

Sept. 7, 1937.

A. LYSHOLM

2,091,998

GAS TURBINE SYSTEM

Filed Feb. 23, 1935

3 Sheets-Sheet 1

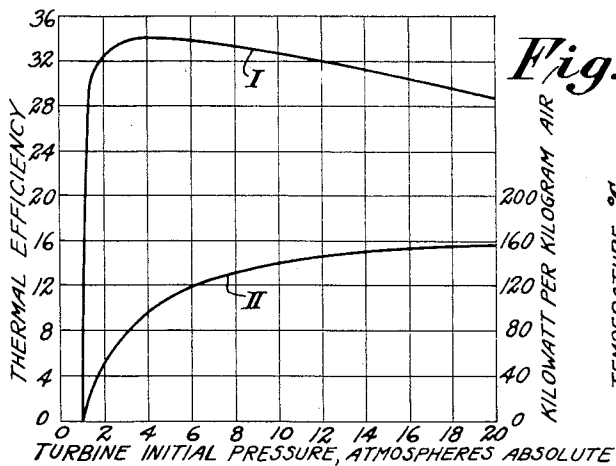


Fig. 1.

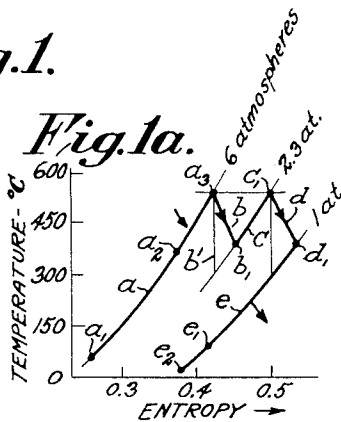


Fig. 1a.

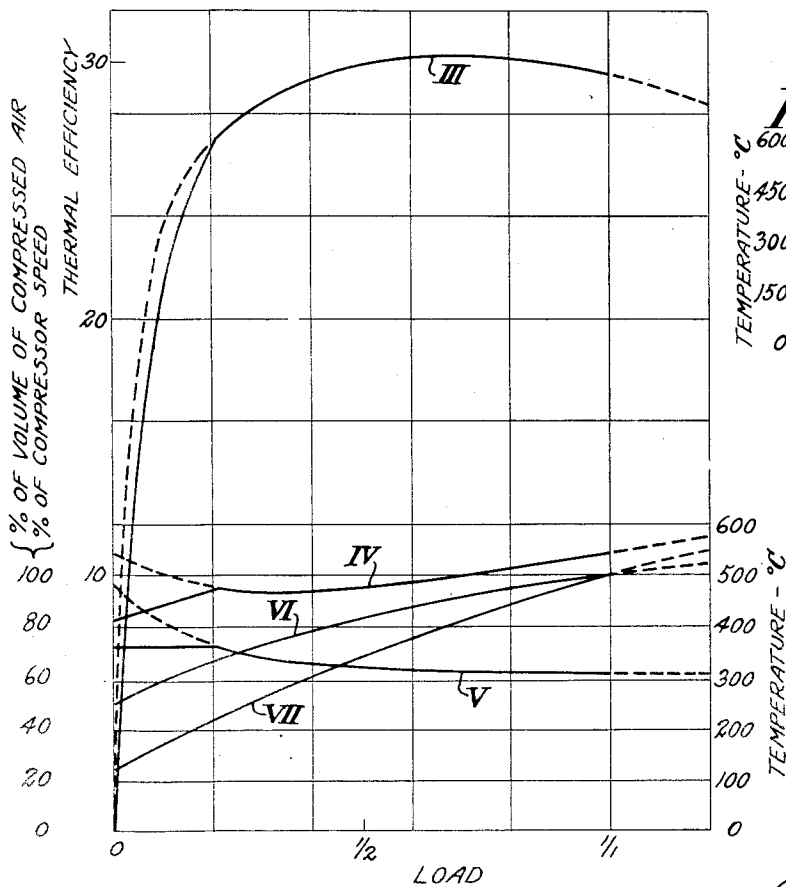


Fig. 2.

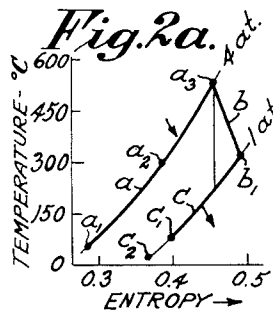


Fig. 2a.

INVENTOR.

Alf Lysholm

BY

H. J. Hallund
his ATTORNEY.

Sept. 7, 1937.

A. LYSHOLM

2,091,998

GAS TURBINE SYSTEM

Filed Feb. 23, 1935

3 Sheets-Sheet 2

Fig. 3.

