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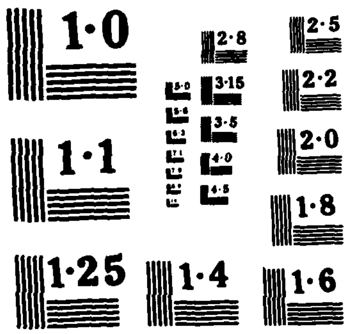

NATIONAL BUREAU OF STANDARDS
MICROCOPY RESOLUTION TEST CHART

ANALYSIS OF THE EFFECTS OF TRANSIENT HEAT TRANSFER ON AXIAL FLOW COMPRESSOR BLADE BOUNDARY LAYERS

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

Charles Wade Wood

June 1985

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ABSTRACT

This ~~investigation~~ determines the magnitude of heat transfer in the high-pressure compressor of a turbofan engine during a "Bodie" throttle transient and estimates the effect of transient heat transfer on compressor blade boundary layer growth. Total stored thermal energy available for release is determined considering compressor blades and roots only. Thermal energy released during a throttle transient is determined and allocated to individual compressor stages and used to estimate blade heat flux. The average heat transfer coefficient at the maximum heat transfer rate is also calculated.

A simple boundary layer analysis is performed assuming zero pressure gradient, compressible turbulent flow over a flat plate. Under free-stream conditions similar to those in the tenth stage of the compressor, this analysis shows increased boundary layer displacement thickness with heat transfer. In zero pressure gradient, the displacement thickness change with heat transfer is small, as is the change in flow deviation angle at the trailing edge of the compressor blade. Heat transfer coefficients are also calculated in the boundary layer analysis and agree with results from the experimental heat transfer allocation procedure to within 30 percent. Results of boundary layer analysis indicate assumptions

made in heat transfer allocation procedure are reasonable and that simple boundary layer analysis with the absence of modifying pressure gradients provides a good first estimate of heat transfer effects on blade boundary layers.

TABLE OF CONTENTS

LIST OF FIGURES

NOMENCLATURE

a	Speed of sound, ft/sec
A	High-pressure compressor inlet area, ft^2
C_f	Local friction coefficient
$\bar{C}_f$	Total friction coefficient
C_h	Stanton number
C_p	Coefficient of specific heat, Btu/lbm-^oR
$\bar{h}$	Average heat transfer coefficient, Btu/ft^2-s-^oR
h_t	Total enthalpy, Btu/lbm
H	Boundary layer shape factor
HP	High pressure
i	Blade incidence angle, deg
LP	Low pressure
m	Mass, lbm
$\dot{m}$	Mass flow rate, lbm/sec
M	Mach number
Pr	Prandtl number
P_s	Static pressure, psf
P_t	Total pressure, psf
Q	Stored thermal energy, Btu
$\dot{Q}$	Heat transfer rate, Btu/s
$\ddot{Q}$	Heat transfer rate per unit area, Btu/ft^2-s
R	Gas constant, Btu/lbm-^oR
Re_x	Reynolds number based on x
T	Temperature, ^oR

T_s	Static temperature, ^oR
T_t	Total temperature, ^oR
u	Local velocity, ft/sec
u_τ	Friction velocity, ft/sec
V	Velocity, ft/sec
$\dot{W}$	Work transfer rate, Btu/s
x	Distance measured along surface, ft
y	Distance measured normal to surface, ft

Greek Letters

α_2	Cascade exit angle, deg
γ	Ratio of specific heats
δ	Boundary layer thickness, ft
δ^*	Boundary layer displacement thickness, ft
Δ	Change in value of parameter
ε^*	Cascade design deflection angle, deg
η_c	Compressor efficiency
θ	Boundary layer momentum thickness, ft
ρ	Density, lbm/ft^3
σ	Defined by Eq. (18)

Subscripts

a	Adiabatic
ht	Heat transfer
idle	Idle throttle setting
max	Maximum throttle setting

w	Conditions at wall
2.5	High-pressure compressor inlet station
3	High-pressure compressor exit station
∞	Free-stream condition

Superscripts

*	Design value
$\overline{}$	Mean value of parameter

response, while the revised finned model gave the most accurate platform temperature response. Maccallum concluded that an eight-element finite difference model or a simple finned model would provide adequate blade temperature response when heat transfer coefficients are known.

Maccallum and Grant [7] developed a method for predicting the effects of heat transfer on a multi-stage axial flow compressor. The method is based on two-dimensional mean blade height calculations and estimates the effect of heat transfer on the compressor by comparing the predicted adiabatic and non-adiabatic characteristics of the compressor. This comparison gives a first estimate of the effect of heat transfer on the entire compressor.

To estimate the effects of heat transfer on a compressor, Maccallum and Grant [7] first developed a prediction method for adiabatic flow in the compressor. The method used row-by-row calculations through the compressor based on two-dimensional cascade data for the stages. While not exact, this method agreed reasonably with experimental data and realistically represented the sources of stall and surge in the compressor.

With an adiabatic prediction method established, Maccallum and Grant [7] estimated the effects of heat transfer on the airflow through a compressor. Air density is reduced as a result of heat transfer from the compressor metal parts to the airflow and has two effects on the

6 seconds, a number in the same range as previously found by Maccallum [3].

Blade boundary layer conditions were not considered in this analysis, and Elder [5] has suggested that a more accurate estimate of cycle time could be obtained if boundary layer conditions were included in this analysis. Elder also concluded that heat soakage effects on airstream temperature and flow conditions were small enough to be neglected in the anslysis of compressor performance during transients.

Prediction of heat transfer and temperature effects in a compressor depends on the correct choice of heat transfer coefficients for the components and selection of a model which accurately represents the action of the blades and support platforms in the compressor. Maccallum [6] developed and compared four heat transfer models ranging from an unfinned model and a simple finned model to a revised finned model and a finite difference model. All models gave adequate predictions of blade and platform temperatures during a transient, with the finite difference model assumed to be the most accurate.

Maccallum [6] compared the predictions of all four models for a second-stage LP compressor blade, an eleventh-stage HP compressor blade, and a second-stage HP turbine blade during an acceleration transient. All three finned models gave similar predictions for the blade temperature above the free-stream air temperature the turbulent boundary layer separation point moves upstream by a small percentage of the blade chord. Grant also concluded that thermal transients change the aerodynamic behavior of compressor blades. While the effect on an individual stage or row of blades is small, the potential exists that the disturbance could propagate through the HP compressor and be amplified in subsequent blade rows.

Elder [5] investigated the changes in axial flow compressor performance resulting from heat soakage into the metal parts of the compressor. The investigation attempted to determine the effect of heat soakage on air-stream temperature and instantaneous flow conditions and to estimate the delay time required prior to accelerating following a rapid deceleration of the engine. Elder considered the delay time to be a function of the time required for the blades to approach a new equilibrium temperature after a change in the airstream temperature.

In developing a "safe" cycle or delay time for acceleration following the rapid deceleration, Elder [5] concluded that allowing the blades to cool to 90 percent of the new airstream absolute temperature would cause flow conditions during the acceleration to be no worse than if there had been no deceleration at all. This cycle time resulted in a typical time constant for the blades of

Maccallum [3] then combined the changes in the running line with the shifts in the surge line to estimate the total effect on the compressor. The combined effects on the engine used for the investigation were a 35-percent reduction in surge margin at the end of the altitude deceleration and a 32-percent reduction following the sea-level deceleration. These changes in surge margins are not considered important during the deceleration itself since the compressor is moving away from surge, but the significant reduction in surge margin could become a factor if an acceleration were attempted immediately following the deceleration.

As a compressor slows down during the deceleration portion of a rapid deceleration-acceleration cycle, the temperatures of the compressor metal parts will be higher than that of the incoming air. Grant [4] suggested that local heat transfer may disturb the aerodynamics of the boundary layer air and be the cause for the apparent stalls during reacceleration.

Grant [4] considered the effect of heat transfer on the blade suction surface boundary layer to be the area of primary interest, and experimented with a convex plate in a low-speed wind tunnel in an attempt to measure the turbulent separation point and estimate the effect of surface heating on separation point movement. From these experiments, Grant determined that for wall temperatures

This investigation also studied the effect of heat transfer on the working line of the engine. Maccallum [3] defined surge margin as the difference between the pressure ratio at the surge line and the pressure ratio at the steady running line for any given mass flow rate. The bulk heat transfer during a transient has two effects which alter the running line of the engine. These effects are non-adiabatic compressions and expansions and alteration of nozzle guide vane effective areas.

Heat transfer during the transients discussed earlier also resulted in non-adiabatic compressions and expansions through the engine. In addition, the heat transfer caused changes in the boundary layer displacement thicknesses of LP and HP turbine nozzle guide vanes which altered the effective discharge area of the guide vanes. The effect of heat transfer in both LP and HP compressors and LP and HP turbines on the steady running pressure ratio of the LP and HP compressors was estimated, as was the effect of effective discharge area changes. The net result during the acceleration was a decrease in the steady running pressure ratio, while the result following the deceleration was an increase in the pressure ratio. These results are shown by the dotted running lines in Figures 1 and 2.

