



Study of gas turbines and development of an experimental gas turbine at Montana State College
by Diyonis V Demirdjoglu

A THESIS Submitted to the Graduate Committee in partial fulfillment of the requirements for the
degree of Master of Science in Mechanical Engineering

Montana State University

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Abstract:

The Gee Turbine , after the Invention of Steam engine by James Watt in 1776, followed by Otto's Internal Combustion engine in 1876 and Parson's and De Level's steam Turbine in 1885, is the fourth type of prime mover invented, backed with experience of long years. The experience gained with Diesel engines since 1895 and the final retouching of steam turbines, form the backbone of modern , constant pressure combustion gas turbine.

Gas turbines known from the time of Hero of Alexandria , 130 B.C. , and the conventional wind mill, entered the family of prime movers quite recently, although different patents on gas turbines start from the time of John Barber of England in 1791. Its successful development is closely associated with the use of high temperatures and the realization of an efficient compressor, hence the progress gained in this field must be directly attributed to the combined efforts of Metallurgists and Aerodynamists.

The inherent simplicity of its design in many respects, ideally adapts the gas turbine for certain classes of service, and it is toward this end that its present development is being directed.

STUDY OF GAS TURBINES
&
DEVELOPMENT OF AN EXPERIMENTAL GAS TURBINE
AT MONTANA STATE COLLEGE
by
DIYONIS V. DEMIRDJOGIU

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P. V. Demirefoyle

ABSTRACT

The Gas Turbine , after the invention of Steam Engine by James Watt in 1776, followed by Otto's Internal Combustion Engine in 1876 and Parson's and De Laval's Steam Turbine in 1885, is the fourth type of prime mover invented, backed with experience of long years. The experience gained with Diesel Engines since 1895 and the final retouching of steam turbines, form the backbone of modern , constant pressure combustion gas turbine.

Gas turbines known from the time of Hero of Alexandria , 130 B.C. , and the conventional wind mill, entered the family of prime movers quite recently, although different patents on gas turbines start from the time of John Barber of England in 1791. Its successful development is closely associated with the use of high temperatures and the realization of an efficient compressor, hence the progress gained in this field must be directly attributed to the combined efforts of Metallurgists and Aerodynamists.

The inherent simplicity of its design in many respects, ideally adapts the gas turbine for certain classes of service, and it is toward this ends that its present development is being directed.

INTRODUCTION

Gas Turbine is the newest and most promising addition in the family of prime movers. Man's ambition to invent a power generating unit simple and efficient with as many less wearing parts, starts from the time of Hero of Alexandria, who had as followers innumerable inventors without any appreciable success up to the present time. And it is logical for one to inquire why, since the Gas Turbine had so much time and effort devoted to its development, it had so little success.

Mechanical Engineers are very familiar with Carnot Cycle which states in a simple expression, the best that can possibly be done in the conversion of heat energy into mechanical energy. The efficiency which is given by Carnot's formula can not be carried out in any actual engine due to inherent mechanical losses in any one, but it is of great value as a criterion of the maximum result approachable.

The mathematical derivation of Carnot's formula is very simple. As it is well known, the efficiency of any power generating unit, is given by the ratio of output over input, or net work over heat supplied. So, we have

$$\text{Eff.} = \frac{Q_2 - Q_1}{Q_2} = \frac{T_2 - T_1}{T_2}$$

From the above expression, it is clearly seen that to improve the efficiency of any engine, even that of the ideal Carnot, either T_2 (maximum temperature in the cycle), must be too high

or T_1 (lower temperature in the cycle) must reach the absolute zero i.e. -460 F. Since the second condition is impossible in this world we live under existing conditions, much effort has been devoted in raising T_1 so that we will have a higher obtainable efficiency.

The superiority of internal combustion engine over the steam unit lies on this fact. In the I.C. engine for a fraction of a time, we have a very high temperature raise, a case which can not exist in a steam turbine or engine; there some parts of the cycle are continuously at the maximum temperature of the cycle and metals can not stand high temperatures as those encountered in the internal combustion engines.

Due to the fact, as it will be seen in the later part of this Thesis, in the Combustion Gas Turbine, the type to attain commercial significance up to the present time, we must have continuous high temperatures, all attempts of the earlier designers and inventors failed due to the inability of the existing materials to withstand the high temperatures necessary to produce suitable high efficiencies. An other important obstacle to the early inventors was the lack of a compressor of adequate efficiency to make the cycle feasible.

The theoretical part of this thesis , deals with the history, the thermodynamic principles , the types and the latest development in the Gas Turbine field, while the experimental

one, explains the developments done in the Mechanical Engineering Laboratories of Montana State College in transforming an aviation turbosupercharger to an open cycle simple gas turbine plant.

HISTORY

Literature on Gas Turbines disclose that in 1791 John Barber of England obtained a patent on the first machine having an intrinsic resemblance to the present days combustion gas turbine plant. John Dumbell, also of England patented in 1808 the first explosion gas turbine. Later we have experiments on gas turbines conducted by Armengaud and Lemale . In those, a fan delivered air under pressure to a combustion Chamber where, the air mixed with gaseous fuel, burned, and the products of combustion cooled by excess air directed in the form of a jet onto a wheel.

After those first patents there are some others the most important of which is by Dr. F. Stolze (1872) of Charlottenburg. Although the unit had striking resemblances with the modern ones, due to the inadequacy of the air compressor it had unfavorable results. Together with the original steam turbine patent by Sir Charles Parsons (1884) mention was done about the gas turbine. Here the compressor was to be of the axial type. Much effort has been spend on those but the idea later was abandoned primarily because of the lower efficiency of axial compressors compared to the centrififigal ones.

Here we might mention the research and projects made by Dr. S. A. Moss (1900) while being a student at the university of California. A Master's Thesis in the Un. of Cal. Library gives account of the proposed project. The same investigator

after that started gas turbine research at Cornell University but not particular results out of these experiments were obtained.

Later we have Karavodine in Paris (1908) having built a 2 hp single stage , impulse turbine operating on the explosion cycle. Four nozzles were circumferentially spaced around the rim of the 6 in. diameter wheel. Connected to each nozzle was a separate waterjacketed combustion chamber , wherein the explosion of the charge caused an increase in pressure and the gases expanded through the nozzle onto the wheel. The cycle repeated itself after the suction effect of the departing gases drawing in fresh charge. The explosions were timed to occur consecutively around the wheel periphery.

An other well known explosion type gas turbine, is the Holzwarth unit (1908). At that time the only practical cycle was this one because of the inadequacy of air compressors. This caused the early designers to investigate cycles that avoided use of air compressors. In fact, in the early designs of Holzwarth gas turbines air at atmospheric pressure was taken in into the combustion chamber, after which the gas was admitted and fired at constant volume , thus raising the pressure to between 70 to 100 psi. Holzwarth turbines have been built by the Korting Company of Hanover, Maschinen -Fabrik Thyssen of Ruhr and Brown Boveri Company of Baden , Switzerland. These turbines work on the Otto cycle, the expansion phase of the cycle extending to

to atmospheric pressure. The explosion occurs upon ignition of a charge of air and gas introduced under pressure into the combustion chamber. The pressure in it increases until it overcomes the action of a spring loaded valve, permitting the gases to a nozzle whence they are discharged at high velocity onto a turbine wheel. The nozzle valve is specially constructed so that it remains open under oil pressure until the combustion chamber is emptied. Expansion is followed by a scavenging operation which clears the combustion chamber of residual burnt gases and also cools the turbine blades. After scavenging a fresh air is admitted and the cycle repeated.

There are very few Holzwarth turbines under operation today. Stodola^I says that the highest overall thermal efficiency obtained in any of the experiments performed up to 1927 is about 13 percent. Aside from those patents and research work carried on gas turbines, there are numerous others. The above ones had a predominant influence on gas turbine progress. Besides, there are a couple of exhaust turbines utilizing the exhaust gases from Internal Combustion Engines; their application is of limited capacity and can not be considered as prime movers.

I- Dr. J.T. Rettaliata, Current Trends in Power Plant Development, Allis Chalmers Bulletin, No: R-6213, 1942

WHAT A GAS TURBINE IS

As far as thermodynamic cycle is concerned, there is little difference between a diesel internal combustion engine and gas turbine referred to as a "rotating diesel". The main difference between those two power generating units is that the diesel repeats the cycle intermittently and has incomplete expansion to save manufacturing cost, while the gas turbine of combustion type has no interruption whatever in the power generation and the expansion of gases is complete.

Those not familiar with the heat cycle think that the term " Gas Turbine " implies a turbine run with gaseous fuel, whereas it is a turbine operated with the products of combustion resulting from the burning of a liquid fuel, pulverized solid and even gaseous for stationary power plants.

The cycle of both internal combustion engine and gas turbine in simple words is the following: Air is compressed, fuel is injected and burned, and finally the high temperature gases under pressure are expanded to a few pounds above the atmosphere or directly to it , producing in the process useful power in excess of that required to compress the air. The internal combustion engine uses one structure for all those functions i.e. the air is compressed, liquid fuel is burned, and gases are expanded all in a cylinder. Because engine makes one structure do all three jobs , it must do them successively, so that the power output is cyclically interrupted.

The gas turbine power unit on the other hand, separates the the three operations, assigning a separate specialized mechanism for each. The air is compressed in a physically separate compressor, fuel is burned in a adjacent combustor, while the gas turbine itself serves only to expand the gases of combustion enabling it to drive the compressor and some useful load. By this system each of the three elements operates continuously, so that there is no interruption in the power output of the turbine. Because each can be designed for a single purpose, and because the speeds can be high, the total weight of the gas turbine power unit for the same output can be less than that of an internal combustion engine.

EXPERIENCE GAINED FOR THE MODERN
GAS TURBINE CYCLE

In the past twenty years the wide research carried and the experience gained from the gas turbines and compressors used in the Houdry cracking process and the Velox boilers served as a guidance for the development of present days gas turbines.

Houdry Cracking Process

In the latter part of 1936, at the Marcus Hook Pa. refinery of the Sun Oil Company the first successful gas turbine was placed in operation in the United States^I, and has been in practically continuous operation since January 1937, except for short periods of inspection every 6 - 8 months. The Houdry catalytic-cracking process is as follows:

Oil is vaporized and passed through a catalyst. As one of the results of the chemical changes which occur, carbon is deposited on the catalyst the efficiency of which is lowered. To remove the carbon they arrange the catalytic containers in such a manner, that while some are regenerated the others operate on the oil cycle. Both cycles from container to container operate on a predetermined basis. The carbonaceous deposit is oxidized by the air from the compressor with 45 psi air. The heat content of the products of regeneration exceeds that required to compress

I - P.R. Sidler, Gas Turbines for Blast Furnace Blowers, Iron and Steel Engineer, April 1945

the air . The excess energy is recovered either as electric power or steam for use in the process.

Up to this date there are in the United States about 35 gas turbines operating in connection with the Houdry process . Most of them are built by Brown Boveri and the rest by Allis Chalmers after a licence granted to them during the War. The nominal rating of some of them in c.f.m. is:

<u>Number</u>	<u>Nom. Rating c.f.m.</u>
12	23,000
13	40,000
1	60,000

The main trouble they had with these units was failure of bearings and so after 1942 they started to install Kingsbury bearings to eliminate to some extent those failures. (Fig. 8) shows a turbosupercharger (Turbocompressor) used in the Houdry Process .

Velox Boilers

Experiments and refinements carried in producing single stage blowers for supercharging 4 cycle diesel engines, led in 1930 to the development of the Velox steam generator in Switzerland. This is a boiler ^I with supercharged combustion space and forced water circulation giving results of high efficiency

I- S. A. Tucker, Gas Turbines, Mechanical Engineering, June 1944, p.365

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and of compact design. In this system, the combustion air is compressed from 25 to 35 psi , the compressor being driven by a multistage exhaust gas turbine. The need for an efficient air compressor to be used with these boilers caused the development of axial flow compressors. Between 1931 and 1940 the Brown- Boveri Company , Baden, Switzerland, built 90 of these and about 20 of larger output 25,000 to 125,000 c.f.m. to be used in the United states for the Houdry catalytic cracking process and for various other purposes.

A Velox boiler installed on the French ship Athos II increased its power from 10,000 shp to 16,000shp replacing one of its 7 original boilers. This shows clearly the high efficiency of Velox steam generators.

TYPES OF GAS TURBINES

The Combustion Gas Turbine is the newest addition in the family of prime movers. This is sometimes referred to as a " constant pressure " machine to be distinguished from the explosion gas turbine or Holzwarth type previously mentioned. Although the last one has been manufactured by Brown-Bovery and other European firms , there are very few in a working condition at present time.

The steady flow process constant pressure gas turbine presently is taking much favor due to its simplicity in design and because it has been proved of higher efficiency and adapts itself better to the various needs present. Generally we have 4 types of gas turbines , if we exclude those used for supercharging internal combustion engines and other used for the Houdry catalytic cracking process and which are not design to produce power by themselves.

The 4 types of gas turbines used for Power Generation include:

- 1- Open Cycle gas turbine
- 2- Closed Cycle gas turbine
- 3- Semiclosed Cycle gas turbine
- 4- Power Gas Cycle gas turbine

Open Cycle Gas Turbine Plant

The cycle in its simplest form consists of a compressor, combustion chamber, and a turbine. The compressor takes in atmospheric air and raises its pressure. In the combustion chamber fuel of any kind burns in this compressed air , raising its

temperature and increasing its heat energy. This produces a working fluid that can be expanded in the turbine to develop mechanical energy. Part of it is consumed by the compressor in raising the air pressure and the remainder is available as useful work.

Closed Cycle Gas Turbine Plant

The closed cycle plant differs from the open one, in that air is recirculated. (Fig. I) Air is heated indirectly in an air heater, corresponding to the boiler in a steam plant, i.e. the air does not come in contact with the combustion products. Then, it expands through the turbine in the usual way and passes to a precoolers corresponding to a condenser in a steam plant. This restored air to original pressure and temperature, enters the compressor again to repeat the cycle. Use of gases other than air as a working medium offers certain advantages in specific output and efficiency.

Most of the advantages seen in the closed cycle grow out of the fact that it operates at pressures above atmospheric. It is characteristic of gas turbines that for each temperature and machine efficiency there is one best pressure ratio. If this is taken as $r = 5$, then instead of 75 psi in an open cycle, here we have 400 psi as a top pressure in the cycle with compressor inlet at 80 psi. These higher pressures mean higher air density and smaller volume. In sequence that means smaller dimensions for the turbine, compressor e.t.c. In turn smaller dimensions for a

given output permit higher temperatures without exceeding conservative stress limits. From the point of heat transfer high density improves heat transfer coefficient, and this coupled with smaller volume, makes for substantial reductions in the size of heat exchangers. Freedom from fouling, because only clean air circulates makes high velocities and small tube diameters practical for further reduction of size. Plant output may be changed by raising or lowering working medium density; this does not change temperatures and according to Carnot 's formula, it gives good part load efficiency.

