer economic life than the diesel: excluding replacement due to obsolescence the life of a locomotive depends largely on the rate at which its maintenance cost rised with age and traditionally the steam locomotive (if well designed) seems to have had some advantage over the diesel due to fewer working parts of coarser construction. Hence component replacement or renovation which must occur with wearing parts is less costly. Surprisingly the steam locomotive in the author's experience may even have longer economic life than electric locomotives. Here it is probably the high cost of manufacturing spare parts for electric locomotives, which are not mass-produced to the extent of diesels, which is responsible.

On balance it is considered that the high installed power and potentially long life of TGS will result in fewer of these locomotives being required to handle a given traffic compared to diesel locomotives, although with electric traction still fewer should be necessary.

The high power built into TGS will be necessary (along with improved track and signalling) if the railways are to become more speed conscious. Whilst the adhesive weight of a locomotive determines its load haulage capacity, its power determines the speed at which the load will be pulled, and TGS offers power at a much lower investment than diesel traction, for example the power equivalence of 1 x (2-10-0 TGS) loco to 2,7 x 3300 (rated) h.p.Deltic diesels.

THE DENOMINATOR OF THE EFFECTIVENESS EQUATION

Having fixed the required work output of the locomotive fleet at the level corresponding to traffic requirements, the cost comparison can be made to determine the most effective locomotives. Of necessity the following must be based on predictions; it is up to the reader to judge for himself how valid he thinks the predictions are.

CAPITAL AND RELATED COSTS

Little data is available on the possible current cost of building steam locomotives. Fig. 1 shows locomotive purchase costs for one railway on the basis of price per unit installed wheel rim power. By refinement of design without increasing the quantity of hardware involved and without undue complexity TGS will achieve a 100% increase in power compared to the First Generation Steam locomotives of this example, which reasonably allows the broken line of Fig. 1 to represent TGS, this being extended to allow for the current upsurge in inflation.

Allowing for the fact that the electricity supply equipment costs are nos included in the electric locomotive curve, it seems reasonably certain that TGS can offer locomotive power at a lower investment than its two principal rivals. - When considering the similar or slightly less utilisation potential of TGS and its anticipated long life this implies a low investment per unit locomotive work.

It should be noted that neither the steam nor electric locomotives of Fig. 1 were mass-produced to the same extent as the diesels; in mass production the steam and electric costs would decrease. Although perhaps not so suitable as the other two for mass production of the complete locomotive, TGS is adaptable to mass production of its components.

Fig 1 is reproduced as Fig 2 on the basis of purchase cost per unit tractive effort: it is seen that the diesel becomes much more economical whilst TGS has no advantage over FGS as the adhesion-limited tractive effort is nominally the same in both cases. Thus for drag service, where no use is made of the high power built into steam and electric locomotives, the purchase costs of diesel traction are competitive.

MAINTENANCE COST

An idea of this relation is given in Fig. 3 which shows the maintenance costs incurred for steam, diesel and electric locomotives of similar vintage operated simultaneously on a certain railway, related to a unit expressing their output in traffic.

The steam locomotive concerned can be classified as "modern FGS"; the electric locomotives are d.c.machines whilst the diesels are U.S.-built standards. Due to design improvements plus the virtual elimination of boiler repairs with modern feed water treatment, TGS must be expected to improve on the FGS performance, which is itself impressively better than diesel.

FUEL COST (INCLUDING WATER COST FOR STEAM LOCOMOTIVES)

As noted in Part I, the high thermal efficiency of TGS plus the ability of the gas producer combustion system to effectively digest a wide variety of fuels, is an important potential economic advantage in a world where energy, and particularly oil, scarcity is a looming spectre.

More has, perhaps, been written in the past few years concerning the future of energy supplies than on any other single technical subject and it is not the author's intention to add to this except to commend to railway managers and government officials a serious study of this problem and its implications for the transport industry.

When, as is expected in 10-20 years time, the supply of oil fuels can no longer match the demand, the price of such fuels will rapidly increase; the necessary increase in demand for alternative fuels will increase their price also and it is impossible to predict at this time what the relative costs between, say, oil, coal and wood fuels may be. However, it is not just a question of costs but of availability and which consumers will have priority in obtaining oil supplies: there appears little doubt that road transport will receive preference and railways may find themselves under government pressure to change to non-oil fuels, as is presently being experiences by the utility companies in the U.S.A.

TGS offers a practical alternative to the present near-total dependance of the railways on oil fuel.

Although water is not a fuel in the thermodynamic sense, it is consummed in the same manner by the steam locomotive and may be regarded as such from the cost viewpoint. It contributes only a small fraction of the overall "fuel" cost of TGS which has low specific steam consumption and less than 100% make-up water due to the returning of condensate to the boiler feed. Fully condensing TGS reduces the water cost to almost zero.

CREW LABOUR CHARGES

The problem of crew labour charges is frequently cited as the most serious drawback to steam traction.

It is noted in Part I that two men will initially be required per TGS locomotive whilst single-manning and remote control are offered as future possibilities retaining the Stephensonian form (i.e. as opposed to turbo-electric or turbo-hydraulic locomotives). We will, however, consider at this stage only the implications of the proposal being offered, i.e. two men per locomotive. Three points are offered.

i. The oil-energy shortage and the long-term probability of a continuing and increasing surplus of labour will alter the relative values of the fuel and crew costs, since both are subject to the laws of supply and demand, the former increasing at a faster rate than the latter. In other words, the importance of rising crew costs may at some state become secondary to the rising fuel bill.

ii. If one TGS locomotive replaces one diesel locomotive or consist the increase in crew costs is either zero or at most is that due to the employment of a fireman. If full use can be made of the greater power capacity of TGS compared to diesel traction, the output (tonne-km) achieved per unit time by a locomotive crew is increased, which implies a reduction in crew costs for a given traffic.

iii. The principles of TGS are capable of incorporation in large locomotives equivalent to diesel consists. There is a long tradition of enlarging the steam locomotive by articulation to cater for increasing tractive effort with limited axle loading, culminating in the successful and well-known Garratt type locomotive still in important use on the African Continent, Fig. 4. These locomotives are regarded by their users as a "double locomotive" under the control of one crew and this concept may be improved and extended by the following means.

a. Extra engine units may be added as required to achieve the desired adhesive weight. At this state 4 units in Mallet-Garratt configuration (say 2-12-12-2 + 2-12-12-2) would seem feasible, this so-called "Super-Garratt" being proposed many years ago by the manufacturer Beyer Peacock. The previous limitation on such a locomotive, i.e. the inability of the boiler to supply sufficient steam, is counteracted in TGS by the low specific steam consumption of the engine units, the high efficiency boiler and high capacity draught on which boiler output so depends. This translates into the small boiler shown in Fig. 3 Part I for the "single" locomotive and it can be appreciated that the Garratt priciple will allow sufficient enlargement of the boiler to provide the steam supply to 4 such engine units, albeit at a probably elevated (yet still reasonable) specific firing rate. Fig 5 contrasts TGS locomotives and diesel consists of the same adhesive weight.

The ultimate extension of this idea is to make the boiler and engine units essentially independent, to be coupled as required for the necessary consist, giving great operational flexibility. This exploits the fact that it is much easier to remote control the functions of the engine units (as is already done with Garratts) than those of the boiler, and a great saving in complexity, capital and maintenance costs is envisaged over any steam proposal having a series of automatic boilers in a consist. This proposal, like other points of TGS, is subject to a research programme. See Fig. 6.

b. Near-constant adhesive weight is achieved by removing the water supply to external tank cars, as practiced in the modern Garratts of the South African Railways.

c. Extremely good adhesion properties will result from the coupling of all driving wheels, by coupling rods within "rigid" wheelbases and a hydraulic system (2) between the articulated sections. From an adhesion viewpoint this will allow heavier trains to be worked by such a locomotive than a diesel consist of the same adhesive weight. Alternatively without hydraulic coupling, anti-slip devices to automatically shut off the steam supply in the event of a slip and subsequently restore it after slipping ceases, without the intervention of the driver, can be fitted to each engine unit, so that a slip in one does not effect any others.

The developments outlined above have greater significance than the saving of crew labour charges alone. Whilst the trend toward lighter, faster and more frequent freight trains is to be encouraged as a means to improve railway competitiveness for which over the generally easily graded double track main lines such as are found in the Eastern U.S.A. the smallest TGS locomotive may be suitable, for the bulk movement of minerals or where single line operation compels the use of long trains to secure the necesary line capacity, the larger TGS locomotives could be used in operations similar to those today performed by diesel consists. The higher power: weight ratio of TGS allows higher train speeds even with very heavy trains.

OTHER COSTS

The stabling, supervision and lubrication costs are generally small in relation to the principal costs and may be expected to be similar for any modern locomotive.

DYNAMIC BRAKING

This is an important aspect of many railway operations.

The principles of dynamic braking from a train handling viewpoint are that the adhesive weight of the locomotive and a retarding torque applied to its driving wheels are used to hold or decelerate the train in the same manner as a positive torque applied to the driving wheels will accelerate the train and sustain its speed. These principles are not confined to electric transmission alone and Koffman (3) describes their application to various types of locomotive.

Dynamic braking of steam locomotives is almost as old as the locomotive itself and developed into the successfull Rigenbach counter-pressure system popular in Europe. This system can be modernized and applied to TGS wherever dynamic braking is thought necessary.

COAL BURNING ALTERNATIVES TO TG STEPHENSONIAN STEAM

Since an important part of the arguments for TGS centres on its ability to burn coal (or other non-oil fuel) it must stand comparison with alternative coal-burning proposals, of which there are currently a variety.