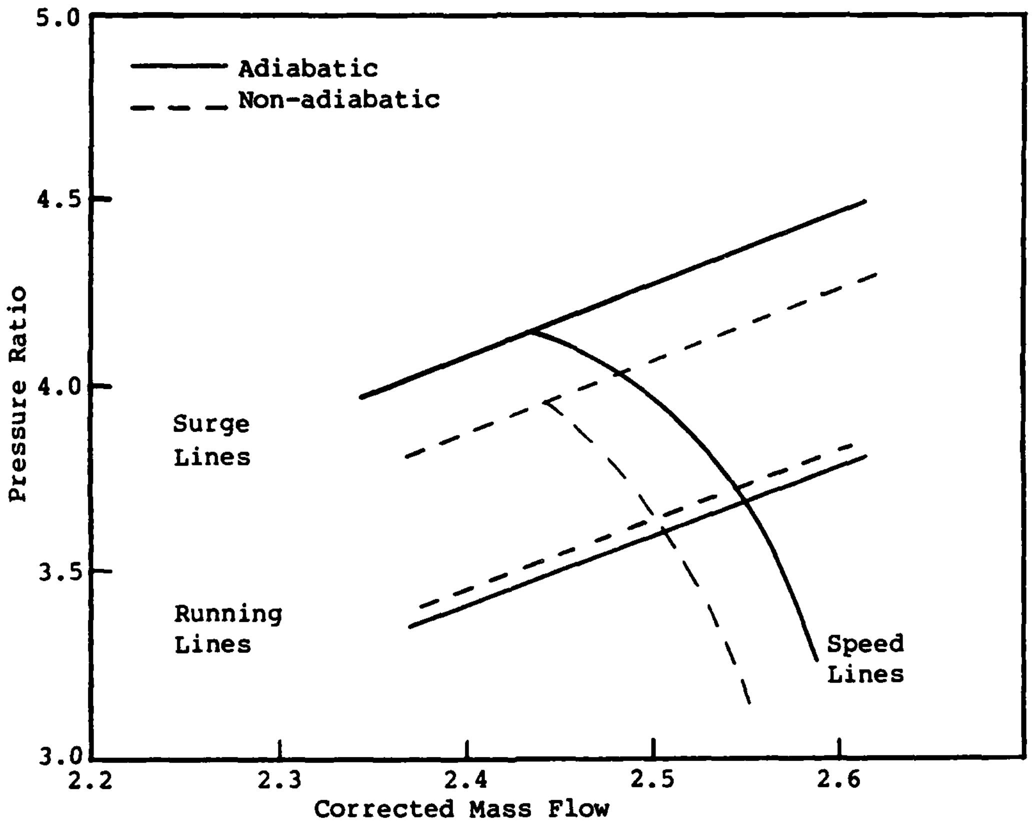

Figure 2. Effects of Heat Transfer on HP Compressor Surge and Steady-running Lines at the End of an Altitude Deceleration [3].

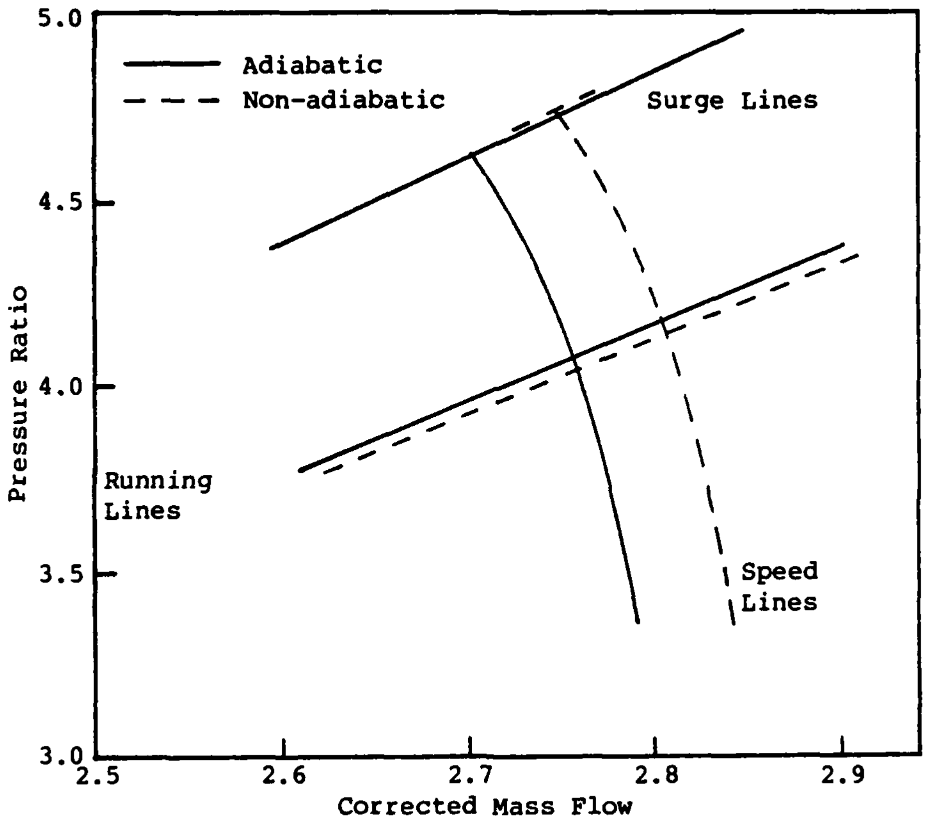

Figure 1. Effects of Heat Transfer on HP Compressor Surge and Steady-running Lines During Sea-Level Acceleration [3].

magnitude of the change in exit temperatures in the LP compressor was small, resulting in negligible change in the LP compressor characteristics. The HP compressor temperature reduction was more significant and was used to predict the change in the HP compressor characteristics as a result of heat transfer during an acceleration.

The prediction model used divided the 12-stage compressor into four groups of three stages and assigned characteristics to each group. All stalls were assumed to be of the full span type because of the high hub-to-tip ratios in the HP compressor stages. It was predicted that heat transfer during an acceleration resulted in negligible surge line movement and an increase in the constant speed line of the HP compressor. Figure 1 shows the shifts in compressor characteristics for this acceleration.

Maccallum [3] performed similar calculations for the three decelerations discussed earlier and found the most significant effects were in the HP compressor during the altitude deceleration from maximum speed to flight idle speed and during the sea-level deceleration from maximum speed to ground idle speed. The same prediction program was used for these decelerations, and Maccallum found that heat transfer caused a reduction in the constant speed line and a significant movement of the compressor surge line from the adiabatic surge line. These changes in surge margin and speed line are shown in Figure 2.

The effect of heat transfer on boundary layer stability was studied by Grant [4] and will be discussed separately. The effect of clearance changes required study of particular design configurations and was not included in Maccallum's [3] work. This paper was limited to a theoretical investigation of the effects of heat transfer on compressor characteristics and the engine working line.

To estimate the effects of "bulk" heat transfer on compressor characteristics, Maccallum [3] studied four transients on a typical twin-spool turbofan engine with an overall pressure ratio of 20. These transients included:

1. sea-level acceleration from ground idle speed to maximum speed,
2. altitude deceleration at 40,000 ft and Mach number of 0.61, from maximum engine speed to flight idle speed,
3. a sea-level deceleration over the same speed range as Item 2, and
4. a sea-level deceleration from maximum speed to ground idle speed.

In all cases the speed change was completed in approximately 10 seconds.

During the acceleration transient, heat transfer occurred from the air to the metal parts of the compressor. This resulted in exit temperatures below the adiabatic exit temperatures in both the LP and HP compressors. The heat transfer to turbine blades and turbine blade cooling, but none dealing with the effects of compressor blade heat transfer. This literature search confirmed the significance of transient heat transfer effects and revealed a lack of published experimental data. The results of the published theoretical investigations are discussed here.

The response of a gas turbine engine to a transient, such as an acceleration, is dependent on the condition of the engine immediately prior to the transient. Maccallum [3] suggested that the reason for this transient response is engine components operating away from their equilibrium characteristics. Reasons given by Maccallum for these shifts from equilibrium are:

1. blade boundary layer stability changes due to heat transfer from the air to the compressor or turbine blades during an acceleration or from the blades to the air during a deceleration,
2. air density changes due to heat transfer and a resultant change in compressor characteristics,
3. engine working line displacement resulting from non-adiabatic compressions and expansions, and
4. compressor and turbine axial and tip clearance changes resulting from different component response rates to the temperature changes.

There are many factors which influence compressor stall. These factors include inlet distortion, Reynolds number effects, transients, and the subject of this investigation, thermal transients.

The objectives of this investigation are:

1. to determine the total amount of thermal energy stored in the HP compressor blades and roots,
2. to quantify the heat transfer rate in the compressor during a deceleration transient,
3. to quantify the individual stage heat transfer rates and individual compressor blade heat flux rates, and
4. to estimate, using a flat plate boundary layer analysis, the effect of blade surface heat flux on the blade boundary layer displacement thickness and the influence of the displacement thickness change on the flow deflection angle at the compressor blade trailing edge.

Literature Survey

An extensive survey of published works, from both foreign and domestic researchers, in the area of transient heat transfer in axial flow compressors produced eight theoretical works, all from foreign sources. A further search for information specific to individual blade level heat transfer produced several references concerned with stall cell propagates through the compressor. In general, the stall zones may cover several blade passages and will rotate, counter to rotor rotation, at about half the engine speed [1]. Rotating stall significantly reduces the efficiency of the compressor. It may also lead to fatigue failure of the compressor blades due to bending stresses caused by the loading and unloading of blades as the stall cell moves through the compressor. The reduction in mass flow during rotating stall operation will result in a fuel-air mixture imbalance which could lead to overtemperature in the turbine and damage to the turbine blades [2].

Where rotating stall is characterized by one or more severely stalled cells rotating around the circumference of the compressor, surge is a large amplitude oscillation affecting the total averaged flow through the compressor. These oscillations are in the axial direction and may result in a reversal of flow within the compressor. During a surge cycle, the compressor may pass in and out of rotating stall as the mass flow through the compressor changes with time. Compressor surging may result in structural damage to the blades and casings similar to rotating stall; however, the frequency of a surge oscillation is usually more than an order of magnitude less than that associated with the passage of a rotating stall cell [2].

compressor blade is defined as a flow breakdown or separation due to a high angle of attack [1][1]. Since a compressor stage has many airfoils, it is likely the stalling process will affect several of these airfoils, resulting in a very complex flow behavior. The upper limit of compressor operation is defined as the surge line. Operation beyond this upper limit can lead to large oscillations in mass flow, called surge, or severe flow distortions which rotate around the compressor annulus, called rotating stall [2].

Rotating stall occurs when a single blade or group of blades within a compressor stage stalls. As the blade or group of blades stalls, the incoming airflow is diverted around the blade passages involved. As a result of this flow diversion, the blade on one side of the stalled blade passage operates at an increased angle of attack, while the blade on the opposite side of the stalled blade passage operates at a lower angle of attack. The blade with the increased angle of attack will eventually stall, causing the flow to be diverted around that blade passage. The flow diversion resulting from the new stalled blade will reduce the angle of attack of the originally stalled blade, causing it to recover. This process continues and the

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[1] Numbers in brackets refer to like-numbered items in the Bibliography.

CHAPTER I

INTRODUCTION

Background

The axial flow compressor used in aircraft gas turbine engines, such as turbojets and turbofans, utilizes alternating rows of stationary and rotating blades to convert shaft work into an increase in total pressure. This total pressure rise is accompanied by a corresponding increase in total temperature.

