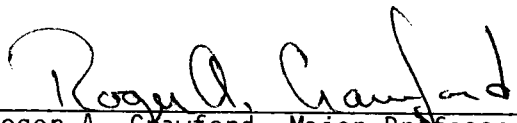
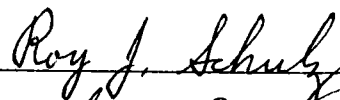
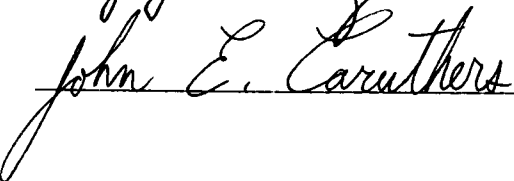


To the Graduate Council:

I am submitting herewith a thesis written by Mark D. Fowley entitled "Quantitative Evaluation of the Effects of Transient Heat Transfer on Axial Flow Compressor Stability." I have examined the final copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering.

  
Roger A. Crawford, Major Professor

We have read this thesis  
and recommend its acceptance:

Accepted for the Council:

  
The Graduate School

QUANTITATIVE EVALUATION OF THE EFFECTS  
OF TRANSIENT HEAT TRANSFER ON AXIAL  
FLOW COMPRESSOR STABILITY

A Thesis  
Presented for the  
Master of Science  
Degree  
The University of Tennessee, Knoxville

Mark D. Fowley  
December 1984

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## ABSTRACT

The purpose of this investigation is to study the effects of transient heat transfer on the stability of axial flow compressors. During speed transients the state of the compressor can be considered non-adiabatic. This is due to heat transfer between the compressor metal parts and the air flow. For instance, during a deceleration, heat is transferred from the hot metal to the cooler air. This phenomenon is detrimental to the stability margin of the compressor. The density changes of the air and boundary layer reactions during heat transfer reduce the margin of stability. When a deceleration is immediately followed by an acceleration the surge margin reduction becomes even worse. The response of a compressor during an acceleration can reduce the surge margin even further. Since the surge margin has already been diminished due to heat transfer, the possibility of the compressor becoming unstable is greater.

First, this investigation thoroughly researched the topic of the effects of heat transfer on compressor stability. Then using engine test data the thermal energy storage capacity for a typical engine and the heat transfer rate during a speed transient were quantified. The transient investigated was a "bodie" transient, which is a standard engine stability test for gas turbine engines. The "bodie" transient is a deceleration followed by an acceleration. Instabilities usually occur during the acceleration part of the transient. Only pre-stall conditions were analyzed, i.e., only the deceleration was

investigated. The heat transfer rate was quantified for four sets of back to back data points that were run at similar operating conditions. One of the tests encountered a stall; the other did not. Through an analysis procedure, it was determined that the stalled test had a greater heat transfer rate than the unstalled test. This demonstrates the link between the occurrence of compressor instability and heat transfer during speed transients.

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## LIST OF SYMBOLS

|                  |                                                                        |
|------------------|------------------------------------------------------------------------|
| $A$ . . .        | Flow area ( $\text{ft}^2$ )                                            |
| $a$ . . .        | Sonic velocity ( $\text{ft/sec}$ )                                     |
| $c, \vec{c}$ . . | Absolute velocity ( $\text{ft/sec}$ )                                  |
| $c_p$ . . .      | Specific heat at constant pressure ( $\text{Btu/lbm-}^\circ\text{R}$ ) |
| $E$ . . .        | Energy                                                                 |
| $h$ . . .        | Enthalpy ( $\text{Btu/lbm}$ )                                          |
| $\dot{m}$ . . .  | Mass ( $\text{lbm}$ )                                                  |
| $m$ . . .        | Mass flow rate ( $\text{lbm/sec}$ )                                    |
| $N$ . . .        | Rotational speed ( $\text{rev/min}$ )                                  |
| $P_s$ . . .      | Shaft power ( $\text{ft-lbf/sec}$ )                                    |
| $p$ . . .        | Pressure ( $\frac{\text{lbf}}{\text{ft}^2}$ )                          |
| $Q$ . . .        | Heat transfer ( $\text{Btu}$ )                                         |
| $\dot{Q}$ . . .  | Heat transfer rate ( $\text{Btu/sec}$ )                                |
| $R$ . . .        | Gas constant ( $\text{ft-lbf/lbm-}^\circ\text{R}$ )                    |
| $T$ . . .        | Temperature ( $^\circ\text{R}$ )                                       |
| $U, \vec{U}$ . . | Rotor velocity ( $\text{ft/sec}$ )                                     |
| $w, \vec{w}$ . . | Relative velocity ( $\text{ft/sec}$ )                                  |
| $\eta$ . . .     | Efficiency                                                             |
| $\Omega$ . . .   | Rotational speed ( $\text{rad/sec}$ )                                  |
| $\rho$ . . .     | Density ( $\frac{\text{lbm}}{\text{ft}^3}$ )                           |
| $\tau$ . . .     | Torque ( $\text{ft-lbf}$ )                                             |

## SUBSCRIPTS

|                |                                                 |
|----------------|-------------------------------------------------|
| a . . .        | Ahead of rotor row                              |
| act . .        | Actual process                                  |
| ad. . .        | Adiabatic process                               |
| b . . .        | Behind the rotor row                            |
| bb. . .        | Behind the stator row                           |
| c . . .        | Pertaining to the compressor                    |
| e . . .        | Exit of the compressor                          |
| ht. . .        | Heat transfer                                   |
| i . . .        | Inlet of the compressor                         |
| s . . .        | Static value                                    |
| st. . .        | Pertaining to the stage                         |
| t . . .        | Stagnation value                                |
| z . . .        | Axial velocity component                        |
| $\alpha$ . . . | Absolute flow angle                             |
| $\beta$ . . .  | Relative flow angle                             |
| $\theta$ . . . | Tangential velocity component                   |
| 2.5 . .        | Pertaining to inlet of high pressure compressor |
| 3 . . .        | Pertaining to exit of high pressure compressor  |

## SUPERSCRIPTS

|         |                    |
|---------|--------------------|
| ' . . . | Isentropic process |
|---------|--------------------|

## INTRODUCTION

The axial flow compressor is a very important component of the modern gas turbine engine. Its stable operation is necessary for the performance of the entire machine. In certain circumstances the axial flow compressor may be forced to operate at pressure ratios and flow rates that cause unstable operation. Compressor instabilities may occur during transient conditions, such as inlet pressure and temperature variations and changes in the rotational speed. The behavior of compressors during speed transients is significant since both the operating characteristics and the stall line vary during the transient. The stable operation of the compressor is essentially dependent on pressure ratio, mass flow, rotational speed, and the turbulent boundary layer behavior on the blades and annulus. During speed transients, heat transfer from the compressor to the airflow tends to increase the thickness of the viscous boundary layer along the annulus and compressor blades which alters the performance of the compressor and increases the possibility of compressor instability. The purpose of this investigation is to study the effect of transient heat transfer on compressor stability; first, by reviewing previous publications concerning this topic, and then by quantifying the heat storage capacity and heat transfer rates from engine test data.

## CHAPTER I

### BACKGROUND INFORMATION

#### Axial Flow Compressors

An axial flow compressor operates by imparting kinetic energy to a working fluid, usually air, by means of alternating rotary and stationary blades called rotor and stator blades, respectively. The combination of a rotor row and stator row makes up a single compressor stage. The rotor row adds kinetic energy to the flow by adding angular momentum in the form of angular velocity and directs the flow to the following stator row. The stator row diffuses the flow which increases the static pressure. Together they produce a pressure rise across the stage. When discussing velocity changes through a stage it is easier to represent that change by a velocity diagram. The velocities within a compressor can be referenced from two frames: the rotor frame, which defines the relative velocity ( $\vec{w}$ ), and the stationary compressor housing frame, which defines the absolute velocity ( $\vec{c}$ ). The vectorial difference between the relative and absolute velocities is the tangential rotor speed ( $\vec{U}$ ):

$$\vec{U} = \vec{c} - \vec{w} \quad (1)$$

Together the velocities represent a velocity triangle and can describe the flow velocities and flow angles entering and leaving a compressor blade row (Figure 1). In a simple mean line analysis of the velocity

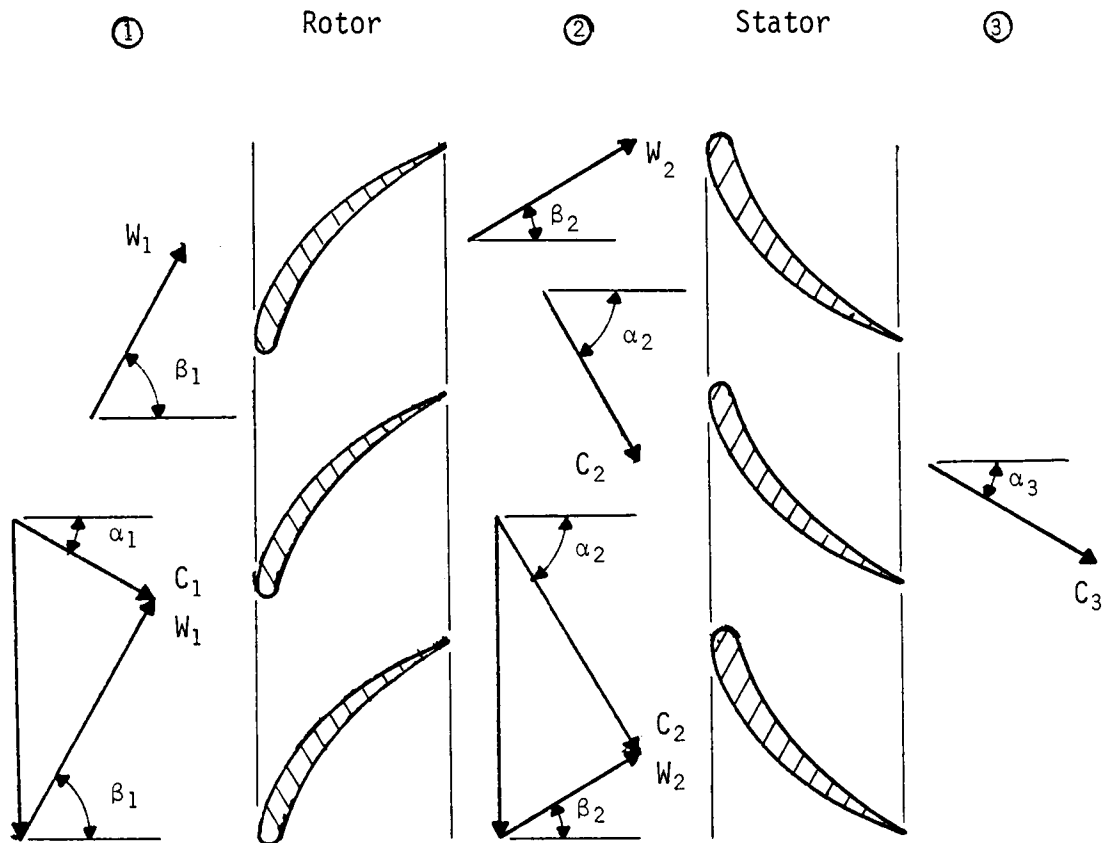


Figure 1. Mean radius section of a compressor stage.

triangles before and after the rotor row, the rotational speed component is considered constant and the axial velocity component can be considered relatively constant. The two triangles can then be compared to visualize a change in the absolute tangential velocity ( $\Delta c_\theta$ ) (Figure 2). The change in absolute tangential velocity represents the amount of flow turning in a blade row and is related to the amount of torque and power delivered by that blade row.

From the conservation of angular momentum the torque acting on the air from the blade row must be equal to the change in angular momentum.

$$r = \dot{m}[(rc_\theta)_b - (rc_\theta)_a] \quad (2)$$

The power of the shaft delivered to the stage can be represented by the product of the torque and angular velocity of the compressor:

$$P_s = r\Omega \quad (3)$$

or,

$$P_s = -\dot{m}\Omega[(rc_\theta)_b - (rc_\theta)_a] \quad (4)$$

The minus sign signifies that the power output of the compressor is negative, i.e., power is delivered to the compressor.

Neglecting body forces and assuming adiabatic flow through the stage the conservation of energy requires that:

$$\dot{Q} = \dot{m}(h_{tb} - h_{ta}) + P_s = 0 \quad (5)$$

$$c_p(T_{tb} - T_{ta}) = \Omega[(rc_\theta)_b - (rc_\theta)_a]$$

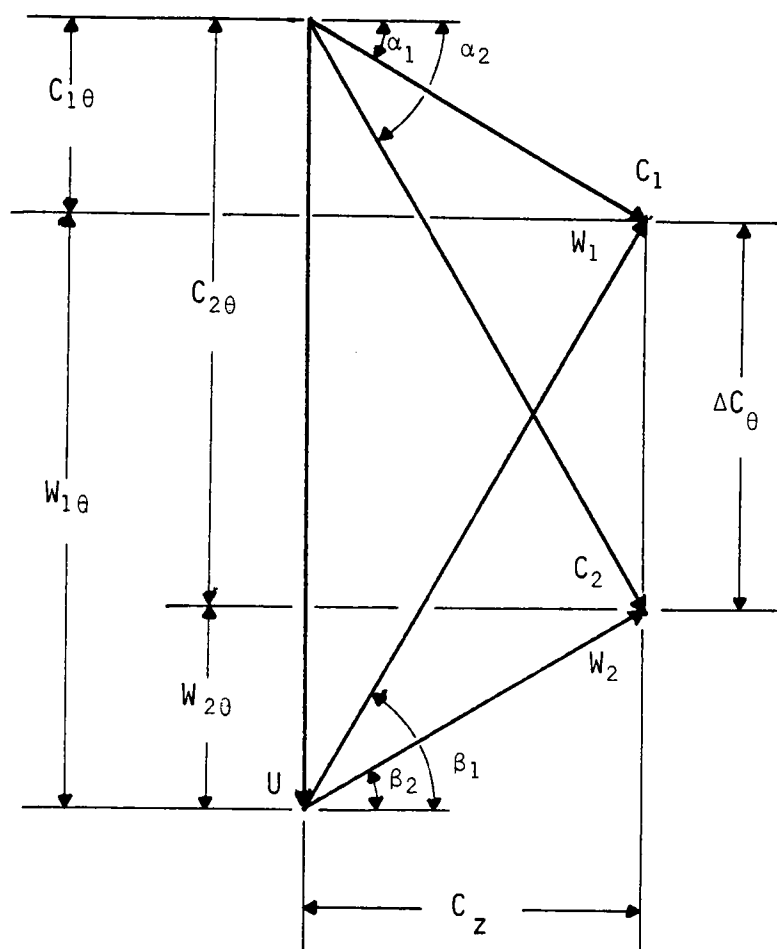


Figure 2. Velocity triangles for a compressor stage.

Applying equation (4), equation (5) becomes:

$$h_{tb} - h_{ta} = \Omega[(rc_\theta)_b - (rc_\theta)_a] \quad (6)$$

Assuming specific heat at constant pressure is constant through the stage,

$$c_p(T_{tb} - T_{ta}) = \Omega[(rc_\theta)_b - (rc_\theta)_a] \quad (7)$$

which correlates a change in total temperature to a change in absolute tangential velocity through the stage.

