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Gas Turbine Recuperator Technology Advancements

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Because of intense development in the aircraft gas turbine field over the last 30 years, the fixed boundary recuperator has received much less development attention than the turbomachinery, and is still proving to be the nemesis of the small gas turbine design engineer. For operation on cheap fuel, such as natural gas, the simple cycle-engine is the obvious choice, but where more expensive liquid fuels are to be burned, the economics of gas turbine operation can be substantially improved by incorporating an efficient, reliable recuperator. For many industrial, vehicular, marine, and utility applications it can be shown that the gas turbine is a more attractive prime mover than either the diesel engine or steam turbine. For some military applications the fuel logistics situation shows the recuperative gas turbine to be the most effective power plant. For small nuclear Brayton cycle space power systems the recuperator is an essential component for high overall plant efficiency, and hence reduced thermal rejection to the environment. Data are presented to show that utilization of compact efficient heat transfer surfaces developed primarily for aerospace heat exchangers, can result in a substantial reduction in weight and volume, for industrial, vehicular, marine, and nuclear gas turbine recuperators. With the increase in overall efficiency of the recuperative cycle (depending on the level of thermal effectiveness, and the size and type of plant), the cost of the heat exchanger can often be paid for in fuel savings, after only a few hundred hours of operation. Heat exchanger surface geometries and fabrication techniques, together with specific recuperator sizes for different applications, are presented. Design, performance, structural, manufacturing, and economic aspects of compact heat exchanger technology, as applied to the gas turbine, are discussed in detail, together with projected future trends in this field.

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C. F. McDONALD

NOMENCLATURE

- a = minimum free flow area, sq ft
- A = exchanger heat-transfer surface area, sq ft
- A_{FR} = exchanger frontal area, sq ft
- b = passage height, in.
- C = flow stream capacity rate, Btu/hr-deg F
- C_P = specific heat at constant pressure, Btu/lb-deg F
- d = hydraulic diameter, tube diameter, in.
- f = fanning friction factor
- g = proportionality factor in Newton's Second Law, ft/sec^2
- h = heat-transfer coefficient, Btu/sq ft-hr-deg F
- hA = thermal conductance, Btu/hr-deg F
- hp = horsepower
- j = Colburn factor
- L = exchanger flow length, in.
- N = fins per inch
- NTU = number of heat transfer units
- P = absolute pressure, lb/sq in.
- Pr = Prandtl number, a fluid properties modulus
- Re = Reynolds number, a flow modulus
- SFC = specific fuel consumption, lb/hr-hp
- Sp hp = specific power, hp/lb/sec
- T = absolute temperature, deg R
- V = exchanger core volume, cu ft
- W = flow rate, lb/sec
- X_T = transverse tube pitch
- X_L = longitudinal tube pitch
- β = surface compactness, sq ft/cu ft
- Δ = denotes difference
- α = denotes proportionality
- ϵ = exchanger effectiveness
- μ = absolute viscosity, lb/hr-ft
- ρ = density, lb/cu ft
- $(\Delta P/P)$ = exchanger pressure loss, percent
- \$ = dollars
- ¢ = cents

In the last ten years, the gas turbine has become established as a reliable and competitive prime-mover in the industrial field. In the first 20 years of its development, the main emphasis had been in the aviation field, and commercial derivatives have benefited from concentrated military activities. In the vehicular, industrial, and power generating fields, the gasoline engine, diesel engine, and steam turbine, with more than 50 years of development, have been the dominant prime-movers, and have demonstrated high reliability and low maintenance costs. Initial plant costs and operating costs, over the life of the machine are important, and with gradual improvement in the turbomachinery components, together with material and manufacturing advancements, the operating cost of the industrial gas turbine is now competitive with the more traditional prime-movers. Highly rated aircraft gas turbines have achieved TBO's of 10,000 hr, and the more conservatively designed industrial plants have demonstrated values much higher than this.

The recuperator initially found acceptance in the large European industrial open- and closed-cycle gas turbines, introduced about 30 years ago. The recuperator was essentially a supplement to the cycle to provide reasonable specific fuel consumptions because of the low component efficiencies of that era. These recuperators, of mainly tubular construction, were designed and manufactured using conventional heat exchanger practice, and this resulted in large and bulky units which were incompatible with the compact nature of the rotating machinery. These plants were conservatively designed and have demonstrated a high degree of reliability and, in many cases, have run virtually maintenance free for over 100,000 hr of operation.

In the open-cycle gas turbine, the recuperator, unlike the ubiquitous compressor and turbine, is not an essential component, and the high initial cost of the heat exchanger has generally been the decisive factor in selecting the simple cycle plant. With an ever-increasing demand for power, and strong emphasis being placed on minimum pollutant emission, the gas turbine looks attractive for many applications. Although current initial plant cost is high, the dominant operating cost of the machine, over the majority of its life, is fuel cost, and, in many cases, the cost of the heat exchanger can be paid for in fuel savings after only a few hundred hours of operation. To be competitive with the diesel engine for industrial, vehicular, and marine application, some form of heat-recovery device in the gas turbine is necessary.

It is expected that the role of the recuperated gas turbine in the next ten years will become significant in the vehicular, industrial, marine, and closed-cycle nuclear gas turbine fields. In spite of a long history of heat exchanger utilization in large gas turbine plants, their success has been fairly limited for small industrial, transportable, or mobile gas turbines. The heat exchanger for a competitive small gas turbine must:

1 Be compact to maintain gas turbine superiority over the diesel engine

2 Have an initial cost low enough to provide overall economic justification

3 Withstand the severe thermal stresses during engine transient operation, and demonstrate a high degree of reliability for extended life

4 Not overly restrict the gas turbines multi-fuel capability.

For many years, the heat exchanger has been the nemesis of the small gas turbine design engineer, but with progressive development, promising results have been achieved by AiResearch with both compact plate-fin and tubular units. The main emphasis in this paper is to show that utilization of compact, efficient heat-transfer surfaces, and fabrication techniques, developed primarily for aerospace heat exchangers, can result in substantial reduction in weight, volume, and cost for vehicular, industrial, marine, and nuclear gas turbine recuperators.

It is beyond the scope of this paper to go into a detailed comparison between the fixed boundary metallic recuperator, and the rotary ceramic regenerator; however, some guidelines are included regarding the selection of gas turbine heat exchanger type. The first heat exchanged gas turbine to go into large volume production will probably be the regenerative engine for truck application in the 300- to 500-hp power range. Utilization of these prime-movers will likely be extended to non-vehicular markets, such as generator sets and varied marine applications. In these areas, consumers will become more aware of the low specific fuel consumption potential of the regenerative or recuperative cycle. For the many industrial operations currently using natural gas fuel, simple cycle gas turbines are utilized, but with the projected increase in fuel costs over the next few years, the economics of gas turbine operation will be substantially improved by incorporating an efficient, reliable recuperator.

Recent work done at AiResearch has shown that a viable, cost-effective recuperator can be produced, which satisfies the performance, structural, manufacturing, and economic goals for a wide range of gas turbine applications.

GAS TURBINE HEAT-RECOVERY SYSTEMS

An obstacle to the wider application of gas turbines has been the relatively low efficiency of simple-cycle units. Inherent in the simplicity of the cycle, and the metallurgical limitations imposed for practical operation, are large specific airflow and high exhaust gas temperature. Some degree of waste heat recovery in the gas turbine is imperative in order to achieve specific fuel consumptions comparable with the diesel engine. Various heat-recovery methods utilized to improve the thermal efficiency of the gas turbine are shown in Fig. 1 and are briefly outlined in the following.

Open Cycle with Recuperator or Regenerator

The simplest form of heat exchanged cycle utilizes a heat-recovery unit to transfer heat directly from the turbine exhaust to the compressor discharge air before it enters the combustor. Throughout this paper, the fixed boundary, direct-transfer heat exchanger is referred to as a recuperator, and the rotary, periodic flow type as a regenerator. For the open-cycle unit, either a recuperator or regenerator can be utilized. The main emphasis in this paper is placed on the recuperator, since it is felt that in the next few years, this type of heat exchanger will be used extensively for vehicular, industrial, marine, and closed-cycle nuclear gas turbines.

Exhaust Heated Cycle

In this type of cycle, the combustor is transposed to a position between the turbine and the heat exchanger. This results in an engine arrangement which is bulkier and less efficient than the first configuration discussed. The high effectiveness heat exchanger will require three or four times the surface area of the conventional type, and will be required to operate at a much higher average temperature. As outlined by Mordell (1),[1] the main attractive feature of the exhaust heated cycle is that the turbine operates on pure air, and this means that very low grade fuels can be used without incurring turbine-blade erosion, etc.

Liquid Coupled System

The liquid-coupled system consists of two independent heat exchangers, linked together by a liquid metal heat-transfer loop. This system affords considerable flexibility in packaging because it is comprised of two separate units, linked by a liquid-metal loop. A completely sealed liquid-metal system would be necessary to obviate oxide-induced corrosion within the loop, thus requiring the use of an electromagnetic pump. For industrial application, the principal disadvantages of the liquid-coupled system are uncertain reliability, complexity, high cost, and volume penalty.

Waste-Heat Cycle

An effective method of improving the thermal efficiency of the industrial gas turbine (for total energy applications) is to utilize a waste heat boiler. Because the turbine exhaust contains nearly all the original oxygen, supplementary firing between the turbine and the boiler can be adopted. The basic gas turbine can be either recuperative or non-recuperative. A combined waste-heat cycle employing a two-phase Rankine cycle, in conjunction with a gas turbine, has many advantages for industrial application. In this combined steam and gas (STAG) turbine, the closed-cycle receives heat from the gas turbine exhaust and contributes shaft power output through a common gear-box. These specialized topics have been reported in depth by Hendrickson (2) and Sheldon (3).

Closed-Cycle Gas Turbine Using Conventional Fuel

As reported by Keller (4), many closed air cycles have been used for power and heating application in the range of 2000 to 30,000 kw. These plants have utilized coal, oil, and natural gas fuels. More recently, a low pollution closed Brayton cycle gas turbine, using air as the working fluid, has been proposed by Pietsch (5) for vehicular application. A unique feature of this cycle is the combined heater/combustor which utilizes premixed fuel and air in a near stoichiometric ratio with a slight excess of air, such that the temperature is limited to minimize the formation of the oxides of nitrogen, while essentially burning all of the hydrocarbons and forming only carbon dioxide.

In these closed-cycle gas turbines, the recuperator is an essential component for high overall plant efficiency, and reduced thermal rejection to environment.

Closed-Cycle Nuclear Gas Turbines

As reported by Keller (4), a development of gas-cooled reactors with high outlet temperatures can be used in conjunction with gas tur-

[1] Underlined numbers in parentheses designate References at end of paper.

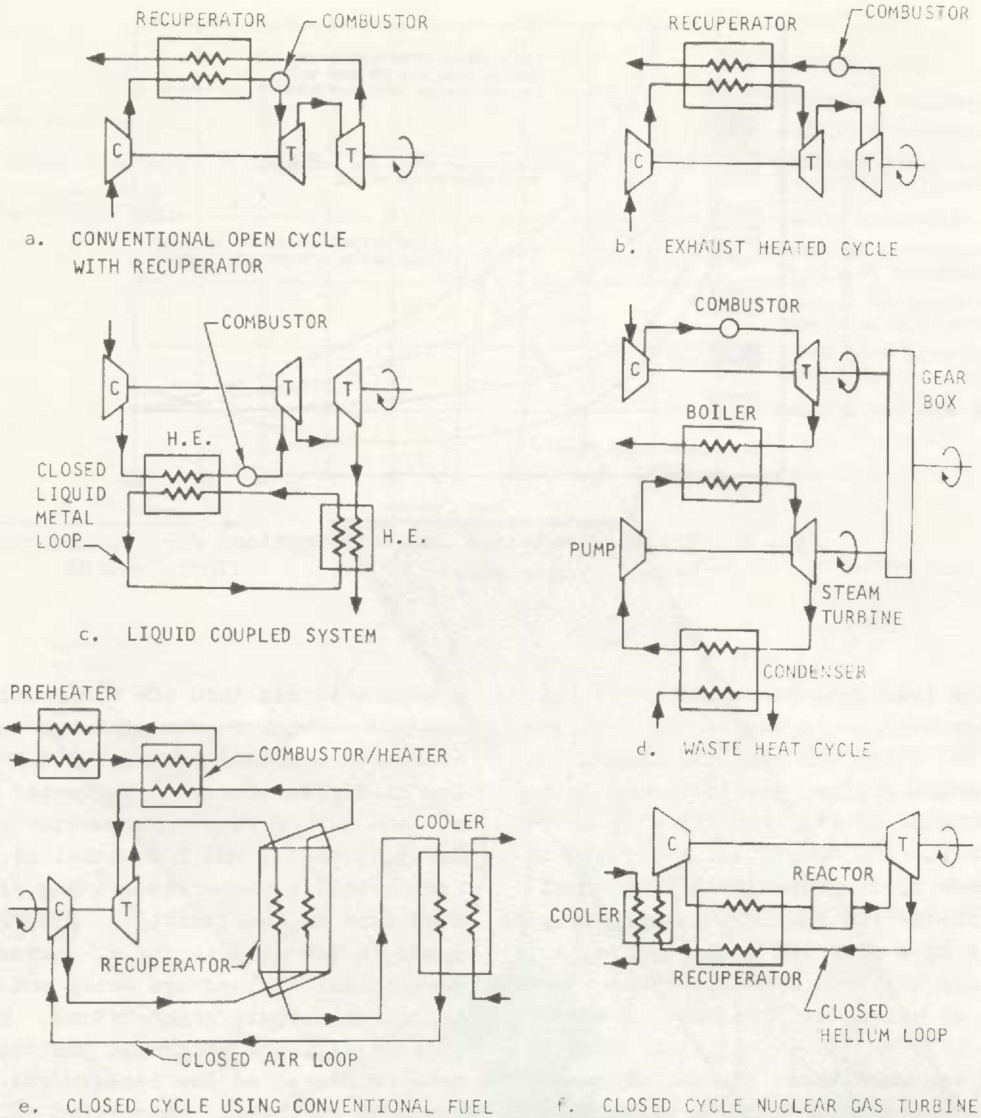


Fig. 1 Heat exchanged gas turbine cycles

bines in a direct circuit, and can thus be utilized without additional pressure and temperature losses, such as are inevitable in the case of intermediate heat exchangers. The high cycle pressure level in these helium gas plants leads to a favorable heat-transfer coefficient, and the recuperator for the closed-cycle gas turbine is in the order of one-third the size of that in air plants. Again, the recuperator is an essential component for high plant efficiency and, hence, reduced thermal rejection to the environment. The cooling water requirement for a closed-cycle helium gas turbine is 1/3 to 1/5 of a comparative steam turbine plant.

SELECTION OF HEAT EXCHANGER TYPE

The specific fuel consumption of a gas turbine can be substantially reduced by applying heat energy from the exhaust gas to preheat the compressed air, thus reducing the temperature rise in the combustion chamber. For vehicular gas turbines the heat exchanger is an essential, if not the most important, component of the power plant. The most important benefit of the heat exchanger, particularly for vehicular application, is the marked reduction in part load fuel consumption. For two-shaft vehicular gas turbines with a free power turbine, a fairly flat fuel consumption curve with load can be realized utilizing a variable power turbine nozzle. Fig. 2 shows a typical comparison in fuel consumption between a high-pressure ratio simple-cycle engine, low-pressure ratio heat-exchanged engines, with and without variable geometry, and a turbo-charged diesel engine. It should be noted that the im-

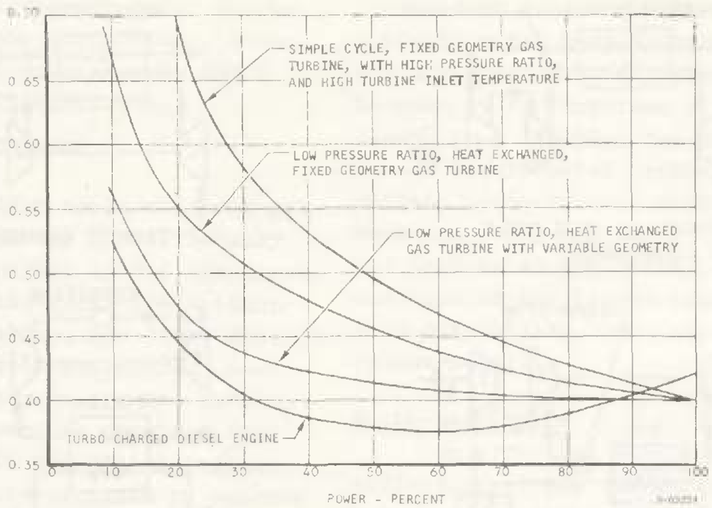


Fig. 2 Typical part-load fuel consumptions for various power plants

provement in part load fuel consumption for the variable geometry machine is due to the interaction of nozzle variation and heat exchanger. In a non-heat-exchanged engine, the influence on part load fuel consumption of variable geometry is significantly less. The beneficial effect of increases in maximum cycle temperature on thermal efficiency is greater for the heat-exchanged cycle. The desirability of a flat SFC curve for vehicular and industrial gas turbines emphasizes the vital need to develop an efficient, reliable, low-cost heat exchanger.

There are two main types of heat exchanger currently being used for gas turbine applications. One is the fixed boundary recuperator, which is often referred to as a conventional direct-transfer heat exchanger, in which the compressor discharge air and the turbine exhaust gas exchange thermal energy directly through, and are separated by, the heat-transfer surface itself. The other is a periodic flow regenerator, like the common Ljungstrom air preheater, in which the heat is alternately absorbed and rejected by a mass of material which rotates through fixed fluid streams, and is exposed periodically to the high-temperature gas and low-temperature air.

The rotary heat exchanger has been utilized for many years in low-pressure air heating systems in the form of the Ljungstrom type. This technology has been extended to regenerator application for vehicular and industrial gas turbine use. One of the projected advantages of the rotary unit was the use of more compact surfaces than the fixed boundary type, to give a reduced heat exchanger volume, for a given effectiveness and pressure drop. Operating at higher differential pressure levels than the boiler air preheater, the gas turbine regenerator has the problem of excessive wear and leakage of the seals between the high-pressure cold compressor discharge air, and the low-pressure hot turbine exhaust gas. Early vehicular and industrial gas turbines utilized metallic regenerator units of both drum and disk type of construction. With recent developments in the ceramic field, current vehicular development engines are being endurance run with twin disk ceramic regenerators. Even after extensive development programs, the metallic-to-ceramic seal problems, of low leakage and long life, have not been resolved. Many types of regenerators have been evaluated over the years, and these include ceramic and metallic disks and drums, toroidal configurations, heat pipe designs, and pebble bed types. It is not the purpose of this paper to fully describe the rotary regenerator, since the many variants have been reported in depth by Penny (6), (7), Lanning (8), Weber (9), Selford (10), Silverstein (11), and Hammond (12).