An other advantage of closed cycle is that with almost constant volume flow, turbine and compressor blading can be designed for maximum efficiency rather for an average one. On the other hand, the closed cycle requires two additional elements; an air heater and a precooler which means additional elements of considerable size and cost. Besides, the cycle is not free from any cold water need although this may be only a fraction of a comparable steam plant.

Semiclosed Cycle Gas Turbine Plant

A semiclosed cycle was proposed by Westinghouse engineers in 1944. In that fuel is burned directly in the circuit. Supply of fresh air needed for combustion comes from an auxiliary compressor driven by a small turbine that operates on gas drawn from the main cycle to offset effect of adding combustion air, pressures suggested being 150 and 600 psi. Direct firing eliminates

the air heater.

A different design has been built by Sulzer Bros in Switzerland. This unit can be considered as a closed and open cycle interconnected. The closed cycle, handling clean air comprises two compressors and their drive turbine, (Fig. 2) a regenerator and a precooler. Fresh air enters the system from an auxiliary compressor, just ahead of the precooler. The heater combines direct and indirect firing. Fuel is burned in air drawn from the cycle just ahead of the heater; combustion products pass through tubes, giving up heat to clean air flowing over the tubes. Clean air supplies the compressor -drive turbine; combustion products power the separate output turbine which exhausts to atmosphere. Cycle pressure are 60 and 240 psi.

Power - Gas Cycle Gas turbine Plant

This cycle is a combination of Internal Combustion Engine with a compressor and a turbine. All three units are mechanically connected. Net output of combination is engine plus turbine output minus compressor load. Of course the primary reason for such a combination is to supercharge the internal combustion engine and naturally includes the disadvantages of reciprocating machinery.

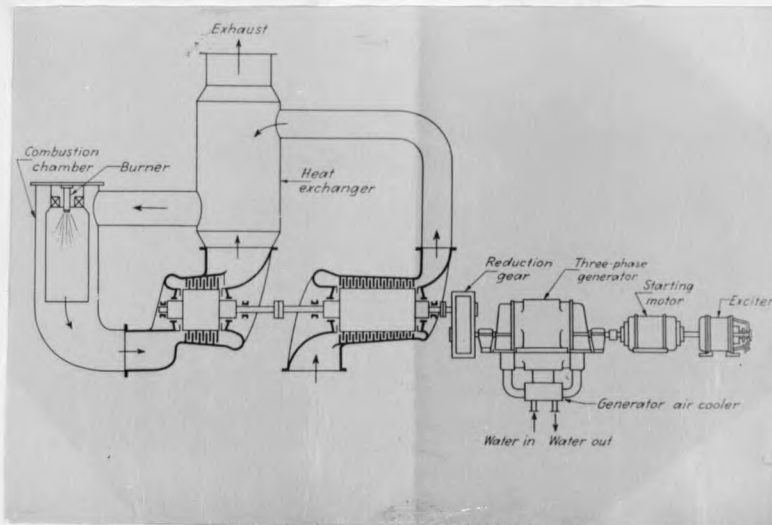


Fig.3- Open Cycle Gas Turbine with Heat Exchanger
(from Power, Sept. 1944)

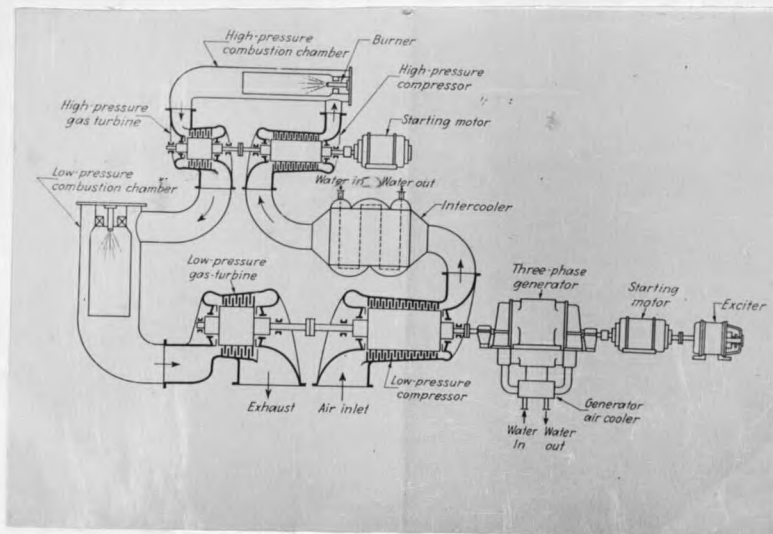


Fig.4- Open Cycle Gas Turbine in Two Shaft arrangement
with Intercooling and Reheating
(from Power, Sept., 1944)

METHODS FOR IMPROVING THE EFFICIENCY OF GAS TURBINES

&

RECENT RESEARCH CONDUCTED ON THOSE

Open Cycle

The most common cycle which has received comparatively more attention than the other gas turbine cycles, is the open cycle gas turbine plant, and this because it is the simplest of all although it has a relatively low efficiency.

Previously it has been said that gas turbines for any set of conditions the thermal efficiency rises to an optimum value and then falls to zero with an increasing pressure ratio. Keeping the compressor ratio constant, a fact which is dependent on compressors, we can raise the efficiency using extremely high temperatures. However, there are three practical ways of improving greatly the gas cycle efficiency, namely, regenerating, intercooling¹ and reheating².

The regenerating gas cycle is one in which a heat exchanger (regenerator) transfers some of the heat from the relatively hot exhaust gases leaving the turbine to the air before it enters the combustor. (Fig. 3) Heating the air by exhaust gases reduces fuel consumption and improves cycle efficiency. The amount of heat obtained from the exhaust gases depends on the size of heat exchanger surface. Calculations indicate that the

1- F.K. Fischer & C.A.Meyer, Combustion, May 1944

2- L.N. Rowlwy & B.G.A.Skrotzki, Power, Oct. 1946

economic size of the heat exchanger will limit the regenerating cycle . at 1200 F inlet temperature, to approximately 75 % recovery of the heat available from the turbine exhaust gases, i.e. (75 % effectiveness) This economic size will be about 0.30 cub.ft. of heat exchanger volume per kilowatt of capacity ^I

Efficiency is further improved when intercooling is added to regenerator as shown in (Fig. 4). As the name implies the intercooling removes the heat of compression from the air passing through the compressor. Water circulating through the intercooler, cools the air, and since the colder air has smaller volume, the compressor work is reduced. This is a well known fact in multicylinder compressors to attain isothermal compression instead of adiabatic and hence consume less work in compressing the air. Other conditions remaining the same, one stage of intercooling will reduce the compressor work by some 15 % This increases the portion of the turbine capacity available as useful output and improves the cycle efficiency. Of course a large number of intercoolers is ideal but probably only a few stages will be practical.

The third method of improving efficiency, the reheating consists of adding heat to the gas as it passes through the turbine (Fig. 5) The gas turbine reheat cycle is the same prin-

^I- F.K. Fischer & C.A. Meyer, The Combustion Gas Turbine Cycle, Westinghouse Engineer, May, 1944

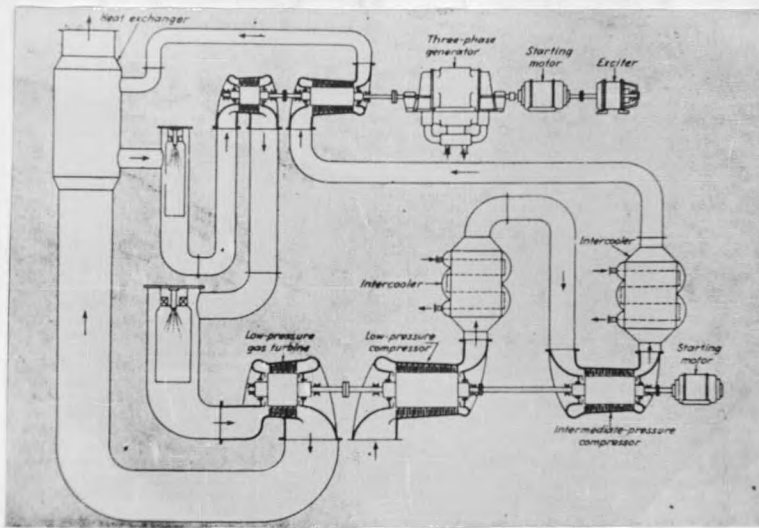


Fig.5- Open Cycle Gas Turbine Plant with Intercoolers
Heat Exchangers and Reaheaters
(from Power, Sept.1944,)

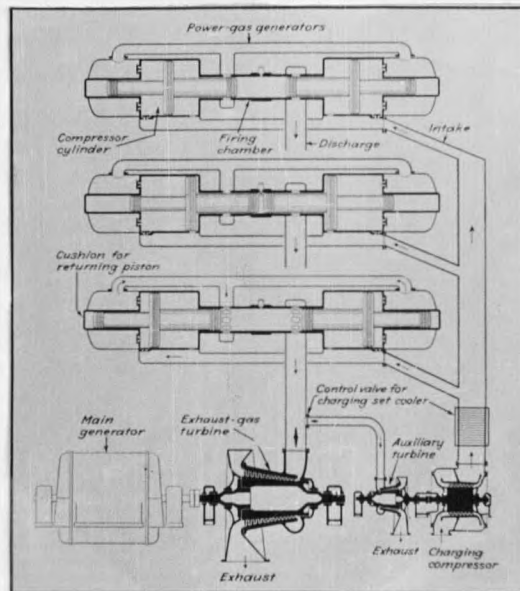


Fig.6- Power Gas Cycle Gas Turbine
(from Power, Nov. 1945)

ciple as the reheat cycle used in many steam plants, but in practice it will bear little resemblance. Reheating in the gas turbine will consist of burning fuel directly in the gas passing through the turbine. Here again the practical number of reheats is limited.

Reheating and intercooling increase the amount of useful energy per pound of working gas passing through the system, thus reducing the number of pounds of working medium circulated. Therefore the size of piping and blade path in the compressor and turbine is reduced.

Closed Cycle

The closed cycle gas turbine plant is the most promising for stationary use due to the high efficiency which it offers using more complicated hookups. Here again as in the open cycle plant we might have intercoolers, regenerators and reheaters and of course all of them being employed outside the working medium. The closed cycle offers the advantage that it has good part load efficiency compared to the open cycle. In the later fluctuations in load are taken care of direct fuel control which means, variable temperature, constant flow operation which has the merit of simplicity but gives poor part load characteristics. In the closed cycle, constant temperature operation can be attained by adjusting working medium density to meet load changes. The mechanism of control is well explained in a paper

by Dr. C. Keller¹. When the load of the plant drops, the working medium i.e. usually air, must be discharged from the cycle (Fig. I) while with increasing load, the working medium has to be admitted to the cycle. A theoretical discussion of this principle is given by F. Saltzmann². In the original operating condition , and in the operating condition which is sought after a change in load has taken place, all pressures in the cycle are in the same ratio to one another. During the transition period, however, matters are different because the pressures do not change with equal rapidity at all points of the cycle. It is those transition periods the paper deals with. A complete consideration of these theories is beyond the scope of this discussion.

A 2000 Kw plant is operating at Zurich , Switzerland with pressure ratio 3-4 and maximum pressure 400 - 500 psi. Closed cycle although it has many similarities compared with steam plant, it has the following advantages over it.

- a- Flexibility of equipment placement allowing compact arrangements with minimum pressure and temperature losses.

1- Dr. C. Keller, Closed Cycle gas turbine looks for further development, Power, January 1946

Dr. C. Keller, The Escher Wyss-Ak Closed Cycle Turbine, Trans. of the A.S.M.E. Novem. 1946

2- F. Saltzmann, Regulation Theory for Thermal Power Plants employing Closed Gas Cycle, A.S.M.E. Tran. May, 1947

b- Limited number of auxiliaries simplifying operating requirements.

c- Smaller cooling water needs, those being only 10-20% of comparable steam station requirements.

d- Absence of fittings for very high pressures.

Designs for 12,000 and 25,000 Kw plants have been projected^I.

Such projects indicate that the length of plants is not greatly influenced by the capacity. Increase in capacity is achieved principally by enlarging the diameters of the equipment components.

Semiclosed Cycle

Several possible arrangements of the combustion cycle have been investigated with three objectives.

a- To obtain full load efficiency substantially better than that of modern steam turbines

b- To develop a marine propulsion unit of substantially constant efficiency at part loads at reduced speeds when geared to a propeller

c- To achieve the maximum in compactness for marine service.

For the second of these objectives, it is essential that the quantity of air flow be reduced for part loads. Best efficiencies are realized if operating temperature remains constant as

I- Power, Novem. 1945 p. 66

load varies. The semiclosed Combustion turbine cycle (Fig. 2) has many points of similarity to the power gas cycle, except that reciprocating machinery is supplanted with rotary equivalents ^I. The operation of that cycle is as follows; Air enters the inner circuit at 60 psi just ahead of the precooler joining with the main flow of air returned from the regenerative heat exchanger. (Fig. 2) Total flow is compressed to 240 psi in two stages with intercooling provided. After compression the air is heated in a regenerator to the highest temperature economically practicable and divided automatically into two streams. One part is led to the combustion chamber, heated to a high temperature by direct combustion of oil, cooled to about 1200 F in a separate heat exchanger and passed to the output turbine at about 240 psi. Remainder of air flow is heated to 1200 F by transfer through the tubes of heat exchanger and then expanded in a turbine to 60 psi to drive the three compressor stages.

Only relatively clean air circulates within the inner circuit, the products of combustion passing through only one heat exchanger and the output turbine. The sole parts subjected to very high temperatures are the combustion chamber itself and the heat exchanger. All the foregoing pressures and temperatures are stated for full load conditions. As less fuel is fired,

I- S.A.Tucker, Free- Piston Compressor Work Leads To New Semiclosed Cycle development.

temperature of air within the closed part of the cycle falls slightly, reducing the compressor drive turbine speed and consequently the quantity of air circulating. Reduced air flow restores equilibrium conditions at about the same temperature as at full load but with reduced pressure throughout the cycle. For starting, the compressor is brought up to moderate speed by an electric motor. Addition of fuel in the combustion chamber result in a net excess of power available to bring the entire unit up to speed.

At 1946 an output turbine with associated combustion chamber was under test at Winterthur, Switzerland, and compared in performance with steam plants of various types.

