May 16, 1967 A. H. BELL III 3,319,931

TURBINE ENGINE

Original Filed July 25, 1963 4 Sheets-Sheet 1

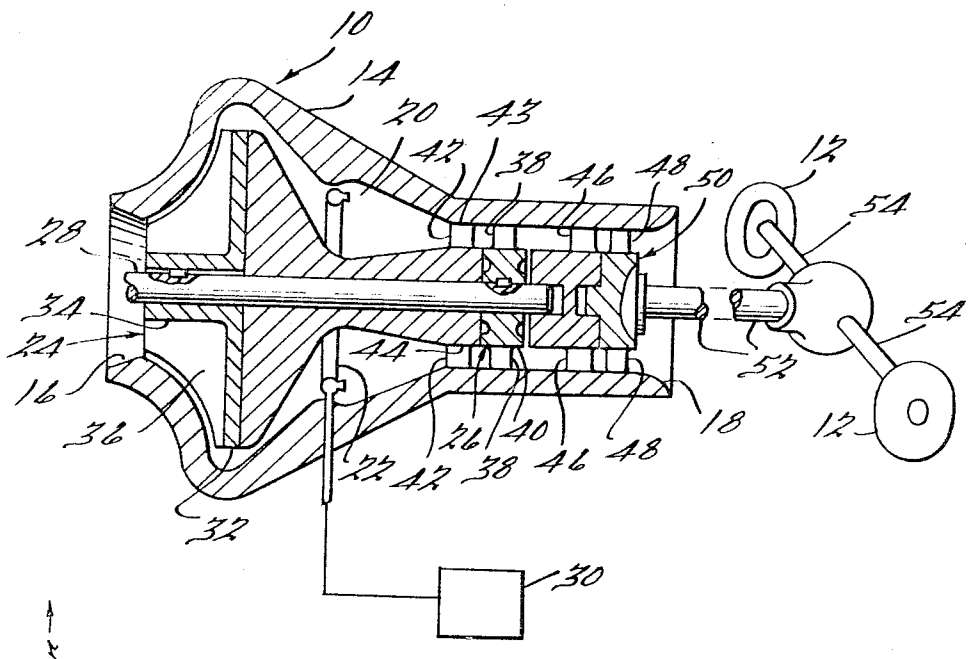


FIG. 1.

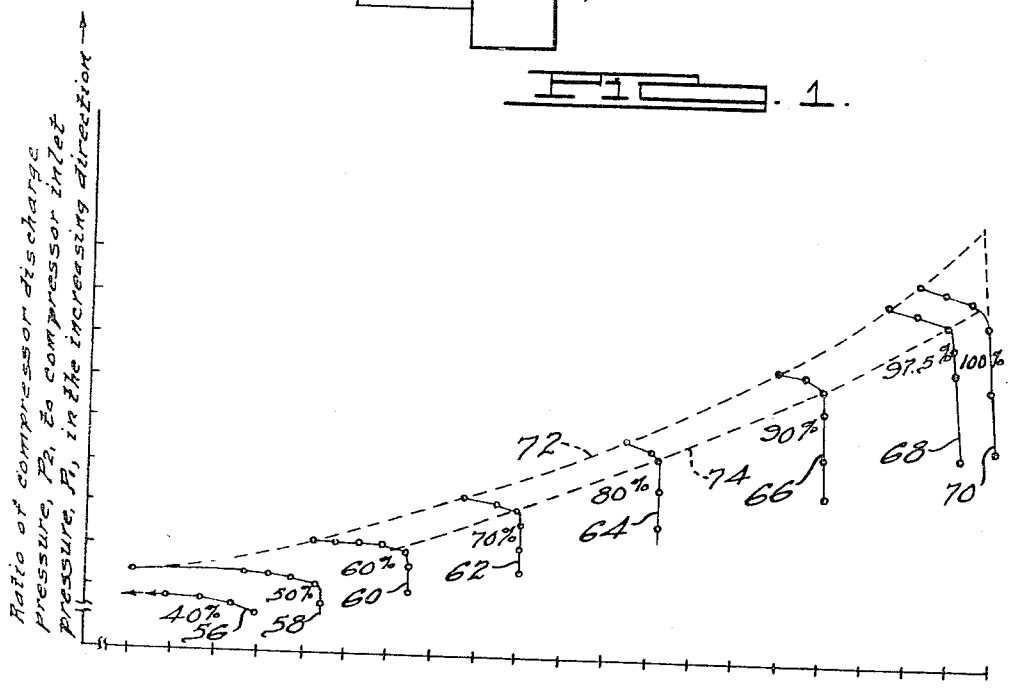


FIG. 2.

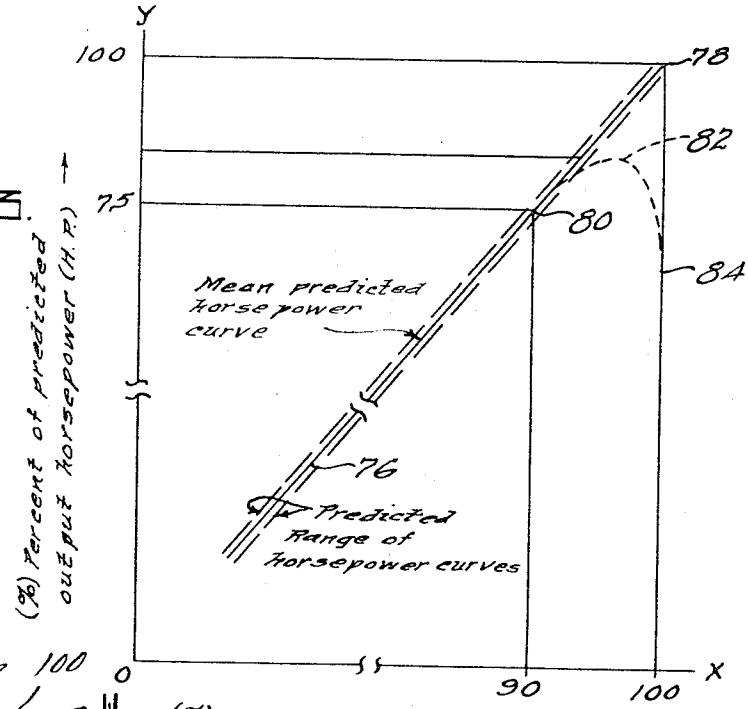
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FIG. 3.