Over the last 30 years, fixed boundary recuperator, of both tubular and plate-fin construction, has been used for industrial, marine, locomotive, aircraft, vehicular, and closed-cycle applications, and has demonstrated a high degree of reliability. The type of heat exchanger to be used, and the material from which it is constructed, essentially depends upon the cycle selected and the application of the gas turbine. Approximate boundaries for gas turbine heat exchanger selection are shown in Fig. 3. The basic curve array relates turbine inlet temperatures, compressor pressure ratio, and heat exchanger gas inlet temperature. Superimposed on this map are bound-

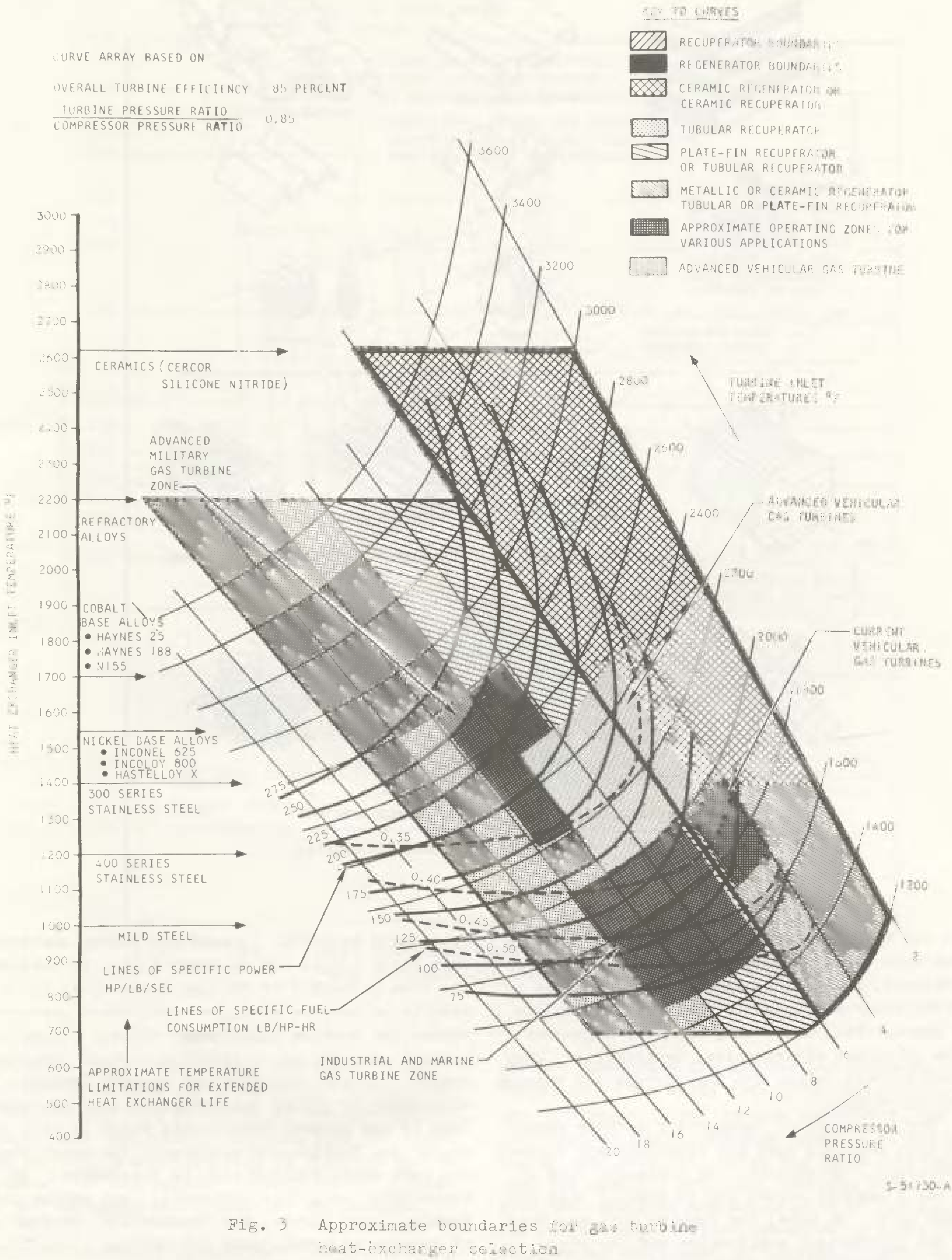


Fig. 3 Approximate boundaries for gas turbine heat-exchanger selection

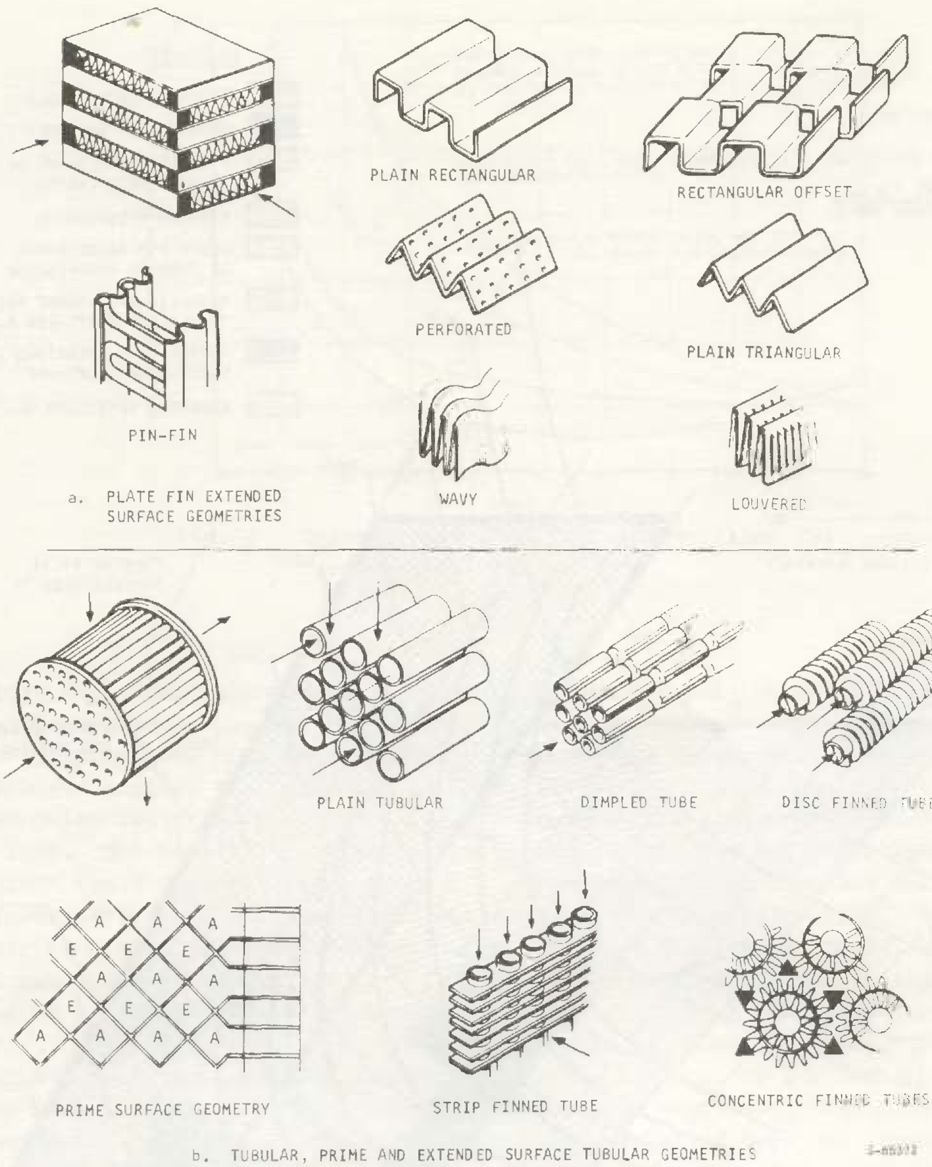


Fig. 4 Recuperator surface geometries

aries for the different heat exchanger types, together with approximate operating temperature limitations, for various materials. The lines of specific power and specific fuel consumption are only approximate, since they are very dependent on the component efficiencies; nevertheless, they do show typical values for the various gas turbine operating regimes.

The two types of recuperator surface geometries most commonly used are plate-fin and tubular construction. In general, the plate-fin type of unit has a smaller volume but is heavier than the tubular unit. To date, the tubular unit has been used for lightweight application and for high-pressure industrial and closed-cycle gas turbine plant. The regenerator, made from either metal or ceramic material, is used for fairly low-pressure ratio cycles (up to about 6:1). To realize the true potential of the gas turbine, it is desirable to operate at higher compressor pressure ratios and turbine inlet temperatures. These increased specific power cycles introduce structural complexity, and seal wear and leakage problems, which tend to offset the potential low-cost advantage of the ceramic regenerator which is best suited for gas turbines of relatively low power that are used where limited life is acceptable. In heavy-duty, vehicular industrial, and marine gas turbine power plants, the regenerator, whether made of ceramic or metal, does not look as attractive as the fixed boundary recuperator.

Clearly, there is no single answer to the

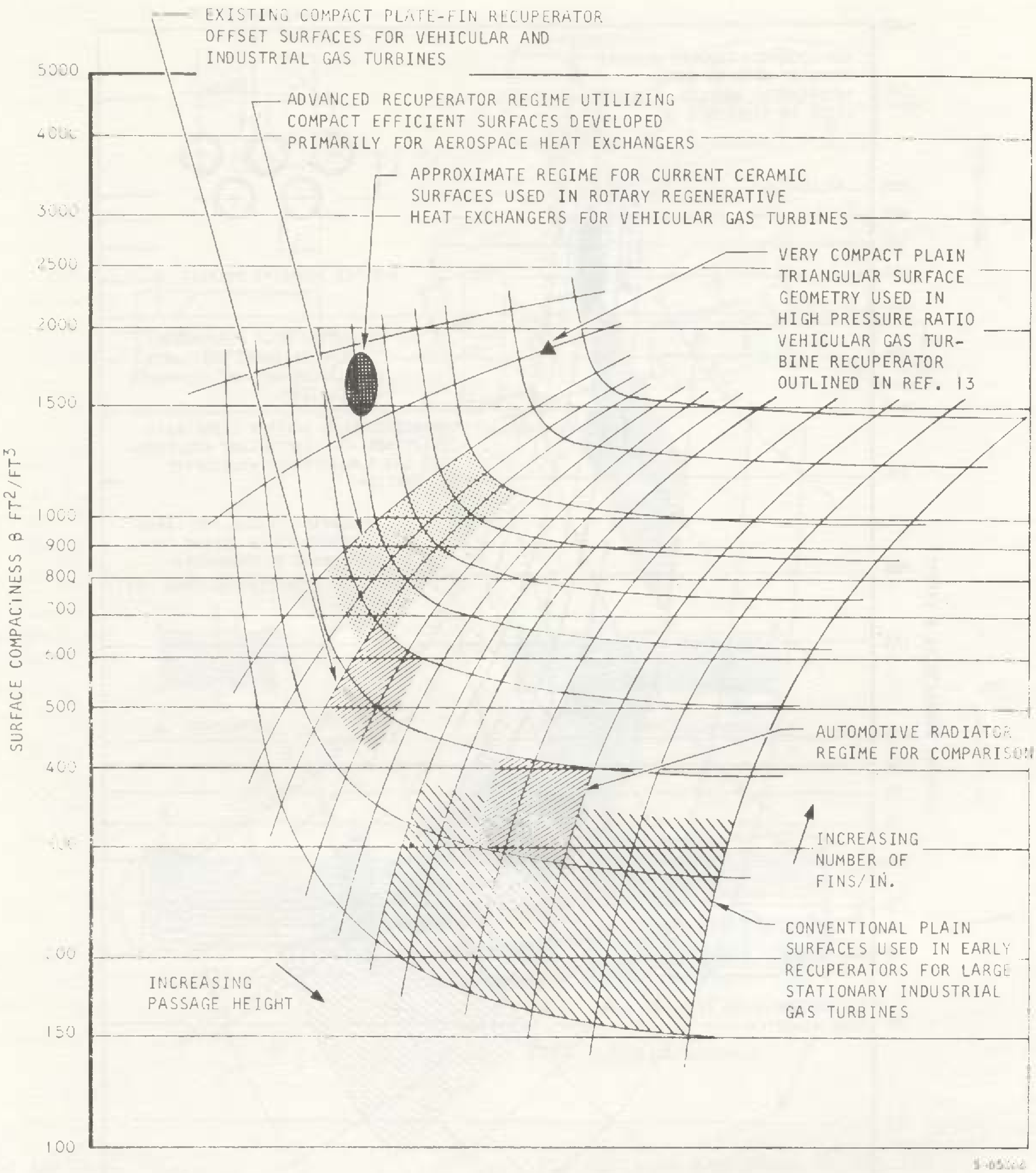


Fig. 5 Heat exchanger surface compactness spectrum for plate-fin recuperator geometries

question of what type of heat exchanger is the most cost-effective for a particular application. To realize the full potential of the gas turbine, considerable development efforts, with materials such as silicon nitride, will be required before a viable low cost ceramic recuperator is available for vehicular and industrial application.

RECUPERATOR SURFACE GEOMETRIES

The two most commonly used recuperator surface geometries are tubular and plate-fin constructions. The early industrial heavy-duty gas turbines utilized tubular geometries, and construction techniques were based on established boiler code practice, where the tubes were welded into the headers, and this, combined with the poor heat-transfer characteristics of the plain low compactness surfaces, resulted in heavy and bulky units which were not compatible with the turbomachinery. Concentrated developments in the aircraft heat exchanger fields have yielded

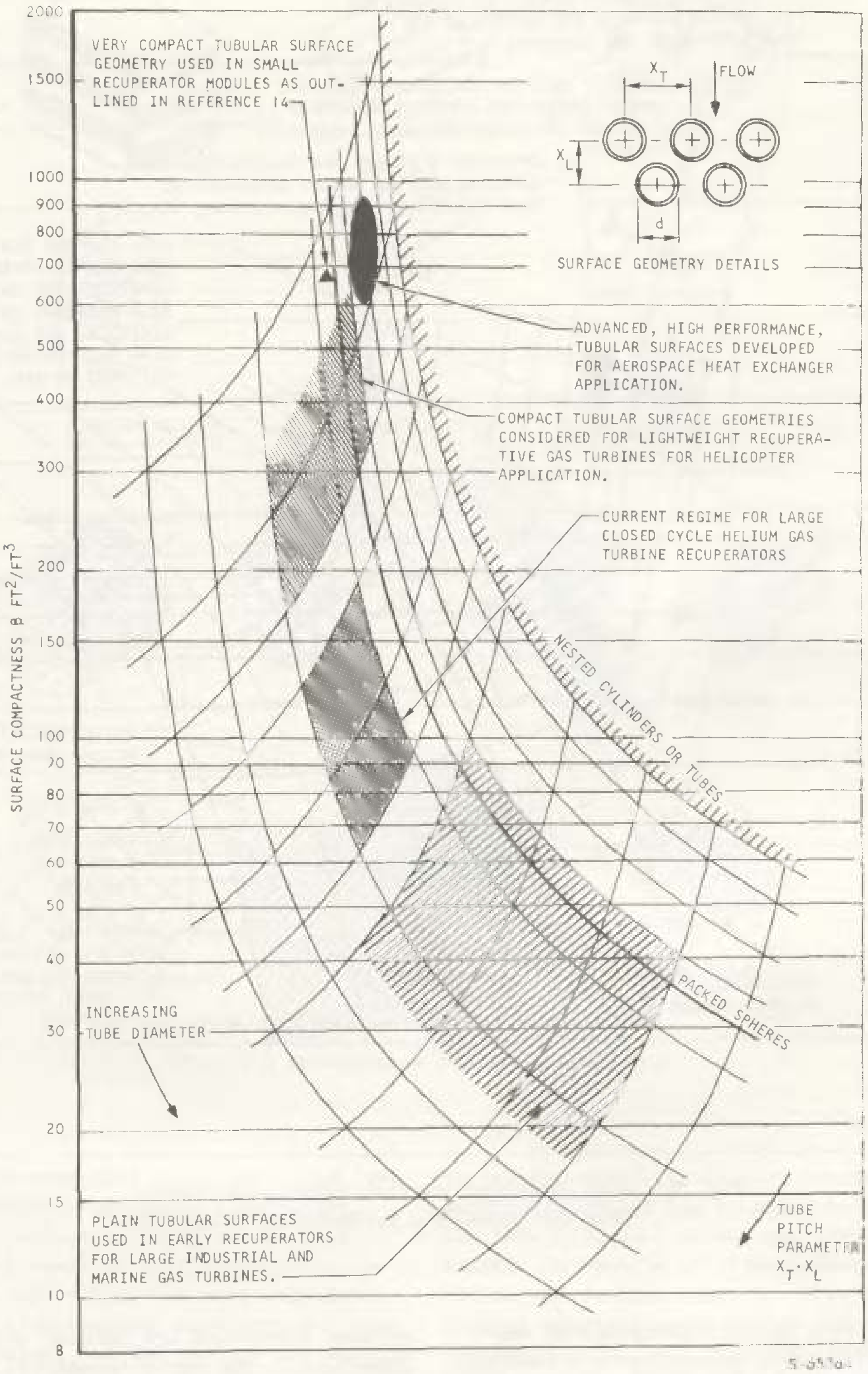


Fig. 6 Heat-exchanger surface compactness spectrum for tubular recuperator geometries

efficient surface geometries, of both plate-fin and tubular types, which have been successfully used in compact recuperators for industrial, vehicular, and aircraft gas turbines. Representative heat exchanger surface geometries are shown in Fig. 4, and are briefly discussed in the following. Heat-transfer and performance aspects of the various surface geometries are discussed

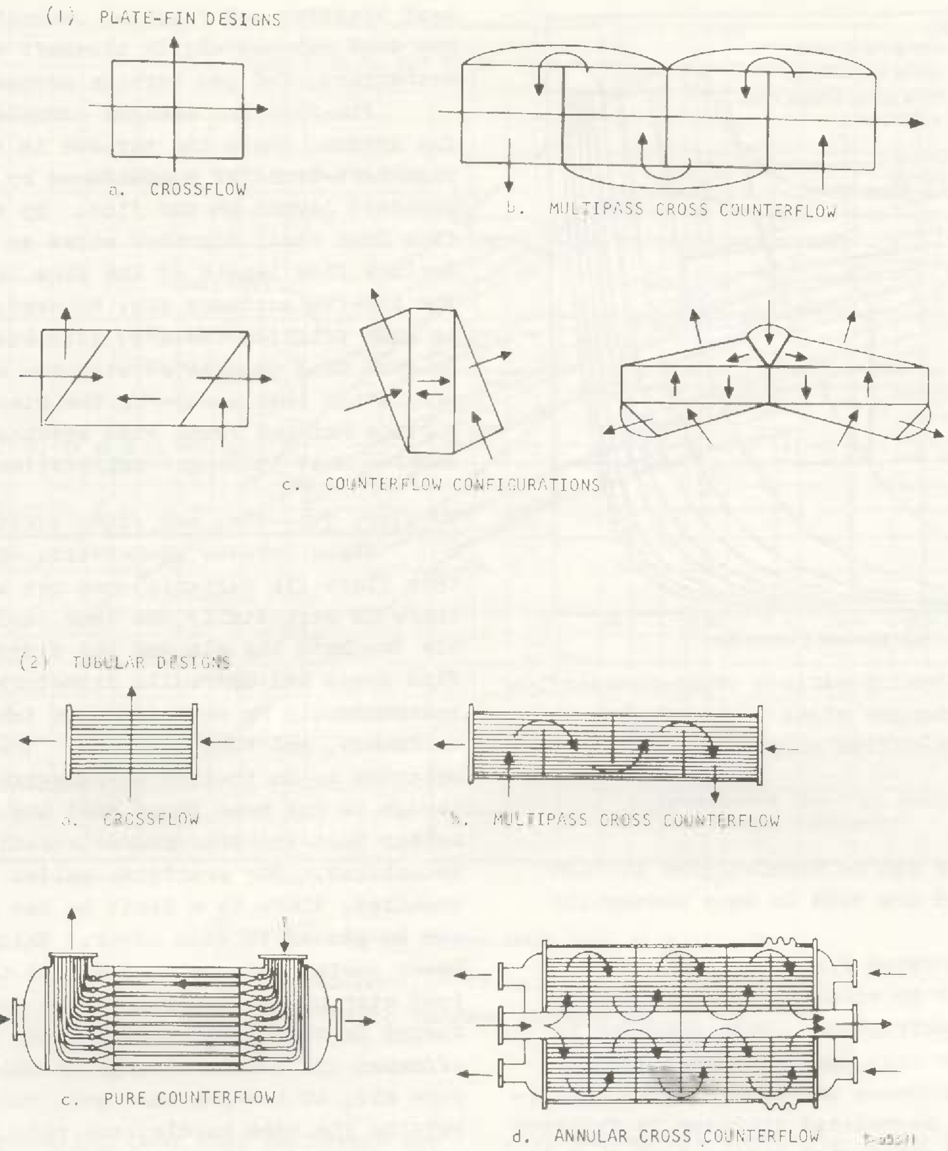


Fig. 7 Recuperator flow configurations

in a later section.

