

Aug. 22, 1933.

A. LYSHOLM ET AL

1,924,062

TURBINE DRIVEN LOCOMOTIVE

Filed Nov. 26, 1932

3 Sheets-Sheet 1

*Fig. 1.*

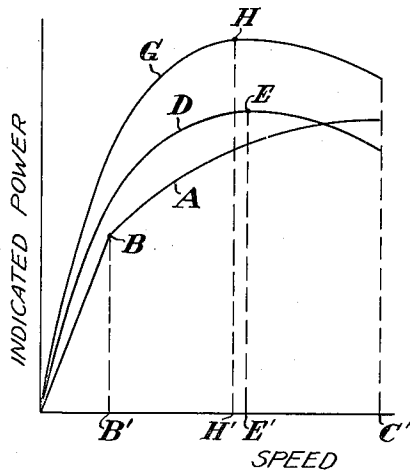


*Fig. 2.*

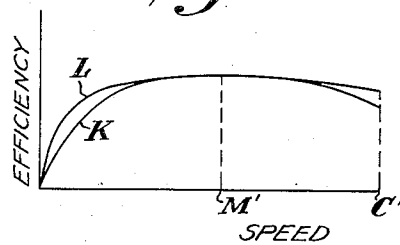


*Fig. 3.*

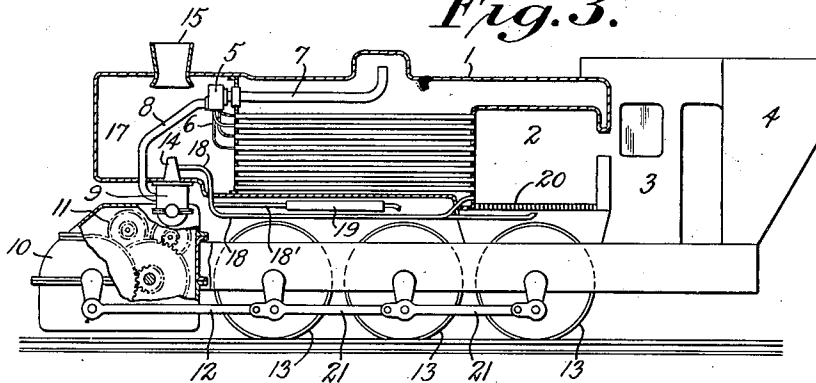
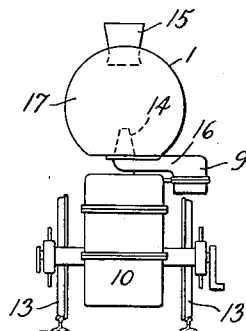


*Fig. 4.*



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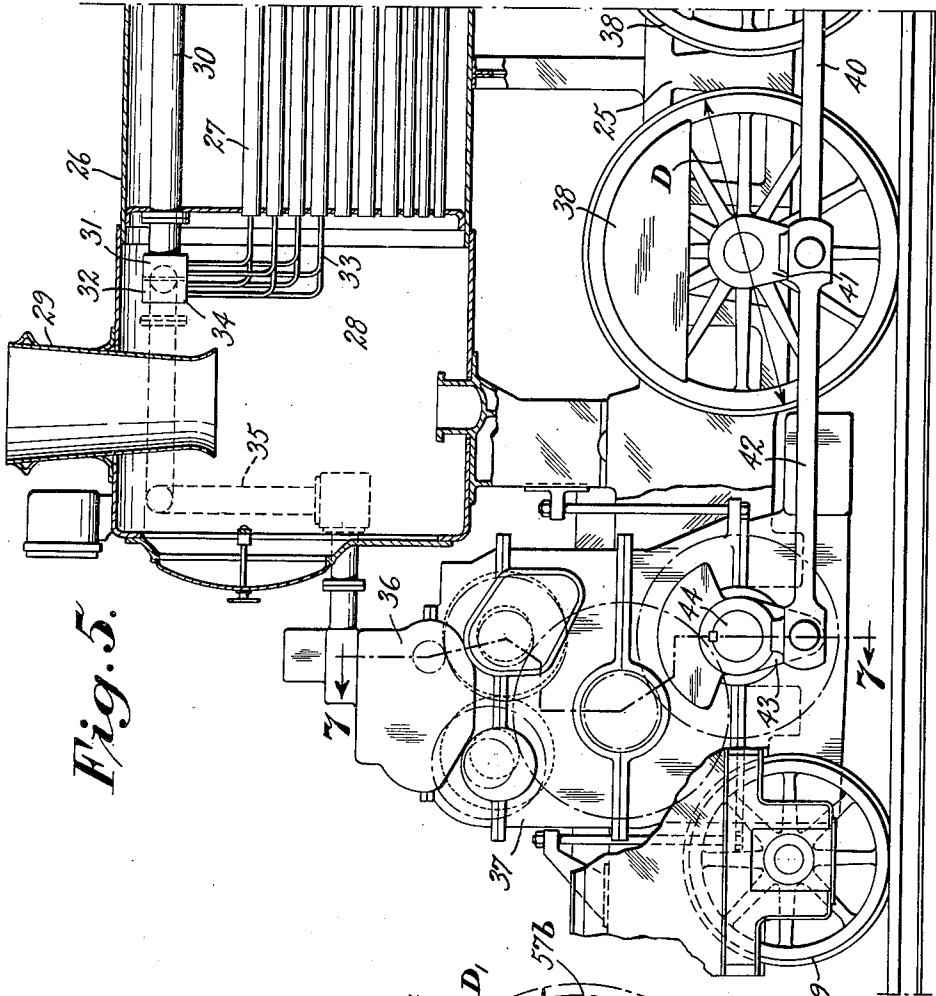
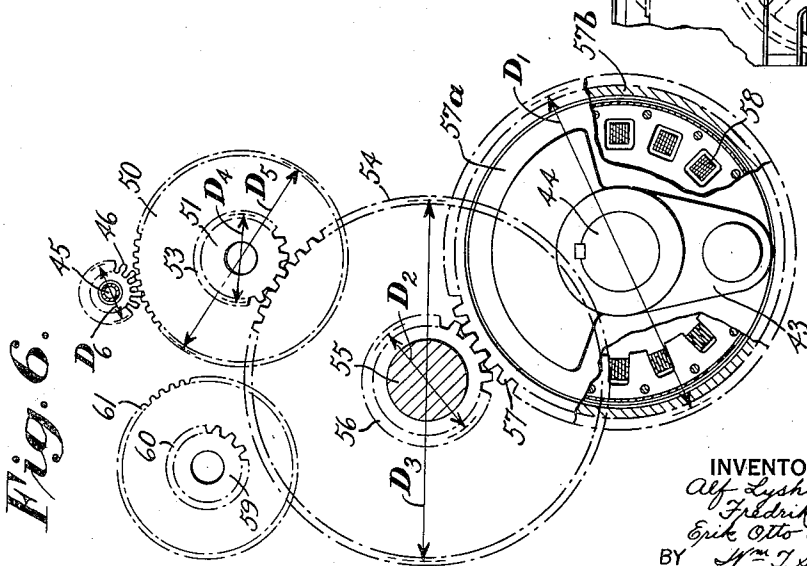
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3 Sheets-Sheet 2