Current turbofan engines have high pressure ratios and consequently high temperature increases through the core compressor. The high temperature increase combined with the thermal capacity of the compressor metal parts results in large amounts of thermal energy being stored in the metal parts of the compressor. If a rapid deceleration occurs, such as a throttle snap from a high power setting to idle, the thermal energy stored in the metal parts will be released to the gas path of the compressor as the operating temperature decreases. This "bulk" thermal energy release causes a disturbance in the compressor flow field and could lead to rotating stall of the compressor if the throttle were readvanced.

Axial flow compressor aerodynamic instabilities are categorized as either stall or surge. Stall of a compressor. The first effect is a "bulk" effect which reduces compressor performance [3]. The second effect is related to blade and annulus wall boundary layer development.

Heat transfer from a wall causes a reduction in local density of the air flowing past the wall which results in more rapid boundary layer development on the wall. If a laminar boundary layer is present, transition from laminar to turbulent flow is also affected by heat transfer. Grant [4] also showed that if flow separation occurs, heat transfer will speed up the separation process moving the separation point upstream.

Blade and annulus boundary layer growth was the second effect of heat transfer suggested by Maccallum and Grant [7]. Slight thickening of the annulus boundary layer would cause losses in the compressor due to blockage. Compressor blade boundary layers would be expected to grow more rapidly under the combined effects of heat transfer from the blades and the adverse pressure gradients present on the suction surface of the blades. The thin boundary layers on the pressure surfaces of the blades would not be affected significantly by the heat transfer.

The combination of a strong adverse pressure gradient and heat transfer from the suction surface of the blades to the gas path results in a thicker boundary layer, a thicker wake behind each blade, and consequently an increase in compressor losses due to increased profile drag. Since cascade stall is related to profile drag, this drag increase also represents a shift toward stall.

Maccallum and Grant [7] performed calculations on cascades typical of axial compressors for adiabatic conditions and for conditions when the blade surface temperature was 55°R above that of the airflow. The region of most rapid boundary layer growth was found to be on the suction surface near the trailing edge of the blade. In that region the boundary layer was found to be fully turbulent. In all cases the result of heat transfer from the blades to the boundary layer was an increase in the displacement thickness of the boundary layer [7].

Maccallum and Grant [7] also analyzed the effect of a hot blade on the wake formed behind the blade. As discussed earlier, heat transfer has no significant effect on the pressure surface boundary layer and, consequently, on the wake component from the pressure surface. On the suction surface the increased boundary layer thickness results in a change in the angle of the wake leaving the trailing edge of the blade. This effect is shown in Figure 3. The effect of heat transfer from the blade to the boundary layer is a thicker wake and a reduction in the cascade deflection angle. From the experimental cascade data used with these prediction methods, Maccallum and Grant developed an analytic expression, Eq. (1),

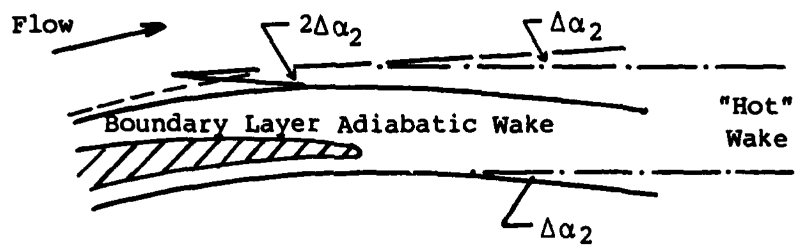


Figure 3. Effect of Heat Transfer on Wake Development [7].

relating the change in outlet angle to the temperature difference betwen the blade surface and the airflow:

$$\Delta\alpha_2/\varepsilon^* = \Delta T(0.0005 + 0.00084(i-i^*)/\varepsilon^*) \ . \qquad (1)$$

The expression did not exactly fit the predictions; however, it did provide an estimate of the effects of heat transfer on the stage deflection angle.

These effects of heat transfer were incorporated into the adiabatic prediction model by Maccallum and Grant [7] and used to predict the effect of a rapid deceleration from maximum speed to flight idle speed on the twin-spool turbofan engine used in previous investigations [3]. Heat transfer from the blades to the airstream caused a reduction in compressor surge margin of about 40 percent. This reduction is shown in Figure 4. The addition of "bulk" heat transfer effects [3] results in an additional reduction of 25 percent in the surge margin. The combined effects are also shown in Figure 4.

Maccallum [8] developed an additional method for predicting the effects of heat transfer on compressor performance. This method used the same stage-by-stage calculation procedure as the method of Ref. [7] with the addition of a factor to account for the development of annulus boundary layers on the hub and casing walls of the compressor.

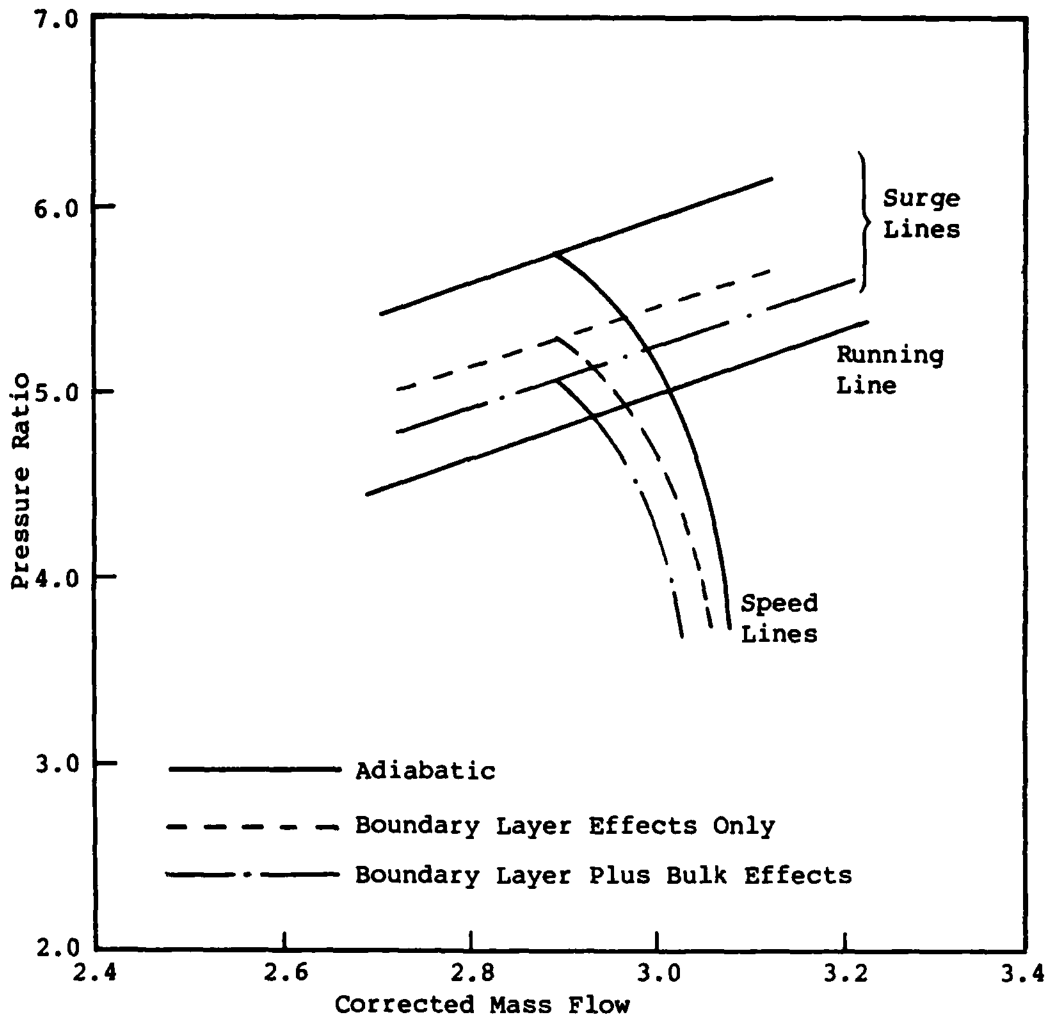


Figure 4. Combined Effects of Heat Transfer on HP Compressor During Deceleration [7].

Applying this revised method to the same HP compressors used in previous analyses [3,7], Maccallum [8] predicted behavior similar to that seen when only blade heat transfer was considered. Inclusion of the end-wall boundary layer effects in the program resulted in prediction of a less severe reduction in surge margin than that predicted by the earlier methods. Maccallum [8] attributed this lower prediction to a change in blade incidence and a reduced likelihood of flow separation on the blade suction surface.

Larjola [9] developed a computer simulation model for studying the effects of transients such as accelerations and decelerations on different types of gas turbine engines. Of particular interest in the simulation was correct representation of the effects of heat transfer between metal parts and the airflow through the gas turbine. Larjola incorporated material from Refs. [3,6,7] into his simulation to correct for heat transfer effects. Results of the simulation were in close agreement with the experimental results reported by Maccallum [10].

The information summarized in this section represents the results of theoretical models and compressor cascade analysis. These models predict a significant influence of transient heat transfer on compressor stability. Blade and end-wall boundary layer effects are driven by the flow Reynolds number, the laminar or turbulent characteristics of the boundary layers, and the actual heat transfer rate from the blade surface. There was no actual transient turbine engine data available to confirm these predictions.

This investigation uses high quality data from turbine engine transient testing to quantify the effects of heat transfer on the compressor and to estimate the effects of blade heat flux on individual blade boundary layers to provide a correlation between the theoretical predictions and the results of actual tests.

CHAPTER II

ANALYSIS

Test Description

The United States Air Force Aeropropulsion Laboratory and Arnold Engineering Development Center (AEDC) conducted an investigation into non-recoverable stall of turbofan engines. This investigation included a series of "Bodie" transients performed on the engines at different altitudes and flight conditions to determine the effect of these transients on the performance of the engine.

A Bodie transient is a standard engine test technique for evaluating the response of an engine to rapid throttle transients. Once the engine has reached a steady operating condition at the maximum power setting, the throttle setting is rapidly decreased, or snapped, to the minimum power setting. After a preselected period of time, the dwell time, has passed, the throttle is rapidly returned to the maximum power setting as shown in Figure 5. The response of the engine to such a deceleration-acceleration cycle will be either (1) a return to operation, (2) a stall with recovery, or (3) a non-recoverable stall of the engine.

