PRINTED IN U.S.A.

69-GT-106

an ASME publication

$1.50 per copy – $.75 to ASME Members

DESIGN AND EXPERIENCE WITH REGENERATORS FOR INDUSTRIAL GAS TURBINES

by

Peter Wolfe
Harrison Radiator, Lockport, N.Y.

Howard F. May
General Electric Co., Schenectady, N.Y.

ABSTRACT

This paper describes the present base load gas turbine regenerator. Covered herein are the basic design parameters, the actual physical configurations of these units along with considerable construction detail, and the service experience covering the eleven years of its existence. The evolution of the design is described.

Contributed by the Gas Turbine Division of The American Society of Mechanical Engineers for presentation at the International Gas Turbine Conference & Products Show, Cleveland, Ohio – March 10-13, 1969.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, UNITED ENGINEERING CENTER, 345 EAST 47th STREET, NEW YORK, N.Y. 10017

GENERAL INFORMATION

Basically a regenerator is used with a gas turbine to save fuel. By definition it is a device which is used with hot air engines in which the incoming air is heated by being passed through a pipe or pipes heated by a flow of hot air or gas escaping in the opposite direction. In a gas turbine cycle the regenerator is used to heat the compressor discharge air with the turbine exhaust gases.

Figure 1 is a typical regenerative gas turbine cycle diagram. As can be seen from the diagram, the compressor discharge air is heated by the regenerator from 500°F to 900°F, a temperature rise of 400°F. Fuel is then burned in the turbine combustion chamber to heat the air from 900°F to 1600°F (T = 700°F). Thus, of a total temperature rise of 1100°F, the regenerator has contributed 36.5 percent of the heat added; in other words, the regenerator theoretically saves 36.5 percent of the fuel a simple cycle turbine would use. For this example, the simplifying assumptions of constant specific heat and zero-cycle air and gas pressure drop result in a net actual fuel saving of approximately 31 percent for an 80 percent air side effectiveness regenerator.

PERFORMANCE

Two terms are used to describe the performance of a regenerator; the air side effectiveness and the total percent pressure drop. The regenerator air side effectiveness is defined as the ratio between the air temperature rise ($T_3 - T_2$) and the inlet gas temperature difference ($T_5 - T_2$) expressed as a percentage. Thus we see in Figure #2 that the regenerator used in the cycle shown in Figure #1 has an effectiveness of 80%.

The second parameter used to describe regenerator performance is percent total pressure drop. This is defined as the sum of the total air side pressure drop divided by the inlet air pressure in psia., plus the gas side pressure drop divided by the gas inlet pressure psia., each expressed as a percentage. Thus we see in Figure #2 that the regenerator used in the cycle shown in Figure #1 has a percent total pressure drop of 3.4%.

The effect of the two performance values, effectiveness and percent total pressure drop, on the overall turbine cycle performance, is shown in Figure #3. An infinite number of effectiveness-pressure drop combinations are possible which allow the gas turbine cycle to operate at or above the design point. Note that higher pressure drops or lower effectivenesses in the regenerator reduce the overall turbine cycle performance. Most Harrison regenerators supplied to date have been designed to an 80% effectiveness - 3.5% pressure drop design point.

HEAT TRANSFER DESIGN & ECONOMICS

Figure #4 shows how regenerator size is affected by the flow pattern. The smallest unit is obtained using the counterflow pattern. Since cost is related to size, the most economical design is also counterflow.

We also observe from Figure #4 that if an effectiveness of 80% requires a size of 100%, an effectiveness of 70% requires a size of 62.5% and an effectiveness of 85% requires a size of 150%. Thus, regenerator cost increases rapidly for effectivenesses in excess of 80%. This explains why most cost studies to date for base load turbine applications justify an 80% effectiveness regenerator as the most economically attractive.

Another factor affecting cost and size is the efficiency of the heat transfer surface. An almost infinite number of combinations of heat transfer surface geometries can be used for heat exchangers or regenerators. These geometries range from the simplest tubular configurations to the more complex forms of indirect heat transfer surface. To date, Harrison has used a plain, indirect heat transfer surface for the exhaust side of the regenerator and a plain air gap for the compressed air side of the unit. However, depending on economics and requirements of various applications, other more compact and efficient surfaces may be used.

REGENERATOR CONSTRUCTION

The Harrison Model TR regenerator is of "plate-fin" construction as shown in Figure #5. The term "plate-fin" means that plates are used to separate the two fluids and fins or centers are used for the indirect heat transfer surface.