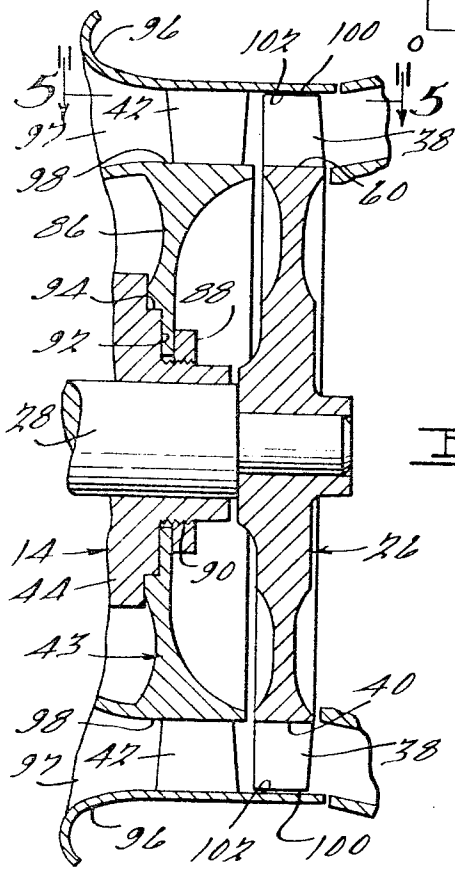


FIG. 4.

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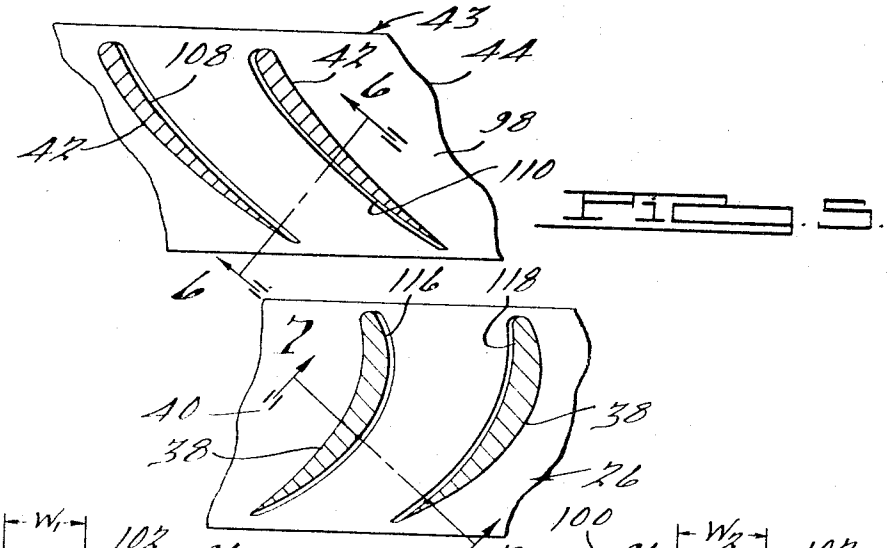


Fig. 5.

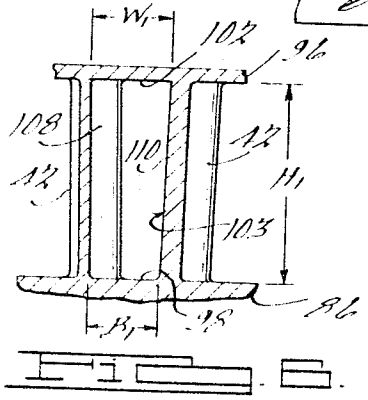


Fig. 6.

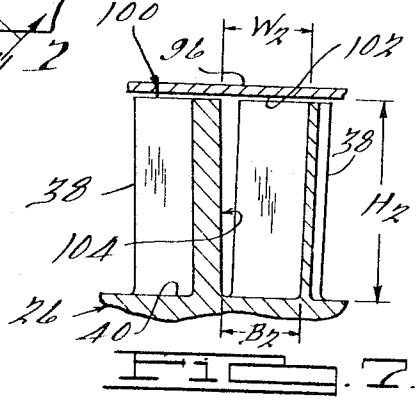


Fig. 7.

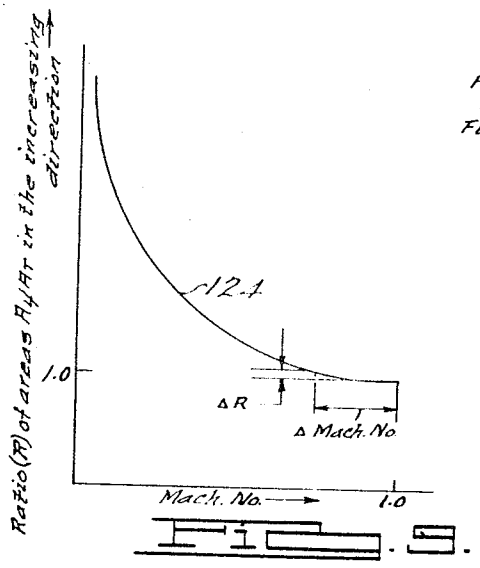


Fig. 9.

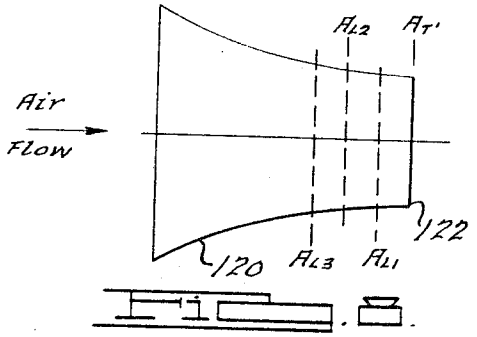


Fig. 8.