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Filed Nov. 26, 1932

3 Sheets-Sheet 3

*Fig. 7.*

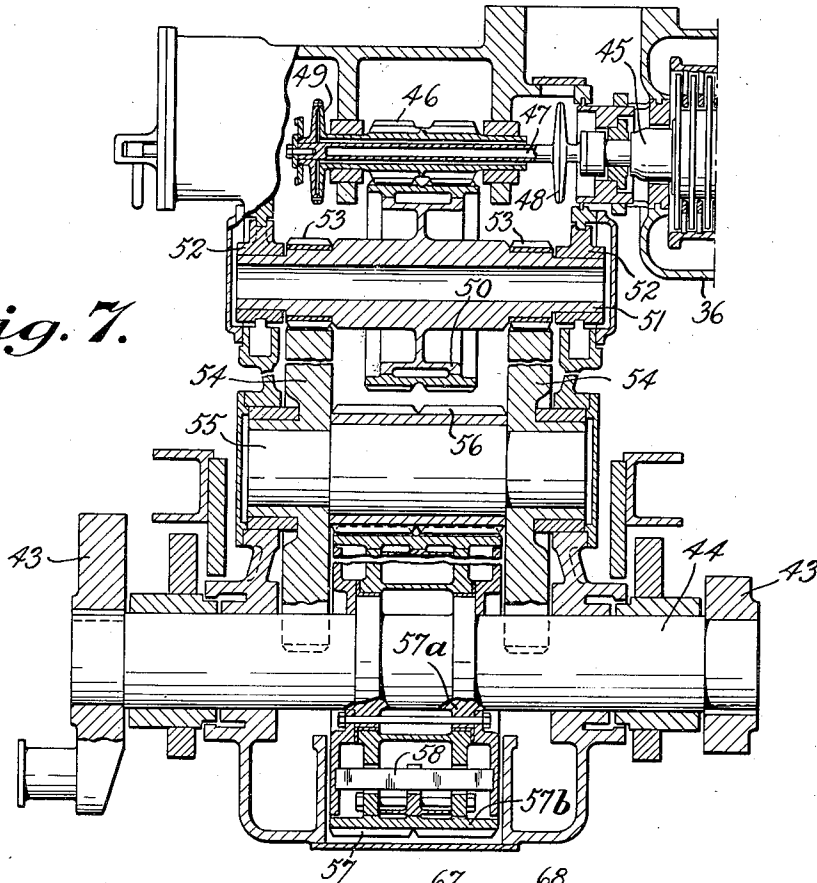
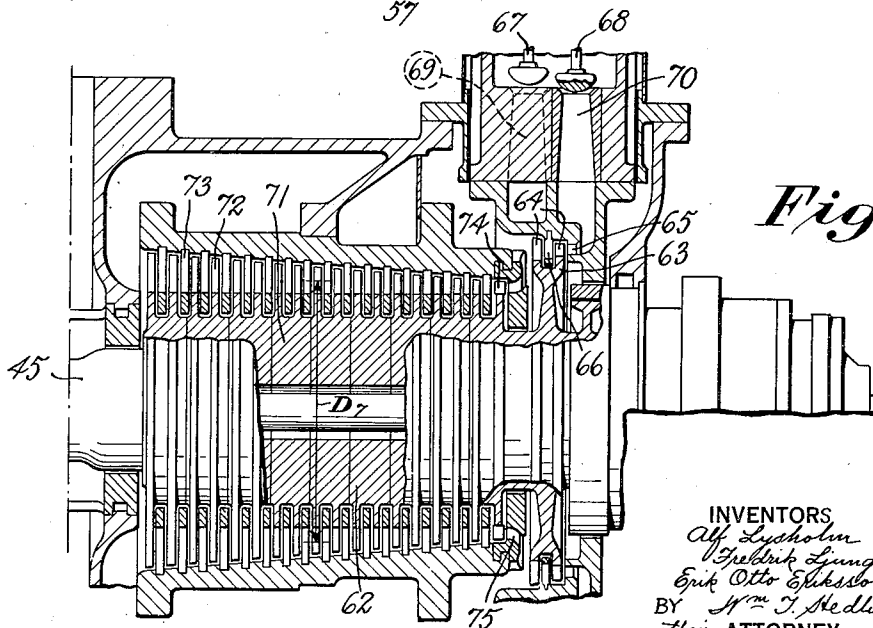


*Fig. 8.*



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## UNITED STATES PATENT OFFICE

1,924,062

## TURBINE DRIVEN LOCOMOTIVE

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Application November 26, 1932, Serial No. 644,474,  
and in Sweden January 14, 1928

8 Claims. (Cl. 105—38)

This application is a continuation in part replacing our copending application Serial No. 333,173, filed January 17, 1929, and, as to all common subject matter, relates back to said application Serial No. 333,173, and to foreign applications corresponding thereto for all dates and rights incident to the filing thereof.

The present invention relates to turbine driven locomotives and has for its general object the provision of a locomotive of the above type capable of fulfilling the requirements of locomotive operation as to size, weight, cost, tractive effort, etc., while at the same time operating with a high degree of economy over the range of speeds encountered in normal locomotive service. More particularly, the invention aims to provide a turbine driven locomotive having the above characteristics which is of the non-condensing type and which may be of a size having relatively great power capacity. Still more particularly the invention aims to provide a locomotive of the above type in which the motive fluid is supplied to the turbine at relatively high pressure and temperature so that the turbine, even though operating as a non-condensing unit, will, due to the condition of the motive fluid as delivered to the turbine and to the other characteristics inherent in the structure when built in accordance with the invention, achieve an operating economy comparable to or improving upon the operating economy of condensing units for driving locomotives, as heretofore constructed, thus providing a locomotive of high operating efficiency without involving the relatively great cost of a condenser.

For the purpose of obtaining the above and other more detailed objects, we provide a turbine locomotive in which the speed of the turbine is so related to the speed of the driving wheels of the locomotive that the speed of rotation of the turbine at which the maximum efficiency is obtained is less than the maximum speed of the locomotive. Preferably, in accordance with the invention, the speed of the turbine at which maximum efficiency is reached is between twenty-five to seventy-five percent of the maximum locomotive speed. By the term "maximum speed" as herein employed, we mean the maximum speed at which the locomotive and its propelling mechanism are intended to operate under normal conditions, as distinguished from excess or abnormally high speeds or "runaway" speed.

Further, and in order to secure the maximum benefit from the invention, we provide a turbine locomotive including a turbine incorporated in the locomotive in a manner providing for operation

with relatively uniform high efficiency over a comparatively wide speed range embracing the most frequently employed speeds of operation of the locomotive.

Stated broadly, we provide, as a principal factor in obtaining the objects of the invention, a locomotive having a turbine of a type suitable for locomotive operation and incorporated in the locomotive in a manner such that the characteristic of the turbine known as the "Parsons characteristic value" (see Stodola, Steam and Gas Turbines) and hereinafter referred to as the Parsons figure, is, when the turbine is operating at maximum speed, greater than 2800 (the factors determining the Parsons figure being expressed in metric units). Preferably, the locomotive is so designed, having regard to the number of stages and speed of operation of the turbine blading, that the sum of the squares of the speeds of the turbine blades at maximum speed is greater than 600,000 meters<sup>2</sup>/seconds<sup>2</sup>. We further provide a locomotive having a boiler and a superheater delivering superheated steam to the turbine, which steam is preferably generated so as to be delivered to the turbine at a pressure of at least twenty atmospheres and superheated to a degree giving a total temperature of at least 400° C.