As shown, the compressed air flows in the channel or gap between adjacent plates and the exhaust gas flows through the gas centers or fins which are sandwiched between the two plates. The centers or fins provide additional surface for heat transfer and give support to the plates to resist the air pressure in the air channel. The fins are attached to the plates by copper brazing assuring a low resistance heat transfer path. The braze joint is only loaded in compression and is not considered a structural joint. The combination of air channel, plates, and gas centers, form the basic tube module.

The number of tube modules required in a regenerator is determined by the turbine flow conditions. The modules are formed by assembling two tube plates, gas centers, closure bars and braze foil into a sandwich as shown in Figure #6.

The plates are then welded to the closure bars as shown in Figure #7, forming the exhaust gas channel or tube. This assembly is then copper brazed in a roller hearth furnance as seen in Figure #8. Placing turning vanes and spacer bars on the exhaust tube forms the air channel as shown in Figure #9 and completes the construction of the basic tube module. Hi-strength low alloy steel is used for the tube modules and pressure vessel steel (SA-204) is used for all supporting structures and "strongbacks". The basic tube module is approximately 28" wide, 150" long and 1" thick.

Basic tube modules are assembled into "bundles" or "packs" by welding them together in a weld fixture as shown in Figure #10. The welds are made two at a time with two oscillating weld heads. The fixture rotates to allow the opposite side of the "bundle" assembly to be welded. Fourteen tube modules make up a "bundle" assembly.

Bundles are welded together to form a core section assembly. As shown in Figures #9 and #11, the outside bundles are fabricated with reinforcements or "strongbacks" which support the tube plates under the compressed air loading. Two core section assemblies placed side by side and separated by four plates, form a core half as seen in Figure #9. The four plates form two plenum chambers which act as manifolds for the inlet and outlet air paths. The upper two plates form the inlet air plenum chamber and the lower two plates form the outlet air plenum chamber. Inlet-outlet flanges, and manways, are added to the inlet and outlet plenum chambers to complete the core assembly.

Legs are added for vertical units and mounting clevises and bearing pads for horizontal units. Two core halves are installed in the field to make a complete unit. In-field erection is minimal since each core half is a complete pressure vessel by itself and can be independently piped. A typical regenerator weighs approximately 56 tons installed.

When the units are installed, they are mounted on spring hangers. For veritcal installations (see Figure #12), two legs are welded to the supporting steel work to form a fixed reference point from which the unit and piping can expand. The remaining six legs are mounted on spring hangers and eyerods. The eyerods allow for both vertical and lateral movement. The eyerods, together with the fixed supports, form a tri-pod arrangement which stabilizes the unit. Guiding the unit at the legs opposite the fixed support prevents the unit from rotating and placing high stresses on the piping.

Horizontal units are installed in a similar manner as shown in Figure #13. A total of eight mounting points are used in these units. One point is used as a fixed reference and is tied solidly to the foundation. Two other points have slide pads which combine with the fixed point for stability. The remaining five mounting points have spring supports which act to uniformly support the unit, allow lateral and vertical expansion, and control the loading on the "lubrite" slide pads.

FIELD EXPERIENCE

In 1957, Harrison Radiator Division constructed the first Model TR regenerator. This unit was installed at Alexandria, La. at a Tennessee Gas Transmission Company compressor station. In 1969 this unit will have 100,000 hours of operation. Since 1957, Harrison has manufactured over 75 regenerators which to date have accumulated over 1,000,000 hours of service.

The regenerator design of today is similar to early regenerator designs in appearance only. Several important changes have been made which allow today's design to give the most reliable, economical operation ever. These changes have produced three distinct regenerator designs. For discussion purposes, we can call these Designs A, B, and C.

Design A was the first regenerator design and includes units manufactured from 1957 to 1961. Design B was an interim design and covers units manufactured from 1962 to 1965. Design C is the current regenerator design and covers units manufactured from 1966 to the present.

The three designs all give comparable heat transfer performance; i.e., they are all designed to the same effectiveness and pressure drop requirements. However, the designs differ in certain structural details described later which affect the performance of the unit during service.

In order to evaluate the merits of one design versus another, certain factors must be measured and compared. In a regenerator the most important factors that are judged are:

1) Air side effectiveness
2) Total % pressure loss
3) Leakage

The air side effectiveness and total percent pressure loss for the regenerator are a function of the heat transfer design of the units. All three regenerator designs, A, B, & C have been identical in this area with the exception of a change in tube module width from 21" to 28" in changing from Design A to B. Tests on the first Harrison regenerator established the heat transfer performance of the designs. These tests are discussed in the ASME Paper No. 61-GTP-12 - "Design Considerations and Operating Experience of Regenerators for Industrial Gas Turbines" by R.F. Caughill. All tests and field experience indicate that the heat transfer performance of the designs is as anticipated. Figure #14 shows the number of tubes or amount of heat transfer surface produced in any one year, remaining in active service as of 1/1/69. These numbers indicate how reliably the heat transfer performance of the regenerator has been maintained. In examining these numbers, the major loss of heat transfer surface sealing off of leaking tubes occurred with Design A. These leakages were a result of cracked welds occurring on or near the hot gas face of the unit. These cracks were found at either end of the lower manifolds where high thermal stresses are encountered during the starting of the gas turbine and near the areas of the mounting feet where high physical loads existed in the original design.