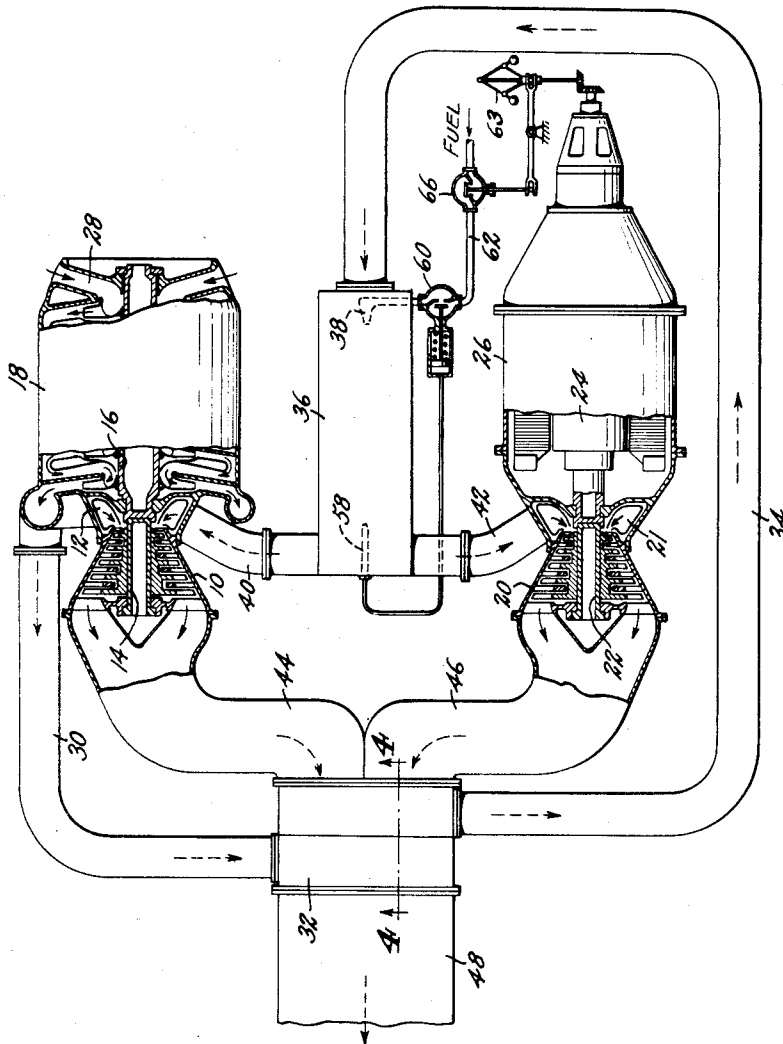
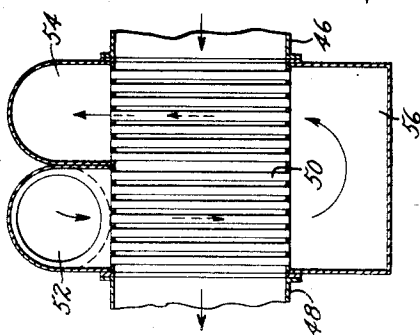


Fig. 4.



BY

INVENTOR.
A. Lysholm
BY *W. J. Hedlund*
his ATTORNEY.

Sept. 7, 1937.

A. LYSHOLM

2,091,998

GAS TURBINE SYSTEM

Filed Feb. 23, 1935

3 Sheets-Sheet 3

Fig. 5.

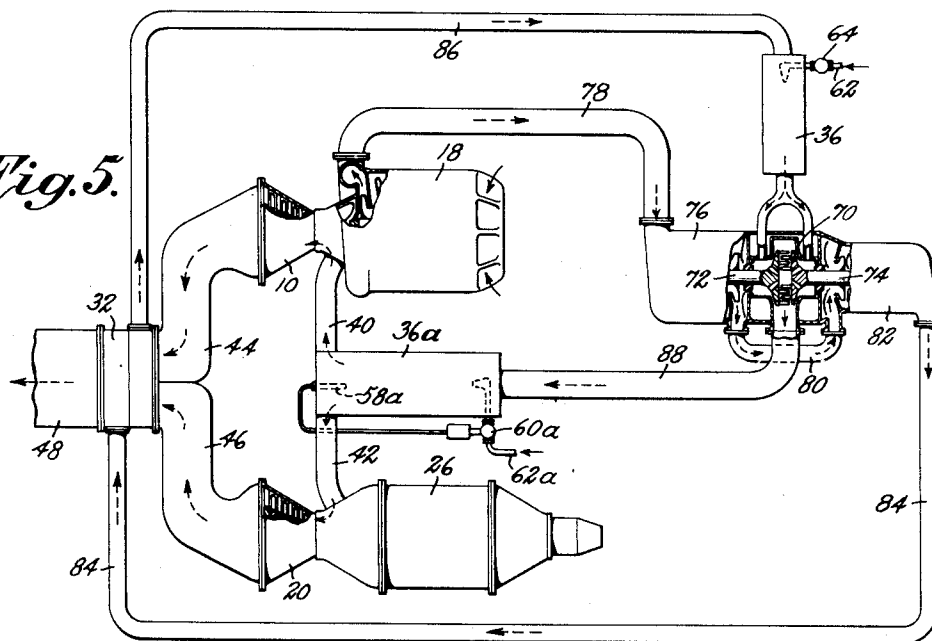
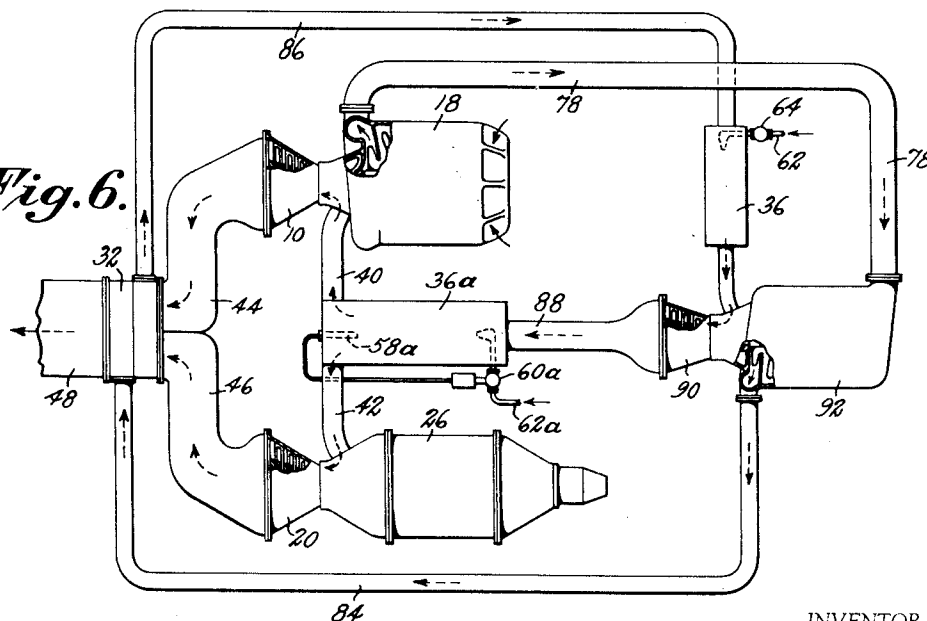