More specifically, we prefer to employ a turbine of the combined type, that is, a turbine having both impulse and reaction blading, and to utilize the different types of blading in a manner hereinafter to be more fully pointed out.

For a better understanding of the nature of the invention and the manner in which it is carried into effect, reference may be had to the accompanying drawings forming a part of this specification and the following description thereof.

In the drawings:

Fig. 1 is a diagram illustrating performance characteristics of locomotives having different kinds of prime movers;

Fig. 2 represents diagrammatically certain locomotive turbine efficiency characteristics;

Fig. 3 is a diagrammatic side elevation, partly in section, of a locomotive suitable for carrying the invention into effect;

Fig. 4 is a diagrammatic front end view of the locomotive shown in Fig. 3;

Fig. 5 is a side elevation, partly in section and on a larger scale, of a portion of a locomotive embodying the invention;

Fig. 6 is a diagrammatic side elevation, on a

still larger scale, of the gearing indicated in Fig. 5;

Fig. 7 is a section taken on the line 7—7 of Fig. 5; and

- 5 Fig. 8 is a longitudinal central section of a preferred form of turbine for use in the arrangement shown in Fig. 5.

In order to facilitate an understanding of the invention, we will compare the operating characteristics of a reciprocating engine locomotive with apparatus in accordance with the invention.

Referring now to Fig. 1, the abscissae of the co-ordinate system represent speed of the locomotive and the ordinates thereof represent the indicated power developed by the engine. Curve A designates the indicated power of a reciprocating engine at various locomotive speeds. From zero value to the point B on the curve, which point corresponds to a certain speed B' of the locomotive, the reciprocating engine operates with constant admission or fixed point of cut-off of steam to the cylinders and with a varying amount of steam. At speeds greater than B', steam admission per working stroke varies inversely to the speed of the locomotive, assuming the quantity of steam to remain constant. The reason for this is that at speeds greater than B', with a given constant quantity of steam, admission of steam to the cylinder is cut off at an earlier point in the working stroke of the piston and the steam expands in the cylinder to a lower pressure. It is well known that decreasing the cut-off to the cylinder increases the indicated work done by the steam in expanding in the cylinder. This is evidenced by the increased area of the indicator diagram and since the increase in work produced is obtained with a constant steam input, the efficiency of the engine is increased. Curve A shows that the indicated power output of the reciprocating engine, with constant steam admission, and thus the efficiency thereof, increases with the speed of the locomotive. The greatest efficiency is obtained at the speed indicated at C' which speed corresponds to the maximum speed of the locomotive.

It is, of course, possible to have a valve setting such that at low speeds of the locomotive, the point of cut-off of steam admission occurs so early in the working stroke of the piston that when the speed of the locomotive is increased to maximum, the cut-off is reduced to such a degree that the indicated work decreases before the maximum speed is reached. Under such conditions, the efficiency of the engine is greater at lower speeds than at maximum speed. However, the last-mentioned conditions are purely theoretical and not practical. A relation such as that above assumed between the cut-off and the maximum speed is not practical because of the very large size of the cylinders required. The size of locomotive cylinders is definitely limited because of the clearance limits imposed upon locomotives. In addition to the difficulty of obtaining, in a locomotive, cylinders of the size required for the above assumed conditions, increase in the size of the cylinders also results in increases in mechanical losses, radiation losses, clearance losses and leakage losses. These losses will be present at all locomotive speeds and so reduce the efficiency of the engine over its entire speed range as to preclude any gain in the over-all efficiency of the engine due to adjustment of the cut-off so as to obtain maximum engine efficiency at a speed below maximum engine speed.

Curve D designates the indicated power output for a turbine locomotive in accordance with the present invention, the curve being based upon the same initial steam pressure and temperature as for the reciprocating engine whose indicated output is shown by curve A. As will be observed from curve D, the maximum power output of the turbine is obtained at point E.

The curve G illustrates the indicated power output of a turbine locomotive also constructed in accordance with the present invention, but designed to operate with a higher initial steam pressure and temperature than that employed in the engine and turbine whose characteristics are shown by curves A and D, respectively. In the present instance, the maximum power output is obtained at point H. From curves D and H, it will be observed that the maximum power outputs of turbines, and thus the efficiencies thereof, are obtained, respectively, at speeds E' and H', which speeds are lower than the maximum speed C'. These curves, which represent the indicated power outputs of the turbines, are plotted on the basis of the use of a constant steam quantity and therefore, are indicative of turbine efficiencies at the various speeds.

In stationary steam power plants where the speed of the prime mover is constant and normal speed is the maximum speed thereof as hereinbefore defined, the prime mover should be designed so that the greatest efficiency is theoretically obtained at its normal operating speed, that is to say, its maximum speed. In actual practice, particularly in the case of turbines, highest efficiency thereof is frequently obtained at a speed somewhat above the maximum speed, because of the fact that to obtain a turbine which will operate with maximum theoretical efficiency at maximum speed involves so much additional cost, as compared with the cost of a turbine operating at somewhat lower than maximum theoretical efficiency at its maximum speed, that the latter type of turbine is economically the most advantageous to employ. In locomotives, on the other hand, the operating conditions are entirely different, the normal average speed usually being considerably below the maximum speed and the normal speed also varying over a considerable range. In accordance with the present invention, the locomotive is designed so that maximum operating efficiency is obtained in the range of speeds, below maximum speed, at which the locomotive is generally operated. Experience has shown that this range includes from between 25% to 75% of the maximum locomotive speed.

To obtain a locomotive turbine having characteristics of operation such as to make it suitable for the varying conditions of load and speed imposed by locomotive operation and also having the desired properties for carrying the present invention into effect, known principles of design may be employed as to the details of the turbine design and within the scope of the invention the turbine may take various specific forms, the details of the turbine design being dictated to a material degree by the operating characteristics desired for the specific type of locomotive in which the turbine is incorporated.

Turbines as known today may be divided into three different classes, impulse turbines, reaction turbines and combined turbines, that is, turbines of the type in which both impulse blading and reaction blading stages are employed. For reasons which are well known, neither the pure impulse turbine nor the pure reaction turbine is

suitable as a driving unit for locomotives, while the combined type turbine is suitable for such purpose since the combined turbine incorporates the characteristics of the impulse and reaction types which are favorable with respect to locomotive operating conditions and minimizes or eliminates the characteristics of impulse and reaction turbines which are not favorable with respect to locomotive operating conditions.

A turbine of the combined type may, therefore, be used most advantageously in carrying our invention into effect. In locomotive practice, and in order to make use of our invention, the locomotive is designed so that the Parsons figure of the turbine is greater than 2800 when the locomotive, and consequently the turbine, is operating at maximum speed.