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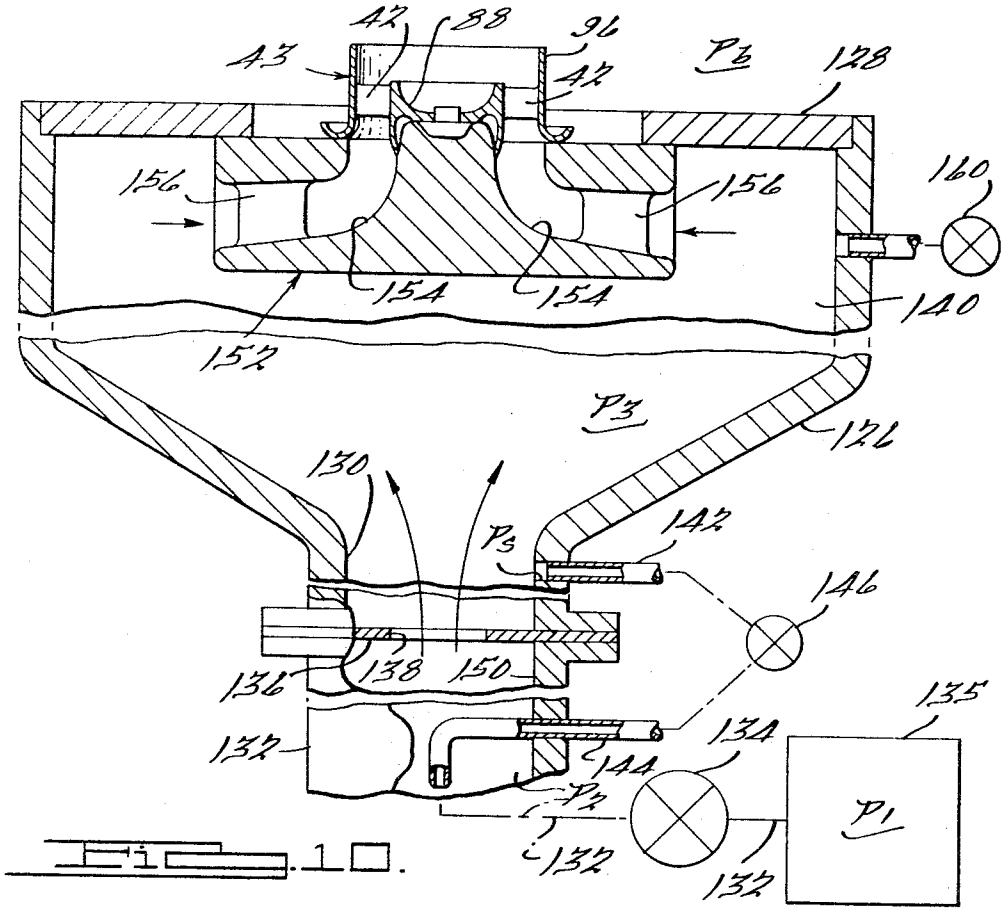


Fig. 10.

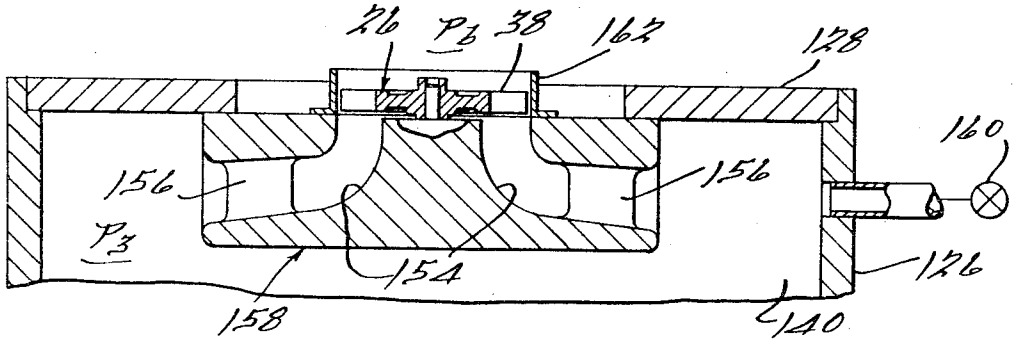


Fig. 11.

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3,319,931

TURBINE ENGINE

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Original application July 25, 1963, Ser. No. 297,538, now Patent No. 3,252,212. Divided and this application Dec. 23, 1965, Ser. No. 516,047
1 Claim. (Cl. 253—78)

This invention relates generally to turbine engines and more particularly to the fixed nozzle vanes and wheel blades employed therein.

This application is a division of my copending application Ser. No. 297,538 filed on July 25, 1963 for Turbine Engine, now Patent No. 3,252,212.

As well known in the art, turbine engines are comprised of a gas generator section consisting primarily of an air compressor, burner portion and a compressor drive turbine wheel which obtains its energy from the gases flowing out of the burner portion and drives the compressor. Some turbine engines, depending on their intended use, also include a power turbine which when placed in the path of the flowing gases, downstream of the compressor drive turbine wheel, serves to provide additional work such as by driving a propeller in a turbojet engine or driving some other output shaft for land or water-based vehicles.

The ultimate output power developed by any turbine engine is primarily dependent on the output of the gas generator section which, in turn, is a function of the compressor speed of that particular gas generator.

In many instances it has been found that substantial variations in output horsepower exist as between any two turbine engines even though the turbine engines are of the same design and rated output power. Such variations, which in some cases have been in the magnitude of thirty percent (30%) power loss, exhibit themselves to the greatest extent in the range of compressor speeds of seventy-five percent (75%) to a hundred percent (100%) of the designed or rated maximum compressor speed. Further, these variations occur most frequently in turbine engines wherein the gas flow through the fixed guide vanes or nozzles and compressor drive turbine wheel blades closely approaches sonic velocity as the compressor speed approaches designed maximum speed.