Plate-Fin Extended Surface Geometries

In this type of construction, the heat transfer takes place between parallel flat plates or channels which incorporate finned secondary surfaces. Considerable flexibility is afforded with this type of geometry in that the air and gas side passage heights and fin form can be varied independently to give an optimized unit for any particular application, the actual choice being dictated by the compactness one can achieve within established design and cost limitations. The most interesting development in plate-fin designs are what can be achieved in the fin form, in the way of promoting heat transfer. Several fin surfaces are shown in Fig. 4(a).

The simplest forms are the plain rectangular and triangular fins, but these are rarely used in recuperators because of their inherently poor heat-transfer characteristics which result in units of excessive volume, weight, and cost. Several simple modifications can be made to these geometries to enhance their heat-transfer characteristics.

In the wavy or herringbone fin, the geometry is formed such that the periodic change in fluid direction breaks up the boundary layer so as to yield higher heat-transfer coefficients. Louvered fin surfaces are characterized by fins that have been cut and bent out into the flow stream at frequent intervals, again to energize the boundary

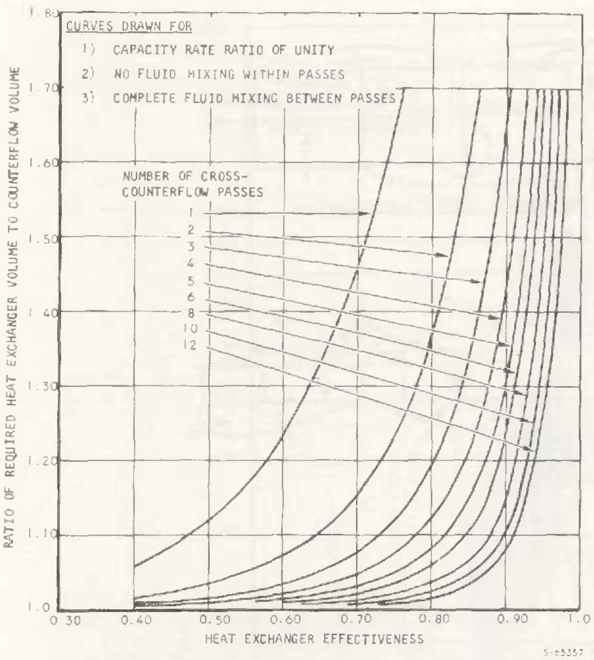


Fig. 8 Curves showing various cross-counterflow heat exchanger sizes compared with a pure counterflow configuration

layer. These fins can be manufactured at high rates of speed and are used in many automobile radiators.

In the perforated fin, holes or slots are cut out of the fin to enhance heat transfer by boundary-layer interruption. Work reported in the literature for very small chemically etched slots showed an increase in heat-transfer coefficient without the associated increase in friction factor. This was attributed to the fact that the thermal boundary layer was disturbed to a greater degree than the hydrodynamic boundary layer. More practical perforated fin designs have been developed by AiResearch in an attempt to minimize frontal area and volume requirements of a Rankine cycle engine condenser for automotive application.

The offset fins are similar in principle to the louvered fin surfaces, the only difference being that the short sections of the fins are aligned entirely with the flow direction. With this strip-fin type of configuration, it is feasible to have very short flow length fins, and thus very high heat-transfer conductances. The performance of these fins, especially the friction factor, is affected by the thickness and character of the fin leading edge. A few ten-thousandths of an inch of scarfing can have a considerable effect on the characteristics of the fin, particularly for very compact surface geometries. AiResearch offset fins exhibit very good heat transfer and friction characteristics and are used extensively in aircraft air-to-air heat exchangers, and gas turbine recuperators.

Pin-fins are another example of the plate-fin system, where the purpose is to achieve very high heat-transfer conductance by maintaining thin boundary layers on the fins. By constructing the fins from small diameter wires as shown, the effective flow length of the fins is very small. The pin-fin surfaces are, however, characterized by high friction factors, attributable primarily to form drag associated with the boundary-layer separation that occurs on the pins. This type of surface has not found wide acceptance for gas turbine heat exchanger application.

Tubular, Tube-Fin, and Prime Surface Geometries

Plain tubular geometries, while simpler than plate-fin variants, are not as flexible since there is essentially the same surface area available for both the air and gas sides of the unit. Flow areas and hydraulic diameters can be changed independently by variations in tube diameter, wall thickness, and tube pitching. The tubes can be oriented in an in-line or staggered pattern, although it has been found that the latter gives better heat-transfer characteristics and bundle geometries. For practical welded and brazed assemblies, there is a limit to how close the tubes can be placed to each other. This results in much lower surface compactness values than can be realized with plate-fin geometries, and will be discussed in the next section. Some flexibility is afforded the tubular design in that the high-pressure air, or low-pressure gas, can flow inside or outside the tube bundle, and this is discussed in a later section comparing the merits of various flow configurations. Various tubular surface geometries are illustrated in Fig. 4(b).

The plain tubular surface geometry has found acceptance in a variety of liquid and gas heat exchangers. For air-to-air units, the main disadvantage of plain tube designs is the poor internal heat-transfer coefficient because of boundary-layer buildup, compared with the much higher external coefficient (because of vortex shedding from the staggered tube rows). For pressure drop limited designs, this effect, combined with the fact that the internal and external areas are essentially the same, results in units of large volume and unattractive geometry proportions because of the thermal conductance unbalance.

For liquid applications (where large pressure losses can be tolerated), the internal coefficient can be improved by using internal strips or turbulators. For recuperator applications, one of the best methods of internal heat-transfer

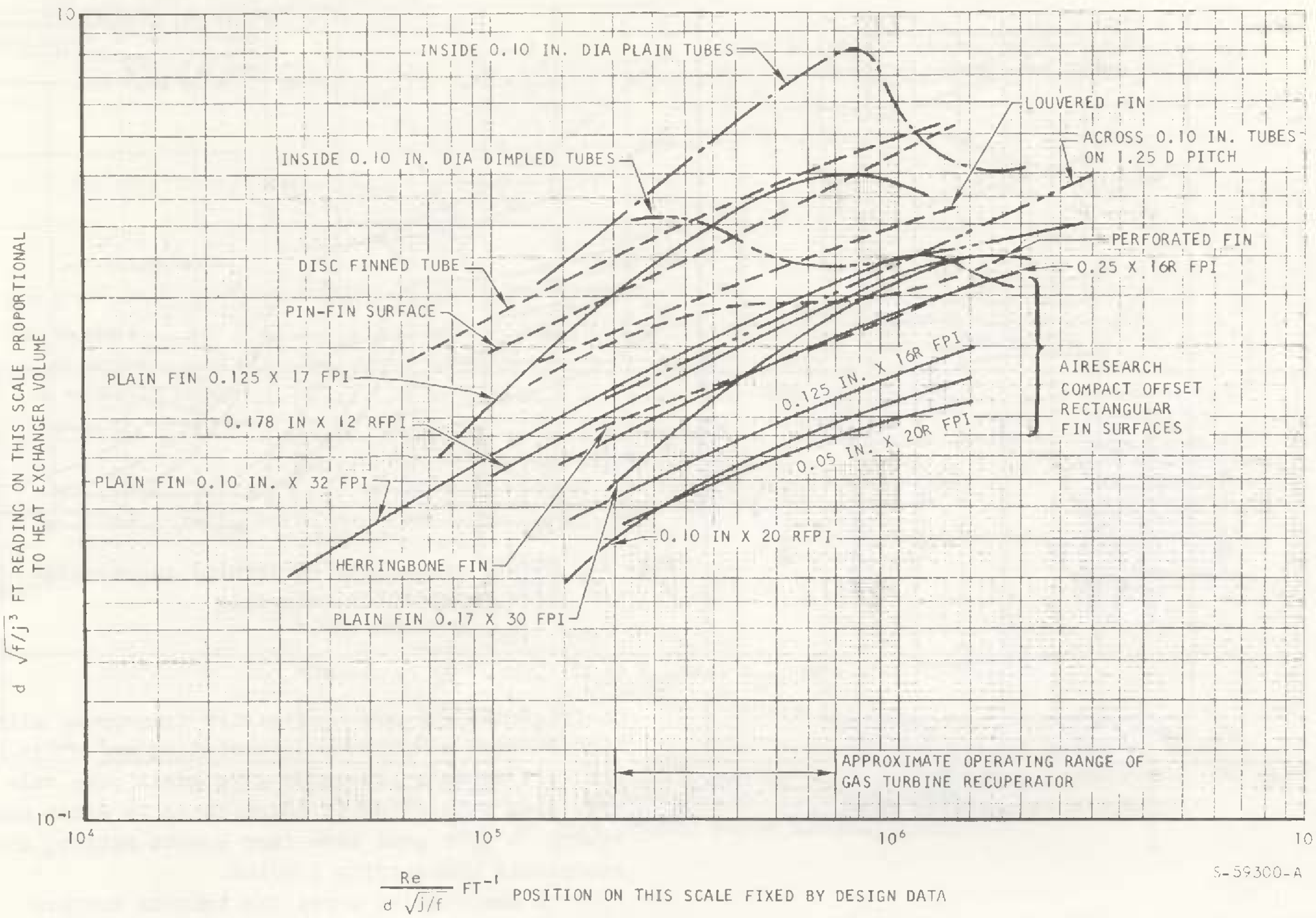


Fig. 9 Quasi-nondimensional comparison of heat-exchanger surface geometries

coefficient augmentation (one that maximizes the ratio of heat-transfer conductance to fluid pumping power) is simple ring dimpling of the tube wall. Local dimpling of the tube wall provides the necessary boundary-layer dissolution to give substantially improved coefficients over a wide range of Reynolds numbers.

For many applications, the addition of secondary surfaces in the form of disk or strip fins can improve the heat exchanger performance. For recuperator application, finned tube geometries are not very attractive, primarily because finning of the tubes adds area to the side of the heat exchanger that already has the higher conductance (i.e., flow across the bundle). The addition of heat-transfer area to the high-conductance side represents an inefficient use of surface area, and thus results in a high weight, expensive design.

In some high-pressure, closed-cycle gas turbines, recuperators of pure counterflow configuration have been successfully demonstrated with concentric strip fin tubes formed as shown into a compact bundle. To prevent bypass, small inserts are necessary with the equilateral triangular tube pitching arrangement. A design requirement, of balanced thermal conductances (for minimum core volume), can be realized with this type of tubular construction.

Several platular prime surface geometries of both brazed and welded construction have been evaluated for gas turbine recuperator application. This type of surface is characterized by a series of formed elements which, when laid together, form the tube functioning channel. Because of the poor heat-transfer coefficients associated with the plain formed channels, and structural problems that have not been fully resolved, the prime surface geometries have not found acceptance for compact, high-performance, low-cost, gas turbine recuperators.

For each application, a wide choice of surface geometries must be reviewed so that a series of recuperator designs can be reviewed for propor-

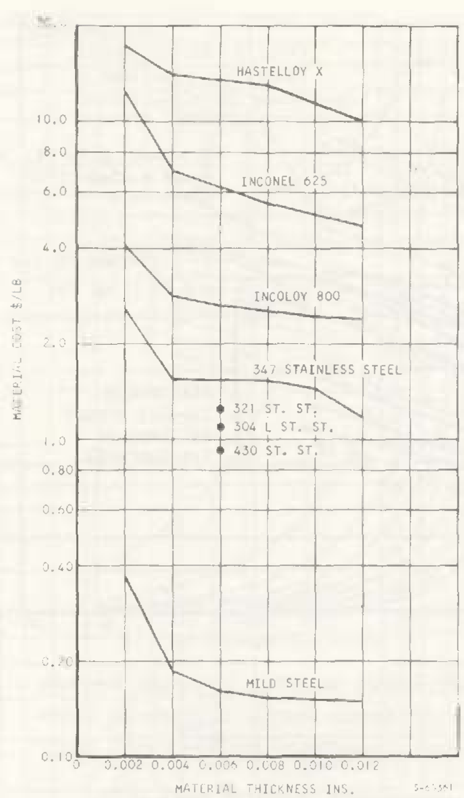


Fig. 10 Sheet metal costs for a range of representative recuperator materials

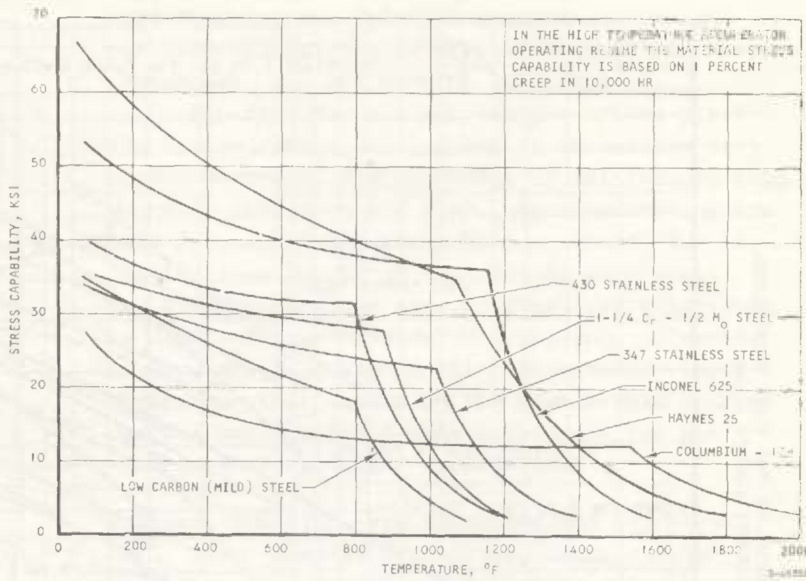


Fig. 11 Stress capability of several representative recuperator materials

tion, cost, and packaging possibilities.

SURFACE GEOMETRY SURFACE COMPACTNESS

A compact heat exchanger is one not necessarily of small bulk and weight, but one which incorporates a heat-transfer surface having a high area density per unit of volume (β). Somewhat arbitrarily, a compact surface can be defined as one with an area density greater than 200 sq ft/cu ft.

A plot of the range of surface compactness for plate-fin surface geometries is shown in Fig. 5. Early plate-fin units had β values around 300 sq ft/cu ft, and this compares with typical automobile radiator surface geometries. Existing industrial recuperators have β values in the order of 600 sq ft/cu ft, and in advanced units, this will probably increase to around 1000 sq ft/cu ft. A very compact recuperator described by Topouzian (13) had a surface compactness approaching 2000 sq ft/cu ft, which is close to the ceramic surfaces being developed for rotary regenerator application. Efficient surfaces with compactness values approaching 2000 sq ft/cu ft have been used in AiResearch heat exchangers for aircraft and cryogenic applications. High heat-transfer coefficients are not necessarily synonymous with high surface compactness geometries, and while the very dense surfaces do give small core volume, some relaxation in compactness is often necessary to give good core face aspect ratios, and acceptable counterflow lengths.

A compactness array for tubular surface geometries is shown in Fig. 6, and it can be seen that early tubular recuperators had β values in the order of 40 sq ft/cu ft. The current large plain tubular recuperators being built for closed-cycle helium gas turbine plates have β values in the order of 100 sq ft/cu ft. Compact tubular geometries used for lightweight recuperative gas turbines for helicopter application have surface compactness around 400 sq ft/cu ft. A very compact tubular recuperator surface geometry described by Wheeler (14) had a β value approaching 700 sq ft/cu ft, which is comparable with advanced surfaces used in lightweight aerospace heat exchangers. Use of very small hydraulic diameters implies a large number of tubes (and welded and brazed joints, etc.), and, as in the case of plate-fin designs, some relaxation is often necessary to get practical core shapes. The limiting compactness boundary for nested cylinders or tubes can be clearly seen on the curve array. Also, a line is superimposed on the compactness map for randomly packed sphere beds, with an average voidage of 40 percent.

From the curves, it can be clearly seen that there is an overlap between the geometries of different types. The motivation for using compact surfaces is to gain high heat-transfer

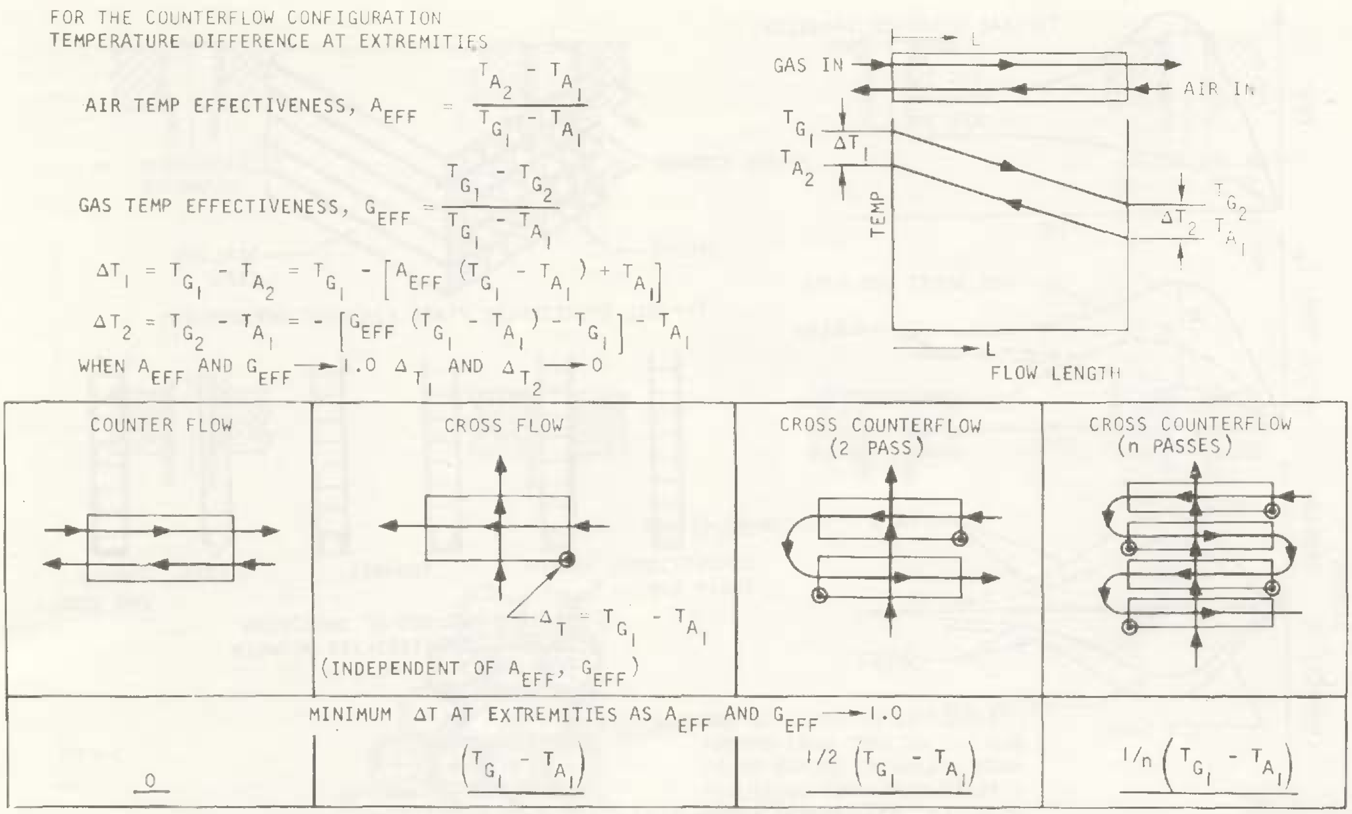


Fig. 12 Effect of heat-exchanger flow configuration on core steady state thermal gradients

performance in a minimum package, with minimum weight and cost. Generally speaking, compact plate-fin surfaces have a much lower cost per square foot of transfer area, and moreover have a higher conductance than the less compact tubular surfaces. Because of the higher ratio of friction factor to heat-transfer coefficient (f/j), the very compact surfaces tend to have smaller dimensions in the flow direction and, consequently, higher temperature gradients in the flow direction.

