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A PERFORMANCE/LOSS EVALUATION OF SSME HPFTP TURBINE

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ABSTRACT

An evaluation of component losses of the High Pressure Fuel Turbopump (HPFTP) in the Space Shuttle Main Engine (SSME) is performed using a mean-line prediciton method. This is accompanied with an extensive review of loss correlations in the literature. The present prediction uses an existing gas path velocity triangle, real LH_2 and H_20 gas properties and loss correlations selected from the literature. The significant losses incurred in the HPFTP turbine include profile loss, secondary loss and tip clearance loss. Results obtained from the present prediction are compared to those calculated from a quasi-three-dimensional numerical analysis. Except for the clearance loss, the present loss coefficients are in general higher than their counterparts from the quasi-three-dimensional analysis. The fact that the mean-line velocity data being unable to represent actual flowfields near the hub and the tip regions is largely responsible for the uncertainty involved in the present method. On the other hand, due mainly to the ad-hoc nature of the studies involved, the correlations currently available in the literature may not be suitable for accurate loss prediction of the particular rocket turbine in the SSME HPFTP. Further studies particularly in the areas of tip clearance loss, coolant loss, secondary loss and their interactions are desirable. Fundamental phenomena concerning flow unsteadiness in wake shedding and turbulence are also important.

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NOMENCLATURE

c	chord length
C_b	base pressure coefficient, see Eq. (7)
C_L	lift coefficient
h	mean blade height
k	tip clearance, see Eqs. (12)-(15)
L	blade surface length in axial direction
M	Mach number
p	pressure
r	ratio of specific heat of constant pressure to that of constant volume, see Eq. (3)
R_T	blade tip radius
R_m	blade mean radius
s	blade pitch
te	blade trailing edge thickness
U	circumferential velocity
V	mean velocity in axial direction
Y	pressure coefficient, see Eq. (1)
Z	Ainley-Mathieson loading parameter, see Eq. (10)

Greek Symbols

α	relative flow angle at blade row, see Fig. 1
β	blade angle, see Fig. 1
δ^*	displacement thickness
θ	momentum thickness
ξ	kinetic energy loss coefficient, see Eq. (2)

Subscript

CL	tip clearance
is	isentropic
m	mean
max	maximum
p	profile loss
s	secondary loss
t	total
te	trailing edge
1	inlet
2	exit

INTRODUCTION AND RESEARCH OBJECTIVES

The estimation of gas turbine performance has attracted extensive research efforts in the past thirty years. Knowledge gained in these efforts has resulted in vast improvements in turbine design technology and understanding of fundamental phenomena involved. Equally significant are the creation of many prediction methods for estimations of turbine performance from these studies. The performance prediction methods can generally divided into two major categories - the overall stage methods and loss component methods.

The overall stage methods deliberately ignore the effects of turbine bladings and aerodynamics. The performance prediction is based on testing data from a number of turbines with similar characteristics. Thus, they are largely ad-hoc in nature and their use is somewhat limited. Typically the turbine stage efficiency is expressed as a function of either flow parameter and loading factor (Smith, 1965) or sprouting velocity and blade angles (Glassman, 1972). These methods have been very viable in performance prediction for turbines designed prior to 1970. Since then, gas turbine designs have evolved considerably. The stage loading has increased while aspect ratios have decreased. In addition, work distribution, blade stacking and blade-profile optimization have become important design parameters. The overall stage methods have lost most of their importance.

The loss component prediction methods use a basic concept that the sum of a number of individual loss components gives the total loss of an entire turbine stage. As a contrast to overall stage methods, knowledge of turbine flow characteristics and blading details is important here. This method initially defines important influence parameters which account for aerodynamic and geometric effects in a stage. These are followed by separate evaluations of individual loss associated with each parameter. Although such a prediction approach appears to be more accurate and acceptable, almost all of the methods developed are based on simplified models which involve critical assumptions. For example, all predictions are based on the mean-line velocity triangle known a priori, it hence assumes that the entire transport process undergone by the working fluid is represented by the velocity characteristics at midspan. Moreover, even with the mean-line assumption, systematic variations of a large number of parameters and accurate evaluations of each individual effects are virtually impossible. In reality, these effects are interactive and fundamentally inseparable. The complexity of actual systems is often far beyond the model's limits. To accurately predict a turbine performance thus requires certain corrections or modifications to the models.

One of the major concerns in our nation's space propulsion program is to improve the performance of space shuttle main engine (SSME). Among many components in SSME, the two-stage, unshrouded, axial turbine in the high-pressure fuel turbopump (HPFTP) has attracted great attention in the past. Nevertheless, continuing research efforts are greatly needed to understand the physical phenomena and to improve the turbine performance. NASA Marshall Space Flight Center (MSFC) is currently constructing a highly instrumented turbine test article (TTA) which is capable of facilitating extensive measurements on loss components and stage efficiency. One primary objective is to develop a viable means for accurate prediction of HPFTP turbine efficiency.