As a consequence of such power variations, it becomes impossible to predict, with any reasonable degree of accuracy, the expected power output of any particular turbine engine. Further, since such variations have been found to exist between engines of identical design, it must be concluded that the engine delivering the lesser power is, at least to some degree, of lesser efficiency and therefore undesirable.

Now it has been discovered that if certain relationships between the elements comprising the turbine engine are considered as being critical and such relationships are established during the process of manufacturing the turbine engine, that the resulting engines will produce a reasonably predictable output horsepower which is consistently at a relatively higher and more efficient value.

Accordingly, a general object of this invention is to provide, in a turbine engine, fixed guide vanes or nozzles and cooperating compressor drive turbine wheel blades having a relationship established therebetween which enables the turbine engine to exhibit an overall higher efficiency.

Another object of this invention is to provide in a turbine engine, fixed guide vanes or nozzles and cooperating compressor drive turbine wheel blades having a relationship therebetween which enables the turbine engine to achieve a predicted output horsepower which is within a reasonable range of tolerances.

Still another object of this invention is to provide a method for achieving a particular critical relationship between fixed guide vanes or nozzles and cooperating compressor drive turbine wheel blades of a turbine engine so as to obtain a turbine engine having a predictable horsepower output.

Other objects and advantages of the invention will become apparent when reference is made to the following description and drawings wherein:

FIGURE 1 is a cross-sectional view of a gas turbine engine adapted to propel driving wheels of a land based vehicle;

FIGURE 2 is a graph illustrating typical compressor curves obtained by plotting the ratio of compressor discharge pressure to compressor inlet pressure against the weight-rate of air flow through the compressor;

FIGURE 3 is a graph illustrating a typical output horsepower curve of a turbine engine;

FIGURE 4 is an enlarged fragmentary portion of the turbine engine of FIGURE 1;

FIGURE 5 is a cross-sectional view taken generally on the plane of line 5—5 of FIGURE 4, looking in the direction of the arrows;

FIGURE 6 is a cross-sectional view taken generally on the plane of line 6—6 of FIGURE 5, looking in the direction of the arrows;

FIGURE 7 is a cross-sectional view taken generally on the plane of line 7—7 of FIGURE 5;

FIGURE 8 illustrates a generally tubular conduit of converging cross-sectional area;

FIGURE 9 is a graph illustrating the relationship between the change in velocity of flow through the conduit of FIGURE 8 for a corresponding change in cross-sectional area; and

FIGURES 10 and 11 illustrate, in cross-section, an arrangement for testing nozzle assemblies and turbine wheels, respectively, in accordance with the teachings of this invention.

Referring now in greater detail to the drawings, FIGURE 1 illustrates a turbine engine **10** adapted for propelling a land-based vehicle having driving wheels **12**. The engine **10** is comprised of a housing **14** having an air intake **16** and exhaust orifice **18**. A combustion chamber **20**, having any suitable fuel distribution means such as fuel distribution ring **22** therein, is located within the housing **14** between the compressor **24** and compressor drive turbine wheel **26**. Compressor **24** and compressor drive turbine wheel **26** are connected to each other by means of a shaft **28** which may also be used for driving or in other ways operating selected engine accessories such as the fuel control schematically illustrated at **30**.

Compressor **24**, as illustrated, is of the radial flow type and is comprised of a generally circular body portion **32** and cylindrical bearing portion **34** with a plurality of generally radially directed compressor vanes **36** therebetween.

Compressor drive turbine wheel **26** is provided with a plurality of circumferentially spaced blades **38** extending radially outwardly from the wheel rim **40**. Upstream of turbine wheel **26**, the engine **10** is provided with a plurality of radially directed guide vanes or nozzles **42**, sometimes referred to as stators, spaced circumferentially about the body portion **44**.

As compressor **24** is rotated, air is drawn in through inlet **16**, compressed by vanes **36** and directed into the combustion chamber **20** where fuel, supplied in accordance with a predetermined schedule, is burned so as to heat the gases therein. The gases flow from the burner chamber **20** through nozzles **42** which accelerate and cause the gases to impinge upon the compressor drive turbine wheel blades **38** in a manner causing rotative motion of turbine wheel **26** which, in turn, drives compressor **24**.

The gases, continuing to flow towards exhaust orifice **18**, pass between a second set of guide vanes or nozzles **46**, similar to nozzles **42**, which direct the gases against blades **48** of an output power turbine wheel **50**. Turbine wheel **50** may be connected, as by any suitable power transmission means **52**, **54** to the ground-engaging driving wheels **12**.

FIGURE 2 is a graph illustrating typical compressor curves **56**, **58**, **60**, **62**, **64**, **66**, **68** and **70** obtained by plotting the ratio of compressor discharge pressure, P_2, to inlet pressure, P_1, against the air flow through the compressor in pounds per second. Each of the curves can be obtained by maintaining the compressor at selected constant speeds while varying the pressure ratio P_2/P_1. The respective compressor speeds are indicated, for illustrative purposes, near each curve in terms of percentages of designed maximum compressor speed.

The area above and to the left of dash line **72** represents that area in which compressor instability or surge is encountered. Compressor instability or surge refers to a condition sometimes referred to as "hunting"; that is, those portions of the compressor curve which tend to flatten out and become more horizontal indicate that slight variations in the ratio of P_2/P_1 result in comparatively large changes in the air flow which give rise to compressor instability. Curve **72** is determined generally by connecting together the respective points, on the individual compressor curves, at which compressor surge is first encountered.

In addition to the compressor surge area, as discussed above, a further important consideration remains. That is, the rapid reduction of the numerical value of the ratio of P_2/P_1 as illustrated by the generally vertically depending portions of each of the compressor curves **56** through **70**. Since the compressor inlet pressure, P_1, may be assumed to be constant, the conclusion must be that the downstream or compressor discharge pressure, P_2, decreases and the rate of P_2 decrease, as illustrated by the slope of the depending portions of the compressor curves, is very rapid as compared to an increment of change in the air flow. The reduction in compressor discharge pressure, P_2, is caused by the choking effect of the air within the compressor. Consequently, a line **74** drawn generally through the points of each of the compressor curves wherein the rate of change of the ratio P_2/P_1 starts to become rapid defines an area, generally below and to the right of line **74**, which might be referred to as the compressor choke area.

