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ABSTRACT

The results of a preliminary analysis of a single shaft regenerative design with a single stage radial turbine are presented to show the fuel economy that can be achieved at high turbine inlet temperatures, with this particular advanced design, if the turbine tip speed and regenerator inlet temperature are not limited. Results are also presented to show the effect of imposing such limitations. The engine size was 100 hp (74.6 kW) for application to a 3500 lb (1588 kg) auto. The fuel economy was analyzed by coupling the engine to the auto through a continuously variable speed-ratio transmission and operating the system at constant turbine inlet temperature over the Composite Driving Cycle. The fuel was gasoline and the analysis was for a 85° F (29° C) day. With a turbine inlet temperature of 2500° F (1644 K) the fuel economy was 26.2 mpg (11.1 km/l), an improvement of 18 percent over that of 22.3 mpg (9.5 km/l) with a turbine inlet temperature of 1900° F (1311 K). The turbine tip speed needed for best economy with the 2500° F (1644 K) engine was 2530 ft/sec (770 m/s). The regenerator temperature was approximately 2200° F (1478 K) at idle. Disk stresses were estimated for one single stage radial turbine and two two-stage radial-axial turbines and compared with maximum allowable stress curves estimated for a current ceramic material. Results show a need for higher Weibull Modulus, higher strength ceramics.

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SUMMARY

This paper presents, first, the results of a preliminary analysis of the fuel economy improvement that can be achieved with a high temperature advanced automotive gas turbine engine. In this analysis, two problem areas were identified: high turbine tip speeds and high regenerator inlet temperatures. The particular engine analyzed was a fixed geometry single shafts regenerative design with a single stage radial compressor and a single stage radial turbine. The engine size was 100 hp (74.6 kW) for application to a 3500 lb (1588 kg) auto. The fuel economy was analyzed by coupling the engine to the auto through a continuously variable speed-ratio transmission and operating the system over the Composite Driving Cycle. The fuel was gasoline and the analysis was for an 85° F (29° C) day. The analysis was performed first without any limitations on the turbine tip speed or the regenerator temperature, then with limitations to determine their effect on the fuel economy.

The results showed that without the limitations, raising the design turbine inlet temperature from 1900° F (1311 K) to 2500° F (1644 K) improved the fuel economy 18 percent, from 22.3 mpg (9.5 km/l) to 26.2 mpg (11.1 km/l). The turbine tip speed needed for best economy with the 2500° F (1644 K) engine was 2530 ft/sec (770 m/s) compared with 2240 ft/sec (683 m/s) for best economy with the 1900° F (1311 K) engine. The maximum regenerator temperature was approximately 2200° F (1478 K) at idle.

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Limiting the turbine tip speed to 1600 ft/sec (488 m/s) by reducing the diameter in both the 2500° F (1644 K) and 1900° F (1311 K) engines reduced the improvement to 11 percent. Limiting the regenerator temperature in the 2500° F (1644 K) engine to 1790° F (1250 K), the value at maximum power, reduced the fuel economy 4.6 percent.

The paper then presents the disk stresses estimated in three sample cases of turbine designs and compares them with estimated maximum allowable stress curves using a hot pressed silicon nitride as representative of a currently available ceramic material. The maximum tensile principal stress in a single stage radial turbine was estimated as 65 000 psi (448 MN/m^2) at the hub. The stress was reduced to 42 000 psi (290 MN/m^2) by reducing the diameter and adding an axial stage. The stress in the axial stage was estimated as 29 500 psi (200 MN/m^2). By reducing the design rotational speed, the stress in the radial turbine of the two-stage design was reduced to 32 000 psi (221 MN/m^2) and 25 000 psi (172 MN/m^2) in the axial stage. Comparison of these stresses with the maximum allowable stresses estimated to provide a component reliability against failure of 99 percent showed that improved ceramic materials are needed.

INTRODUCTION

The ERDA and NASA are engaged in a joint program to develop an automotive gas turbine engine/vehicle system as an alternative to the spark-ignition engine/vehicle system. The long range goal of this program is an advanced gas turbine engine/vehicle system that will achieve a major improvement in efficiency, use minimum scarce materials, be capable of using alternate fuels, have low emissions, and be competitive in cost with a comparable size spark-ignition engine/vehicle system.