Power- Gas Cycle

Some ten years ago Sulzer Bros, Switzerland began to aim its research program at the most promising fields of application of exhaust turbines used in conjunction with reciprocating machines as well as continuous combustion gas turbines. As the supercharging pressure^I of reciprocating engines is increased, above 30 to 45 psi now commercially developed in Europe, the proportion of work done by the exhaust gas turbine rises until at about 75 to 90 psi the engine output is just sufficient to drive the compressor, and the whole useful output is produced at the tur-

I- See footnote on previous page

bine shaft. Under these conditions, the free piston compressor unit proves a most satisfactory machine for producing the power gas cycle

A comprehensive explanation of the cycle is given by Dr. Adolph Meyer.^I The free or floating piston machine as built by Junkers in Germany, Pescara in France, and Sulzer Bros in Switzerland has been successfully developed for this purpose by the last named firm. Three floating piston engines (Fig. 6) combined with two gas turbines and two axial compressors give a net output of 7,000 Hp. The smaller turboset is used for supercharging the piston compressors which provide the air, further compressed for the opposed floating piston Diesel engines. The exhaust gases of these engines flow to the main gas turbine at the left , which produces the net output. Such a set may give an overall efficiency of more than 40 % as compared with 34 % which Brown Boveri guaranteed on a two stage 27,000 Kw set designed and built for a Swiss power Station. The last unit is under construction and it will have the highest efficiency ever obtained from a gas turbine set. The Sulzer high pressure set profits from the high efficiency of the Diesel drive which is due to the high initial temperature with which the Diesel process works , inherently has the disadvantages of a reciprocating engine.

I- Dr. A. Meyer, Recent Developments in Gas Turbines, Mech. Engineering, April 1947, p.273

Several attempts have been made to obtain similar conditions in a rotating machine. Brown Boveri some 20 years ago obtained a patent but it proved to be of low efficiency. A much more interesting machine of this kind called "Compres" has been incorporated in the design of the power set of a new gas turbine locomotive which is now on the test bed of Brown Boveri in Switzerland.

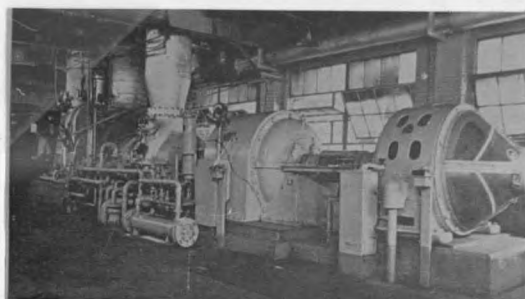


FIG. 1 A SUN OIL COMPANY 40,000-CFM TURBOCOMPRESSOR INSTALLATION WITH MOTOR STARTER

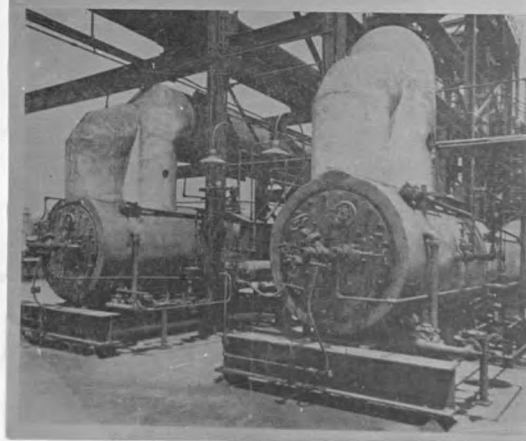


Fig.7- Houdry Units
(from Mech.Eng.Sept.1945 p.594)

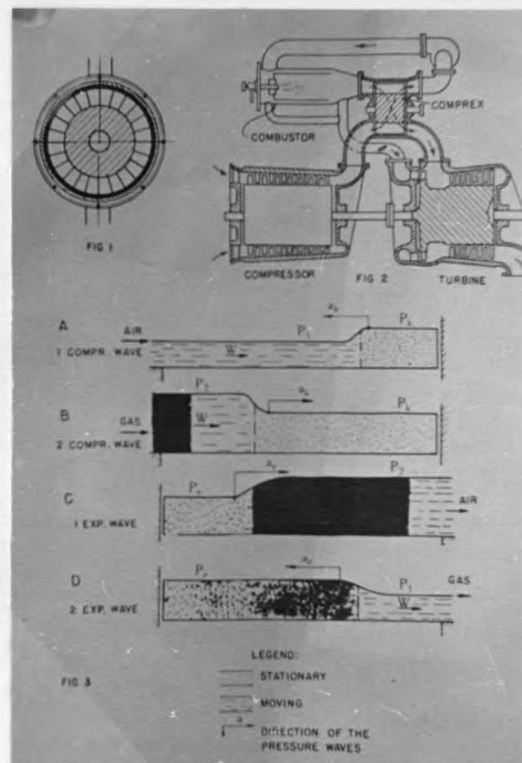


Fig.8- Principle of Compress
Pressure Exchanger
(from P.P.Eng.July 1947,p.1280)

THE COMPREX

The comprex was invented in 1940 by Claude Seippel then chief of gas turbine department of Brown Boveri. The name comprex originates from the fact that the same rotating machine compresses air and expands gas, having the same merits as a Diesel engine; to be subjected to the average temperatures existing between the gases to be compressed and to be expanded. The flow of air and gas through the gas turbine cycle using Comprex is as follows:

From the outlet of the axial flow compressor (Fig. 8) the air enters the bottom cell of the comprex and while its rotor turns one half revolution, the air enclosed in this cell has reached due to pneumatic ram a substantially higher pressure. It is then exhausted or pushed out through the circulating fan which is designed to overcome friction losses between the comprex and the combustion chamber. Entering the combustion chamber and being heated by burning fuel, the air and gas mixture again reaches the top cell of the comprex and while this cell makes one half turn to the bottom, the pressure is reduced by expansion in the enclosed cell, whereupon the gas enters the reaction section of the gas turbine and expands further down to atmosphere.

The closing and opening of the cells is produced by their rotation between the fixed shields of the comprex which are pro-

vided with two openings not in line, on each side through which air and gas enter and escape.

A complete explanation of the principle is given by Paul R. Sidler,¹ president, Brown Boveri Corp. New York and by Dr. Adolph Meyer², Chief Consulting Engineer of the same company in Switzerland.

1- P.R. Sidler, A pressure exchanger for gas turbines, Power Plant Eng. July 1947 p. 128

2- Dr. A. Meyer. Recent Developments in Gas Turbines, Mech. Eng. April 1947 p. 273

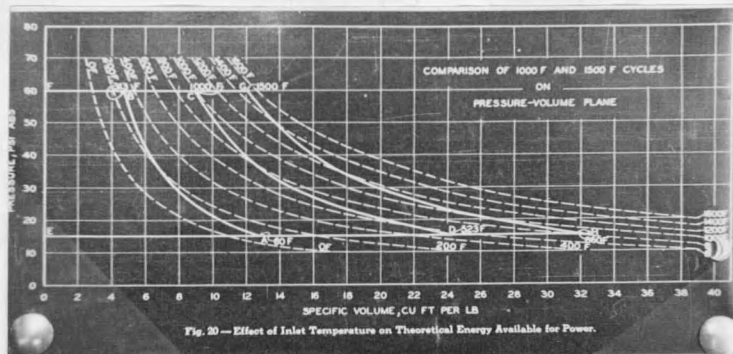


Fig.9- Effect of Inlet Temperature on Theoretical Energy Available for Power
(from Allis Chalmers Bul.No:7,R-62I3)

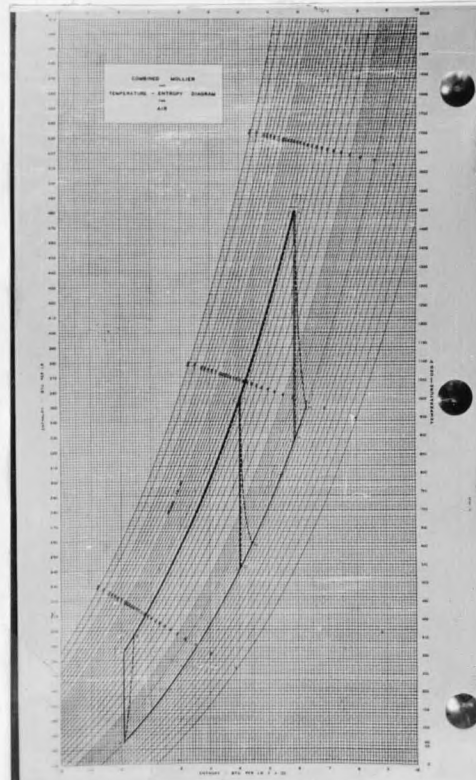


Fig.10- Mollier Diagram for Air
(from Allis Chalmers Bul. No; 7,R-62I3)

THE THERMODYNAMIC CONSIDERATIONS
OF GAS TURBINES

The thermodynamic cycle of Gas turbine power plant has been analyzed by various authors in detail and mathematical expressions have been developed. The most important of those are by Dr. J. T. Retallia, ¹ Dr. A. Stodola ² . Dr. J.I. Yellott ³ in his series of papers in Power, discusses gas turbine cycles in simple thermodynamic calculations without much mathematics in it.

It is beyond the scope of this discussion to include all those mathematical formulas together with their derivation and the method of their use. However, here is a simple thermodynamic consideration of the constant pressure, continuous combustion gas turbine cycle, which sometimes is called continuous Diesel cycle, or rotating Diesel engine.

In a steady flow reversible process, area FBAE (Fig. 9) represents the amount of energy theoretically required to compress one pound of air in the compressor, while the theoretical energy created by the expansion of one pound of gas in the tur-

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- 1- J.T. Retallia, Current Trends in Power Plant Development
Allis- Chalmers Bulletin Number: 7, 1942
J.T. Retallia, The Combustion Gas Turbine, Trans, of the
A.S.M.E. Febr. 1941 p.115
2- A. Stodola, Steam and Gas Turbines, vol. II McGraw-Hill
Book Comp. 1927 p1173- 1318
3- J.T. Yellott, Gas Turbine Power Plant Calculations, Power
Plant Eng. January 1947.

bine is shown by the area FCDE Therefore the difference between these two areas, ABCD, would represent the theoretical excess energy available for power purposes per pound of fuel.

It has been said that the greater the inlet temperature in the turbine wheel, the greater the energy available for power purposes. (Fig. 9) shows this effect for temperatures 1000 F and 1500 F. Here a pressure ratio of 4 has been used with air entering the compressor at 15 psia and 60 F.

(Fig. 10) shows the Gas cycle diagram on the combined S - T and H - T plane for air. The broken lines represent actual cycles in which losses have been incorporated, and the solid line indicates a theoretical cycle without losses. The thermal efficiency, that is, the ratio of the heat equivalents of the useful output and fuel supplied in a given time as affected by pressure ratio and turbine inlet temperature is given in (Fig. 12a). On (Fig. 12b) is shown the theoretical regenerative cycle with infinite surface heat exchanger i.e. all the available heat from the turbine exhaust gases have been consumed by the compressor exhaust air, on T - S and H - S plane. The shaded area KYDN, shows the amount of heat available, and area EBCC the amount of heat consumed. Of course in this case those two areas are equal considering we have infinite surface of heat exchanger.

The same diagram but this time with actual regenerative cycle is shown in (Fig. 13a). A totally actual case with

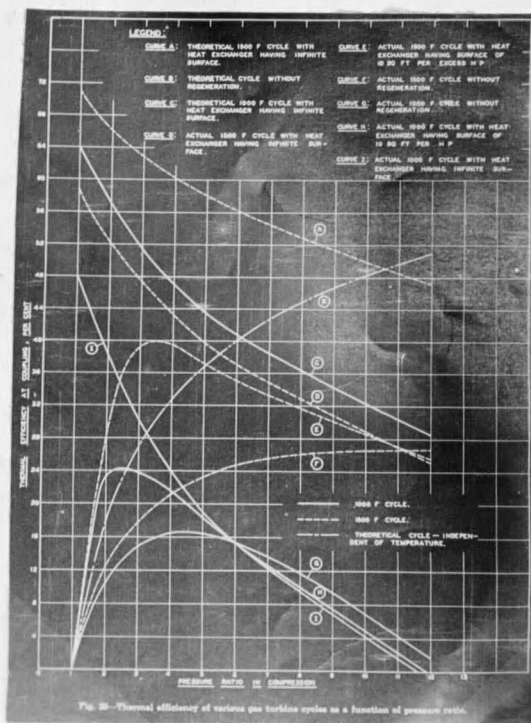


Fig.II- Thermal Efficiency of Various Gas Turbine Cycles as a Function of Pres.Ratio
(from Allis Chalmers Bul.Num.7,R-62I3)

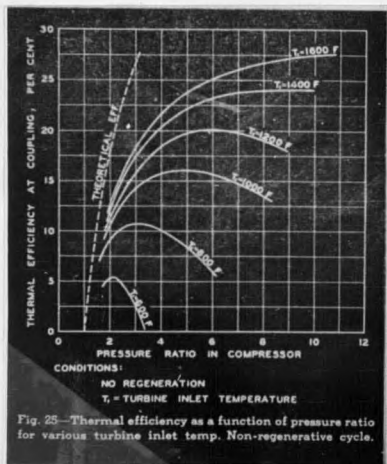


Fig. I2a- Thermal Efficiency as a function of Pr. Ratio

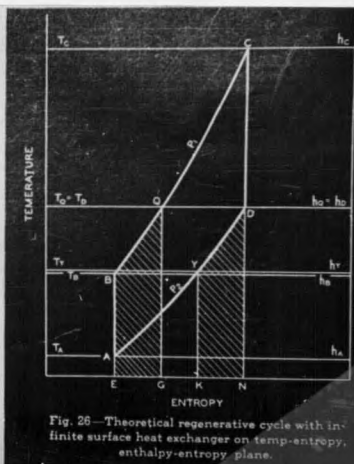


Fig. I2b- Theoretical regenerative cycle

(from Allis Chalmers Bul.Num.7, R-62I3)

actual regenerative cycle with finite surface heat exchanger on T - S and H - S plane is shown in (Fig.I3b)

In previous discussions it has been pointed out that the higher the pressure ratio used and the more elevated turbine inlet temperature is, the better the thermal efficiency of the gas turbine cycle. (Fig.I2a) shows the effect of different pressure ratios and turbine inlet temperatures on thermal efficiencies.

Of course high pressure ratios are attained with modern air compressors developed quite recently, and higher temperatures with new alloys which permit an increase of gas cycle efficiency reaching that of Diesel engine.