Examples known to the author comprise water tube boilers fired by (i) coal in a fluidised bed (ii) coal-in-oil slurry and (iii) pulverised coal. The steam generated drives turbines usually exhausting to a condenser at atmospheric pressure. The turbines may drive the road wheels through mechanical, electrical or hydraulic transmission, the latter two giving diesel-type characteristics whilst electric transmission can use the normal form of dynamic brake. A feature of all proposals is that boiler operation is automatic so that individual units, not requiring a fireman to attend to each boiler, can be worked in consists as with diesel and electric traction.

Judging by the publicity given to these proposals through papers and articles they seem to generate more interest than modern reciprocating steam and there is no engineering doubt that they can be developed into workable propositions both to produce the drawbar work units necessary for traffic haulage and to reduce the railways' dependancy on oil fuel. Given this the comparison with TGS reduces to their relative costs for a given traffic and the following is offered concerning the principal costs.

Capital Cost.- The complexity of the turbine proposals seems formidable; the hardware to drive the hydraulic or electric transmission (themselves a source of heavy first cost and maintenance) is more than the diesel engine it is replacing, so that it is difficult to predict a reduction in capital cost from that of diesel traction even when related to power output, which will give the advantage to TGS.

Maintenance Cost.- The same applies as for capital cost. In addition, it is noted that the turbine proposals assume a smooth transfer of industrial equipment not tested in railway conditions onto a locomotive. The life of a locomotive is a hard one; it is exposed to vibration and shocks due to motion and coupling, constantly changing speed and load demands, ambient temperatures from + 40°C to - 40°C, the techniques and temperaments of different crews, and is usually without careful attention. It is not surprising that under such conditions there is a long history where successful industrial equipment, from pipe joints to complete diesel engines, has failed to transplant successfully to the rough and tumble of locomotive life, and it is anticipated that a period no less lengthy or costly than the proposed TGS research programme will be necessary to get the turbine locomotives operating reliably. In contrast TGS is based on that form of steam power which has evolved under and as a consequence of the hardships of locomotive life.

Fuel Cost.- Suppossing that any of the coal burning alternatives.to TGS is able to burn the same low grade of fuel with the same effectiveness as has been already demonstrated in practice with the gas producer combustion system, the relative fuel costs will be in inverse proportion to the drawbar thermal efficiencies of the locomotives.

It has not yet been demonstrated to the author's knowledge that any of the turbine proposals can equal the drawbar thermal efficiency of TGS: indeed the anticipated thermal efficiencies of some proposals are much lower. This not only translates into a higher fuel bill but is not acceptable in view of the coming energy shortage.

The reasons for the lower efficiency are briefly that:

(i) The upper and lower thermodynamical limits of the Rankine cycle and degree of regeneration are not markedly different between the various locomotives which means a similar ideal cycle efficiency for all.

(ii) The turbine is a less efficient converter of the thermal energy in the steam to mechanical work than a well-designed reciprocating engine operating between the same pressure limits.

(iii) The turbine is more sensitive to load and speed changes than the reciprocating engine which is why the latter always finds favour in transport applications with mechanical transmission where the engine speed fluctuates widely. Thus if the turbine is geared to the road wheels a marked falling off in efficiency must be expected outside the desing conditions. With electric or hydraulic transmission this is reduced but the efficiency of the transmission system is lower causing further power losses.

(iv) The auxiliary power consumption may be greater due to the larger quantity of auxiliaries in some designs.

(v) If, as expected,the power:weight ratio is less than TGS the fraction of the wheel rim output required to move the locomotive around is naturely greater, reducing the available drawbar output and efficiency, especially at high speed.

Crew Cost.- The previous remarks on this subject apply. Although the attractiveness of the turbine proposals may stem from their capacity for multiple operation as with diesel locomotives, it has been shown that TGS can develop high tractive effort under the control of a single crew.

This cursory outline of the possible relative costs of TGS and alternative coal-burning proposals does not point to any advantage for the latter; rather the contrary. Yet the comparison is worthy of a more detailed study than was possible in this paper.

CONCLUSIONS TO PART II

We now return to our original question in the introduction to Part II - what do you, the railway managers and operators and government officials, expect to gain by adopting TGS?

Firstly, and of particular importance, you will be operating a locomotive that does not burn oil, oil which you may not be able to get at any price in the foreseeable future.

Although there are alternative locomotives dispensing with oil, notably electric traction, TGS achieves this with the minimum investment. Can you afford to electrify your railway? The answer is almost certainly no. Bust just as you replace life-worn diesel locomotives with new diesels so you can replace them with TGS locomotives. New maintenance and supplies services will have to be erected but their cost is negligable compared to that of stretching catenary from Atlantic to Pacific.

Secondly, you will have a locomotive that can haul the traffic in any way you want it, long trains or short, at the lowest overall cost of any motive power in many situations, especially for railways serving the coal producing areas.

Although it may look like the traditional steam locomotive it will be the same machine in looks only. By the methods outlined in Part I, it will have approximately 300% better year-round drawbar thermal efficiency than its predecessors, which is not negligeable. Do not disbelieve this figure (try to disprove the validity of the calculations of Part I if you can!). It is the result of sound thermodynamical understanding of the steam locomotive, often sadly lacking in the past. There is a general belief that the laws of nature limit the possible efficiency of this form of locomotive to the destructively low figures usual with FGS. This is not so. It was rather the inability of designers and operators of locomotives to make proper use of the machine's potential efficiency which produced such low efficiencies. Most FGS locomotives built in the late 1940's or 1950's had a potential test drawbar thermal efficiency of around 15% but could manage only about half that figure and were much worse still on a year-round basis. Yet it was considered at the time that no significant improvement was possible.

High thermal efficiency does not end with a reduced fuel bill - it means less fuel to transport, thus increasing line capacity for revenue-earning traffic, it means fewer coaling stations of smaller capacity decreasing the investments necessary for the changeover; it means less fueling operators to pay; it means smaller tenders to pull along; it means being able to meet anti-pollution laws.

The same applies to watering facilities; non-condensing TGS can be expected to have about 25% of the water consumption of FGS for the same duties, hence 75% less water to purchase, treat and distribute.

Clearly there was, in fact, a high development potential for the Stephensonian steam locomotive and the work to improve its efficiency has exploited this potential. Second generation steam (SGS) technology already offers some 200% increase in year-round drawbar thermal efficiency over previous steam without the need for further research, whilst TGS is one stage further. The next step in this logical progress is to obtain greater work output by expanding the steam to below atmospheric pressure, which we also expect to require some research programme before it becomes a practicality.

Therefore the steam locomotive still has considerable potential for development, by contrast, diesel and electric traction have already been developed to a high degree (as witnessed by their actual performance approaching far closer to the ideal than has hitherto been the case with steam), indeed their performance level may be approaching the asymptote denoting maximum performance.

It should not be thought that because the steam locomotive has not reached the ultimate performance we think it capable of its introduction should be delayed. On the contrary it is most likely that development work will procede faster with a healthy demand and output for such locomotives. This means that the railways will be prepared to meet the oil crisis and that a better answer to energy saving on the railways will be available sooner. It gives the necessary time if such a project is to proced at the correct pace so that mistakes are found on the drawing board not on the orad. It allows industry to more easily meet the production demand (as for industry, too, there is a benefit: someone must build the locomotives).

There are, of course, alternatives to this strategy, such as the electrification that cannot be afforded. You are invited to think of others which are better. The easiest is to do nothing. Let us wait to see if the oil crisis being predicted is a myth or reality. If it is real we can act. Only then it may be rather late.

RECOMMENDATIONS

Enough arguments and calculations have been made to prove the case for TGS on paper. Further progress must take the form of translating this into hardware.

The recommendation of this paper is that a fully modern Stephensonian steam locomotive prototype is built and tested to prove the practical validity of its claims. This could be a TGS locomotive but a SGS locomotive, not requiring a research programme, could be constructed more cheaply and would have a suitably improved all-round performance compared to traditional steam and diesel to convince the sceptics. TGS and condensing TGS locomotives coulw follow as a logical sequence of development.

The author considers that such a prototype should take the smallest practical form because the advantages it is intended to prove can be demonstrated on such a locomotive and it is obviously cheaper to build, whilst teething problems and operating inconveniences (e.g. the need for fueling and watering facilities) will multiply with the size of the locomotive. It makes sense to construct, say, a 2-8-0 which would stand comparison with a single 4-axle diesel or electric. It may be claimed that this is not a representative traffic machine, but it is not meant to be so; a prototype will be the subject of intense engineering tests in conjunction with the dynamometer car to prove the anticipated performance level whilst simultaneously maintenance problems and overall reliability must be closely monitored. Such testing is not best conducted by putting the locomotive into normal traffic. The larger locomotives required for regular traffic would follow as a result of the successful testing of the prototype.

Clearly the cost of building and testing a prototype is negligeable compared to the anticipated benefits and being in a field of particular interest should qualify for active government support.

Lastly it is recommended that action on this project is not delayed.

Any views expressed in this paper are those of the author.

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(1) KIEFER P.W., . "A practical evaluation of railroad motive power". Steam Locomotive Research Institute Inc., New York, 1947.

(2) PORTA L.D., "A system for coupling both Mallen engine units and its extension to other articulated locomotives" Buenos Aires 1977, (Unpublished)

(3) KOFFMAN J. "The dynamic braking of steam, diesel and gas turbine locomotives" paper 505, Journal of the Institution of Locomotive Engineers, London, 1951.