Experimental data used in this analysis was obtained from four non-recoverable stall tests, at

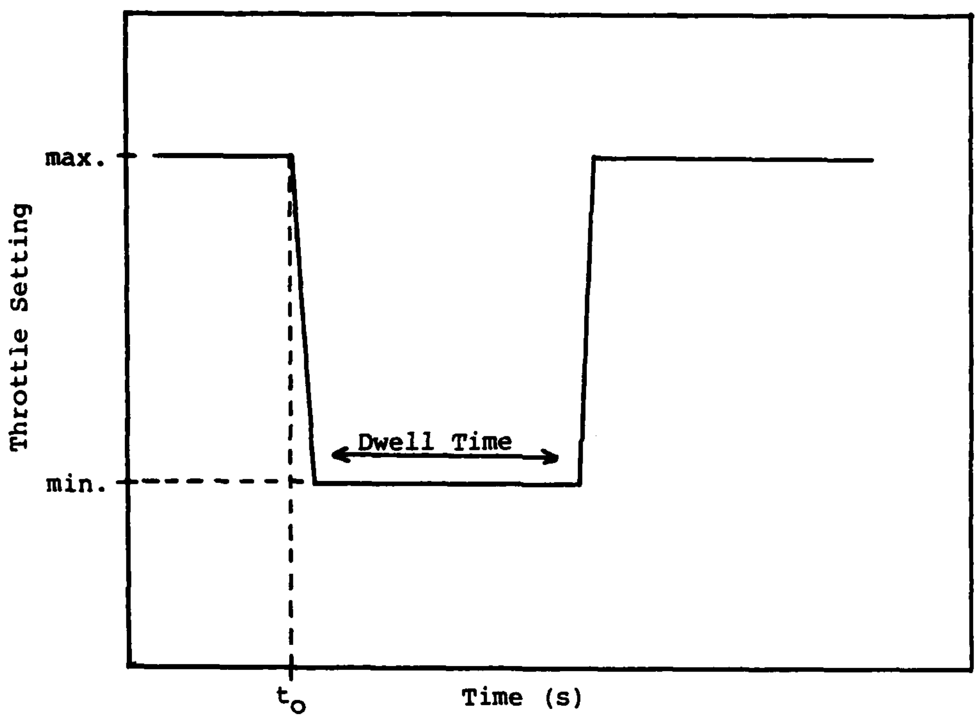


Figure 5. Bodie Transient.

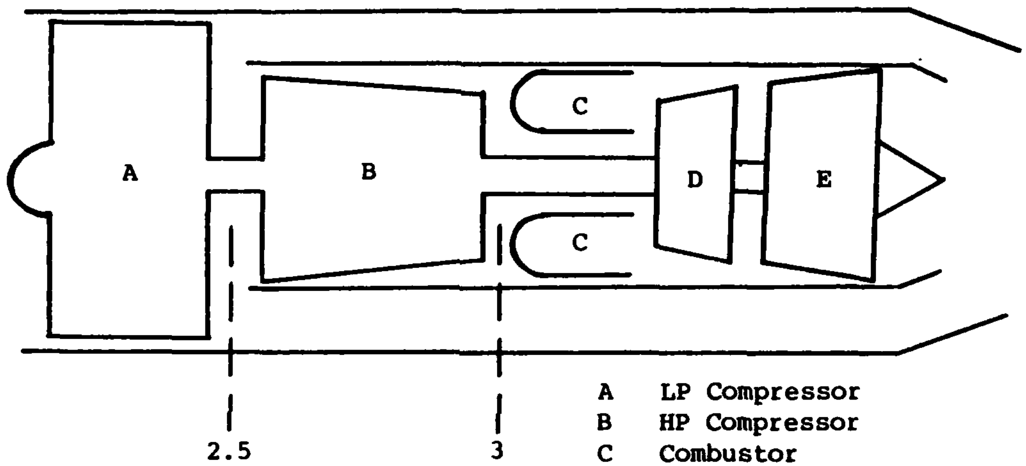
20,000 ft altitude and flight Mach number 0.8, of a typical twin-spool turbofan engine with a 10-stage HP compressor. This data included temperatures and pressures at the inlet, station 2.5, and exit, station 3, of the HP compressor as functions of time during the transient as shown in Figure 6.

Stored Thermal Energy Calculations

Current turbofan engines have high pressure ratios and, consequently, a high temperature rise through the HP compressor. This temperature rise results in thermal energy being stored in the blades, casings, and discs of the compressor. If a rapid deceleration occurs, such as in a Bodie transient, the thermal energy stored in the metal parts will be released into the gas path of the compressor.

In analyzing the effect of heat transfer on the HP compressor, it is necessary to determine the amount of thermal energy stored in the metal parts. This stored thermal energy results from the difference in temperatures in the compressor stages at the maximum throttle setting and at the flight idle setting. Figure 7 shows the temperature rise per stage for one of the four transients studied.

The total stored thermal energy to be released by the HP compressor during a transient can be calculated by



A LP Compressor
B HP Compressor
C Combustor
D HP Turbine
E LP Turbine
2.5 HP Compressor Inlet
3 HP Compressor Exit

Figure 6. Turbofan Engine Block Diagram.

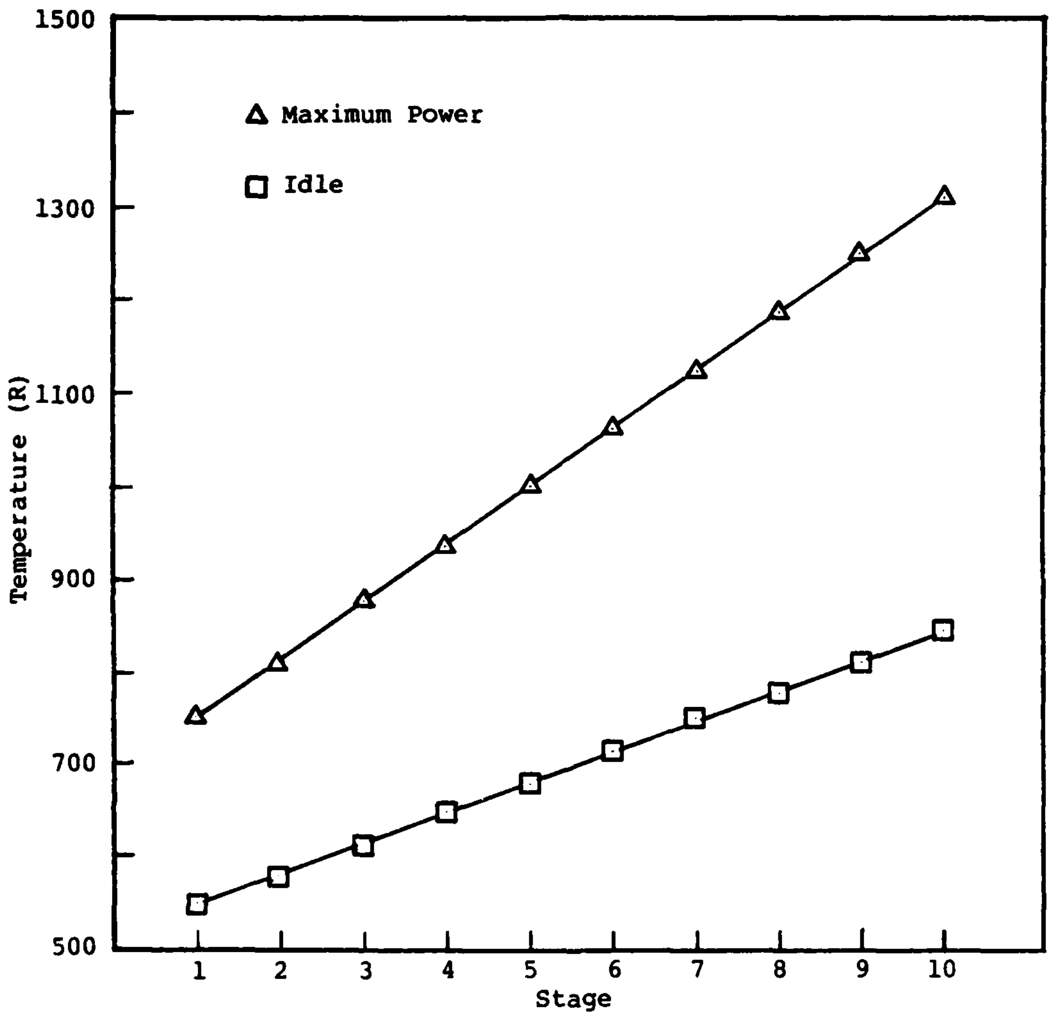


Figure 7. Total Temperature Rise Through HP Compressor.

summing the thermal energy stored in each stage as shown in Eq. (2):

$$Q = \sum [mC_p(T_{max}-T_{idle})]_{stage} \quad . \tag{2}$$

In this analysis, the stage mass, m, includes the masses of the blades and blade roots only. The temperature differences, in this typical transient, ranged from 204^{o}R in the first stage to 465^{o}R in the tenth stage. Values of C_p for the metals used in this compressor varied from 0.12 to 0.14 Btu/lbm-^{o}R. Table 1 shows the results of these calculations for one of the transients. The resultant stored thermal energy in the blades was approximately 2,400 Btu.

Consistent with the transient response times noted by Maccallum [3] of 0.5 to 10 seconds for compressor blades, and 40 seconds for a turbine disc, the current analysis was restricted to examining the effects of blade heat transfer on the HP compressor. Including discs and casings in the stored thermal energy analysis would at least double the amount of stored thermal energy available for release to the gas path during a transient.