Fig. 6.



INVENTOR.

Alf Lysholm

BY

W. J. Hedlund
his ATTORNEY.

UNITED STATES PATENT OFFICE

2,091,998

GAS TURBINE SYSTEM

Alf Lysholm, Stockholm, Sweden, assignor to
Aktiebolaget Milo, Stockholm, Sweden, a cor-
poration of Sweden

Application February 23, 1935, Serial No. 7,693
In Germany February 24, 1934

12 Claims. (Cl. 60—41)

The present invention relates to gas turbine systems of the continuous combustion type, sometimes referred to as constant pressure systems in contradistinction to intermittent or explosion type systems.

In the attempts to develop a practical continuous combustion system the principal difficulty has been to prevent failure of the turbine blades due to the high temperature of the motive fluid without reducing the initial temperature of the motive fluid to a degree such that the possible thermal efficiency of the system is too low to make it practical.

Until recently it was thought that the problem of the continuous combustion gas turbine system could be solved only by using initial gas temperatures so high that known materials available for turbine blading could not withstand such temperatures. Recently it has been found possible to evolve a practically operable gas turbine system by the utilization of double rotation radial flow turbines of high efficiency operating with gases the initial temperature of which is in the range of from approximately 800° C absolute to 1000° C. absolute. This solution, however, has certain definite limitations when the radial flow turbine alone is employed. This is due primarily to the fact that the relatively very small area for flow of motive fluid at the inlet to the blading of this type of turbine precludes its economical utilization in many instances where large volumes of motive fluid must be handled. In gas turbine systems, the volume of motive fluid necessary to effect a given heat input into the system is very much larger than the volume of steam in a steam plant of corresponding heat input. This is because the specific heat of internal combustion gases is less than the specific heat of steam. Consequently the development of a continuous combustion gas turbine presents a problem involving the handling of large volumes of motive fluid in a plant of given size. There thus arises the problem as to whether or not it is possible to use the axial flow type of turbine in a system of this character, since the axial flow turbine is better adapted to handle large volumes of motive fluid than is the radial flow type of turbine. However, because of inherent structural limitations the axial flow turbine will necessarily have such high losses if operated in any manner that is practical with temperatures such as those above mentioned as being used with the radial flow turbine, that the turbine efficiency will be too low to give an acceptable overall thermal efficiency for the system. Consequently, it has heretofore been

generally believed that the axial flow turbine is generally unsuitable for this type of system.

In accordance with the present invention, however, the axial flow turbine, with its desirable characteristic of ability to handle large volumes of motive fluid, is employed, and the method of operation of the system and the means whereby such method may be carried into effect, will now be explained.

It is known that modern types of axial flow turbines can be successfully operated with motive fluid having an inlet temperature of as high as 540° C., and that with such turbines at average inlet pressures of 15 atmospheres there will be obtained comparatively high thermo-dynamic efficiency which may be as high as 86% or even higher.

I propose to utilize this type of turbine which, if efficiency of the above order is to be obtained, must be of the multiple stage type, and which is preferably of the multiple stage reaction type, and to use this turbine with gaseous motive fluid the initial temperature of which is within the range of which the upper limit is approximately 540° C. It would appear, if the teachings of the prior art were accepted, that such utilization of this type of turbine would be obviously unsuccessful because of the low initial temperature, but I have discovered that a practical system giving acceptable overall thermal efficiency can be provided by utilizing this type of turbine, operating with gaseous motive fluid at this relatively low initial temperature, provided that it is possible to recover by regeneration a sufficient quantity of heat from the motive fluid as exhausted from the system and to return this recovered heat to the motive fluid in a manner such that it can be effectively utilized.

In order to effect the regeneration necessary to make my proposed method of operation successful, it is necessary to maintain the pressure of the motive fluid as admitted to the turbine at a relatively low value. The reason for this is that if the initial pressure of the motive fluid is relatively high the amount of the heat drop in the motive fluid passing through the turbine is so great that the exhaust temperature is not sufficiently high for my purposes. In fact, in cases of high initial pressures it is even possible that the exhaust temperature will be so low that regeneration is impossible because of the fact that the exhaust temperature is below the temperature of the air when the latter is compressed to the initial temperature of the motive fluid. Because of this circumstance, I propose, in accordance

with my invention, to utilize for expansion in an axial flow turbine of the character described, motive fluid the initial temperature of which is within a range not exceeding approximately 540° C. and having an initial pressure within a range of which the lower limit is approximately two atmospheres and the upper limit of which may be approximately six to eight atmospheres. For this range of pressures a radial flow turbine would be relatively undesirable because of the fact that with gaseous motive fluid comprising gases of combustion the specific volume of the gases is too great to be most effectively handled by the radial flow turbine. When operating a turbine with an initial pressure as low as that contemplated in the arrangement above mentioned and with the turbine exhausting at not less than substantially atmospheric pressure, which must be the case when uncondensable gases constitute the motive fluid, the heat drop of the motive fluid in the turbine is comparatively so little that high exhaust temperature is a necessary result. With high exhaust temperature, a very considerable amount of regeneration may be effected, and I propose, in accordance with my invention, to utilize regeneration to return to the compressed air which constitutes one of the constituents of the motive fluid, an amount of heat which is at least as much and which may be more than the heat equivalent of the work extracted from the motive fluid in the turbine or turbines from which the exhaust gases are derived.