Accordingly, with turbine engines, precautions are taken to assure the operation of the compressor within the limits defined by the surge or stall line **72** and choke line **74**. In order to achieve this, it has been the practice, generally, to aerodynamically match the compressor drive turbine wheel and its associated nozzle assembly, as a subassembly, to the compressor. This is sometimes referred to as gross matching. In some instances other limits similar to the choke line **74** and the stall line **72** are determined and employed for aerodynamic reasons. (The aircraft industry has also employed additional means for avoiding at least portions of the compressor stall or surge areas by providing compressor bleed valves which at times and in accordance with selected operating parameters, vent some of the compressor air to the atmosphere.) A turbine engine having such a gross-matched compressor, compressor drive turbine wheel and nozzle assembly, so as to operate between the stall line **72** and choke line **74**, should exhibit a characteristic horsepower curve **76**, obtained generally by plotting output horsepower and the percent of designed maximum compressor speed along a logarithmic Y-axis and an arithmetic X-axis, respectively, of the graph of FIGURE 3.

However, it has been found that substantial unpredictable variations in output horsepower still exist between

gas turbine engines which have been constructed in accordance with the prior art.

For example, with reference to FIGURE 3, it has been found that while one turbine engine might deliver full rated power, another engine of the same design and rating might well develop substantially less horsepower than that predicted. That is, a particular engine might deviate from the predicted mean curve **76**, as at some point **80**, and after reaching a maximum output as at point **82** (substantially lower than the rated value represented by point **78**) steadily decrease to some point **84** which may represent a developed horsepower less than that of point **80**.

Further, it has been found that this phenomenon of horsepower variation, or lost horsepower, will at times occur in the same gas turbine engine which had originally developed the full rated horsepower. As was previously stated, the phenomenon of unpredictable horsepower variation occurs most frequently in gas turbine engines wherein the velocity of gas flow through the compressor drive turbine wheel and the compressor turbine nozzle assembly is in the transonic range as the compressor approaches its designed maximum speed.

In the past, various proposals have been tried in unsuccessful attempts to overcome the problem of horsepower variation. For example, the redesigning, modification and/or inspection of such areas as lubrication systems, gear trains, horsepower demands of engine driven accessories, alignment of turbine components and engine air leakages in no way alleviated the problem of horsepower variation.

Therefore, assuming point **84** to be the lowest horsepower ever achieved, it can be seen that turbine engines, constructed in accordance with the prior art and having the compressor drive turbine wheel and nozzle assembly grossly matched, as a subassembly, to the compressor, can be expected to produce an output horsepower value ranging anywhere between the limits defined by points **78** and **84**. In view of this, it becomes evident that the gross aerodynamic matching of the compressor drive turbine wheel and its nozzle assembly, as a subassembly, to the compressor is in itself insufficient for assuring reasonable attainment of predicted horsepower and that still another influencing factor, heretofore not apparent to those skilled in the art, remains as an important consideration in the design and construction of the turbine engine.

It has been discovered that such an influencing factor is the relationship between the compressor drive turbine wheel blades **38** and the nozzle or stator vanes **42**. More specifically it is the relationship of the total area available for gas flow between the nozzle vanes **42** as compared to the total area available for gas flow between the compressor drive turbine wheel blades **38**.

FIGURE 4 illustrates in greater detail the arrangement and construction of the nozzle assembly **43** and compressor drive turbine wheel **26**. Preferably, the nozzle or stator vanes **42** are made integrally with a main body portion **86** which is secured to a portion **44** of the general engine housing **14**. This may be accomplished as by means of a nut **88** internally threaded so as to cooperate with a threaded portion **90** in axially urging the stator body **86** against the radial surface **92** of a mounting shoulder **94**. Radially outwardly, an annular shroud **96** is formed preferably integrally with the nozzle vanes **42**. The shroud **96**, in cooperation with the outer surface **98** of the stator body **86**, defines an annular passage **97** in which the vanes **42** are located for the directional control of the gases flowing therethrough.

The compressor drive turbine wheel **26**, secured to shaft **28** for rotation therewith, has its blades **38** generally within the confines of shroud **96**. The clearance between the outermost ends **100** of moving compressor drive turbine wheel blades **38** and the inner stationary surface **102** of shroud **96** is kept to a minimum. The outer

surface **40** of compressor drive turbine wheel **26** and the inner surface **102** of shroud **96** also define an annular passage, generally coaxial with passage **97**, for the flow of gases therethrough.

The flow area as between any two adjacent nozzle vanes **42** will be the smallest cross-sectional area between such adjacent blades. For example, assuming that a plane indicated by line **6—6** of FIGURE 5 were passed through vanes **42** so as to have the cross-sectional area between the vanes a minimum area, the flow area **103** therebetween could be determined, generally, from the dimensions H_1, W_1 and B_1 as illustrated in FIGURE 6. Similarly, the flow area of the turbine blades **38** could be determined by passing a plane indicated by line **7—7** of FIGURE 5 through points resulting in a minimum area **104**. The flow area between such turbine blades would then be determined, generally, by the dimensions H_2, W_2, and B_2 as illustrated in FIGURE 7. (Because of the relatively small clearances existing between the ends **100** of turbine blades **38** and the inner surface **102** of stationary shroud **96**, the height, H_2, in FIGURE 7 can be considered as the distance from surface **40** of compressor drive turbine wheel **26** to the inner surface **102** of the shroud **96**.)

As previously stated, the gross matching of the nozzle assembly **43** and compressor drive turbine wheel, as a sub-assembly, to the compressor, even though well known in the art, only assures the engine of operation within the limits defined by the stall line **72** and the choke line **74** of FIGURE 2. Such gross matching does not, however, give any assurance that the engine will attain the maximum horsepower as represented by point **78** of FIGURE 3, but rather that the horsepower attained will be somewhere between the desired point **78** and the low limit of point **84**.