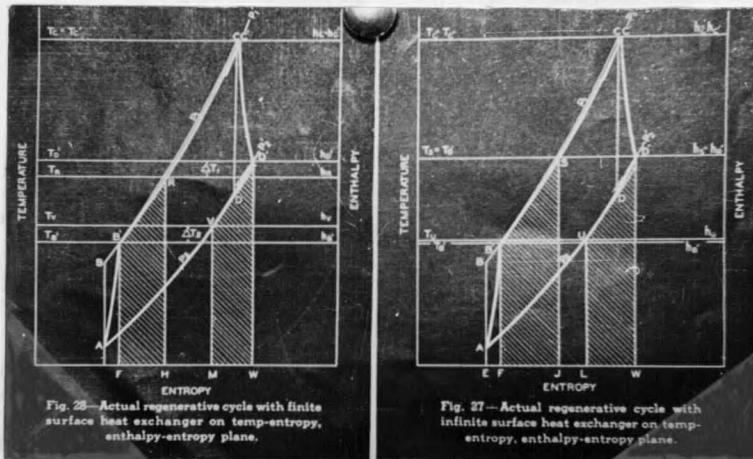


Fig. I3a Fig. I3b
(from Allis Chalmers Bul. No: 7, R-62I3)

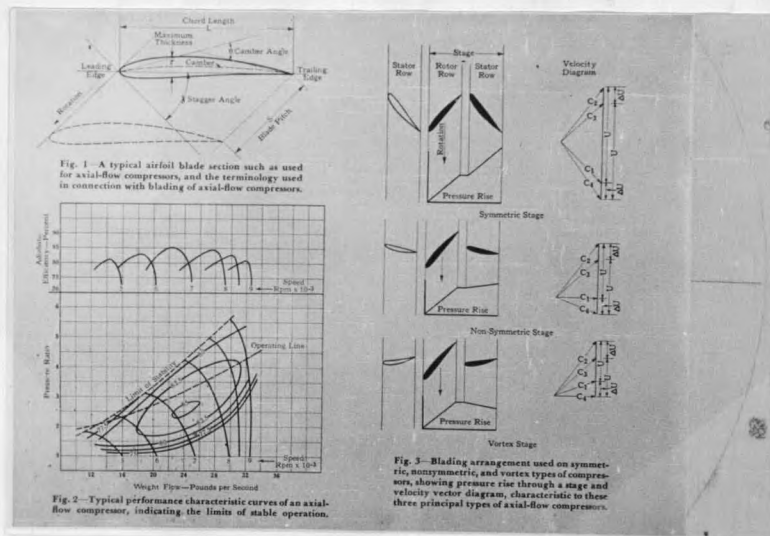


Fig. I4- Characteristics of Axial Flow Compressor
(from Allis Chalmers Bul. No: 7, R-62I3)

COMPRESSORS USED IN GAS TURBINE CONSTRUCTION

The recent development of gas turbine cycle is primarily due to the efforts of the aerodynamists to improve the axial flow compressor. But before discussing this one in detail, let us see the other existing types of compressors and their capacity to be used in connection with the gas turbines.

Since gas turbines require continuously large amount of air flow, in a relatively small compression ratio, 4 - 6, the positive displacement piston type compressors can not be used in gas turbine cycle, not satisfying the above requirements except that it can create large pressure ratios. The other existing types are:

Centrifugal Compressor

The centrifugal compressors have been and still are widely used in aircraft gas turbines and turbosuperchargers. The first centrifugal compressor ever built is that of Switzerland in 1906^I. Work on centrifugal blowers, for which many applications were found, led to the development of superchargers for four cycle diesel engines, consisting of a single stage blower wheel, driven by a single stage exhaust gas turbine on the same shaft. Those may be considered as forerunners of modern po-

I- P.R.Sidler, Turbines for Blast Furnace Blowers, Iron and Steel Engineer, April 1945

wer gas turbines and have indeed contributed to the body of experience, necessary for successful gas turbine work. The centrifugal compressors work as follows:

Air trapped between rotor blades is thrown out past blade tips by centrifugal force. A stationary diffuser closely fitted about the rotor converts the velocity energy into pressure by deceleration. The vacuum that tends to form causes atmospheric air to enter the rotor to replace air thrown out by centrifugal action, thus maintaining air flow. The maximum pressure ratio attainable with centrifugal compressors is about 3-4 with comparatively low volume flow.

Lysholm Compressor

The recently introduced Swedish Lysholm compressor ¹ is a rotary lobe machine which can compress air before discharging it. It is a positive displacement, highly efficient compressor and is stable throughout its operating range ², although it has a low pressure ratio ³ of about 2 - 3. Its pressure is independent of speed and is suitable for direct coupling to the turbine shaft.

The Elliott Company built a gas turbine arrangement with

1- Alf Lysholm, Chief engineer, Aktiebolaget Ljungstroms Angturbin, Sweden, after which the compressor was named

2- R.T. Sawyer, The Modern Gas Turbine, Prentice-Hall Inc. 1945, p II6

3- A. Lysholm & R.B. Smith, The Elliott - Lysholm Supercharger, Paper Presented at Meeting of A.S.M.E. in Detroit Mich. January 14, 1943

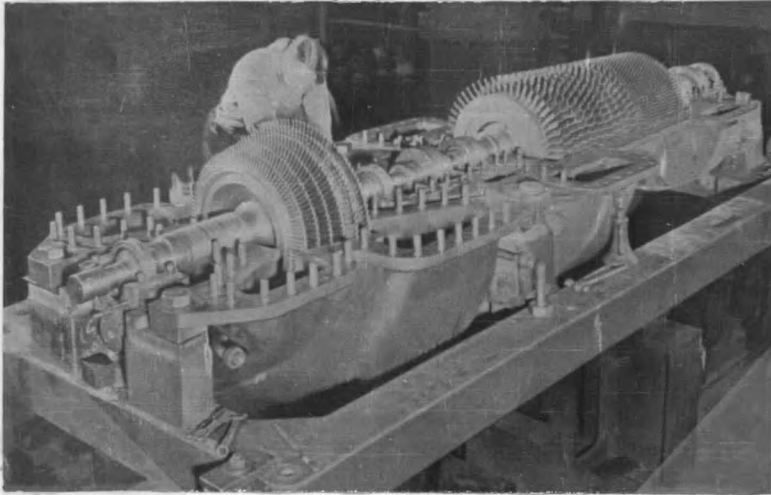


Fig.I5- Axial Flow Compressor with Top Casing Removed
(from Trans. A.S.M.E., Feb. 1941 p.115)

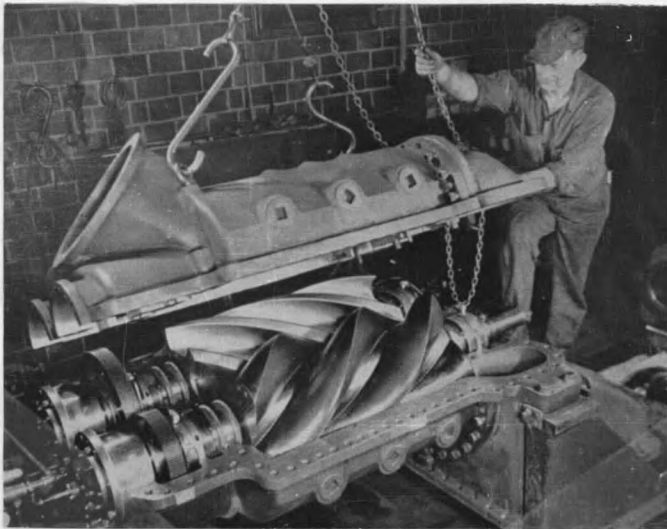


Fig.I6- Elliott-Lysholm Compressor
(from Powerfax by Elliott Co. Autumn 1945)

Lysholm Compressors for Marine Service. The adiabatic efficiency of the Lysholm compares favorably with the axial compressor. Several small units of 400-500 e.p.m. have been built by the same company to be used as superchargers of four cycle Diesel engines. They run at rotative speeds of about 2000-3000 rpm and they can supercharge the biggest Diesel engines that have been designed. Two-stage units operating at discharge pressures as high as 100 psi, have also been built. Light-weight units for aircraft service are also feasible, and are, in fact, in the experimental stage of operation abroad. See (Fig.16) for general arrangement of Lysholm Compressor.

Axial - Flow Compressor

The most promising type of compressor for gas turbine cycle use is the Axial-flow Compressor. Essentially it is a "turbine" driven by external means which increases the pressure of the fluid supplied to it, rather than decreasing it, as in the case of an ordinary steam turbine. The designers of axial-flow compressors used all the information developed by aerodynamists in the fluid dynamics field. The Brown Bover Company¹ pioneered in the development of this

I- J. K. Salisbury, The Basic Gas Turbine Plant and Some of its Variants, Paper Presented by the Author at a meeting of A.S.M.E. at Tulsa, Oklahoma, May 8, 1944.

modern kinetic-type compressor and has made many useful contributions to the design of such machines. In 1928, Westinghouse¹ applied an axial-flow blower to the ventilation of a large turbine generator at the Hell Gate Station of the Consolidated Edison Company. Velocity diagrams may be drawn for the axial-flow compressor stage in much the same manner as they are drawn for a steam-turbine stage. (Fig. I4) shows the blade positions together with the velocity diagrams of the existing three types of axial-flow compressors, namely symmetric, non-symmetric and Vortex stage.

It is particularly interesting to note that the blade section of an axial-flow compressor (Fig. I4) are similar to those used in conventional airfoils. This section, when arranged in a grid similar to turbine nozzles, possesses the property of deflecting through a small angle the fluid which passes through it. This property is utilized in increasing the pressure of the fluid. Above it has been said that axial-flow compressor resembles a steam turbine. However, the flow path of an axial-flow compressor decreases in area in the direction of the flow to accommodate the diminishing volume as the compression progresses from stage to stage. In entering a blade row, the gas flowing in a generally axial

¹- A. I. Ponomareff, Principles of the Axial-Flow Compressor, March, 1947 issue, Westinghouse Engineer.

direction is deflected through a small angle in the direction of rotation. This change in direction of flow is accompanied by a decrease in relative velocity with resultant pressure rise through diffusion. The change in the tangential component of air velocity, when multiplied by the blade velocity at the same point, represents the change in momentum and is proportional to the power input to the compressor. In the mechanism of a pressure rise in an axial-flow compressor through diffusion unassisted by centrifugal force, lies the principal difference between the axial-flow and centrifugal type of compressors. The flow pattern in an axial flow compressor, through a series of diffusing passages formed by a proper arrangement of blades in each row, differs radically from that found in the reaction steam turbine, where the flow occurs in the direction of the pressure gradient. This fundamental difference in flow explains why the development of the axial flow compressor did not parallel that of the steam turbine.

(Fig. I4) shows the performance characteristics of an axial-flow compressor. It will be noted that it is essentially a constant-volume-flow device at any given speed. Increase in the discharge pressure causes only a slight reduction in the volume flow passed by the compressor. Eventually, however, increase in the discharge pressure causes excessively high "lift" of the airfoil sections and results in

a sudden breaking down of the diffusing properties. At this time, the compressor is likely to become unstable, and destructive vibratory stresses may be set up.

As it has been mentioned above, there are three types of axial-flow compressors. The symmetrical-stage uses blade tip velocities exceeding the velocity of sound, and axial velocities up to 600 feet per second. The pressure rise in both rotating and stationary runs results in fewer stages for a given compression ratio. High axial and blade velocities lead to small physical dimensions and high rotative speeds. A $7\frac{1}{2}$ inch diameter, jet propulsion compressor is designed for 5000 cfm of atmospheric air at a pressure ratio of three when operating at 34,000 rpm.

The non-symmetric type of axial flow compressor uses blade tip speeds up to 250 feet per second and axial velocities up to 400 feet per second. It has an efficiency of about 70% compared to 85% of symmetrical. Compressors built by Brown Boveri in oil refineries are of this type.

The third type is the vortex stage compressor because the air acquires a whirling velocity before the rotating blades. Air velocities of about 200 feet per second and blade tip speeds as low as 450 feet per second are used in this type, resulting in relatively low operating speeds and large physical dimensions. Eseler-Wyss advocates the vortex type air compressor for a closed cycle gas turbine plant.

As a conclusion, we might say that the axial flow compressor is ideally suitable for gas turbine plant operation, and especially for use in aviation due to its small frontal area. Due to unstable characteristics at high pressure rise per stage, there is a need for multistage compressors even for moderate pressure ratios. However, in contrast to the multiple stage centrifugal unit, staging of the axial flow compressor does not involve any appreciable losses. The only disadvantage of the axial flow compressor being that it can not handle efficiently small volumes, say $500 \text{ ft.}^3/\text{min.}$, especially at high pressure ratios. The centrifugal might be used for low air flow with moderate pressure ratio, and in case of high pressure ratios, the positive displacement one, such as the piston or rotary type (Lysholm) compressor should be used.

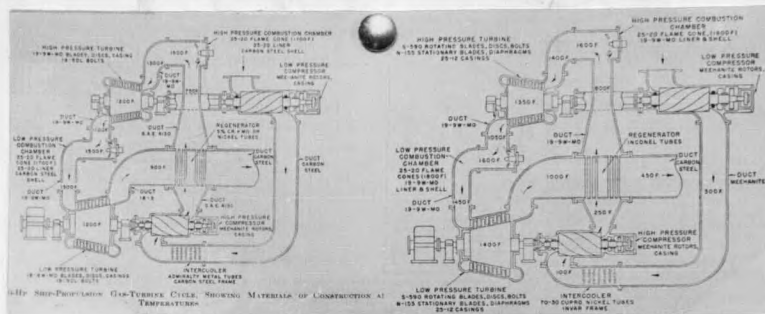


Fig. 17-Elliott 2500 and 3000 Hp. Ship Propulsion Gas Turbines (from Trans. A. S. M. E., Aug. 1947)

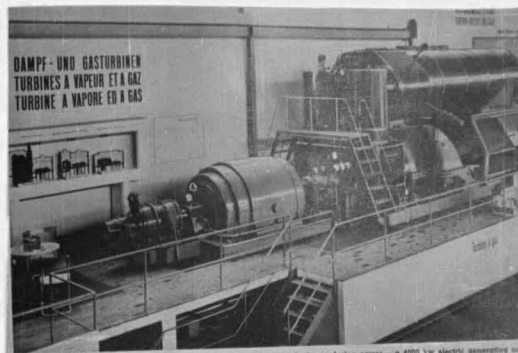


Fig. 31 - The first modern combustion gas turbine designed primarily for producing power - a 4000 hp electric generating unit.

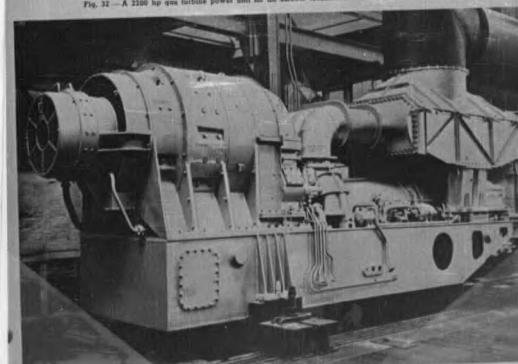


Fig. 32 - A 2200 hp gas turbine power unit for an electric locomotive for the Swiss Federal Railways.