As a prelude to the up-coming testing with TTA, it is necessary to review and evaluate different models and correlations for turbine losses/efficiency currently available in the literature. Knowledge gained from such a study will provide a valuable baseline information to clarify future research directions with TTA. The work described here is primarily for this purpose. Here, the HPFTP turbine efficiency is assessed using the mean-line loss component method, as mentioned earlier. The prediction chooses correlations from previous studies and is considered to be suitable for HPFTP turbine. However, these correlations are generally obtained from experiments with conditions substantially different from those in HPFTP turbine. The major difference includes blading geometry, loading conditions and fluid properties, and, therefore certain levels of deviation and uncertainty are expected.

Results obtained from the presently chosen correlations will be compared with corresponding results calculated from the NASA developed, MERIDL-TSONIC-BLAYER (MTB) method. The MERIDL and TSONIC are quasi three-dimensional codes for inviscid flowfields (Katsanis, 1969; Katsanis and McNally, 1977), and BLAYER is an integral solver for the viscous boundary layer (McNally, 1970). The MTB method is widely used for present day turbine design, largely because it is much less costly in computing time and technically simpler than a full three-dimensional, Navier-Stokes simulation. However, its accuracy remains uncertain, particularly for the predictions of flowfields. Povinelli (1985) has suggested that improvement in velocity prediction near blade trailing edge and the suction surface is required.

By definition, the total loss in a turbine stage is expressed in terms of total loss of dynamic pressure evaluated at the stage exit, i.e.

$$Y = (p_{t1} - p_{t2}) / (p_{t2} - p_2) \quad (1),$$

where Y is the total pressure loss coefficient, and p_{t1}, p_{t2} and p_2 represent the total pressures at stage inlet and exit and static pressure at exit, respectively. The loss component method assumes that the total pressure loss is the sum of profile, secondary and tip leakage losses, i.e. $Y = Y_P + Y_S + Y_{CL}$. For modern gas turbine involved significant blade cooling, e.g. film cooling, coolant loss is often considered. Another loss measure is defined as the loss of kinetic energy as compared to the isentropic condition, i.e.

$$\xi = (V_{2is}^2 - V_2^2) / V_2^2 \quad (2)$$

where ξ is the kinetic energy loss coefficient, and V_{2is} and V_2 are the velocities at exit under isentropic and actual situations, respectively. The major difference between these two coefficients lies their response to Mach number (M) variations. ξ is insensitive to Mach number variation; while Y increases with the magnitude of Mach number. The relation between these two loss coefficients can be expressed as,

$$Y / \xi = [1 + (r - 1) \cdot M^2/2]^{r/(r-1)} \quad (3)$$

where r is the ratio of specific heat at constant pressure to that at constant volume. It is clear that Y and ξ are practically equal for the low-Mach number flow, and it is generally the case for HPFTP turbine. The maximum value of Mach number in HPFTP turbine at full power level is approximately 0.4.

The following describes the loss components and reviews the corresponding correlations reported in the literature. To aid the description, The blade terminology is shown in Fig. 1.