Compressor Heat Transfer

Fowley [2] investigated the effects of transient heat transfer on compressor stability. That analysis

Table 1. Stored Thermal Energy in HP Compressor Blades

Stage	Mass (lbm)	C_p (Btu/lbm-^oR)	ΔT (^oR)	Q (Btu)
1	15.07	0.13	204	390.99
2	11.74	0.13	233	344.24
3	6.96	0.13	262	232.82
4	5.57	0.13	291	207.83
5	5.71	0.12	320	219.26
6	4.85	0.12	349	203.12
7	4.29	0.12	378	194.60
8	3.99	0.12	407	194.87
9	3.71	0.12	436	194.11
10	4.17	0.12	465	232.69
			Σ =	2414.53

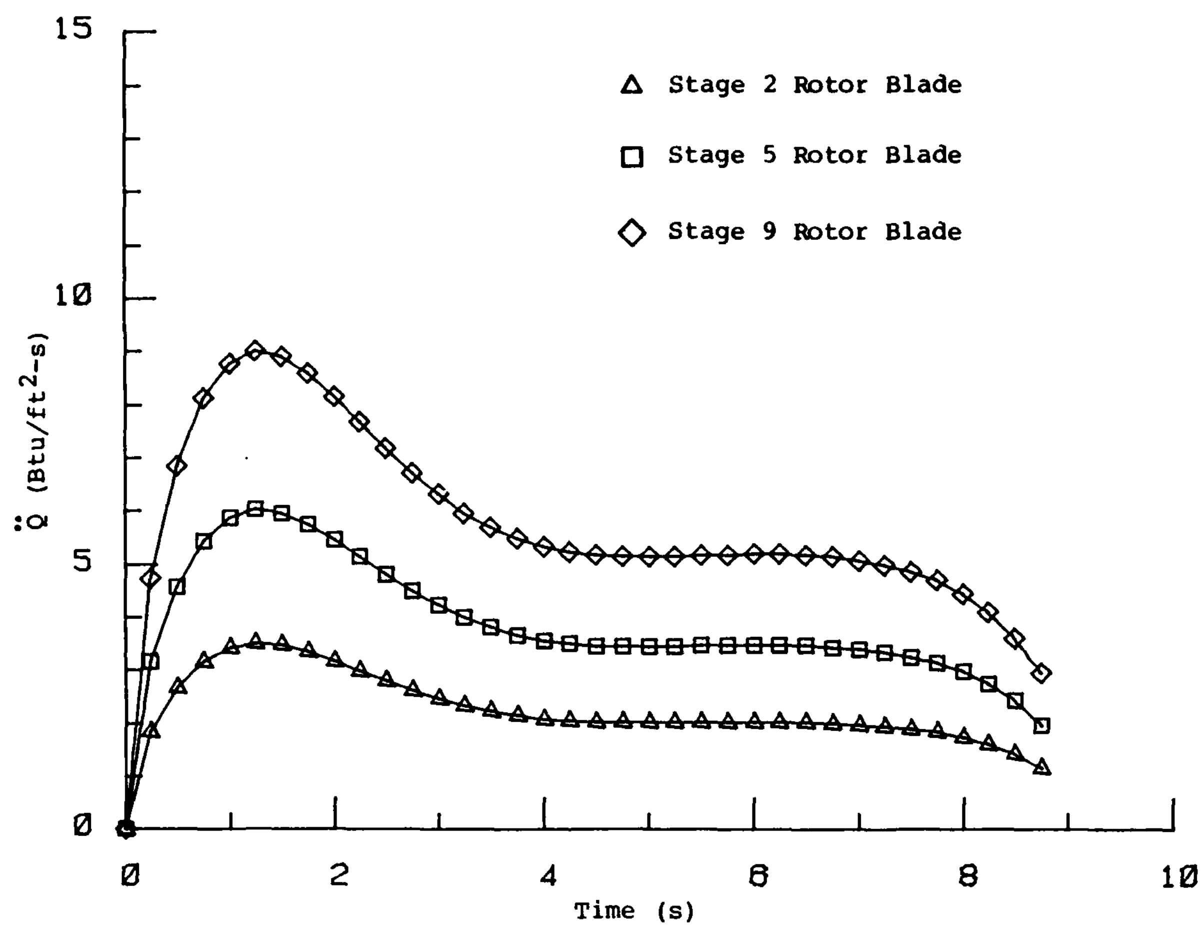


Figure 12. Typical Rotor Blade Heat Flux During Transient.

allocation method have the same heat transfer rate per stage.

Estimation of individual blade transient heat transfer per unit area followed from the stage-by-stage allocation results. Blade areas were determined from average height and chord data assuming the blades were of rectangular planform and negligible thickness. This single blade surface area was then multiplied by the total number of blades to determine the total blade surface area per stage. This total area was then used with the allocated stage heat transfer rate to determine the transient heat transfer per unit area for the blades.

Figure 12 shows the results for the rotor blades of the second, fifth, and ninth stages during the transient under consideration. In contrast to Figure 11 showing higher rates for the earlier stages in the compressor, the heat transfer per unit area is higher in the smaller blades near the rear of the HP compressor. The results shown in Figure 12 do not agree with the conclusion reached by Elder [5] that the compressor blades would complete the release of stored thermal energy in approximately 6 seconds. At 6 seconds into the transient, the compressor blades are still releasing a significant amount of thermal energy.

As in the analysis of the overall transient heat transfer, repeatability is an important consideration in the stage and blade analysis. Comparison of Figure 9,

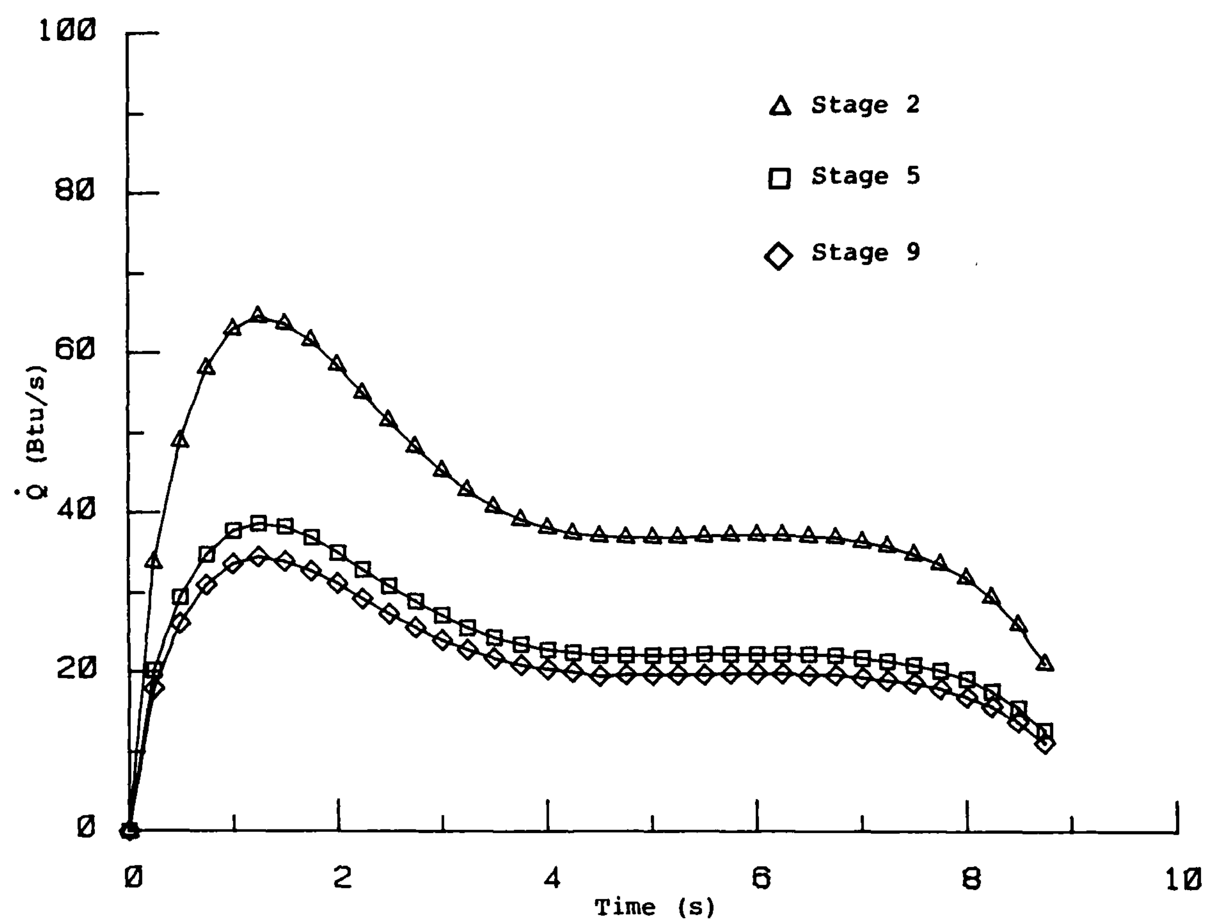


Figure 11. Typical Stage Heat Transfer Rates.

blade heat transfer will focus on the 8.75-second dwell time transient discussed earlier. Variation in the results for the other transients will be discussed at the end of this section.

It was decided that the heat transfer rate for the entire compressor would be allocated to the 10 stages based on individual stage contributions to the total stored thermal energy. Since the calculated stored thermal energy was for the blades only and accounted for approximately 50 percent of the total thermal energy dumped during this transient, the allocation of the overall heat transfer rate was based on 50 percent of the total thermal energy released. The remaining 50 percent of the thermal energy released comes from slower response time components such as discs, casings, and seals.

Figure 11 shows the results of the stage allocation for the second, fifth, and ninth stages during the transient under consideration. These stages were selected to show the heat transfer rates near the front of the HP compressor, at the midpoint into the compressor, and near the exit. The results in Figure 11 include both the rotor and stator blade masses in the selected stages. When the rotors of these three stages are considered without the stators, these three stages make similar percentage contributions to the total stored thermal energy, and under this position of the compressor stator vanes for surge margin control. In addition, the control system adjusts the flight idle setting to maintain either a minimum rotor speed or a minimum combustor pressure. The effect of this idle speed variation on the heat transfer rate would be a small change in the amount of stored thermal energy available for release during transients conducted at identical inlet and flight conditions. The major control system effect on the heat transfer rate would come from stator position changes. Stator position changes cause changes in the mass flow through the compressor. Lack of precise repeatability in stator position during the transient would cause mass flow variations which would cause variations in the transient heat transfer rate.