RECUPERATOR FLOW CONFIGURATIONS

For both plate-fin and tubular recuperators, a wide range of flow arrangements can be utilized, the choice being much dependent on effectiveness level, turbomachinery gas flow paths, and packaging envelope. Typical flow configurations are shown in Fig. 7. Because of the limiting effectiveness level of 50 percent (at a capacity rate ratio of unity), parallel flow arrangements are not normally used. An exception, however, would be for very high-temperature application (exhaust heated cycle, for example), where, by using parallel flow, the metal temperature in the heat exchanger can be kept substantially below the gas inlet temperature.

For the plate-fin variant, either crossflow, cross counterflow, or counterflow configurations can be used. The single pass crossflow and multipass cross counterflow arrangements are simple, in that the heat exchanger is virtually self-headering. As will be mentioned later, the simple crossflow unit is rarely used, because, in addition to the volume penalty at high effectiveness levels, the large temperature gradients in the core adversely affect the structural integrity. In Fig. 8, the ratio of heat exchanger volume to counterflow volume is shown as a function of effectiveness and number of passes for a cross counterflow arrangement. From the figure, it can be seen that at high levels of effectiveness, where most recuperators are designed to operate, a large penalty in exchanger volume has to be paid if flow configuration, other than counterflow, was used. Several plate-fin counterflow variants are shown in Fig. 7. With a counterflow configuration, end headering sections are necessary to direct the gas and air to and from the pure counterflow portion of the core. These triangu-

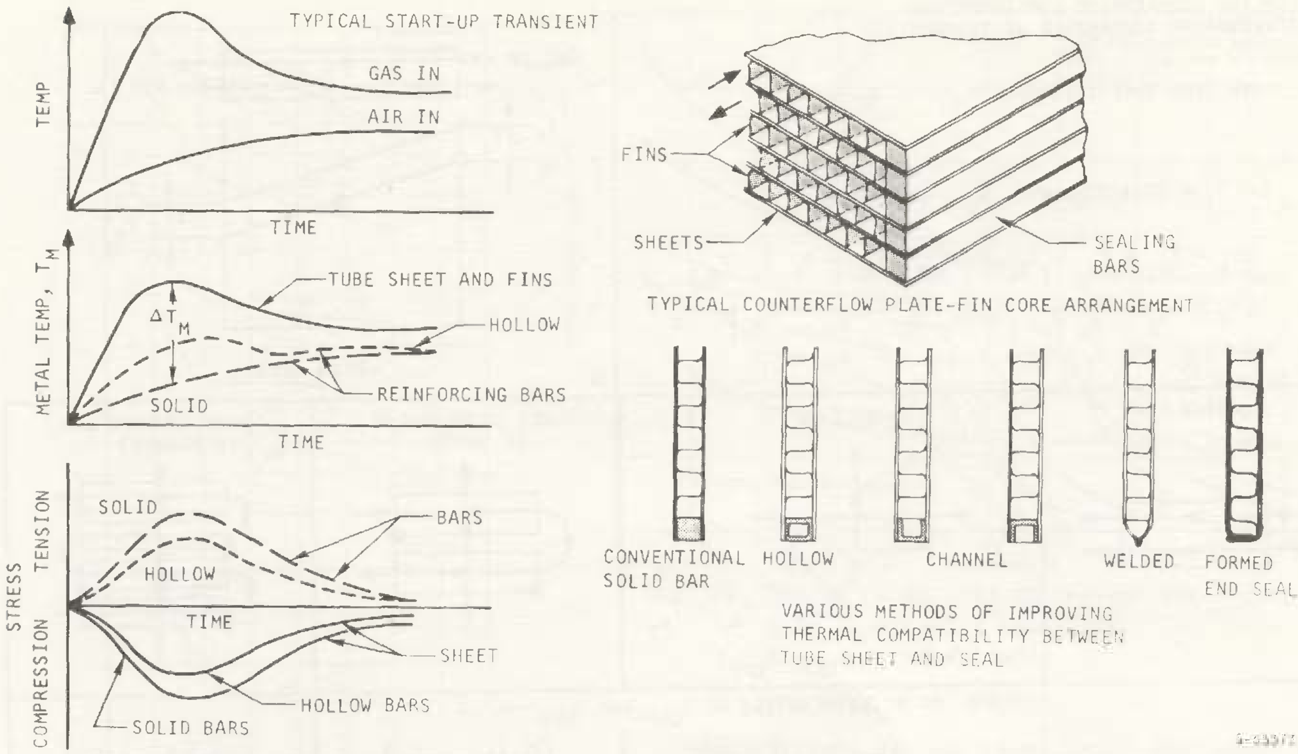


Fig. 13 Transient thermal stresses in plate-fin recuperator core matrix

lar or rectangular sections are integrated with the counterflow portion, and the core can be fabricated as one assembly. To minimize pressure loss, and yet satisfy structural requirements, low compactness geometries are used in the end sections, and this, combined with the essentially crossflow arrangement in these areas, implies that very little of the overall thermal duty takes place in the headering sections. Detailed analyses are nevertheless required in these areas to give minimum volume headers that satisfy the structural, low-pressure loss, and good airflow distribution requirements necessary for high effectiveness level units.

For similar reasons to those discussed in the foregoing, single plass crossflow tubular units are rarely considered. For plain tube designs, the pure counterflow configuration is not attractive because of the low heat-transfer coefficients associated with the axial flow outside the tubes. This type of flow configuration has been used in large closed-cycle gas turbine recuperators where the concentric tube type of geometries, with finned secondary surfaces, are assembled in the form of tube nests. This type of construction results in large core volumes and is not acceptable for compact, small gas turbines.

For industrial and aircraft gas turbine application, the most commonly used flow configuration is multipass cross counterflow.

A degree of flexibility exists as regards whether the high-pressure air should flow inside the tubes or across the tube bundle. Where the high-pressure air flows through the plain tubes, their number will be small and their length large, owing to the fact that the flow resistance of the tube bundle crossed by the low density gas is high, so that small velocities and, hence, large flow areas are needed for this gas. This arrangement, nevertheless, can yield low volume, lightweight designs of practical dimensions by utilizing ring-dimpled tubes to improve the inside heat-transfer coefficient. Since the exhaust gas is the lower pressure fluid, lighter weight recuperators generally result with the exhaust gas outside the tubes, because in addition to the above heat-transfer considerations, the shell weight will be lower, and the tubes will have thinner walls when loaded in tension with internal pressure than when loaded with external pressure.

The baffles separating the shell side passes can be either arranged such that the flow is deflected within the core or in annular passages at the bundle outer diameter. Perhaps the most prac-

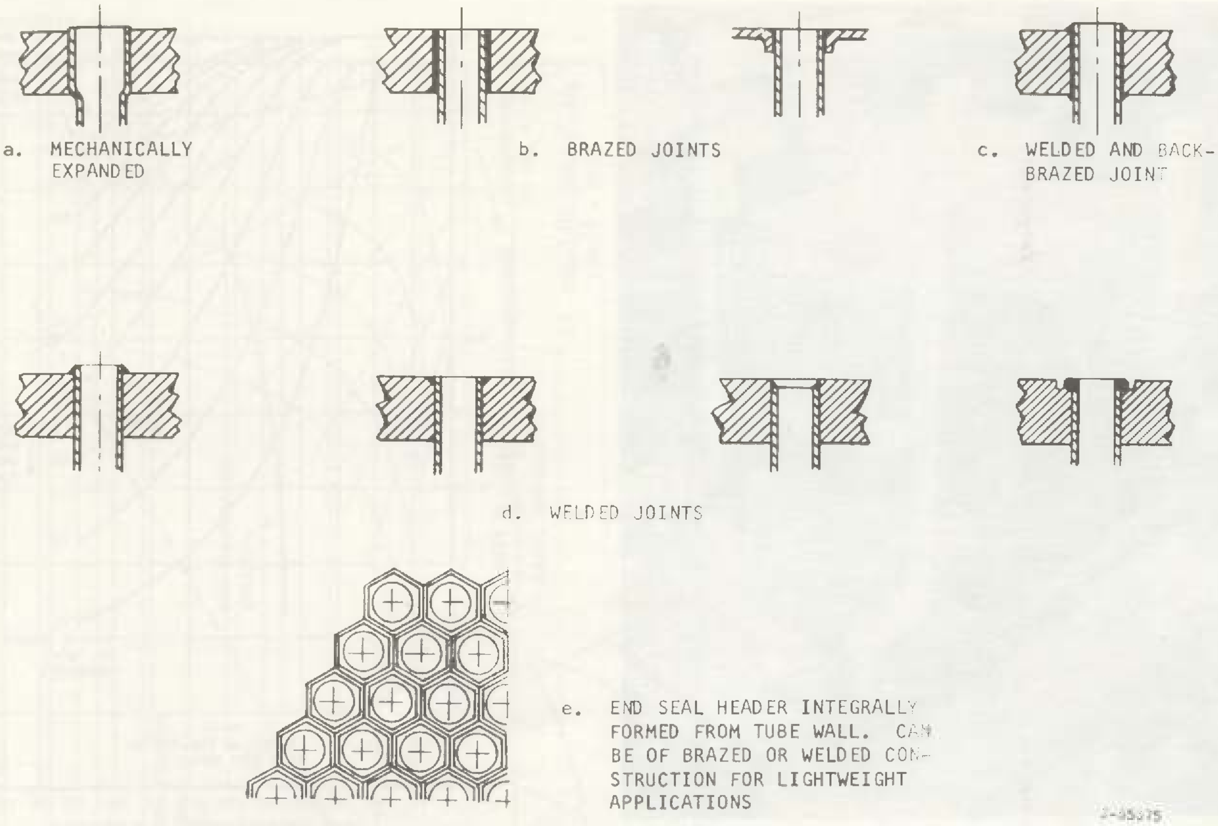


Fig. 14 Typical end seals used in tubular heat exchangers

tical flow configuration is the annular design shown in Fig. 7. In this type of flow pattern, the interpass turns occur in plain annular passages at the core inner and outer diameters. This type of design is practical for large mass flow applications (such as the helium closed-cycle gas turbine), since large flow frontal areas can be accommodated in a small volume. The number of shell side passes can be increased (hence approach a pure counterflow design for high effectiveness levels) to give low-temperature gradients per pass and minimize the stresses resulting from the variation in tube wall metal temperature throughout the annular bundle.

There is obviously no single criteria which dictates the flow configuration, but it has been shown that counterflow, or multipass cross counterflow, is necessary for high effectiveness application. The counterflow design, whith axial temperature gradients in the flow direction only, is the best from the structural standpoint, and this will be discussed in a later section.

HEAT-TRANSFER CONSIDERATIONS

The design of a gas turbine heat exchanger involves detailed considerations of both heat transfer between the fluids, and the pumping power expended to overcome fluid friction in moving the fluid through the matrix elements. The process of choosing the optimum recuperator cannot be independently carried out, but must include considerations of the affects of the heat exchanger on the overall power plant. In many cases, preliminary designs based on turbomachinery considerations alone have "fixed the cycle conditions, and specified the heat exchanger," without considering the overall packaging and cost penalties that may occur as a result of this design philosophy. For a given problem statement, the relative effect of cycle parameters on exchanger size is independent of recuperator type, and thus, detailed analyses can be carried out for various surface geometries to identify a solution that satisfies the requirements for the particular application.

Detailed optimization techniques are not outlined in this paper, since each company involved in heat-exchanger development has its own comprehensive computer programs to identify the optimum unit from the standpoints of minimum volume, weight, etc., for each particular application. Analytical techniques for recuperator optimization

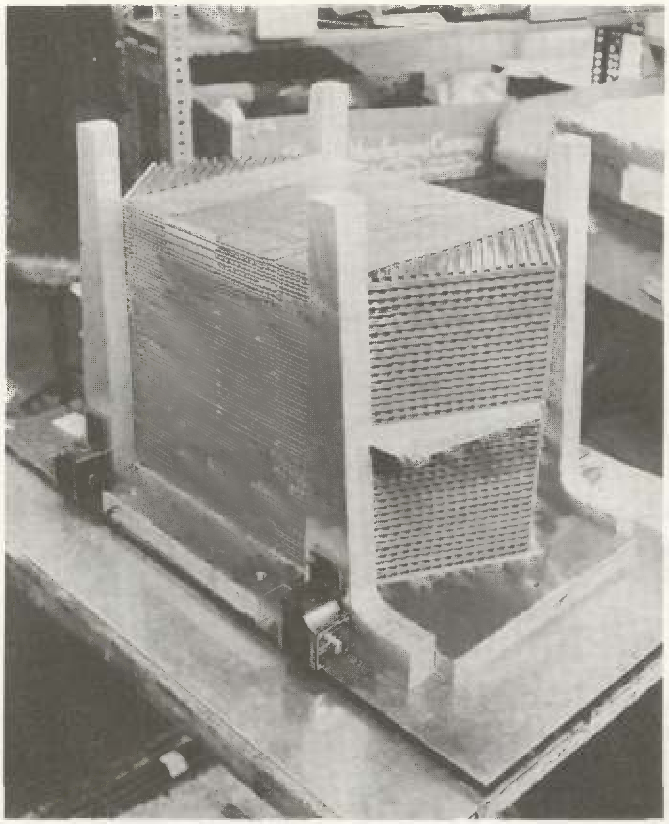


Fig. 15 Plate-fin recuperator core stack assembly

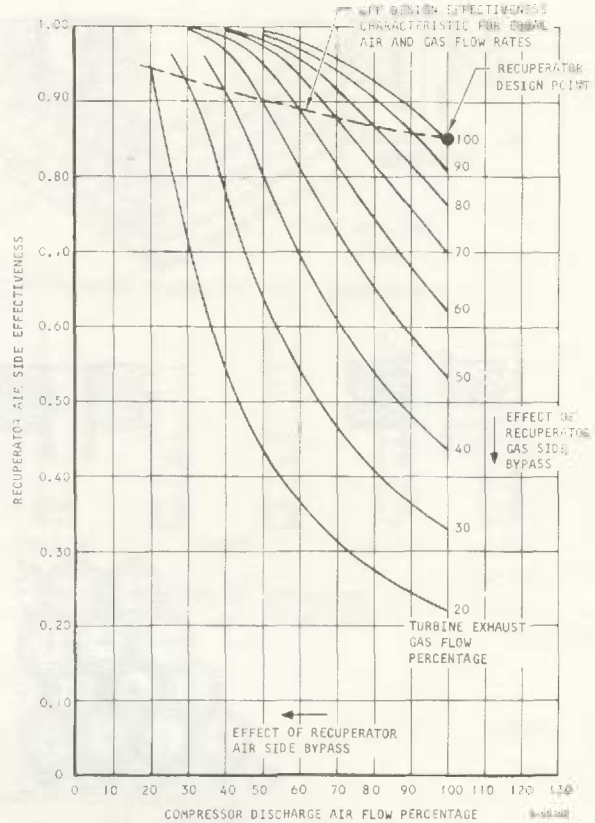


Fig. 16 Typical effectiveness performance map for compact plate-fin recuperator

have been reported by Aronson (15,16), Rohsenow (17), Shepherd (18), Hryniszak (19), and Ohlgren (20).

Detailed investigations of various exchanger surface geometries have been reported by Kays and London (21), Gel'fenbeyn (22), Larkin (23), Smith (24), Bergwerk (25), and Burrow (26). It is difficult to envision which combination of surface geometries would give a unit of minimum volume, or weight, by merely studying the heat-transfer (j) and friction (f) characteristics of various surfaces. A simple technique is developed in Appendix 1 which enables direct surface comparisons to be made at the same recuperator thermodynamic and pressure loss conditions. This analysis shows a number of relations which make the trends more obvious (e.g., effect of changing corrugation type). In the use of Fig. 9, the design data fixes a point on the horizontal scale, with high Reynolds number applications toward the right-hand end. The volume contributions will then be proportional to the reading on the vertical scale, provided the shape of the matrix is not restricted.

From Fig. 9, it can be seen that the offset corrugations give the greatest advantage in the medium Reynolds number range where, because of pressure loss limitations, a gas turbine recuperator would be designed to operate (Reynolds number less than 1000). In the majority of compact recuperators for industrial, marine, and vehicular gas turbine application, offset type of finned surfaces are utilized for minimum volume and weight. With this type of surface, the fins are systematically pierced in the direction of flow and offset normal to the direction of flow. This provides periodic interruption of the boundary layers and thereby increases the heat-transfer coefficients. Boundary-layer dissolution incurs a smaller pressure drop penalty (lower f/j) than artificial turbulence promotion, such as wavy, louvered, or perforated fin configurations. In comparing the characteristics of the various types of surfaces, it has been observed that the compact offset rectangular fins result in matrices of minimum volume and weight, and thus represent a very effective type of secondary surface.