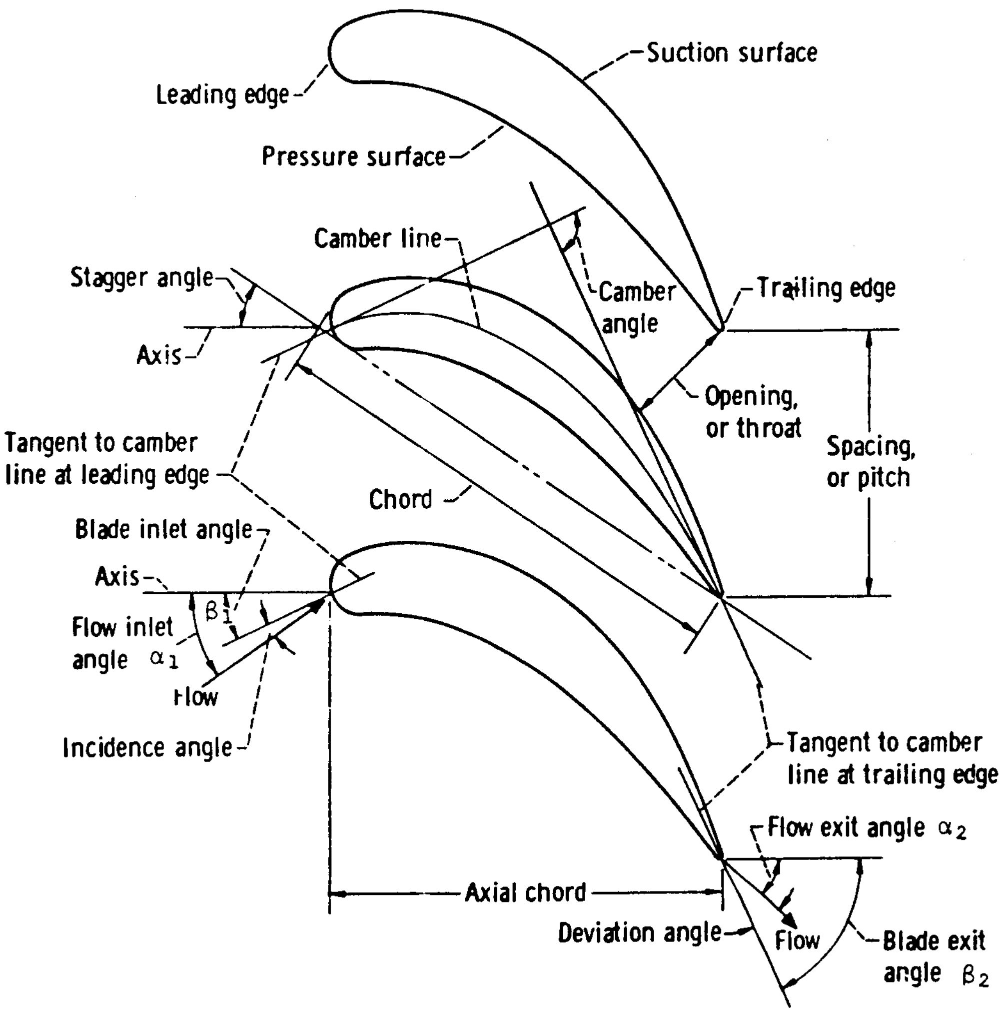


FIG. 1 BLADE TERMINOLOGY

DESCRIPTIONS OF LOSS COMPONENTS

1. PROFILE LOSS

The profile loss is composed of two parts. The first is the friction loss associated with boundary layer development on the blade surfaces, so-called the "basic profile loss". The second part represents the aerodynamic loss due to wake mixing near the blade trailing edge when the blade has a non-zero thickness. This leads to the name - "trailing edge loss". The phenomenon of this wake mixing is caused by sudden expansion of flow area and non-uniform pressure distribution in the spanwise (radial) direction at a stage exit.

1.1 Basic Profile Loss

Fundamentally, the basic profile loss is proportional to the magnitude of momentum thickness at the stage exit. Hence, it depends on stage geometry, i.e. blade shape and pitch-to-chord ratio, and Reynolds number. Since the turbine flow is largely turbulent, it is also affected by blade surface roughness. To a more detailed extent, curvature of blade surfaces influences the boundary layer developments on both pressure and suction sides of the blade. This is particularly important near the blade leading edge where a transition boundary layer exists. However, the curvature effects on the concave and convex sides of the blade are considered to offset each other, so the flat-plate approximation is a reasonable assumption.

Results concerning the basic profile loss prior to 1970 has been extensively reviewed by Denton (1973). In this review, Denton compared seven different correlations which, in addition to his own, include those well-known ones from studies by Ainley and Mathieson (1951), Stewart (1955), Traupel (1966), Balje and Binsley (1968), and Craig and Cox (1970). Among them three frequently mentioned correlations are listed as follows.

Ainley and Mathieson (1951)

$$Y_p = [\, Y_{\beta_1=0} + (\beta_1/\alpha_2)^2 \cdot (Y_{\beta_1=\alpha_2} - Y_{\beta_1=0})] \cdot (t_{max}/0.2\,c)^{\beta_1/\alpha_2} \tag{4}$$

Stewart (1955)

$$Y_p = 1 - [\, \sin^2\alpha_2 (1 - \delta^* - \delta^*_{te} - \theta^*)^2 / (1 - \delta^* - \delta^*_{te})^2 + \cos^2\alpha_2 (1 - \delta^* - \delta^*_{te})^2]$$
$$\{1 + 2\cos^2\alpha_2 [\, (1 - \delta^* - \delta^*_{te})^2 - (1 - \delta^* - \delta^*_{te} - \theta^*)]\} \tag{5}$$

Traupel (1966)

$$\xi_p / (1 - \xi_p) = 0.006\,(L/s) \cdot (V_m/V_2)^3 \cdot [\,1 + 3/4\,(U/V_m) \cdot (s/L)]^2 / \cos\alpha_2 \tag{6}$$

The comparison made by Denton (1973) shows surprisingly large deviations among these correlations. This is mainly caused by the ad-hoc nature of each individual test and aerodynamic improvement toward modern blade design. The latter is evidenced by the fact that correlations proposed in earlier days tend to over-predict the loss for turbines developed later. Kacker and Okapuu (1982) suggest that the losses predicted by the correlation of Ainley and Mathieson (1951) should be multiplied by a factor of 2/3 to account for the progress in blade design. In addition, all the studies aforementioned use stationary cascade models which neglect the rotational effects in actual turbine conditions. Dejc and Trojanovkij (1973) propose a correction factor to transform data with stationary cascades for use in actual turbine stages.