It has been discovered that what is still further required, in addition to the gross matching with reference to the compressor, is a critical or selective matching of the nozzle assembly **43** to the compressor drive turbine wheel assembly **26** based on the respectitve effective flow areas of each.

Referring to FIGURES 4, 5 and 6 it can be seen that if plane **6—6** determines the smallest cross-sectional area between two adjacent nozzle vanes **42** then the cross-sectional areas taken at local points upstream of the plane **6—6** will define progressively large areas generally as the distance between the local area under construction and the minimum area **103** increases. In other words, the progressively larger areas can be considered as defining, generally, the converging conduit **120** of FIGURE 8 wherein the throat area, A_I, would be a dimensional equivalent of the minimum area **103** defined by W_1, B_1 and H_1 of FIGURE 6 and the various local areas, R_{L1}, A_{L2}, A_{L3}, etc., would constitute the increasing cross-sectional areas upstream of the minimum area **103**. Further, just as the throat area, A_T, will determine the maximum volume rate of air flow through conduit **120**, so will the minimum area **103** between nozzle vanes **42** determine the maximum volume rate of gas flow therethrough for any given condition. Generally, it can be said that the absolute maximum rates in both arrangements will occur when the velocity of air flow through the most restricted areas is sonic.

Assuming that conduit **120** is under a condition of sonic flow, that is, the velocity of air flow through throat **122** is Mach-one, a relationship can be established between the respective local areas and the velocity of flow through such areas. For example, an exponential curve, **124** illustrated in FIGURE 9 can be obtained by graphically plotting the ratio of the local cross-sectional areas, A_L, to the throat area, A_T, against the velocity of flow at that particular local area.

From an inspection of curve **124**, it can be seen that when the ratio of areas, A_L/A_T, is 1.000 then the velocity of the flow at the local area is equal to Mach-One. It should also be observed, however, that the change in local air velocity, Δ Mach No., is large as compared to a relatively small change in the local cross-sectional area, as reflected by ΔR, in that range where the ratio of areas, A_L/A_T, approaches 1.000.

Even though curve **124** of FIGURE 9 has been discussed primarily with reference to the converging conduit **120** of FIGURE 8, the principles involved apply equally well to the nozzle assembly **43**. That is, as previously stated, the minimum area **103** determined by W_1, B_1 and H_1 is the equivalent of the throat area, A_T, of throat **122**, and the various local cross-sectional areas upstream of the minimum nozzle area **103** are the equivalents of the local cross-sectional areas A_{L1}, A_{L2}, A_{L3}, etc.

Accordingly, it becomes evident that if the size of the minimum nozzle area **103** is calculated to transmit gas therethrough at sonic velocities and at a specific weight-rate of flow, that very slight deviations from the calculated size of the minimum nozzle area as illustrated by FIGURE 9 will result in relatively large reductions in the velocity of air or gas flow through the minimum nozzle area **103**. The immediate effect of such velocity losses is a drastic reduction in the energy available for driving the compressor drive turbine wheel **26** and ultimately the power turbine **50**.

It has been discovered that in some instances turbine engines, having a nozzle assembly wherein the blade surfaces **108** and **110** (FIGURE 5) were .003 inch further away from each other than the ideal calculated dimension, experienced about a ten percent (10%) power loss. It has also been found that such turbine engines could be made to produce their full rated horsepower if the nozzle assembly and compressor drive turbine wheel **26** in the engine were replaced by a nozzle assembly and compressor drive turbine wheel which were selectively matched to each other.

For example, let it be assumed that it has been empirically determined that a particular engine requires the effective flow area of the nozzle assembly and the effective flow area of the compressor drive turbine wheel to be in an ideal ratio of 1.000 to 1.500, respectively, as determined at a particular pressure differential across each. (The effective or equivalent flow areas are determined by total flow and velocity experienced through the entire nozzle assembly 43, that is the summation of the effective flow areas between the individual nozzle vanes, or the entire compressor drive turbine wheel, as the case may be.)

It is well known that dimensional tolerances are required in any manufacturing operation. Therefore, it becomes impossible to randomly choose any one nozzle assembly out of many and similarly choose a compressor drive turbine wheel and have any degree of assurance that the two will have their respective equivalent flow areas in precisely the assumed ideal ratio of 1.000; 1.500. Therefore, the very tolerances necessary for the production of the nozzle assembly and compressor drive turbine wheel cause the respective equivalent flow areas to vary to the degree sufficient to cause flow velocity losses as described with reference to FIGURES 4–9. Such variations resulting from the manufacturing tolerances also exhibit themselves in causing the respective equivalent flow areas to vary from the assumed ideal ratio of 1.000; 1.500.

However, the discovery of the cause of the problem of variations in and loss of engine power, does not in and of itself suggest a necessarily practical solution of that problem because dimensional manufacturing tolerances cannot be eliminated nor can they be reduced to a degree which results in prohibitive manufacturing costs.

It has been discovered that the solution of the problem resides in the selective matching of a nozzle assembly to a cooperating compressor drive turbine wheel by means of matching the flow performance of each of them.

For purposes of illustration, let the following be assumed to be the ideal constants:

(1) ratio of the total nozzle assembly flow area to the total compressor drive turbine wheel flow area= 1.000/1.500;
(2) total nozzle assembly flow area=10.000 sq. inches;
(3) total compressor drive turbine wheel flow area= 15.000 sq. inches;
(4) velocity of gas flow through nozzle assembly at full rated engine power=Mach 0.94;
(5) velocity of gas flow through compressor drive turbine wheel at full rated engine power=Mach one.

Further, let it be assumed that because of the small dimensional differences obtained due to reasonable manufacturing tolerances, that relatively large changes occur (as previously discussed with reference to FIGURES 8 and 9) in the equivalent flow areas of both the nozzle assembly and the compressor turbine, so as to result in the following:

(6) the effective or equivalent total nozzle assembly flow area varies, as between assemblies, from 9.700 to 10.300 sq. inches and
(7) the effective or equivalent total compressor drive turbine flow area varies, as between turbines, from 14.500 to 15.500 sq. inches.