This process can be shown on a temperature-entropy diagram as an isentropic rise in total temperature. But due to losses within the blade row the real process is not isentropic and results in a loss of total pressure (Figure 3). If a stage efficiency is defined as:

$$\eta_{st} = \frac{h_{tb}' - h_{ta}}{h_{tb} - h_{ta}} \approx \frac{T_{tb}' - T_{ta}}{T_{tb} - T_{ta}} \quad (8)$$

then by using the isentropic flow relation, the total pressure rise can be related to the total temperature rise across the stage by the stage efficiency (the approximate relation between enthalpy and temperature definitions is due to minor specific heat variations through the stage). Hence:

$$\frac{p_{tb}}{p_{ta}} = \left(1 + \eta_{st} \frac{\Delta T_t}{T_{ta}}\right)^{\frac{\gamma}{\gamma-1}} \quad (9)$$

Knowing the amount of flow turning in a blade row for an ideal stage, the pressure rise for a single stage can be determined from the temperature rise, and vice versa.



The operation of the entire compressor can similarly be shown on a temperature-enthalpy diagram where the isentropic and actual processes between the inlet and exit can be represented (Figure 4). The adiabatic efficiency of a compressor can be defined as follows, "The ratio of the work required in an isentropic process to that required in the actual process, for the same stagnation pressure ratio and inlet state: [1],

$$\eta_c = \frac{h_{te}' - h_{ti}}{h_{te} - h_{ti}} \approx \frac{T_{te}' - T_{ti}}{T_{te} - T_{ti}} \quad (10)$$

Using the isentropic flow relations the total pressure rise across the compressor can be determined:

$$\frac{p_{te}}{p_{ti}} = \left(1 + \eta_c \frac{\Delta T_t}{T_{ti}}\right)^{\frac{\gamma}{\gamma-1}} \quad (11)$$

which relates the pressure and temperature rise through the entire compressor.

#### Axial Flow Compressor Stability

The operation of a compressor can be depicted on a component map called the compressor map (Figure 5). This map shows the relationship of the compressor total pressure ratio to mass flow, rotational speed, and compressor adiabatic efficiency. The operating range of a compressor is bounded by aerodynamic and mechanical limitations. It is limited mechanically by its rotational speed. High rotational speed can cause damaging high centrifugal forces. The other limitation is due to aerodynamic and aeroelastic instabilities of the blades

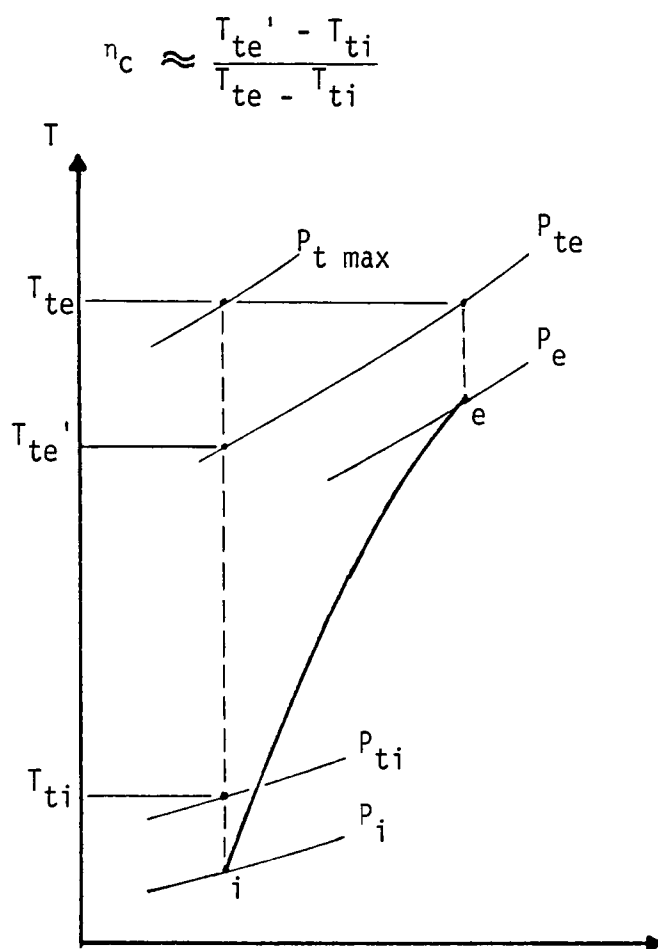


Figure 4. Overall compressor compression process.

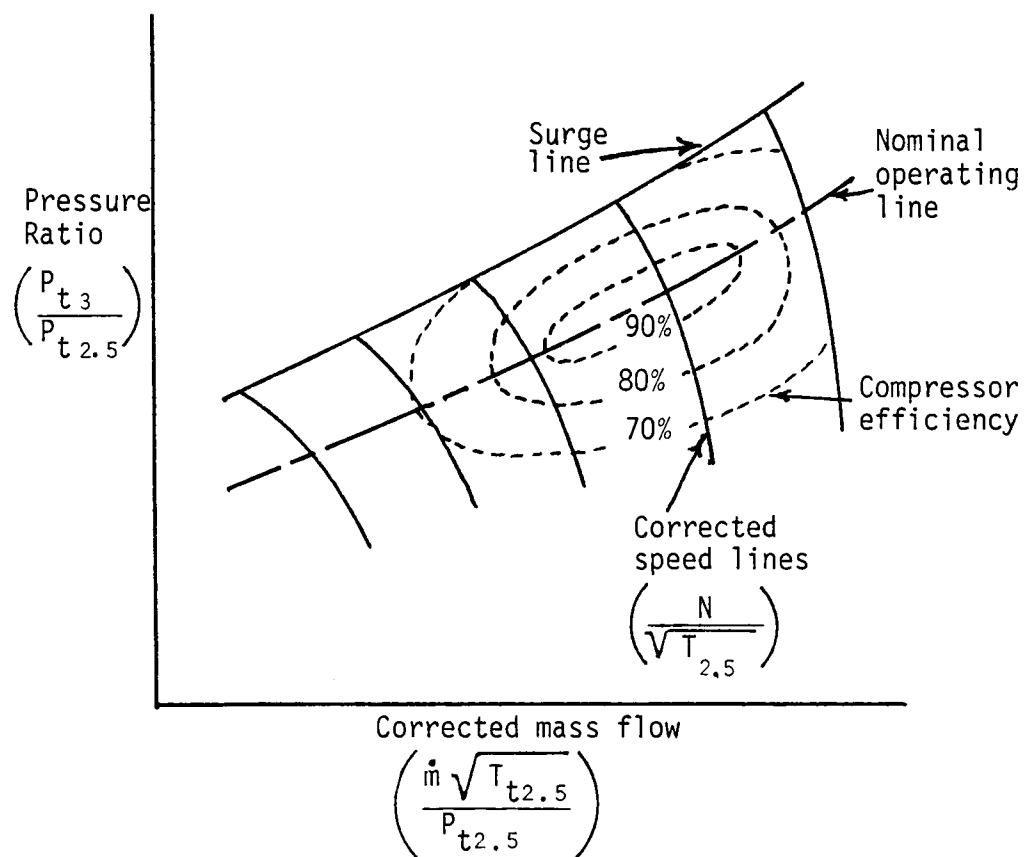


Figure 5. Typical compressor map.

themselves. Aerodynamic instabilities manifest themselves in several ways, and can be categorized as either stall or surge. A stall occurs as a breakdown or separation of the flow over an airfoil usually due to a high angle of attack. A compressor blade row contains many airfoils so the occurrence of stall will affect several of these airfoils and its behavior will be very complex. On the compressor map the upper region is bounded by a stall or surge limit line. Operation beyond this line is characterized by large oscillations in the mass flow called surge or severe self-induced flow distortions that rotate around the annulus called rotating stall [2].

Rotating stall is characterized as a stall cell or group of stall cells that rotate around the annulus of the compressor in the direction of rotation and at about half the rotational speed. Rotating stall usually begins as a stall of a single blade (Figure 6). As the single blade stalls, flow is diverted around the stalled blade passage. The blade below the stalled blade receives a lower angle of attack and the blade above the stalled blade receives a higher angle of attack. The blade that receives a higher angle of attack will eventually stall and the flow will then be diverted around that blade passage. The blade that originally stalled will unstall since the diverted flow from the stalled blade above will reduce its angle of attack, and the process will repeat itself as the stall cell advances up the blade row. The above description discusses the stall cell as a single blade passage but the stall cell can grow to envelope half of the flow annulus or more. Under rotating stall conditions the compressor will operate in a semi-steady condition even though the mass flow in the stall cell

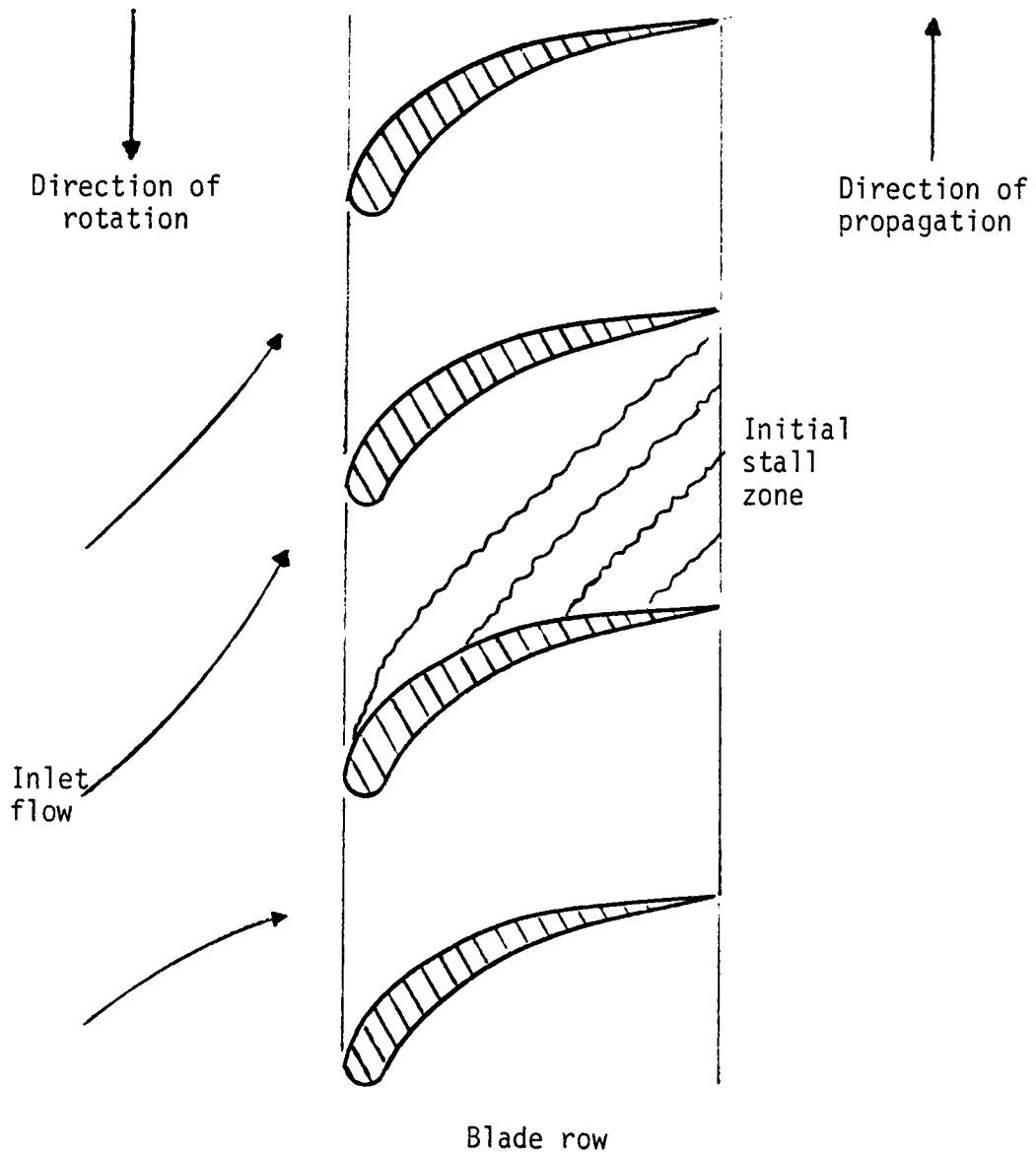


Figure 6. Rotating stall propagation.

is considerably less or even reversed. Rotating stall has many detrimental effects. First, it greatly reduces the performance of the compressor. It also can cause structural damage to the blades due to the bending and unbending of the blade as the stall cell passes. While the compressor is in a rotating stall mode the increases in the fuel-air ratio from the reduction in mass flow can cause temperature levels in the turbine high enough to damage the turbine blades.

During a surge cycle the mass flow and pressure ratio undergo large oscillations even to the extent that the flow within the entire compressor reverses. These large oscillations can induce extreme structural damage to the blades and the casing of the compressor similar to the rotating stall oscillations. But, the frequency of the surge cycle is usually an order of magnitude less than that of the rotating stall.

In the design of a compressor the designer chooses a set of steady state operating points correlating to various rotational speeds that will define an operating line on the compressor map. The operating line is affected by the entire engine but the stall line is determined primarily by the compressor. The margin between this steady state operating line and the stall is termed the stall margin. Compressor designers go to great lengths to increase the stall margin which ensures safe operation. However, this margin can be affected many ways: inlet distortion, Reynolds Number effect, control tolerances, transients, and the subject of this investigation, thermal transients (Figure 7).

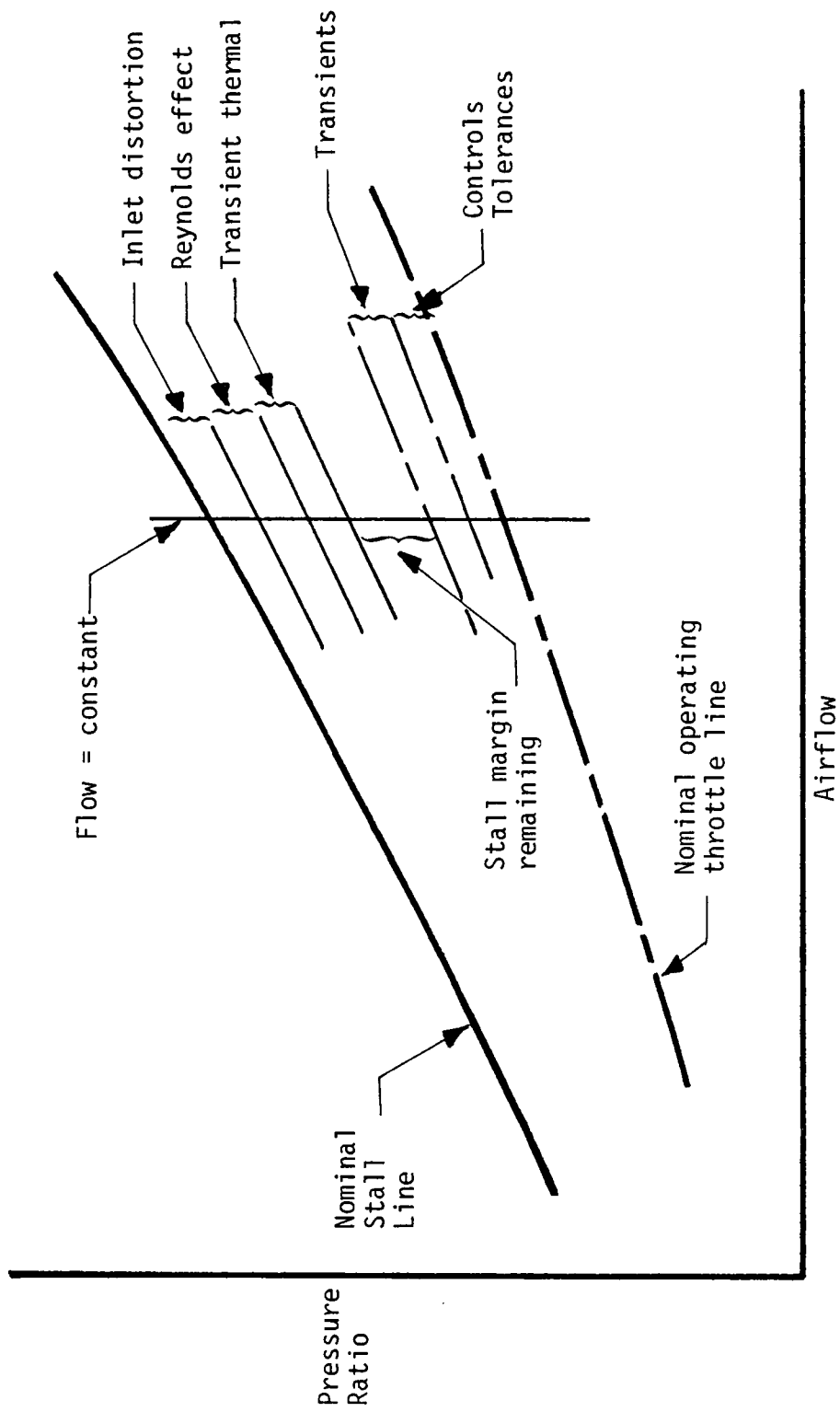


Figure 7. Stability margin audit.