The Parsons figure is well known and has been widely used for many years in the turbine art as a ready means of indicating turbine characteristics without the necessity of having to take into account details of turbine design. It is in the nature of a comparator and the value of the figure is obtained by dividing the sum of the squares of the peripheral velocities of the turbine blading, or, expressed in another way, the mean square of the peripheral velocities of the blading times the number of stages, by the adiabatic heat drop in the turbine. The peripheral velocity of the turbine blading we will designate by the letter "u". The adiabatic heat drop in the whole turbine is designated by  $\lambda$ . The Parsons figure, designated by X, equals

$$\frac{\sum u^2}{\lambda}$$

or, expressed in words, the sum of the squares of the blade speeds divided by the adiabatic heat drop in the whole turbine. For the values of the Parsons figure as herein employed, "u" is expressed in terms of meters per second and  $\lambda$  is expressed in kilogram calories. It is this Parsons figure of

$$\frac{\sum u^2}{\lambda}$$

which we make greater than 2800, and the factor  $\sum u^2$  which we prefer to make greater than 600,000 meters<sup>2</sup>/seconds<sup>2</sup>, at maximum locomotive speed.

The Parsons figure is used in connection with the present invention in preference to speed or velocity triangles because of the fact that this figure is characteristic of the turbine as a whole, while velocity diagrams are usually different for different stages in the same turbine. Thus by the use of the Parsons figure, those skilled in the art may determine readily the manner in which the locomotive must be designed in order to provide the turbine characteristics necessary in order to carry the invention into effect. At the same time, the locomotive designer is not limited in his choice of different arrangements involving differences in the details of the design of the locomotive as a whole or of the turbine, which details may be governed by well known principles of design.

In order to understand the bearing on the present invention of the value of 2800 for the Parsons figure, reference may be had to Fig. 2 in which are plotted the more or less diagrammatic curves K and L, in a coordinate system in which locomotive speeds are represented by abscissæ and efficiencies of the turbine are ordinates. The curve K indicates the efficiency of a turbine in which allowance is not made for losses due to impact in the blade system. The upper curve L indicates the efficiency of a turbine

in which allowance is made for such impact losses. On the latter curve, the maximum efficiency of the turbine occurs at a locomotive speed M' which is substantially half of the maximum speed C'. As hereinbefore mentioned, the combined type of turbine is the most suitable type for locomotive propulsion, but for different types of locomotives the relation between the amount of impulse blading and reaction blading may be different if best results are to be obtained. Generally speaking the higher the initial steam pressure, the greater should be the percentage of the impulse blading employed, but we have found and consider 50% impulse blading to be a maximum for satisfactory performance. With relatively low initial steam pressures as little as 10% impulse blading may be employed and for locomotive boiler pressures of the order now often employed, that is to say, about 12 to 14 atmospheres, 30% impulse blading represents a desirable proportion.

Considering the Parsons figure for a moment, it can be shown theoretically that for impulse blading, maximum efficiency will be obtained when the Parsons figure has a value of about 1800 and for reaction blading, maximum efficiency will be reached when the value of the Parsons figure is about 3800. Thus, for a combined type turbine in which the maximum amount of impulse blading is employed, maximum efficiency is to be expected when the Parsons figure has a value of approximately 2800 and this value therefore becomes in the nature of a critical value for the purposes of the present invention, since as the percentage of impulse blading is decreased, the value of the Parsons figure at which maximum efficiency is obtained, increases.

The benefits of our invention are obtained in the manner which we contemplate by designing the locomotive so that the Parsons figure of the turbine at maximum locomotive speed is greater than 2800. In order to secure the maximum benefits from the invention, it is advantageous to design the locomotive so that the value of the Parsons figure at maximum locomotive speed is materially greater than 2800, and, as will hereinafter be more fully explained, at least 4000. Thus, in the more or less diagrammatic curves which we have used for illustrative purposes in Fig. 2, the Parsons figure for the turbine at maximum locomotive speed, that is, at speed C', which is not quite double the speed M', is approximately 10,000 for a turbine the maximum efficiency of which is reached when the value of the Parsons figure is 2800, since the Parsons figure varies approximately as the square of the speed of the blading, due to the presence of the factor  $u^2$  in the formula

$$\frac{\sum u^2}{\lambda}$$

and the fact that mere change in speed has little effect upon the value of the factor  $\lambda$ .

From the character of the curves indicated, it will be evident that if the speed is less than M', the Parsons figure for the turbine at such lesser speed will be less than 2800 and if the speed is greater than M' the Parsons figure for the turbine at the greater speed will be greater than 2800. From this it follows that if a locomotive is designed so that at maximum speed the Parsons figure is less than 2800, the locomotive will be working always on the descending part of the efficiency curve to the left of M'. This, obviously, will result in low average efficiency of operation. On the other hand, if the locomotive is designed so that the Parsons figure at maximum speed exceeds 2800, when at a speed such as M'

the Parsons figure will be 2800, and the maximum efficiency of the turbine will be reached at this speed  $M'$ , which may be and preferably is materially lower than the maximum speed of the locomotive. The locomotive will then operate at relatively high efficiency, due to the fact that the normal operating range of the locomotive will be between maximum speed and the speed  $M'$  at which maximum efficiency occurs. This range, due to the efficiency characteristics of turbines with variations in speed, will give a very much higher average overall efficiency than will the range on a curve which reaches its maximum efficiency at the maximum speed of the locomotive.

As is well known in the art, blades for steam turbines should be designed in such a manner that the inlet angle of the motive fluid relative to the blades, for the normal speed of the blades, is such that only minimum shock losses are incurred. In locomotive turbines, the blade speed will vary over a comparatively wide range and for this reason it is advantageous, if the best efficiency is to be obtained, to select a type of blade which is adapted to be used for widely varying inlet angles of the motive fluid relative to the blading without incurring appreciable losses due to eddies and the like in the flow of the motive fluid. A blade fulfilling the above requirement for variable speed locomotive turbines is disclosed in United States Reissue Patent No. 18,485 granted to Alf Lysholm May 31, 1932.

The curve K in Fig. 2 is based on the use of such a blade, and from this curve it will be evident that the difference in efficiency incident to the use of this blade as compared with the efficiency incident to the use of this blade as compared with the efficiency obtained under the conditions represented by the curve L, in which curve allowance is made for impact losses, will be negligible, particularly in the normal operating speed range for the locomotive.

Where blading of this type is used, it is advantageous to use partial admission for controlling the speed and power of the turbine.

From a comparison of curves K and L in Fig. 2, it will be observed that when impact losses in a variable speed turbine are taken into account, the specific design of the turbine blade will have some influence on the specific degree of slope or curvature of the efficiency curve of the turbine, but for a given type of turbine the general characteristics of the turbine with respect to the speed at which maximum efficiency is reached will not vary materially for a given Parsons figure, provided that the design of the turbine is in accordance with known principles of turbine design and that the details of the blading are worked out with normal known factors of design in mind. It is thus evident that while, for a variable speed turbine, advantage may be derived from the employment of a special type of blade such as that disclosed in the above mentioned patent, the advantage so derived is due to the fact that the efficiency curve produced when such a blade is employed is relatively flat for a greater distance on each side of the point of maximum efficiency than would be the case if what may be termed a normal blade profile were used in the same turbine. In both cases, however, the normal characteristic of the efficiency curve of the turbine will be the same with respect to the speed at which maximum efficiency of the turbine is reached.