To achieve the major improvement in efficiency and thus fuel economy (more miles per gallon), the advanced gas turbine for this system will have to be designed for much higher turbine inlet temperatures than are considered for current or near term automotive gas turbine engines, which are around 1900° F (1311 K). Temperatures as high as 2500° F (1644 K) are being considered.

Ceramics are proposed for use at these temperatures to eliminate or at least minimize the need for cooling and hence the performance loss that usually accompanies cooling. Minimizing the need for cooling also will allow a simpler engine design which very likely will make it cheaper to produce. In addition the use of ceramics will conserve scarce materials, reduce inertia of rotating components, and reduce overall engine weight.

In a preliminary analysis (1) at the Lewis Research Center of the fuel economy improvement that can be achieved with a high temperature advanced automotive gas turbine engine, two problem areas were identified: high turbine tip speeds and high regenerator inlet temperatures. The particular engine analyzed was a fixed geometry single shaft regenerative design with a single stage radial compressor and a single stage radial turbine. The engine size was 100 hp (74.6 kW) for application to a 3500 lb (1588 kg) auto. The fuel economy was analyzed by coupling the engine to the auto through a continuously variable speed-ratio transmission and operating the system over the Composite Driving Cycle. The fuel was gasoline and the analysis was for a 85° F (29° C) day. The analysis was performed first without any limitations on the turbine tip speed or regenerator temperature, then with limitations to determine their effect on the fuel economy. The problem associated with high turbine tip speeds is high stresses in the turbine disk. The problem assicated with high regenerator inlet temperature is the general integrity and durability of the regenerator, its drive system and seals.

To get an idea of the magnitude of the turbine stress problem, the stresses were estimated in three cases of turbine designs: a single stage radial, and two two-stage turbines using a radial and axial combination with lower tip speeds.

The purpose of this paper is

(1) to present some of the results that show the fuel economy potential of the design considered and the effect of the turbine and regenerator limitations,

(2) to present the estimates of the turbine disk stresses, and

(3) to show a comparison of these stresses with the maximum allowable stresses estimated for a current ceramic material.

The objective is to emphasize the need for higher strength, higher quality ceramic materials than are currently available.

FUEL ECONOMY POTENTIAL

The single shaft gas turbine that was analyzed in reference 1 for fuel economy potential is shown schematically in figure 1. It consisted of a fixed geometry single-stage radial compressor, a conventional combustor, a fixed geometry single-stage radial turbine and a regenerator. All parts in the hot section of the engine were assumed to be ceramic and to need no cooling. The engine was sized for a maximum net power output to the transmission of 100 hp (74.6 kW).

For analysis over a driving cycle, the engine was coupled to a continuously variable speed-ratio transmission (CVT) through a speed-reduction gear box and applied to a compact car. The CVT was a hydromechanical type. The weight (loaded) of the car was 3500 lb (1588 kg). The analysis was first conducted at constant turbine inlet temperature without turbine tip speed or regenerator temperature limitations. Then the design was examined for the effect of imposing such limitations.

A complete discussion of the assumptions and methods of analysis can be found in reference 1.

Without Limitations on Turbine or Regenerator

The fuel economy potential of the engine/vehicle system without limitations on the turbine or regenerator is shown in figure 2. The fuel economy in miles per gallon (mpg) and kilometers per liter (km/l) is shown as a function of constant turbine inlet temperature.

The engine/vehicle system was evaluated over the Composite Driving Cycle. This cycle is a combination of city and highway driving such that the average vehicle speed is 33 mph (53 km/hr). The results are based on an 85° F (29° C) day at sea level using gasoline as the fuel. The regenerator effectiveness was assumed to be 0.90.