1.2 Trailing Edge Loss

According to conservation equations for continuity, momentum and energy, the trailing edge loss is the sum of two terms, i.e.

$$Y_{te} = [te / (1 - te)]^2 \cdot \cos^2 \alpha_2 + C_b \cdot te \tag{7}$$

The first term represents the loss caused by sudden expansion of flow area at the stage exit, and the second term is the additional mixing loss induced by the difference of pressures at blade base and average pressure across the exit span. C_b is the so-called "base pressure coefficient" which, in fact, bears a complex nature and varies strongly with blade geometry and flow conditions. The flow parameters affecting the value of C_b include the ratio of momentum thickness to blade trailing edge thickness and the Mach number of exit flows on both sides of the blade. Sieverding (1980) reported C_b data for flow exit-angle, $\alpha_2 = 60^o$ to 70^o and te/c = 0.06 to 0.15. His results confirm the strong dependency of C_b on parameters mentioned above.

Kacker and Okapuu (1982) observed a distinct difference of trailing edge loss between stators and impulse blades. They claimed that the difference in the thickness of boundary layer is the major cause of this phenomenon. Impulse blades, with their thick boundary layer, have lower (less negative) base pressure coefficients and thus have lower trailing edge loss. The correlation they proposed is

$$\xi_{te} = \xi_{te\,(\beta_1=0)} + |\beta_1 / \alpha_2| \cdot (\beta_1 / \alpha_2) \cdot [\xi_{te\,(\beta_1=\alpha_2)} - \xi_{te\,(\beta_1=0)}] \tag{8}$$

An important aspect pertaining to trailing edge loss, that is missing in the past research, is the effects of flow unsteadiness. The wake mixing inherits the flow characteristics of vortex shedding in various frequencies and blade interactions. These unsteady flow effects are generally non-linear with respect to time, and thereby have accumulative influence on the transports of momentum and energy. From the standpoint of dimensional analysis, the wake mixing and trailing edge loss are functions of many parameters, in particular Strouhal number and Reynolds number. Further studies in this regard should be emphasized.

2. SECONDARY LOSS

The secondary loss is due to the secondary flow in the blade passage. To gain a better understanding of the passage flow, it is usual to consider an ideal "primary" flow which may, for example, be two-dimensional potential flow. The difference between this primary flow and actual flow is then termed secondary flow. There are two mechanisms which have a major influence on the generation of secondary flows in turbine passages. The first mechanism is the effects of rotation and curvature, which cause the development of vortices in the streamwise direction. This is the similar mechanism of three-dimensional flow patterns existing in curved ducts, pipe bends and rotating channels. The vortices formed under this condition is the so-called "passage vortices" in turbine cascades. The second mechanism is the roll-up of endwall boundary layer in front of a blade. The flow pattern, due to its particular shape, is called "horseshoe vortex," similar to its classical sense of cross flow over a wall-attached cylinder. The significance of horseshoe vortex in a turbine passage flow has been recognized only very recently. Fig. 2 displays an artistic sketch of passage secondary flow pattern originally presented by Klein (1966) and Langston (1980).

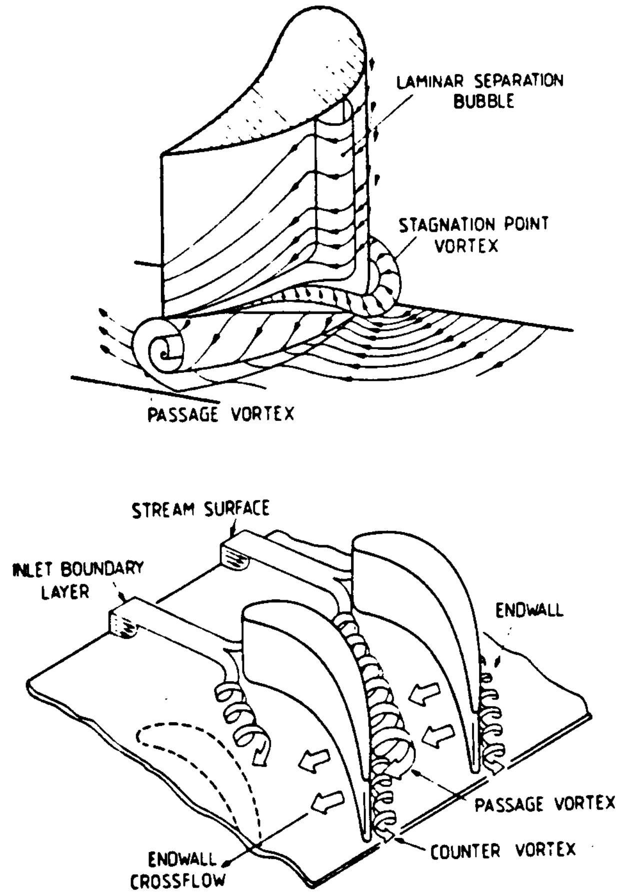


Fig. 2 SECONDARY FLOW MODEL BY KLEIN (1966) AND LANGSTON (1980)

The secondary loss, often amounting to nearly half of total stage loss, has been the subject of extensive research in the past. However, despite the large number of loss correlations available in the literature, it remains difficult and is risky to extrapolate the existing results for a generalized loss prediction. This is mainly due to the ad-hoc nature in passage geometry, blade aspect ratio, and flow condition for all the studies reported.