We have further found that for non-condens-

ing locomotive turbine drives, the benefits of the present invention may be enhanced if the turbine is constructed so that the maximum efficiency of the impulse blading is reached at a different and lower locomotive speed than that at which the reaction blading reaches maximum efficiency. While at first thought, it might appear that this relation of the locomotive speeds at which maximum efficiency of the two types of blades is reached follows from the fact that the value of the Parsons figure for maximum efficiency of impulse blading is lower than that for maximum efficiency of reaction blading, such is not necessarily the case. The reason for this is that in the case of the Parsons figure, the speed involved is peripheral velocity and the value of this speed as modified by the adiabatic heat drop is determinative of the value of the Parsons figure.

By relating the factors determinative of the speed of the blading so that the impulse blading reaches maximum efficiency at a locomotive speed lower than that at which the reaction blading reaches maximum speed, a flatter characteristic is obtained for the efficiency curve of the combined turbine than would be the case if both the impulse and the reaction blading reached maximum efficiency at the same locomotive speed, and when the value of the Parsons figure at maximum locomotive speed is in the range contemplated by us, it will be evident that in the normal operating speed range of the locomotive, the efficiency will not only be high but will also vary but little as compared with the variations in efficiency usually obtained.

In connection with the preceding discussion, it will be understood by those skilled in the art that while for purposes of explaining the nature of the present invention we have considered impulse and reaction blading from a theoretical standpoint, each being considered as converting energy solely in accordance with its respective principle of operation, the practical operation of turbine blading is such that in impulse blading some conversion of energy unavoidably occurs due to reaction and in reaction blading some conversion of energy unavoidably occurs due to impulse. For this reason there may be, as will hereinafter appear, a minor difference between the purely theoretical values obtainable and the results obtained in practice, but this difference in no wise affects the general principles of the invention or the manner of carrying it into effect.

It will further be understood that the determination of the Parsons figures as herein considered is based, in accordance with usual practice, on full steam admission and unthrottled heat drop.

Turning now more particularly to Figs. 3 and 4, we have shown a locomotive in diagrammatic fashion for illustrative purposes. The locomotive illustrated is of the non-condensing type and comprises a boiler 1 having a fire box 2. Reference numeral 3 indicates the usual cab and 4 the tender space for carrying fuel. Reference character 5 designates the header of the usual fire tube type of superheater, the superheating elements connecting the inlet and discharge chambers of the header being indicated at 6. Saturated steam from the boiler is supplied to the superheater through the dry pipe 7 and superheated steam is conducted through the pipe 8 to the turbine 9. A gear box indicated generally at 10 contains a set of speed reducing gears through which power is transmitted to a lay shaft connected by means of connecting rods 12 (one

only being shown in Fig. 3) to the driving wheels 13 of the locomotive, the latter wheels being connected by side rods 21 in the usual manner.

The exhaust steam from turbine 9 is conducted through the exhaust steam conduit 16 to the exhaust steam nozzle 14 which may be of conventional form and which acts as a draft creating ejector by discharging the exhaust steam upwardly through the locomotive stack 15.

If desired, some of the exhaust steam may be utilized for the purpose of preheating feed water and in assisting combustion. By way of example, we have shown diagrammatically the exhaust steam branch conduit 18 leading to the locomotive fire box and having a sub-branch 18' for conducting exhaust steam to the feed water heater indicated diagrammatically at 19.

In order to secure the best results from the present invention, it is desirable to utilize steam generated at high pressure and superheated to a relatively high degree. We have found that if the invention is embodied in a locomotive adapted to supply steam to the turbine at a pressure of at least 20 atmospheres and the steam is superheated to at least 400° C. It is possible to obtain a fuel economy comparable to that obtainable with present condensing turbine locomotives. We are, therefore, under these conditions, enabled to provide a locomotive giving the economy to be expected from present condensing locomotives but very much lower in cost and very much more practical in operation because of the elimination of the necessity for a condenser. It will be understood, however, that the invention is not limited in its application to so-called high pressure locomotives, and is also productive of materially improved results in locomotives having boilers delivering steam at relatively moderate pressures.

Turning now more particularly to Figs. 5 to 8 inclusive, we have shown by way of example a form of locomotive adapted to be operated with steam at the pressure and temperature given above and with a turbine having a Parsons figure at maximum locomotive speed which is well above the minimum value of 2800, so that maximum efficiency is reached at a speed materially below maximum speed. For purposes of illustration we have chosen a combined type of turbine which, as hereinbefore stated, we consider to be the most suitable type of turbine for meeting the various demands of locomotive service and which type of turbine we have found to be successful in a full sized locomotive which we have constructed in accordance with the present invention. It is to be understood, however, that the specific form of construction to be described is given merely by way of example for illustrating what we consider to be an advantageous form of general design. Our invention, however, is not to be considered as in any way limited to the general type of or specific design herein illustrated in these figures.

Referring now more particularly to Fig. 5, the locomotive comprises a frame 25 which may be of conventional construction and which serves to support the boiler 26 which, in the illustrated embodiment, is of the usual fire tube locomotive type. Combustion gases from the fire box (not shown) flow through the flues 27 to the smoke box 28 and from the latter pass to the atmosphere through the stack 29.

The steam generated in the boiler is conducted through the dry pipe 30 to the inlet compartment 31 of a superheater header 32 and from this compartment flows through the superheater elements

to the discharge compartment 34. The superheater may be of any suitable form. The type of superheater illustrated is that most commonly used in which the steam is superheated in its passage from compartment 31 to compartment 34 by flowing through a multiplicity of elements 33 extending into flues 27 and connecting these compartments in parallel.

Superheated steam is delivered from the superheater through the delivery pipe indicated at 35 to the admission ports of the turbine, presently to be described.

Turning now to the propelling mechanism for the locomotive, this comprises a turbine, the casing of which is indicated generally at 36, from which turbine power is transmitted through a set of speed reducing gears contained in the gear housing 37. The turbine casing and gear housing constitute a rigid unit which may be mounted on the locomotive in any suitable manner. In the present embodiment, this unit is carried by a forward extension of the main frame 25 of this locomotive. This extension may, depending upon the character of the locomotive, be rigid with frame 25 or articulated with respect thereto. Frame 25 is supported in the usual manner upon the main driving axles of the locomotive, which axles carry the driving wheels 38. The forward extension of the frame may be advantageously supported by a separate axle upon which are mounted bogie wheels 39.

The driving wheels 38, of which there may be a variable number depending upon the type of the locomotive, are connected by the usual side rods 40 pivoted to wheel cranks 41. A pair of connecting rods 42 serve to connect one pair of driving wheels 38 with the crank arms 43 on a lay shaft 44 which comprises the slow speed shaft of the reduction gearing in housing 37.

Turning now more particularly to Figs. 6 and 7 we have shown on enlarged scale a suitable form of reduction gearing for connecting the turbine with the lay shaft. This gearing may be constructed in accordance with designs known in the art.