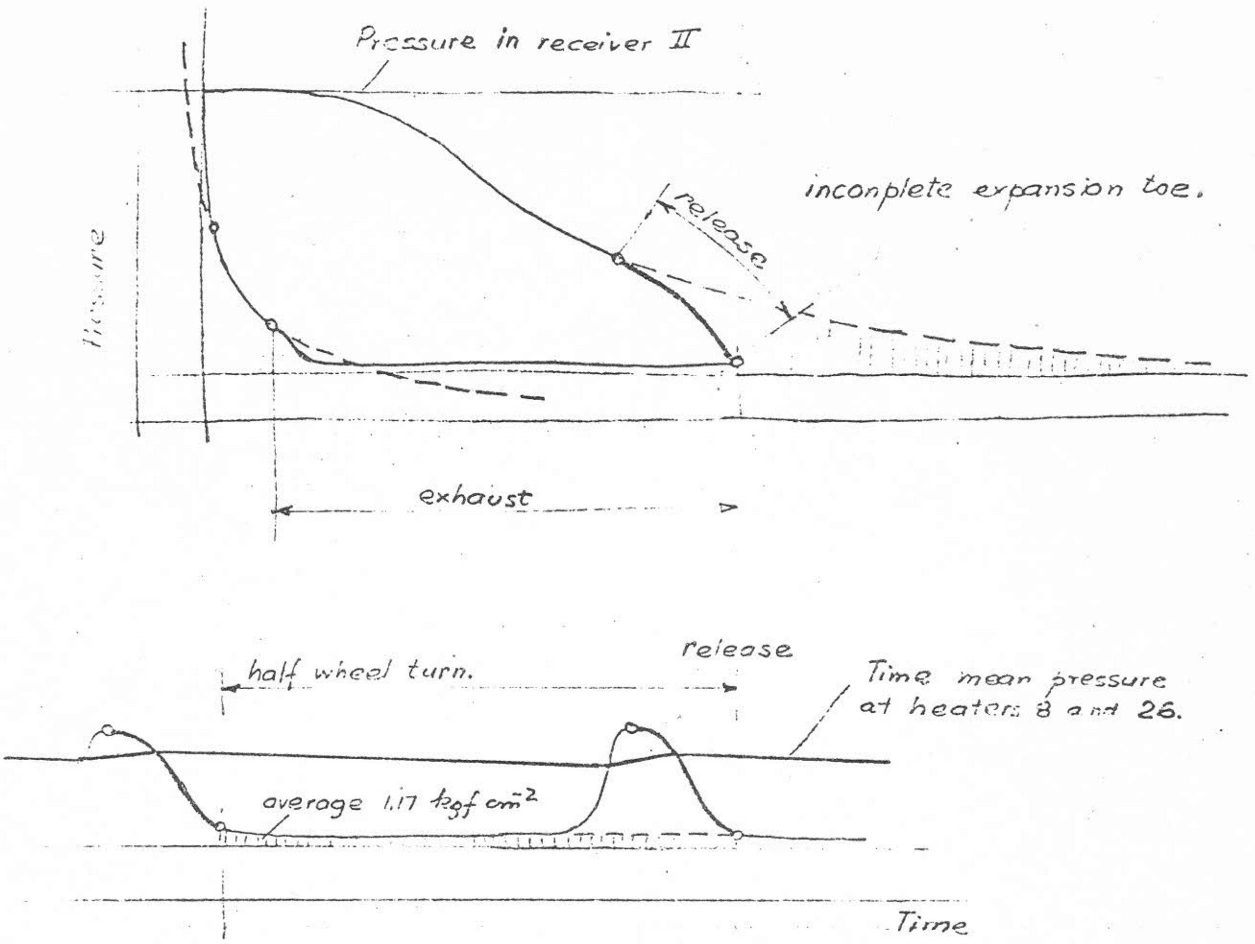


Fig. 2 LP indicator diagram

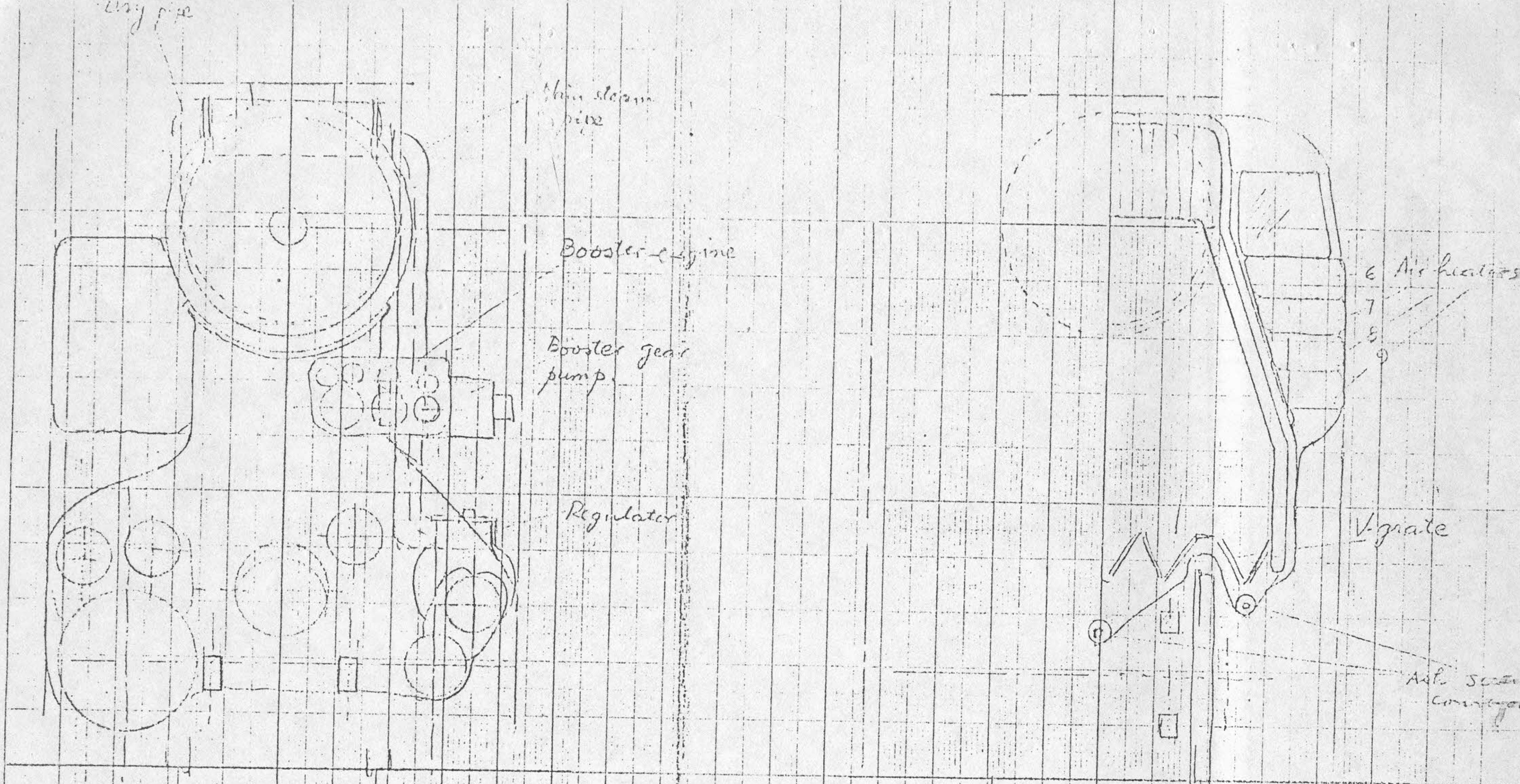


Fig 36. 6000 Drawbar Horsepower "Third-Generation" Steam Locomotive.