When the system is constructed in accordance with my invention, it is possible, by means of regeneration, to add as much heat energy to the air by regeneration as is added by the combustion of fuel with the air, which fuel is injected or otherwise supplied to convert the compressed air into gaseous motive fluid. The necessity for such extreme use of regeneration requires comparatively very large heat exchange apparatus which, moreover, should be constructed with particular care so as to afford high heat transfer efficiency. Such apparatus involves a very considerable additional expense in the construction of the system, but the additional expense required for this equipment is more than made up for by the high operating efficiency and the relatively low rate of fuel consumption for the whole system when operating in accordance with the invention.

While for the sake of simplicity I have described the above consideration as applying to a system in which the initial pressure of the motive fluid for the entire system is within the comparatively low pressure range mentioned, it is to be understood that the invention is not applicable solely to low pressure systems, but is equally applicable to systems in which a plurality of turbines operating at different pressure ranges are employed, and in which systems the invention is applicable to the turbine or turbines operating in the lowest pressure range in the system. Such turbines may conveniently be called end turbines.

As will hereinafter be more fully explained, high pressure turbines which may be of the radial flow type or of the axial flow type may be connected in series with such end turbine or turbines, but regardless of the character of such high pressure turbine or turbines, the characteristics of the regeneration and of the initial temperature and pressure of the motive fluid as admitted to the end turbine or turbines, remain in substantially the same relationship when the invention is adhered to. Stated in another way, the amount of heat transferred to the compressed air by means

of regeneration is at least as much or may be more than the heat equivalent of the work extracted from the motive fluid in the end turbine or turbines.

I further propose, in accordance with my invention, to so design the system that the overall thermal efficiency of the system attains its maximum value at part load, which is preferably in the range of from 50 to 75% of the normal full load for which the system is designed to operate. The reason for this will later be more fully explained.

In order to more fully explain the nature of the invention and the manner in which it is carried into effect, I have shown in the accompanying drawings certain thermal diagrams and also suitable embodiments of apparatus for carrying the invention into effect, which I will now proceed to describe.

Fig. 1 is a diagram showing certain efficiency characteristics of a gas turbine system embodying the invention;

Fig. 2 is another diagram illustrating other characteristics of such a system;

Fig. 1a is a temperature entropy diagram illustrating a thermal cycle typical of one embodiment of a system embodying the invention;

Fig. 2a is another temperature entropy diagram illustrating the thermal cycle of another embodiment;

Fig. 3 is a more or less diagrammatic plan view, partly in section, of a gas turbine system embodying the invention;

Fig. 4 is a section taken on the line 4-4 of Fig. 3;

Fig. 5 is a more or less diagrammatic plan view of a system embodying the invention and having a separate double rotation radial flow turbine acting as a high pressure turbine and exhausting to the portion of the plant with which the present invention is particularly concerned; and

Fig. 6 is another more or less diagrammatic plan view of a third embodiment similar to that shown in Fig. 5, but utilizing a different type of high pressure turbine.

In the diagrams of Figs. 1 and 2, I have shown certain characteristic curves illustrating the conditions which will obtain in gas turbine systems constructed in accordance with the present invention.

Fig. 1 refers to a gas turbine system of the type illustrated in Fig. 6 and the thermal cycle of which is shown in Fig. 1a. The ordinates represent percent thermal efficiency or useful power output of the system, as the case may be, and the abscissae represent initial pressure of the motive fluid, the initial pressure being considered as the pressure at the inlet of the end turbine or turbines of the system. Curve I indicates thermal efficiency of the system, and curve II indicates the useful power output in kilowatts per kilogram air compressed in the compressor means. The curves shown are based on calculated data assuming motive fluid temperature at the inlet of the turbine of 540° C., a thermal-dynamic turbine efficiency of 90%, compressor efficiency (isothermic) of 70%, regenerator efficiency of 92%, and a temperature of the compressed air leaving the compressor of 60° C.

This diagram shows that with increase of turbine inlet pressure the useful power output per kilogram compressed air of the system increases, but that with conditions of the nature assumed the total or overall thermal efficiency is very much more favorable in the field of relatively very low initial pressure than in the field of high pressures.

This is due, in part, to the nature of the utilization of heat by the regenerator under varying capacity conditions.