Figure 2 shows the significance of increasing the turbine inlet temperature to obtain improvement in fuel economy. The pressure ratios for best economy at three temperature levels are noted on the figure. For example, at 2500° F (1644 K), the best pressure ratio is 5. With these design conditions, the fuel economy potential is 26.2 mpg (11.1 km/l). With a 1900° F (1311 K) turbine inlet temperature and a compressor pressure ratio of 4.4, the fuel economy potential is 22.3 mpg (9.5 km/l). Thus raising the turbine inlet temperature from 1900° F (1311 K) to 2500° F (1644 K) improved the fuel economy by 18 percent. With a temperature of 2200° F (1478 K) and a pressure ratio of 4.5, the improvement would be 10 percent, to 24.5 mpg (10.4 km/l).

If, for comparison, 19.5 mpg (8.3 km/l) is taken as an average mileage figure for 1976 models of comparable weight from EPA listings, the 26.2 mpg (11.1 km/l) figure for the 2500° F (1644 K) engine is a 34 percent improvement. At the same time, it should be noted here that the advanced gas turbine that was analyzed was with fixed geometry and with a regenerator effectiveness of 0.90. With variable geometry to improve performance at part power, the fuel economy figure could be improved, but at the expense of complexity. If a higher regenerator effectiveness could be assumed, the fuel economy certainly would be improved, but at the sacrifice in size and weight.

Some pertinent design details of the 2500° F (1644 K) engine are shown in Table I. Particularly important to note is the high turbine tip speed of 2530 ft/sec (770 m/s). Although not shown in Table I, operation at a constant 2500° F (1644 K) also resulted in high regenerator operating temperatures. For example, at idle conditions it was 2200° F (1478 K).

Effect of Limiting Turbine Tip Speed

High tip speeds imply high rotor disk stresses, so it is important to look at the effects of limiting the tip speeds. The effect on fuel economy is shown in figure 3 where the fuel economy with various turbine inlet temperatures is plotted as a function of turbine tip speed. The tip speed was reduced by reducing the diameter of the turbine while maintaining the rotational speed. Because the loading on the turbine is increased as the turbine tip speed is decreased, the efficiency of the turbine and the engine decreases and consequently the fuel economy decreases as shown. Imposing a limit of 1600 ft/sec (488 m/s) on both the 2500° F (1644 K) and 1900° F (1311 K) engines, for example, reduces the fuel economy improvement of the 2500° F (1644 K) engine to 11 percent over that of the 1900° F engine (1311 K).

Turbine-tip speed, and hence turbine stress, can also be reduced without sacrificing performance by adding another turbine stage and reducing the loading on the turbine. Reducing the loading allows reducing the diameter of the turbine and consequently the tip speed. Another way of reducing tip speed is to lower the rotational speed of the turbine. This approach, most likely, will adversely effect the performance of both the compressor and turbine and thereby compound the effect on the performance of the engine resulting in substantially poorer fuel economy.

The actual effect on engine performance and fuel economy of going to two turbine stages and of lower rotational speed were not analyzed in reference 1. However, the effects of these approaches on disk stresses were considered for this paper and are presented under "TURBINE STRESS ESTIMATES."

Effect of Limiting Regenerator Inlet Temperature

The fuel economy of the engine/vehicle system over the driving cycle was so far, presented for operation with constant turbine inlet temperature and with no limitations imposed on the regenerator inlet temperature. Material considerations, however, may require limits on this temperature. The results if such limitations are imposed are shown in figure 4 for the case of turbine inlet temperature of 2500° F (1644 K). With the turbine inlet temperature held constant at this temperature, the temperature out of the turbine and into the regenerator varies from 1790° F (1250 K) at full power to 2200° F (1480 K) at idle. If the temperature is limited to that at idle, which in essence is no limitation, the maximum economy is realized, that is, 26.2 mpg (11.1 km/l). If the temperature is limited to 500° F (278 K) below that of the turbine inlet temperature, that is, 2000° F (1367 K), the fuel economy is reduced to 26 mpg (11.1 km/l) or a negligible amount. But, if the temperature is limited to that at full power, that is 1790° F (1250 K), so that the turbine inlet temperature must be reduced over the entire power range, the fuel economy is reduced 4.6 percent to 25 mpg (10.6 km/l). So to realize the full fuel economy potential of the engine with 2500° F (1644 K) turbine inlet temperature, it is important that the regenerator material have capabilities of withstanding temperatures from 1800° F (1256 K) to 2000° F (1367 K), and possibly to 2200° F (1478 K).