Fig. 18-The First Modern Gas Turbine (from Brown Boveri Bul. Switzerland, No: I584E-II.9(XI.46)LI044)

MATERIALS USED IN GAS TURBINES

It has been said that the present successful development of gas turbines is primarily due to the combined progress in the aerodynamical and metallurgical fields. The first contributed in the development of the axial flow compressor and the second in the advancement of high temperature stress resisting alloys. The August, 1947 issue of the Transactions of the A.S.M.E. is full of descriptions and of constituents of new alloys found especially during the second World War and secrecy restrictions on the so-called "super" heat-resisting alloys were lifted early in 1946. It is not the purpose of this discussion to give detailed information on all new alloys. However, the most important of those will be mentioned emphasizing their primary constituents. (Fig. 17) shows the diagram ¹ of a 2500 Hp and a 3000 Hp gas turbine ship propulsion cycles showing the materials of construction and operating temperatures. The 2500 HP unit was essentially completed in 1943 and is based on 100,000 hour life. After extensive tests, it was accepted by the United States Navy in December, 1944, the first complete gas turbine power plant to operate successfully in the United States. This plant developed a thermal efficiency of 29%. It will be noted that

1- C. T. Evans, Materials for Power Gas Turbines, Transactions of the A.S.M.E., August, 1947, Elliott Gas Turbine.

19-9 W-Mo was used throughout the turbines except for the bolts which are 19-9. A 25-20 alloy was used throughout the combustion chambers for the cone and liners, details of which are shown in (Fig.20). The use of high temperature materials creates so many manufacturing problems that the only possibility of successful construction comes through extremely close cooperation of the manufacturing design departments. Castings, though simple and convenient to design, are hard to produce in high temperature alloys and do not have the high temperature properties of rolled or forged material of the same analysis. Because riveted joints depend primarily on tension in the rivet, due to the high temperatures used, which causes elongation in the rivets, they cannot be used except in minor attachments. In general, the only recourse in building such machines as this is to use rolled plate and arc welding, and by this method to fabricate many pieces into one permanent, single assembly. This method of fabrication was used in all of the duct work and the combustion chambers in this gas turbine plant. (Fig. 17) shows also the diagram of the 3000 Hp ship propulsion gas turbine. Three plants are at present under construction in Elliott's Comp. shops; two for the Navy Department, Bureau of Ships, and one for the Maritime Commission. It will be noted that a wider variety of materials will be present in the new turbines. There are several reasons for this, including

TABLE I TYPICAL ANALYSES OF ALLOYS USED IN
ELLIOTT POWER GAS TURBINES

	19-9 W-Mo	19-9 DL	19-9 WX	Ni Re- sist	18-8	18-8 Colum	Cast 25-20	Wrought 25-20	S-590	Low C N-155	Cast 25-12	Inco- nel
C	0.10	0.30	0.10	3.00	0.07	0.07	0.35	0.12	0.40	0.15	0.35	0.09
Mn	0.60	0.75	1.50	1.25	1.00	1.00	0.75	1.25	0.75	1.00	1.00	0.75
Si	0.50	0.50	0.50	1.50	0.50	0.50	1.00	1.75	0.65	0.50	1.25	0.30
Cr	19.00	19.00	21.00	2.00	18.50	18.50	25.00	25.00	20.00	21.00	24.00	13.50
Ni	9.00	9.00	9.00	14.00	9.50	11.00	20.50	20.50	20.00	20.00	14.00	78.00
Co	----	----	----	----	----	----	----	----	20.00	20.00	----	----
Mo	0.40	1.25	0.50	----	----	----	----	----	4.00	3.00	----	----
W	1.25	1.25	1.50	----	----	----	----	----	4.00	2.00	----	----
Cb	0.40	0.40	1.15	----	----	10 X C min.	----	----	4.00	1.00	----	----
Ti	0.35	0.35	0.20	----	----	----	----	----	----	----	----	----
N	----	----	----	----	----	----	----	----	----	0.12	0.12	----

Oth-
ers (Iron Remainder) Cu = 6.25

material availability and cost.

(Table I) gives the typical analysis of alloys used in Elliott Power gas turbines. ¹

1- For more information on Gas Turbine Materials see Literature Listed at the End of Thesis.

LATEST DEVELOPMENT OF GAS

TURBINE CYCLE

1. In The United States of America

Here in the United States the development and the manufacturing of gas turbines has been undertaken by Allis-Chalmers, Elliott, General Electric and Westinghouse Companies.

a. Allis Chalmers

During the last war years, the Allis-Chalmers Manufacturing Company was licensed to produce Brown Bover types of gas turbines for the Houdry Cracking process. Still many of those are in successful service including a 60,000 cfm unit.

Since 1941, the Navy has been engaged in the development of the gas turbine power plant with Allis Chalmers and other commercial firms. This gas turbine was of experimental nature and it was rated to 3500 Hp with the first test running at 1350° F. and later tests at 1500° F were planned. This unit was installed at the Naval Experiment Station at Annapolis since 1944, but for reasons of safety, it was disclosed until late in 1946. This unit was designed to provide information for designers at high inlet temperature from 1350° F. to 1600° F. The cycle selected for this experimental unit involves the use of a parallel turbine arrangement, one turbine driving the compressor and the other the propeller through electric transmission. The compressor

Aside from that, the same company in 1944 started the design of a 4800 hp gas turbine driven locomotive as discussed by J. T. Rettaliata.¹ The load is divided in two 2400 hp and it is coupled to two generators. The type used is almost the same as those of Houdry Oil Refinery process.

Another important construction is the 4000 hp coal burning locomotive being almost completed in Allis Chalmers Shop.² It consists of 21-stage axial compressor discharging through a regenerator at a maximum pressure of 25 pounds to the combustion and fly-ash separation system.

b. The Elliott Company

The Elliott Company together with the Allis-Chalmers is associated in building a 3750 coal Burning locomotive. This unit is built for the Baldwin Locomotive Works, while the one of Allis Chalmers is for the American Locomotive Company. Also, the Elliott Company built a 2500 and a 3000 hp ship propulsion plant details of which are given in the chapter on Materials for Gas Turbines. The Elliott Company is characterized by us-

1- Rettaliata, J. T., Gas Turbine Road Locomotive, Mechanical Engineering, November, 1944, p. 697.

2- See page () for further discussion of this design.

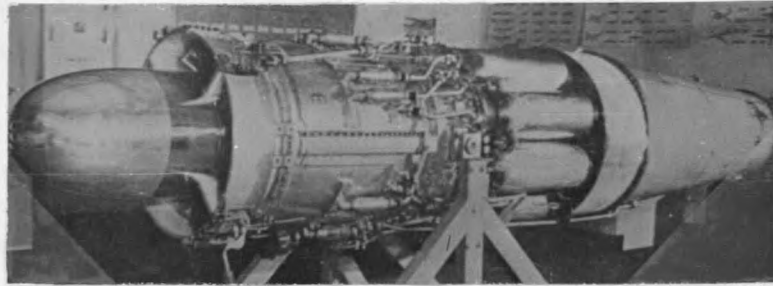


Fig.19-General Electric, T.G.-180,Aircraft Gas Turb.)
(from Power, August 1947 p.68)

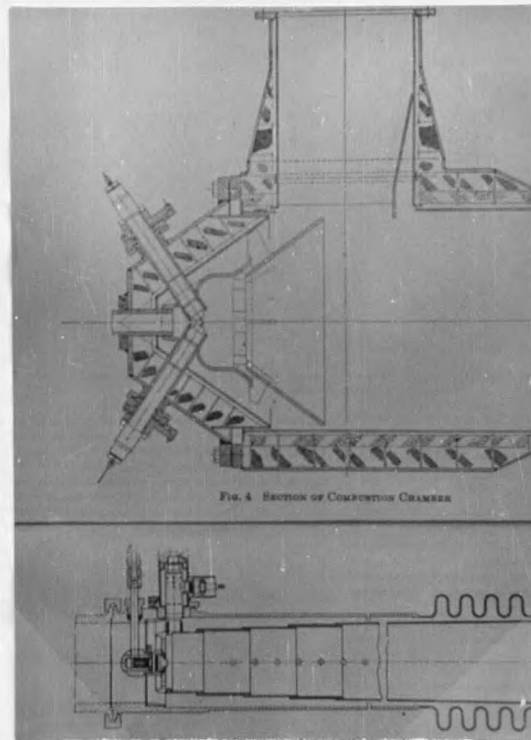


Fig.20- Combustion Chambers
(from Trans.A.S.M.E.p.603)

ing Lysholm Type compressors for both the 2500 Hp and the 3000 Hp marine service gas turbines. (Fig, I7)

c. General Electric

The General Electric Company has important contributions to the development of gas turbines as a prime mover. Shortly before Pearl Harbor in 1941 , the United States Army Air Forces engaged the G.E. Company to design, construct and test gas turbines for propeller drive, after the British Whittle Engine. The power plant then designed was G.E type T. G. - I00 now known as XT- 3I. Fifty of these were mounted on the Vultee Xp-8I assisted by I-40 jet engine.^I

General characteristics of the T.G. -I00 are that it has I4 stage compressor, runs at I3,000 rpm and drives the propeller at II45 rpm and the thrust at 2400 Hp including the effect of the jet thrust, if the weight is I975 lbs.or less than I lb/hp. (Fig.I9) shows the T.G. -I80 type. The jet engine diagram used with these gas turbine propeller drives is shown in (Fig.2I) Although details are still restricted , it has been described as having I2 stage axial compressor with a pressure ratio of 7;I. The turbine is designed for I350 F inlet temperature and it is expected that this will be raised.

I- Howard Alan, An Aircraft Gas Turbine for Propeller Drive
Mechanical Engineering, October 1947, p.827

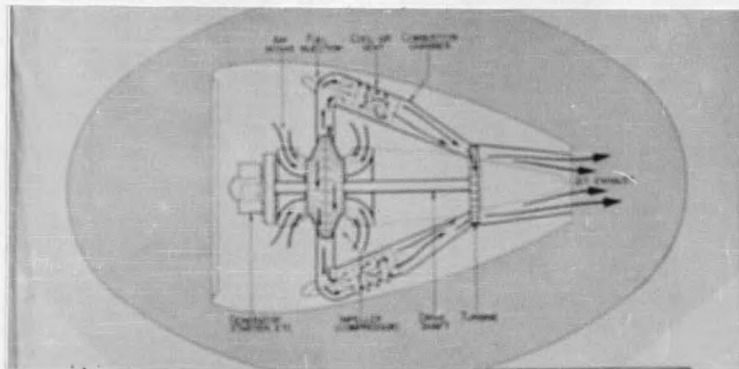


Fig.21- General Electric Jet Engine
(from Power, May 1945)

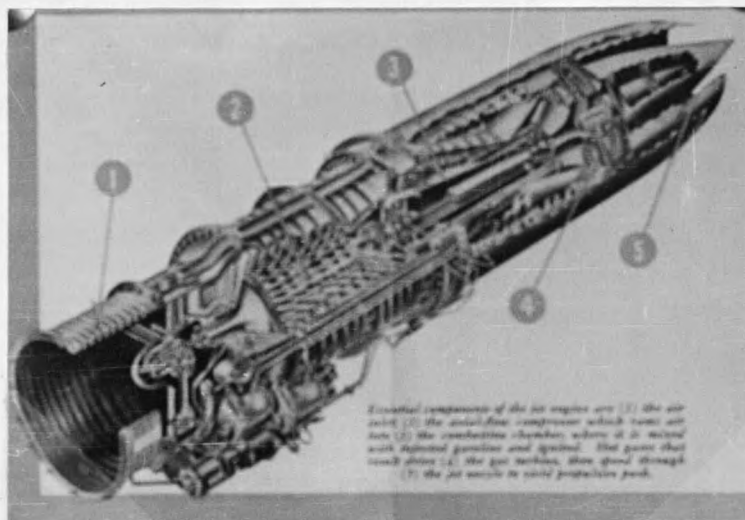


Fig. 22- Westinghouse Jet Engine
(from Westing. Bul.Aug.1947,B-3974)

The General Electric Company, before the war, undertook the development of gas turbine power plants, but their work was delayed due to the incoming war. The company's first post war design is a plant for locomotive and other applications. General arrangement is similar to that of TG-180 aircraft engine shown in (Fig. 19). It is rated at 4800 hp, runs at 6700 rpm with 1400° F turbine inlet temperature.¹ It has 15 stage axial flow compressor with 70,000 cfm capacity through a pressure ratio of about 6:1, working on the vortex principle. The turbine has an overall efficiency of 17%.

d. Westinghouse Corporation

As far as the Westinghouse Corporation is concerned, it has contributed to a considerable extent to the advancement of the gas turbine in this country. Late in August, 1941, it was disclosed that the Navy's highest performance military aircraft, "Banshee", making 600 mph, was equipped with Westinghouse jet propulsion machines. The Pirate has a single J-34 engine in the fuselage, with an overall diameter of 24 inches. (Fig. 22) shows the general arrangement of the Westinghouse jet-gas turbine

1- Howard, Alan, Design Features of a 4800 hp Locomotive, Gas-Turbine Power Plant, Mechanical Engineering, April, 1948, p. 301.

power plant. Another type of 19 inches in diameter is installed on the Phantom, giving it a net thrust of 1600 lbs.

Aside from these, in 1943 Westinghouse decided to build an experimental gas turbine of 2000 hp, working on the simple cycle. (Fig. 25) shows the turbine on the test bed. The compact unit lends itself to a narrow in-line arrangement particularly desirable for locomotive use. Its general characteristics are a 25,000 cfm capacity axial flow compressor in 20 stages with 5:1 pressure ratio. Up to March, 1948, it had run 1000 hours successfully. The turbine is designed for an inlet temperature of 1350° F and it has a thermal efficiency of 16.7%. Work on this turbine is still going on according to T. J. Putj¹, Manager of Gas Turbine Engineering, Westinghouse Corporation.

2. In Europe

Abroad, Switzerland, England and Germany are the countries where gas turbines have been taken into consideration. England and Germany, during the World War years, developed gas turbines primarily for aviation use. The British "Meteor"

1- Putj, T. J., The 2000 Hp Gas Turbine on Trial, Westinghouse Engineering, March, 1948, p. 39. 1350° F, 2000 Hp Gas Turbine Runs on Test Stand for 1600 Hours, Power, March, 1948, p. 80.