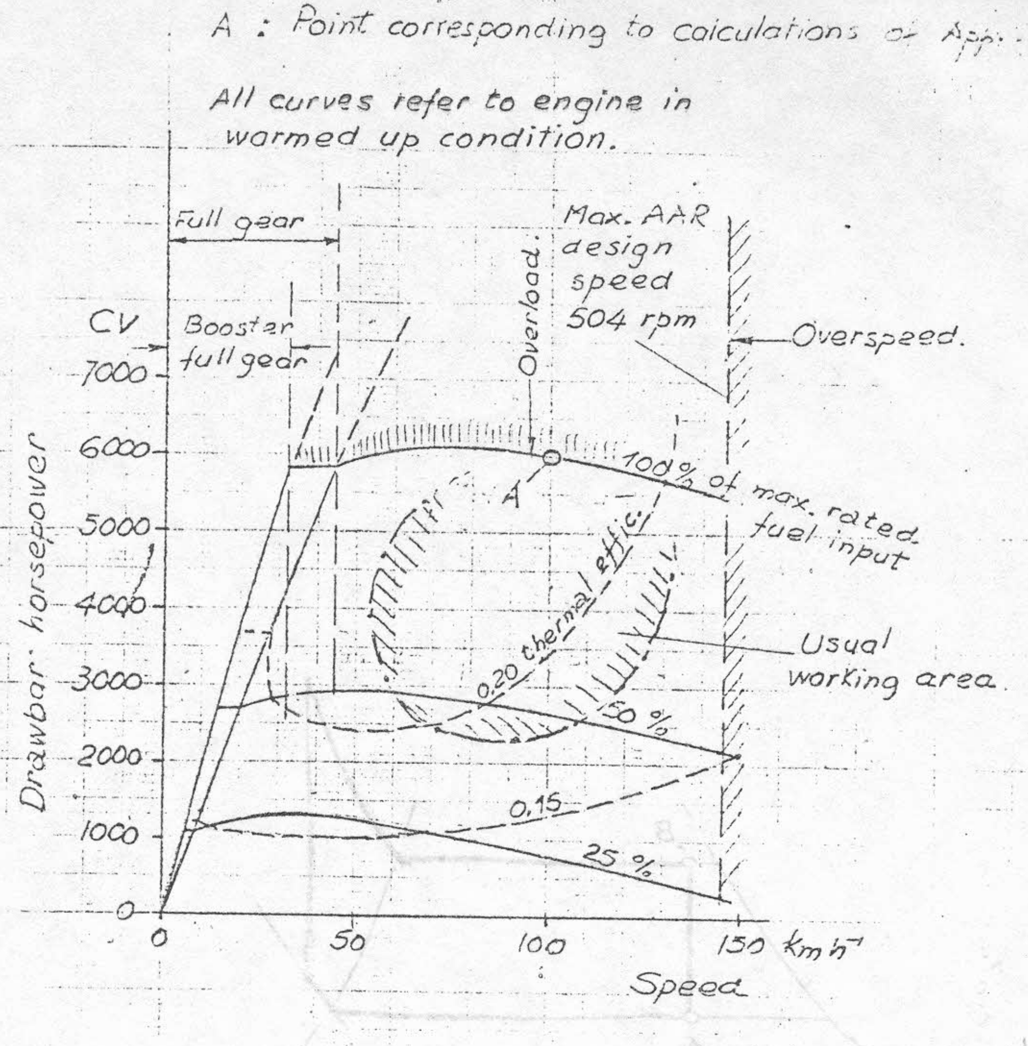


Fig. 4 Drawbar power characteristics

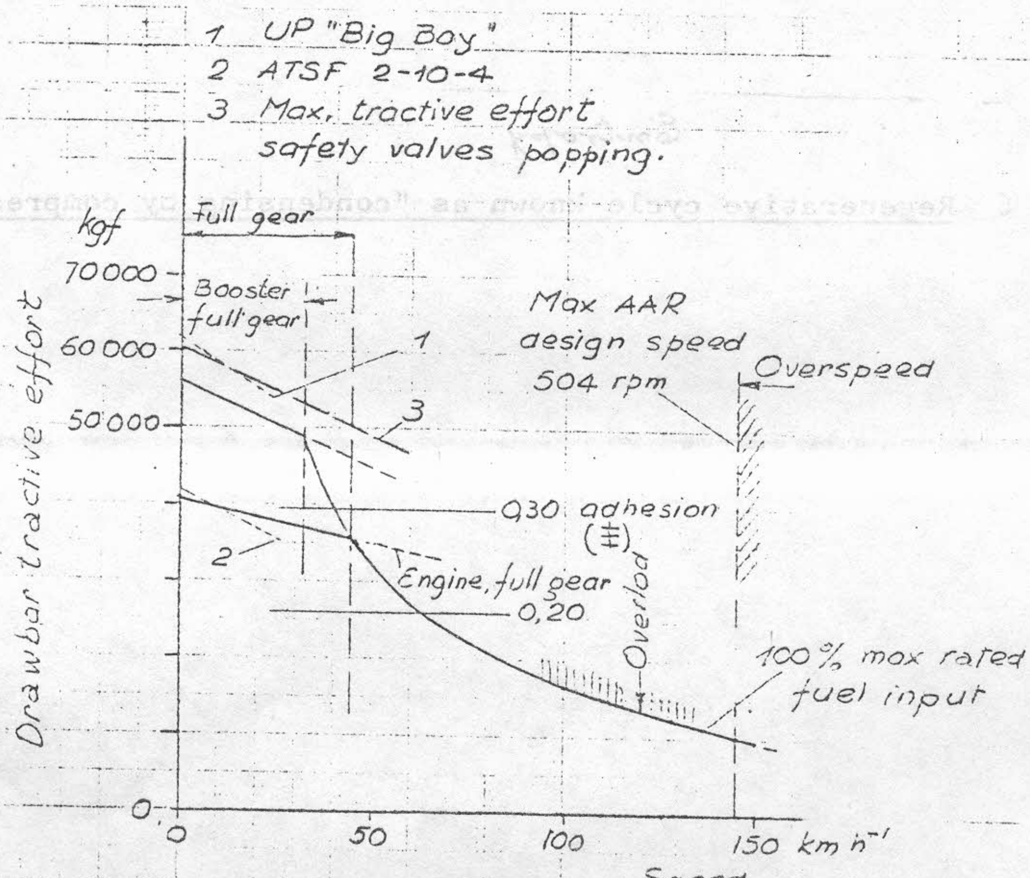


Fig. 5 Drawbar tractive effort characteristics.

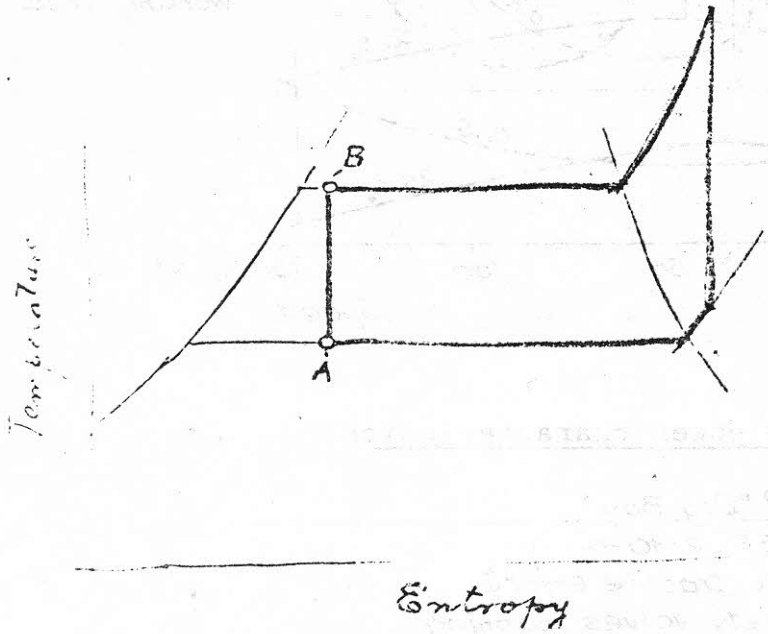


Fig. 6 Regenerative cycle known as "condensing by compression"

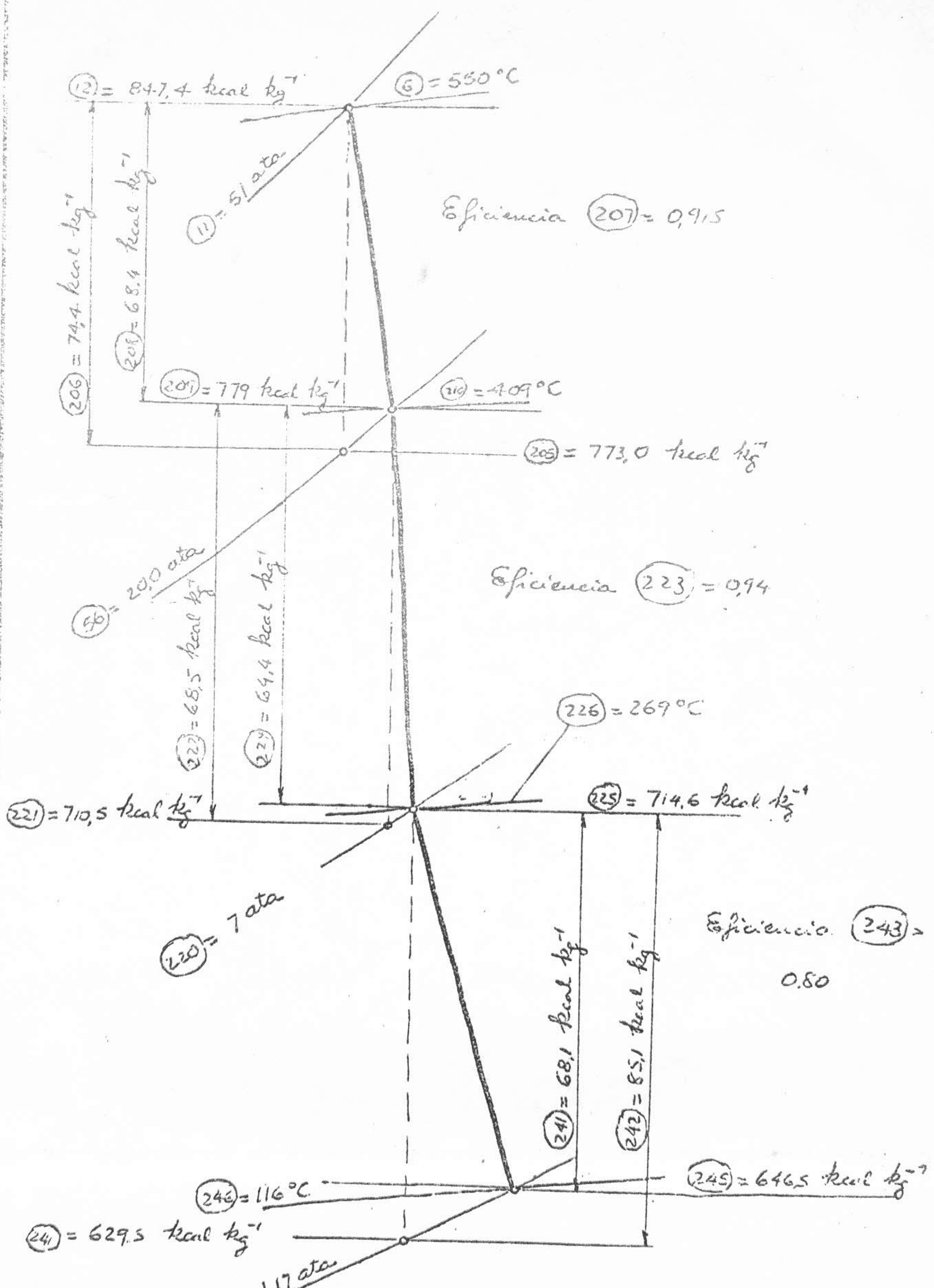


Fig A1.1. Evolución del vapor en el MOLLIER (Fuera de escala)