TURBINE STRESS ESTIMATES

Three turbine designs were investigated for estimates of the level of stresses that would be encountered in these small high speed gas turbines. The three cases with some pertinent design details are listed in Table II. Case A is a single stage radial (1R) design with high tip speed. Case B is a two-stage turbine with one radial (1R) stage and one axial (1A) stage designed for the same rotational speed as in Case A. The reduction in loading which results from adding a stage and dividing the work permits a reduction in the tip speed without a sacrifice in performance. The lower tip speed is, of course, reflected in a smaller diameter turbine. Adding a stage, as the table shows, has significantly reduced the tip speed.

Case C is the same arrangement as Case B but with the tip speed further reduced by reducing the rotational speed. Here, however, there will be a sacrifice in efficiencies of both the compressor and turbine.

Typical profiles of the turbine disks in these cases of turbine designs is shown in figures 5 and 6 which are for Case B. Figure 5 is the radial turbine and figure 6 is the axial turbine. Only the stresses in the disks were estimated in this study. The disk shapes were arrived at by starting with a 20 000 psi (138 MN/m^2) constant stress disk and superimposing the load of the blades on it. A phantom outline of the blades is shown in figures 5 and 6. An axisymmetrical finite element computer program was used in calculating the stresses. The program computes the stresses in the radial, tangential, and axial directions and the shear stresses in the radial-axial plane. These stresses are then combined by the distortion energy theory into equivalent stresses. Profiles of these stresses are shown in figures 5 and 6 with the maximum and minimum values noted and located. The maximum values are listed for each turbine stage in Table II. The stresses in each direction and the shear stress were also used to compute the maximum positive (tensile) principal stress in each element. This stress is also listed for each turbine stage in Table II.

A temperature profile was assumed through each disk to account for stresses due to thermal gradients on a steady state basis. For example, the temperature profile on the forward face of the radial turbine in figure 5 was assumed to vary linearly from 2500° F (1644 K) at the tip to 1500° F (1089 K) at the hub. On the forward face of the axial turbine in figure 6, the temperature profile was assumed to vary linearly from 1840° F (1278 K) at the rim to 1400° F (1033 K) at the hub. The physical properties of the material used in the stress analysis are listed in Table III. They are for NC-132 hot pressed silicon nitride (2).

The results of the stress analysis, as expected, show a very high stress for the single stage radial turbine with the high tip speed. Adding an axial stage and reducing the tip speed by either reducing the diameter or the speed, reduces the stresses substantially. To judge the severity of the levels, it is necessary to look at the strength characteristics of the material.

CERAMIC DESIGN STRESS ESTIMATES

Estimates of design stress curves for NC-132 hot pressed silicon nitride are shown in figure 7. The figure shows a plot of the average 4 point Modulus of Rupture (MOR) stresses for the material versus temperature, based on data from reference 2. The Weibull Modulus, m, a measure of scatter in the MOR test data was estimated to be about 10. The mean stresses of the material decrease gradually with temperature until 1500° F (1089 K) at which point they drop off rapidly. Two estimated design stress curves, arrived at by the approach in reference 3, are included on the figure. Also plotted are the maximum positive principal and equivalent stresses from "TURBINE STRESS ESTIMATES."

The lower design stress curve in figure 7 is considered a rough estimate of the allowable stresses which will result in a component in-service reliability against failure of

99 percent, based on MOR data for a material with a Weibull Modulus of 10 and an effective volume ratio of 400. The upper design stress curve is for a Weibull Modulus of 20. It is included to show roughly how the allowable stress can be increased by improving the quality of the ceramic. The lower curve shows an almost linear variation in the allowable stress from about 47 000 psi (324 MN/m^2) at room temperature to 43 000 psi (296 MN/m^2) at 1500° F (1089 K) and 16 000 psi (110 MN/m^2) at 2500° F (1644 K). By improving the Weibull Modulus to 20, the allowable stress at room temperature can be raised to 58 000 psi (400 MN/m^2), at 1500° F (1089 K) to 54 000 psi (372 MN/m^2), and at 2500° F (1644 K) to 20 000 psi (138 MN/m^2).