Blade Heat Transfer

As discussed previously, the only experimental measurements available were temperatures and pressures at the inlet and exit of the HP compressor. Determination of heat transfer effects on the blade boundary layer required the knowledge of the heat transfer per unit area from the [illegible] blade surfaces. Since no inter-stage measure- [illegible] were available, it was necessary to find a means of [illegible] the individual stage heat transfer rate, and [illegible] estimate the heat transfer per unit area for an [illegible] blade. The remainder of this discussion of indicated a total thermal energy release of 4,870 Btu during the transient. The difference in this number and the stored thermal energy in the blades, as seen in Table 1, page 28, is attributed to the discs, casings, and seals which were not considered in the stored thermal energy calculations.

In order to evaluate the repeatability of these heat transfer effects from the test data, four transients at the same inlet conditions, 20,000 ft altitude and flight Mach number 0.8, were analyzed. Examination of Figure 9 shows that at the peak heat transfer rate there is a difference of 200 Btu/s about a mean value of 925 Btu/s. At 5 seconds into the transient the four tests show a variation of approximately 95 Btu/s about the mean value of 516 Btu/s. These variations appear too large to be explained by instrumentation or measurement uncertainty.

The transient heat transfer rate can be affected by both internal and external factors. The primary external effects are flight conditions such as altitude and inlet Mach number. These parameters will directly influence the mass flow into the compressor due to air density variation with altitude and changing inlet velocity with changes in Mach number. The primary internal influence on transient heat transfer is the engine control system. As the engine operates, and in response to changes in flight conditions, the control system adjusts the

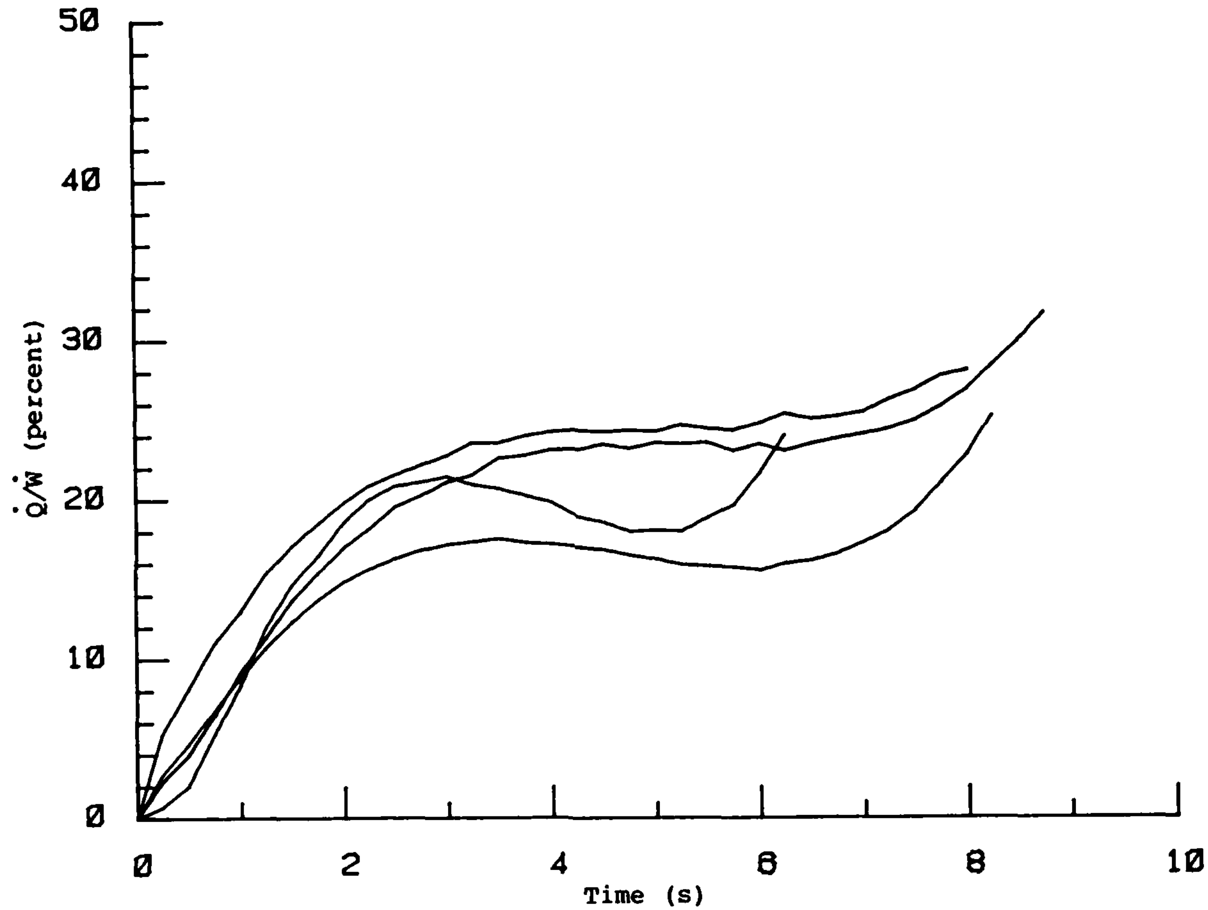


Figure 10. Heat Transfer Percentage of Work Transfer During Transients.

and occurred from 0.75 to 1.25 seconds into the transients. In each transient, heat transfer calculations were halted when the throttle was readvanced and the engine stalled.

Heat transfer rates shown in the curves of Figure 9 are the result of applying a least-squares smoothing technique on mass flow, $\dot{m}$, and the exit temperature difference, $(T_{t3}-T_{t3a})$, to allow easier integration or differentiation of the data [2]. Unsmoothed curves show some irregularity due to the uncertainty and precision of the experimental data. A $\pm$10-percent accuracy on the transient heat transfer rate is considered acceptable.

As the engine slows down during the deceleration portion of a throttle transient, the adiabatic work transfer from the air decreases more rapidly than the heat transfer from the metal components to the air. This is due to the more rapid decrease in the adiabatic exit temperature as shown in Figure 8 for one of the four transients. Figure 10 shows the changes in the ratio $\dot{Q}/\dot{W}$ for the four transients. This ratio corresponds to Maccallum's F factor [3], and behaves in a similar manner, approaching a maximum near the completion of the deceleration.

Information on the transient with the longest dwell time, 8.75 seconds, was used to determine the stored thermal energy in the HP compressor blades. Numerical integration of the area under this curve in Figure 9

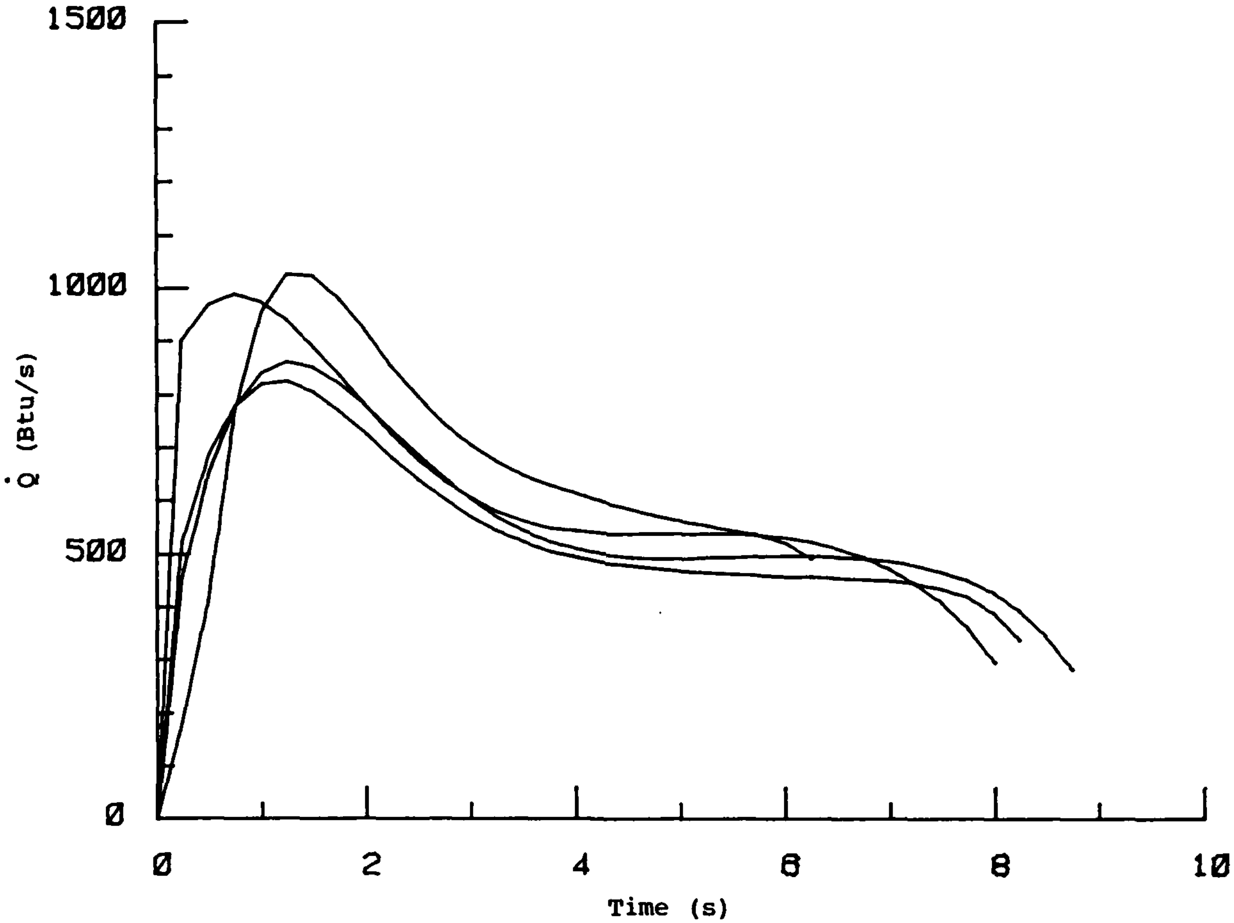


Figure 9. Transient Heat Transfer Rates.