Secondary loss correlations prior to 1970 has been critically reviewed by Dunham (1970). His review lists 14 different correlations for incompressible flow. Secondary loss in these correlations is expressed by the total pressure loss coefficient defined as

Y_S = stagnation pressure loss / exit dynamic pressure.

In addition, all correlations are composed of a loading term, representing the effect of cascade loading or deflection and a length-scale ratio term. A typical form under this concept is expressed as

$$Y_s = (c/h) \cdot (\cos \alpha_2/\cos \beta_1) \cdot f(\delta_1^*/c) \cdot Z \tag{9}$$

where , Z is the loading parameter given by Ainley and Mathieson (1951),

$$Z = (C_L/(s/c))^2 \cdot \cos^2 \alpha_2/\cos^2 \alpha_m \tag{10}$$

where

$$\alpha_m = \tan^{-1} [(\tan \alpha_1 + \tan \alpha_2)/2] \tag{10a}$$

$f(\delta_1^*/c)$ represents the endwall boundary layer effect at the passage inlet.

Correlations presented later than 1970 still follows the same formula as Eq. (9); but with a more explicit form of the length-scale ratio term, i.e.

$$f(\delta_1^*/c) = C_1 + C_2 \cdot (\delta_1^*/c)^n \tag{11}$$

where C_1, C_2 and n are constants, and their values vary with different studies. As an example,

$C_1 = 0.0055$, $C_2 = 0.078$, $n = 0.5$ (Dunham and Came, 1970),
$C_1 = 0.011$, $C_2 = 0.294$, $n = 1.0$ (Morris and Hoare, 1975)
$C_1 = 0.034$, $C_2 = 0$ $n = 0$ (Kacker and Okapuu, 1982)

In Eq. (11), the first term accounts for loss due to boundary layer growth on the endwall. The second term, a function of boundary layer thickness, represents the loss resulting from the horseshoe vortex formation near a blade leading edge.

3. TIP CLEARANCE LOSS

Aerodynamic loss due to flow leakage through the narrow gap between blade tip and adjacent outer casing represents a major efficiency penalty in a turbine rotor. The tip leakage flow is induced by the pressure difference between pressure and suction sides of a blade. The flow pattern is further complicated by the effect of relative wall movement with respect to the blade tip, which generates an additional secondary vortex, so-called the scraping vortex. Fig. 3 shows a schematic view of tip leakage flow. In modern turbine design, the tip leakage flow is controlled by maintaining close tolerance on tip clearance and/or geometric treatment of blade tip. Instead of plain tip, the latter commonly uses geometries such as winglet, squealer or groove tip.

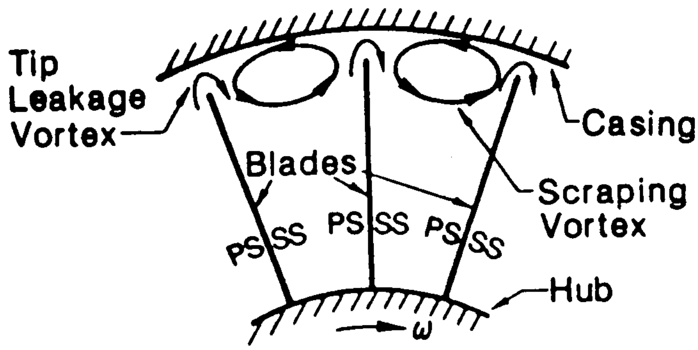


FIG. 3 TIP LEAKAGE FLOW

The fundamental mechanism of tip leakage flow that results in performance loss is not understood in detail. This, in part, is due to strong flow interaction and its effects on transport processes. Hence, for an unshrouded rotor stage, a clear demarcation between secondary and tip clearance losses is fundamentally impossible. This is particularly true for HPFTP turbine where the blade aspect ratios are small, nearly approximately unity. Under this condition, tip clearance loss is defined as the difference between total losses with and without clearance, i.e. $Y_{CL} / Y_{CL=0}$. Peacock (1982) has assessed the mechanism of tip clearance loss by considering the following four factors.

(1). Pressure difference between pressure and suction surfaces of a blade - primarily an inviscid effect. This is the same mechanism leads to the tip vortices on an aircraft wing. The tip vortices progressively induces loss in lift of the wing section towards the wing tip.