The turbine shaft 45 (Fig. 7) is connected to and drives at shaft speed the pinion gear 46 which, in the embodiment illustrated, is in the form of a dual gear provided with resilient teeth for reasons which will hereinafter appear. The connection between shaft 45 and pinion 46 is advantageously through a flexible connection of known form comprising shaft 47 extending through pinion 46, the latter being hollow. Shafts 45 and 47 are connected by a flexible coupling indicated at 48 and shaft 47 and a suitable extension on the pinion 46 are connected by a second flexible coupling 49. Pinion 46 meshes with gear 50 fixed centrally to an intermediate shaft 51, this shaft being mounted in eccentric bearings 52 in the housing 37. Shaft 51 has fixed thereto the gears 53 which mesh with gears 54 fixed to a second intermediate shaft 55. Shaft 55 is suitably journaled in the gear casing 37. Intermediate the gears 54, shaft 55 has fixed thereto the gear 56 which meshes with gear 57 mounted on the lay shaft 44. Shaft 44 is also suitably journaled in the housing 37 and at its outer ends has fixed thereto the crank arms 43 which are advantageously disposed so that the crank pins carried thereby are angularly offset with respect to each other.

In the specific form of gearing illustrated the gear 57 comprises a web portion indicated at 57a which portion is rigidly secured to shaft 44,

and a rim portion 57b having gear teeth thereon and adapted to have limited turning movement peripherally with respect to the portion 57a. Force transmission through the gear is effected by a plurality of sets of springs 58 which engage both portions of the gear and which provide a certain degree of resiliency in the line of force transmission from the turbine to the lay shaft. The type of gear above described is known and need not be described herein in further detail.

In order to reverse the direction of drive of the locomotive without a separate reversing turbine, the intermediate shaft 51 is advantageously mounted, as previously described, in eccentric bearings 52 which permit this shaft to be moved from the position shown in dot-and-dash lines in Fig. 5, which position corresponds to the position shown in Fig. 6 in which gears 53 are in mesh with gears 54, to the position shown in dotted lines in Fig. 5 in which these gears are out of mesh. A set of reversing gears is provided, which gears are carried on an intermediate shaft 59 (see Figs. 5 and 6), these gears comprising a pair of gears 60 adapted to engage gears 54 and a gear 61 adapted to mesh with gear 50.

In Fig. 6, the gears are shown in mesh for forward running. For reverse running the shaft 51 is moved out of engagement in the manner described and in a similar manner shaft 59, which is also mounted in eccentric journals in the housing 37, is moved from the position shown in dot-and-dash lines in Fig. 5, which position corresponds to that shown in Fig. 6, to the position shown in dotted lines in Fig. 5. With shafts 51 and 59 moved to the dotted line positions indicated in Fig. 5, it will be evident that the line of drive from the turbine to the lay shaft will include an additional set of gears which will operate to reverse the direction of rotation of the lay shaft with respect to the direction of rotation of the turbine shaft.

The above described form of reversing gear is known in the art as is also the means for effecting movement of the eccentric bearings to bring the desired gears into engagement. Further description thereof in this specification is, therefore, not deemed to be necessary. It will be evident that insofar as the present invention is concerned, a separate reversing turbine of the usual character may be employed instead of the reversible gearing hereinbefore described by way of illustration. Since the use of separate reversing turbines, and various ways of effecting disconnectible driving connection between them and the gearing are known in the art, a detailed description of such an arrangement becomes unnecessary.

Turning now to Fig. 8, we have illustrated in longitudinal cross-section a form of turbine which, when employed in a locomotive such as that above described, is adapted to advantageously carry the present invention into effect. This turbine is of the axial flow combined type and comprises a rotor designated generally at 62 mounted on shaft 45. The rotor 62 comprises a wheel portion 63 carrying rows of blading 64 of the impulse or Curtis type. In the specific design illustrated, two rows of impulse blading are carried by this wheel, steam being admitted to the first row of moving blading through the fixed nozzles 65 mounted in the turbine casing. A row of fixed blades 66 serves to guide the steam to the second row of moving blading 64.

As hereinbefore noted, it is advantageous to

use partial admission for controlling turbines of the character in question and the turbine shown in Fig. 8 is constructed for this type of admission. Flow of steam from the pipe 35 to the turbine blading is controlled by a plurality of valves of the type indicated at 67 and 68, each valve controlling a separate inlet passage such as passages 69 and 70, which passages admit steam to independent groups of nozzles 65.

The above described Curtis blading constitutes the high pressure portion of the turbine and the low pressure portion thereof is provided by the drum-like portion 71 of the turbine rotor, which portion is provided with blading of the reaction type. In the form illustrated the drum carries a plurality of rows of moving blades 72 between which rows are situated the usual rows of stationary guide blades 73 fixed in the turbine casing. The first row of blades on the drum portion of the rotor, indicated at 74, may be advantageously of the impulse type, the steam exhausted from blades 64 being admitted to blades 74 through the guide nozzles 75 in the casing.

The above mentioned combined type of turbine, of which type the turbine illustrated is a typical example, is one which is known to be particularly suitable for locomotive propulsion because of the operating characteristics thereof but the specific form of the turbine may be varied in many different ways within the scope of the present invention.

In order to secure the advantages of the invention, it is necessary to so relate the several factors determinative of the blade speed, having reference to the adiabatic heat drop in the turbine, that the sum of the squares of the blade speeds will, at maximum locomotive speed, result in a Parsons figure for the turbine greater than 2800 and preferably, materially greater.

If we now consider the above described embodiment with reference to the manner in which the desired value of the Parsons figure is obtained we find that the following relation of the various factors involved will give the desired results.

Let the diameter of the driving wheels be represented by  $D$ ; the diameter of the lay shaft gear 57 by  $D_1$ ; the diameter of the intermediate shaft gear 56 by  $D_2$ ; the diameter of the intermediate shaft gears 54 by  $D_3$ ; the diameter of the intermediate shaft gears 53 by  $D_4$ ; the diameter of the intermediate shaft gear 50 by  $D_5$ ; the diameter of the high speed pinion 46 by  $D_6$ , and the mean diameter of all of the rows of moving blades in the turbine by  $D_7$  (see Fig. 8).

For the particular locomotive design of the present example, which represents a locomotive suitable for passenger train service, the maximum speed of the locomotive is 112 kilometers per hour and the maximum efficiency is at 65 percent of maximum speed, which is approximately 73 kilometers per hour.

The diameter of the driving wheels is 1700 mm. The diameters  $D_1$ ,  $D_2$ ,  $D_3$ ,  $D_4$ ,  $D_5$  and  $D_6$  are 1000 mm. 465 mm. 1000 mm. 250 mm. 600 mm. and 150 mm., respectively. With this diameter for the driving wheels and with the diameters given above for the several gears, the maximum locomotive speed of 112 km/hr corresponds to a speed of the driving wheels of 350 R. P. M., while the speed of 73 km/hr, at which maximum efficiency is obtained, corresponds to a speed of 227 R. P. M. of the driving wheels. With the values given above for the diameters of the several speed reducing gears, the ratio of the speed of pinion 46 with respect to the speed of gear 57

will be 34.4:1 and this same ratio will represent the ratio of the speed of the turbine shaft 45 with respect to the speed of rotation of the driving wheels, since pinion 46 and shaft 45 rotate 5 at the same speed and the driving wheels 38 rotate at the same speed as that of the lay shaft gear 57.

With the above given dimensions, it will be found that the maximum speed of the turbine is 10 12,000 R. P. M. and that the speed of the turbine for maximum efficiency is 7,800 R. P. M. These speeds are sufficiently high so that resiliency of drive through the gear train is desirable and to this end we prefer to employ the resilient type of tooth on high speed pinion 46 15 in addition to the spring gear 57.