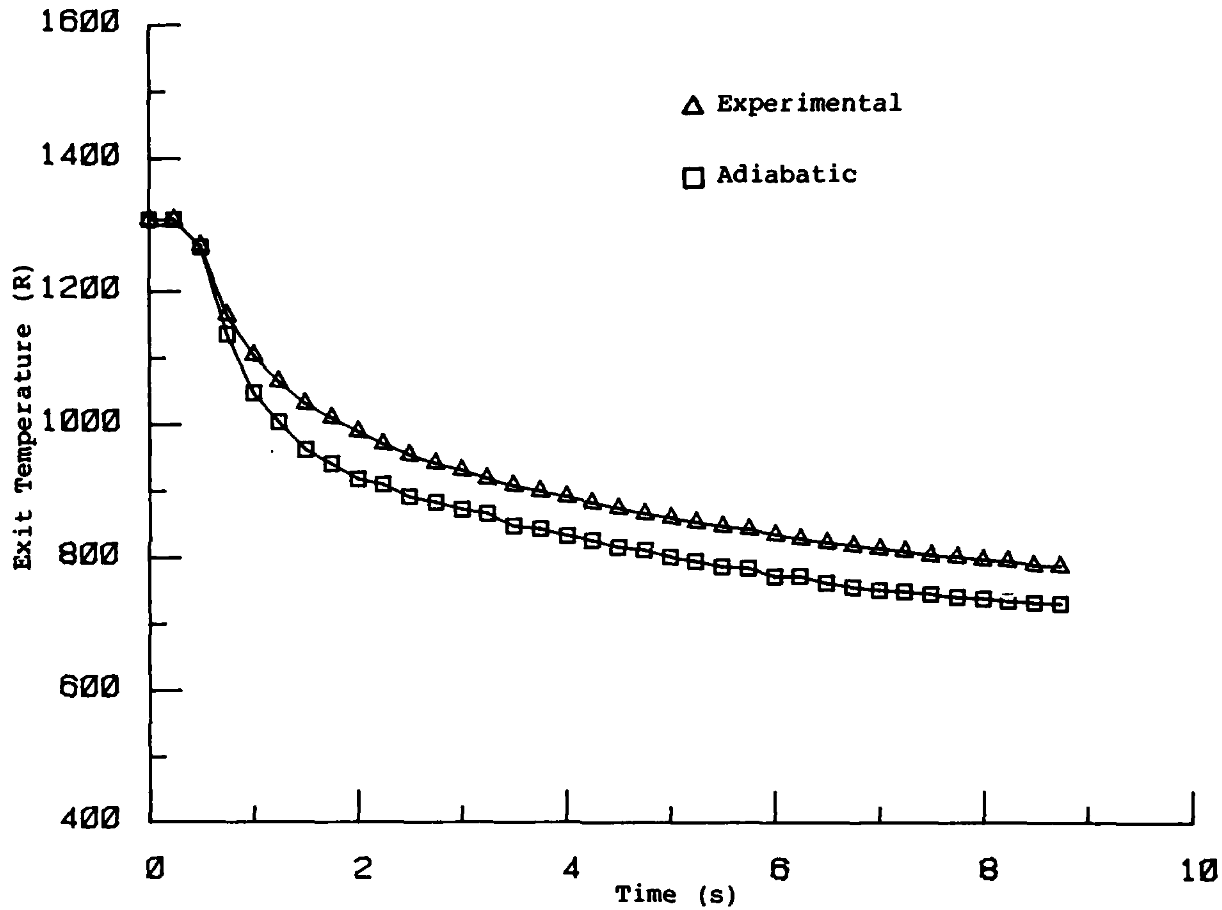


Figure 8. Experimental and Adiabatic Exit Temperatures During Transient.

Assuming constant specific heat, C_p, across the compressor, the energy equation can be written

$$\dot{Q} + \dot{W} = \dot{m}C_p(T_{t3}-T_{t2.5}) \ , \tag{13}$$

where the temperatures are the experimentally measured inlet and exit temperatures of the compressor. The work transfer from the air, $\dot{W}$, is calculated based on the transient mass flow determined from Eq. (10) and the adiabatic exit temperature calculated from Eq. (11):

$$\dot{W} = \dot{m}C_p(T_{t3a}-T_{t2.5}) \ . \tag{14}$$

Substituting Eq. (14) into Eq. (13) and solving for the heat transfer rate gives

$$\dot{Q} = \dot{m}C_p(T_{t3}-T_{t2.5}) - \dot{m}C_p(T_{t3a}-T_{t2.5}) \ , \tag{15}$$

which simplifies to

$$\dot{Q} = \dot{m}C_p(T_{t3}-T_{t3a}) \ , \tag{16}$$

and can be used to solve for the heat transfer rate during the transient. Figure 8 shows the variations of the experimental and adiabatic exit temperatures during one of the transients studied.

The results of the calculations of Eq. (16) for the four transients studied are shown in Figure 9. The maximum heat transfer rates ranged from 824 to 1,025 Btu/s,

$$M = \left(\frac{2}{\gamma-1}\left(\frac{P_t}{P_s}\right)^{\frac{\gamma-1}{\gamma}} - \frac{2}{\gamma-1}\right)^{1/2} . \tag{9}$$

Substituting into the mass flow equation gives:

$$\dot{m} = AP_t \left(\frac{\gamma}{RT_t}\right)^{1/2} \left(\frac{P_t}{P_s}\right)^{-\frac{(\gamma+1)}{2\gamma}} \left[\frac{2}{\gamma-1}\left(\frac{P_t}{P_s}\right)^{\frac{\gamma-1}{\gamma}} - \frac{2}{\gamma-1}\right]^{1/2} . \tag{10}$$

Calculation of transient heat transfer rates also requires knowledge of the adiabatic efficiency of the HP compressor. During a transient this efficiency will not remain constant; however, for purposes of this analysis, it is assumed to be constant and equal to the value at the initiation of the transient as determined from Eq. (11):

$$\eta_c = \frac{\left[\left(\frac{P_{t3}}{P_{t2.5}}\right)^{\frac{\gamma-1}{\gamma}} -1\right]}{\left[\frac{T_{t3}}{T_{t2.5}} -1\right]} . \tag{11}$$

A calculated efficiency of 82 percent was used for the remaining heat transfer calculations. This efficiency value is also used in Eq. (11) with the measured P_{t3}, $P_{t2.5}$, and $T_{t2.5}$ to solve for the adiabatic exit temperature, T_{t3a}.

Equation (12) shows the energy balance for the HP compressor:

$$\dot{Q} + \dot{W} = \dot{m}(h_{t3} - h_{t2.5}) . \tag{12}$$

significant during a relatively slow transient such as an acceleration or deceleration; however, the storage effects can be important during stall or surge. The mass flow storage and release will be particularly important during the large amplitude mass flow oscillations associated with a surge cycle. This analysis is concerned with deceleration transients only and assumes the HP compressor is operating under steady-flow conditions.

Using the steady-flow assumption, the transient mass flow at the HP compressor inlet can be determined from the measured temperatures and pressures, the one-dimensional mass flow equation, and the isentropic flow relations:

$$\dot{m} = \rho A V \ , \tag{3}$$

$$\rho = \frac{P_s}{R T_s} \ , \tag{4}$$

$$\frac{P_t}{P_s} = \left(1 + \frac{(\gamma-1)M^2}{2}\right)^{\frac{\gamma}{\gamma-1}} \ , \tag{5}$$

$$\frac{T_t}{T_s} = \left(\frac{P_t}{P_s}\right)^{\frac{\gamma-1}{\gamma}} \ , \tag{6}$$

$$V = Ma \ , \tag{7}$$

$$a = (\gamma R T_s)^{1/2} \ , \tag{8}$$

compared the heat transfer rates in back-to-back data points from AEDC non-recoverable stall testing. The preliminary results indicate a possible correlation between the magnitude of the transient heat transfer and the onset of compressor instability such as stall. In particular, transients with higher heat transfer rates encountered compressor instabilities, while those with the lower rates did not. This investigation uses the calculation procedure developed by Fowley [2] as the starting point for an analysis of the effects of heat transfer during a Bodie transient on the compressor stages and individual blades.

Once the total stored thermal energy in the blades was determined, further analysis required calculation of the transient heat transfer rate during a Bodie transient. This calculation requires knowledge of the transient mass flow through the HP compressor. As previously discussed, the available experimental data consisted of $P_{s2.5}$, $P_{t2.5}$, $T_{t2.5}$, P_{t3}, and T_{t3}. Since no transient mass flow information was available from the tests, it was necessary to develop a calculation procedure using only the available experimental data.

During rapid compressor operating condition changes, transient mass flow will be stored and released from the compressor. This mass is stored in the primary flow path of the compressor as well as in trapped internal areas of the compressor. Mass flow storage effects will not be page 34, and Figure 11 show that the individual stages behave in much the same way as the overall compressor. This is to be expected since stage heat transfer rates were determined as a percentage of the overall heat transfer rates. At the peak heat transfer rate there are differences of 15 Btu/s about a mean of 69 Btu/s in the second stage, 9 Btu/s about a mean of 42 Btu/s in the fifth stage, and 8 Btu/s about a mean of 37 Btu/s in the ninth stage. At 5 seconds into the transient, the four tests show variations of 7 Btu/s about a mean of 39 Btu/s in the second stage, 4 Btu/s about a mean of 23 Btu/s in the fifth stage, and 4 Btu/s about a mean of 21 Btu/s in the ninth stage.

Boundary Layer Analysis

In addition to the overall heat transfer effects, a first approximation of the effects of heat transfer on the blade boundary layer displacement thickness was desired. For the purposes of this analysis, the compressor blades have been represented by finite flat plates at zero angle of attack and zero pressure gradient.

For these boundary layer calculations, the laminar boundary layer region and the transition region have been neglected and the flow assumed fully turbulent from the leading edge of the plate. This analysis uses the compressible, turbulent flow equations for a flat plate in zero pressure gradient.

None of the HP compressor blades were instrumented, so there was no way to determine the actual ratio of surface temperature to free-stream temperature, T_w/T_∞, during the transient. Two temperature ratios in the same range as those studied by Grant [4] were selected for this analysis. They are $T_w/T_\infty = 1.1$ and $T_w/T_\infty = 1.2$. A free-stream Mach number of 0.395 was selected since it represents the flow conditions at the point of maximum heat transfer in the HP compressor. Free-stream conditions used in this analysis are from the tenth stage of the HP compressor since measured values of temperatures and pressures were available there.