### Transient Compressor Operation

Transient operation is a general term associated with compressor behavior as a function of time. Two transients that are of great interest are accelerations and decelerations, or more specifically, "slam" accelerations and "slam" decelerations. The term "slam" refers to a sudden throttle or, in engine testing vernacular, power lever angle (PLA) movement. For instance, a "slam" acceleration is a sudden throttle movement from an idle level to an intermediate or maximum level and vice versa for a "slam" deceleration. The path the transient operating line takes during a speed transient depends on the nature of the transient. During a deceleration the path of the transient operating line tends to fall below the path of the steady-state operating line on the compressor map (Figure 8). At the initiation of a deceleration the transient line path drops, nearly tracing a speed line. This is due to a fast reduction in compressor's pressure ratio and a slow reduction in rotational speed. When the throttle is cut back the amount of fuel going to the combustor is reduced. This decreases the temperature in the combustor and causes the specific volume of the gas within the combustor to decrease. The static pressure in the combustor decreases with the specific volume of the gas which will unload the compressor and reduce the pressure ratio. This process occurs rapidly, while the reduction in rotational speed is restrained by the momentum of the rotor. As the pressure ratio and rotational speed continue to drop, the transient line begins to seek a new steady-state point corresponding to a new throttle position. For an acceleration the transient

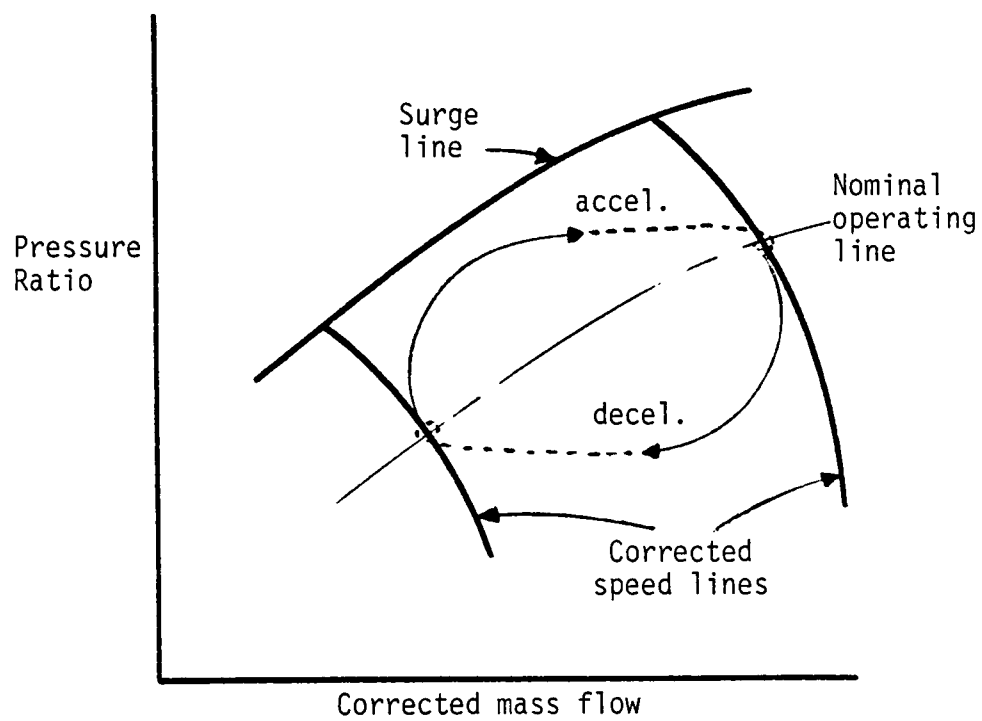


Figure 8. Operating lines during speed transients.

line tends to rise above the steady-state line (Figure 8). Again, this is due to the faster response of the pressure ratio to the speed transient, only this shift in the operation line causes a reduction in the surge margin and drives the compressor towards surge.

### Thermal Transients

Several things occur during speed transients. During an acceleration, part of the turbine power is lost to speed up the shaft rotation, and during a deceleration the rotating shaft delivers power. The mass flow entering and leaving the compressor may be different due to pressure gradients, temperature gradients, and temperature storage. The compressor may also be considered non-adiabatic during a transient due to heat transfer between the airflow and metal parts [3]. During an acceleration, the increase in pressure ratio results in a corresponding increase in temperature ratio. This process will cause a difference in temperature between the air and the compressor metal, creating a driving force for heat transfer. During a deceleration the same thing occurs only opposite. The reduction in pressure ratio causes a reduction in air temperature creating a difference in temperatures between the air and the metal parts. This difference in temperatures causes heat to be transferred from the metal to the air.

Therefore, speed transients cause thermal transients. Not only will the surge margin be affected by the shift in the transient operating line, but it will also be affected by the influence of heat transfer. The problem of surge margin reduction is then two-fold for speed transients.

## CHAPTER II

### LITERATURE SEARCH

#### Sources

The non-adiabatic effect during speed transients is of concern and interest to compressor stability and has not been thoroughly investigated. An extensive literature search was conducted through the University of Tennessee Space Institute (UTSI) research library, which included domestic as well as foreign sources. The search was directed at the topic of the effect of transient heat transfer on compressor stability. Considering the extent of the search the number of papers found were relatively few, and all of the papers were of foreign origin. This fact revealed that although the subject of thermal transients within compression systems is widely known the amount of written literature, analytic work and experimental data is very limited.

#### Summary

Three significant papers were found and reviewed. They all discussed the effects of transient heat transfer on the operation of axial flow compressors. The effects resulted in either an increase or decrease in the surge margin, depending on the type of transient. An acceleration caused an increase and a deceleration caused a decrease in the surge margin. The papers were authored or co-authored by the same person. Therefore, the prediction methods in the three papers were

similar in analysis procedure but unique in the assumptions taken and in the application of heat transfer effects.

The analysis procedure was a simple stage-stacking model similar to the one described by Huppert and Bensor [4], and it was applied to a two-spool turbofan of pressure ratio 20. This method utilizes a "mean line" or "pitch line" analysis. If the hub-to-tip ratio of a blade row is relatively high (close to unity) the flow can be considered two-dimensional. That is, there is no significant variation in the pressure ratio and blade speed from the root to the tip of the blade. The performance of the entire blade row can then be calculated at the mean or pitch line. Each successive blade row can be correspondingly analyzed and "stacked" together to represent a complete compressor.

Maccallum [5] utilized this model to study the effect of "bulk" heat transfer on compressor surge margins. "Bulk" heat transfer manifests itself as a general reduction in the air density due to temperature changes produced by thermal transients. By inserting temperature changes between the stages Maccallum was able to account for the "bulk" heat transfer effects during a typical transient. The transients investigated were an acceleration at sea level and three decelerations at sea level and altitude conditions. For the case of the acceleration it was determined that the heat transfer reached a maximum value midway through the transient. Therefore, conditions were examined at that point. For the deceleration it was determined that the thermal effects were at a maximum value at the end of the transient. Hence, conditions were investigated at that point. The surge point was determined from

compressor stage characteristics in the form of pressure rise coefficients and efficiencies plotted against a flow coefficient. The point of maximum pressure rise coefficient was said to be the point at which stall occurs. At low rotational speeds surge is associated with stall in the first stages and at high rotational speeds surge is associated with stall in the latter stages. From the definition of non-dimensional speed ( $\frac{N}{\sqrt{T}}$ ) the change in the temperature due to heat transfer causes the speed line to shift. MacCallum attributes the shift in the steady running lines to heat transfer effects in the low pressure turbine (LPT). The changes in temperature due to heat transfer causes the turbine nozzles to choke, from thermal expansion, at a different flow rate. This alters the pressure ratio as well as the mass flow rate through the LPT, and affects the steady running line of the compressor.

During sea level accelerations the surge margin was increased in the high pressure compressor (HPC) and low pressure compressor (LPC). In the HPC there was a notable increase in the non-dimensional speed line and nominal reduction in the steady running line. In the LPC there was only a nominal movement in the running line (Figure 9). In two cases of deceleration for sea level and altitude conditions there was a shift in the surge line, speed line, and running line for the HPC and a shift only in the running line for the LPC (Figures 10 and 11). The combined effect of the shift in the surge, speed, and running lines was a 35% reduction in surge margin for the altitude deceleration and a 32% reduction in surge margin for the sea level deceleration for the HPC. For the LPC there was an 18% and 8% decrease in the surge margin for altitude and sea level deceleration, respectively. These combined

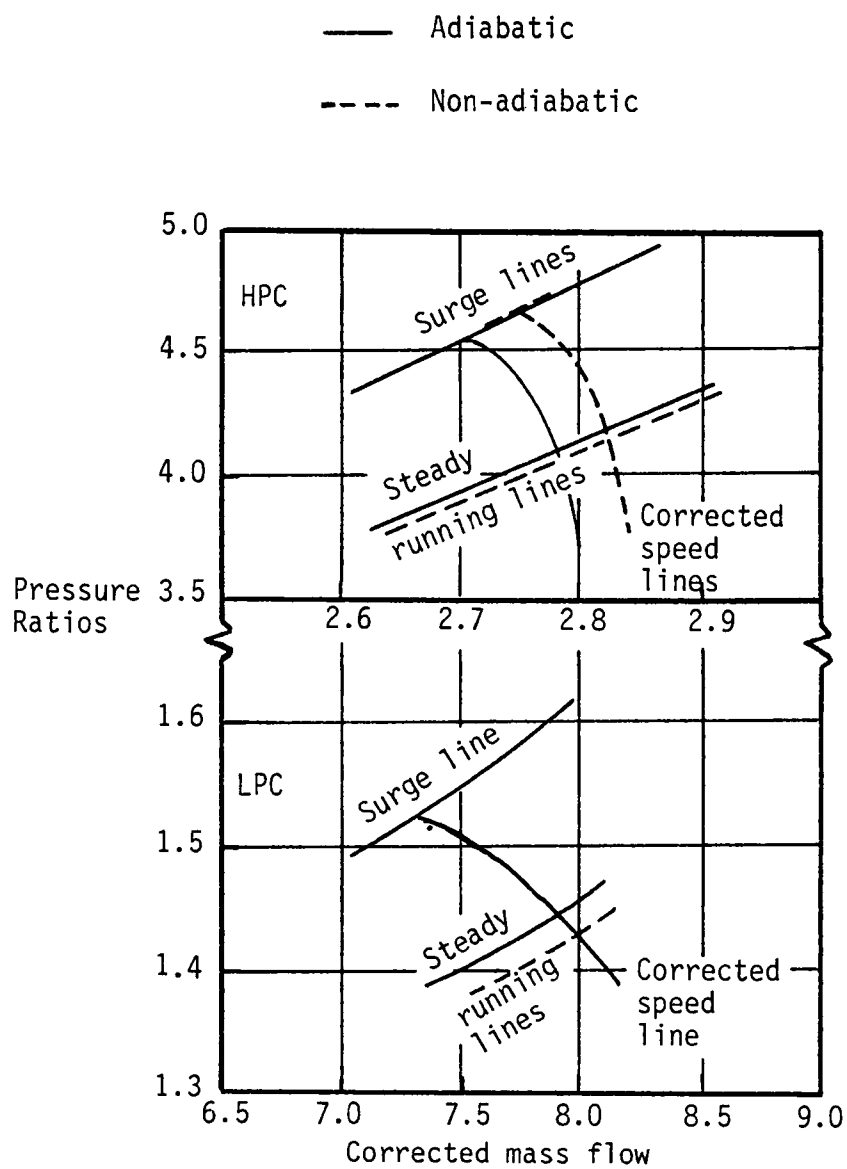


Figure 9. Effects of heat transfer midway through sea level acceleration.

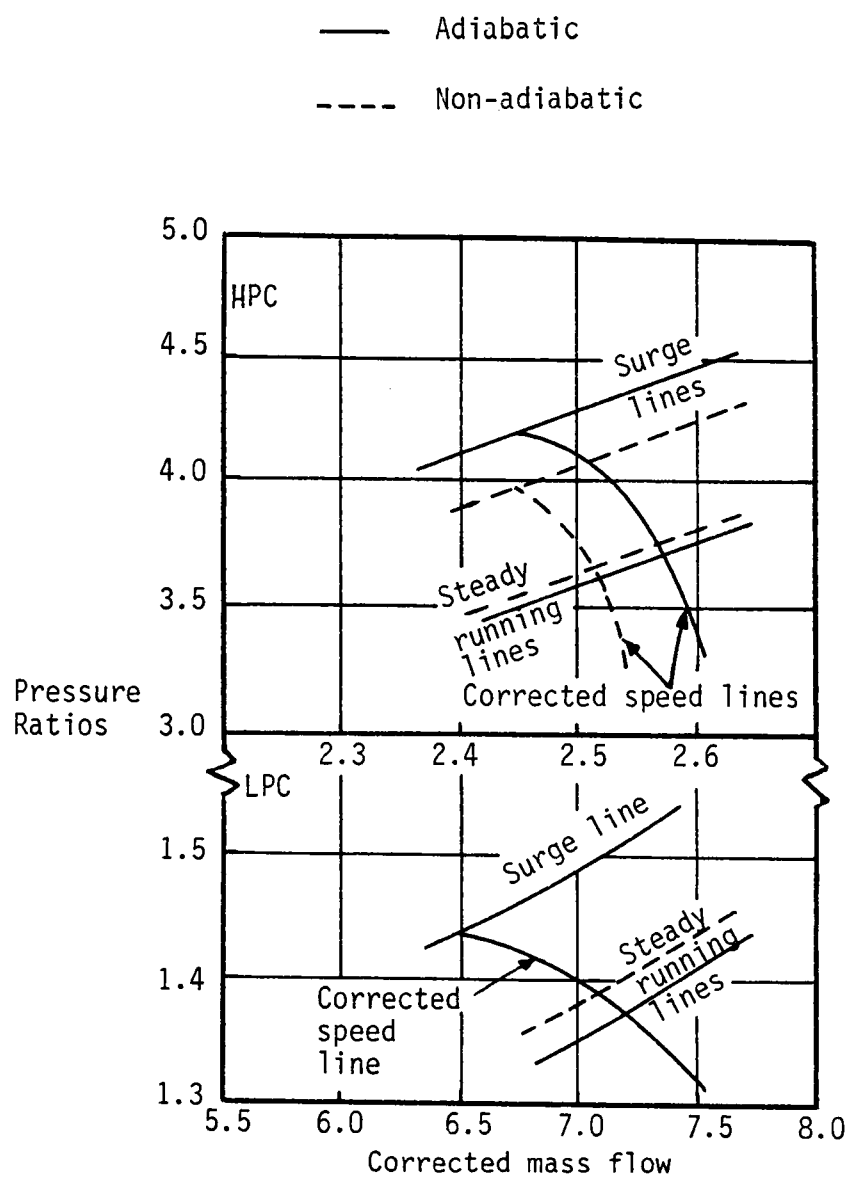


Figure 10. MacCallum's effects of heat transfer at end of altitude deceleration.