The mean diameter of the turbine blading (D<sub>r</sub>) is 355 mm. and if we take the speed of the turbine at maximum efficiency we find that 20 with this mean diameter, the mean of the blade "u" is 145 meters/second and the mean square of the blade speeds is 21,000 meters<sup>2</sup>/seconds<sup>2</sup>. The number of rows of moving blades, which corresponds to the number of stages in the turbine, is 20, from which it follows that  $\Sigma u^2 = 420,000$  25 meters<sup>2</sup>/seconds<sup>2</sup>.

A fair value for the exhaust pressure of a non-condensing locomotive turbine is 1.2 atmospheres and the adiabatic heat drop of the steam in the turbine when it is admitted at 20 atmospheres 30 pressure and 400° C. pressure and exhausted against a back pressure at 1.2 atmospheres is 150 kilogram calories. Dividing the factor  $\Sigma u^2$  by 150 we find that the Parsons figure is 2800.

It will be noted that the above Parsons figure of 2800 is obtained at the comparatively low locomotive speed of 73 km/hr only by providing a design in which the factor  $\Sigma u^2$  is extremely high. This is provided in the present design by gearing 40 the turbine to the driving wheels through a three stage reduction gearing to obtain a very high rotor speed for the turbine and further by providing a relatively large number of stages in the turbine, as compared with the minimum number of stages 45 required to efficiently extract the energy from the steam if the locomotive were designed solely from the standpoint of obtaining high maximum turbine efficiency. Obviously, the desired high value for  $\Sigma u^2$  may be obtained by designs in which the 50 relation of the several elements determinative of this factor vary widely, since it will be appreciated that the diameters of the driving wheels, the ratio of gear reduction, the mean diameter of the rows of turbine blading and the number of rows of the blading each contribute to the determination of the value of this factor.

Regardless, however, of the manner in which these individual factors of gear diameter, gear ratio and turbine rotor diameter are varied, it will be evident that the advantages of the present invention can be obtained only by the provision of a turbine and gearing arrangement which results in the incorporation in the locomotive of a turbine operating either with extremely high rotor speed or having an abnormally large number of 65 stages for the given steam conditions, or both. We mention only the factors of high rotor speed and large number of stages for obtaining the desired high value of  $\Sigma u^2$  for, while the high value of  $\Sigma u^2$  which we desire can be obtained, theoretically, with high rotor speed and a few stages of large diameter, such an arrangement is impractical because of the inability of present day materials to withstand the resulting stresses imposed by centrifugal force. In order to obtain either the ex-

treme turbine speeds or the abnormally large number of turbine stages which are required if a turbine locomotive embodying the present invention is to be constructed, advantageous factors of cost and peak efficiency possessed by designs not 80 incorporating the invention must be sacrificed, but we have found that in the particular field to which the present invention relates, the advantageous factors which must be sacrificed in structures embodying the present invention are 85 more than compensated for by the advantages to be gained.

As previously noted, the invention is productive of improved results when applied to locomotives operating at moderate pressure. As illustrative of the application of the invention in this field, we give in the following the principal data with respect to a locomotive constructed in accordance with the invention, which locomotive has been built and is at present in successful commercial 95 operation. This locomotive is a freight locomotive of relatively low speed and to enable the operators of the locomotive to compare the performance of the locomotive with their reciprocating engine locomotives, the turbine locomotive 100 was constructed with a boiler designed to operate at the same pressure as that of their reciprocating engine locomotives of the same class. Steam is delivered to the turbine at a pressure of about 10.5 atmospheres and at a temperature of approximately 400° C. The driving wheels of the locomotive are of somewhat smaller diameter than those given in the previously described example, being 1350 mm. in diameter. The total gear reduction between the turbine shaft and the axles 110 of the locomotive is about 50:1, the gearing being substantially identical with that shown in Fig. 6 of the drawings which shows a set of gears giving this ratio of reduction. The turbine is of the same type as that shown in Fig. 8. 115

Because of the relatively low initial steam pressure, we extract only approximately 30 percent of the total adiabatic heat drop in impulse blading, and for maximum turbine efficiency, which occurs at a turbine speed of 8400 R. P. M. and a locomotive speed of 42 kilometers per hour, the Parsons figure is 3400, which approximates the strictly theoretical value of 3200 for the Parsons figure for a turbine having 30 percent impulse blading. The maximum turbine speed of 14000 R. P. M. corresponds 125 to a locomotive speed of 69 km/hr and at this speed the Parsons figure of the turbine has a value of 9400.

It is to be noted that in the construction of this locomotive, the maximum efficiency is obtained at about 57 percent of maximum speed. Ordinarily a freight locomotive will operate at relatively low speeds, compared to its maximum speed, for a greater percentage of its operating time than will a passenger locomotive and it is therefore desirable to increase the Parsons figure of the turbine at maximum speed to a value resulting in maximum efficiency of operation at a speed very considerably below maximum locomotive speed. 130

In this turbine, as well as in the turbine previously described for operation at 20 atmospheres initial steam pressure, we provide impulse blading which operates at maximum efficiency at a lower locomotive speed than that at which the reaction blading operates at maximum efficiency, and in the locomotive we have constructed, the impulse blading reaches maximum efficiency at a locomotive speed of about 35 kilometers per hour while the reaction blading operates at 135

highest efficiency when the locomotive speed is 47 kilometers per hour.

In the preferred embodiment of our invention, the locomotive speed at which the reaction blading reaches highest efficiency, as well as the speed at which the highest efficiency of the turbine as a whole is reached, should be below the maximum speed of the locomotive, so that the largest possible part of the relatively flat efficiency curve produced by having the impulse blading and the reaction blading reach peak efficiency at different locomotive speeds, is brought within the normal operating speed range of the locomotive. Inasmuch as the theoretical value of the Parsons figure for reaction blading at highest efficiency is about 3800, it follows that the preferred embodiment of our invention comprises an arrangement in which the turbine at maximum locomotive speed has a Parsons figure above 3800. Minor variations from the purely theoretical results should be allowed for, but we find that if the value of the Parsons figure at maximum locomotive speed is made at least 4000, the desired operating characteristics of the preferred form will be achieved.

While in the foregoing description we have confined our consideration to locomotives in which the turbine comprises a single unit, it will be understood that the invention is not confined to this specific arrangement but may equally well be embodied in locomotives in which two or more separate turbine rotors are employed in expanding the motive fluid down from boiler pressure to final exhaust pressure. In cases where a plurality of separate turbine rotors are employed, operating in series with respect to motive fluid, the several turbine rotors are to be considered as turbine sections and the value of the Parsons figure is to be considered with respect to the turbine as a whole, that is, all of the turbine sections taken together.

What we claim is:

1. A turbine driven locomotive comprising driving wheels, a turbine having rotatable blade carrying means, blades on said means, reduction gearing for causing said means to drive said wheels, the ratio of speed reduction of said gearing being so related to the diameter of said driving wheels that the speed of rotation of said means at which maximum turbine efficiency is produced corresponds to a locomotive speed less than the maximum speed thereof, and the diameter of said means, the number of rows of moving blades, and the speed of rotation of said means as determined by the diameter of said driving wheels and the ratio of said gearing being so related to the difference in heat content of the motive fluid as admitted to and as exhausted from the turbine that at speeds within normal operation the ratio between the sum of the squares of the values of the blade speeds, expressed in meters per second, and the adiabatic heat drop of the motive fluid in the turbine, expressed in kilogram calories, is greater than 2800:1.