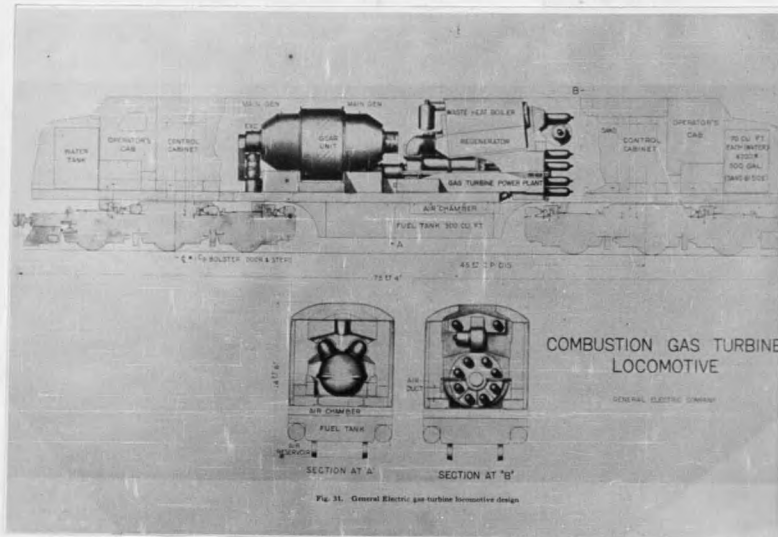


Fig.23-General Electric Gas Turbine Locom.Design)
(from Gen.Elec.Bul.Filing No: 400)

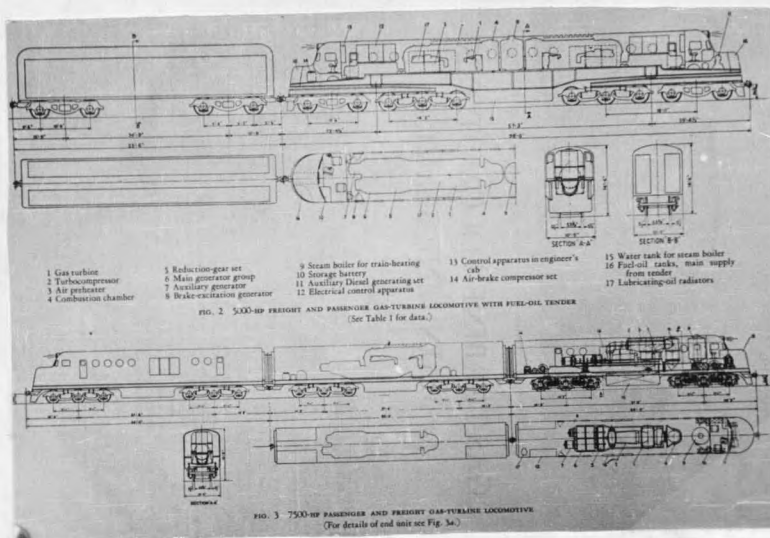


Fig.24-Brown Boveri Locomotive Design
(from Mech.Eng.Nov.1944 p.690)

plane is a typical example in the use of those turbines.

Switzerland is the country in Europe where gas turbines received extensive research and development. The first gas turbine ever installed for power generation is that of Neuchatel ¹ of Switzerland, (Fig.31), in 1939. Its main characteristics are: output at generator terminal at p.f = .8, 4000 Kw, and speed at 3000 rpm. In spite of the simplicity of the arrangement, it has shown an efficiency of 17.38%. This was designed solely as a standby unit for use in case of an aerial attack on Switzerland. Later, in May, 1943 and June, 1944, a 2200 hp was put on the rail and operated on a regular daily schedule over one of the largest secondary lines in Switzerland, which is not yet electrified. During this period, the locomotive covered 50,000 miles in spite of the oil shortage in Switzerland during the war years. Later, the same company designed a 2500 hp "standard" locomotive (Fig.24) for a 7500 hp passenger and freight gas turbine locomotive. In 1946, they were testing a locomotive of 4000 hp.

At present, the Brown Boveri Company has under construction 13 gas turbine power plants, ranging from 1650 Kw terminal output to 27,000 Kw. The last unit is scheduled to be delivered in the winter of 1948 to the Northwest Power Company, Bejnav, Switzerland. It has a heat exchanger and two-stage

1- Stodola, Dr. A., Load Tests of Combustion Gas Turbine, Brown Boveri Bulletin 1584E - 11.9 (xl.46) L 1044.

gas turbine with expected efficiency of 34%. Tests have been completed on the 10,000 Kw unit for Bucharest, Rumania, designed for peak loads. The unit is of 2-shaft design with reheating and intercooler, but not regenerator. It showed a thermal efficiency of 23% against 22% guaranteed.

Along with the Brown Boveri Company in Switzerland, there is the Sulzer Brothers, developing primarily the closed cycle gas turbines and the Maschinenfabrik Oerlikon on small open cycle units.

COAL AS A FUEL FOR THE GAS TURBINE

As it has been previously mentioned, lately there is a trend in the United States of America to use coal as a fuel in the gas turbine cycle. In this field, the United States seems to be ahead of Europe and mainly of Switzerland because reports given by S. A. Tucker¹ state that although the Brown Boveri Company experimented with pulverized coal as early as 1941 on a 1500 Kw machine, the available information indicates that the combustion of solid fuel cannot be hoped for in the near future.

Since May, 1945, the Locomotive Development Committee of Bituminous Coal Research, Inc., Baltimore, Md., has been developing a coal burning gas turbine, two plants of which are under construction as previously mentioned by Elliott and Allis-Chalmers Companies. The cycle diagram for the proposed plants is shown in (Fig. 26). As it is shown, the cycle works on the open cycle principle. (Fig. 29) shows a diagram of pulverized coal combustion chamber adapted from the British Fuel Research Station's vortex design with cyclone for removing flyash. Fuel and primary air enter more or less tangentially. Secondary air flows through admission ports around the periphery, moving across the fuel in such a way as

1- Tucker, S. A., Gas Turbines, Present and Future Status, Mechanical Engineering, June, 1944, p. 363.

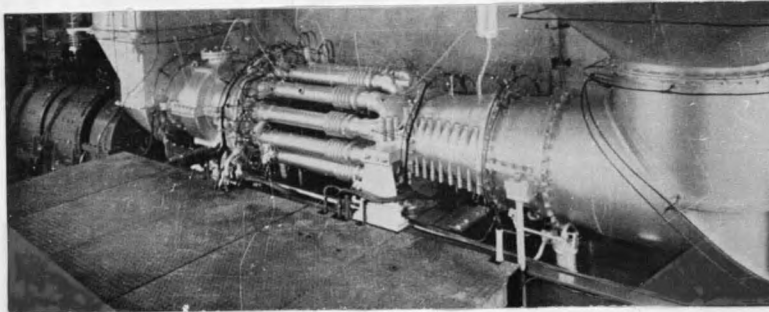


Fig.25- Westinghouse 2000 Hp. Gas Turbine
(from West. Eng. March 1948 p.39)

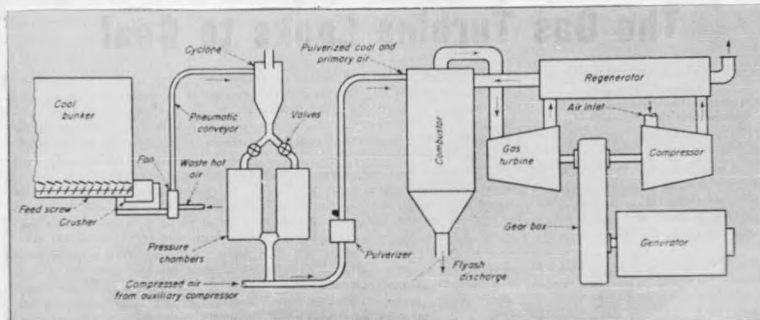


Fig.26- Coal Burning Gas Cycle Diagram
(from Power Aug. 1945 p.74)

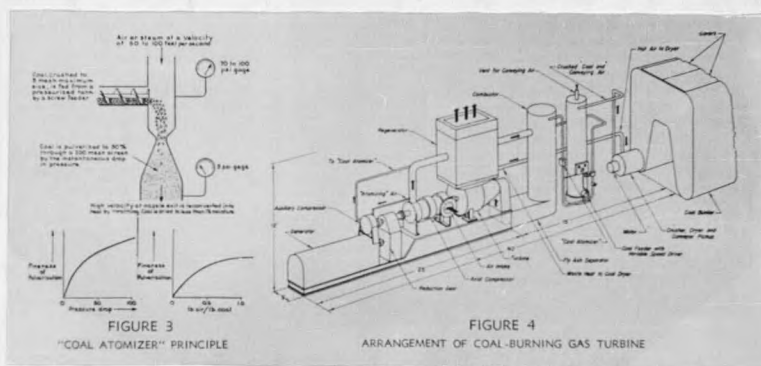


Fig.27- Coal Atomizer and Turbine arrangement
(from Illin. Tech. Bul. Summer 1946)

to cause combustion to follow a spiral path. High relative velocity in passing fuel particles insures rapid, complete combustion. Flyash particles, except those light enough to follow the gas streamline, drop out at the bottom. According to Dr. John I. Yellott ¹ the economics of fuels in the United States are such that coal becomes competitive with Diesel, when the Thermal efficiency of the coal burner apparatus is as low as 25% of the efficiency of the Diesel which means an approximate gas turbine efficiency of 10%.

Coal handling system is now being tested ² with full scale equipment, following successful operation of small scale apparatus at the Dunkirk N. Y. pilot plant. Besides, full scale work on combustion is now under way at Battelle Institute at atmospheric pressure, and at the Northrop-Houdy Company's test site in the Kaiser Steel Works at Fontana, California. It is expected that the removal of flyash will render the combustion products virtually nonabrasive, as has been shown with small mechanical apparatus at the Institute of Gas Technology. Tests made by the Allis-Chalmers Company show that particles five microns in diameter are relatively harmless to turbine blade material when carried at high tempera-

1- Yellott, John I., Director of Research, Locomotive Development Committee of Bituminous Coal Research.

2- A.S.M.E. Technical News, Mechanical Engineering, March, 1948, p. 248.

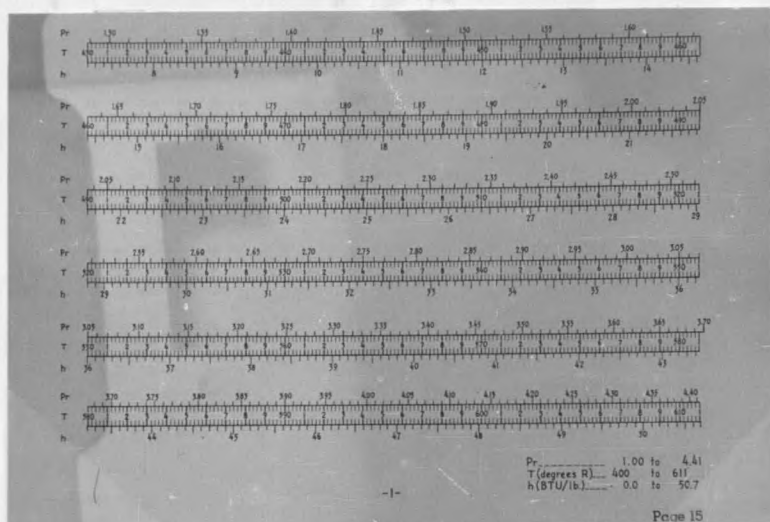


Fig.28-Combustion Gas Chart
(from Research and Stan.Branch,Bureau of
Ships,Navy Dept.Wash.D.C.No;6-44)

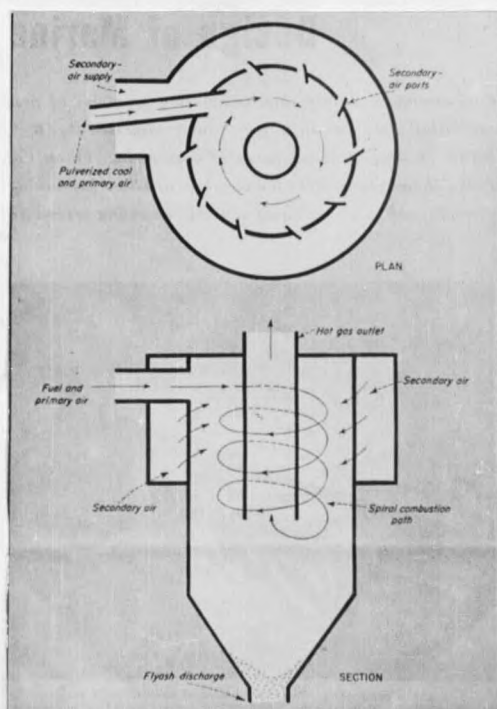


Fig.29-Cyclone Coal Burner
(from Power, Aug. 1945, p.75)

ture air stream. The general arrangement of the apparatus to be used is shown in (Fig. 27). The coal atomizer principle is also shown in (Fig. 27). By releasing the pressure on the coal it has been found that it is possible to pulverize crushed coal. This process is called coal-atomization and is a product of the research program of the Institute of Gas Technology at Illinois Institute of Technology in Chicago. A comprehensive discussion of the subject is given on a mimeographed paper by John I. Yellott, Peter R. Broadley and Charles F. Kottcamp.¹

1- Yellott, John I., Broadley, P. R., Kottcamp, C. F., The Coal-Burning Gas Turbine Locomotive, Bituminous Coal Research, Inc., Baltimore, Md., 1947 A.S.M.E. Annual Meeting, Paper No. 47 - A-118.

ADVANTAGES OF GAS TURBINES OVER THE
OTHER TYPES OF PRIME MOVERS
AND COMPARISON WITH THOSE

Thus far, almost everything concerning gas turbines has been discussed, and the advantages over the other types of prime movers have been pointed out. Different figures and illustrations presented in this paper clearly show that the gas turbine can be used effectively on land, on sea, and in the air, replacing almost all the existing types of prime movers with their simplicity and reliability and with their relative high efficiency. If we consider that the research on gas turbines started very late as compared with the other types of prime movers, there is much hope that, in the near future, the gas turbine engine will hold the first position in the family of up to the present invented power generating units.

In comparing the gas turbines with the other forms of power plants, we have:

1. Advantages compared with Steam Plant

- a. Less expensive.
- b. Simpler in design. No boiler, auxiliaries and feed water supply.
- c. No condensing plant and, consequently, no cooling water necessary.
- d. Simple to operate.
- e. Maintenance costs are low.

f. Plant can be quickly put into service.

2. Advantages compared with Diesel Engines

a. Less expensive.

b. Taking account of the lower grade fuel it can use and of the relatively small amount of lubricating oil it requires, a gas turbine thermal efficiency of 21% on the dollar basis corresponds to about 38% of Diesel efficiency.

c. No cooling water necessary.

d. Running smoothly, it does not require expensive foundations.

e. Maintenance costs are low.