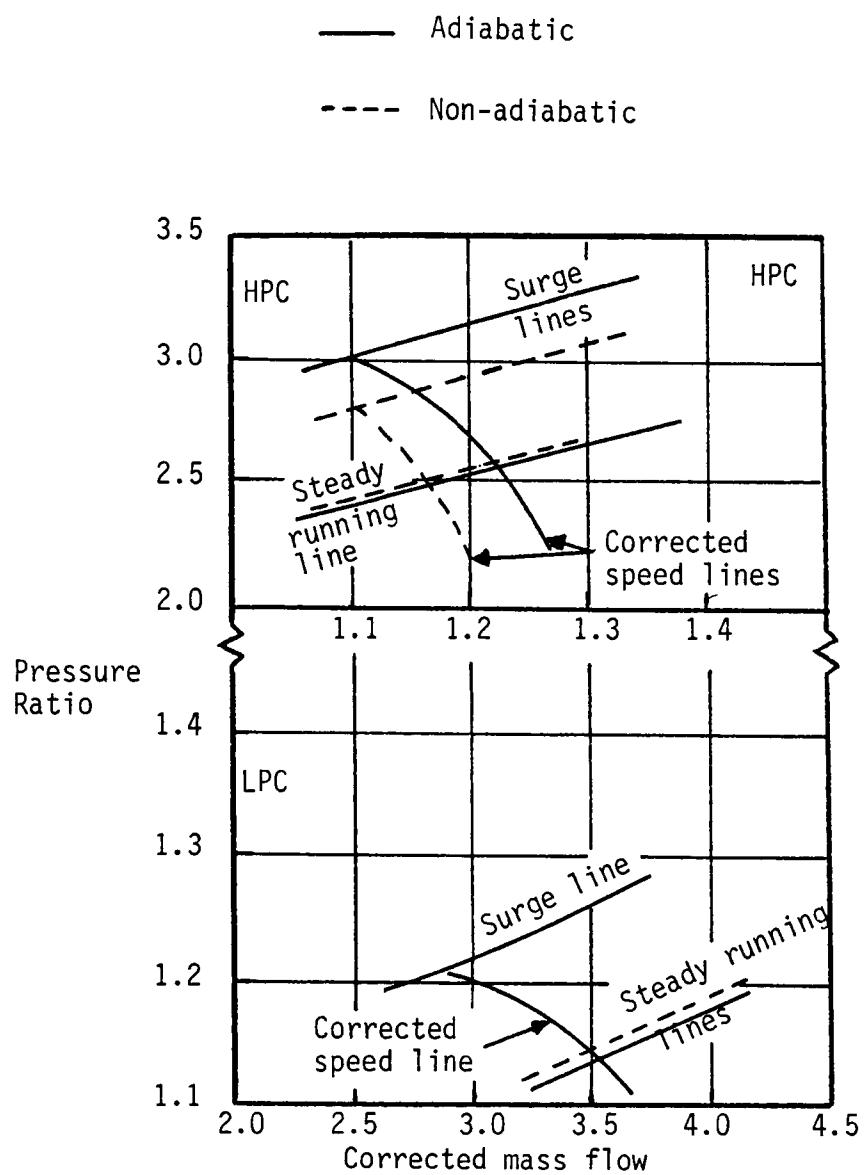


Figure 11. Effects of heat transfer at end of sea level deceleration.

effects are relatively important for deceleration alone, but when an acceleration immediately follows a deceleration the reduction in surge margin becomes very important.

A second paper by Maccallum and Grant [6] used a similar stage stacking routine to show the effect of boundary layer changes due to transient heat transfer on axial flow compressor performance. The transient studied was a rapid altitude deceleration which produced heat transfer from the compressor metal to the airstream. The model included two-dimensional cascade data to represent compressor blade operation. The effects of heat transfer on boundary layer development were first theoretically analyzed, then were incorporated into the prediction method. The prediction method was used to predict the changes in the compressor characteristics that described the compressor operation. These changes in the compressor characteristics were then incorporated into the model. The model also included "bulk" heat transfer effects similar to the previous paper. From previous work it was determined that heat transfer will affect the HPC more than the LPC, therefore only the HPC was investigated.

The effects of heat transfer on the boundary layer are important. Based on a boundary layer analysis using the momentum integral equation the influence of a reduction in density increases the rate at which the boundary layer develops. The thicker boundary layer influences the flow over both the annulus and blades. On the annulus the thicker boundary layer creates extra losses due to a constriction of the flow area. On the blades the shape of the boundary layer profile and presence of pressure gradients affect each side of the blade differently

(Figure 12). On the pressure side, the concave shape and the relatively small adverse pressure gradient cause little effect on the boundary layer growth due to heat transfer and can be neglected. But on the suction side the convex shape of the blade and large adverse pressure gradient influence the boundary layer immensely. First, there is an increased profile drag of the blade. Then there is a thicker "hot" wake created by the blade. There also is a deviation in the flow turning angle of the blade. This deviation is caused by an upstream movement of the flow departure point on the blade. In general, these effects cause a loss in compressor performance and a tendency to stall at a lower pressure ratio. All these effects were incorporated into the prediction method.

Maccallum's model indicated a 40% reduction in the surge margin due to boundary layer effects only and an additional 25% reduction due to "bulk" heat transfer effects (Figure 13). This amounts to a total of a 65% reduction in the surge margin, and a better chance of encountering unstable operation.

A third paper by Maccallum [7] investigated how axial compressor characteristics may be affected by thermal transient operation. Two methods were employed and compared in this investigation: a pitch line method with stall cells, and a pitch line method with end-wall boundary layer effects. The first method was similar to the one used in the first two papers. The surge point for the non-adiabatic case was determined from the adiabatic case. The adiabatic surge point was defined as the maximum pressure ratio point on a stage characteristics curve. The non-adiabatic surge point was defined as a point where a group of

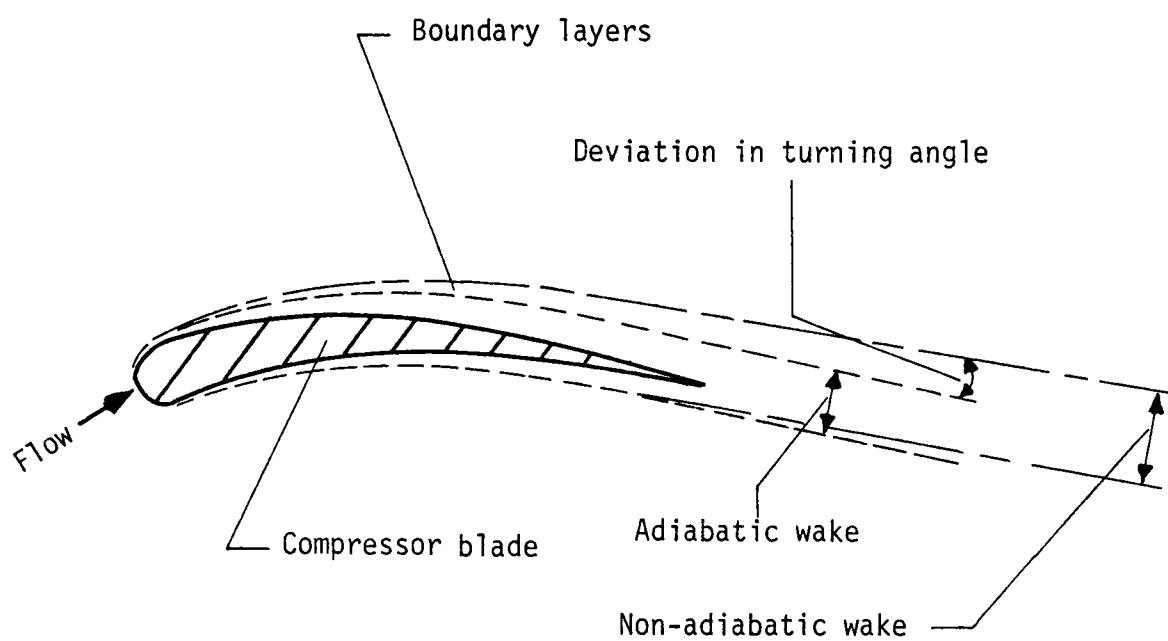


Figure 12. Effect of heat transfer on blade boundary layer.

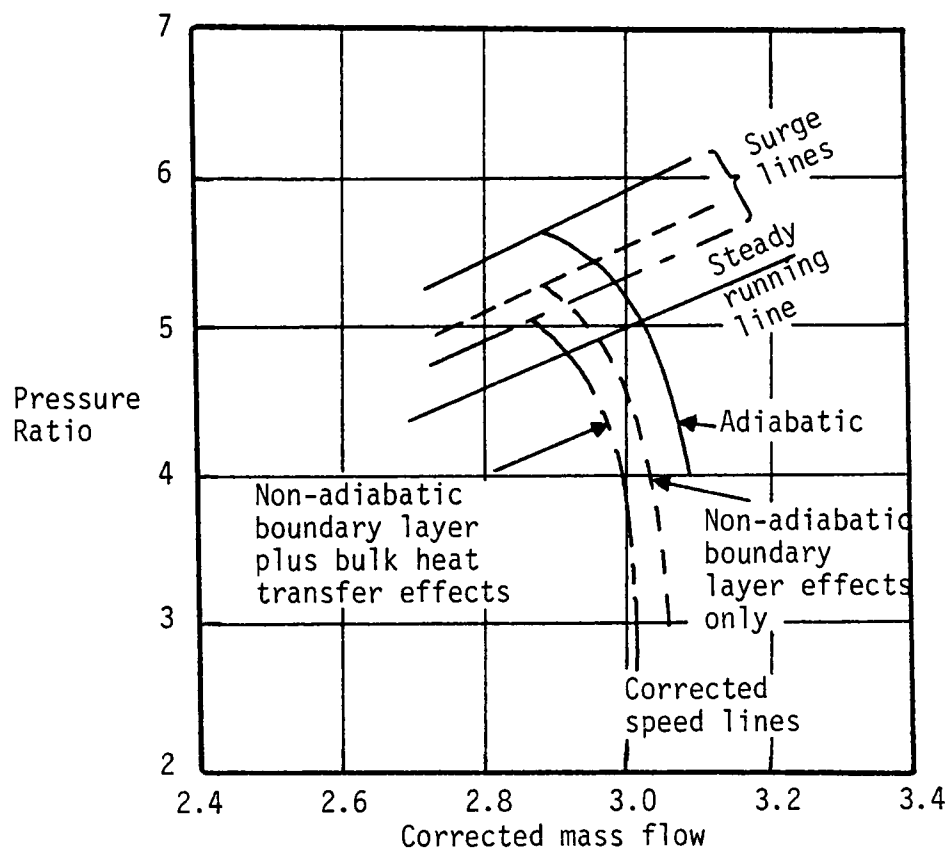


Figure 13. Effects of heat transfer at end of altitude deceleration.

the four most heavily loaded stages have an average axial velocity to blade speed ratio equal to that of the same four stages when at surge in the adiabatic case. The four heavily loaded stages constituted a stall cell. The second method considered the effect of end-wall boundary layer thickness on the effective flow area, and the resulting effect on the stage performance. As previously described, the growth of the boundary layer at the end-wall (annulus) caused a reduction of the flow area. This changes the flow rate in the compressor and also the velocity of the flow over the blades. The surge point definition in this method is identical to the surge point definition in the previous method. These boundary layer effects were introduced into the second method.

The results of the first method for the case of an altitude deceleration was a 50-60% reduction in surge margin due to density and blade boundary layer changes (Figure 14). The results of the second method for the same transient was a less severe reduction in the surge margin (20-25%) (Figure 15). The second method was considered to be the more accurate of the two methods. It combined end-wall effects and blade boundary layer effects, which better described the three-dimensional flow in the blade passage.

### Conclusions

All the papers reviewed predicted similar results: a shift in the surge line, speed line, and running line of a compressor for different transients. The worst case of the transients evaluated was an altitude deceleration. In one paper a 50-60% reduction in the surge margin

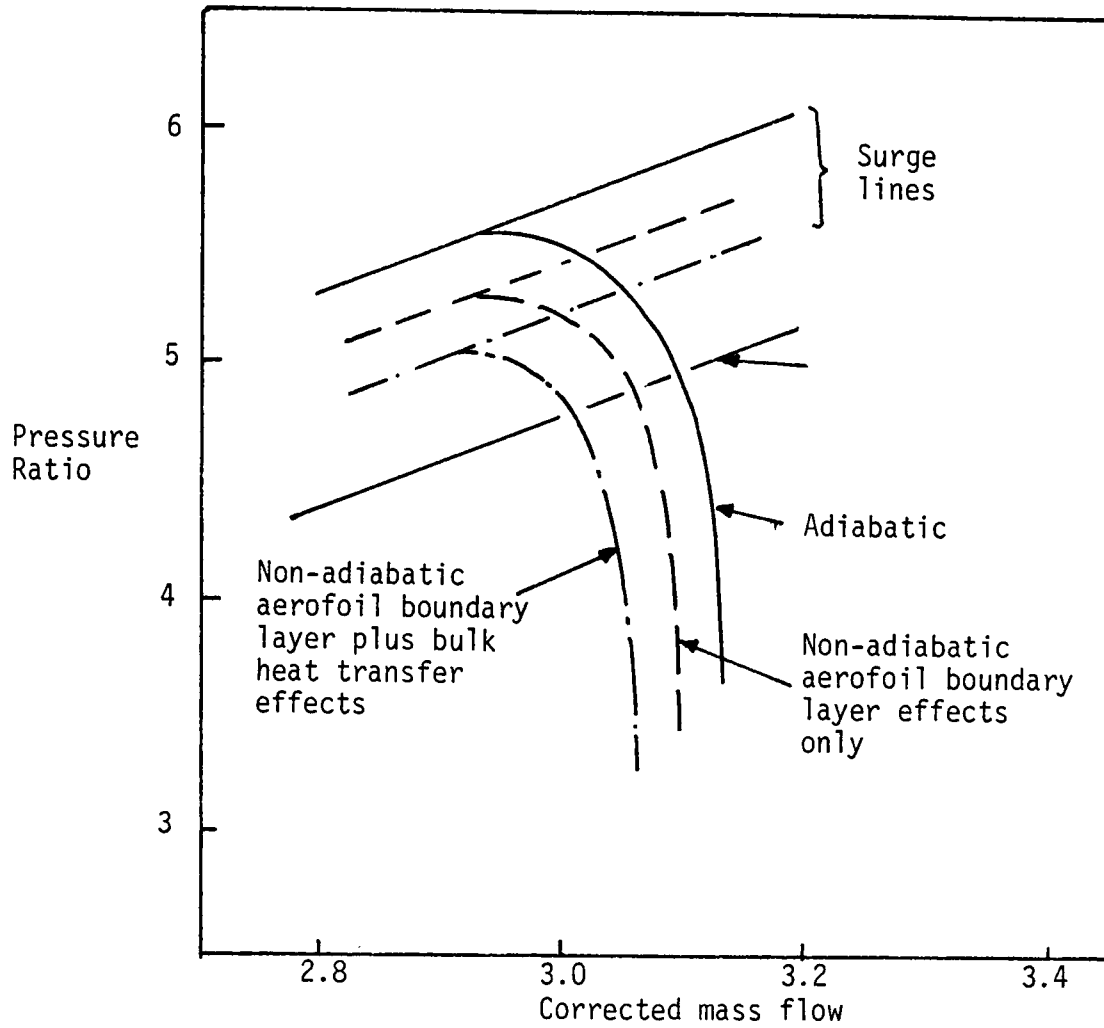


Figure 14. Effects of heat transfer at end of altitude deceleration--Method 1.