With the above assumptions it can be seen that a nozzle assembly and compressor drive turbine wheel randomly selected from a large group of each could yield the following equivalent flow area combinations having the corresponding indicated equivalent flow area ratios, which would still satisfy the gross matching requirements as between the compressor and the nozzle assembly and compressor drive turbine wheel as a subassembly:

Combination	Equivalent Flow Area of—		Actual Area Ratio	Ideal Area Ratio
	Nozzle Assy.	Comp. Turbine		
I	9.700	14.500	1/1.495	1/1.500
II	9.700	15.500	1/1.598	1/1.500
III	10.300	14.500	1/1.408	1/1.500
IV	10.300	15.500	1/1.505	1/1.500

Combination	Actual Area Ratio	Numerical Equivalent of—		Percent Variation of Actual Area Ratio from Ideal Area Ratio
		Actual Area Ratio	Ideal Area Ratio	
I	1/1.495	0.6689	0.6667	0.33
II	1/1.598	0.6258	0.6667	6.13
III	1/1.408	0.7102	0.6667	6.52
IV	1/1.505	0.6645	0.6667	0.33

Combinations I and IV result in equivalent flow areas which are very near the assumed ideal area ratios.

It has been discovered that, in gas turbine engines wherein the velocity of motive gas flow through the nozzle assembly or through the compressor drive turbine wheel is in the range of Mach 0.9 to Mach 1.0 as the compressor approaches its designed maximum speed, the numerical equivalent of the actual area ratio may vary in the range of plus or minus 1.0% from the numerical equivalent of the ideal area ratio. Therefore, combinations I and IV would result in the engine developing its full rated horsepower at point 78 of the horsepower curve of FIGURE 3.

However, the same nozzle assemblies combined in reverse order with the compressor drive turbine wheels, as illustrated by combinations II and III, result in equivalent flow area ratios having a substantial deviation from the assumed ideal equivalent flow area ratio. Combinations II and III, even though satisfying the gross matching requirements of the compressor, would, nevertheless, cause the engine to produce an output horsepower much less than the rated output of the engine such as represented by point 84 of the curve of FIGURE 3.

Accordingly, in order to insure proper engine performance so as to consistently achieve a reasonably predictable horsepower output it becomes necessary to selectively match the nozzle assembly and compressor drive turbine wheel to each other.

A method of selectively or critically matching the nozzle assembly and compressor drive turbine wheel is disclosed with reference to the apparatus of FIGURE 10 which is comprised of an outer housing **126** provided with a cover **128** at one end thereof and an inlet passage **130** at the other end. Conduit **132**, having a valve **134** serially connected therewith, communicates at its one end with a source **135** of pressurized air while its other end serves to retain an orifice plate **136** in proper position so as to maintain the orifice **138** within the path of the air flowing from conduit **132** and into the plenum chamber **140** defined generally by housing **126** and cover **128**.

Conduits **142** and **144** are provided so as to communicate the pressures on opposite sides of the orifice plate **136** to a differential pressure gage **146**. Conduits **142**, **144**, orifice **138** and pressure gage **146** provide means for determining the mass rate of flow through the orifice **138** and therefore the entire system. The mass rate of flow, W, can be determined by the application of the following general compressible mass flow equation:

$$G=\frac{W}{CA}=P_0\sqrt{\frac{\overline{m}}{T_0}}\cdot\sqrt{\frac{2g}{\overline{R}}\cdot\frac{n}{n-1}}\cdot\frac{1}{r^n}\sqrt{1-\frac{n-1}{r^n}}$$

where:

(1) G=flow per unit of area;
(2) W=mass rate of flow;
(3) A=cross-sectional flow area of orifice **138**;
(4) C=the discharge coefficient of orifice **138** (a calibrated function of the Reynold's number as well known in the art);
(5) $P_0=P_2$=total pressure upstream of orifice **138**;
(6) $\overline{m}$=molecular weight of the fluid flowing;
(7) T_0=total temperature upstream of orifice **138**;
(8) g=acceleration due to gravity;
(9) n=the ratio of the specific heat of the fluid at constant pressure, to the specific heat of the fluid at constant volume.
(10) $\overline{R}$=universal gas constant; and
(11) r=pressure ratio of downstream static pressure, P_{s2}, to upstream total pressure, P_0.

A testing fixture **152** having an annular passage **154** formed therein is secured in sealing engagement with cover **128**. A plurality of guide vanes **156** are provided within the annular passage **154** so as to impart a swirling motion to the air passing therethrough in order to simulate the angle of air impingement experienced by the nozzle vanes in the actual engine. The nozzle assembly **43** to be tested is mounted, as illustrated at the exit of the annular passage **154**.

During testing of the nozzle assembly **43** valve **134** is opened to the degree necessary to establish a desired differential, ΔP, between the plenum chamber pressure, P_3, and the barometric pressure, P_B. This pressure differential and effective cross-sectional flow area of the nozzle assembly will then determine the velocity of flow through the nozzle assembly.

Since nozzle assembly **43** and orifice **138** are in series with each other, it becomes evident that the mass-rate of flow, W, must be the same for both the nozzle assembly **43** and the orifice **138**. Accordingly, once the mass-rate of flow, W, is determined for orifice **138**, the same mass-rate of flow can be employed in the general compressible mass flow equation as applied to the nozzle assembly. The terms of the equation, as applied to the nozzle assembly will have the following meanings:

(1) A=actual cross-sectional flow area of nozzle assembly **43**;
(2) W=mass-rate of flow through the nozzle assembly (the same as the mass-rate of flow through orifice **138**);
(3) $P_0=P_3$=pressure upstream of the nozzle assembly;
(4) m=molecular weight of the fluid;
(5) T_0=total temperature upstream of the nozzle assembly;
(6) C=the discharge coefficient of the nozzle assembly;
(7) g=acceleration due to gravity;
(8) n=the ratio of the specific heat of the gas at constant pressure, C_p, to the specific heat of the gas at constant volume, C_v;
(9) $\overline{R}$=universal gas constant; and
(10) r=pressure ratio of downstream pressure, P_B, to upstream pressure, P_3.