Fig. 2 illustrates conditions obtaining in a turbine system such as illustrated in Fig. 3, the thermal cycle of which is represented in Fig. 2a. In this diagram the abscissae represent percentage of load, and the ordinates represent thermal efficiency, temperature of motive fluid, number of revolutions of the compressor in percentage of normal full speed or volume of compressed air in percentage of normal volume at full load, as the case may be. In this diagram, curve III represents the thermal efficiency, curve IV the temperature of the motive fluid at the turbine inlet, curve V the temperature of the motive fluid as exhausted from the turbine, or exhaust gas temperature, curve VI the speed of the compressor, and curve VII the volume of air compressed. From these curves it will be evident that with increasing load on the system the speed of the compressor and the volume of air compressed both increase, but that at the same time the temperature of the motive fluid as exhausted from the turbine decreases. In this connection, it is to be noted that in the range of very light loads, approaching idling load, curves III, IV and V are shown with both full line and dotted line portions. The dotted line continuation of curves IV and V indicates the rapid rise of exhaust and inlet temperatures which accompany such light load operation if no control is provided to take care of this. These temperatures in many instances exceed safe values. In the curves illustrated, the no-load exhaust temperature would be nearly 500° C., and this temperature is too high for the exhaust end of the turbine blade system. Consequently, control is preferably provided whereby exhaust temperature at light load may be limited to a predetermined maximum value, and the solid line portions of curves IV and V indicate the conditions existing when such control is employed. A control of this nature is not germane to the present invention and has therefore not been illustrated in this application. Such control is fully disclosed and claimed in and forms the subject matter of my copending application Serial No. 749,006, filed October 19, 1934. The solid line portion of curve III indicates the effect on the overall efficiency of the system due to the employment of such exhaust temperature limiting control. Increasing load, however, means reduction in regenerator efficiency due to increase in quantity of air and decrease in the exhaust gas temperature.

It is also characteristic of heat exchangers of the type employed for regenerators that the efficiency of heat transfer increases as the load decreases from a value representing capacity operation of the regenerator to a value representing partial capacity operation.

The curves shown in Fig. 2 are based upon a turbine inlet pressure of four atmospheres and with a temperature of motive fluid at the inlet which is in the range of from 410° C. to 540° C. As will be seen from these curves, the efficiency curve is comparatively flat throughout the major range of operation of the system with the maximum efficiency lying within the range of 40 to 80% of full load. With the above pressure and temperature at the inlet of the turbine, it will be seen that the nature of the characteristics of operation of the units of the system provides an overall thermal efficiency, under the assumed conditions, which is of acceptably high value

throughout the major portion of the operating load range of the system.

Turning now to Fig. 3, the system shown therein is of the kind in which the turbines which separately drive the useful power means and the air compressor are connected in parallel with respect to the flow of motive fluid through the turbines, and in which these turbines, which are end turbines in the sense in which I employ this term, are also the highest pressure turbines in the system. In other words, in this system there is but one pressure range of expansion of the motive fluid, as will be seen from the thermal cycle represented in Fig. 2a which will be referred to later on.

In the system illustrated, 10 indicates an axial flow multiple stage gas turbine which is preferably provided with blading of the reaction type, and which preferably is constructed to afford full admission of motive fluid to the inlet of the turbine blading from a central admission chamber 12. Rotor 14 of the turbine is connected to the rotor 16 of air compressor 18, which is preferably of the multiple stage centrifugal or rotary type. Turbine 20, which advantageously is similar in construction to the turbine 10, has its rotor 22 connected to the rotor 24 of a generator 26, which delivers the useful power produced by the system.

The invention is not limited to the use of an electrical generator for delivering the useful power from the system. For convenience, I will refer to turbine 10 as a compressor turbine and to turbine 20 as a useful power turbine.

Air is drawn into compressor 18 through the inlet 28 and is delivered in compressed state to conduit 30 connected to the regenerator indicated generally at 32. From the regenerator the air is delivered by means of conduit 34 to the combustion chamber 36, to which liquid or other suitable fuel is supplied through fuel nozzle 38. Conduits 40 and 42 supply heated gaseous motive fluid, produced by the combustion of fuel in chamber 36, to the admission chamber 12 of turbine 10 and the admission chamber 21 of turbine 20 respectively, the flow of motive fluid being in parallel to the turbines. The exhaust conduit 44 of turbine 10 and the exhaust conduit 46 of turbine 20 lead to the regenerator 32, and from the regenerator exhaust gases are finally discharged 50 through conduit 48.

The specific design of the regenerator may be varied widely within the scope of the present invention. For purposes of illustration I have shown more or less diagrammatically a regenerator of the recuperative type in which a plurality of tubes 50 are disposed across the path of flow of the exhaust gases in their passage from conduits 44 and 46 to the discharge conduit 48. The conduit 30 delivers compressed air from the compressor and is connected to a header chamber 52 communicating with one end of a part of the tubes 50. A header chamber 54 provides communication between the air delivery conduit 34 and one end of the remaining tubes, and the opposite ends of all of the tubes are in communication with a common reversing header or chamber 56.

The flow of air and motive fluid through the system will be more or less evident from the drawings. Assuming the system to be in operation, air is drawn into the compressor and delivered in compressed state to the regenerator, passing in two passes through the tubes 50 and receiving heat from the exhaust gases passing

through the passage into the regenerator through which the tubes 50 extend. The heated compressed air is then delivered to the combustion chamber, where combustion of the fuel admitted to the chamber further increases the temperature and provides heated motive fluid which is delivered to the turbines. The exhaust gases from the turbines passing through the regenerator deliver a substantial part of their heat to the air passing from the compressor to the combustion chamber.

As previously pointed out, the specific form of construction of the regenerator may vary within the scope of the invention, but in order to make proper use of the invention the capacity of the regenerator must be sufficiently great so that when any given system is operating under conditions productive of maximum thermal efficiency for the system the regenerator will transfer from the exhaust gases of the system to the air at least as much heat as the heat represented by the heat drop of the motive fluid in the turbines. Stated in another way, the capacity of the regenerator must be such that at least as much heat is added to the compressed air by regeneration as is represented by the heat equivalent of the work done by the motive fluid in expansion in the turbine. This capacity may, of course, be readily calculated for any given design of regenerator, since the exhaust temperature of the gases may be determined by calculation under the conditions of operation which are productive of maximum efficiency of the system, and the temperature of the air delivered to the regenerator may also be determined for any given design of compressor.