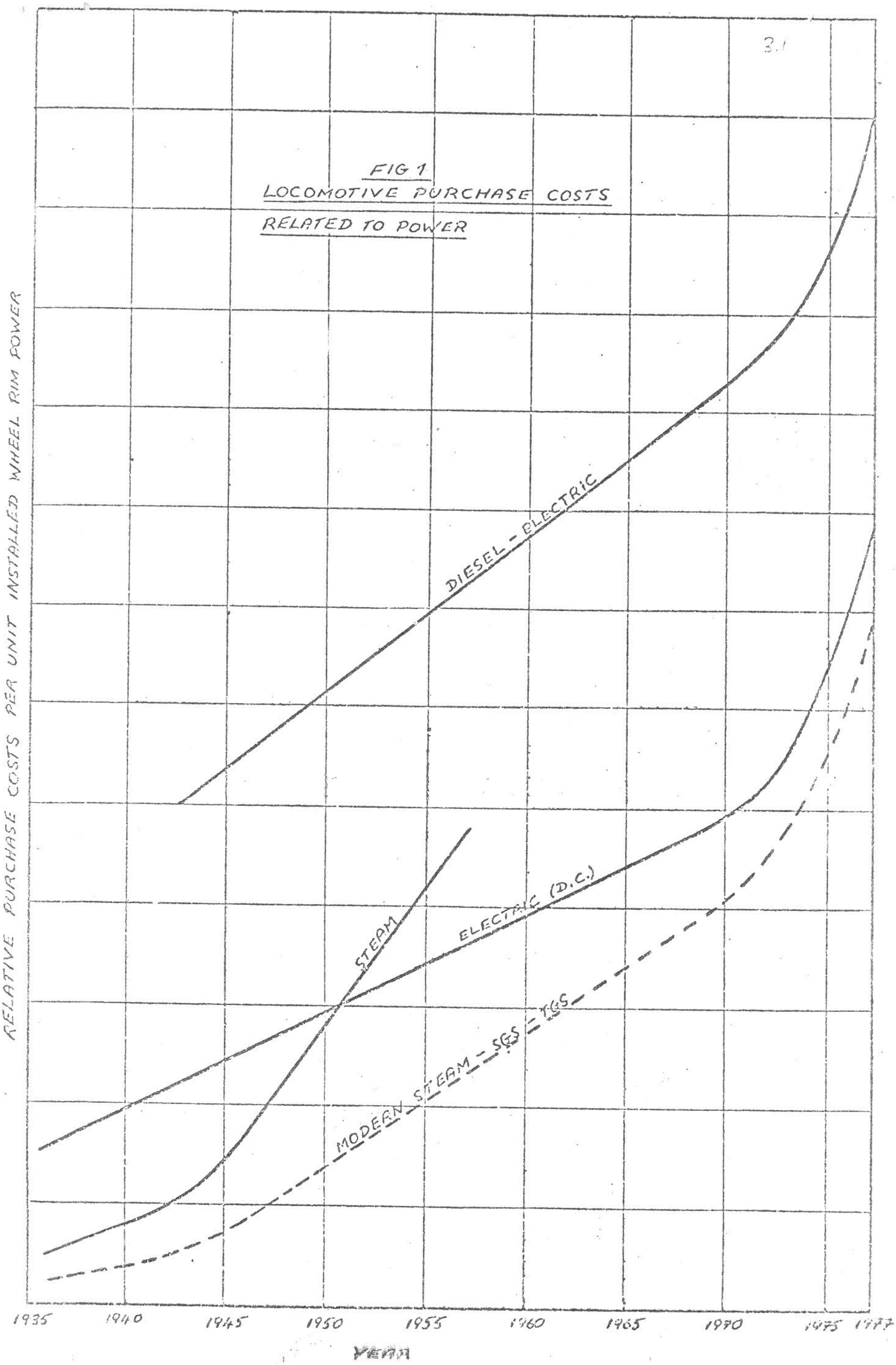


FIG 1
LOCOMOTIVE PURCHASE COSTS
RELATED TO POWER
RELATIVE PURCHASE COSTS PER UNIT INSTALLED WHEEL RIM POWER
DIESEL - ELECTRIC
STEAM
ELECTRIC (D.C.)
MODERN STEAM - SGS - TGS
1935
1940
1945
1950
1955
1960
1965
1970
1975
1977
YEAR

3,2

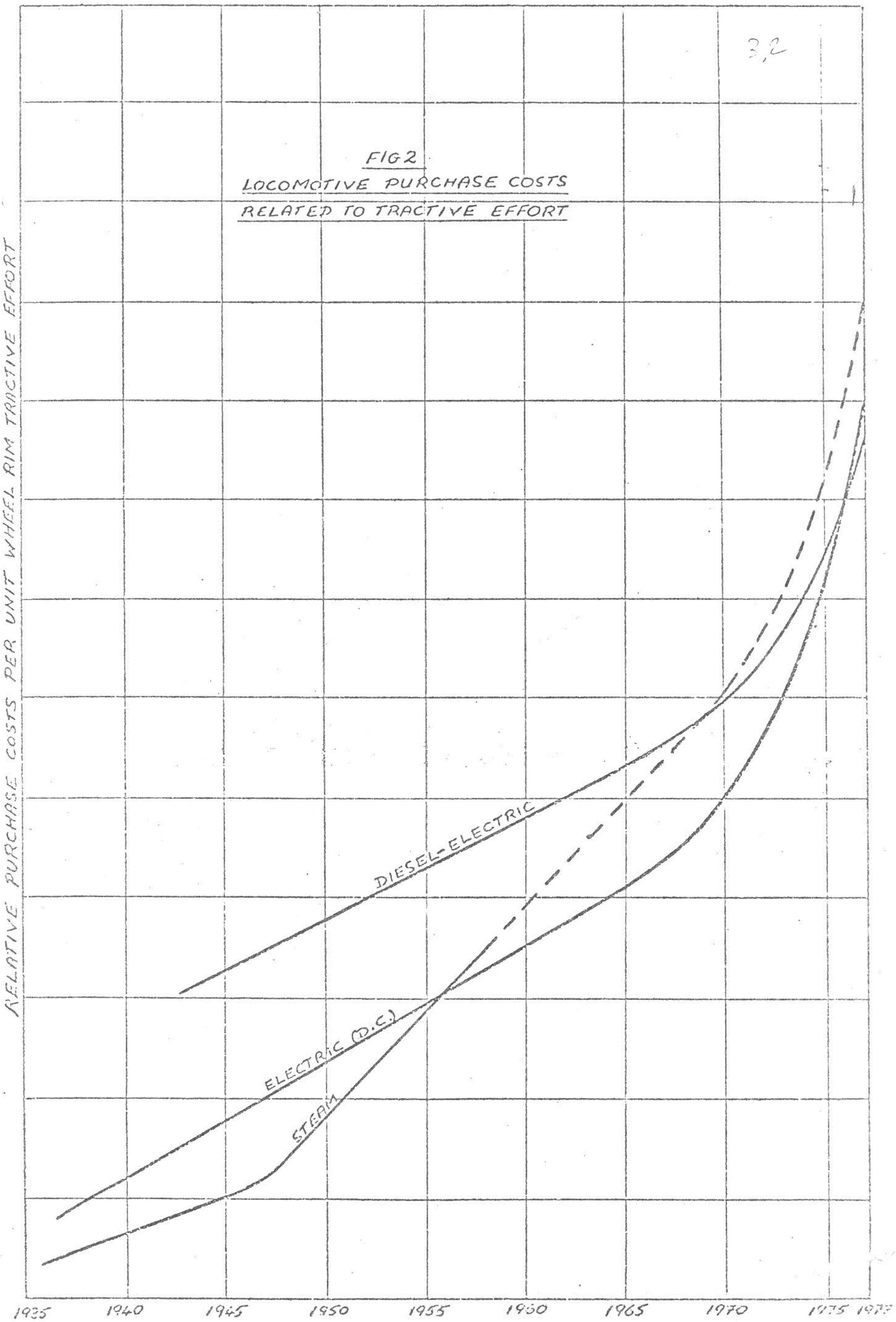
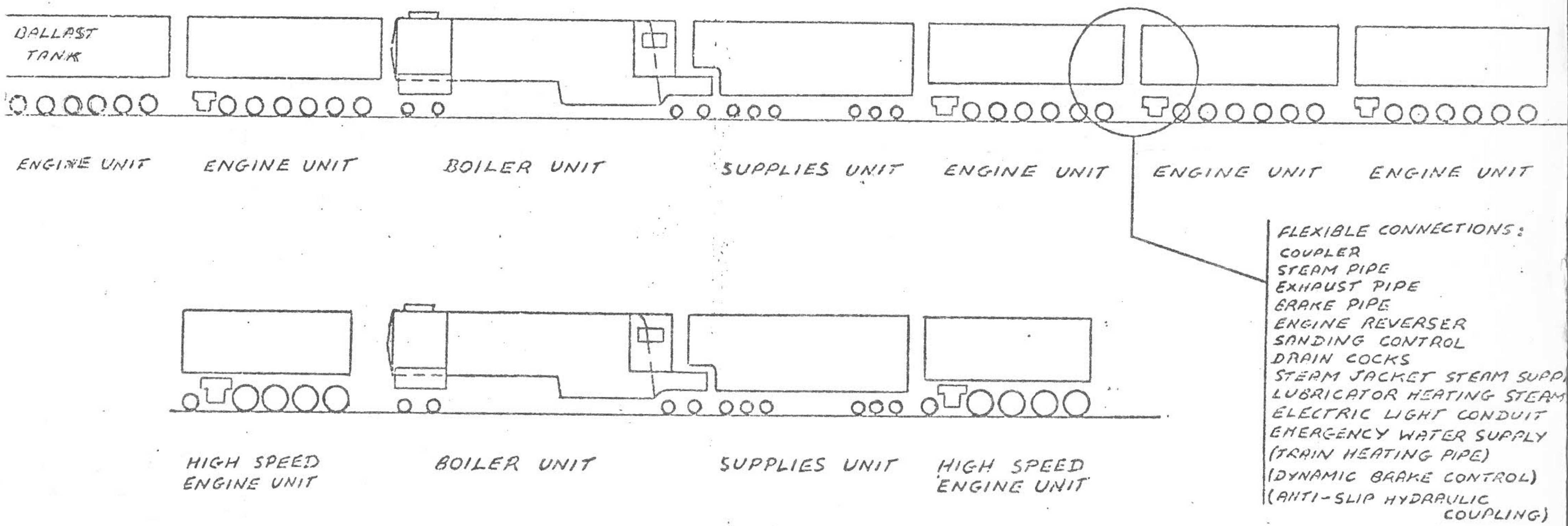