Comparing the maximum principal stresses estimated in the three cases of turbine designs, Table II, with these curves, assuming the temperature at the point of maximum stress is around 1500° F (1089 K), it is clearly seen that the stress in the single stage radial is much too high. A material with higher strength, or an m greater than 20 would be needed to bring the stress within that allowable. The stresses in the radial turbines in Cases B and C fall within the lower curve but if the temperature is higher than assumed the stresses would be marginal and a material with a higher Weibull modulus would be needed. The stresses for the axial turbines appear to be well within the curve unless the temperature is significantly higher than assumed.

Comparing the maximum equivalent stresses plotted in figure 7 as solid symbols at an assumed temperature of 1550° F, with these curves, now shows more definitely that the radial turbine in Case B would need a better material. The radial turbine in Case C is just below the lower curve and would be considered marginal. The axial turbines still fall within the lower curve. But these are first estimates. Before more confident judgements can be made, more detailed stress analysis would have to be made. Also, time dependent failure modes and such factors as oxidation, erosion, impact, cyclic loading, creep and methods of attachment would have to be considered.

The evidence here does show, however, that to realize such engine designs as considered here, particularly the simple design with a single stage turbine, it is necessary to raise the allowable stress curve. The most effective way is to increase the Weibull Modulus of the material. This means improving the quality of the material by reducing the maximum flaw size and assuring a uniform flaw distribution. A Weibull Modulus greater than 20 is needed.

CONCLUDING REMARKS

This paper has illustrated the fuel economy improvement that can be achieved with a high temperature advanced automotive gas turbine. The particular engine used for illustration was a single shaft, regenerative design with fixed geometry, a single stage radial compressor, and a single stage radial turbine. It has been shown that the fuel economy in this particular design was improved 18 percent, from 22.3 mpg (9.5 km/l) to 26.2 mpg (11.1 km/l), by raising the design turbine inlet temperature from 1900° F (1311 K) to 2500° F (1644 K). The engine was sized for 100 hp (74.6 kW) and applied to a compact car of 3500 lb (1588 kg) weight through a continuously variable speed-ratio transmission. The engine/vehicle system was evaluated over the Composite Driving Cycle at constant turbine inlet temperature using gasoline as the fuel on an 85° F (29° C) day. To get this improvement, a turbine tip speed as high as 2530 ft/sec (770 m/s) was needed with a single stage turbine. An estimate of the stresses in such a turbine showed levels in excess of the allowable stress estimated for a current hot pressed silicon nitride ceramic where the allowable stress is limited by the scatter in properties. A material with a Weibull Modulus greater than 20 would be needed. Two stage turbine designs reduced the stress substantially but could benefit by better quality ceramics. Thus, the results here emphasize the need for developing higher Weibull Modulus, higher strength ceramics if the potential of advanced gas turbines such as were analyzed here are to be realized.

REFERENCES

1. Klann, J. L.; and Tew, R. C.: Analysis of Regenerated Single-Shaft Ceramic Gas-Turbine Engines and Resulting Fuel Economy in a Compact Car. NASA TM X-3531, 1977.
2. Design Fabrication and Spin Testing of Ceramic Blade-Metal Disk Attachment. NAS3-19715, Pratt & Whitney Aircraft, Division of United Technologies.
3. D. G. S. Davies, "The Statistical Approach to Engineering Design in Ceramics," in D. J. Godfrey, Editor, Ceramics for Turbines and Other High-Temperature Engineering Applications, p. 429-451, British Ceramic Soc. (1973).