Loss of surface resulted from the original repair methods which sealed off both ends of any tube emitting leakage air. **Simple welding procedures have now been developed permitting field repair of tube leaks.** It is now possible to maintain units at 100% performance levels.

If the units can be maintained at 100% capacity with the proper repair technics, then the leakage becomes the prime method of determining when unit service is necessary. The first warning of excessive leakage would be output. Slide 15 shows the effect of leakage on both output and thermal efficiency. If a machine is running properly, there is no reason to pressure test the regenerator. If, however, an unexplained loss of performance occurs, leakage is a prime suspect. Testing of the unit is done by breaking the combustion air piping at the regenerator flanges, capping the regenerator flanges and pressurizing the unit.

Rate of leakage is found by pressurizing and sealing off the unit. The rate of pressure decay combined with the internal volume of the unit makes it possible to calculate the true leakage at operating conditions. The units are built to a leakage of .1% of rated flow maximum. It is felt that leakage below .2% is acceptable. We do not recommend pressurizing of the unit unless machine performance warrants it. Leaks are found thru the use of soap bubbles on the outside surfaces of the unit or die penetrant checks in the lower manifolds which may be entered thru the rear manways.

DESIGN DUSCUSSION

To the person unfamiliar with regenerator design, the regenerator is thought of as a large, static, heat exchanger which when compared to the sophisticated gas turbine it is used with, should have an almost infinite, troublefree life and present uncomplicated design problems. To those acquainted with the details of regenerator design however, some complicated problems are encountered. These problems are not always immediately evident but become apparent with service. As a result, design changes are made which improve the regenerator, increasing the reliability. The

Harrison Model TR regenerator has undergone several design changes since the first unit was placed in service. The changes have been made to increase the reliability and reduce the leakage of the unit. Most of the changes were made to alleviate thermal stresses and resulted in very subtle but very important modifications in the structural details of the regenerator.

The first major change Figure 17 made in the regenerator design involved the mounting system. Design A was initially installed with slide pads on six of the eight legs. The slide pads were designed to allow for free thermal expansion and contraction of the regenerator during heatup and cooldown. Experience and field testing showed that the slide pads were not functioning properly and were "hanging up", causing high tensile stresses in the tube to tube welds resulting in weld fractures. As a result, a new spring hanger mounting system was devised to allow for unrestrained expansion of the regenerator.

Another problem area was discovered at the junction of the two core halves on the gas inlet face. In order to effect an exhaust gas seal, it had been the practice to weld a solid closure bar to each core half to seal the bypass area between core halves. Due to uneven heatup and resulting uneven thermal expansion, large tensile stresses were transmitted from core half to core half through the closure bar, resulting in fractures in the air header welds allowing high pressure air leakage. Substituting an angle iron expansion joint for the solid closure bar limited the level of stresses that could be developed and eliminated fractures in the air header welds.

The above two changes describe the major differences between Designs A and B. Design B also had tie straps or reinforcing bars installed on the hot end of the unit to reduce stress levels in welds.

Experience with Design B showed that an acceptable design had been reached. This, however, was not the stopping point in the design improvement. Some minor cracking of tube to tube welds near the strongbacks was still evidenced by field inspections. Although leakage was minor, analysis showed a structural modification would be helpful. The analysis disclosed transient thermal shear stresses between the strongback and regenerator core causing eventual weld fracture. A change in the method of attaching the strongback to the core allowed a limited amount of two directional differential expansion to occur between the strongback structure and regenerator core, reducing transient stress levels to acceptable levels. This change is shown in Figure #17 and consisted of scalloping the side plate and partial welding of the support ribs, resulting in a two axis expansion feature. This change was incorporated into Design B as an improvement.

Design C differs from B in a minor but very important manner. Field inspection of units of Design B revealed minor but consistent fractures of the tube side bars at the rear of the unit adjacent to the outlet air plenum chamber. The construction at the rear differed from that in the front in the manner in which the air outlet plenum chamber was closed off. A detailed thermal analysis of the area was made with the aid of computers at General Motors Research. The analysis involved determining the transient temperature history of the regenerator in the failure area during startup and shutdown. From this information, thermal stress levels were determined which were applied to a low cycle fatigue analysis and the resulting cyclic life was determined. The analysis showed that severe thermal gradients occurred in the rear manifold closure area during startup which were of a magnitude sufficiently high to cause failure in about 40 startup cycles. Redesign of the area showed an increase in cyclic life of the failure area of over 10 times. Therefore, the regenerator was redesigned in the failure area to take advantage of the possible improvement. The redesign, together with the addition of inspection manways in the same area, constitute the differences between Designs B and C. Tests on Design C are very favorable with leakage levels of less than .005% after one year being reported.