FIG 2
LOCOMOTIVE PURCHASE COSTS
RELATED TO TRACTIVE EFFORT

FIG. 6

THE SEPERABLE STEAM LOCOMOTIVE

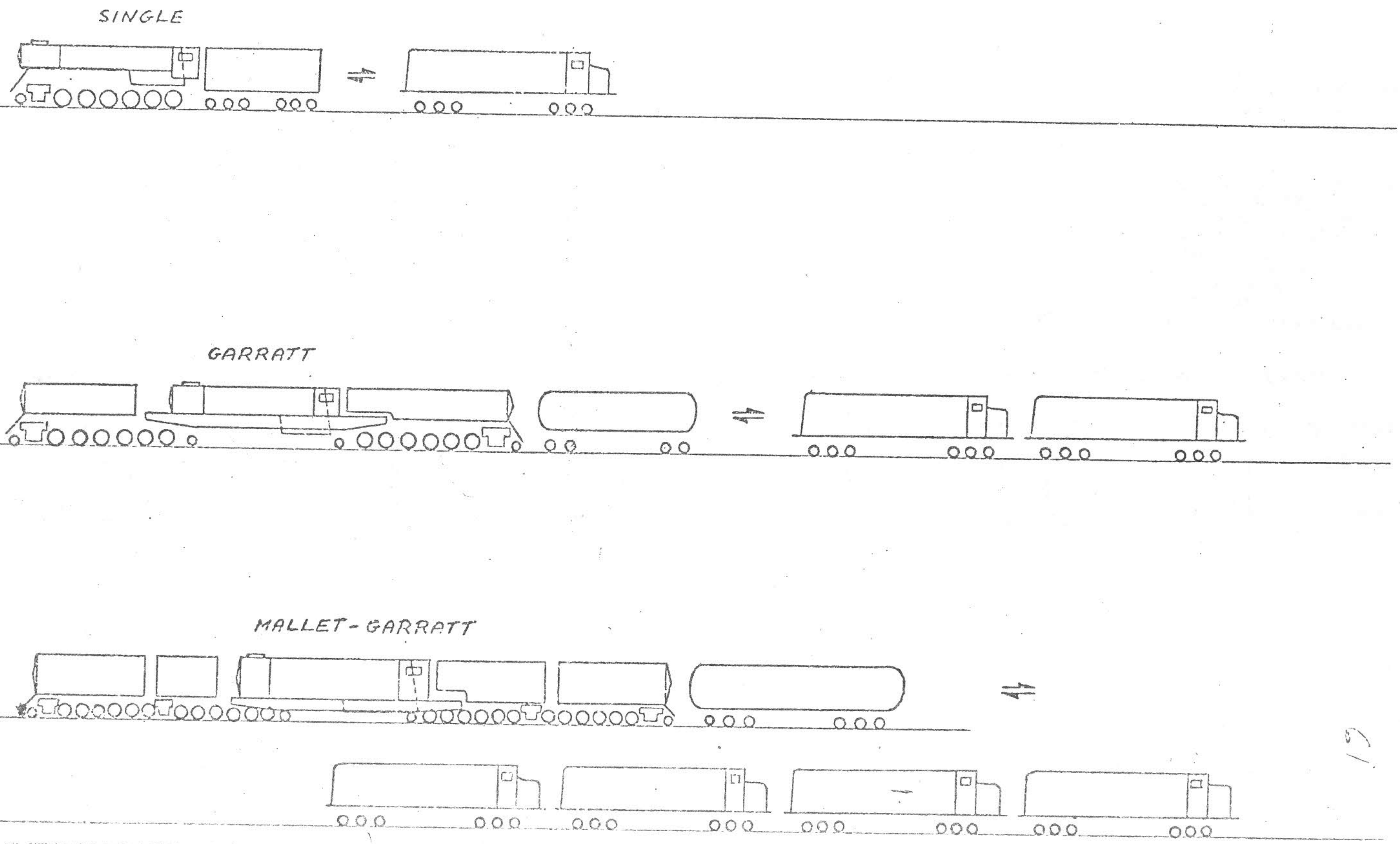


BOTH LOCOMOTIVES HAVE THE SAME BOILER UNIT & WHEEL RIM POWER. THE UPPER LOCOMOTIVE CAN DELIVER THIS POWER AT LOW SPEED & IS FOR VERY HEAVY TRAINS. THE LOWER LOCOMOTIVE, FOR FAST FREIGHT, DISPENSES WITH THE ADDITIONAL LOCOMOTIVE WEIGHT WHICH REDUCES DRAWBAR POWER AT HIGH SPEED.

FIG 5

ENLARGING THE STEAM LOCOMOTIVE.