For tubular surface geometries, it can be seen that enhancing the inside coefficient by means of tube wall dimpling will result in a considerable reduction in matrix volume while, at

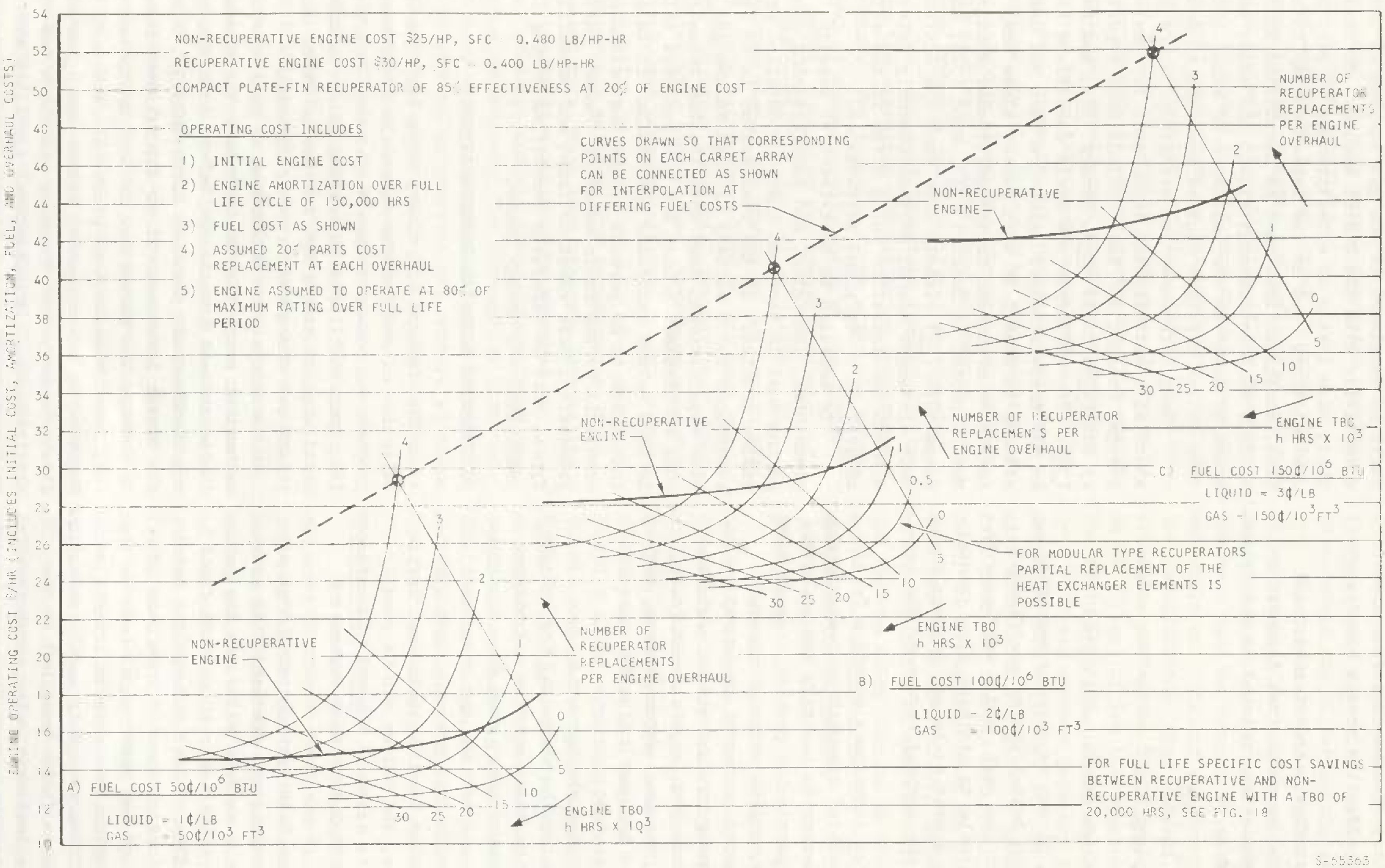


Fig. 17 Comparison of operating costs between a recuperative and non-recuperative 3500-hp industrial and marine-type gas turbine

the same time, giving a core of more attractive dimensions (reduced tube length) and better structural characteristics. The benefit from the dimpling feature is effective over a wide range of Reynolds numbers.

For a given airflow, effectiveness, and pressure loss, it can be shown from Appendix 1 that the core geometries are related to the following surface parameters.

Frontal Area $A_{FR} \alpha \sqrt{f/j}$

Core Volume $V \quad \alpha\ d \sqrt{f/j^3}$

Core Weight W (and material cost) $\alpha \sqrt{f/j^3}$

From the foregoing, it can be clearly seen that a surface with a high heat-transfer coefficient (j) is necessary for low cost, and a surface with a low (f/j) is desirable to minimize flow frontal area, and thus give a heat exchanger package that is compatible with the compact nature of the turbomachinery.

Modern computer techniques permit a "vary every parameter possible" approach to heat-exchanger design in an endeavor to identify the so-called optimum unit for a particular problem statement. For plate-fin units, a wide range of passage heights, fin geometries, material types and thicknesses, and flow configurations can be evaluated for each thermodynamic problem statement (i.e., flow, temperature, pressure, effectiveness, and pressure loss). For tubular units, the tube diameter, dimple or turbulator geometry, tube pitch, wall thickness, and flow configuration must be evaluated. For counterflow variants, the optimum geometry of the end sections must be established to give minimum pressure loss, good flow distribution into the core, and satisfy the structural requirements. The effect of pressure loss split between the two streams is a variable that influences the core shape, and values can be readily identified which give balanced thermal conductances and, hence, minimum heat exchanger volume and weight.

In counterflow heat exchangers of high effectiveness, the temperature gradient that exists in the matrix is at a maximum value, as the hot end of the core is essentially at maximum fluid temperature. The flow of heat through the metal in this type of situation results in a loss of heat from the hot end and addition of heat to the cold end, both of which have adverse effects on heat-exchanger performance. This axial conduction effect, which has been fully described in the literature, must be included in the recuperator design, and, for units made from the very compact surfaces, with large frontal areas and small flow lengths, the conduction effect can be significant. It has been shown earlier that a flat SFC-power curve is necessary if the gas turbine is to be competitive with the diesel engine. To realize this, high levels of effectiveness at part load are necessary, and in the heat-transfer optimization, configurations must be identified which can satisfy this requirement (i.e., the basic heat-transfer data and the influence of axial condition at low airflows must be examined in detail).

Many references are available in the literature with optimization techniques for heat-exchanger design, and they include work reported by Fax (27), Snell (28), Hendry (29), and Fairall (30). The influence the cycle conditions have on the heat-exchanger selection have not been considered in detail in this paper, since they are very much dependent on the particular application. Work reported by Demetri (31), Wood (32), and Gasparovic (33) clearly show that the optimum heat exchanger cannot be independently selected without consideration being given to the effect on overall power plant performance.

In addition to heat-transfer aspects, structural, packaging, and manufacturing considerations must be factored into the detailed optimization studies. If the recuperator is to find acceptance for a variety of high volume gas turbine applications, then optimization techniques must be directed toward the minimum cost solution. In high volume vehicular production, it is assumed that close to 80 percent of the heat-exchanger cost will be the basic material; hence, surface geometries, core constructions, and material types, must be carefully identified during the analytical and design phase.

RECUPERATOR MATERIAL CONSIDERATIONS

The factors of primary importance in the selection of recuperator materials are mechanical properties, hot-corrosion resistance, fabricability, compatibility with brazing filler metals, metallurgical stability, and cost. These considerations are discussed in the following.

A recuperator material must have adequate mechanical properties to withstand the stresses due to thermal transients, and to both fluctuating and steady-state pressure differentials, during its design lifetime. It is insufficient merely to measure the properties of new, unexposed material, because mechanical properties are most always degraded by environmental attack and by metallurgical changes, such as aging reactions and carbide precipitation. Metallurgical stability and corrosion resistance cannot be considered apart and are important in evaluating mechanical properties.

The recuperator alloy must be capable of resisting simultaneously the effects of high temperatures and high stresses, particularly in the presence of contaminants. In such cases, corrosion can take place in certain materials.

Hot-corrosion is the attack on metal alloy components caused directly, or indirectly, by contact with products of combustion of gas turbine engines. Included in this term are all synergistic effects that contribute to hot-corrosion, such as sulfidation, oxidation, erosion, stress corrosion, and both static and cyclic stresses. In spite of their sensitivity to stress corrosion, the austenitic stainless steels used in many recuperators have proved to be excellent in operation at 1300 F. The creep strength and corrosion resistance of steel can be greatly improved by the addition of certain alloying elements. The addition of such elements as nickel, cobalt, and molybdenum improve the strength characteristics at elevated temperature. The most important element for increasing the corrosion and oxidation resistance is chromium. The chromium forms a tightly adherent layer of chromium-rich oxide on the surface of the metal, retarding the inward diffusion of oxygen and inhibiting further oxidation. In plain carbon steel, the amount of scaling in air is negligible below about 1000 F, and many recuperators for large industrial gas turbines, operating below this temperature, have been made from mild steel, and have demonstrated a high degree of reliability.

Materials selected for formed parts, such as fins and pans, must have adequate formability at room temperature and must be amenable to brazing, welding, and all other contemplated manufacturing operations. The strength at time and temperature, or creep properties, must be high enough to prevent deformation and rupture of these thin foil formed parts, made to the gages selected for the particular design application.

Since brazing is the most economical means of producing recuperator structures, the compatibility of structural materials with candidate brazing filler metals is of primary importance. The selected base metal/filler metal combinations must satisfy general requirements which include the following: A compatible brazing temperature must exist to avoid metallurgical changes in the parent material during the braze cycle. There should be minimum erosion and penetration into the base metal, fins, plates, tubes, or headers. The joint should have adequate strength and corrosion resistance and little or no embrittlement of the parent material due to braze alloy diffusion.

The mechanical properties and corrosion resistance of the chosen materials must not be detrimentally affected by internal metallurgical changes, such as carbide precipitation, sigma formation, internal oxidation, or diffusion reactions between dissimilar materials, e.g., between base metals and filler metals or coatings.

Cost is a factor of prime importance in all recuperators. For the recuperator to find a wide range of acceptance in gas turbine applications, the cost must be reduced by utilizing lower cost materials and types of construction to minimize the manufacturing labor content. For high volume vehicular production it is projected that close to 80 percent of the heat-exchanger cost will be the basic material. Thus, it is mandatory to select the best surface geometry to give the lowest specific weight (lb/hp) and the best material to give acceptable recuperator specific costs ($/hp). Cost data for several materials, in the thin foil form, that have been used for plate-fin recuperators are shown in Fig. 10. Descriptions are given in the following of representative materials from the various alloy classes in order of ascending costs, increasing oxidation resistance, and elevated-temperature strength.

Mild Steel

Many gas turbine recuperators, of both tubular and plate-fin geometries, have been manufactured using mild steel with brazed and welded types of construction. Most of these units have been used in conservatively designed, large industrial plants, where the gas temperatures have been below 1000 F and oxidation has not been a problem. These units of the types described by Caughill (34), Wolfe (35), Howitt (36), and Holm (37) have accumulated high degrees of reliability. Mild steel has outstanding attributes of low cost, good fabricability, good weld and brazeability, but is temperature limited to 1000 F because of the poor oxidation resistance.

Treated Mild Steel

As mentioned previously, the oxidation resistance of carbon steel can be substantially improved by the additon of chromium. This can be done by adding the chromium during the steelmaking process or by surface enrichment treatment of the mild steel by the aluminizing or chromizing process. Aluminized sheet is not considered a likely candidate because it is not easy to braze, but chromized mild steel demands serious consideration, since it has excellent brazing characteristics and is much cheaper than the various 400 and 300 series stainless steels. Should the chromized layer be incomplete, the base material would have no oxidation resistance, but this is not considered a serious problem, since these imperfections would

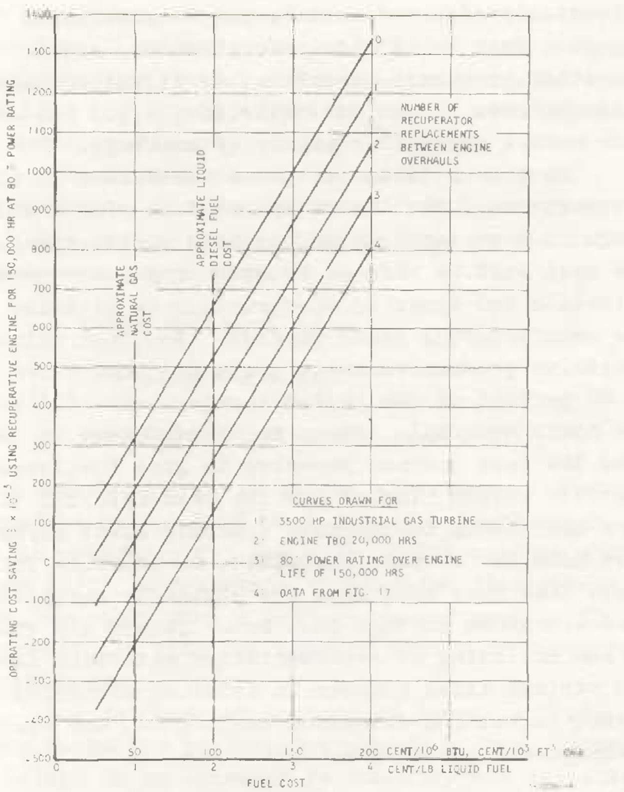


Fig. 18 Influence of fuel price on the operating cost differential between a recuperative and nonrecuperative 3500-hp industrial gas turbine

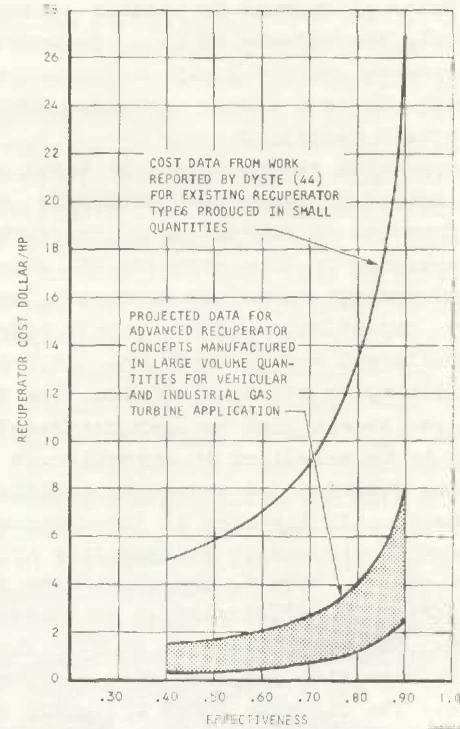


Fig. 19 Plate-fin recuperator cost data

be covered by a layer of filler metal during the braze process.

Ferritic Stainless Steels (400 Series)

Typical of the ferritic stainless steels are Types 430, 434, 436, 405, and these steels have approximately one-third of the rupture and creep strength of austenitic stainless steels. These alloys have excellent oxidation resistance, a low thermal expansion, are readily formable, and can be easily brazed. A strong advantage of the 400 series alloys (in addition to the low cost) is the fact that they do not contain nickel. Exclusion of such a critical element is very important for the projected high volume vehicular gas turbine market.

Type 347 Stainless Steel

Type 347 stainless steel is an excellent recuperator material and has been used widely in these applications. Carbide precipitation in grain boundaries, and consequent sensitization to chemical corrosion at low temperature, is minimized by the addition of columbium, which forms stable carbides. Brazeability of Type 347 is excellent, and formability is good. Type 347 has good creep strength and hot-corrosion resistance for long-time service at elevated temperatures.

Incoloy 800

Incoloy 800 is a low-cost, iron-base superalloy containing a nominal 32 percent nickel and 21 percent chromium. The high percentage of nickel keeps the alloy austenitic and ductile after long exposure to elevated temperature, contributes to oxidation and general corrosion resistance, and makes the alloy nearly immune to stress corrosion. Because of its chromium content, the alloy displays outstanding oxidation resistance and good resistance to hot corrosion. The iron content imparts resistance to sulfidation.

Inconel 625

Inconel 625 is a relatively new alloy, which is similar in composition to Hastelloy X and has comparable strength and oxidation characteristics. The distinguished feature of Inconel 625 is its improved ductility at elevated temperatures. Its resistance to low-cycle fatigue, therefore, is exceptionally good. Inconel has been used for a number of aircraft heat exchangers and lightweight gas turbine recuperators, but its high cost makes it prohibitive for industrial, vehicular, and commercial application.

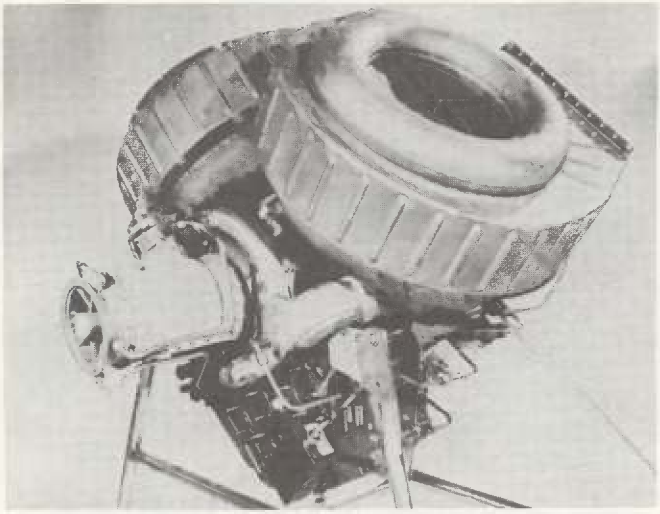


Fig. 20 Recuperative gas turbine for helicopter propulsion

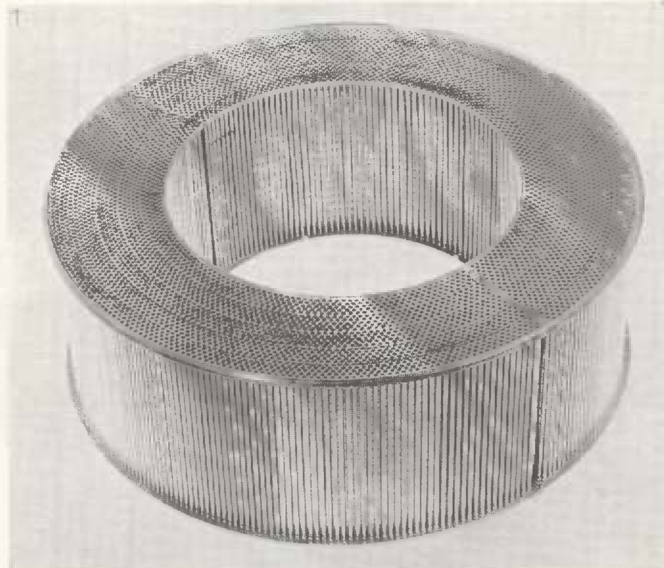


Fig. 21 Compact, lightweight tubular recuperator matrix

Hastelloy X

Hastelloy X is a favorite sheet metal alloy for use in combustion liners and hot ducting in gas turbine engines. It is a high-cost alloy and is produced in large quantities. The high cost has virtually eliminated it as an attractive material for gas turbine recuperators.

Cobalt Base Alloys

N-155 is a mixed-base alloy containing nominally 20 percent nickel, 20 percent cobalt, 20 percent chromium, and the balance iron. While this alloy, together with Haynes 25 and 188, are widely used in gas turbines, their high cost makes them unattractive for commercial recuperator application.

Ceramics

Ceramic materials offer unlimited temperature potential and low cost, but, to date, they have been utilized in the rotary regenerator only. With poor thermal conductivity and the high pressure loading and manifolding problems in fixed boundary units, considerable development work will be required before a viable ceramic recuperator can be utilized for vehicular and industrial gas turbines. Because these materials lack ductility, new design approaches must be taken to match the unique characteristics of the particular ceramic considered. Utilization of ceramic materials for various stationary gas turbine parts (nozzles, combustors, scrolls, and ducting), as well as the rotating components, have been outlined by McClean (38), and it is expected that similar development efforts will be applied to the fixed boundary heat exchanger.

RECUPERATOR STRUCTURAL CONSIDERATIONS

The structural design of the tubular or plate-fin core matrix must consider combined stresses due to internal pressure loadings, and thermal stresses incurred by steady-state and transient metal temperature differentials, between the elements in the core. In order to retain structural integrity during the design life of the unit, the level of steady-state, long-term operating stresses should be kept below the stress for 1 percent creep in 10,000 hr (or the specified design life) and, where possible, the peak short-term stresses kept below the yield strength of the material. In each case, the material strength at the appropriate metal temperature should be considered. The stress capability of several representative recuperator materials is shown in Fig. 11. Where local transient stresses exceed the material yield strength, a plastic analysis must be used to demonstrate that the number of cycles to failure is well above the expected number of load cycles. Also, where vibratory stresses are involved, the relevant steady and alternating stress levels must be evaluated to demonstrate fatigue resistance. The recuperator must be designed for sustained pressure operation at maximum operating

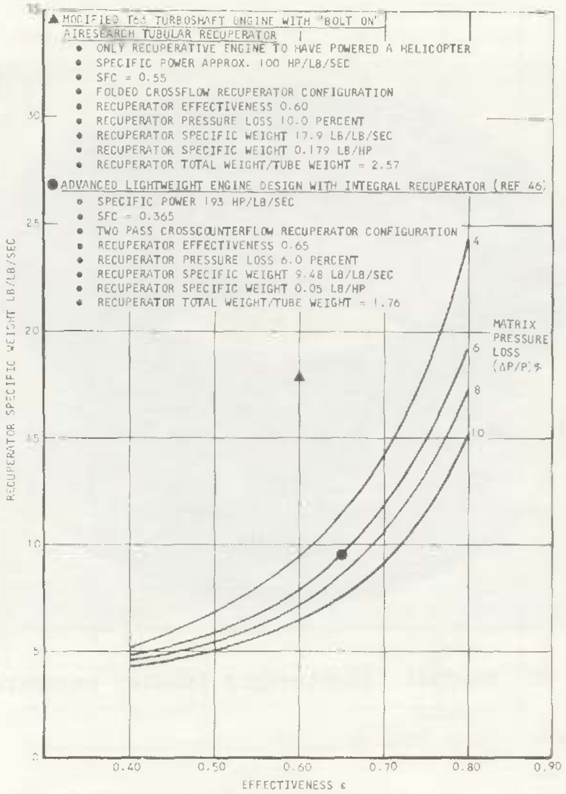


Fig. 22 Specific weights of lightweight, compact tubular recuperators for aircraft gas turbine application

temperature throughout the entire design life. Established analytical techniques are used to ensure that acceptable pressure stress levels exist in the core, headers, and manifolds.