<u>SUMMARY (FIGURE 18)</u>

You have listened to a description of the modern regenerator and its evolution over the past 10 years. This unit is a highly sophisticated design with over 85% of its weight involved in the heat transfer process. It represents a nice balance between physical demands-100 psig-1000°F-25 year life and weight and space-110,000#-1275 ft^3. Its duty is comparable to a steam boiler producing 36,000#/hr. and on a weight-space basis, the comparison is quite favorable in spite of the penalty of gas as a heat transfer medium on both sides.

Although the present unit is highly satisfactory to meet present requirements the future will bring new and different demands; and I should like to discuss this subject briefly.

1. Higher Exhaust Temperatures

   In the foreseeable future, there will be needs for units which can operate at 1150° exhaust temperatures. This will require improvement in base materials, braying alloys, and manufacturing techniques. There is considerable encouragement in these areas to think that these problems can be solved at reasonable cost increments.

2. Modification of Basic Surface

   Some further refinement of the basic surfaces are possible which would result in weight and space reductions of some 20%. This modification will have no effect on the basic physical design of the unit.

3. Better Integration of the Regenerator & Turbine

   It is recognized that regenerators are large, heavy objects and require large piping. There is a constant effort in the area of arrangement to improve both the simplicity and appearance of regenerator installations.

REGENERATIVE CYCLE TURBINE

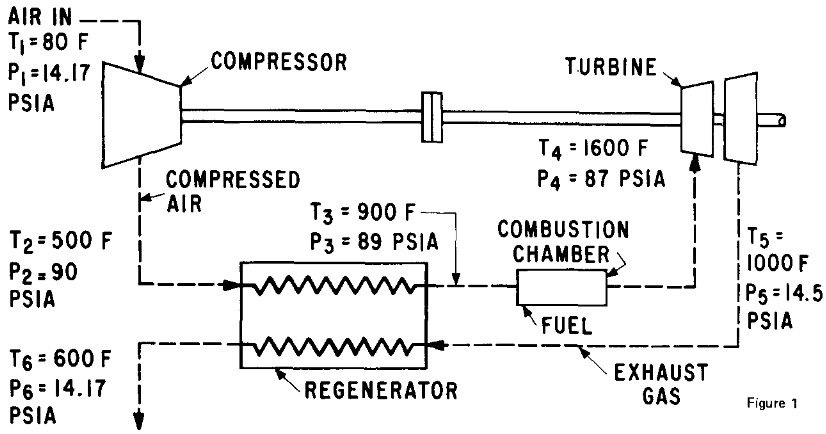
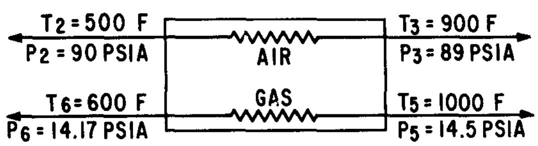


Figure 1

REGENERATOR PERFORMANCE PARAMETERS



AIR SIDE EFFECTIVENESS

$$\frac{T_3 - T_2}{T_5 - T_2} = \frac{900-500}{1000-500} = 80\%$$

TOTAL PRESSURE LOSS

$$\frac{P_2 - P_3}{P_2} + \frac{P_5 - P_6}{P_3} = \frac{90-89}{90} + \frac{14.5-14.17}{14.5} = 3.4\%$$