⇌ MEANS EQUAL IN ADHESIVE WEIGHT



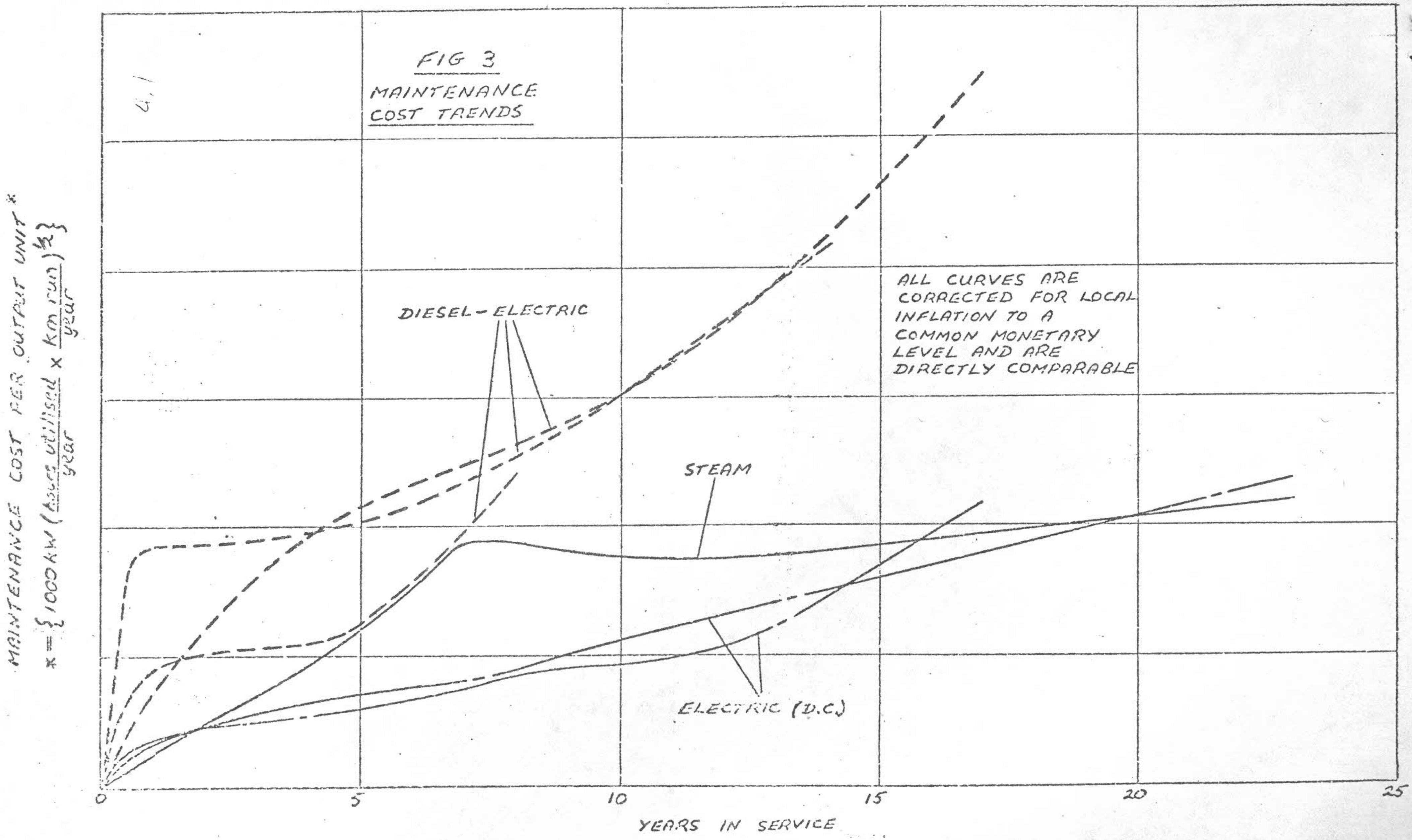
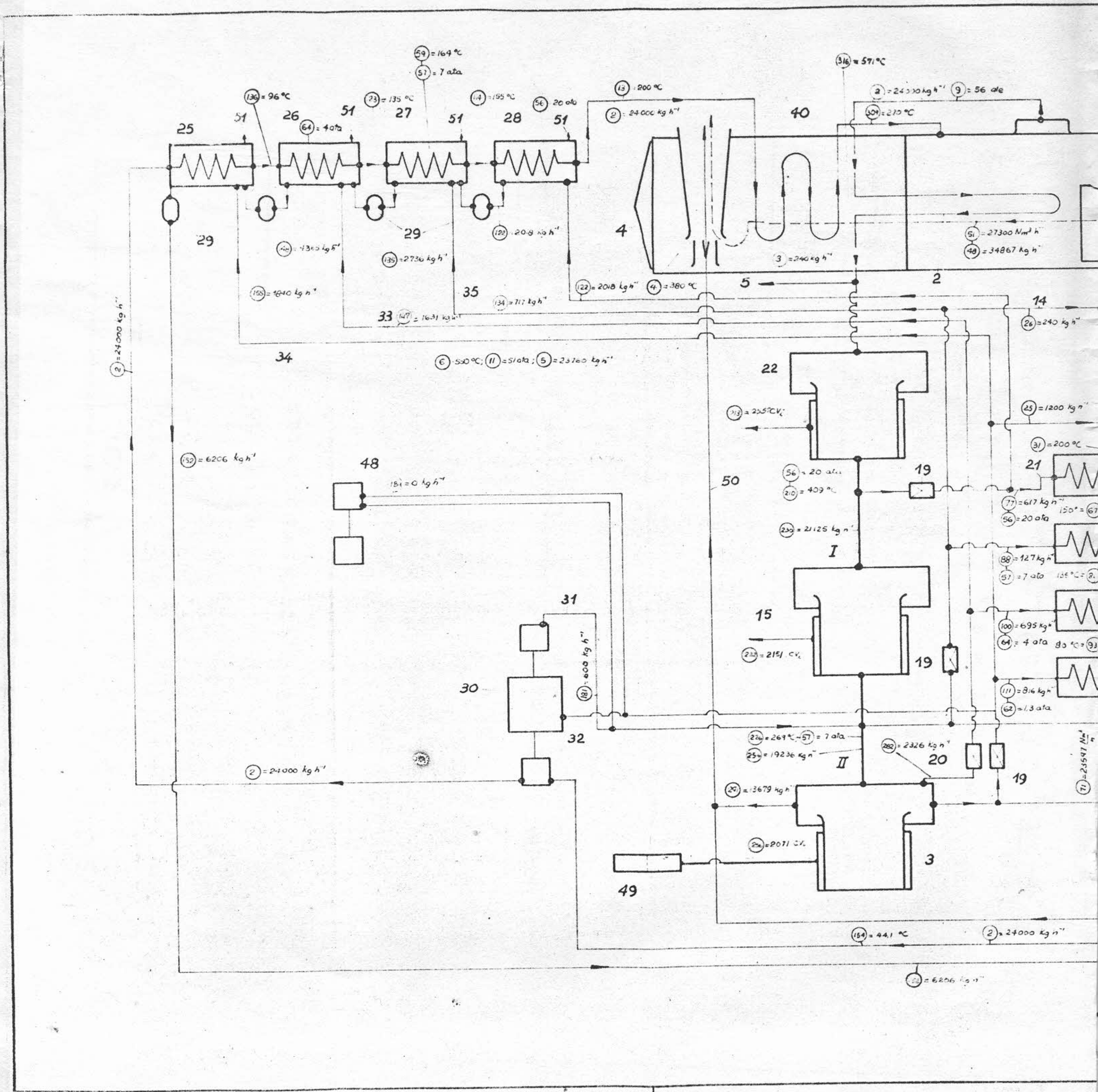
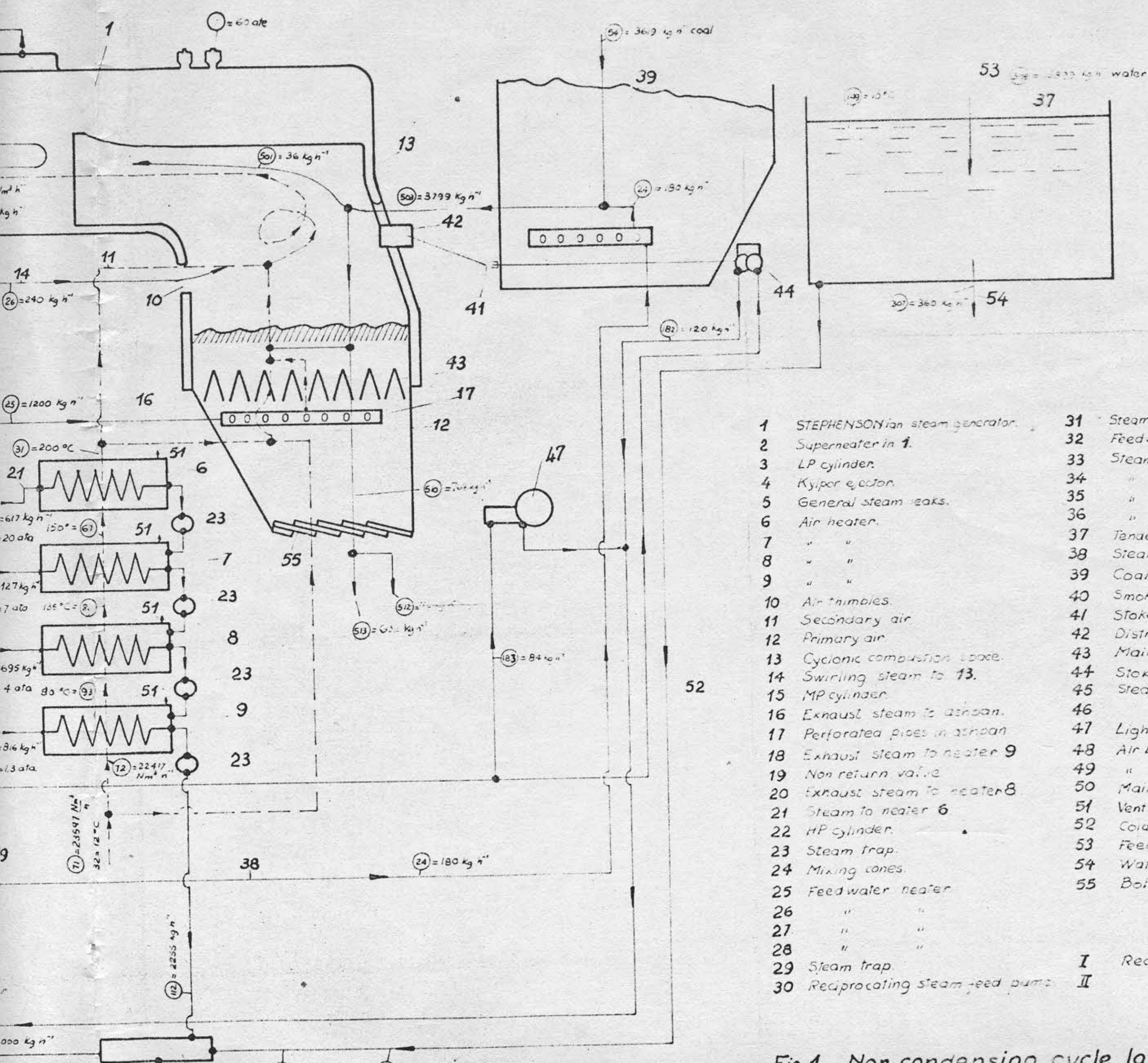


FIG 3
MAINTENANCE COST TRENDS



(59) = 164 °C
(57) = 7 ata
(136) = 96 °C
(73) = 135 °C
(14) = 155 °C
(56) 20 ata
(13) = 200 °C
(2) = 24 000 kg h⁻¹
(316) = 571 °C
(2) = 24 000 kg h⁻¹ (9) = 56 ata
(301) = 270 °C
25
26
27
28
51
(64) = 4 ata
40
4
29
(135) = 2736 kg h⁻¹
(122) = 2018 kg h⁻¹
(3) = 240 kg h⁻¹
(51) = 27300 Nm³ h⁻¹
(48) = 34867 kg h⁻¹
(155) = 1840 kg h⁻¹
35
(134) = 717 kg h⁻¹
(122) = 2018 kg h⁻¹ (4) = 380 °C
5
2
14
33 (147) = 1631 kg h⁻¹
(26) = 240 kg h⁻¹
34
(2) = 24 000 kg h⁻¹
(6) = 550 °C; (11) = 51 ata ; (5) = 23760 kg h⁻¹
22
(25) = 1200 kg h⁻¹
(31) = 200 °C
21
(152) = 6206 kg h⁻¹
48
(56) = 20 ata
(210) = 409 °C
19
50
(77) = 617 kg h⁻¹
(56) = 20 ata
(230) = 21125 kg h⁻¹
I
(88) = 127 kg h⁻¹
(57) = 7 ata
31
15
(100) = 695 kg h⁻¹
(64) = 4 ata
30
19
(111) = 816 kg h⁻¹
(62) = 1,3 ata
32
(57) = 7 ata
(254) = 19236 kg h⁻¹
(282) = 2326 kg h⁻¹
II
20
19
(2) = 24000 kg h⁻¹
(71) = 23597 Nm³ h⁻¹
(291) = 13679 kg h⁻¹
3
49
(154) = 44,1 °C
(2) = 24000 kg h⁻¹
= 6206 kg h⁻¹