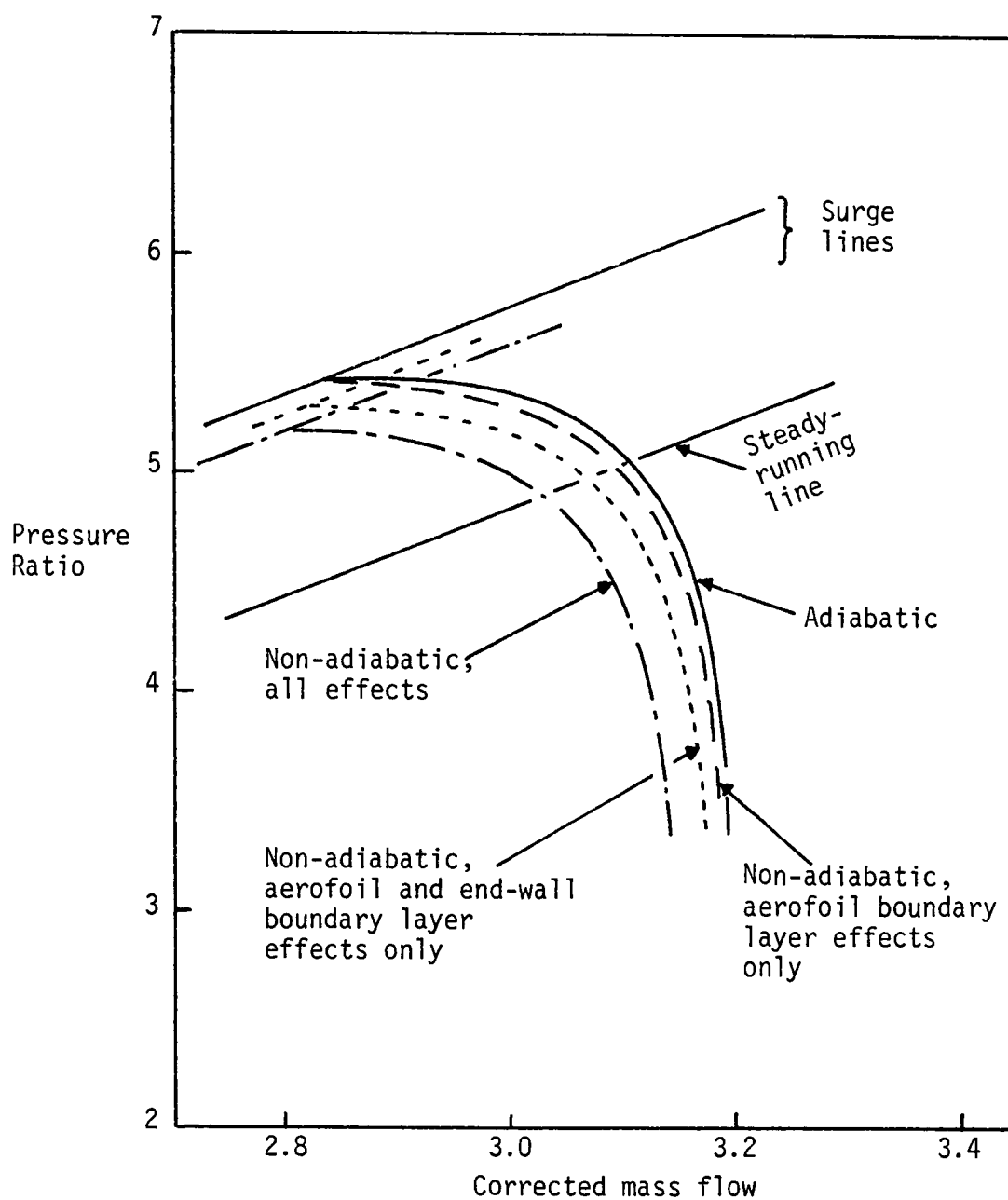


Figure 15. Effects of heat transfer at end of altitude deceleration--Method 2.

was predicted. In a more conservative approach the shift in the surge margin was 20-25%. Still a relatively large value. Thus transient heat transfer can be a significant influence on stable compressor operation during speed transients.

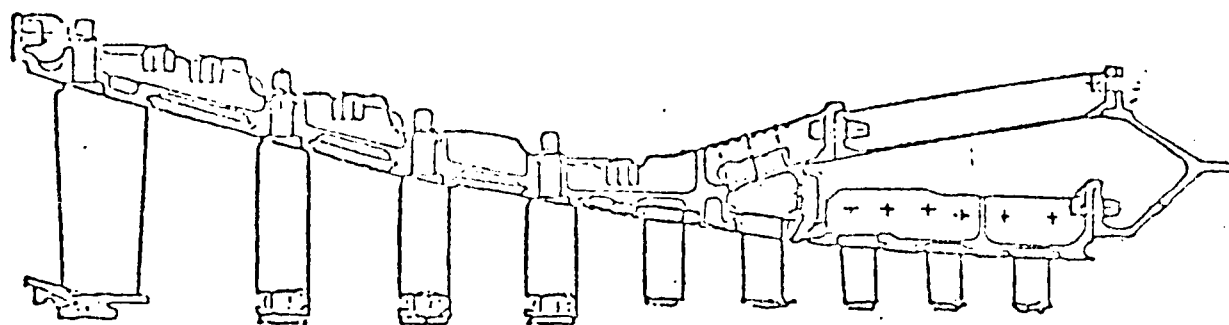
The information presented here represents analytical data based on stage-stacking model results and cascade characteristics data. The results to this point have been purely theoretical. No real engine data has yet been investigated. By analyzing real engine data actual results can be determined and a correlation between the theoretical and actual results can be made.

## CHAPTER III

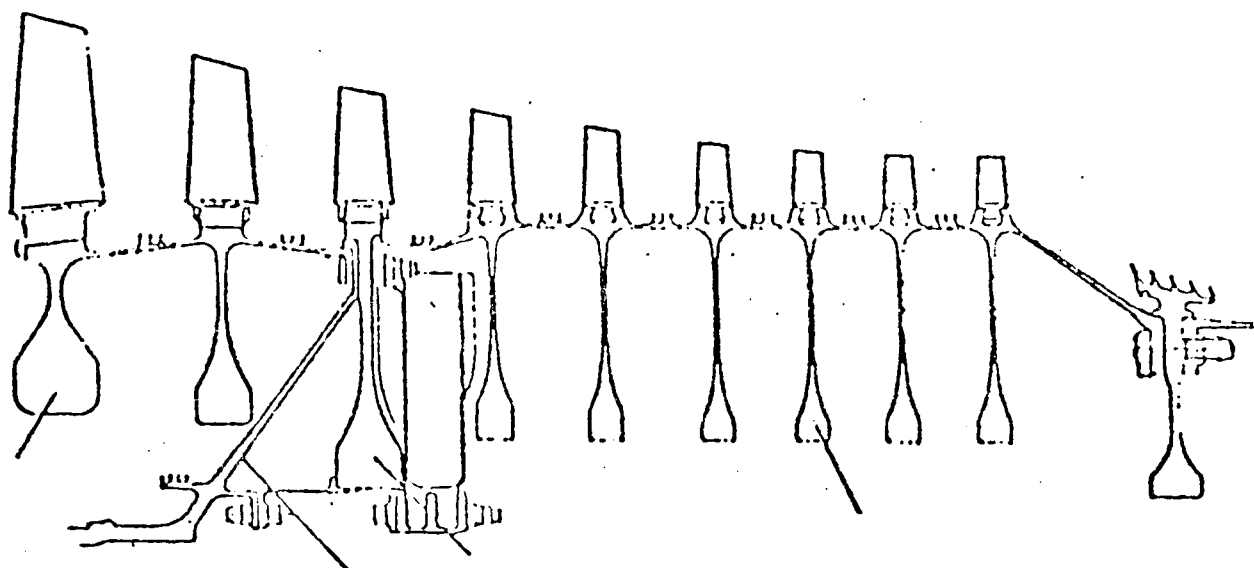
### QUANTITATIVE EVALUATION OF STORED THERMAL ENERGY

#### Compressor Metal Parts

When a compressor is operating at a maximum power level setting the temperature of the metal parts within the compressor that are in contact with the air flow or in the near vicinity will stabilize at the adiabatic wall recovery temperature of the flow. As the compressor rotational speed decreases during a deceleration the heated metal parts will release their stored thermal energy back into the air flow which is now cooler than the metal parts. The amount of thermal energy released and the time taken to release it depends on the location of the part and the part itself. The rotor and stator blades and their roots are in direct contact with the air flow and can be thought of as extended fins on a platform. These extended fins are subject to forced convection and will release their thermal energy relatively quickly. The metal of the annulus, also in contact with the air flow, contains more metal than the blades but has less metal in contact with the flow and is under different forced convection conditions. So its thermal energy release will be slower but will last longer (Figure 16). But, as explained in the literature research, the thermal energy release from the annulus will be contained within the boundary layer and will only cause an increase in its thickness possibly causing flow blockage



Stator Blades and Case



Rotor Blades, Disks and Seals

Figure 16. Typical high pressure ratio compressor rotor-stator design.

in the latter stages. Therefore, the thermal energy release of the blades is more important when regarding the flow separation of the blades.

#### Compressor Internal Temperature Distribution

The stored thermal energy in each blade row that can be released during a transient is found from the difference in thermal energy levels corresponding to a high and low power level setting during a deceleration. From sample test data the difference in temperatures for the inlet and exit of a HPC were determined for two different power level settings. The temperatures at the inlet and exit for a typical low power level at nominal conditions (engine inlet mach number of 0.8 and altitude of 20K) were about 75°F and 430°F, respectively. For a typical high power level at nominal conditions the temperatures at the inlet and exit were about 225°F and 900°F, respectively. This gives a typical temperature variation at the inlet of about 150°F and the exit of about 470°F. Assuming that the pressure rise per stage is approximately equal, the temperature rise per stage will also be approximately equal. Hence, a temperature difference corresponding to a high and low power level setting can be determined for each stage (Table 1).

#### Stored Thermal Energy Analysis

From sample engine geometry data concerning this investigation obtained from Arnold Engineering Development Center (AEDC), a typical compressor was modeled to evaluate the stored thermal energy. Using a 10-stage compressor design with inlet guide vanes the weights of the blades and blade roots were determined from analysis of the available

Table 1. Temperature difference corresponding to a high and low power level setting.

| Station       | $\Delta T_{\text{idle} - \text{max}} (^{\circ}\text{F})$ |
|---------------|----------------------------------------------------------|
| IGV           | 150                                                      |
| Rotor 1 (R1)  | 166                                                      |
| Stator 1 (S1) | 182                                                      |
| R2            | 198                                                      |
| S2            | 214                                                      |
| R3            | 230                                                      |
| S3            | 246                                                      |
| R4            | 262                                                      |
| S4            | 278                                                      |
| R5            | 294                                                      |
| S5            | 310                                                      |
| R6            | 326                                                      |
| S6            | 342                                                      |
| R7            | 358                                                      |
| S7            | 374                                                      |
| R8            | 390                                                      |
| S8            | 406                                                      |
| R9            | 422                                                      |
| S9            | 438                                                      |
| R10           | 454                                                      |
| S10           | 470                                                      |

data (Table 2). The stored thermal energy to be released by a particular blade row during a transient can be evaluated from the following equation:

$$Q_{\text{per stage}} = mc_p(T_{\text{high}} - T_{\text{low}}) \quad (12)$$

where  $m$  represents the mass of the blades and the blade roots. From the mass data and typical values of specific heats corresponding to blade and blade root material the amount of stored thermal energy that can be released for this standard transient can be determined (Table 3). The summation of the thermal energy released per stage amounts to quite an impressive number (about 2500 Btu), which could account for the occurrence of separation along the blade.

In this analysis only the blade and their roots were considered. Obviously, the other metal parts such as the casings and discs will also release their stored thermal energy into the airstream. But, as explained previously, it will be released slowly, a greater part being conducted through the blades. These effects were omitted since it would be difficult to estimate their heat transfer values accurately.

Table 2. Weights for blades and blade roots.

| Station       | (lbm)<br>Weight/Item | Number/Engine | (lbm)<br>Weight/Engine |
|---------------|----------------------|---------------|------------------------|
| IGV           | 0.04                 | 90            | 3.6                    |
| Rotor 1 (R1)  | 0.10                 | 50            | 5.0                    |
| Stator 1 (S1) | 0.15                 | 65            | 9.75                   |
| R2            | 0.05                 | 70            | 3.5                    |
| S2            | 0.13                 | 60            | 7.8                    |
| R3            | 0.03                 | 85            | 2.55                   |
| S3            | 0.06                 | 75            | 4.5                    |
| R4            | 0.03                 | 90            | 2.7                    |
| S4            | 0.04                 | 80            | 3.2                    |
| R5            | 0.03                 | 95            | 2.85                   |
| S5            | 0.02                 | 100           | 2.0                    |
| R6            | 0.03                 | 95            | 2.85                   |
| S6            | 0.02                 | 105           | 2.1                    |
| R7            | 0.03                 | 90            | 2.7                    |
| S7            | 0.01                 | 120           | 1.2                    |
| R8            | 0.03                 | 80            | 2.4                    |
| S8            | 0.01                 | 130           | 1.3                    |
| R9            | 0.03                 | 90            | 2.7                    |
| S9            | 0.01                 | 135           | 1.35                   |
| R10           | 0.02                 | 95            | 1.9                    |
| S10           | 0.02                 | 95            | 1.9                    |

Table 3. Stored thermal energy calculation.

| Station       | (lbm)<br>Weight | (Btu/lbm °F)<br>$C_p$ | (°F)<br>$\Delta T$ | (Btu)<br>$Q$    |
|---------------|-----------------|-----------------------|--------------------|-----------------|
| IGV           | 3.6             | 0.14                  | 150                | 75.6            |
| Rotor 1 (R1)  | 5.0             | 0.14                  | 166                | 116.2           |
| Stator 1 (S1) | 9.75            | 0.14                  | 182                | 248.4           |
| R2            | 3.5             | 0.14                  | 198                | 97.0            |
| S2            | 7.8             | 0.14                  | 214                | 233.7           |
| R3            | 2.55            | 0.14                  | 230                | 82.1            |
| S3            | 4.5             | 0.14                  | 246                | 155.0           |
| R4            | 2.7             | 0.13                  | 262                | 99.1            |
| S4            | 3.2             | 0.13                  | 278                | 115.6           |
| R5            | 2.85            | 0.13                  | 294                | 108.9           |
| S5            | 2.0             | 0.13                  | 310                | 80.0            |
| R6            | 2.85            | 0.13                  | 320                | 120.9           |
| S6            | 2.1             | 0.13                  | 342                | 93.4            |
| R7            | 2.7             | 0.13                  | 358                | 125.7           |
| S7            | 1.2             | 0.12                  | 374                | 53.7            |
| R8            | 2.4             | 0.12                  | 390                | 112.3           |
| S8            | 1.3             | 0.12                  | 406                | 63.3            |
| R9            | 2.7             | 0.12                  | 422                | 136.7           |
| S9            | 1.35            | 0.12                  | 438                | 71.0            |
| R10           | 1.9             | 0.12                  | 454                | 103.5           |
| S10           | 1.9             | 0.12                  | 470                | 107.2           |
|               |                 |                       |                    | $\Sigma$ 2399.9 |

## CHAPTER IV

### EXPERIMENTAL EVALUATION OF TRANSIENT THERMAL ENERGY RELEASE

#### Bodie Transients

Analysis of real transient engine performance was conducted using test data from aircraft gas turbine engines. The tests were conducted at AEDC in their altitude test cells. The engines tested were low by-pass, high performance turbofan engines with thrust augmentation (Figure 17). The standard components of an afterburning turbofan engine are the inlet, the low pressure compressor (LPC), the high pressure compressor (HPC), the burner, the turbines (high and low pressure), the afterburner, and the nozzle. The data were collected by use of fore and aft facing pressure probes that measured the total and static pressures respectively, and high response temperature probes that measured values of total temperature, all located at selected stations within the engine. For this investigation the component of the greatest interest is the HPC, since it will experience the most instability during transient conditions. Due to the sensitivity of the data the actual numerical values of the results will not be disclosed but the general trends of the results will be presented.