Thermal stresses in recuperator modules are of: (a) the steady-state, long-term type which exist during most of the operational life, and (b) the transient type which occur during engine start-ups, shutdowns, and load changes. Fig. 12 is included to illustrate that the steady-state thermal gradients are a minimum for counterflow recuperators by showing a comparison with the other various configurations discussed in an earlier section. Transient metal temperature differentials are more elusive and are the most critical in determining the gages and materials that must be used to give the required life.

Transient metal temperature differentials occur when the recuperator is subjected to rapid temperature fluctuations. These differentials occur during thermal transients and are caused by the temperature-time lag of the relatively heavier sections in the core. Typical components in the plate-fin matrix that have inherent metal temperature lag are the heavier reinforcing bars and heavy gage manifolds. Unless extreme care is

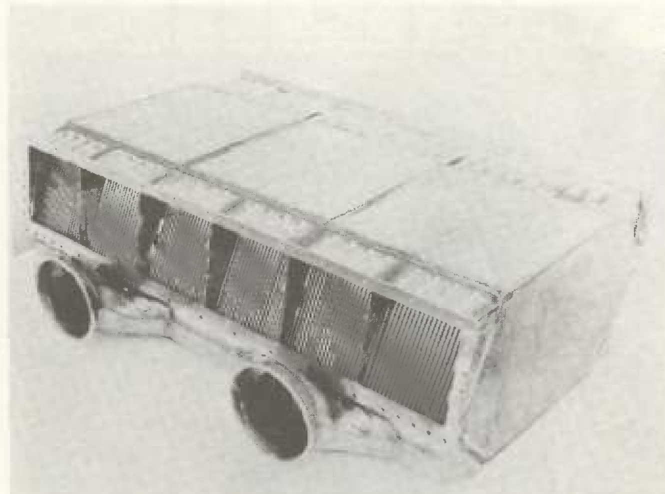


Fig. 23 Vehicular gas turbine plate-fin recuperator

taken, a critical area in the core is the tube sheet adjacent to the sealing bar near the matrix extremity where the transient thermal stress can often exceed the material yield strength. The typical effect of transient thermal stresses in a plate-fin recuperator matrix is shown in Fig. 13. For long life, it is obviously mandatory to achieve thermal inertia compatibility between the various elements of the core and parts attached to the core. Methods of reducing the time constant of the sealing bars include utilization of hollow and channel sections, and, more ideally for complete compatibility, the seals can be integrally formed from the plate itself.

RECUPERATOR FABRICATION PROCESSES

In previous sections, design approaches for surface geometry and flow path optimization have been outlined; however, some departures from the ideal heat-transfer solution are often necessary to give practical mechanical designs that can be readily constructed. In the simplest form, tubular designs are assembled into drilled header plates or can be designed to interlock or integrally form the header at the tube ends. The joining process can be done either using welding or brazing techniques, and typical end seals for tubular heat exchangers are shown in Fig. 14. In the lightweight tubular recuperators developed for aircraft gas turbines, small tube diameters and compact geometries (close tube pitching) are utilized, and this virtually necessitates brazing the assembly together. For this type of application, extremely lightweight designs can be achieved

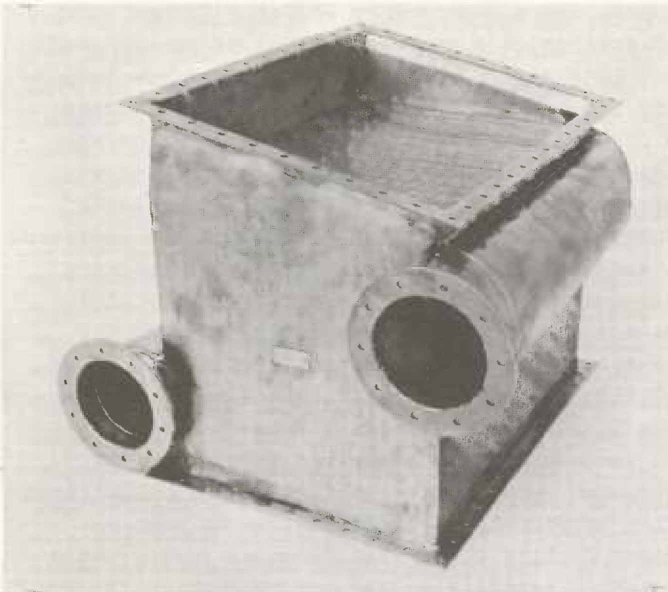


Fig. 24 Counterflow plate-fin unit for vehicular gas turbine

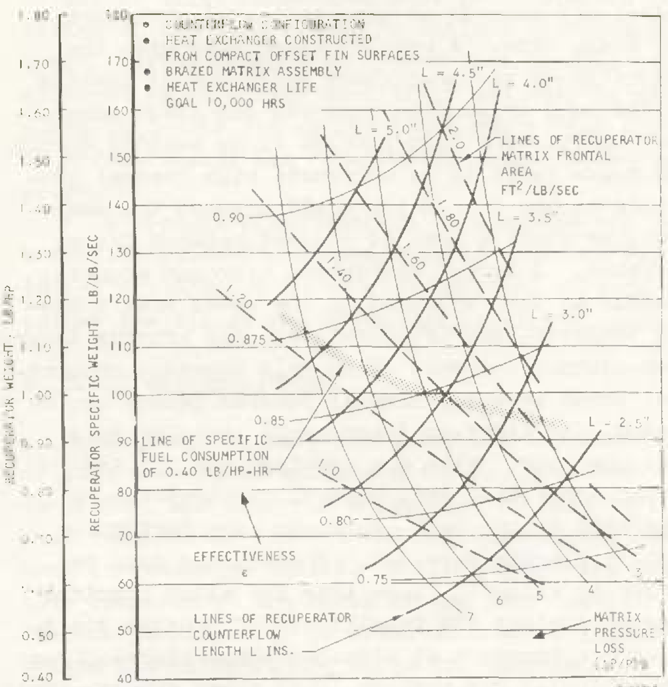


Fig. 25 Specific sizes and weights of compact plate-fin recuperators for vehicular gas turbine application

by utilizing very compact geometries, in which the tube ends are extruded into the form of squares of hexagons and brazed together to form the header. For the large, closed-cycle, nuclear gas turbines, the surface compactness of the tubular units is limited by the requirement for welded tube-to-header joints. The resultant large tube pitch arrangement gives units of extremely large volume rather like steam boilers, where a similar all-welded construction has been called for traditionally.

For the efficient secondary surface designs, the highest degrees of specialization are associated with the operations required to produce the heat-transfer surfaces themselves. There the object is to make the desired part from thin gage raw foil at a high rate of production. As mentioned in the surface geometry section, high-performance fins of the rectangular offset type demand forms and shapes more critical in terms of die wear than the other types of surfaces. These die-forming machines are special presses that have combined braking, feeding, cutting, and forming capabilities.

The assembly of plate-fin recuperators, is done with the aid of special tools and fixtures. In these fixtures, the parts are assembled to form the recuperator matrix with its alternating high- and low-pressure passages. For low volume production, the cores are manually stacked in the type of assembly fixture shown in Fig. 15. For high volume production on, as envisioned for the vehicular gas turbine market, the component forming and assembly operations would be done by automatic machine tools.

The final joining of the matrix elements can be done using brazing or welding techniques. The majority of recuperator cores are designed for brazing, and welding of the basic matrix is only used where there are only a few parts and are readily accessible. Braze alloys for recuperator applications are typically nickel-base and available in powder form. This material in the powder or slurry form may be sprayed, brushed, or rolled on the surfaces to be joined, and this could be either on the tube plate itself or on the corrugation nodes. Again, in high volume production, this process would be automated. There is an obvious cost incentive to use the minimum amount of braze alloy to give the required joint structure properties, and, as London (39) points out, the roughness associated with the braze alloy layer can adversely affect the heat-transfer and friction characteristics for very compact surface geometries. The resultant slight increase in friction factor will be tolerated in practical recuperators, since the rich layer of braze alloy on the fins and plates gives good oxidation resistance, and will help to give long life to units made from the cheaper, low-cost steels.

Brazing can be carried out either in a vauum furnace or in a hydrogen atmosphere to remove oxides and provide an uncontaminated environment for braze flow. A problem associated with the conventional vacuum-furnace technique of brazing is the long cycle time required for large recuperator cores. The main problem to be avoided during the braze cycle is to eliminate high thermal gradients in the core which could distort the components or cause a loss of contact between faying surfaces. A forced convection hydrogen process, similar to that utilized in the steel heat-treating industry, has been developed for brazing large heat-exchanger cores. With this process, recuperator cores weighing several hundred pounds can be brazed with floor-to-floor cycle times of less than one hour. With the combined radiant and forced convection heat inputs, the temperature gradients in the heat-exchanger core during the braze cycle are very low. The reduced time required to bring the core into the alloy liquidus range minimizes the possibility of erosion due to excessive exposure at elevated temperatures. For the vehicular gas turbine, this braze process could be developed in a semi-automated operation that would be capable of a high rate of production. The importance of good fixturing and a controlled braze operation to minimize distortion and hold the heat transfer elements within close tolerance should be emphasized, since London (40) has shown that in laminar flow, the local passage deformation and tolerance stack-up can result in adverse heat transfer and friction effects.

For future vehicular and industrial gas turbines, designs that offer low material costs, reduced number of parts, reduced labor effort, and adaptability to high volume production methods are required to demonstrate the viability of the fixed boundary recuperator.

ENVIRONMENTAL ASPECTS

Over the years, many questions have been asked about the applicability of the fixed boundary recuperator in open-cycle engines, where uncertainties regarding the effects of fouling, fires, dust congestion, and marine salt spray, have existed. Pertinent comments in each of these areas are outlined in the following.

With the early large, heavy oil (Bunker C) burning gas turbine plants, deposits of a carbonous soot material in the heat-exchanger gas side passages were experienced, and periodic soot removal by means of steam jets was necessary. With today's vehicular, industrial, and marine gas turbines burning lighter fuels (Diesel No. 2), careful attention to the combustor design has more or less eliminated the recuperator fouling problem. Work reported by Miller (41) has shown that fouling is a strong function of recuperator metal temperature. An increase of surface temperature above about 700 F very rapidly eliminates any previously accumulated deposits, typically within seconds of running time. In such cases, heat transfer and pressure loss performance was returned to initial values typical of new surfaces. The simplest and most effective way of removing fouling deposits is to incorporate an air side heat-exchanger bypass duct. This provides the capability of periodic heating of the core to essentially full gas temperature and thus removing accumulated carbon deposit. This feature has been successfully incorporated in a number of AiResearch recuperative engines and has eliminated engine performance degradation due to heat exchanger fouling for a wide range of fuels. With increasing emphasis on cleaner combustion for low pollutant emission, the problem of recuperator fouling will be completely eliminated.

As outlined by McDonald (42), situations have arisen where fires have occurred in gas turbine recuperators. Most of the fires reported were due to either inadequacies in the engine design, which allowed excess fuel to become trapped in the ducts and heat-exchanger matrix, or to control malfunction. To eliminate the possibility of fires in gas turbine recuperators under normal modes of operation, the following should be observed. A good drainage system should be incorporated which prevents fuel from becoming trapped in the ducts and matrix for all engine altitudes. A relay in the starter system should be provided to prevent repeated start attempts, after an aborted start, until some of the excess fuel has drained away or vaporized. As a further precaution after an aborted start, the system should be purged thoroughly by motoring the engine over on the starter for a period of 20 to 30 sec. By utilizing good design practice, AiResearch has gained many thousands of hours of recuperative engine experience without the advent of a fire in a compact stainless-steel plate-fin recuperator.

To obtain information on the effect of erosive contaminants, a sand and dust test was performed on an AiResearch plate-fin recuperator. The sand and dust were injected into the gas turbine engine at an average concentration of 0.0008 gm/cu ft of engine inlet airflow. After 150 hr of operation, no plugged passages were observed on either side of the heat exchanger. No appreciable amount of dust was deposited on the actual heat-transfer surfaces, and no signs of erosion or appreciable change in the performance characteristics were observed.

To check the effect of a marine environment, the AiResearch plate-fin recuperator was removed for inspection from an Orenda OT-4 600-hp engine that had been used aboard the Minesweeper USS Kingbird. The marine service life had been two calendar years, and the engine was reported to have had 1753 hr of operation. This record indicates that the engine components were subjected to both static and dynamic conditions in a marine environment. Although severe corrosion was observed on some of the external engine components, the recuperator core showed no sign of corrosion, erosion, or fouling. Leakage and pressure drop tests showed no change from the new recuperator values. Samples of the stainless-steel core were selected from the highly stressed areas of the unit and mounted for micro-examination. No corrosion was found in any of the brazed fin-to-plate joint sections. The metallurgical condition of the core was excellent, and scaling damage was very slight (in the range of 0.0000 to 0.0001 in.). The weld joints throughout the core were in good condition and showed no signs of corrosion damage.

RECUPERATOR PERFORMANCE CONSIDERATIONS

For the accurate assessment of recuperator performance, the heat-transfer characteristics of the selected surface geometries must be known over a range of Reynolds numbers. Good flow distribution is important where high effectiveness and low-pressure losses are required. Where high-flow rate designs are considered, several recuperator modules may be required. As the number of modules increases, the complexity of the headering and manifolding also increases, and greater attention must be given to the problem of securing uniform flow distribution. There are several techniques for achieving proper flow distribution, including the use of contoured headers, splitter vanes, and variable resistance through the matrix. Variations of the flow pattern at engine off-design conditions must also be considered in accurate recuperator performance calculations.

In gas turbine recuperators, where high effectiveness necessitates a counterflow arrangement, the axial temperature gradient that exists in the core is at a maximum. The axial flow of heat through the metal in this situation results in a loss of heat from the hot end and addition of heat to the cold end, both of which have adverse effects on recuperator performance. This heat flow in the walls by conduction in the axial direction is undesirable, since it tends to unify the wall temperature. This adverse effect on thermal performance is dependent on material conductivity, conduction path length, cross-sectional area, and flow configuration. In the thermal analysis, the effect of axial conduction must be considered in sizing of the recuperator to satisfy performance goals. Axial conduction effects are usually significant at part flow conditions and must be examined in detail for applications where high levels of effectiveness are required at engine part-load power.

A typical effectiveness performance map for a compact plate-fin recuperator is shown in Fig. 16. The influence on effectiveness of varying either the air or gas flows can be clearly seen.

ECONOMIC CONSIDERATIONS

In the preceding sections, the importance of such aspects as surface geometry, core construction, and material selection, as they influence recuperator size, cost, and durability, have been discussed in detail. While engines are usually compared on the basis of the same specific fuel consumption, it is often more meaningful to relate actual operating costs and examine, for instance, the influence of recuperator effectiveness on plant initial cost and operating cost over the life of the machine. The validity of comprehensive economic analyses are very dependent on the particular application, and load profile, fuel cost, life requirement, and initial plant cost. Gas turbine initial plant cost data is usually of a proprietary nature and, again, is very dependent on the application and the production quantities involved. In work reported by Gasparovic (43), optimal surfaces were identified to give power plant and fuel costs of minimum value. In the work reported by Dyste (44), optimum recuperator effectiveness values for maximum savings were shown to be very sensitive to fuel costs and life requirements.

A comprehensive survey of operating costs is obviously beyond the scope of this paper, because it involves a multitude of combinations of fuel costs, varying load profile, power levels, overhaul periods, and time between recuperator replacements for a wide range of applications. For the purpose of this paper, it was, therefore, necessary to make some assumptions in an attempt to identify a method of comparing operating costs of simple-cycle and recuperative engine variants. It is necessary to emphasize that the data presented has the basic objective of showing trends that are to be expected, and that a specific application and its anticipated duty cycle must be studied in detail to give actual economic data.

A comparison of operating costs between a recuperative and nonrecuperative industrial type of gas turbine, in the 3500-hp class, is shown in Fig. 17. The assumed engine has a compact plate-fin recuperator of 85 percent effectiveness, with an initial cost of 20 percent of the overall engine cost. The estimated engine operating costs are based on an 80 percent, power rating over the life of the machine, together with the other assumptions given in the figure. It can be seen that the operating economics of the gas turbine are very sensitive to fuel cost, and if fuel costs increase substantially [as indicated in reference (45)], then the recuperative cycle will become more attractive to the prime-mover owner. Since the reliability and life of heat exchangers has not reached the level of the rotating machinery, the influence of recuperator expendability on power plant operating cost is shown in Fig. 17. Again, the effect is sensitive to fuel cost, but it can be seen (based on the defined assumptions) that even at the low-fuel costs, the recuperator does not necessarily have to be designed for the full engine life to show an overall economic advantage over the simple-cycle engine. For this industrial type of engine, a modular recuperator approach could be provisioned as necessary. For the same engine, the influence of fuel price on the operating cost differential between the recuperative and nonrecuperative variants is shown in Fig. 18. Again, the economic effect of recuperator replacements between engine overhauls can be seen to be a strong function of fuel cost. The effect of downtime inconvenience for recuperator replacement has not been included in the foregoing data.

In the two curves described in the foregoing, there has been no attempt to directly duplicate a particular manufacturers specific engine problem statement, but instead a simple technique has been demonstrated to relate operating costs, rather than merely specific fuel consumptions. In the preliminary design phase of a new engine, a family of simple curve arrays of the type shown enable a meaningful assessment to be made of the relative recuperative-nonrecuperative engine economics.

For some applications, volume and weight considerations must be considered in conjunction with economic goals. For marine applications, for instance, the volume of the engine plus fuel is important, and while the heat exchanged engine itself occupies a larger space, it can be readily shown that the additional volume of the recuperator is recoupled in a few hours of operation due to the reduced fuel consumption. For military gas turbine generator sets, required to operate in isolated battle areas, engine plus fuel weight is important, since the units (with their fuel supply) must be flown in. The additional weight of the recuperative variant is made up for in fuel savings after only a few hours running, and this alleviates the fuel logistics problem.

In this paper, no attempt has been made to identify the recuperator effectiveness to give maximum cost savings, since this is very dependent on the type of engine and the load profile for a particular application. Fig. 19 is included to show the influence of effectiveness on specific cost ($/hp). Again, cost factors are related to engine type and volume production, and, hence, bands of data are shown. The lowest cost data are projected values for recuperators to be used on automotive type gas turbine power plants, where very high volume production quantities are postulated. Clearly, recuperator sizes, weights, and costs increase significantly at effectiveness levels above about 80 percent. Very high levels of effectiveness (in excess of 90 percent) are utilized in closed nuclear cycle gas turbines, for example, where, in addition to achieving high levels of cycle efficiency, the size of the precooler at the compressor inlet can be substantially reduced.