TABLE I. - PERTINENT DESIGN DETAILS OF 2500° F ENGINE

Compressor	
Pressure ratio	5
Airflow, lb/sec (kg/sec)	.73 (.33)
Specific speed, dimensionless	.775
Tip diameter, in (cm)	3.12 (7.92)
Efficiency, percent	78.6
Tip speed, ft/sec (m/s)	1602 (488)
Turbine	
Specific speed, dimensionless	.581
Tip diameter, in (cm)	4.93 (12.52)
Efficiency, percent	.85
Tip speed, ft/sec (m/s)	2530 (770)
Regenerator effectiveness	.90
Engine speed, rpm	117 700
Horsepower output, hp (kW)	100 (74.6)
SFC lb/hp-hr (gm/hr-W)	.360 (.219)

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TABLE II. - TURBINE DESIGNS FOR STRESS ESTIMATES

Case	A	B		C	
Stages	IR	IR	IA	IR	IA
Speed, rpm	122 000	122 000		74 000	
Tip, diameter, in. (cm)	4.66 (11.84)	3.31 (8.41)	3.31 (8.41)	4.80 (12.19)	4.80 (12.19)
Tip speed, ft/s (m/s)	2479 (756)	1763 (537)	1763 (537)	1550 (472)	1550 (472)
Airflow, lb/s (kg/s)	.66 (.30)	.66 (.30)		.72 (.33)	
Maximum equiv. stress, lb/in^2 (MN/m^2)	82 000 (565)	49 000 (338)	37 400 (258)	40 400 (279)	31 000 (214)
Maximum princ. stress, lb/in^2 (MN/m^2)	65 000 (448)	42 000 (290)	29 500 (200)	32 000 (221)	25 000 (172)

TABLE III. - PHYSICAL PROPERTIES OF NC-132 HP SILICON NITRIDE

Density	lb/in^3 (g/cc)		.12 (3.2)
Modulus of elasticity	lb/in^2 (MN/m^2) lb/in^2 (MN/m^2)	68° F (293 K) 2500° F (1644 K)	46×10^6 (320×10^3) 35×10^6 (240×10^3)
Coeff. of thermal expansion	in/in-°F (m/m-K)	68°-1800° F (293-1273 K)	1.8×10^{-6} (3.2×10^{-6})
Thermal conductivity	Btu/hr-ft-°F (W/m-K) Btu/hr-ft-°F (W/m-K)	68° F (293 K) 2500° F (1644 K)	19 (33) 9 (16)
Specific heat	Btu/lb-°F (J/gm-K)	68° F (293 K)	.16 (.67)

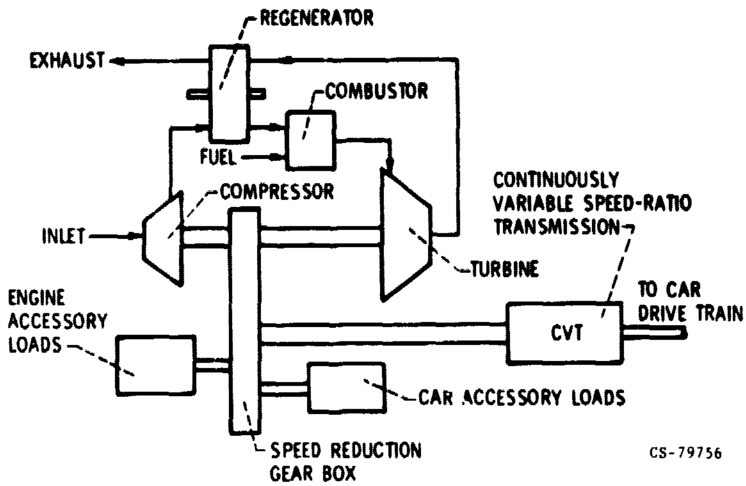

Figure 1. - Schematic diagram of engine/transmission arrangement.