A "bodie" transient is a standard stability test for gas turbine engines. It represents real flight conditions and requirements. By utilizing a series of speed transients the "bodie" test examines the

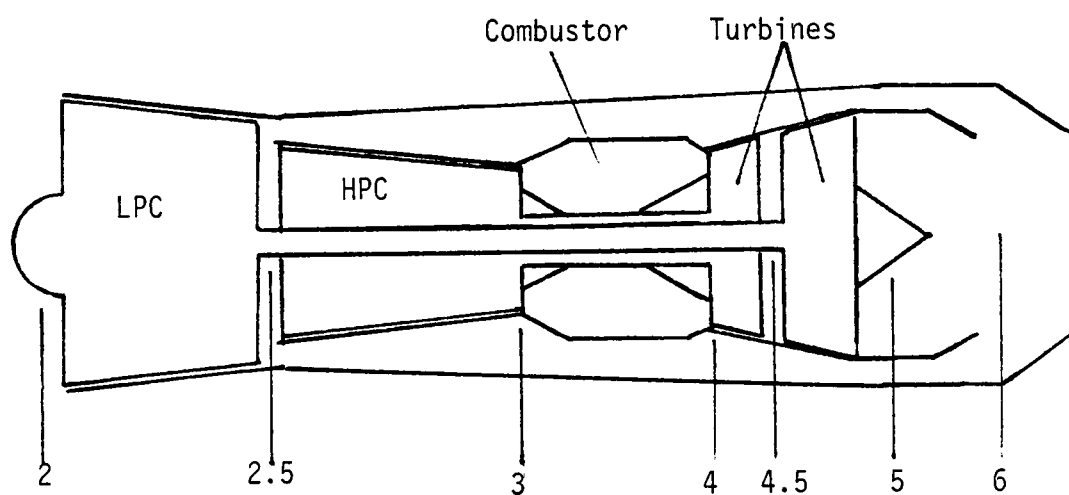


Figure 17. Simple schematic of typical low by-pass turbofan.

engine's stability (Figure 18). The engine is initially run at a high power level setting with controlled inlet conditions corresponding to a mach number and altitude. The engine is run at this setting until stable operation is attained. Stable operation at the initiation of the transient is required to eliminate any outside factors that may contribute to instability. From stable operation the engine is quickly decelerated. This operation is called a "throttle chop." The engine is decelerated to a lower power level setting corresponding to a lower power lever angle. It is run at this setting for a predetermined amount of time called the "dwell time." At the end of the "dwell time" the engine is quickly re-accelerated, or "snapped" to its former high level setting. Instabilities usually occur when the engine is re-accelerated. During the deceleration, heat transfer occurs between the hot compressor metal and the cooler air. As mentioned in the literature search, there is a corresponding reduction of the surge margin. When the engine is re-accelerated the surge margin is further reduced due to the shifting of the transient operating line. If the surge line is crossed during the acceleration the compressor will surge or stall.

From the "bodie" stability investigation, pairs of data points were selected that were of interest to this investigation. These pairs of data points consisted of two tests that were run under similar conditions but met with different results. In a pair of data points, one test encountered a stall upon re-acceleration and the other did not. It was believed that with this information a correlation between transient heat transfer and the occurrence of stall could be determined. Four pairs of similar data points from two different engines were

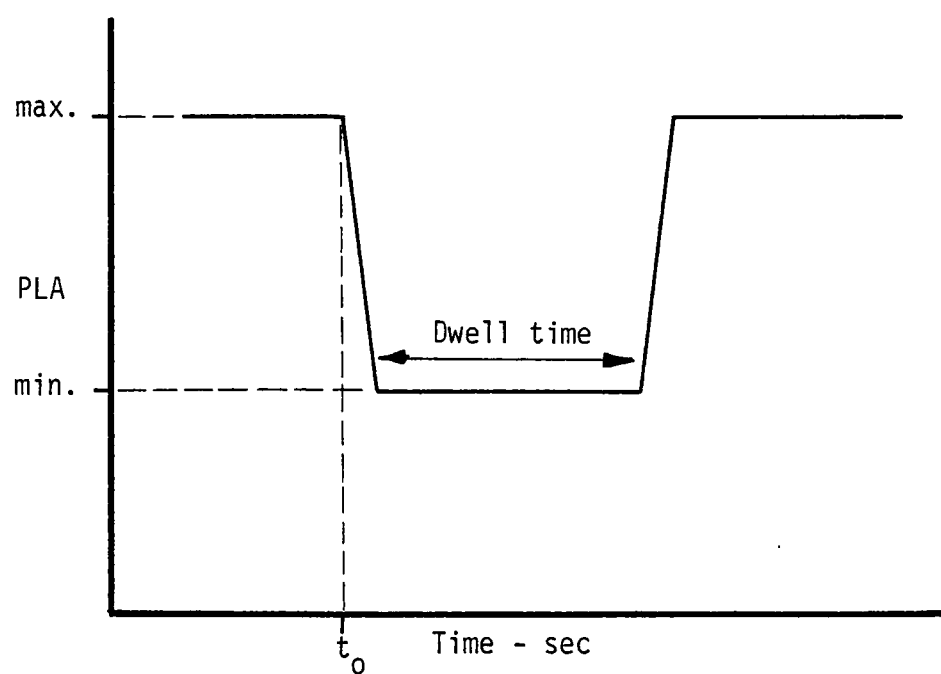


Figure 18. Bodie transient.

chosen. Using one-dimensional, compressible flow relations at the inlet and exit of the HPC the amount of heat transfer may be quantified given certain flow parameters at these locations. These parameters consist of total and static pressures and total temperatures at the inlet of the HPC, the total values of pressure and temperature at the exit of the HPC, various values of air properties at each location, and the total flow area at the inlet of the compressor.

#### Transient Mass Flow

To evaluate the transient heat transfer rate it is first necessary to calculate the transient mass flow for each time interval using the one-dimensional, mass flow relation:

$$\dot{m} = \rho AV \quad (13)$$

By incorporating the following flow relations:

$$\rho = \frac{p_s}{RT_s} \quad (14)$$

$$\frac{p_t}{p_s} = \left( \frac{T_t}{T_s} \right)^{\frac{\gamma}{\gamma-1}} \quad (15)$$

$$\frac{p_t}{p_s} = \left( 1 + \frac{\gamma-1}{\gamma} M^2 \right)^{\frac{\gamma}{\gamma-1}} \quad (16)$$

$$M = \frac{V}{a} \quad (17)$$

$$a = \sqrt{\gamma RT} \quad (18)$$

the mass flow relation can be expanded to the form:

$$\frac{\dot{m}}{A} = \frac{p_{t2.5} \sqrt{\gamma}}{\sqrt{RT_{t2.5}}} \left( \frac{p_{t2.5}}{p_{s2.5}} \right)^{\frac{-(\gamma+1)}{2\gamma}} \left[ \frac{2}{\gamma-1} \left( \frac{p_{t2.5}}{p_{s2.5}} \right)^{\frac{\gamma-1}{\gamma}} - \frac{2}{\gamma-1} \right]^{\frac{1}{2}} \quad (19)$$

This equation relates the mass flow per unit area to the ratio of total to static pressure, total pressure and temperature, and various air properties, all at the inlet of the compressor. Given a flow area at the inlet the mass flow can be evaluated for each time interval. With this information a transient operating line can be determined and displayed on a pressure ratio versus mass flow diagram (Figure 19). This diagram represents the deceleration only. Any instability would occur at the moment of re-acceleration and this investigation is only concerned with pre-stall conditions.

#### Adiabatic Efficiency and Temperature

Combining definition of adiabatic efficiency and enthalpy for processes with constant specific heat,

$$\eta_c = \frac{h_{t3}' - h_{t2.5}}{h_{t3} - h_{t2.5}} \quad (20)$$

$$\Delta h_t = c_p \Delta T_t \quad (21)$$

and the isentropic flow relation between pressure and temperature:

$$\frac{T_{t3}'}{T_{t2.5}} = \left( \frac{p_{t3}}{p_{t2.5}} \right)^{\frac{\gamma-1}{\gamma}} \quad (22)$$

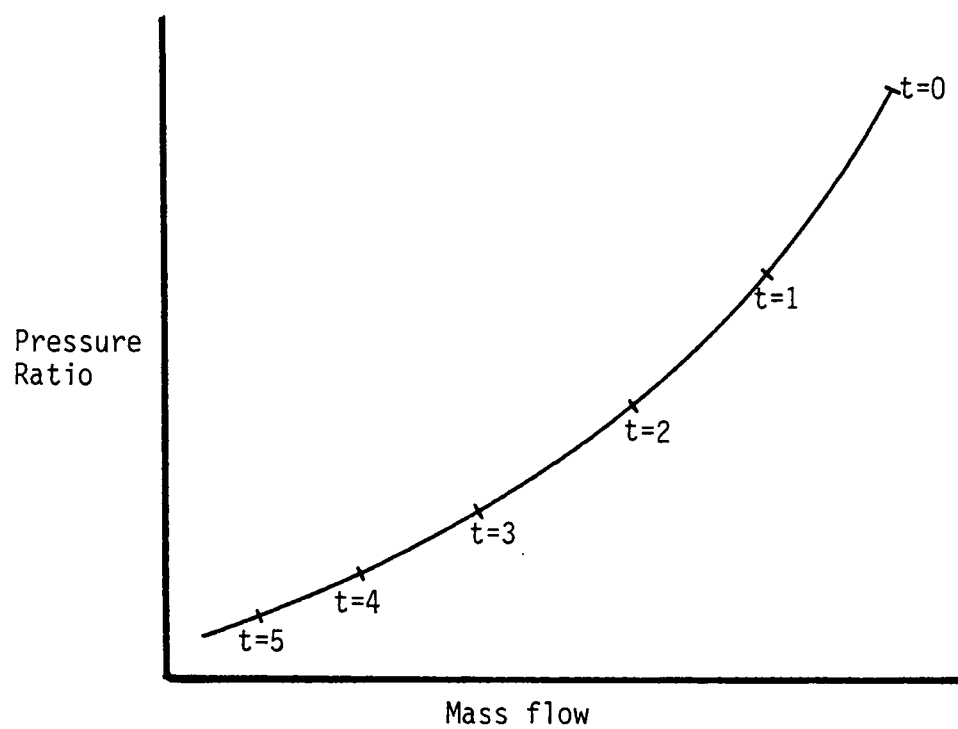


Figure 19. Pressure ratio-- Mass flow relation during speed transient.

the following equation relating the ratio of total temperatures to the ratio of total pressures and adiabatic efficiency through the HPC can be derived:

$$\frac{T_{t3}}{T_{t2.5}} = \frac{1}{\eta_c} \left[ \left( \frac{p_{t3}}{p_{t2.5}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] + 1 \quad (23)$$

Assuming that the compressor has reached an adiabatic state during the time just before the "bodie" transient (which is a good assumption since the compressor is operating at a stable, steady state at the time), an adiabatic efficiency can be determined. During the transient the adiabatic efficiency will not be constant, but for these initial calculations it was assumed to be the value just prior to the deceleration. Therefore, using equation (23) and holding the adiabatic efficiency constant, an adiabatic exit temperature can be found:

$$T_{t3ad} = \frac{T_{t2.5}}{\eta_c} \left[ \left( \frac{p_{t3}}{p_{t2.5}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] + T_{t2.5} \quad (24)$$

This adiabatic exit temperature is the temperature that would occur if the compression process were adiabatic. When comparing this temperature to the actual exit temperature a difference appears which is due to the presence of heat transfer in the actual case (Figure 20). When the deceleration begins, the difference in actual and adiabatic exit temperatures increases due to thermal energy being transferred to the air. As the heat transfer rate maximizes and begins to diminish, the difference in temperatures will also maximize and begin to decrease. If the engine were allowed to reach a steady condition at the lower

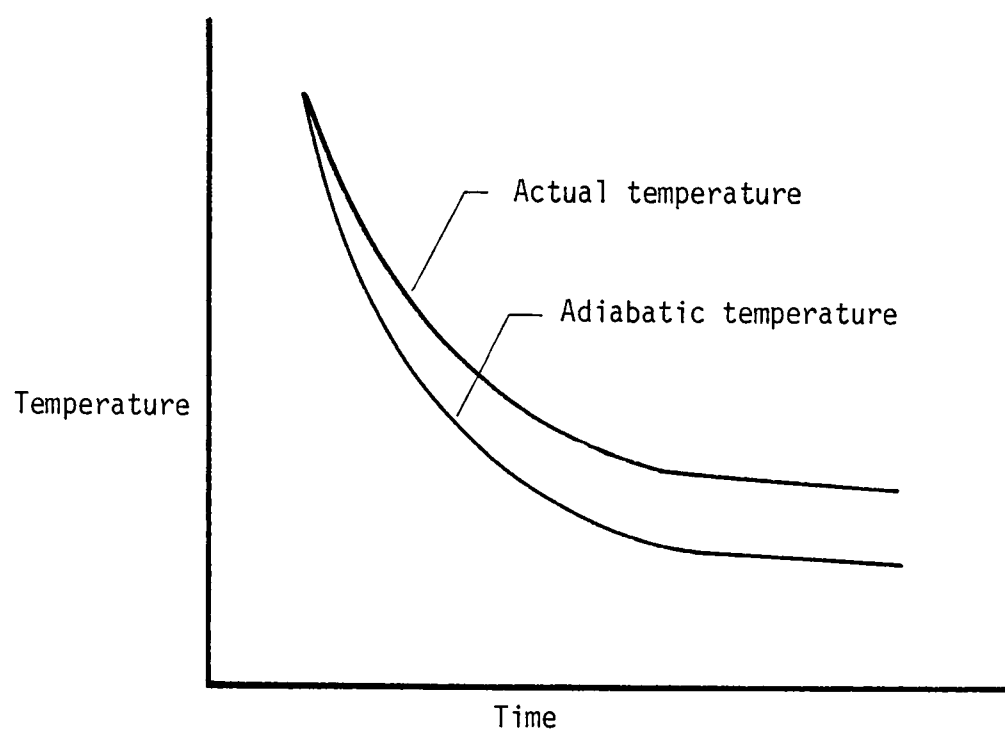


Figure 20. Actual versus adiabatic exit temperature.

power level setting, the difference in adiabatic and actual exit temperatures would disappear as the engine approached a new adiabatic, steady state.