(2). Presence of boundary layer on turbine casing - a viscous effect. The boundary layer itself affects the tip region performance by inducing aforementioned scraping vortices which, in turn, results in a three-dimensional separation contributing to the loss of efficiency.

(3). Relative movement between blade and boundary layer on the casing - primarily a viscous effect. For a turbine blade, this movement is in the opposite direction of tip leakage flow, thus it is in opposition to the mechanism stated in (1).

(4). Size of clearance. For blades with large tip clearances, this effect overwhelms the combined mechanism of (1) to (3). This is significantly detrimental to the turbine efficiency due to blade unloading. For small clearance, the pressure force is balanced by the viscous force in the leakage flow, and the effects of (1) to (3) prevail.

For unshrouded, plain-tip, correlations of tip clearance loss prior to 1950 are expressed as linear functions of reduced flow area or mass flow rate in the presence of tip clearance. These correlations, largely based on steam turbine data, have been reviewed by Ainley and Mathieson in 1951. About the same time, Ainley (1951) proposes a correlation involving both clearance size and blade loading, i.e.

$$Y_{CL}/Y_{CL=0} = 0.5 \cdot (k/h) \cdot Z \tag{12}$$

where k and h are clearance size and blade height, respectively, and Z is the Ainley Loading factor. Hong (1962) later using an annular flow model reported a similar correlation, i.e.

$$Y_{CL}/Y_{CL=0} = 1 - 2.2 \cdot (k/h) \tag{13}$$

Dunham and Came (1970) modified Ainley's correlation (Eq. (12)) and suggested that conventional linear dependence of loss on clearance size should be replaced by a power law. Their correlation for a plain tip is

$$Y_{CL}/Y_{CL=0} = 0.47 \cdot (k/h)^{0.78} \cdot Z \tag{14}$$

More recently, Kacker and Okapuu (1982) have claimed that correlations developed earlier overpredicts the tip clearance loss of recently developed turbines. They proposed instead

$$Y_{CL}/Y_{CL=0} = 0.93 \cdot [k/(h \cos \alpha_2)] \cdot (R_T/R_m) \tag{15}$$

where R_T and R_m represent the tip radius and blade mean radius.

Note that all correlations up to date, including Eqs. (12)-(15), are derived based on simplified, non-rotating cascade models. Hence, they fail to include some essential effects, i.e. rotational force, scraping vortices, and relative wall-movement.

4. OTHER LOSSES

Depending on details of specific turbine conditions, there are other loss components which are significant and need to be addressed. Each of these (other) losses directly relates to one or more of those major losses as discussed in previous sections. In the literature, losses belonging to this category include the effects of blade thickness (Roelke and Hass, 1983), flow angles of incidence (Flagg, 1967), Mach number (Ainley and Mathieson, 1951; Kacker and Okapuu, 1982), Reynolds number (Traupel, 1966, 1977), and coolant flow (Hartsel, 1972; Tabakoff and Hamed, 1975; Ito, Eckert and Goldstein, 1980).

The influence of Reynolds number on the turbine performance is two-fold. The first effect lies on the development of boundary layer attached to both surfaces of the blade. Near-wall velocity profiles and momentum thickness is determined by the value of Reynolds number. The second effect relates to the blade surface roughness or finish. A turbulent boundary layer, sharply different from its laminar counterpart, is very sensitive to the surface condition. Thus, the degree of surface roughness along with Reynolds number determines the friction characteristics on a blade surface. This, in turn, will affect the basic profile loss to a certain extent.

The Mach number effect lies mainly in shock loss near a blade trailing edge. The effect varies strongly with blade thickness and passage shape. The shock loss is particularly important near the trailing section of a blade suction surface. Thus, Mach number is an additional parameter to be considered for the trailing edge loss. A further complication pertaining to trailing edge loss is due to the coolant ejection (Haas and Kofskey, 1977). The ejection of blade internal coolant often exists in modern gas turbines; however, all turbine blades in SSME do not have this feature.

The coolant loss in gas turbine, in a conventional sense, is referred to the turbine performance loss due to coolant injection of film cooling. Film cooling is one of the most effective means for blade cooling in modern gas turbine engines. In this case, coolant (compressed air in gas turbine) is directly injected into the hot mainstream, through either slots or discrete holes located on blade surfaces. As a result of momentum interaction between the injection and mainstream, a coolant film is formed covering the blade surface to be protected from hot-gas exposure. The direct impact of coolant injection is to thicken the boundary layer, so the profile loss increases.