Figure 2

REGENERATOR PRESSURE DROP AND EFFECTIVENESS vs CYCLE EFFICIENCY

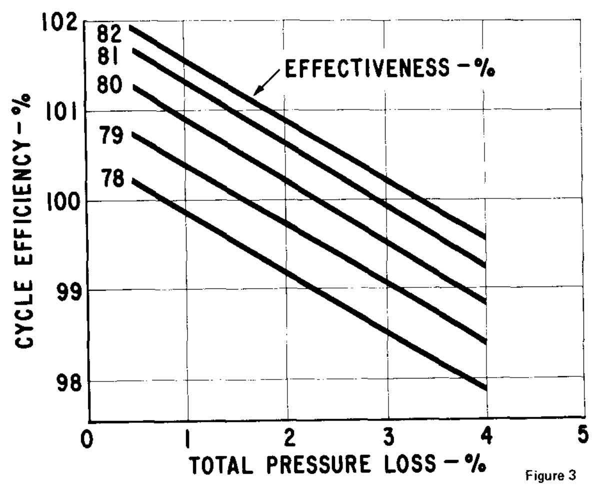


Figure 3

EXCHANGER PERFORMANCE

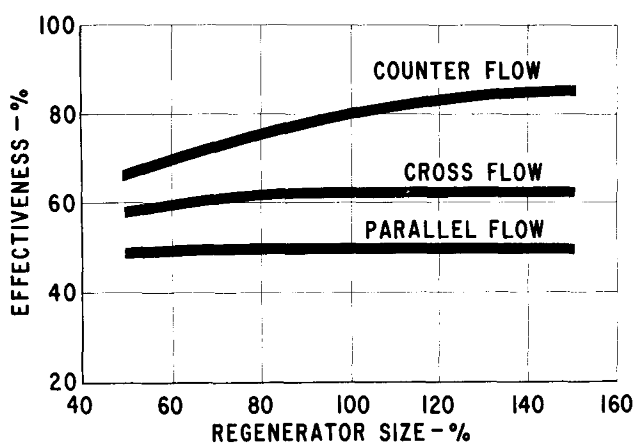


Figure 4

TUBE CONSTRUCTION

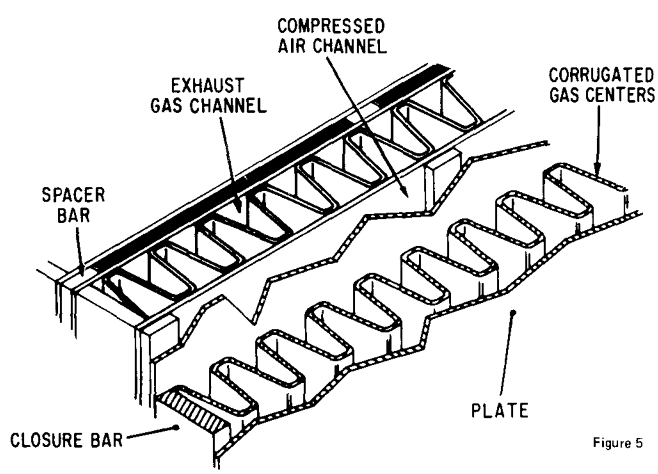
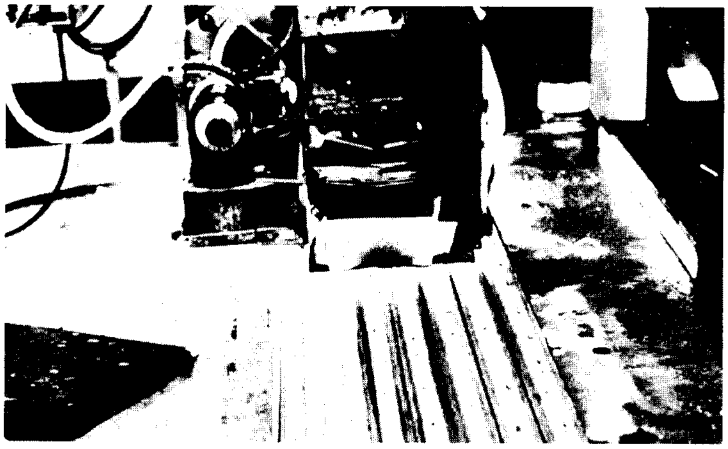