GAS TURBINE GAS CHARTS

The research and standard branch, Bureau of Ships, Navy Department, Washington, D.C. on December 1944 published the above named charts. (Fig.28) shows a page of those which express the relation between temperature, enthalpy and relative pressure function Pr . The relative pressure is evaluated from

$$A R \ln Pr = \int_{T_0}^T c_p \frac{dT}{T}$$

If any of the above quantities is known the others can be found directly from the charts.

For instance, if H and T for an adiabatic compression of 6 is to be found, given the initial temperature, say 540 R first the relative pressure is found. In this case $Pr_I = 2.862$. Then multiplying this value with the given pressure ratio 6 we have $Pr_2 = 6 (2.862) = 17.172$. With the aid of those charts the new temperature and enthalpy is thus found.

Auxiliary Combustion Charts are provided to take care of the moisture content of the air used,

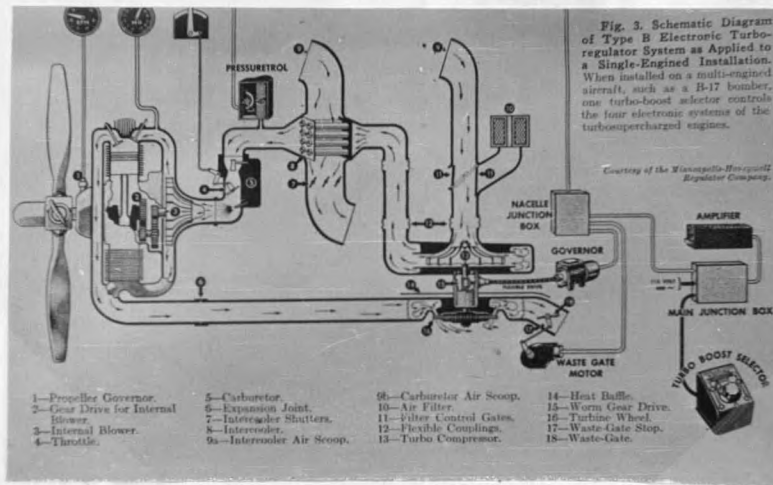


Fig.30- Turbosupercharger and Engine Arrangement
((from Modern Gas Turbine by R.T.Sawyer,
Prentice Hall, N.Y.1945,p.155)

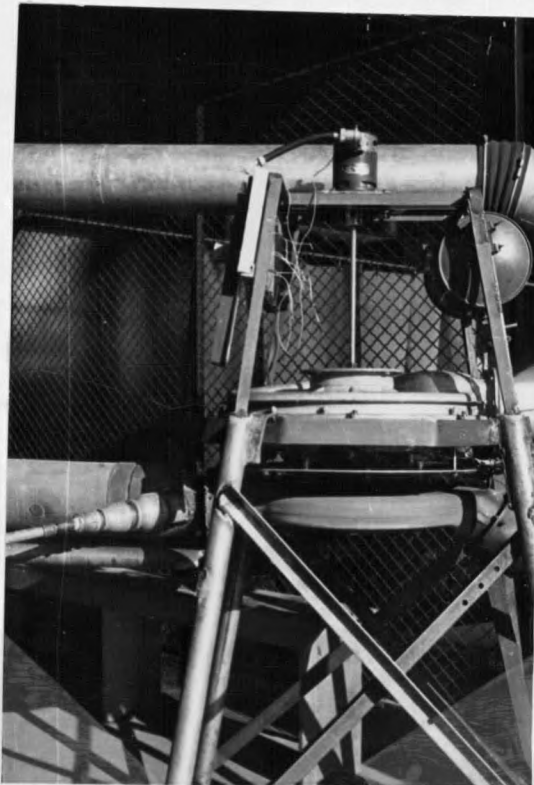


Fig.3I-Labor.Gas Turbine Installation

GAS TURBINE DEVELOPMENT IN
MONTANA STATE COLLEGE
LABORATORIES

As it is clearly stated in the introduction, the aim of the work done in the Mechanical Engineering Laboratories of Montana State College was to transform a turbosupercharger to an engine working on the open cycle gas turbine principle.

Before proceeding further, let us examine the general characteristics of turbosuperchargers as a whole and particularly the one which has been used in this project.

Turbosuperchargers were first used in France just after the first World War and have been taken into serious consideration a while later by the General Electric Company in the United States. The Army Air Corps pressed its development and so during the second World War, the United States alone had undertaken possession, through turbosupercharging, the best planes for high altitude operation.

Turbosuperchargers of the G.E. type were installed on Boeing Fortressess (B-17), and Super Fortresses (B-29), the Consolidated Liberator B-24, Lockheed Lightning P-38 and others. The main advantage of using these turbosuperchargers is to develop the rated engine power at altitudes up to 25,000 and 30,000 feet high. (Fig. 30) shows the arrangement of the turbosupercharger in connection with the airplane engine.

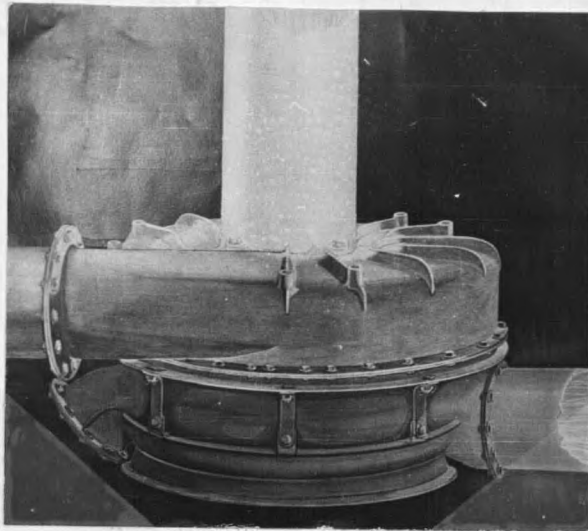


Fig.32-B-Type Supercharger
(from Tran.A.S.M.E.July 1944,p.361)



Fig.33- Turbine Rotor Wheel

The flow diagram is clearly seen. The engine acts as a generator of hot gas under pressure to supply the turbine, the turbine being connected to the same shaft as a centrifugal compressor, which compresses the air and supplies the engine cylinders.

Suppose the plane flies at 25,000 feet altitude where the pressure is about 11.1 inches of Hg. To supply the engine with 29.92 inches Hg, the compressor must run at a pressure ratio of $29.92/11.1 = 2.69$. According to the calculations given by the General Electric Company¹, the power required by the compressor is 1.21 Hp per lb of air per minute. On the other hand, the exhaust engine gases at 1500° F and 29.42 inches Hg at the exhaust will develop 2.88 Hp per lb per minute. Since we have an efficiency of $1.21/2.88 = 42\%$, it is totally practicable even if we allow for different losses around the cycle.

Of course, the back pressure set up in the engine of about $29.92 - 11.1 = 18.82$ inches Hg is not desirable from the engine's viewpoint, but this makes the inlet charge be compressed and thus the engine has a net gain. As the altitude increases, naturally the compressor must develop greater pressure ratio, which requires more power. At the same time,

1- Standerwick, R. G., A. S. M. E. Transactions, Vol. 66, January, 1944, p. 61.



Fig.34-Turbine Nozzle Box

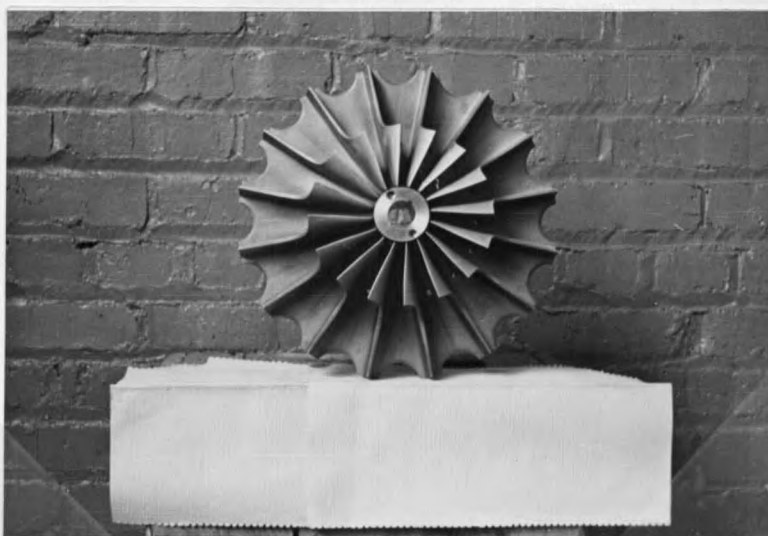


Fig.35-Compressor Impeller Blades

the pressure ratio across the turbine also increases with altitude, so that the power available continues to balance the power required. The limit of balance is either reached by maximum turbospeed or by loss of efficiency at excessive high pressure ratios.

The turbosupercharger to be transformed to gas turbine was taken from the "Lady Lilian", which was a surplus B-17 Flying Fortress. A considerable amount of time was spent for the dismounting from the No. 2 engine of the plane. This is a General Electric, type B-2 made by Ford Motor Company. The outside appearance of the B type turbosuperchargers is shown in (Fig.32). The turbo was afterwards taken to the laboratory and disassembled to examine the way it was built and the position in which it should be installed. (Figures 33-37) show the different parts of the turbo. Specifically, (Fig.33) shows the turbine wheel with impulse blades, (Fig.34) shows the turbine nozzle box, (Fig.35) is the centrifugal compressor blades, (Fig.36) shows the turbine wheel together with the bearing casing, and (Fig.37) shows the lubricating oil pump. The unit is in a compact form. It has roughly the shape of a 25 inch diameter cylinder. On one end (Fig.32) there is the turbine wheel 12.5 inches D and on the other the compressor impeller of about 10 inches in diameter. The total width is about 14 inches.

After a careful examination of the mechanisms involved,

it has been found that the lubricating pump will not operate in a vertical position from the rotor axis, although it can for some time when it is installed on a flying plane, due to the centrifugal force developed. The lubricating oil pump sucks only from the bottom of the bearing casing.

The unit was installed as it is shown in (Fig. 3I). The general arrangement of the gas turbine as viewed from two different angles is shown in (Fig. 38). A partly aluminum and galvanized sheet metal pipe induced the compressed air from the compressor outlet to the combustion chamber. This is made the same as the other pipes, of 3.5 feet long and 6.5 inches in diameter, stainless steel pipe taken from the airplane exhaust stack.

As it is the case with all kinds of gas turbines, the unit should come to a speed of about 10% of the rated one,¹ say 2130 rpm, with outside means, before it will be able to run itself.

Several schemes were considered to start the engine, namely to use a motor, compressed air, and steam. The latter seemed to be the most convenient one, and it was applied with considerable success. A conic section of $\frac{1}{2}$ inch regular water pipe enclosing 4 out of 46 nozzles existing in the nozzlebox, when supplied with steam of about 50 pounds per square inch

1- Sawyer, R. Tom, Gas Turbine Construction, Prentice-Hall, Inc., New York, 1947, p. 162.

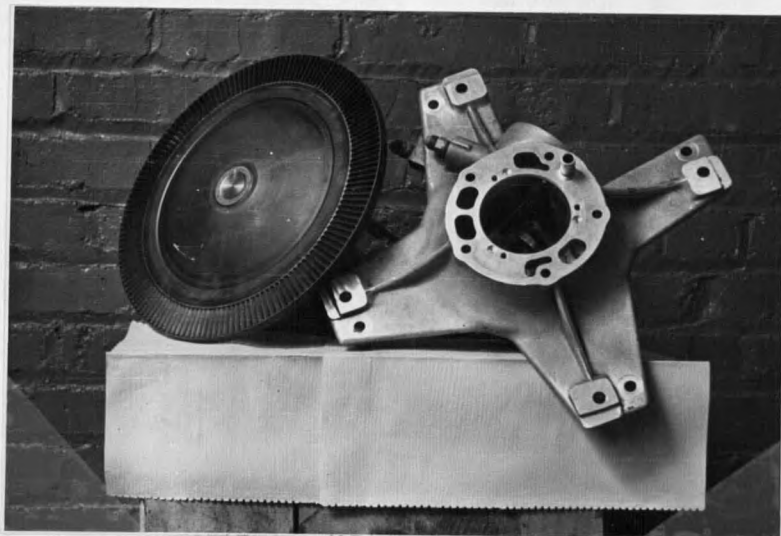


Fig.36- Turbine Rotor and Casing



Fig.37- Lubricating Oil Pump

gage, it was able to run the turbine at 2200 rpm.

The first test was run without any primary air control in the combustion chamber. Natural gas with a heating value of approximately 1100 Btu per cubic foot, at 27 pounds per square inch gage pressure, was introduced through the $\frac{1}{2}$ inch pipe shown in the (Fig.38a). The burner consisted of 15 holes $\frac{1}{16}$ inch in diameter, so that the gas would be spread and well mixed with the air. A spark plug furnished the ignition requirements. Through a series of tests, the maximum attainable rpm was 5200, the turbine running completely with the products of combustion.

After this success, several other trials were made to improve the combustion chamber. The combustion chamber was divided in two. Another pipe 4 inches in diameter, (Fig.38), was installed in the $6\frac{1}{2}$ inch stainless steel combustion chamber. A butterfly valve working like a carbureter throttle was regulating the air through the 4 inch pipe where the burner and the spark plug were located. After a series of tests were run, improvement could be seen through the increased number of turbine revolutions. (Fig.39) shows the diagram of the combustion chamber having as a result, the maximum attainable rpm of 7000 or $\frac{1}{3}$ of that rated for the type of turbosupercharger we were using.

As far as the measurement of the power generated is concerned, this is obtainable through a 0.5 horsepower generator

hooked to the turbine-compressor rotor, (Fig.3I). This was a motor running at 4000 rpm - the only one available - with 60 volts terminal voltage, and it was converted to a self-excited generator, so that there will be no doubt of the generated power.

The turbine as it now stands has 3 banks of 2 in parallel 28 volts 0.1 ampere light-bulbs, and a voltmeter with an ammeter to measure the developed power. It also has a pilot tube in the compressed air pipe connected to a U-tube, which gives readings and data for the measurement of energy input to compress the combustion air. The rpm of the turbine and compressor rotor can be counted through the extension shaft of the generator.