2. A turbine driven locomotive comprising driving wheels, a turbine having rotatable blade carrying means, rows of blades on said means, reduction gearing for causing said means to drive said wheels, the ratio of speed reduction of said gearing being so related to the diameter of said driving wheels that the speed of rotation of said means at which maximum turbine efficiency is produced corresponds to a locomotive speed between 25 and 75 percent of the maxi-

um speed thereof, and the diameter of said means, the number of rows of moving blades, and the speed of rotation of said means as determined by the diameter of said driving wheels and the ratio of said gearing being so related to the difference in heat content of the motive fluid as admitted to and as exhausted from the turbine that the ratio between the sum of the squares of the values of the blade speeds at maximum speed, expressed in meters per second, and the adiabatic heat drop of the motive fluid in the turbine, expressed in kilogram calories, is greater than 2800:1.

3. A turbine driven locomotive comprising driving wheels, a turbine having rotatable blade carrying means, rows of blades on said means, reduction gearing for causing said means to drive said wheels, the ratio of speed reduction of said gearing being so related to the diameter of said driving wheels that the speed of rotation of said means at which maximum turbine efficiency is produced corresponds to a locomotive speed less than the maximum speed thereof, and the diameter of said means, the number of rows of moving blades, and the speed of rotation of said means as determined by the diameter of said driving wheels and the ratio of said gearing being so related to the difference in heat content of the motive fluid as admitted to and as exhausted from the turbine that the ratio between the sum of the squares of the values of the blade speeds at maximum speed, expressed in meters per second, and the adiabatic heat drop of the motive fluid in the turbine, expressed in kilogram calories, is greater than 2800:1, said sum of the squares of the blade speeds being greater than 600,000 meters<sup>2</sup>/second<sup>2</sup>.

4. A turbine driven locomotive comprising a boiler, a superheater, driving wheels, a non-condensing turbine having rotatable blade carrying means, blades on said means, reduction gearing for causing said means to drive said wheels, the ratio of speed reduction of said gearing being so related to the diameter of said driving wheels that the speed of rotation of said means at which maximum turbine efficiency is produced corresponds to a locomotive speed less than the maximum speed thereof, and the diameter of said means, the number of blades and the speed of rotation of said means as determined by the diameter of said driving wheels and the ratio of said gearing being so related to the difference in heat content of the motive fluid as admitted to and as exhausted from the turbine that the ratio between the sum of the squares of the values of the blade speeds at maximum speed, expressed in meters per second, and the adiabatic heat drop of the motive fluid in the turbine, expressed in kilogram calories, is greater than 2800:1, said boiler and said superheater supplying steam to said turbine at a temperature of at least 400° C.

5. A turbine driven locomotive comprising a boiler, a superheater, driving wheels, a non-condensing turbine having rotatable blade carrying means, blades on said means, reduction gearing for causing said means to drive said wheels, the ratio of speed reduction of said gearing being so related to the diameter of said driving wheels that the speed of rotation of said means at which maximum turbine efficiency is produced corresponds to a locomotive speed less than the maximum speed thereof, and the diameter of said means, the number of rows of moving blades, and the speed of rotation of said

means as determined by the diameter of said driving wheels and the ratio of said gearing being so related to the difference in heat content of the motive fluid as admitted to and as exhausted from the turbine that the ratio between the sum of the squares of the values of the blade speeds at maximum speed, expressed in meters per second, and the adiabatic heat drop of the motive fluid in the turbine, expressed in kilogram calories is greater than 280:1, said boiler and superheater supplying steam to said turbine at a pressure of at least 20 atmospheres and a temperature of at least 400° C.

6. A turbine driven locomotive comprising a boiler, driving wheels, a turbine having rotatable blade-carrying means, rows of blades on said means comprising impulse blades and reaction blades arranged in the order named with respect to steam flow through the turbine, the amount of impulse blading with respect to the amount of reaction blading being such that the adiabatic heat drop in the impulse blading does not exceed one half of the total available adiabatic heat drop, and reduction gearing for causing said means to drive said wheels, the ratio of speed reduction of said gearing with respect to the impulse blading as compared with the reaction blading and the diameters and numbers of rows of the different types of blading being related so that the speed of the locomotive at which the impulse blading operates at maximum efficiency is different from the speed of the locomotive at which the reaction blading operates at maximum efficiency.

7. A turbine driven locomotive comprising a boiler, a superheater, driving wheels, a non-condensing turbine having rotatable blade-carrying means, rows of blades on said means comprising impulse blades and reaction blades arranged in the order named with respect to steam flow through the turbine, means for supplying superheated steam to the impulse blading from the superheater, means for exhausting steam from the reaction blading to the atmosphere, the amount of impulse blading with respect to the amount of reaction blading being such that the adiabatic heat drop in the impulse blading does not exceed one half of the total available adiabatic

ic heat drop, and reduction gearing for causing said means to drive said wheels, the ratio of speed reduction of said gearing with respect to the impulse blading as compared with the reaction blading and the diameters and numbers of rows of the different types of blading being related so that the speed of the locomotive at which the impulse blading operates at maximum efficiency is lower than the speed of the locomotive at which the reaction blading operates at maximum efficiency.

8. A turbine driven locomotive comprising a boiler, a superheater, driving wheels, a non-condensing turbine having rotatable blade carrying means, rows of blades on said means comprising impulse blades and reaction blades arranged in the order named with respect to steam flow through the turbine, means for supplying superheated steam to the impulse blading from the superheater, means for exhausting steam from the reaction blading to the atmosphere, the amount of impulse blading with respect to the amount of reaction blading being such that the adiabatic heat drop in the impulse blading does not exceed one half of the total available adiabatic heat drop, and reduction gearing for causing said means to drive said wheels, the ratio of speed reduction of said gearing with respect to the impulse blading as compared with the reaction blading and the diameters and numbers of rows of the different types of blading being related so that the speed of the locomotive at which the impulse blading operates at maximum efficiency is lower than the speed of the locomotive at which the reaction blading operates at maximum efficiency and the diameters of the driving wheels, the ratio of speed reduction of the gearing and the diameters and numbers of rows of blades of the whole turbine being related to the total available adiabatic heat drop so that at maximum speed of operation of the locomotive, the Parsons figure for the whole turbine is at least 4000, expressed in metric units.

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### CERTIFICATE OF CORRECTION.

Patent No. 1,924,062.

August 22, 1933.

ALF LYSHOLM, ET AL.

It is hereby certified that error appears in the printed specification of the above numbered patent requiring correction as follows: Page 4, lines 36 and 37, strike out the words "as compared with the efficiency incident to the use of this blade"; page 5, line 25, for "It" read "it"; page 9, line 10, claim 5, for "280:1" read "2800:1"; and that the said Letters Patent should be read with these corrections there in that the same may conform to the record of the case in the Patent Office.

Signed and sealed this 26th day of September, A. D. 1933.

F. M. Hopkins

Acting Commissioner of Patents.

(Seal)