Numerous different specific methods of controlling operation of the system in accordance with variations in load may be employed within the scope of the invention. Since, in accordance with the present invention, the inlet temperature of the motive fluid to the end turbines is maintained at a relatively low value, some means for automatically limiting this temperature may advantageously be employed, and for purposes of illustration and by way of example I have indicated in the combustion chamber 36 a thermostatic element 58 arranged to operate a valve 60 in fuel line 62 so as to throttle the fuel supply if the maximum permissible value of the temperature of the motive fluid is exceeded. Variations in load on the system are, in a system of this character, most advantageously taken care of by means of control of the fuel supply, and the control may advantageously be made responsive to load by means of a suitable governor 63 driven from the shaft of the useful power turbine and controlling a main fuel supply valve 66 so as to reduce the amount of fuel supplied to the combustion chamber upon a reduction in load on the system. It will be obvious that other refinements of control may be added, and that the system may be controlled by manual operation of such controls as are provided.

As heretofore mentioned, the end turbine or turbines with which the present invention is primarily concerned need not constitute the entire turbine structure of the system.

In Fig. 5 I have illustrated by way of example a turbine system which is adapted to operate with expansion of motive fluid in a plurality of pressure stages or, in other words, with a high pressure expansion stage ahead of the expansion stage represented by the end turbines. In this system the end turbines 10 and 20 drive respec-

tively a compressor 18 and generator 26, and exhaust respectively through conduits 44 and 46 to the regenerator 32. As in the embodiment shown in Fig. 3, these turbines are connected in parallel by means of conduits 40 and 42 for admission of motive fluid from combustion chamber 36a. In this instance, however, a high pressure turbine 70 of the double rotation radial flow type is employed in addition to the end turbines. Turbine 70 comprises oppositely rotating shafts 72 and 74, shaft 72 driving the rotor of a compressor 76, the inlet of which is connected by means of conduit 78 with the outlet of compressor 18. The outlet of compressor 76 is connected by conduit 80 with the inlet of compressor 82, the latter being driven by shaft 74 of turbine 70 and having its outlet connected by means of conduit 84 with the regenerator 32. Compressed air from regenerator 32 is conducted by means of conduit 86 to combustion chamber 36, to which fuel is supplied through pipe 62 under the control of a control valve 64 which may be manually controlled or which may be governor controlled from the useful power turbine as shown in Fig. 3. From combustion chamber 36 the motive fluid generated in this chamber is delivered to turbine 70 and this turbine exhausts through conduit 88 to the combustion chamber 36a. Fuel is supplied to the combustion chamber 36a through pipe 62a, and the maximum temperature of the motive fluid delivered from chamber 36a may be controlled by means of a thermostat 58a controlling a valve 60a.

The operation of a system of this kind will be largely evident from the preceding description of the system shown in Fig. 3. The low pressure end of the system operates in substantially the same manner as that described. The difference in the two systems is that by means of the compressors 76 and 82 connected in series with compressor 18, the air is compressed to a higher pressure than where a single compressor is used, and the motive fluid at this higher pressure which is generated in the combustion chamber 36 is first partially expanded in the double rotation radial flow turbine 70, which drives compressors 76 and 82. Because of the higher pressure of the motive fluid, the volume to be handled in the high pressure expansion stage is less than would be the case with a relatively low initial pressure, and for this reason the radial flow turbine may advantageously be employed.

Also, if desired, the radial flow turbine enables motive fluid to be utilized having a higher initial temperature than would be practical for use in an axial flow turbine. Consequently, with this arrangement, advantage may be taken of higher pressure and temperature characteristics of the motive fluid. The motive fluid exhausted at reduced pressure and temperature from turbine 70 is conducted to the combustion chamber 36a, which, in this embodiment, acts as a reheater. Reheating of the motive fluid in this combustion chamber may be effected in known manner by combustion of additional fuel with excess air remaining in the motive fluid as delivered from the primary combustion chamber 36. As in the case where no high pressure turbine stage is employed, the temperature of the motive fluid leaving the combustion chamber 36a is limited to a relatively low temperature and the pressure is also low so that the heat drop in the end turbines is comparatively small, with resulting high exhaust temperature, thus permitting the large amount

of heat recovery by regeneration which takes place in the regenerator 32.

In Fig. 6, still another embodiment is illustrated, which is similar in arrangement to that just described in connection with Fig. 5, the only difference being that instead of utilizing as the high pressure turbine a double rotation radial flow turbine, an axial flow turbine 90 of the same general type as turbines 10 and 20 is employed. Turbine 90, in this instance, operates a compressor 92 connected in series by means of conduits 78 and 84 between the low pressure compressor 18 and the regenerator 32. In this arrangement, the axial flow turbine 90 is utilized as the high pressure turbine of the system, but when a turbine of this sort is employed instead of the radial flow type of turbine it will be evident that because of operating characteristics which I have previously discussed, the initial temperature of the motive fluid as delivered from the primary combustion chamber 36 must be maintained at a relatively low value, that is, within the range of which the maximum is approximately 540° C.

The embodiments shown in Figs. 5 and 6 have been illustrated primarily only for the purpose of indicating that the present invention is not confined solely to low pressure systems. It will be understood by those skilled in the art that the invention may be incorporated in many different specific designs of systems intended to operate with a plurality of different ranges of expansion in different turbines, and that the type, number and arrangement of the turbines of higher pressure in such systems may vary without affecting the principles of the invention.