In SSME HPFTP turbine, the coolant loss is fundamentally different from that of gas turbines as described above. First of all, current HPFTP design does not use film cooling or any other enhancement cooling to protect the blades. The coolant here is referred to the liquid hydrogen (LH_2), a cryogenic fluid, which is mainly used for cooling of rotating disks. Due to rotation, LH_2 gains radial component and flows outward. The coolant can bleed into the turbine passage through the gap (approximately 0.01") between rotor and stator. However, it is unclear that the flow passing through this gap is whether inward or outward, due mainly to the complex pressure distributions in HPFTP. The amount of flow rates are also uncertain. According to the present knowledge largely gained from numerical simulations, the combusted gas flows radially inward in the first stage (ahead of both stator and rotor) and LH_2 bleeds outward in the second stage. In either case, the predominant influence of this leakage flow lies in the region near a blade leading edge as it affects the structure of oncoming boundary layer before separation. The nature of leading-edge separation and horseshoe vortices existing in the blade-endwall section can vary significantly. As a result, the effects on the secondary loss is expected. So far, studies relating to this unique aspect of coolant loss in HPFTP have never been undertaken.

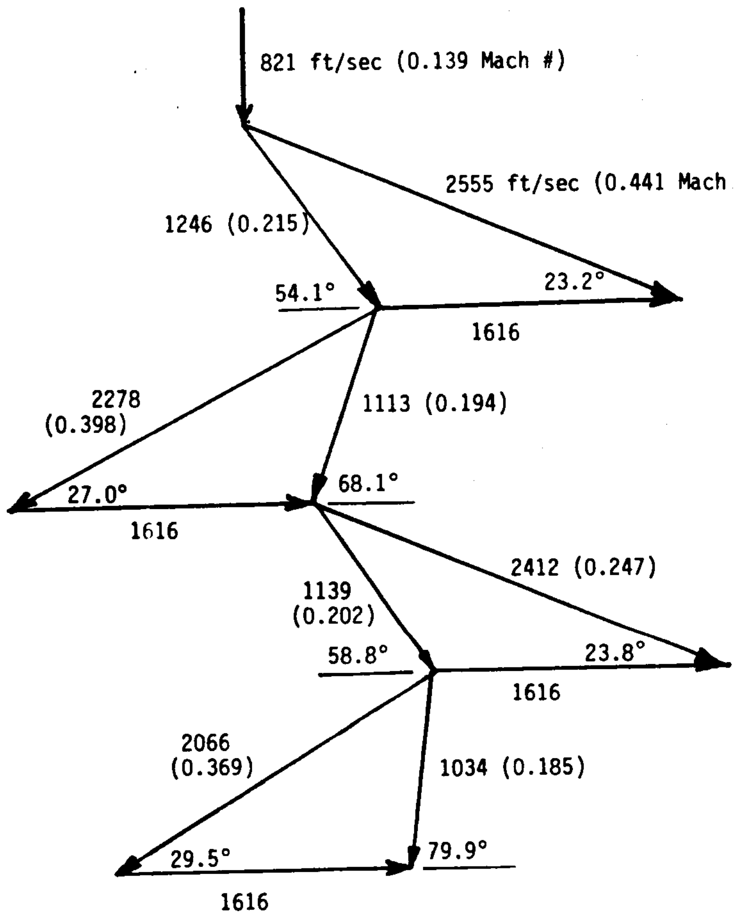
MEAN-LINE LOSS ESTIMATION

The present mean-line prediction uses the velocity triangle shown in Fig. 4. This full power-level velocity diagram was provided by Rocketdyne engine-balance calculation on May 21, 1987. Also listed in the figure is information pertaining to the specific turbine operation, e.g. pressure, temperature, mass flow rate, and rotating speed. The height of blade (h) is 0.884 and 0.984 inch for the first and second stage, respectively. The mean axial width is about 20% larger than the height for the first stage nozzle; nevertheless, this feature is completely reversed for the second stage rotor. The first stage rotor and second stage nozzle have a width-to-height ratio of nearly unity. The passage pitch-to-height ratio is approximately 0.88, 0.58, 0.86 and 0.55 for the first nozzle, first rotor, second nozzle and second rotor, respectively. The tip clearance used here is 0.019 and 0.016 inch respectively for the first and second stage rotor, and each of these amounts to 2 and 1.5% of the corresponding blade height.

To calculate the transport properties of working fluid in HPFTP, the present study uses correlations of real gas properties reported recently by Harloff (1987). These correlations include all important fluid properties in polynomial forms of temperature with a fixed gas mixture ratio of 86.87% H_2 and 13.13% H_2O. Using these polynomials removes the limitation of using air-equivalent conditions. Moreover, properties presented in polynomial forms are required as an input for loss predictions using the MERIDL-TSONIC-BLAYER (MTB) computer code. In the same study, Harloff has performed such a computation, and his results will be compared with the present ones.