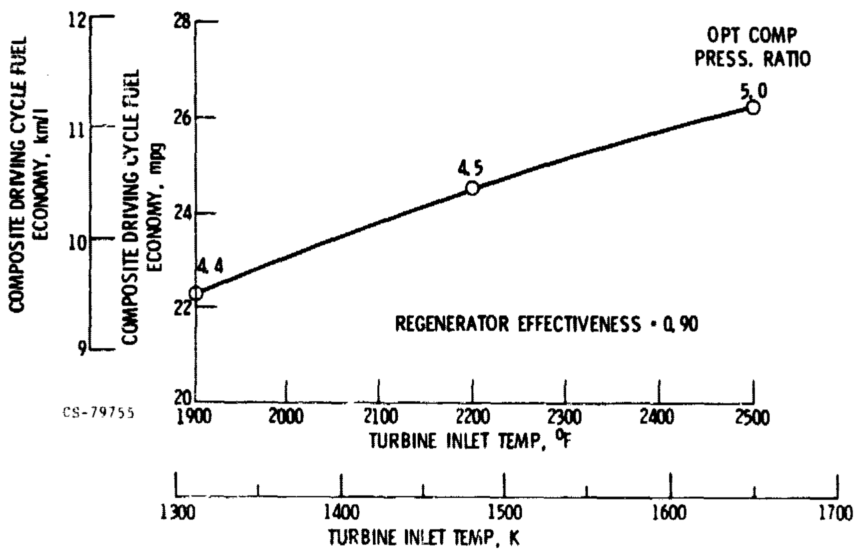

Figure 2. - Variation in fuel economy with turbine inlet temperature.

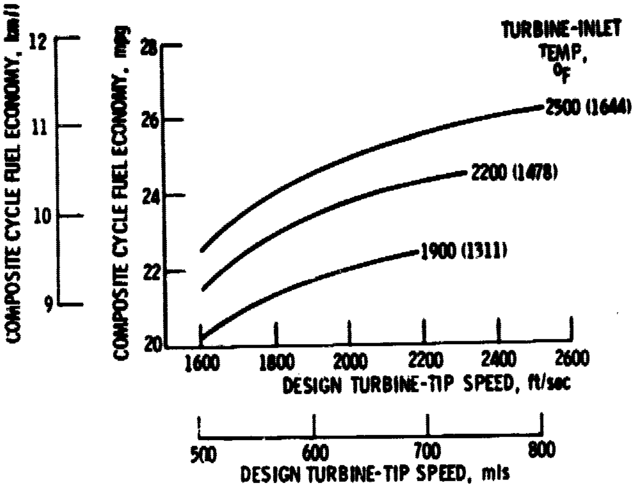

Figure 3. - Effect of reduced turbine diameter on design tip speed and fuel economy. Gas-turbine operation at constant turbine-inlet temperature; regenerator effectiveness, 0.90.

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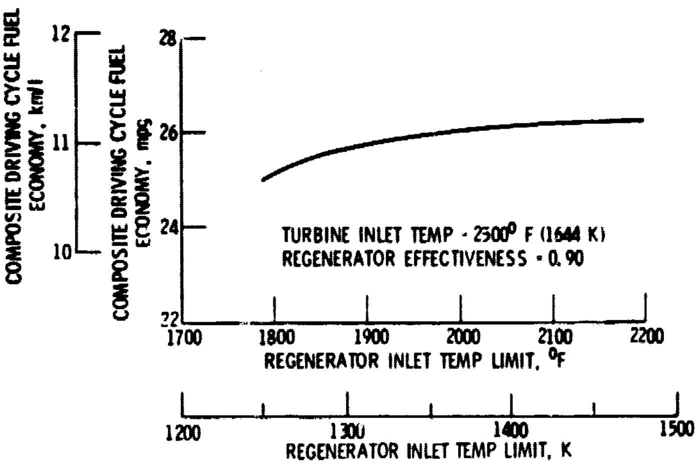

Figure 4. - Effect of regenerator temperature limit on fuel economy.

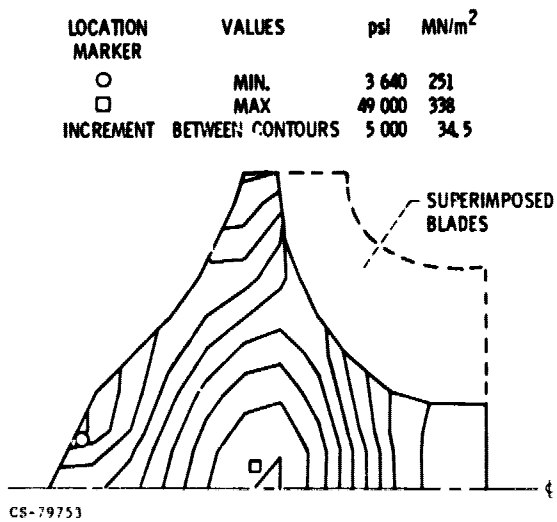

Figure 5. - Equivalent stress contours in radial turbine, case B (table II).