In order to further illustrate the character of the thermal cycle in a system such as that shown in Fig. 6, I have shown in Fig. 1a this cycle in a temperature entropy diagram in which the ordinates are temperature and the abscissae entropy. In this diagram point e_2 represents the temperature of the air as drawn in by the compressor and the constant pressure line a represents the portion of the cycle during which heat is added to the air, point a_1 representing the temperature of the air as delivered from the high pressure compressor, the point a_2 representing the temperature of the air as delivered from the regenerator, and the point a_3 representing the temperature of the motive fluid as delivered from the combustion chamber. Line b represents the expansion of the motive fluid in the high pressure turbine, line b' indicating the theoretical perfect adiabatic expansion line, and the point b_1 representing the temperature of the motive fluid as exhausted from the high pressure turbine. The constant pressure line c represents the addition of heat to the motive fluid in the reheater, point c_1 representing the temperature of the reheated motive fluid which, in this instance, is substantially the same as the temperature at the inlet of the high pressure turbine. Line d represents the low pressure expansion stage in the end turbines to a temperature d_1 , and that part of the constant pressure line e from d_1 to e_1 represents the drop in temperature of the exhaust gases in the regenerator which effects the heating of the air along line a from the point a_1 to the point a_2 .

As will be seen from the diagram represented in Fig. 1a, the amount of heat transferred by the regenerator to the compressed air is more than the heat equivalent of the work done by the motive fluid in expanding in the end turbines, the first-named amount of heat corresponding to line a from point a_1 to point a_2 , and the last-named

amount of heat corresponding to line d from point c_1 to point d_1 . It will further be evident that in this instance the amount of heat added by regeneration and corresponding to line a_1 — a_2 is also more than the heat added by combustion corresponding to line a_2 — a_3 .

In Fig. 2a, I have illustrated the thermal cycle of a system of the kind shown in Fig. 3, in which there is but one expansion stage. In this diagram, the constant pressure line a from point a_1 to point a_2 corresponds to the heat added to the air in the regenerator and the same line from point a_2 to point a_3 corresponds to the heat added to the air in the combustion chamber. Line b represents the expansion in the turbines to a temperature at point b_1 , and the constant pressure line c from point b_1 to point c_1 represents the heat drop of the exhaust gases in the regenerator.

By way of example, I give below the principal characteristics of the apparatus for a gas turbine system of the kind shown in Fig. 3 and embodying the principles of my invention. For such a system having a full load capacity of 2200 kilowatts, the compressor is preferably designed to deliver 20 kilograms of air per second at 4 kilograms per square centimeter pressure, and at a temperature of approximately 60° C. Such compressor is advantageously designed to operate at full capacity at a speed of 3000 R. P. M., the compressor turbine advantageously comprising six rows or stages of blading and being adapted to operate at 3000 R. P. M. The turbines are designed for an inlet temperature of approximately 540° C. at full load, the path for flow of motive fluid through the turbines being so designed that the turbines expand under full load operating conditions approximately 20.2 kilograms of motive fluid per second from a pressure of 4 kilograms per square centimeter to an exhaust pressure of 1 kilogram per square centimeter, and from the inlet temperature to an exhaust temperature of approximately 320° C. The regenerator has a heat transfer surface of 1100 square meters capable of transferring from the above mentioned quantity of exhaust gases sufficient heat to raise the temperature of the air delivered by the compressor from its compressor outlet temperature of 60° C. to a temperature of 300° C.

While in compliance with the patent statutes, I have described my invention in connection with certain illustrative examples of apparatus for carrying it into effect, it will be understood that the invention is equally applicable to many different variations of continuous combustion gas turbine systems in which the number and disposition of the several turbines of the system may be quite different.

The invention is accordingly to be understood as embracing all forms of apparatus and variations in method of operation falling within the scope of the appended claims, which are to be construed as broadly as is consistent with the state of the prior art.

Certain features of construction relating to the turbo-compressor and turbo-generator units shown but not claimed herein are claimed in my copending application, Serial No. 140,639 filed May 4, 1937.

What I claim is:

1. A gas turbine system of the continuous combustion type having end turbine means and a regenerator, said end turbine means being of the axial flow type and having a plurality of expansion stages, means including a compressor

for supplying heated gaseous motive fluid to said turbine means at a pressure not exceeding approximately 8 kg. per sq. cm., and means for conducting the exhaust gas from said turbine means to said regenerator to heat the air compressed by said compressor, said regenerator having a capacity enabling it to transfer to said air an amount of heat equal to at least the heat equivalent of the work done by said motive fluid in expanding in said turbine means.

2. A gas turbine system of the continuous combustion type having end turbine means and a regenerator, said end turbine means being of the axial flow type and having a plurality of expansion stages, means including a compressor for supplying heated gaseous motive fluid to said turbine means, means for limiting the temperature of said gaseous motive fluid as admitted to said turbine means to within a range of which the upper limit is approximately 540° C., and means for conducting the exhaust gas from said turbine means to said regenerator to heat the air compressed by said compressor, said regenerator having a capacity enabling it to transfer to said air an amount of heat equal to at least the heat equivalent of the work done by the motive fluid in expanding in said turbine means.

3. A gas turbine system of the continuous combustion type having end turbine means and a regenerator, said end turbine means being of the axial flow type and having a plurality of expansion stages, means including a compressor for supplying heated gaseous motive fluid to said turbine means at a pressure not exceeding approximately 8 kg. per sq. cm., means for limiting the temperature of said gaseous motive fluid as admitted to said turbine means to within a range of which the upper limit is approximately 540° C., and means for conducting the exhaust gas from said turbine means to said regenerator to heat the air compressed by said compressor, said regenerator having a capacity enabling it to transfer to said air an amount of heat equal to at least the heat equivalent of the work done by the motive fluid in expanding in said turbine means.