It was also necessary to assume a velocity profile and density relationship for this analysis. Maise and McDonald [13] used a general law-of-the-wall, law-of-the-wake velocity profile for incompressible flow, and used van Driest's [14] transformation to establish a velocity profile for compressible, flat plate boundary layers. This profile is given by:

$$\frac{\bar{u}}{u_\infty} = \frac{1}{\sigma^{1/2}} \sin \left\{ - \frac{u_I}{u_\infty} \sigma^{1/2} \left[- 2.5 \ln \left(\frac{y}{\delta}\right) + 1.25\left(1 + \cos \pi \left(\frac{y}{\delta}\right)\right)\right] + \arcsin \sigma^{1/2} \right\}, \quad (17)$$

where

$$\sigma = \frac{[(\gamma-1)/2]M_\infty^2}{1 + [(\gamma-1)/2]M_\infty^2} \tag{18}$$

and

$$\frac{u_\tau}{u_\infty} = \left(\frac{\overline{C}_f}{2}\right)^{1/2} . \tag{19}$$

The density profile selected is one given by van Driest [14], and is:

$$\frac{\overline{\rho}}{\rho_\infty} = \frac{T_\infty/T_W}{1 + B(\overline{u}/u_\infty) - A^2(\overline{u}/u_\infty)^2} , \tag{20}$$

where

$$A^2 = \frac{\frac{(\gamma-1)}{2} M_\infty^2}{T_W/T_\infty} \tag{21}$$

and

$$B = \frac{1 + \frac{(\gamma-1)}{2} M_\infty^2}{T_W/T_\infty} - 1 . \tag{22}$$

For turbulent flow past a flat plate with zero pressure gradient, the von Karman momentum integral relation reduces to [12]:

$$\frac{d\theta}{dx} = \frac{C_f}{2} , \tag{23}$$

where θ, the boundary layer momentum thickness, is given by

$$\frac{\theta}{\delta} = \int_0^1 \frac{\bar{\rho}}{\rho_\infty} \frac{\bar{u}}{u_\infty} \left(1 - \frac{\bar{u}}{u_\infty}\right) d\left(\frac{y}{\delta}\right) . \tag{24}$$

The boundary layer shape factor, H, is:

$$H = \frac{\delta^*}{\theta} , \tag{25}$$

and the boundary layer displacement thickness is:

$$\frac{\delta^*}{\delta} = \int_0^1 \left(1 - \frac{\bar{\rho}}{\rho_\infty} \frac{\bar{u}}{u_\infty}\right) d\left(\frac{y}{\delta}\right) . \tag{26}$$

Solving Eq. (25) for δ^*, taking its first derivative with respect to x, and substituting Eq. (23), gives the following expression for the local rate of change of the boundary layer displacement thickness along the flat plate:

$$\frac{d\delta^*}{dx} = H \frac{C_f}{2} . \tag{27}$$

The boundary layer shape factor, H, was determined by substituting Eqs. (17) and (20) into Eqs. (24) and (26) and numerically integrating Eqs. (24) and (26) for each of the temperature ratios selected. The resulting boundary layer thickness ratios were used in Eq. (25) to determine the shape factors. For $T_W/T_\infty = 1.1$, the shape factor was 1.60927, and for $T_W/T_\infty = 1.2$, the shape factor was 1.74293.

Values of the local and total friction coefficients required in this analysis were obtained from the semiempirical calculation procedure developed by Spalding and Chi [15]. This method allows simple determination of the friction coefficients when the Reynolds number, Mach number, and temperature ratio are known. This procedure was used with a Mach number of 0.395 and a Reynolds number, based on blade chord, of 5.01×10^5. The resulting friction coefficients for $T_w/T_\infty = 1.1$, are $\overline{C}_f = 0.00478$ and $C_f = 0.0038$. For $T_w/T_\infty = 1.2$, the results are $\overline{C}_f = 0.00473$ and $C_f = 0.00375$.

Once the friction coefficients and shape factors were determined, Eq. (27) was used to estimate the rate of change of the boundary layer displacement thickness at the trailing edge of the flat plate. For the lower temperature ratio, Eq. (27) gave $d\delta^*/dx = 0.00306$, while for the higher temperature ratio, the result was $d\delta^*/dx = 0.00327$. This indicates that as the ratio of surface temperature to free-stream temperature increases, the boundary layer displacement thickness will develop at a faster rate. Maccallum and Grant [7] showed similar results in their theoretical study of the effects of heat transfer on compressor performance. This increased displacement thickness resulting from heat transfer from the blades to the airflow will result in increased flow deviation angle at the blade trailing edge and a reduction in the performance of the compressor stage.

In addition to determining the change in displacement thickness, the average heat transfer coefficients can be calculated using the Reynolds analogy modified for non-unity Prandtl number [12]:

$$C_h = \frac{\overline{C}_f}{2\ Pr^{2/3}} \ . \tag{28}$$

For a turbulent Prandtl number of 0.86, and using the previously determined total friction coefficients, the Stanton number was found to be 0.00264 for $T_w/T_\infty = 1.1$, and 0.00261 for $T_w/T_\infty = 1.2$. The average heat transfer coefficients can be determined using Eq. (29):

$$\overline{h} = C_h \rho_\infty u_\infty C_p \ . \tag{29}$$

For $T_w/T_\infty = 1.1$, the value of $\overline{h}$ was 0.07173 Btu/ft^2-s-^{o}R, and for $T_w/T_\infty = 1.2$, the value of $\overline{h}$ was 0.07092 Btu/ft^2-s-^{o}R. The average heat transfer coefficient calculated from the heat transfer allocation procedure at the peak heat transfer rate, in the tenth stage of the HP compressor, was 0.10362 Btu/ft^2-s-^{o}R. This experimental heat transfer coefficient is for a temperature ratio, T_w/T_∞, of approximately 1.08. The heat transfer coefficients determined from the simplified flat plate boundary layer analysis are in acceptable agreement with the heat transfer coefficient determined from the allocation of overall HP compressor transient heat transfer to individual compressor blades.

The boundary layer analysis presented here represents a first approximation for the effects of heat transfer on HP compressor blade boundary layers. A more accurate determination of the effects of heat transfer on transient performance of the compressor would require consideration of both the favorable and the adverse pressure gradients which are found on compressor blades.

CHAPTER III

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

The objectives of this study were to determine the magnitude of heat transfer in the compressor during a throttle transient and to estimate the effect of transient heat transfer on compressor blade boundary layer growth.

Thermal energy stored in the HP compressor will be released during a deceleration, contributing to a possible temperature-induced boundary layer instability in the compressor flow field that may cause compressor stall if the throttle is readvanced before the thermal energy is dissipated. The magnitude of stored thermal energy available for release during a transient is a function of inlet conditions and engine idle setting. Compressor blades, casings, and discs all have capacity for storing and releasing thermal energy. Consistent with results found in the literature survey work by Maccallum [3], this analysis considered only the blades and blade roots in the stored thermal energy calculations. The resulting stored thermal energy for a typical transient in this analysis was approximately 2,400 Btu for a typical jet engine compressor. Adding the contributions of the other metal parts in the compressor would at least double the amount of stored thermal energy available for release during a transient.

The overall heat transfer rate in the HP compressor during a "Bodie" throttle transient was determined using a previously developed transient heat transfer analysis program [2]. Numerical integration of the area under the overall heat transfer rate curve gave a total thermal energy release, from throttle chop to throttle readvance, of approximately 4,800 Btu. This thermal energy release is double the available amount determined from the calculations for the blades and blade roots. The difference in thermal energy amounts is due to neglecting the discs and casings in the stored thermal energy analysis.

Calculation of the effects of heat transfer on individual compressor blades required estimation of the heat flux on an individual compressor blade. The overall heat transfer rate was allocated to the ten stages of the HP compressor based on the percentage each stage contributed to the calculated total stored thermal energy. With rotor and stator masses of each stage combined, this allocation procedure resulted in higher heat transfer rates in the stages near the front of the compressor which had larger stators and, thus, larger combined masses.

Individual blade heat flux rates were determined using the allocated stage heat transfer rates. Each compressor blade was assumed to be of rectangular planform and of negligible thickness. Total blade area was calculated from this individual blade area and used with the stage heat transfer rate to determine the blade heat flux rate. Highest heat flux rates were found in the small blades in the rear stages of the compressor.

A simple boundary layer analysis was performed in an attempt to confirm the assumptions made in the transient heat transfer allocation procedure and analysis. These boundary layer calculations were for a fully turbulent boundary layer in zero pressure gradient, compressible flow over a flat plate. Free-stream conditions for the calculations were the same as those found in the tenth stage of the HP compressor at the peak heat transfer rate. The calculations were performed for wall to free-stream temperature ratios, T_W/T_∞, of 1.1 and 1.2.

Average heat transfer coefficients were also calculated for the two temperature ratios. Comparison of these calculated values with the heat transfer coefficient determined in the heat transfer allocation process gives agreement to within 30 percent between the heat transfer coefficients. The agreement in these coefficients indicates that the assumptions made in the heat transfer allocation procedure are reasonable, and that a simple boundary layer analysis with the absence of any modifying pressure gradient provides a good first estimate of heat transfer effects on blade boundary layers.

This boundary layer analysis also confirmed that boundary layer displacement thickness increases with heat transfer from the blade to the gas path. In this zero pressure gradient analysis, the change in displacement thickness with heat transfer was small, as was the change in flow deviation angle at the trailing edge of the blade. Further understanding of the effects of heat transfer on displacement thickness and deviation angle requires inclusion of the blade pressure gradients.

Recommendations

This theoretical investigation has provided an estimation of the effects of transient heat transfer on the HP compressor, including a first approximation of the effects of blade surface heat flux on boundary layer displacement thickness and flow deviation angle at the blade trailing edge. Analytic correlation between blade heat transfer results and stage stability requires a non-adiabatic viscous flow model or an experimental study.

A more detailed theoretical analysis is desirable. This analysis of cascade stability should include the effects of blade curvature and realistic compressor blade pressure gradients. Consideration should also be given to the entire regime of boundary layer behavior from the laminar flow region, through transition, to fully turbulent flow [11]. This theoretical investigation should also provide information on the effects of blade heat flux on compressor stability, including boundary layer displacement thickness changes and resultant flow deviation angle changes.

This investigation has given first estimates of the blade surface heat flux present in a compressor stage during a transient. Further information on how this heat flux affects compressor stability could be provided by a two-dimensional compressor cascade study. Such a study, using heated compressor blades, could provide confirmation of the data from the heat transfer allocation procedure as well as a better indication of the effects of surface heat flux on cascade performance.

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