Figure 5

TUBE ASSEMBLY Fig. 6



BRAZING TUBE IN ROLLER HEARTH FURNACE

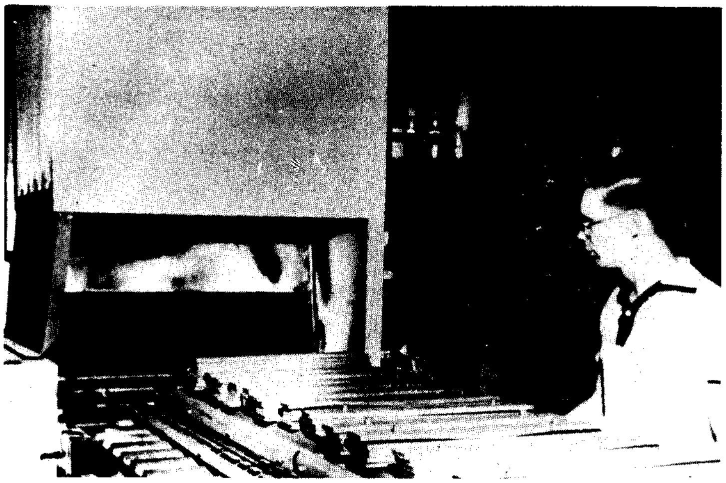
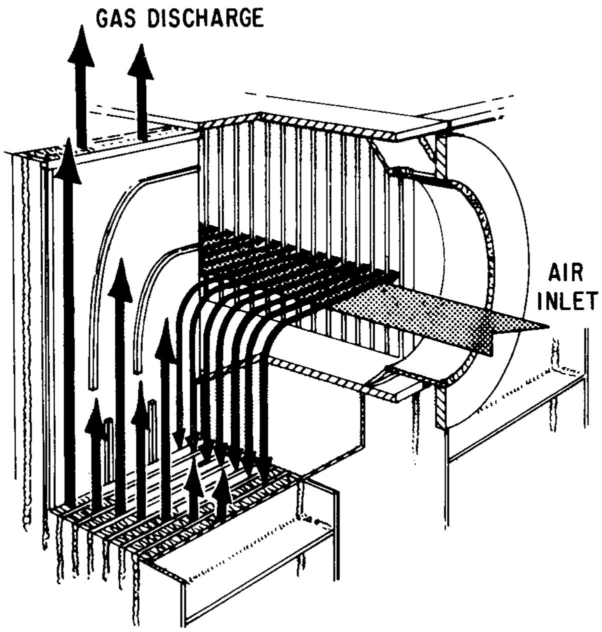
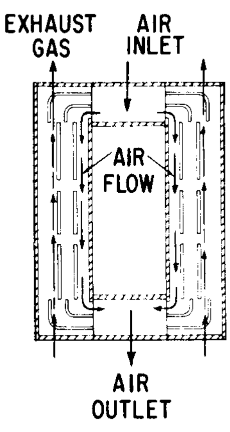