4. A gas turbine system of the continuous combustion type having end turbine means, means including a compressor for supplying heated gaseous motive fluid, means for supplying said motive fluid to said end turbine means at a pressure not exceeding approximately 8 kg. per sq. cm., a regenerator for heating the air compressed by said compressor, means for conducting to said regenerator the exhaust gas from said end turbine means, said regenerator having a capacity enabling it to transfer to said air an amount of heat equal to at least the heat equivalent of the work done by the motive fluid in expanding in said end turbine means, and said system being constructed and arranged to operate at maximum efficiency at part load.

5. In the operation of a gas turbine system of the continuous combustion type having end turbine means of the axial flow multiple stage type and a regenerator, that improvement which consists in forming motive fluid by compressing air and burning fuel therewith, expanding said motive fluid in said turbine, conducting the exhaust gas from said end turbine means to the regenerator, limiting the temperature of the motive fluid at the inlet of said end turbine means to a relatively low temperature, limiting the pressure of the motive fluid at the inlet end of said end turbine means to a relatively low pressure, and adding to said compressed air in said

regenerator an amount of heat equal to at least the amount of heat represented by the heat drop of the motive fluid due to its expansion in said end turbine means.

6. In the operation of a gas turbine system of the continuous combustion type having end turbine means of the axial flow multiple stage type and a regenerator, that improvement which consists in forming motive fluid by compressing air and burning fuel therewith, expanding said motive fluid in said turbine, conducting the exhaust gas from said end turbine means to the regenerator, conducting the motive fluid to said end turbine means at a pressure not exceeding approximately 8 kg. per sq. cm., and adding to said compressed air in said regenerator an amount of heat equal to at least the amount of heat represented by the heat drop of the motive fluid due to its expansion in said end turbine means.

7. In the operation of a gas turbine system of the continuous combustion type having end turbine means of the axial flow multiple stage type and a regenerator, that improvement which consists in forming motive fluid by compressing air and burning fuel therewith, expanding said motive fluid in said turbine, conducting the exhaust gas from said end turbine means to the regenerator, limiting the temperature of the motive fluid at the inlet of said end turbine means to a value not exceeding approximately 540° C., limiting the pressure of the motive fluid at the inlet end of said end turbine means to a value not exceeding approximately 8 kg. per sq. cm. whereby to obtain a relatively low heat drop of the motive fluid in said end turbine means, and adding to said compressed air in said regenerator an amount of heat equal to at least the amount of heat represented by the heat drop of the motive fluid due to its expansion in said end turbine means.

8. In a gas turbine system of the continuous combustion type, end turbine means comprising a compressor turbine and a useful power turbine, said turbines being of the axial flow multiple stage type, a compressor driven by said compressor turbine, a combustion chamber, a regenerator, conduits connecting said combustion chamber, said turbines and said regenerator for flow of motive fluid from the combustion chamber in parallel through the turbines to the regenerator, and means including said compressor, said regenerator and said combustion chamber for supplying motive fluid to said turbines, said regenerator having a capacity enabling it to transfer to the air compressed by said compressor an amount of heat equal to at least the heat equivalent of the work done by the motive fluid in expanding in said turbines.

9. In a gas turbine system of the continuous combustion type, end turbine means comprising a compressor turbine and a useful power turbine, said turbines being of the axial flow multiple stage type, a compressor driven by said compressor turbine, a combustion chamber, a regenerator, conduits connecting said combustion chamber, said turbines and said regenerator for flow of motive fluid from the combustion chamber in parallel through the turbines to the regenerator, and means including said compressor, said regenerator and said combustion chamber for supplying motive fluid to said turbines at a pressure not exceeding approximately 8 kg. per sq. cm. and at a temperature not exceeding approximately 540° C., said regenerator having a capacity enabling it to transfer to the air compressed by said compressor an amount of heat

equal to at least the heat equivalent of the work done by the motive fluid in expanding in said turbines.

5 10. In a gas turbine system of the continuous combustion type, end turbine means of the axial flow multiple stage type, means including an air compressor for supplying gaseous motive fluid to said end turbine means at low pressure, and a regenerator receiving exhaust gas from said end turbine means for heating air delivered by said compressor, said regenerator having a capacity enabling it to transmit to the air at least as much heat as the heat equivalent of the work done by the motive fluid in expanding in said end turbine means.

10 11. In the operation of a gas turbine system of the continuous combustion type, that improvement which consists in extracting work from gaseous motive fluid in a low pressure end range of expansion, and adding to a constituent of the motive fluid by regeneration from the exhaust gases from the system at least as much heat as that represented by the heat drop in the motive fluid in said range of expansion.

12. In the operation of a gas turbine system of the continuous combustion type having end turbine means of the axial flow multiple stage type and a regenerator, that improvement which consists in forming motive fluid by compressing air with increase of temperature due to compression and burning fuel with the compressed air, expanding said motive fluid reactively in said end turbine means, conducting the exhaust gas from said end turbine means to the regenerator, limiting the temperature of the motive fluid at the inlet of said end turbine means to a value not exceeding approximately 540° C., limiting the pressure of the motive fluid at the inlet end of said end turbine means to a value not exceeding approximately 8 kg. per sq. cm., whereby to extract a relatively small proportion of the available heat of the motive fluid in said end turbine means and adding to said compressed air in said regenerator an amount of heat equal to at least the amount of heat extracted from said motive fluid due to expansion thereof in said end turbine means.

ALF LYSHOLM.