In applying the general equation to the nozzle assembly, the equation may be rearranged into the following form:

$$CA=\frac{W}{P_0\sqrt{\frac{\overline{m}}{T_0}}\cdot\sqrt{\frac{2g}{\overline{R}}\cdot\frac{n}{n-1}}\cdot\frac{1}{r^n}\sqrt{1-\frac{n-1}{r^n}}}$$

It should also be noted that W, $\overline{m}$, T_0, g, n and $\overline{R}$ are the same values as when the general equation was applied to the orifice **138**. Further, r, is determined from the pressure differential, P_3-P_B. The only values which are not known are C and A. However, the solution of the equation, as rearranged above, yields a value which is the product of C, the discharge coefficient, and A, the actual cross-sectional flow area or, in other words, the effective or equivalent flow area of the nozzle assembly **43** which is precisely the value desired.

Similarly, the compressor drive turbine wheel **26** can be secured at the outlet end of a test fixture **158** (FIGURE 11) which also has an annulus **154** and guide vanes **156**. A shroud **162** located about compressor drive turbine wheel **26** and secured atop fixture **158** is provided in order to duplicate the confining effect that the shroud **96** will have within the engine. The remaining portion of the apparatus of FIGURE 11 can be made so as to be identical with that disclosed in FIGURE 10.

The general procedure employed in testing the nozzle assembly is followed in the testing of the compressor drive turbine wheel **26**. That is, the valve **134** is opened to the degree necessary to again establish the differential, ΔP, between P_3 and P_B.

The mass-rate of flow, W, through orifice **138** is then determined by the application of the general compressible mass flow equation. Once, W, is determined it then becomes possible to solve the same equation, as applied to the compressor drive turbine wheel, for the combined term, CA, (as discussed with reference to the nozzle assembly of FIGURE 10) of the turbine wheel **26**.

By employing the above method it then becomes possible to test a plurality of nozzle assemblies, one at a time, and determine and record thereon the equivalent or effective flow area exhibited by that particular nozzle assembly. Likewise, the compressor drive turbine wheels can be tested and the equivalent flow area, so determined, recorded on the individual compressor drive turbine wheels. It then becomes a matter of simply selectively matching a nozzle assembly and compressor drive turbine wheel, based on their respective equivalent flow areas, which will result in an equivalent flow area ratio within the critical limits defined on both sides of the ideal equivalent flow area ratio. Such a selectively matched set, when placed within the turbine engine will insure the engine of not only operating within the limits of the choke and stall lines of FIGURE 2 as achieved by gross aerodynamic matching to the compressor, but will further insure the engine of producing an output horsepower at least very closely approaching the designed maximum output horsepower as represented by point **78** of FIGURE 3.

Further, it has been discovered that as between any two nozzle assemblies or compressor drive turbine wheels, slight differences may exist between the respective curves defining the mass-rate of gas flow therethrough as a function of the pressure differential across such nozzle assemblies or compressor drive turbine wheels. This is believed due to the slightly varying aerodynamic characteristics of the particular nozzle assembly or compressor drive turbine wheel under consideration. Accordingly, the method for determining and critically selecting a particular nozzle assembly to a cooperating compressor drive turbine wheel can be modified so as to even avoid such slight discrepancies arising out of such varying aerodynamic characteristics.

That is, preferably, when the ideal equivalent flow areas are empirically determined, they should be determined by employing a pressure differential substantially equivalent to that pressure differential which the nozzle assembly or the compressor drive turbine wheel, as the case may be, will experience within the engine as during full rated power operation or maximum designed compressor speed.

Subsequently, the pressure differential employed during calibration of nozzle assemblies would vary from that pressure differential employed for calibrating compressor drive turbine wheels. For example, with nozzle assemblies, a sufficient pressure differential may be indicated when pressure gage 160 indicates a value which is 0.75 times the then existing barometric pressure, whereas, in the case of the compressor drive turbine wheel a reading of 1.17 times the then existing barometric pressure may indicate a proper pressure differential.

In each event, however, the calibration is conducted employing a pressure differential substantially equivalent to the pressure differential actually experienced by the test piece under conditions of actual engine operation thereby avoiding any influencing factors which might otherwise exhibit themselves due to the aerodynamic characteristics of the compressor drive turbine wheel or nozzle assembly.

The drawings and the foregoing specification constitute a description of the invention in such terms as to enable any person skilled in the art to practice the invention, the scope of which is indicated by the appended claims.

I claim:

In a gas turbine engine having a compressor; a compressor drive turbine wheel; and a turbine wheel inlet nozzle assembly; wherein said turbine wheel and nozzle assembly define, each by means of a plurality of radially directed circumferentially spaced blades, turbine wheel and nozzle motive gas flow areas; wherein said nozzle motive gas flow area is less than said turbine wheel motive gas flow area; and wherein the velocity of motive gas flow through at least one of said flow areas is in the range of Mach 0.9 to 1.0 during periods of maximum engine speed operation; the improvement of having said turbine wheel and said nozzle assembly critically matched to each other so that the ratio of said turbine wheel motive gas flow area to said nozzle motive gas flow area is within one percent of that ratio between said turbine wheel motive gas flow area and said nozzle motive gas flow area which enables said turbine engine to produce its maximum power.

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JULIUS E. WEST, *Primary Examiner.*

UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,319,931 May 16, 1967

Albert H. Bell III

It is hereby certified that error appears in the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 5, line 47, for "construction" read -- consideration --; line 53, for "R_{L1}" read -- A_{L1} --.

Signed and sealed this 3rd day of December 1968.

(SEAL)
Attest:

Edward M. Fletcher, Jr.
Attesting Officer

EDWARD J. BRENNER
Commissioner of Patents