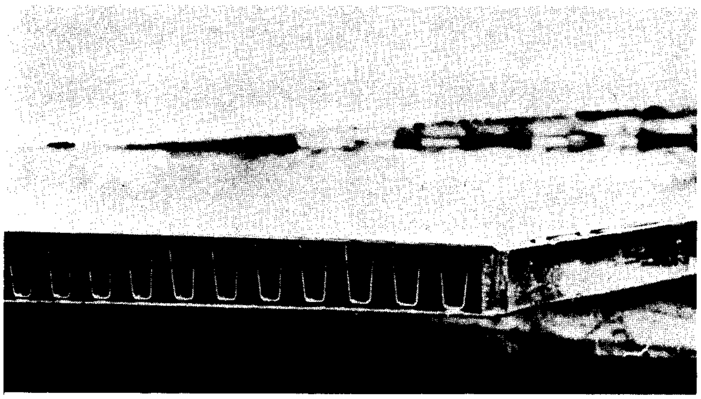
Fig. 8

TUBE ASSEMBLY Figure 7





FLOW ARRANGEMENT Figure 9



BUNDLE ASSEMBLY

WEIGHT — 110,000 LB

$T_{GAS\ IN}$ = 1000 F

W_{GAS} = 100 LB/SEC

W_{AIR} = 98 LB/SEC

η = 81 %

$\frac{\Delta P}{P}$ = 3.5 %

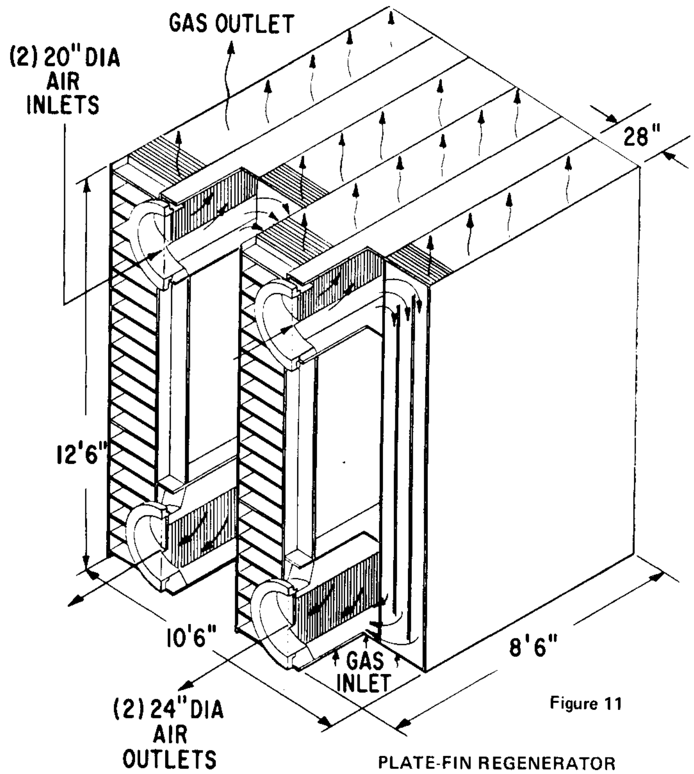
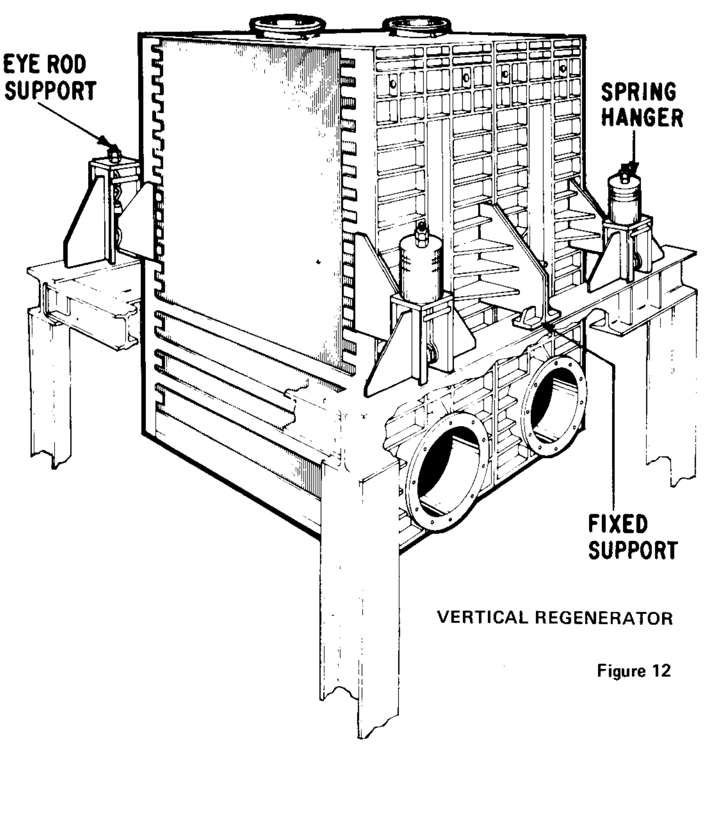


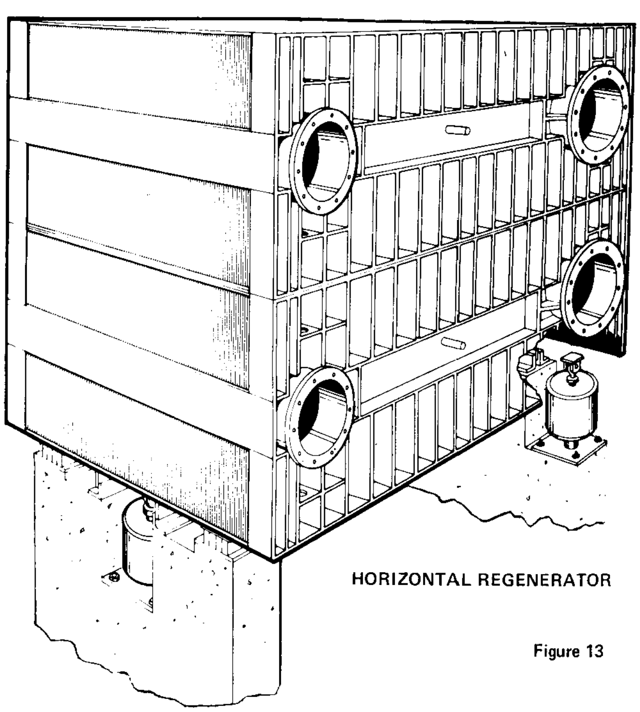
Figure 11

PLATE-FIN REGENERATOR



VERTICAL REGENERATOR

Figure 12



HORIZONTAL REGENERATOR

Figure 13

OPERATION EXPERIENCE

YEAR	NUMBER OF TUBES PLACED IN SERVICE	PERCENT OF TUBES OPERATIONAL 1-1-69
1957	440	97.50
58	2200	96.14
61	1320	99.17
62	1188	100
63	1700	100
64	1360	100
65	1020	99.41
66	6600	100
67	3520	100
68	2352	100
		AVG. = 99.22 %