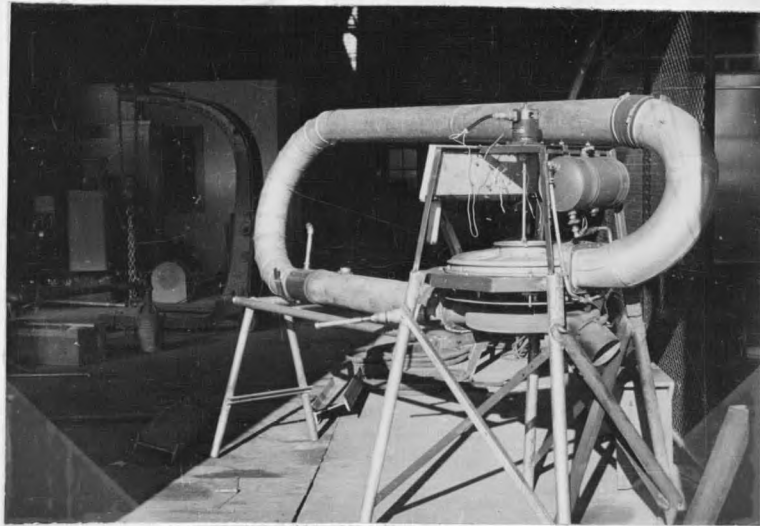


Fig.38a- Experimental Gas Turbine in Lab.

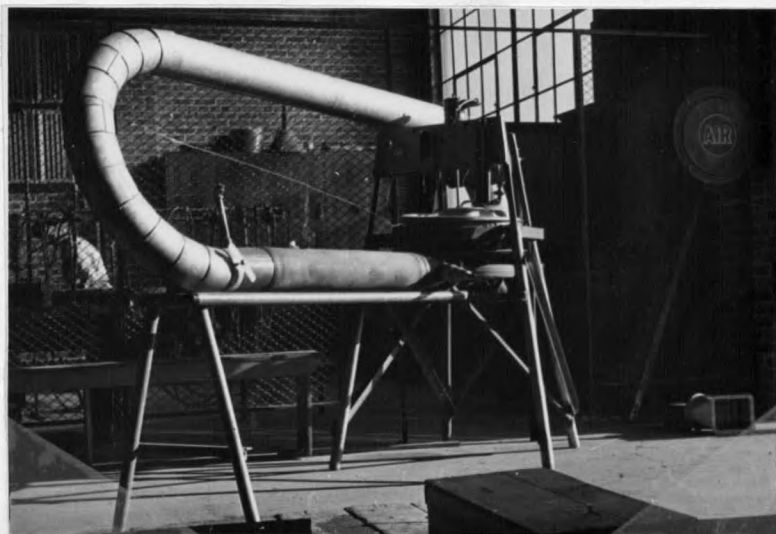


Fig.38b- Experimentan Gas Turbine in Lab

OPERATING PROCEDURE OF THE GAS TURBINE UNIT

Although it has been pointed out in many instances that gas turbines even now, when they are in the wide development stage, have shown a reliable performance without any power cut-off during operation, due to the fact that the one developed here is not hundred percent scientifically designed, certain precautions should be taken in operating the unit.

The most critical instant in running the said unit is when starting it. Explosion or back-firing (a term used in automotive engines when the ignition switch is turned off with the engine running geared to the drive shaft on a sloping road) may occur, when the fuel valve is turned on after the turbine is speeded to about 2000 rpm with the steam nozzle, even though the ignition switch is on.

To prevent any accidents, operating personnel must not be around the turbine that is very close to it during this period.

The following procedure should be followed in starting the unit:

1. Open oil lubricating valve beneath the oil tank.
2. Open the steam valve and see that the turbine runs at about 2000 rpm.
3. Turn on the ignition switch for the spark plug in the combustor.

4. Turn on, slowly, the natural gas supply to the burner and wait for the gas to catch fire.

After the above steps have been followed and the turbine runs with the products of combustion, the steam supply may be cut off. It is wise, though, to leave a thin stream of steam to pass through the steam nozzle so that the melting of it will be avoided. It has been said that this nozzle is made of simple water pipe, and although it is not subjected to any stress, it might give way in the high temperature flow of combustion products.

By adjusting the primary air valve of the combustion chamber with the fuel supply valve, the turbine rotative speed may easily rise to 7000 rpm or even higher. When the engine is running, there is no fear of explosion under normal conditions. Different measurements may be taken such as the turbine rpm, pitot tube readings, and, when the engine is loaded, voltmeter and ammeter readings.

EXPERIMENTAL DATA AND RESULTS

Although the results given below they are not complete in every respect, in a way they show and give some performance characteristics of the experimental gas turbine unit.

The unit as (Fig. 31) shows is equipped with a 0.5 Hp generator and a pitot tube (not shown) on the straight portion of the pipe above the combustion chamber. To have representative readings of the pitot tube which was connected to a mercury U-tube manometer, different readings along the section of the pipe were taken and the average value for each compressor RPM is given below. The Generator during the first two runs was loaded only with approximately 18 watts ; those of the light bulbs , and on the last one it was loaded to full capacity with an external variable resistance. No appreciable slow down of the engine was noticed as the unit was loaded, although the gas supply was kept constant. This fact shows that the unit can be loaded a considerable amount more if it will be equipped with the necessary equipment.

The unit was gradually loaded so that the armature current would not exceed the rated one ; the generator used had a rating of 10 minutes and armature current 8.3 amperes. The voltage remained constant which is an other proof that the unit did not slow down. As it is mentionned before the generator is self excited . The resistance used for that purpose in series with the shunt field is so arranged that although the generator is rated

60 volts at 4,000 rpm, actually it generated 57 volts at 6,600 rpm.

Table: II - Test Data

Rpm	Total Pres. inches Hg	Static Pres. inches Hg	Differ. in. Hg	Volts	Amp.
5,800	1.00	0.60	0.40	47	0.3
6,000	1.20	0.65	0.55	55	0.3
6,600	1.50	0.70	0.80	57	3.0

Barometric Pressure: 25.22 Inches Hg

Temperature: 85 F

Throat area: 30.2 sq.inches

Table: III - Test Results

Rpm: 6,600
Load 456 watts
Air Horsepower 8.93

Computations:

$$\begin{aligned} P &= H \times W \\ \text{lb/ft}^2 &= \text{ft} \times \text{lb/ft}^3 \\ &= \text{in} \times \text{ft/in.} \times \text{lb/ft}^3 \end{aligned}$$

$$\begin{aligned} P &= \text{in.} \times 62.30/\text{I}2 \\ H \times W &= \text{in.} \ 5.192 \\ H &= 5.192 \text{ h/W} \end{aligned}$$

$$\begin{aligned} v &= \sqrt{2gH} \\ &= \sqrt{2g \ 5.192 \text{ h/W}} \\ &= 18.27 \sqrt{h/W} \\ &= 18.27 \sqrt{\frac{0.80 \times 13.56}{0.0728}} \\ &= 220 \text{ ft/sec.} \end{aligned}$$

$$\begin{aligned} \text{Air Horsepower: } & \frac{220 \times 60 \times 30.2 \times (1.50 \times .4912) \times 144}{144 \times 33,000} \\ &= 8.93 \text{ Hp} \end{aligned}$$

Discussion of test results

The writer does not claim that the results presented are decisive in determining the performance of the gas turbine unit. However, they are the best which could be attained under existing conditions and facilities. Besides, the purpose of the work done was to demonstrate the principle of the new developed motive power unit entering the family of the existing up to date principal prime movers. No attempt has been made to find the thermal efficiency, but here is an estimate of that.

Considering that most of the turbine work goes to compress the air, since the air Horsepower at 6,600 rpm was 8.93 hp assuming a value for the bearing losses of the turbine rotor, work consumed by the lubricating oil pump and the pressure drop through friction in the pipe from the compressor outlet to the point where the measurements were taken, we have as approximate power to the turbine rotor about 10 hp plus the measured output i.e. 456 watts. The rest of the Heat energy of the fuel supplied goes to exhaust gases and a relatively small percent to convection losses and radiation to atmosphere.

Certainly the Heat energy taken away by the exhaust gases constitutes a big percent of the total. Comparing this with a Diesel engine which exhausts gases at about 800 F, having a 28 % heat rejection to atmosphere we might assume a safe value of about 35 % for heat rejection through exhaust gases, and about 10 % for losses through convection, radiation and miscellaneous others.

On the other hand assume that the 55 % of the turbine output is the measured 8.93 Hp, plus 456 watts, plus friction losses, all-together forming 7916 watts.

Accordingly the estimated thermal efficiency of the unit is about

$$\text{Ther. Effic.} = \frac{456}{7916 - 6480} = 3.17 \%$$

The above figure although low, is acceptable for the developed experimental gas turbine unit. The reader should be reminded that the first modern gas turbine built in Switzerland in 1939 working on the simple open cycle gas turbine principle had an efficiency of 17.3 %. The one built here is of primitive nature for demonstration purposes only, not built to develop any actual useful output.

GAS TURBINE DEVELOPMENTS IN OTHER
NORTHWESTERN COLLEGES OF THE
UNITED STATES

As far as the writer of this thesis knows, two other institutions of the Northwest are in the stage of developing gas turbines of the same nature. These are Oregon State College and the University of Utah.

From reports and papers read in the A. S. M. E. Annual Regional Conference of the Pacific Northwest Student Branches in the State College of Washington, Pullman, Washington on May 5-8, 1948, we learn that Oregon State College developed a gas turbine using a turbosupercharger. A paper read by George Frank of Oregon State College stated that in constructing this turbine, they used a regular combustion chamber made by Allis-Chalmers, burning Diesel oil as a fuel. It was said that the turbine ran at about 15,000 rpm and that the compressor worked against a 27 inch mercury pressure. No power was taken from that unit, and George Frank stated that everybody was satisfied there, having the turbine running on the open cycle principle. Starting of the unit was accomplished with a similar nozzle as used here, but using compressed air instead of steam. They planned to demonstrate the jet principle and measure the possible thrust, so they made some changes in the lubricating oil system of the unit,

and had the rotor axis in the horizontal plane.

The writer can count the following as possibilities for the better performance of the Oregon State College unit.

Either:

- a. The unit was of the type ¹ B-11, B-22, B-31, B-33, which run at 24,000 rpm, or
- b. The combustion chamber, since it was factory made, had a better performance, or
- c. For both the above reasons.

The gas turbine being constructed at the University of Utah was then at the primary stage, so nothing was said about its performance. However, those interested planned to purchase a combustion chamber similar to the one at Oregon State College.

1- Sawyer, R. T., Gas Turbine Construction, Prentice-Hall, Inc., New York, 1947, p. 162.

SUGGESTIONS TO IMPROVE PERFORMANCE
OF EXPERIMENTAL GAS TURBINE UNIT

The developed gas turbine, although it successfully runs with the products of combustion without any outside assistance and produces some useful work, there is much room for refinement of the unit. Obviously, the component which needs retouching is the combustion chamber, to be able to substitute as closely as possible the 9 cylinder airplane engine from the products of combustion of which - exhaust - it was turning.

At present the turbine rotor spins at 7,000 Rpm which is $1/3$ of the rated value for the B-2 type of supercharger we are using. First thing which comes to mind in trying to find ways to boost the turbine Rpm and by doing so, to increase the percentage of the useful output, is to burn more fuel in the combustion chamber. During a series of tests, it has been noticed that as the fuel supply was gradually increased, instead of having an increased Rpm reading which would mean more air to combustion chamber to burn the supplied fuel, flames of combustion process were escaping through the turbine rotor blades. This of course means incomplete combustion and waste of extra fuel without return. There are three ways to remedy this ; either,

- a- Increase the length of Combustion space, so that more fuel will be burned before the combustion products will reach the turbine blades, or

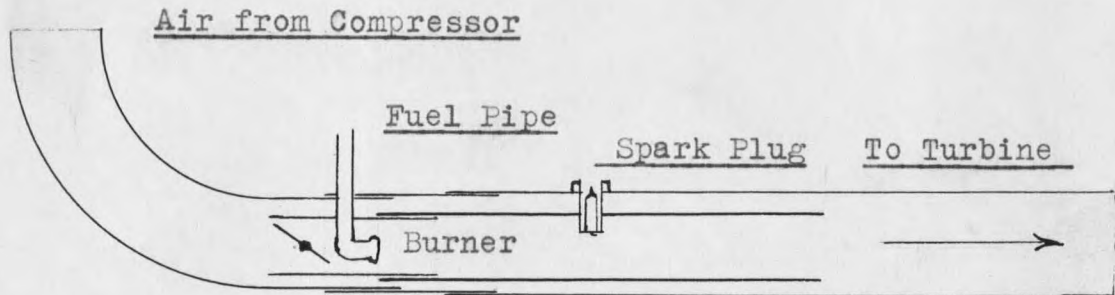


Fig. 39 - Sketch of Combustion Chamber in use
at present. Scale: 1" = 1'

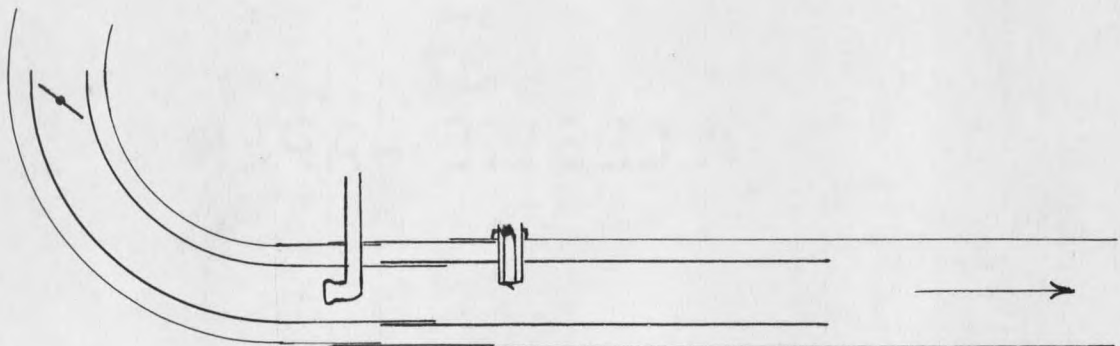


Fig. 40 - Sketch of proposed Combustion Chamber
Scale: 1" = 1'

b- Provide a better mixing of the fuel with the combustion air, or

c- Do both the above refinements.

Considering literature on Gas Turbine Combustion Chambers, the length of the present one seems to be enough. However, it seems that it is narrow; it has the same size as the pipe which conducts compressed air from the compressor outlet. The Combustion air which comes at a considerable velocity can not have the chance to be completely mixed with the fuel supplied to it. The velocity is still increased considering the increased volume due to the addition of heat to it.

Accordingly it is thought that if the 6 $\frac{1}{2}$ inch combustion chamber width will be increased to at least 12 inches, a much better performance of the present unit can be expected. In such a case the present stainless steel outside shell of combustion chamber can be used in the place of the present 4 inch pipe used as a primary combustion space.

A simpler but not as good as the first one alternative would be to use the construction shown in the sketch of (Fig. 40). With such a construction the fuel will be better mixed with the primary air for the control of which a butterfly valve is provided. Notice the position of burner in (Fig. 40). Of course an ideal case would be to use both of the refinements mentioned above.

The writer feels pretty sure that if the above suggestions will be applied, they will have as effect the boosting of turbine-compressor Rpm and better overall performance of Gas Turbine unit.

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