### Heat Transfer Rate

Applying an energy balance to the compressor the heat transfer rate may be quantified. During an actual transient the total amount of energy transferred to the air is obtained from the difference in amounts of energy between the inlet and exit. This total amount of energy can be separated into two forms: adiabatic compression, and heat transfer:

$$E_{act} = (E_3 - E_{2.5})_{act} = E_{ad} + E_{ht} \quad (25)$$

The adiabatic compression can be determined by the difference in energies between the entrance and the exit of the compressor during an adiabatic process:

$$E_{ad} = (E_3 - E_{2.5})_{ad} \quad (26)$$

From the general energy equation adiabatic compressor work can be represented in the following manner, assuming constant mass flow:

$$E_{ad} = \dot{m} \left( h_3 - h_{2.5} + \frac{V_3^2 - V_{2.5}^2}{2g} \right) \quad (27)$$

or:

$$E_{ad} = \dot{m} (h_{t3} - h_{t2.5})_{ad} \quad (28)$$

The actual compressor work can be represented in a similar manner:

$$E_{act} = \dot{m} (h_{t3} - h_{t2.5})_{act} = (E_3 - E_{2.5})_{act} \quad (29)$$

From equation (25) the rate of heat transfer can be found:

$$E_{ht} = (E_3 - E_{2.5})_{act} - E_{ad} \quad (30)$$

or:

$$E_{ht} = \dot{Q}_{ht} = \dot{m} (h_{t3} - h_{t2.5})_{act} - \dot{m} (h_{t3} - h_{t2.5})_{ad} \quad (31)$$

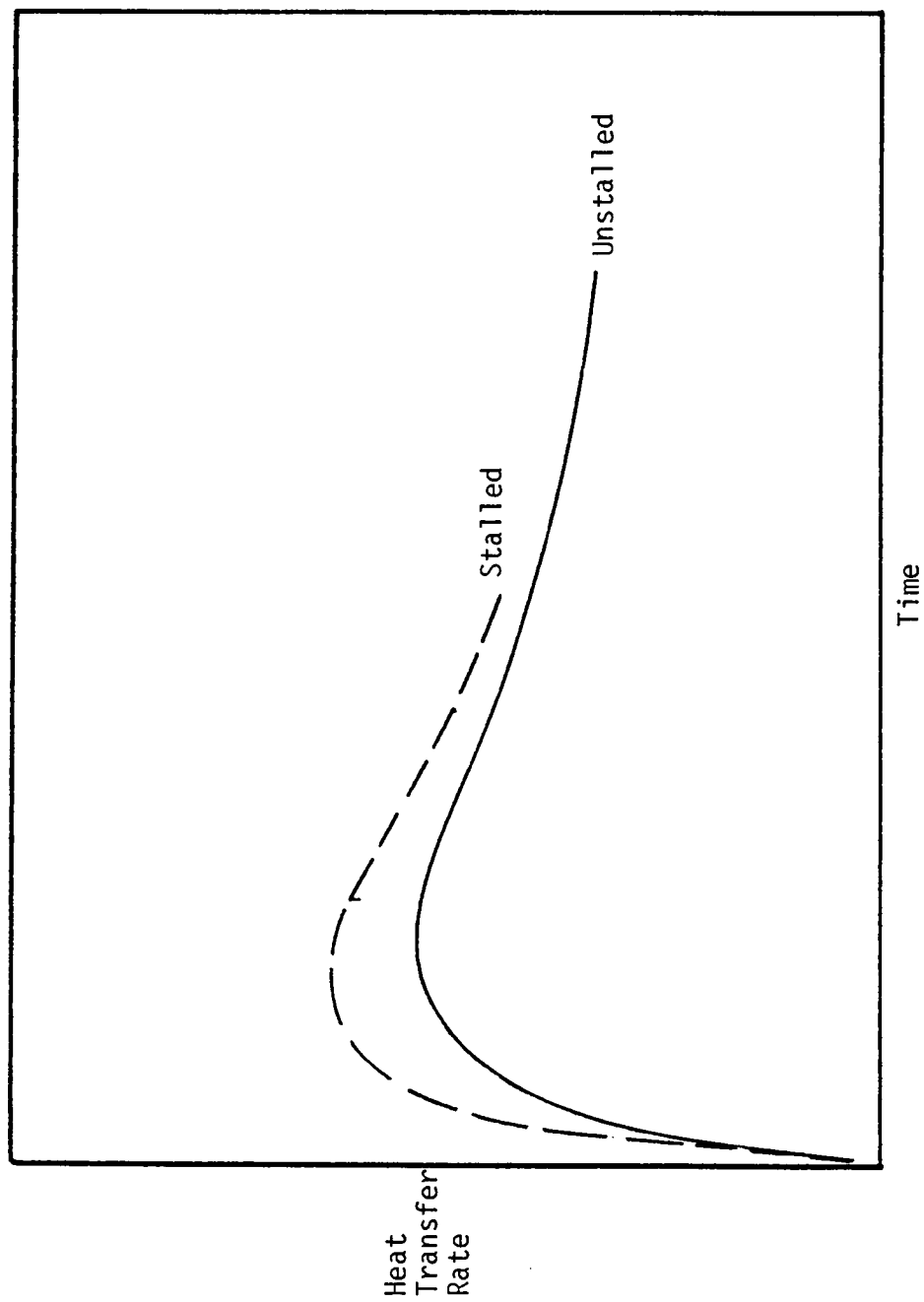
The conditions at the inlet of the HPC will not be affected by the heat transfer within the compressor, therefore, the adiabatic and actual values of enthalpy at that station will be equal, and assuming constant specific heats, equation (31) reduces to:

$$\dot{Q}_{ht} = \dot{m} c_p (T_{t3act} - T_{t3ad}) \quad (32)$$

Using equation (32) the heat transfer rate within the compressor can be estimated for each time increment knowing the transient mass flow, and the actual and adiabatic temperatures at the exit of the HPC.

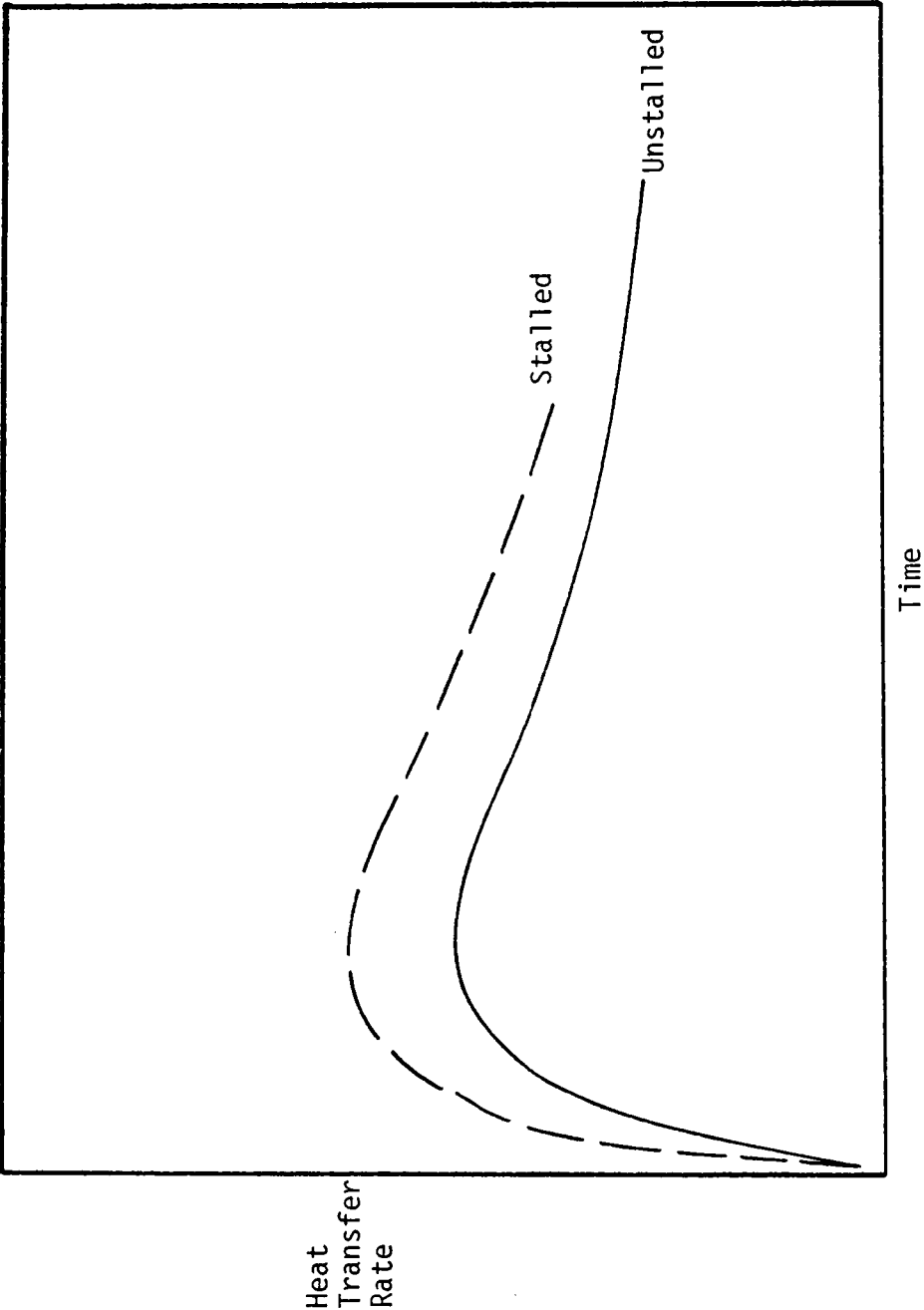
### Quantitative Results

This method was applied to the four pairs of engine test data. In two out of the four cases the stalled test showed a greater heat transfer rate than the unstalled test (Figures 21a and 21b). In the other two cases the heat transfer of the stalled test was similar to or initially less than the heat transfer of the unstalled test. But,



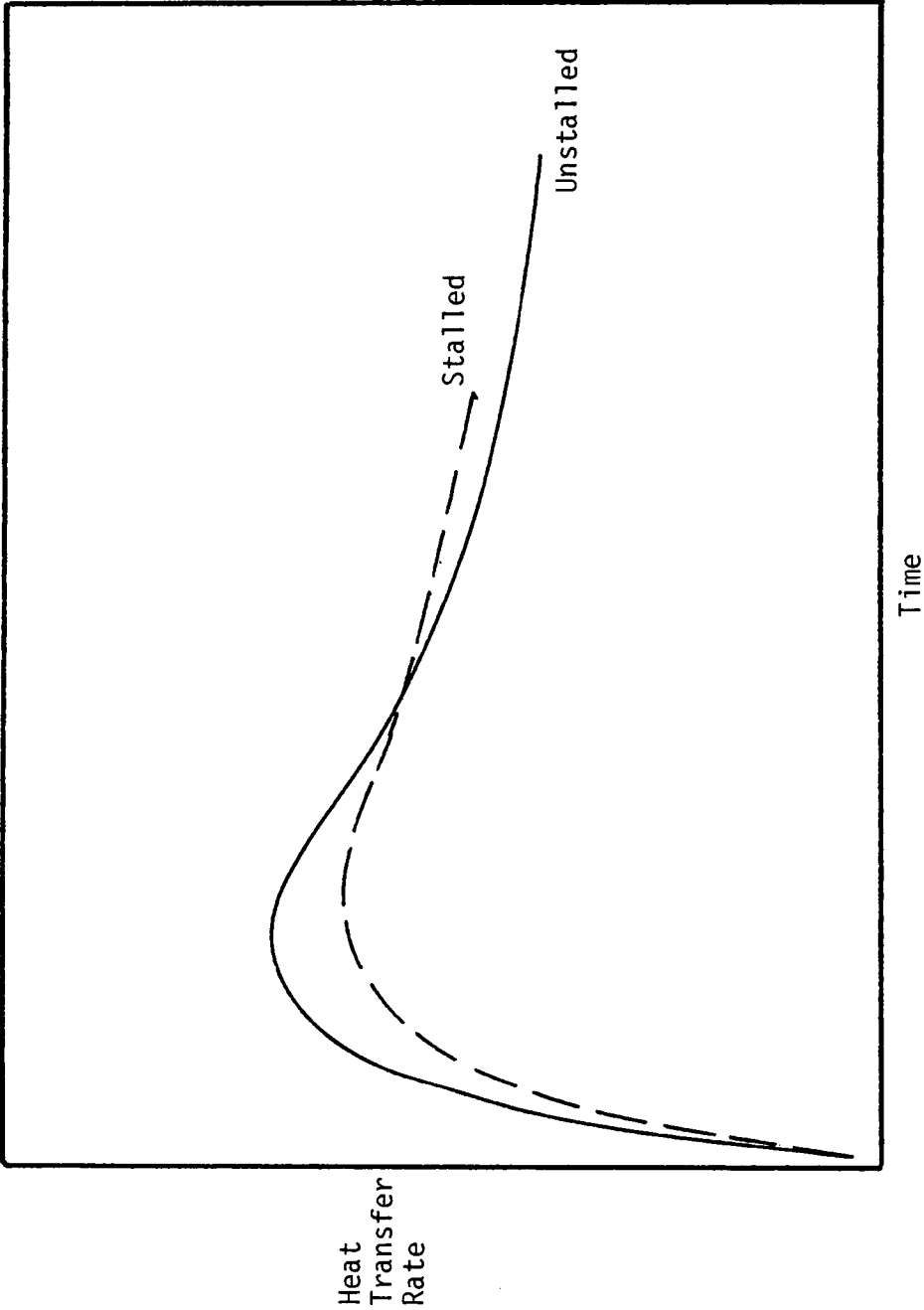
(a)

Figure 21. Comparison of heat transfer rates between stalled and unstalled tests.