GAS TURBINE RECUPERATOR APPLICATIONS

The various gas turbine applications, where fixed boundary recuperators of various types have been utilized, are briefly outlined in the following.

Aircraft Turboshaft Gas Turbines

Recuperators have not found acceptance for helicopter or fixed-wing aircraft application, in spite of considerable engineering and experimental programs to develop lightweight heat exchangers. The only recuperative engine to have powered a helicopter was a modified T63 engine, with an AiResearch tubular recuperator, and this engine is shown in Fig. 20. A more detailed view of the lightweight recuperator, constructed from dimpled thin-walled tubes, is shown in Fig. 21. This engine, with essentially a bolt-on recuperator, performed satisfactorily, demonstrated the structural integrity of the heat exchanger, and increased the specific range of the helicopter by 25 percent. In work reported by McDonald (46), it has been shown that lightweight engines can be designed when the recuperator is considered as an integral part of the engine during the design phase. Specific recuperator weights for aircraft gas turbines are shown in Fig. 22. For flight times in the order of 2 hr, maximum net weight savings can be realized with effectiveness levels between 60 and 70 percent. One aspect, which may be influential in selection of recuperative engines for military helicopters in the near future, is the reduced IR signature compared with the

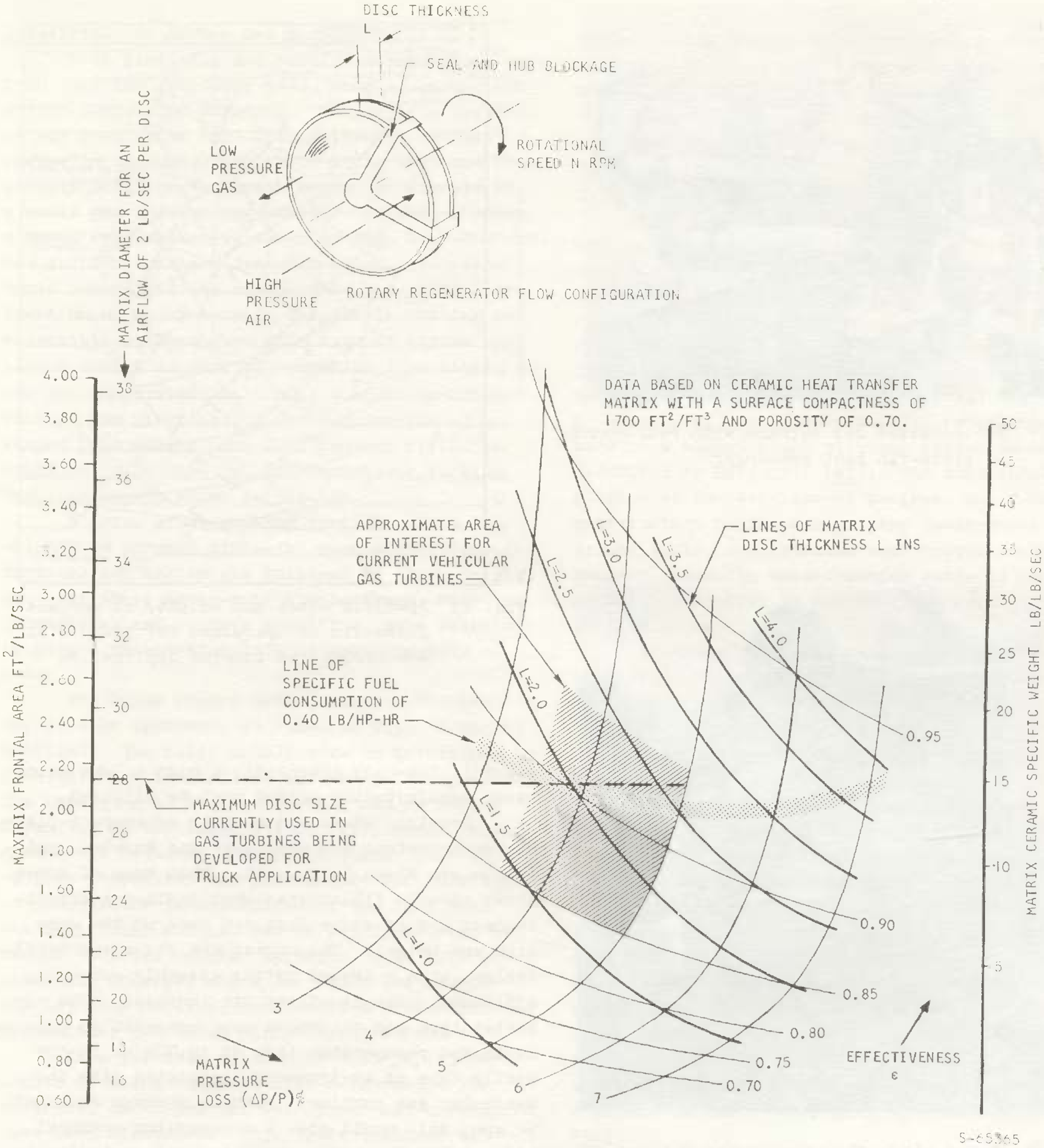


Fig. 26 Specific sizes of rotary ceramic regenerator for typical low-pressure ratio vehicular gas turbine engine cycle

nonrecuperative variants.

Vehicular Gas Turbines

Perhaps the biggest challenge for the heat-exchanger manufacturer is the vehicular gas turbine field which includes automotive, truck, and off-highway equipment. Utilization of low-cost materials, efficient heat-transfer surfaces, and high volume manufacturing techniques, as described in other sections of this paper, are necessary to keep the initial cost of the recuperator to an acceptable value. Counterflow, plate-fin units

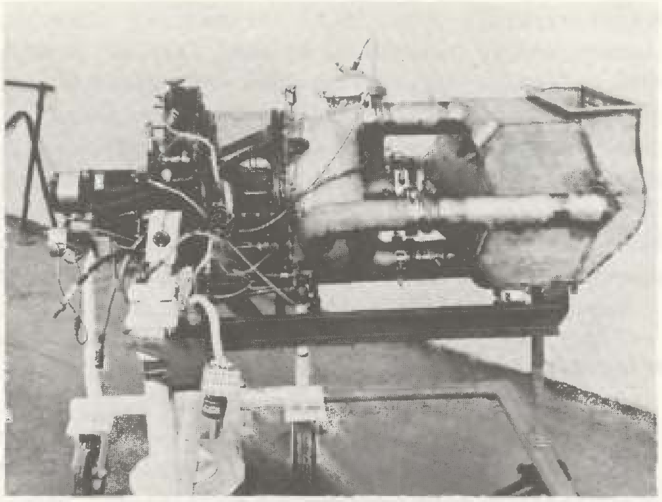


Fig. 27 Recuperative gas turbine with rear-mounted plate-fin heat exchanger

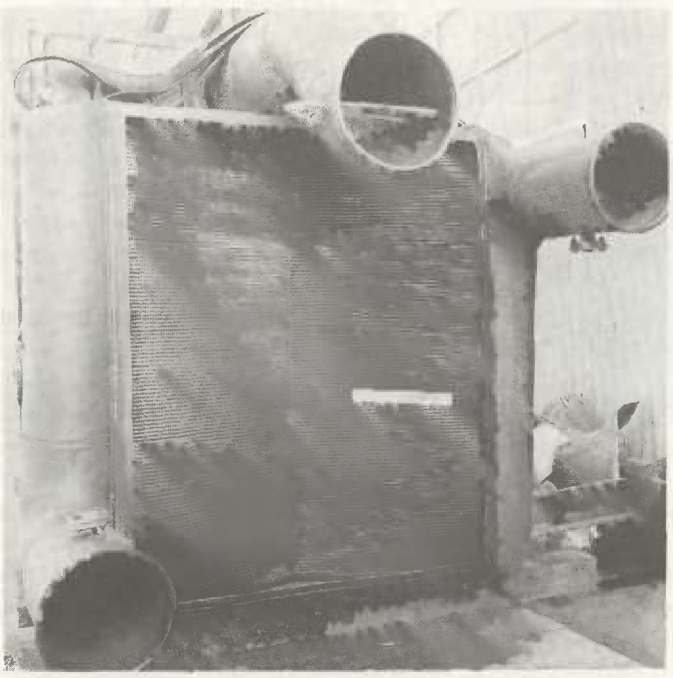


Fig. 28 Large plate-fin recuperator for industrial gas turbine application

of the types shown in Figs. 23 and 24 have been developed for vehicular gas turbines and have satisfied the required performance goals and structural integrity. Because of the small quantity involved, traditional manufacturing methods involving a high labor content have been used, and this has resulted in high recuperator costs. To realize

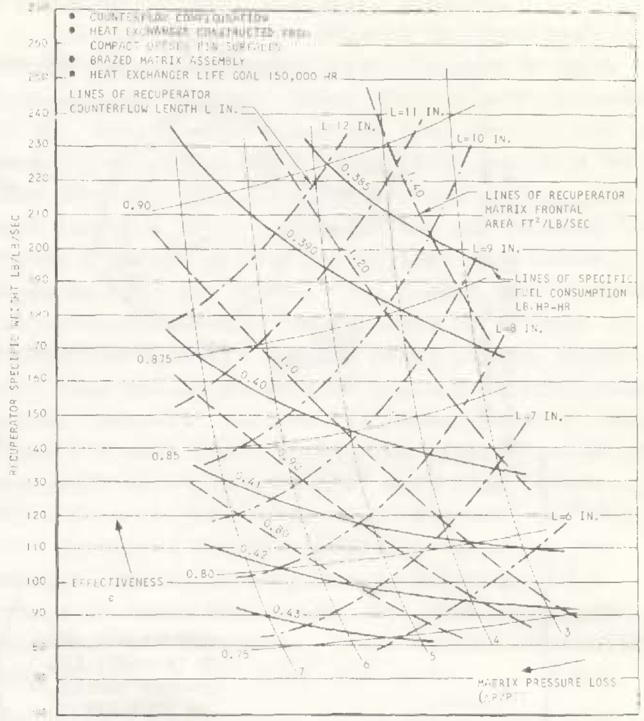


Fig. 29 Specific sizes and weights of compact plate-fin recuperators for industrial and marine gas turbine application

the full low-cost potential, a high volume automated manufacturing method must be utilized.

Specific sizes and weights of compact plate-fin recuperators for vehicular gas turbine application are shown in Fig. 25. This type of curve array clearly illustrates what influence effectiveness and pressure loss can have on the core size and weight. The curves are for a counterflow design, with a brazed matrix assembly embodying efficient, compact offset fin surfaces. The material type and thickness were selected to give an estimated recuperator life of 10,000 hr in the cyclic type of environment associated with the vehicular gas turbine. With an average speed of 50 mph, this would give a recuperator overhaul life of 500,000 miles for a typical vehicular application.

Although it has not been the purpose of this paper to make direct comparisons between rotary regenerators and fixed boundary recuperators, Fig. 26 is included to show specific regenerator sizes for typical low-pressure ratio vehicular gas turbine cycles. For these cycles, with specific powers in the order of 100 hp/lb/sec, the current maximum disk diameter of 28 in. limits the power to 400 to 500 hp for the twin disk variants with effectiveness values in the order of 90 percent.

Industrial and Marine Gas Turbine

Both plate-fin and tubular recuperators have been used for generator sets, pumping units, and marine propulsion turbines. In previous sections it has been shown that utilization of compact plate-fin surfaces results in units that are compatible with the turbomachinery. An example of a small gas turbine regenerator set with a rear-mounted recuperator is shown in Fig. 27. The unit has quick disconnect features which facilitate rapid removal of the recuperator for routine inspection or maintenance. The simple ducting and accessible combustor on this type of engine enables a change in mode of operation from simple cycle to recuperated, with only a minor modification to the plumbing. A further example of a larger recuperator made from compact plate-fin surfaces, also used for industrial gas turbine application, is shown in Fig. 28.

A curve array showing specific sizes and weights of compact plate-fin recuperators for industrial and marine gas turbines is shown in Fig. 29. For these heavy-duty applications, material thicknesses and surface geometries were selected to give a recuperator life of approximately 20 years.

For large engine application, a modular recuperator approach, as shown in Fig. 30 can be utilized. The basic module size is determined by manufacturing limitations and economic aspects. The packaging of the basic cores can be varied depending on the application, performance requirements, and required engine envelope. Ease of routine inspection and maintenance for this type of recuperator is emphasized by the fact that the individual modules can be easily removed as shown. For the modular approach, this type of packaging concept, where the basic box forms the gas side ducting, represents a cost-effective recuperator for large engine application.

In a previous section of this paper, the merits of using compact, high-performance fin surfaces, developed for aerospace heat exchangers, have been discussed. Fig. 31 is included to show a comparison of specific core volume and weight between existing low compactness plain surfaces, and the high-performance surfaces mentioned in the foregoing. It can be seen that the volume and weight of the large plain surface type of recuperators currently in service on large industrial and marine gas turbines can be reduced by an order of magnitude by using high-performance, heat-transfer surfaces.

Closed Nuclear Cycle Gas Turbines

For the small Brayton cycle space power systems being developed, a recuperator is necessary to give good cycle efficiency and, hence, reduced heat rejection from the radiator. High effectiveness recuperators of plate-fin construction have been successfully built and tested to meet the demanding performance, structural, and life requirements for space application.

For the many fossil burning closed-cycle gas turbines, developed and built in Switzerland over the years, tubular recuperators have been utilized. Plate-fin recuperator technology has been developed to the point where units designed for high-pressure, closed-cycle systems offer substantial volume savings and can be more readily integrated with the turbomachinery and heat source. An example of a compact plate-fin recuperator used in a closed-cycle nitrogen system has been described by Bridgnell (47). The successful completion of the development program, and subsequent satisfactory performance of the recuperator in the system tests, demonstrated that recuperators using compact plate-fin heat-transfer surfaces have definite potential in closed-cycle gas turbine systems.

As higher temperature nuclear reactors are developed, direct gas turbine cycles become attractive, in that lower capital costs should be possible by using a simple gas cycle instead of two loops (one gas loop to cool the reactor core and one steam cycle for power generation). A 25-MW(e) helium closed-cycle gas turbine plant is presently under construction in Germany, and the recuperator details have been reported by Bammert (48). Even with the very high system pressure levels and helium working gas, both of which result in high heat-transfer coefficients, the recuperators constructed from plain, small diameter tubes are very large compared with the rotating machinery. Various recuperator types and flow configurations have been reported by Lys (49) and Malherbe (50), and the potential of the direct gas turbine cycle for advanced gas-cooled reactors has been reported by Melese-d'Hospital (51).

High levels of recuperator effectiveness are necessary to improve the cycle efficiency and, hence, reduce the thermal rejection to the environment, which can be significant for the large power plants of the future. While it is hard to relate the essentially thin-foil recuperator technology described in this paper with these large tubular recuperators, it is felt that by utilizing enhanced surface geometries of the type outlined, the unit sizes can be reduced. Close attention to the headering areas, manifolding, and ducting should result in a structurally sound

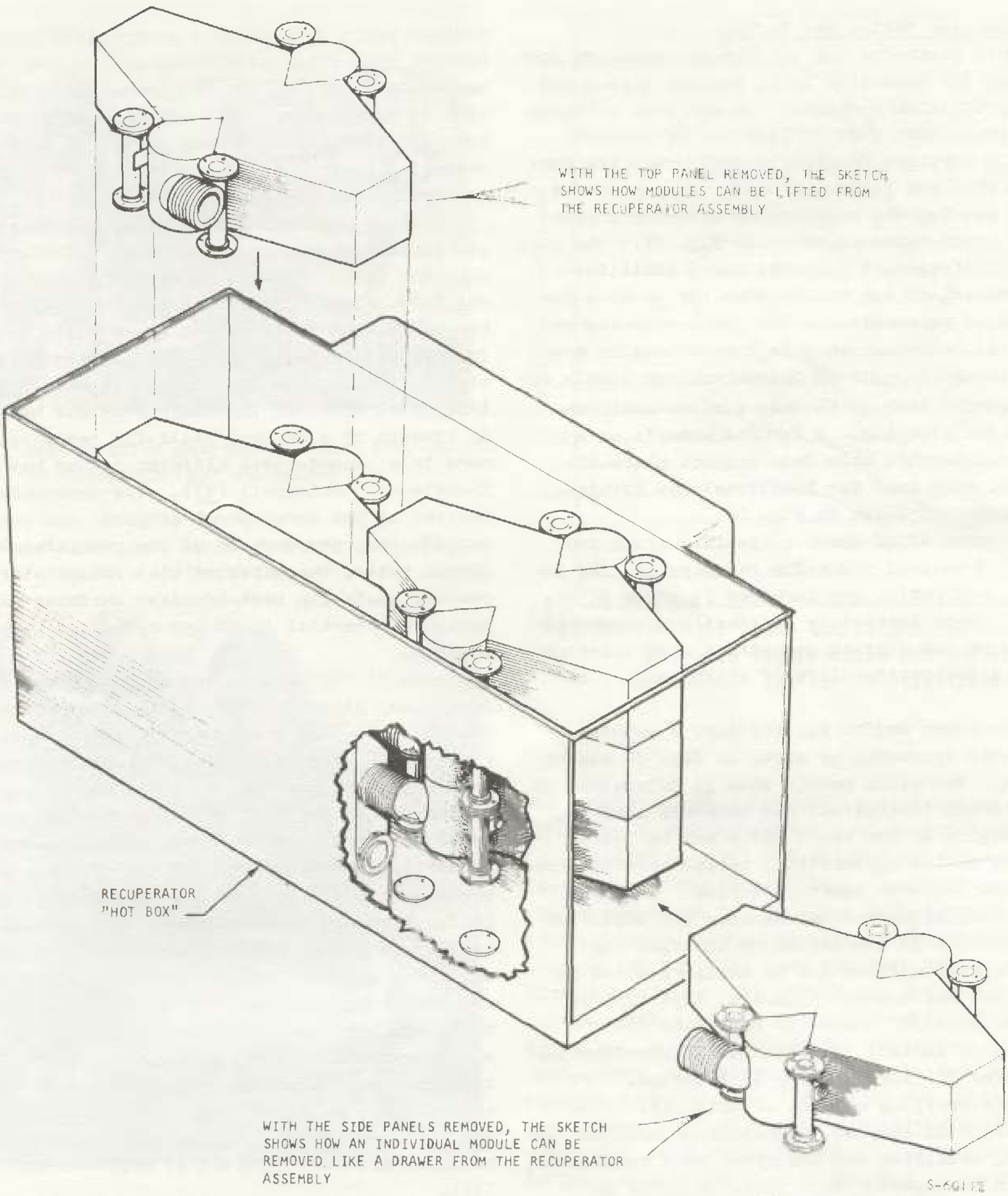


Fig. 30 View showing a modular plate-fin recuperator assembly for large industrial and marine gas turbine application

heat exchanger of reduced volume (and initial cost), and this is important for the integrated type of plant.

