RN-TM-0166

10 September 1965

REON TECHNICAL MEMORANDUM

MASTER

MARK III, TURBINE ASSEMBLY

THERMAL ANALYSIS

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ABSTRACT

Presented herein are the results of a thermal analysis run to determine the steady-state temperatures of the Mark III turbine. Because of the uncertainty associated with the flow patterns of the bearing coolant once it reaches the main drive gas stream, two methods of analysis were used in determining gas recovery temperatures. The actual distributions will be between those obtained by these two extreme methods.

This analysis was performed prior to the failure of TPA PN 287900-179 S/N 013 (first stage rotor) and temperature distributions have been published for the conditions of failure. The purpose of this report is to demonstrate that the most severe thermal gradients are obtained by assuming that the bearing coolant flow mixes completely with the main drive gas before entering the first stage rotor rather than assuming that the coolant exists as a discrete layer while passing through the first stage rotor.

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LIST OF SYMBOLS

A	Area - ft^2
C	Specific heat - $\text{Btu/lbm}^\circ\text{R}$
D	Diameter - feet
D_h	Hydraulic diameter - feet
h	Heat transfer coefficient - $\text{Btu/ft}^2\text{hr}^\circ\text{R}$
k	Conductivity - $\text{Btu/hr ft}^\circ\text{R}$
N_{u_r}	Angular Nusselt number - dimensionless
P	Pressure - psia
Pr	Prandtl number - cu/k - dimensionless
Re	Reynolds number - $DV\rho/\mu$ - dimensionless
Re_w	Angular Reynolds number - $\frac{w r^2}{\nu}$ - dimensionless
R_t	Recovery factor
S	Film thickness - feet
T	Temperature - $^\circ\text{R}$
T_r	Recovery temperature - $^\circ\text{R}$
V	Velocity - ft/sec
X	Distance downstream from origin - feet

Greek Letters

η	Efficiency - percent
μ	Viscosity - lb/ft sec
ν	Kinematic viscosity - ft^2hr
ρ	Density - lb/ft^3
ω	Angular velocity - sec^{-1}

Subscripts

aw Adiabatic wall (recovery)
b Bulk
c Cold gas
g Hot gas
n Stage number
o Stagnation conditions
oi Inlet stagnation condition
s Static condition
w Wall
oo Freestream - hot gas

I. INTRODUCTION

The object of this report is to present the results of a steady-state thermal analysis of the Mark III turbine assembly for turbine inlet drive gas stagnation temperature and flow rate conditions of 1350°R and 9.06 lb/sec. Hydrogen at a temperature of 120°R is assumed to flow from the bearing coolant flow regulating labyrinth at a flow rate equal to 7% of the turbine drive gas flow rate.

II. SUMMARY AND CONCLUSIONS

A. The IBM Computer Program, 278B (Reference 1), which features a two-dimensional finite difference solution, was used in obtaining the steady-state temperature profiles. Because of the complexity of the flow patterns, at the point where the bearing coolant meets the main gas stream, two methods of analysis were used in determining the gas recovery temperatures.

B. The first method of analysis assumes complete mixing of the turbine drive gas and bearing coolant at the first stage rotor entrance. Figure 1 is a plot of the gas recovery temperature versus location and Figure 2 gives the steady-state temperature profiles for the rotor assembly. The most severe temperature gradient (900°R/inch) occurs in the outer portion of the first stage rotor disk. Figures 3 and 4 are the steady-state temperature profiles for the stator and rotor vanes, respectively, assuming complete mixing. The temperature gradients which exist across the vanes are due to the change in gas properties with area ratio. In this analysis, the vanes are treated as converging-diverging nozzles and gas properties are calculated assuming one-dimensional isentropic flow. Sonic flow is assumed at the nozzle throat with a shock at the exit.

C. The second method of analysis uses a correlation equation developed by Tribus and Klein (Reference 2) in their study of gas-film-cooled nozzles and assumes that no mixing occurs between the bearing coolant flow and the main drive gas. The bearing coolant gas film is assumed to enter the first stage rotor vanes as a discrete layer (no mixing with hot turbine gases). This layer then clings to the first stage rotor inner diameter, goes through the second stage nozzle labyrinth seal and then clings to the second stage rotor inner diameter. Figure 1 presents the gas recovery temperature versus location obtained from the Tribus-Klein correlation. Using the Tribus-Klein solution, the most severe temperature gradient (1200°R/inch) occurs at the root of the first stage rotor vane; whereas, assuming complete mixing, the most severe gradient occurs in the outer portion of the first stage rotor disk. Less confidence is placed in the no mixing results because of the assumptions made in developing the correlation equation (see Technical Discussion).

D. The main problem encountered in this analysis was the determination of bulk temperatures in the rotor assembly plenum regions; therefore, all estimated bulk temperatures were chosen to yield the most severe thermal gradients.