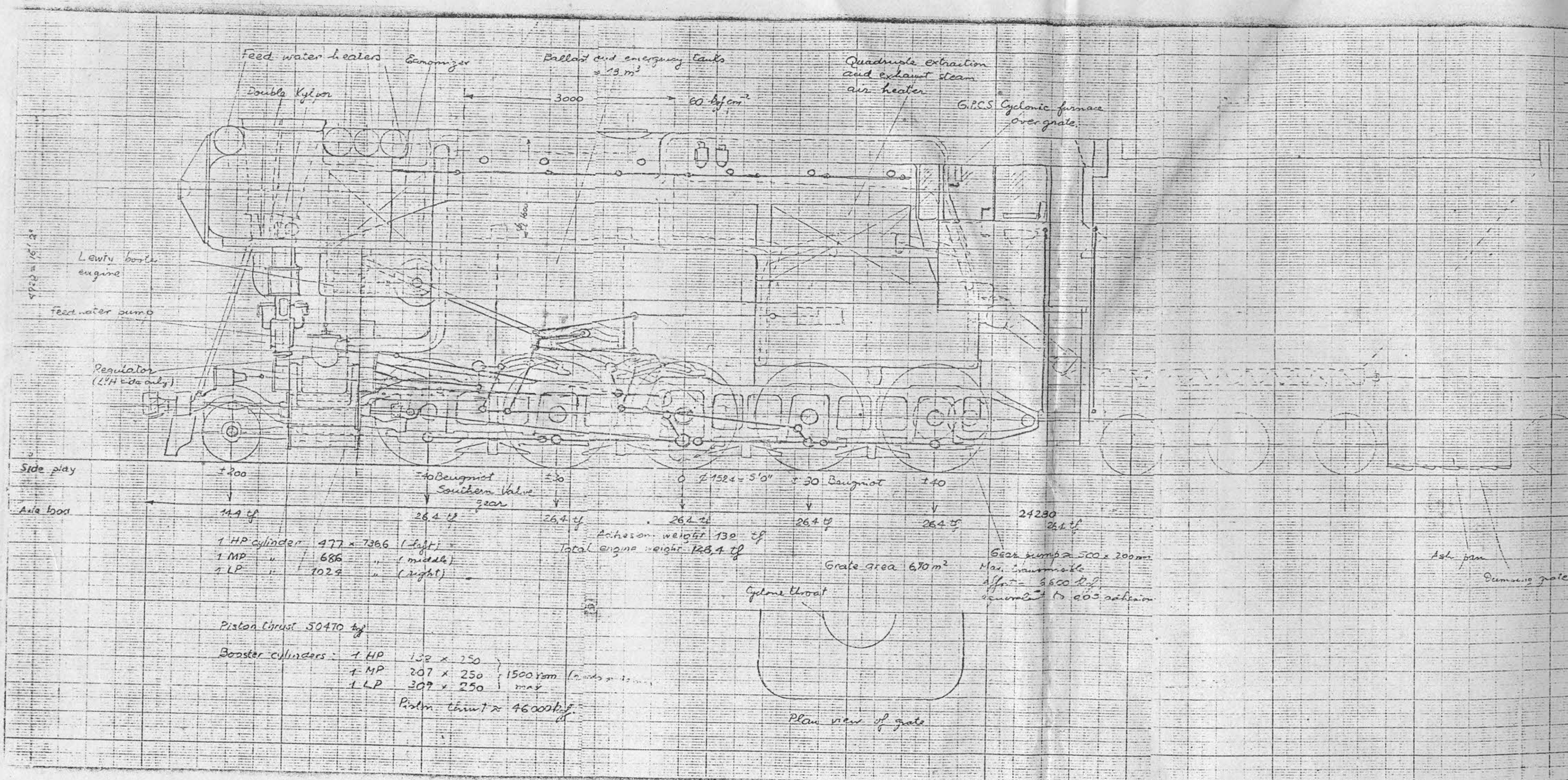


1 STEPHENSONian steam generator.
2 Superheater in **1**.
3 LP cylinder.
4 Kylpor ejector.
5 General steam leaks.
6 Air heater.
7 " "
8 " "
9 " "
10 Air thimbles.
11 Secondary air.
12 Primary air.
13 Cyclonic combustion space.
14 Swirling steam to **13**.
15 MP cylinder.
16 Exhaust steam to ashpan.
17 Perforated pipes in ashpan.
18 Exhaust steam to heater **9**
19 Non return valve
20 Exhaust steam to heater **8**.
21 Steam to heater **6**
22 HP cylinder.
23 Steam trap.
24 Mixing cones.
25 Feedwater heater.
26 " "
27 " "
28 " "
29 Steam trap.
30 Reciprocating steam-feed pump.
31 Steam feed to **30**.
32 Feedwater pump exhaust.
33 Steam feed to heater **26**
34 " " " " **25**
35 " " " " **27**
36 " " " " **28**
37 Tender tank.
38 Steam to wet coal.
39 Coal on the tender.
40 Smokebox economizer.
41 Stoker screw.
42 Distributing head.
43 Main grate.
44 Stoker engine.
45 Steam to **44**.
46
47 Light generator engine
48 Air brake pump.
49 " " booster pump
50 Main exhaust piping
51 Vent.
52 Cold water pipe
53 Feed to tender tank
54 Water for various uses
55 Bottom grate.

I Receiver.
II "

Fig 1 Non condensing cycle layout and quantities.

Indicated quantities refer to engine in warm up


Feed water heaters
Economizer
Ballast and emergency tanks ≈ 19 m³
Quadruple extraction and exhaust steam air heater
Double Kylpor
3000
60 kgf/cm²
G.P.C.S. Cyclonic furnace over grate.
4920 = 16'2"
Lewis booster engine
Feed water pump
Regulator (L.H. side only)
Side play
Axle load
±200
±40 Beugniot
Southern Valve gear
±30
0
⌀ 1524 = 5'0"
± 30 Beugniot
±40
14.4 tf
26.4 tf
26.4 tf
26.4 tf
26.4 tf
26.4 tf
24280
26.4 tf
Adhesion weight 130 tf
Total engine weight 148.4 tf
1 HP cylinder 477 × 7366 (left)
1 MP " 686 " (middle)
1 LP " 1024 " (right)
Grate area 6,70 m²
Gear pump ≈ 500 × 200 mm
Cyclone throat
Ash pan
Dumping grate
Piston thrust 50470 kgf
Booster cylinders : 1 HP 132 × 250
1 MP 207 × 250
1 LP 309 × 250
1500 rpm max
Piston thrust ≈ 46000 kgf
Plan view of grate

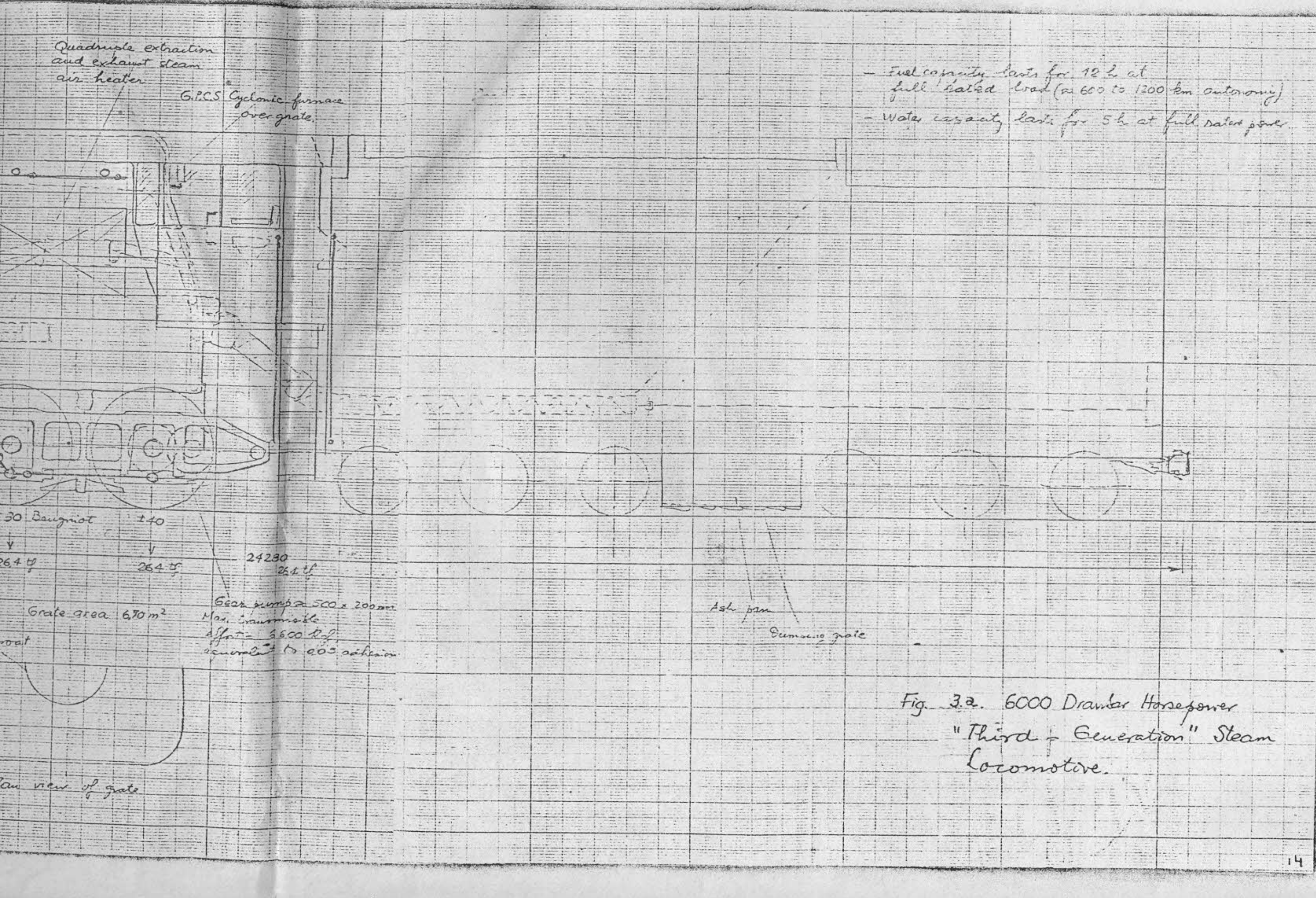


Fig. 3a. 6000 Drawbar Horsepower "Third - Generation" Steam Locomotive.