Table 1 shows the loss-component breakdown and efficiency for each individual stage and entire turbine. Also shown in the parenthesis is the corresponding results from Harloff's (1987) MTB numerical computation. Mainly because of limited flow information, particularly the flow angles, the present mean-line prediction neglects the losses contributed by the trailing edge mixing, flow incidence, and effects of Reynolds and Mach number. Here, the estimation of basic profile loss uses the Stewart (1955) correlation, Eq. (5), since it bears many important fundamental features and is also used in the Harloff's simulation. The correlation used for secondary loss is the one developed by Kacker and Okapuu (1982), Eq. (8), which is considered to be appropriate for turbines designed later than 1970. The tip clearance loss uses the correlation by Dunham and Came (1970), Eq. (14), which involves power-law relationship, instead of the conventional linear one, with the length scale ratio term.

A comparison between the results from two studies shows that, except for the tip clearance loss, the present mean-line method gives higher values of loss prediction. In both studies, the greatest loss contribution comes from the profile loss in the stator, and tip clearance loss in the rotor. The tip clearance loss accounts for nearly one-half of the total rotor loss. The differences in profile and secondary losses are substantially large, often exceeding 100%. The entire, 2-stage, turbine efficiency is estimated to be 76.1%, compared to 81.6% reported by Harloff. The Rocketdyne design efficiency is cited as 79.1%. The present mean-line method appears to be the most conservative one, and it may overpredict the losses. Note that the velocity diagrams among all three studies are slightly different. However, the effects on the loss prediction results are considered to be very minor.



	Temperature R	Pressure PSIA
Inlet Total	1892	5525
1st Nozzle Exit Static	1829	4812
1st Rotor Exit Static	1794	4393
2nd Nozzle Exit Static	1749	3939
2nd Rotor Exit Static	1720	3648

Rotating Speed (RPM)	36353
Power (HP)	72903
Flow Rate (LB/SEC)	164.9

FIG. 4. VELOCITY TRIANGLE

	1st Stator	1st Rotor	2nd Stator	2nd Rotor
1. Profile Losses Strwart (1965)	0.022 (0.014)	0.025 (0.015)	0.022 (0.014)	0.021 (0.016)
2. Secondary Losses Kacker & Okapuu (1982)	0.017 (0.012)	0.026 (0.010)	0.022 (0.011)	0.029 (0.011)
3. Tip Clearance Losses Dunham and Came (1970)		0.047 (0.063)		0.033 (0.047)
4. Incidence Losses	(0.001)	(0.006)	(0.001)	(0.014)
5. Stage Efficiency	0.867 (0.902)		0.878 (0.905)	
6. Turbine Efficiency		0.761 (0.816)		

Rocketdyne Design Efficiencey: 0.791

TABLE 1. LOSS BREAKDOWN

CONCLUSIONS AND RECOMMENDATIONS

An extensive literature review on the component loss prediction for unshrouded axial turbine has been performed in this study. All the correlations presently available are generally ad-hoc in nature and may be inappropriate for accurate prediction of turbine performance in SSME. Future testing with Turbine Testing Article at Marshall Space Flight Center should be directed towards collecting sufficient data base as well as gaining fundamental understanding of transport phenomena in turbine flow. The data base is required for validation of computer codes, and the fundamental information is needed for theoretical model developments. The models developed must be capable of providing physical insight into both momentum and energy transport, and also improve the performance predictions for other types of turbines. The following recommends future research directions which are important and need to be studied in greater depth.

1. Effects of Reynolds number. The primary concern lies on its influence on the boundary layer structure with various degrees of blade surface finish. This has direct influence on the basic profile loss. The combined and/or interactive effects between the surface roughness and curvature are also of interest.

2. Loss due to flow unsteadiness and turbulence. The wake shedding associated with periodic blade interaction is recently known to have significant influence on the time-averaged blade surface heat transfer. This has been a major issue in turbine research community for the past two or three years. The analogy between the energy transfer and momentum transfer implies that such a flow unsteadiness can induce an excessive loss in turbine aerodynamic performance. A viable prediction method in the future must incorporate this aspect.

3. Tip Clearance Loss. Based on the fact that the tip clearance loss amounts to nearly 50% of total rotor loss, correlations with much higher accuracy are desirable. The correlations currently available are fundamentally inaccurate, as they fail to include some predominant features in the system, e.g. relative wall movement, scraping vortices, and unloading near the blade trailing edge. Study recommended requires detailed pressure and flow measurements near the tip region. Ultimate research results should include recommendations of optimal tip geometries for future SSME turbines. In addition, the interaction between the tip clearance loss and secondary loss also needs to be emphasized, particularly for the present HPFTP turbine having low aspect ratios.

4. Coolant Loss. Investigation in this aspect must be directed to the special features of present HPFTP design, as discussed earlier. A rough estimation of the coolant velocity through the gap between the stator and rotor disks is approximately 150 ft/sec (nearly 5 to 10% of the passage mean flow velocity), that is sufficiently strong to affect the flow near the blade endwall region. Despite its thermal effects, it is certain that the secondary loss will be affected to a great extent. Due mainly to the complex nature involved, it would be desirable to initiate the study by conducting experiments in a non-rotating environment. Having established the stationary model, the rotating effects can then be enforced.

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