III. TECHNICAL DISCUSSION

A. Methods of Analysis

1. Complete mixing - In this method of solution, complete mixing of the gases is assumed to occur at the first stage rotor vane entrance and the gas bulk temperature between the vanes of the rotors and the second stage nozzle are given by:

$$T_{ob} = \frac{T_{on} + 0.07T_c}{1.07} \quad (1)$$

The gas recovery (adiabatic wall) temperature is then computed from:

$$T_r = T_s + R_t (T_{ob} - T_s) ; R_t = 0.88 \quad (2)$$

Heat transfer coefficients in the vane regions were computed from the following correlation:

$$h = 0.048 (\rho V)_b^{0.8} (D_h)^{-0.2} \left(\frac{T_b}{T_w} \right)^{0.5} \quad (3) \quad \text{Reference 4}$$

Heat transfer coefficients along the rotor disks were calculated from the following correlation:

$$N_{u_r} = \frac{h_c r}{k} = .0195 \left(\frac{u_r^2}{\nu} \right)^{.8} \quad (4) \quad \text{Reference 5}$$

The properties used in the above correlations are evaluated at the gas bulk temperature which is assumed to vary as shown in Figure 1. A hand iterative heat balance would involve the assumption of such variables as the turbine inlet manifold radiation temperature and the hot gas velocity in plenum regions, thus lessening the confidence level of the results.

2. No mixing - The Tribus-Klein correlation for nozzle-film cooling was derived using cylindrical straight-pipe flow theory. In applying this theory to turbine thermal analyses, the bearing coolant was treated as a discrete layer (analogous to injected fluid in Tribus-Klein studies) which clings to the rotor disk peripheries and the disk rim at the vane roots. The following equation developed by Tribus and Klein (Reference 2) was used in calculating recovery temperatures:

$$\frac{T_r - T_{oo}}{T_o - T_c} = 5.75 (Pr_g)^{2/3} (Re_c)^{0.2} \left(\frac{C_c}{C_g} \times \frac{V_g}{V_c} \frac{\rho_g}{\rho_c} \right) \left(\frac{\mu_c}{\mu_g} \right)^{0.2} \quad \text{Reference 2}$$

Since the above equation was developed for straight-pipe flow, the results yield a lower temperature than occurs in nozzle flow (see Reference 6).

Another factor that lowers the confidence level of results using the Tribus-Klein correlation is that the coolant (cold gas) layer thickness is not

known to the degree of accuracy required for optimum use of the above equation.

Table I is a summary of main drive gas interstage pressures and temperatures used in calculating heat transfer boundary conditions for both methods of analysis. A comparison of resulting gas recovery temperatures versus location for both methods of analysis is shown in Figure 1.

B. Assumptions

1. Complete Mixing

a. Hot and cold gases completely mixed at first stage rotor vane entrance.

b. Cold gas flow clings to periphery of first stage rotor disk until reaching rotor vane entrance.

c. All materials isotropic.

2. No-Mixing Method

a. Cold gas film exists as a discrete layer (no mixing with main turbine gases).

b. Temperature gradient of film does not change rapidly in downstream direction.

c. Correlation developed for straight-pipe flow.

d. All materials isotropic.

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TABLE I

INTERSTAGE CONDITIONS, MARK III MOD 3 TURBINE

$$T_{TT1} = 1350^{\circ}\text{R}, N = 27,500 \text{ rpm}$$

First Stage

Nozzle Inlet	T = 1350°R P = 700 psia	
Nozzle Exit	T = 1290°R P = 440 psia v = 3789 ft/sec c = 500 ft/sec	t = 1210°R p = 340 psia M = 0.75
Rotor Inlet	Assumed to be same as Nozzle Exit	
Rotor Exit	T = 1238°R P = 280 psia v = 4630 ft/sec c = 3460 ft/sec	t = 1175°R p = 226 psia M = 0.53

Second Stage

Nozzle Inlet	Assumed to be same as 1st stage Rotor Exit	
Nozzle Exit	T = 1190°R P = 195 psia v = 3140 ft/sec c = 4330 ft/sec	t = 1130°R p = 151 psia M = 0.675
Rotor Inlet	Assumed to be same as Nozzle Exit	
Rotor Exit	T = 1145°R P = 140 psia v = 4000 ft/sec c = 2890 ft/sec	t = 1100°R p = 117.5 psia M = 0.46

$$\dot{W}_t = \frac{P_{TT1} (0.475)}{\sqrt{T_{TT1}}} = \frac{700 (0.475)}{\sqrt{1350}} = 9.06 \text{ lb/sec}$$

7% $\dot{W}_t = 0.635 \text{ lb/sec}$, $\dot{W}_t$ is measured upstream of the turbine inlet

$$\dot{W}_t = 9.06 - 0.635 = 8.425 \text{ lb/sec}$$

$\dot{W}_t$ = total turbine weight flow

T = total temperature, t = static temperature, v = relative velocity
P = total pressure, p = static pressure, c = absolute velocity, M = Mach number

GAS RECOVERY TEMPERATURE vs LOCATION MARK III MOD 3 TURBINE

FIGURE 1

2/9/65

MARK III MOD III TURBINE ANALYSIS
 STEADY STATE TEMPERATURES
 $T_0 = 1350^\circ R$
 COMPLETE MIXING

NOTES

- 1 ALL TEMPS. $\sim^\circ$ RANKINE
- 2 SCALE 1/1

2/5/65

FIGURE 2

MARK III MOD III TURBINE ROTORS
STEADY STATE TEMPERATURES
 $T_0 = 1350^{\circ}\text{R}$

1 ST STAGE

NOTES:

1. TEMPS. $\sim ^{\circ}\text{R}$

2. SCALE: $\sim 3/1$

2ND STAGE

2/3/65

FIGURE 3

MARK III MOD III TURBINE STATORS
STEADY STATE TEMPERATURES
 $T_0 = 1350^{\circ}R$

1ST STAGE

NOTES:

1. TEMPERATURES $\sim^{\circ}R$
2. SCALE: 5/1

2ND STAGE

2/3/65
FIGURE 4

NO MIXING

2 SCALE: 1/1

2/5/65

FIGURE 5