Figure 14

AIR LEAKAGE vs POWER OUTPUT & EFFICIENCY

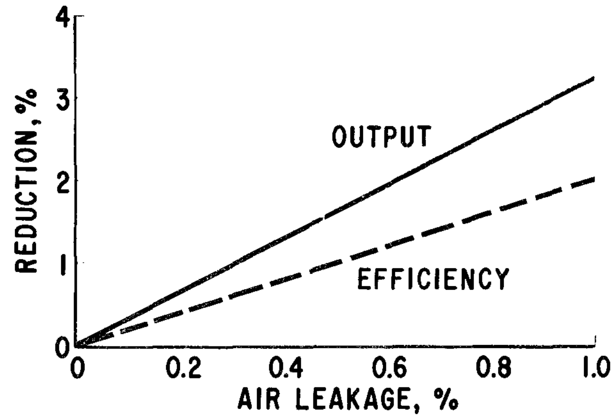


Figure 15

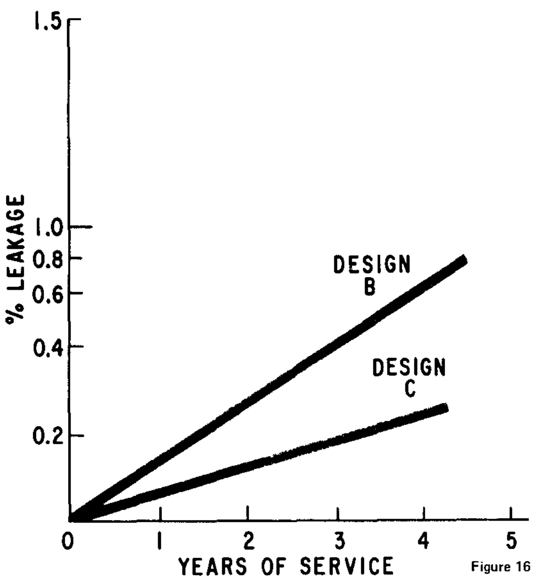


Figure 16

VERTICAL REGENERATOR

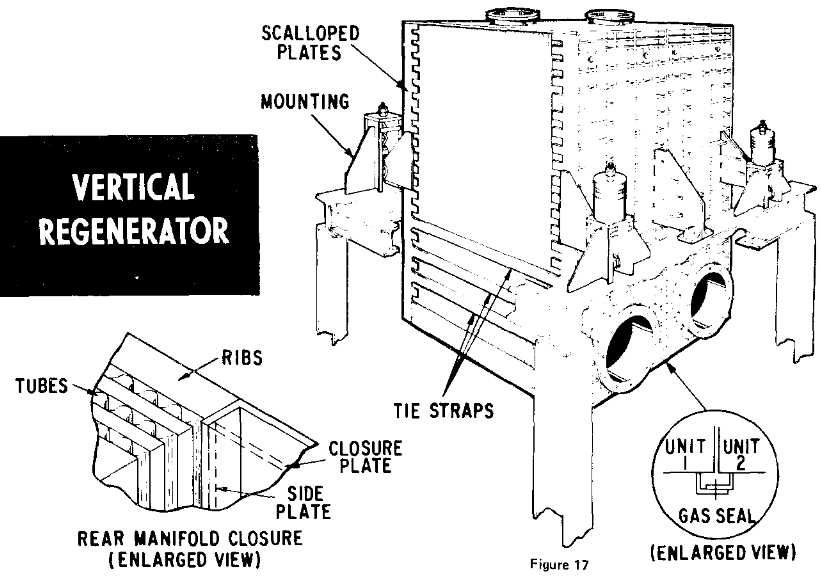


Figure 17

OUTDOOR PACKAGE GAS TURBINE

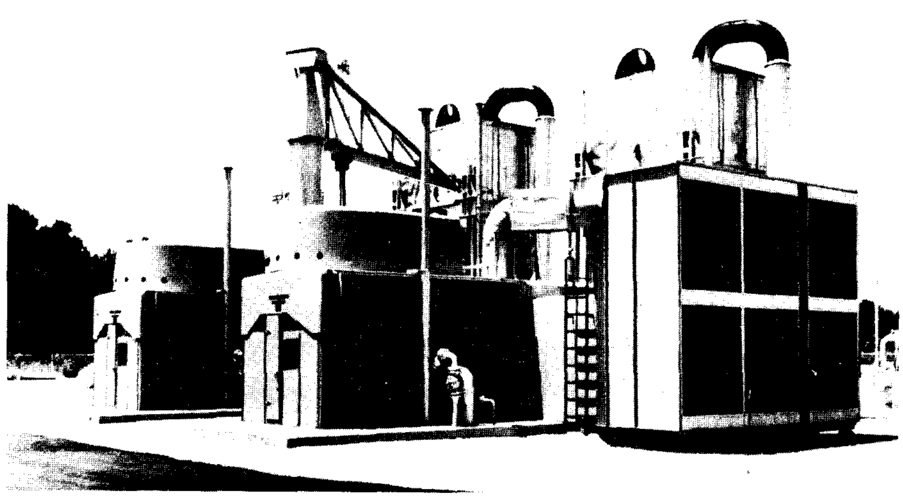


Figure 18