DISCUSSION

Excluding the aircraft field, all other forms of gas turbine plant can benefit from the inclusion of a heat-recovery device to improve the cycle efficiency. For the open-cycle gas turbine, the utilization of a heat-recovery system is optional and is very dependent on the application, the load profile, and fuel cost, etc. For the closed-cycle gas turbine, using either air, helium,

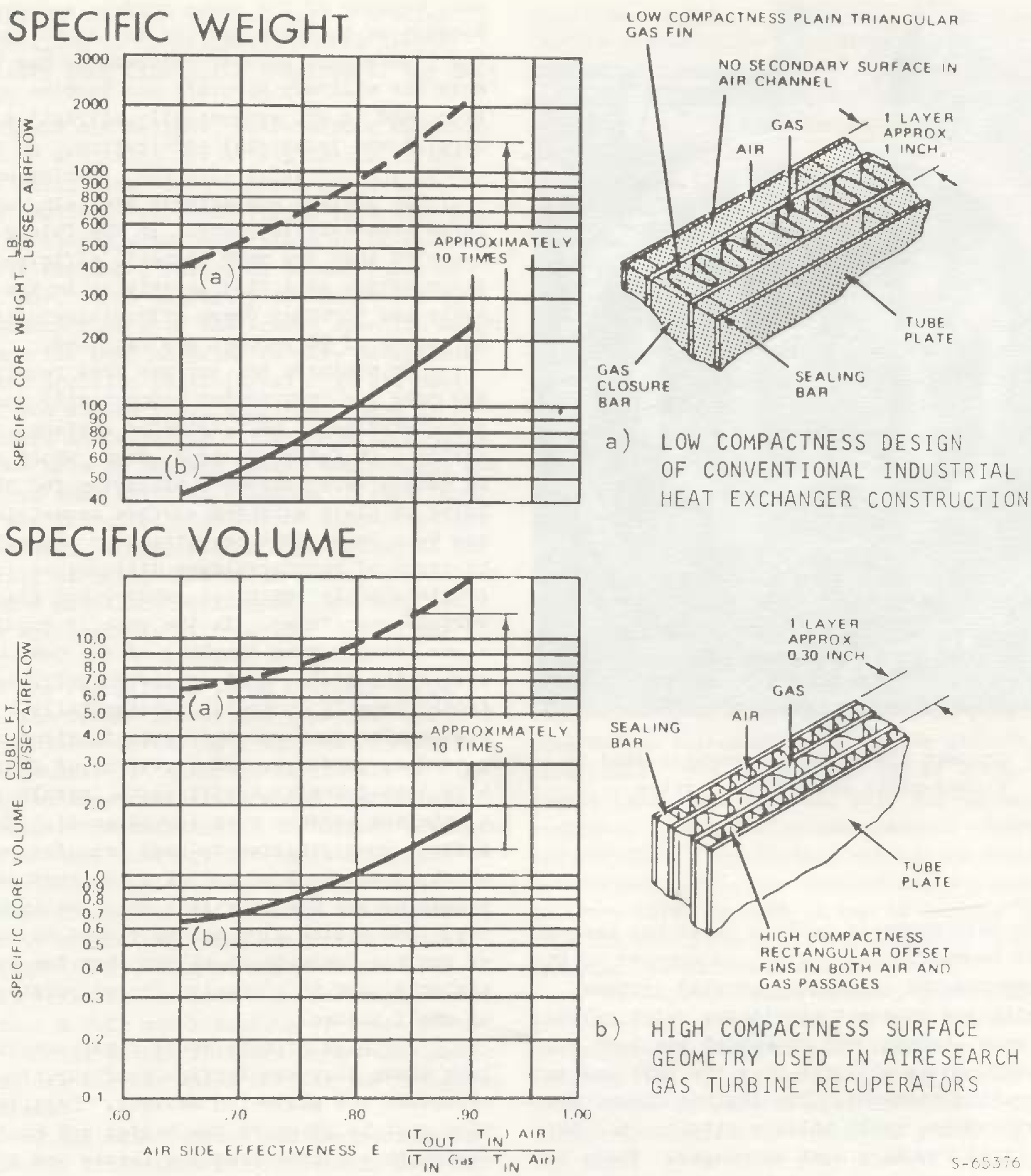


Fig. 31 Comparison of plate-fin recuperator specific sizes for industrial and marine gas turbine applications using conventional surfaces and high-performance surfaces developed from aerospace heat-exchanger technology

or monatomic gas mixtures as the working fluids, a recuperator is essential to give good cycle efficiency and hence, reduce the heat rejection to the environment. It has been shown that the recuperative cycle, in conjunction with variable geometry turbomachinery, gives a relatively flat specific fuel consumption over a wide range of power, and this is important, particularly for the vehicular gas turbine, where a considerable portion of the operating life may be at part-load

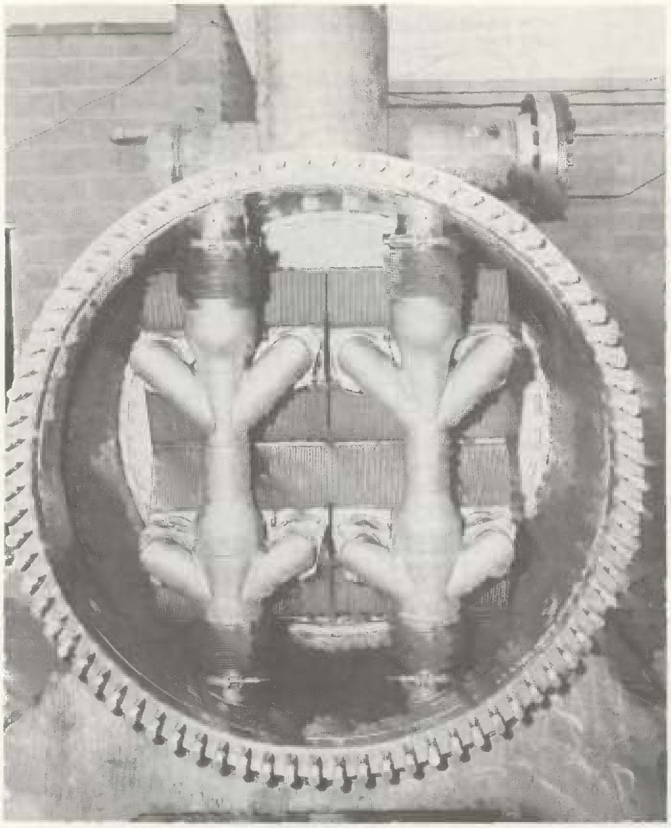


Fig. 32 Compact plate-fin recuperator used in closed-cycle gas turbine system

condition.

The main emphasis in this paper has been on the fixed boundary recuperator, as opposed to the rotary regenerator and liquid coupled systems, since, with its inherent simplicity, high reliability, zero leakage, and potential low cost, the recuperator can be utilized over the full gas turbine operating spectrum. The low horsepower vehicular gas turbine could utilize either a periodic flow or fixed boundary heat exchanger. There is no single criteria that dictates which of the two types offers the most viable solution. The regenerator, made from either metal or ceramic material, is currently being used in fairly low-pressure ratio cycles (up to about 5:1). To realize the true potential of the gas turbine, it is desirable to operate at higher compressor pressure ratios and turbine inlet temperatures. These increased specific power cycles tend to favor the recuperator because of the increased structural complexity, seal wear, and leakage problems associated with current rotary designs.

It has been shown that tubular designs, using small hydraulic diameter thin-walled tubes, are lighter than plate-fin variants but larger in volume, because of the lower surface compactness values. Because of the high cost of the tubes themselves, the lightweight tubular recuperator has been used only for military aircraft gas turbine application, and is not economically attractive for commercial and industrial application. At the other end of the operating spectrum, tubular units of very low surface compactness are being used for large closed-cycle plant. In the future, it is expected that the more compact, efficient, plate-fin surfaces will find acceptance in these closed-cycle gas turbines where effectiveness levels in excess of 90 percent are required.

To minimize the surface area requirements and make the recuperator economically acceptable, high-performance heat-transfer surfaces are necessary. In future designs, there would seem to be only a very limited application for plain tubular or plain extended surface geometries. It has been shown that heat-transfer augmentation by means of boundary-layer disturbance can be provisioned in practical tubular and plate-fin surface geometries. In the case of tubular designs, simply ring dimpling of the tube wall improves the inside heat-transfer coefficient (which usually controls) substantially. Offset rectangular fins are the most efficient of all secondary surfaces, and by virtue of their very high heat-transfer coefficients, result in units of minimum surface area (hence cost). They also possess good friction-to-heat transfer relations, which result in low frontal areas that facilitate packaging and compatibility with the turbomachinery. The offset fins can be formed in much higher surface compactness values than the tubular variants, and this results in recuperator cores of small volume.

For high effectiveness requirements, it has been shown that counterflow configurations are necessary for plate-fin designs. Detailed attention must be given to the header and manifolding design to minimize pressure losses and give good flow distribution into the pure counterflow portion. For high effectiveness tubular designs, multi-pass cross counterflow configurations are necessary, and the internal flow circuitry (i.e., high-pressure air inside or outside the tubes) is much dependent on structural and packaging requirements for the particular application.

One of the main problems in early recuperators was that of prolonged structural integrity in the cyclic environment associated with gas turbine operation. Many designs experienced fatigue failures, resulting from the high transient thermal stresses associated with the thermal inertia incompatibility of the core components. In current recuperators, careful attention to the core design has eliminated the high thermal stress problem and has resulted in configurations that satisfy the necessary long life goals required of the heat exchanger.

With the new designs, where strong emphasis is placed on achieving low stress levels in the core, the utilization of the lower grade alloys becomes a reality. For high volume vehicular production, it is projected that close to 80 percent of the heat-exchanger cost will be the basic material. Thus, it is mandatory to select the best surface geometry to give the lowest specific weight (lb/hp) and the best material to give acceptable recuperator specific costs ($/hp). In all heat-exchanger and engine cycle analyses, there is a big incentive in selecting designs and components that do not have to contain nickel and cobalt, and exclusion of such critical elements is very important for the projected high volume automotive gas turbine market. A much closer examination of the practicality of using mild steel (surface enriched to give good oxidation resistance) for many recuperator applications will be observed in the future.

For the recuperator to be a viable component for high volume production vehicular gas turbines, designs involving a minimum of manual labor are mandatory. Although it is hard to draw an analogy between recuperators and commercial heat exchangers, manufacturing techniques developed by the automotive industry for radiators, oil coolers, condensers, etc., will be applicable.

Problems experienced in early recuperators in the areas of fouling, dust congestion, fires, and operation in a marine environment have all been fully evaluated in both rigs and engine hardware. Since a full understanding of these phenomena now exists, design features have been incorporated in new engines which obviate the foregoing problem areas.

One of the main reasons the recuperator has not found wide acceptance for industrial gas turbine plant has been the high initial cost of the heat exchanger. This has resulted from the custom design of recuperators for each engine application and the very low production level. Rather than merely comparing specific fuel consumptions of various engines, an approach based on operating costs for recuperative and nonrecuperative engine variants has been presented. The economics of engine operation are very dependent on the assumptions made by each manufacturer, and the values given in this paper do not correspond to any particular problem statement, but are included to show a conveninet way of assessing engine economics during the preliminary design phase. It is very apparent that operating cost is very sensitive to fuel cost, and for a continuous duty machine, burning a relatively expensive fuel, the operating cost rapidly approaches the basic fuel cost per hour. It has been shown that for this type of operation, the heat-exchanged variant is far more economic, even allowing for periodic recuperator replacement. If the projected increases in both liquid and natural gas fuel costs are significant, then the recuperative cycle will find more acceptance in the industrial gas turbine market. The added cost of the heat exchanger can often be paid for in fuel savings after only a few hundred hours of operation. Similarly, the extra weight and volume of the recuperator can be recouped fairly quickly, and this is important for military and marine applications where fuel logistics may dictate the choice of prime-mover.

For each application, detailed investigations are required to define the engine-heat-exchanger interface to give the most cost-effective package that satisfies the particular envelope. In the current rotary regenerative engines, the heat exchanger is integrated with the turbomachinery, but it is not obvious that this is the most attractive configuration from the standpoint of simple maintenance. The heat exchanger must be located relative to the turbomachinery to give attractive gas flow paths. The recuperator can be either integrated (and could possibly form the turbomachinery backbone) or be an attached element. Certainly, for vehicular, industrial, and marine applications, the recuperator in the form of bolt-on module(s) is attractive, in that it can be quickly removed from the power plant for replacement, routine inspection, or maintenance.

As regards future trends, there will be continued efforts to identify more efficient heat-transfer surfaces and lower cost methods of construction. Long-term goals must include the utilization of ceramic materials, which offer virtually unlimited temperature potential and low cost. With their poor thermal conductivity and the high-pressure loading and manifold problems in the core, considerable development work will be required before a viable ceramic recuperator can be utilized for vehicular and gas turbines. Because these materials lack ductility, new design approaches must be taken to match the unique characteristics of the particular ceramic considered. Such work on an industrial gas turbine that utilizes a cycle with an "equi-pressure" ceramic recuperator has been reported by Miwa (52).

A fairly comprehensive list of references is included in this paper to cover various recuperator technology advancements over the last 30 years.

CONCLUSIONS

It is postulated that the gas turbine will find an ever-increasing role in the vehicular, industrial, marine, and closed nuclear cycle markets and may, by virtue of its low pollutant emissions, high reliability, and compact nature, emerge as the dominant prime-mover in the future.

The expected increase in both liquid and natural gas fuel costs makes the recuperative cycle look economically more attractive, and the associated increase in thermal efficiency will result in reduced thermal rejection to the environment. An obvious goal is to reduce current engine size and lower the initial cost ($/hp) by utilizing higher specific power cycles (i.e., increased compressure pressure ratios). For the higher pressure ratio cycles, the fixed boundary recuperator will offer a more reliable heat-exchanger solution, because of the increased structural complexity and seal leakage problems associated with the rotary ceramic regenerator.

It has been shown that utilization of compact, efficient heat-transfer surfaces (together with design and manufacturing techniques) developed primarily for aerospace application, can significantly reduce the volume and weight of the recuperator, and hence lower initial plant costs. While compact plate-fin and tubular recuperators have limited application to date, related high-performance aircraft heat exchangers, operating under high stress conditions in high-temperature environments, have accumulated millions of operating hours and have demonstrated a high degree of reliability. Technology, gained from related aerospace projects, has been utilized in the design and manufacture of recuperators currently being developed for a wide range of gas turbine applications.

To achieve attractive specific costs ($/hp) for the vehicular, industrial, and marine markets, utilization of the low-cost alloys, as opposed to the superalloys ($1.50 to $6/lb) used in the current recuperators is necessary.

Plate-fin recuperator designs lend themselves to the use of efficient compact surfaces, are adaptable to automation in high volume production, and can be made in forms possessing excellent heat transfer and flow friction characteristics. There is no reason to doubt that with development, the recuperator reliability will be made equal to that of the other major engine components. This being the case, the maintenance costs of the recuperative engine should essentially be the same as the simple-cycle engine.

Looking into the future, it would seem that the plate-fin type of construction will find increasing utilization by virtue of its high performance per unit of volume and cost and will replace tubular units currently being used for some gas turbine applications. Current development programs at AiResearch on advanced recuperator concepts have shown that by departing from traditional construction methods, significant performance, cost, reliability, and manufacturing advancements can be realized. With these new designs, plate-fin recuperator weights considerably less than 1 lb/hp can be achieved, and for high volume production, the recuperator cost should be no more than 10 percent of the overall engine cost.

Utilization of low-cost ceramic materials for all the high-temperature components, including the fixed boundary recuperator, will enable the full thermodynamic potential of the gas turbine to be achieved, while at the same time satisfying the demanding economic goals.

APPENDIX 1

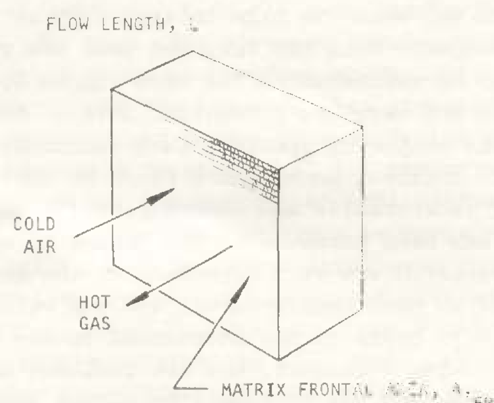
VOLUME CONTRIBUTION IN PLATE-FIN CONSTRUCTION

In order to make the method as general as possible, and apply to both primary and secondary surface construction, the fin efficiency in the extended surface case is assumed equal to unity, and the blockage due to finite fin thickness is considered negligible. The hot and cold sides of the heat exchanger should be treated separately and the contribution to each side to the total volume obtained.

Considering the cold side only, the following corrugation properties are defined:

Hydraulic diameter $d = \frac{4aL}{A}$

Core Volume $V = A_{FR}\ L$



COUNTERFLOW MODULE

It is now necessary to express this volume contribution in terms of the required heat transfer, allowable pressure drop, and fluid properties of the cold side of the heat exchanger. As mentioned previously, the heat-transfer requirements are best expressed as the product $(hA)_c$ for the cold side. The heat-transfer coefficient is defined as:

$$h = j \frac{W}{a} Cp/Pr^{2/3}$$

Substituting for h and A,

$$(hA) = \frac{j}{Pr^{2/3}} \cdot W \cdot \frac{4}{d} \cdot L \cdot C_p \qquad (1)$$

The pressure drop is given by the usual equation

$$\Delta P = \frac{4fL}{d} \left(\frac{W}{a}\right)^2 \cdot \frac{1}{2g\rho} \qquad (2)$$

From equation (1)

$$L = \frac{hA\ Pr^{2/3}\ d}{4j\ W\ C_p} \qquad (3)$$

From equation (2) $\left(\frac{W}{a}\right)^2 = \frac{2g\ \rho d\ \Delta P}{4fL}$ (4)

Combining equations (3) and (4) gives

$$a^2 = \frac{2f}{j} \frac{hA}{\Delta P} \frac{W \cdot Pr^{2/3}}{g\rho C_p}$$

$\therefore$ Heat Exchanger Frontal Area A_{FR}

$$\alpha \sqrt{f/j} \sqrt{\frac{W \cdot hA}{\Delta P}} \qquad (5)$$

(proportionality assumes constant fluid properties)

Combining equations (3) and (5) gives the core volume, V

$$\therefore \quad V \ \alpha \ d \sqrt{f/j^3} \sqrt{\frac{(hA)^3}{W\ \Delta P}} \qquad (6)$$

(for a given flow, effectiveness, and pressure loss)

and since $hA = W \cdot C_p \frac{\varepsilon}{1-\varepsilon}$ for $C_{min}/C_{max} = 1.0$, it follows that $hA \ \alpha \ W$ and equation (6) becomes

$$V \ \alpha \ d \sqrt{f/j^3} \sqrt{\frac{W^2}{\Delta P}} \qquad (7)$$

The core volume of a heat exchanger is equal to the heat-transfer area divided by the surface compactness which, in turn, is inversely proportional to the hydraulic diameter, i.e., Area α V/d, and since material weight and cost are related to area, it follows that:

$$\text{Core weight W (hence core material cost \$)} \ \alpha \sqrt{\frac{f}{j^3}} \sqrt{\frac{W^2}{\Delta P}} \qquad (8)$$

From the foregoing, it has been shown that the volume contribution is proportional to

$$d \sqrt{\frac{f}{j^3}}$$

However, this parameter varies with the type of surface and Reynolds number. The Reynolds number is implicit in the velocity as determined by equations (1) and (2).

$$Re = \frac{W\ d}{a\mu} \qquad (9)$$

Combining equations (1) and (2) gives the relationship:

$$Re \ \alpha \ d\sqrt{j/f} \qquad (10)$$

If the parameter, $d\sqrt{f/j^3}$, on a vertical scale is plotted against $\frac{Re}{d}\sqrt{f/j}$ on a horizontal scale, a given design problem immediately fixes a point on the horizontal scale, irrespective of the surface used. The vertical reading is then proportional to the volume contribution of various surfaces giving identical thermal conductance and pressure loss values.

In Fig. 9, values of $d\sqrt{f/j^3}$ (suitably corrected in the case of tubular geometries) have been plotted against

$$\frac{Re}{d}\sqrt{f/j}$$

for a number of different surface geometries. These surfaces, which include tubular and a variety of plate-fin variants, are shown merely as examples of the relative merit of different geometries for gas turbine recuperator application.

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