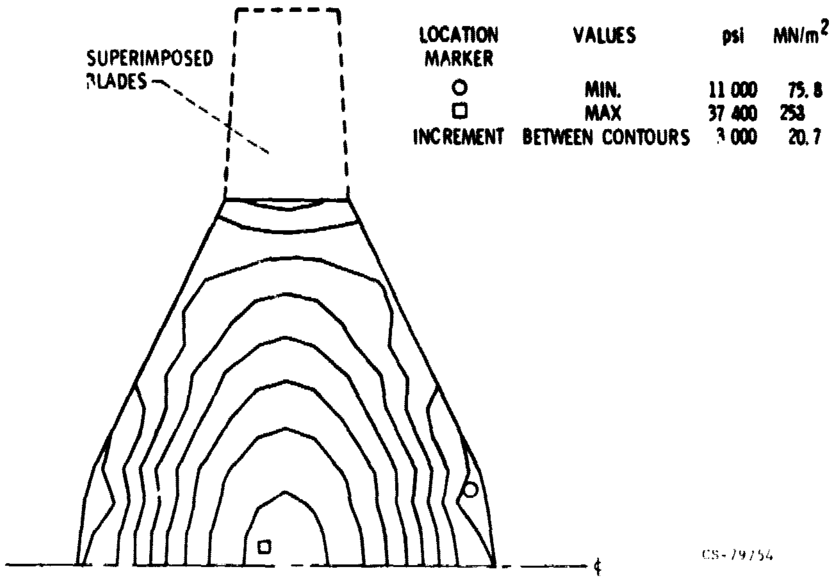

Figure 6. - Equivalent stress contours in axial turbine case B (table II).

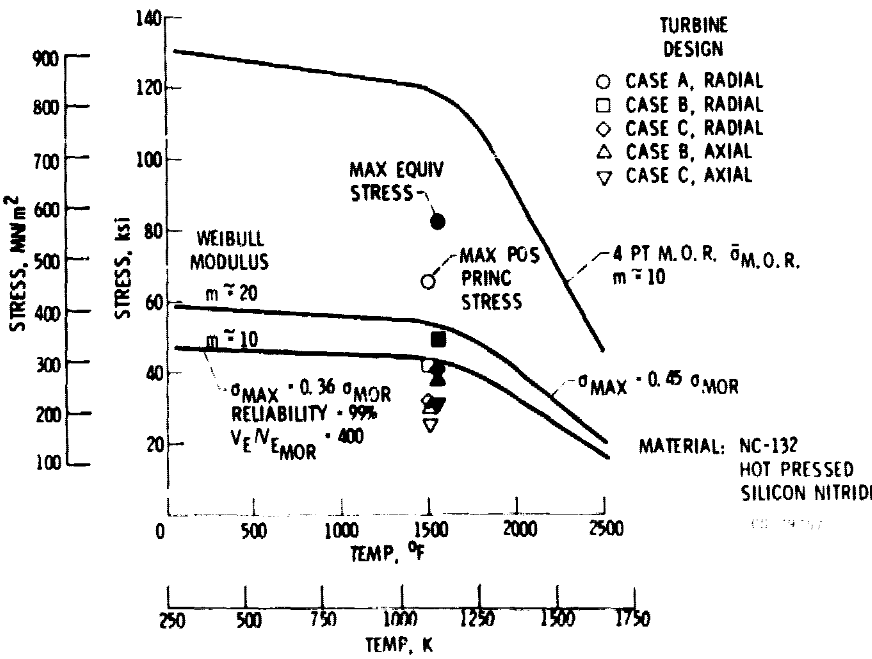

Figure 7. - Estimated design stress curves.