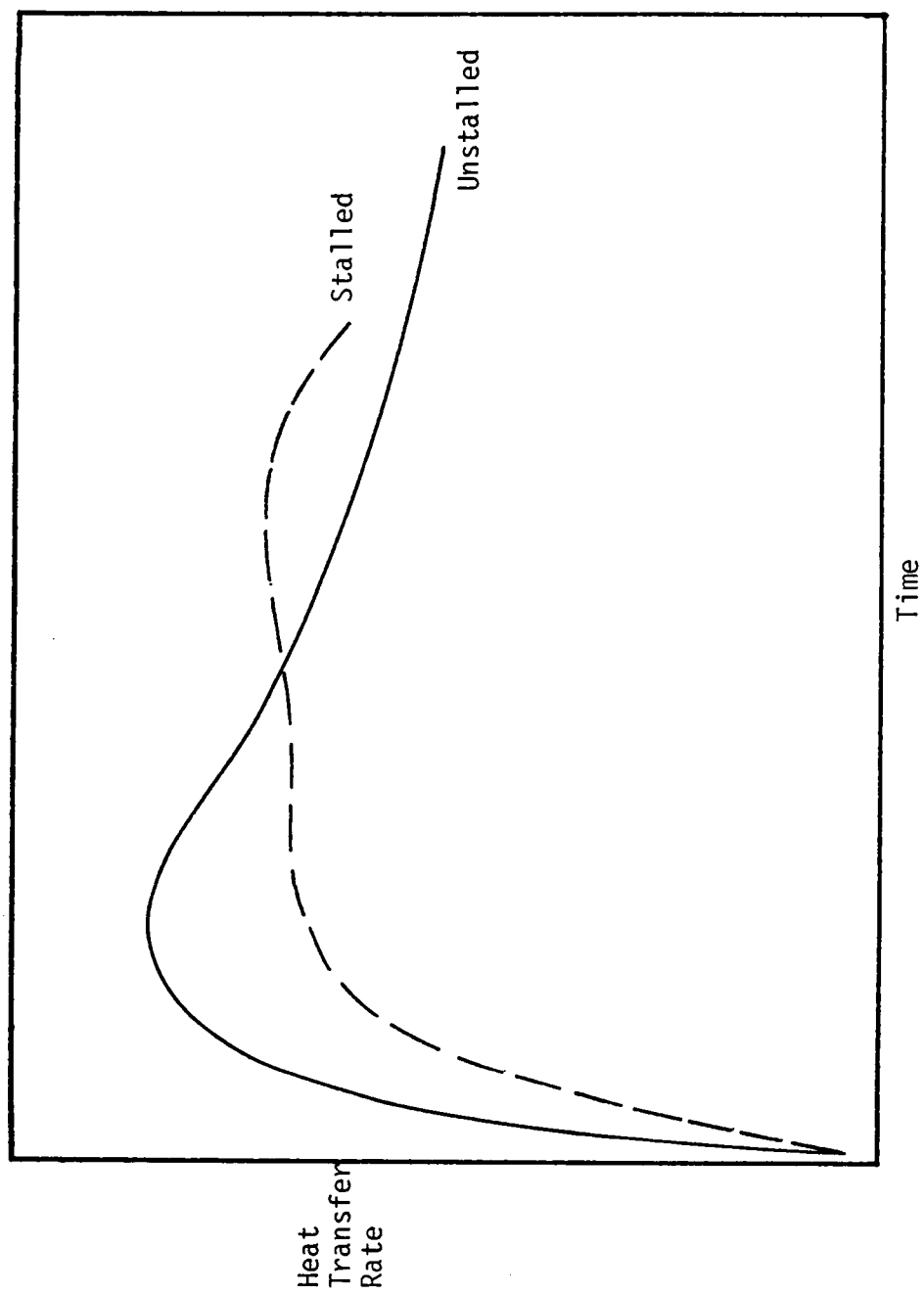


(b)

Figure 21 (continued)



(c)  
Figure 21 (continued)

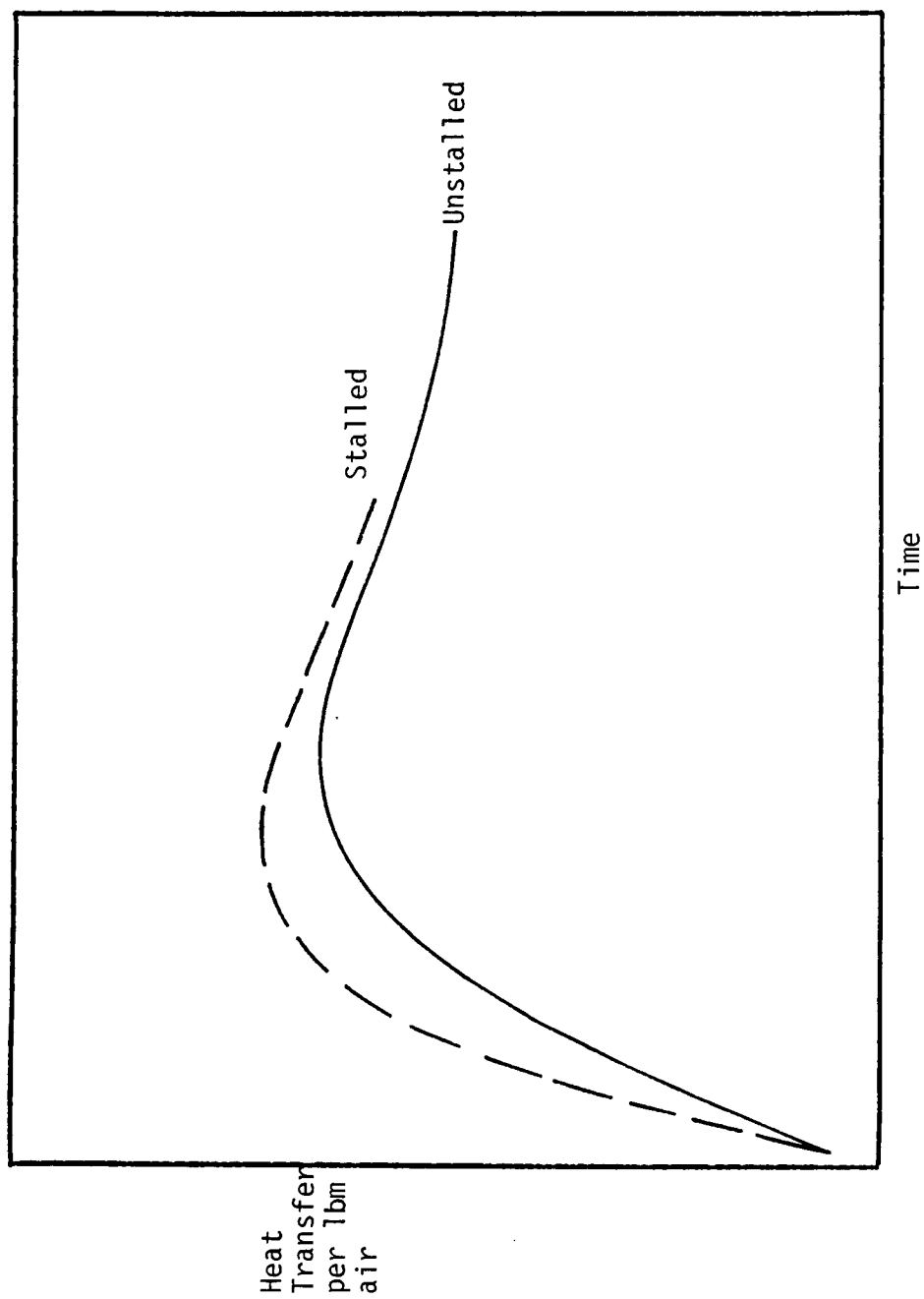


(d)  
Figure 21 (continued)

eventually the heat transfer rate of the stalled test became greater (Figures 21c and 21d). In all cases the heat transfer rate of the stalled test was greater than that of the unstalled test at the moment of re-acceleration.

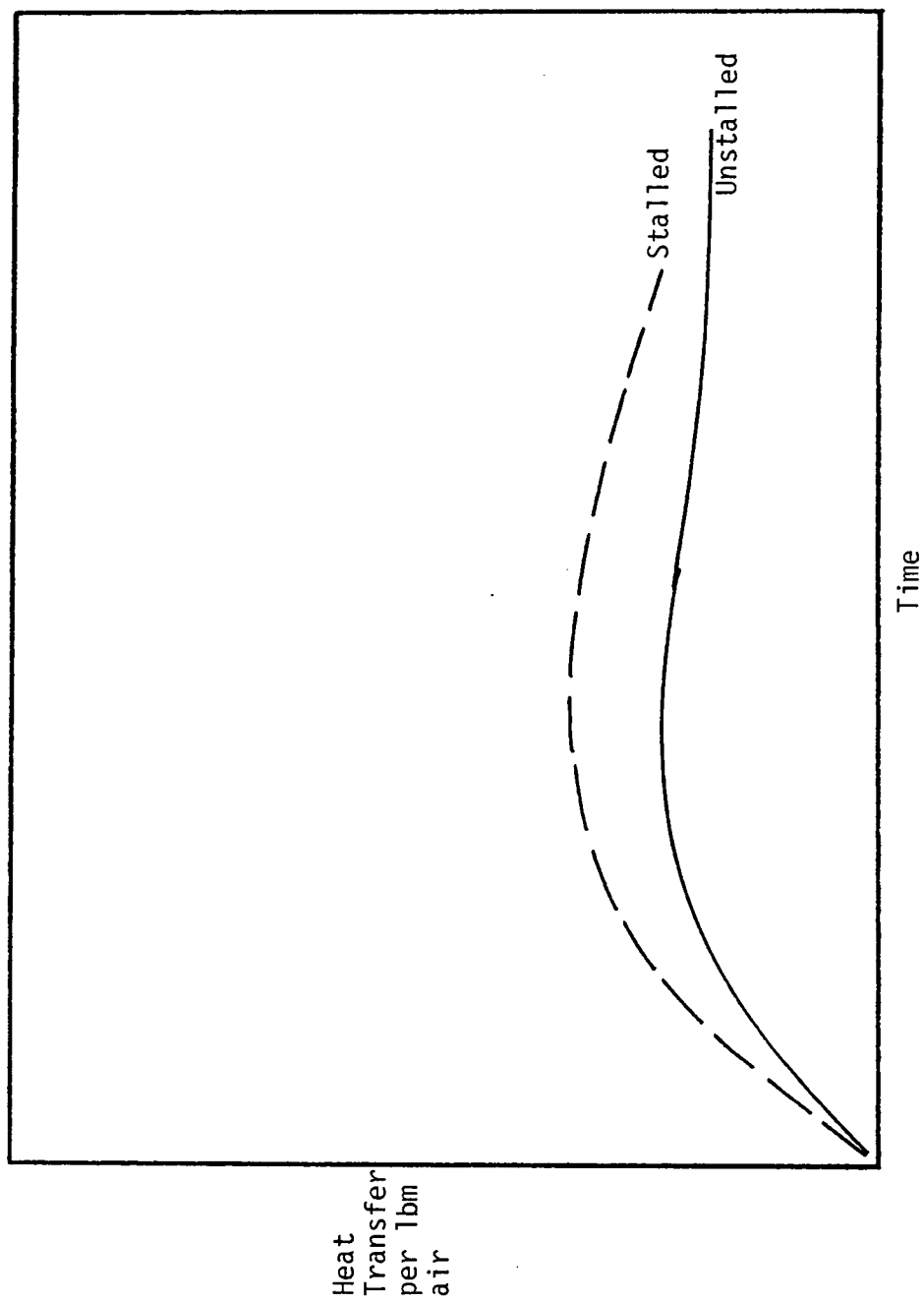
At the start of deceleration the heat transfer rate rises sharply to a maximum value of about 500-1250 Btu/s in about 3-4 seconds as the hot metal releases its stored thermal energy. The sharp increase is due to the blades releasing their stored energy immediately while the annulus, discs, and other internal parts release their stored energy at a slower rate. After a maximum rate is attained there is a steady decline in the heat transfer rate as the hot metal cools. By representing these preliminary results in the form of heat transfer per pound mass of air the effect of input of heat to the air flow can be better seen (Figure 22). Upon deceleration the heat transfer rate quickly climbs to a maximum, however, at that instance the mass flow rate is dropping but still relatively high. Therefore, the heat transfer per pound mass of air reaches a maximum of about 20-40 Btu/lbm more gradually (5-7 seconds) than the total heat transfer rate. After the maximum value is reached, the heat transfer rate declines along with the mass flow rate and the heat transfer per pound mass of air reduces at a slow rate, more slowly than the absolute heat transfer rate.

To show the relative effects of heat transfer on compressor work, the amount of temperature rise through the compressor due to adiabatic compression and due to heat transfer can be compared (Figure 23). During a typical deceleration the temperature rise due to adiabatic compression will decrease as the speed and pressure ratio of the



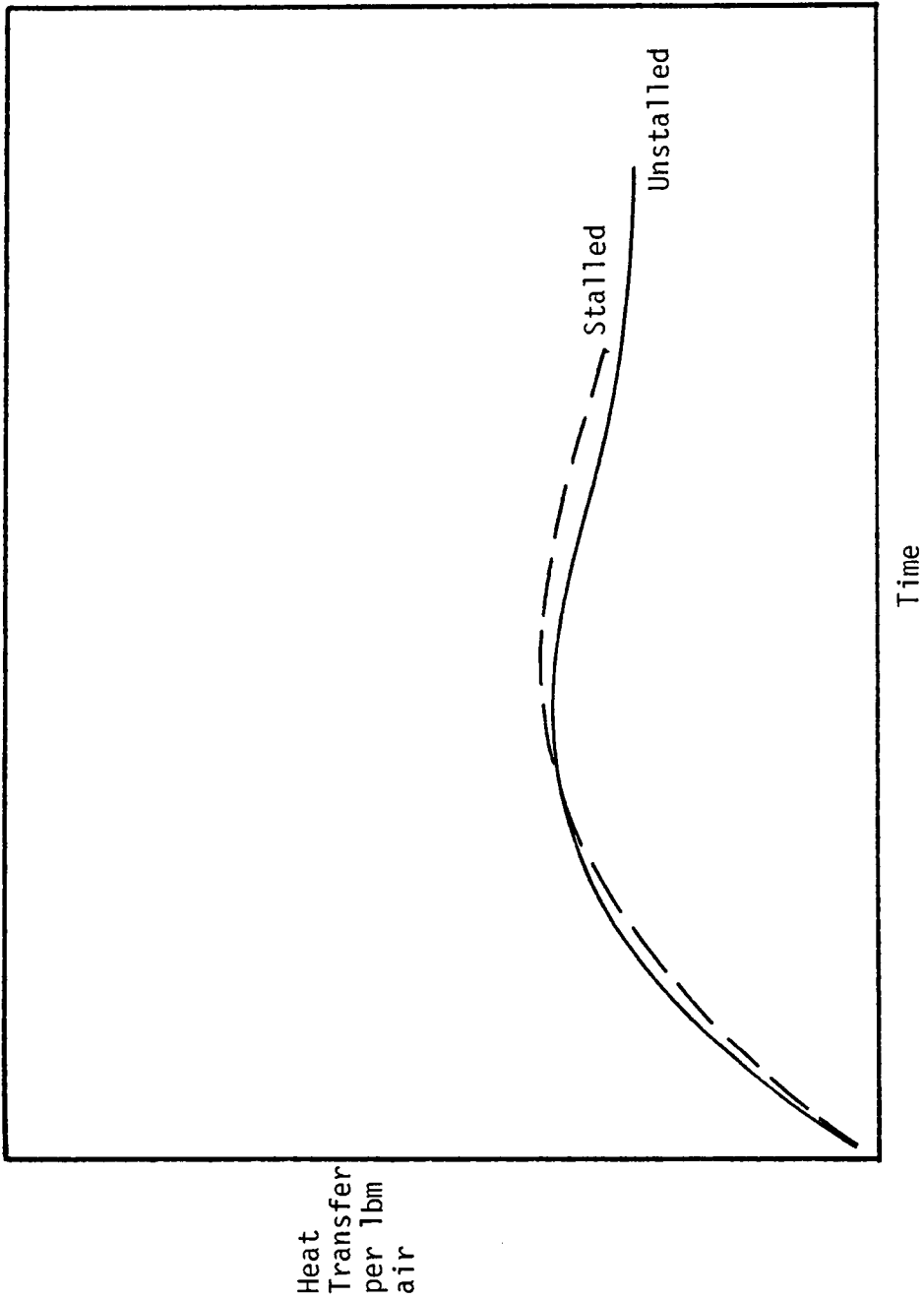
(a)

Figure 22. Comparison of heat transfer per pound mass of air between stalled and unstalled tests.

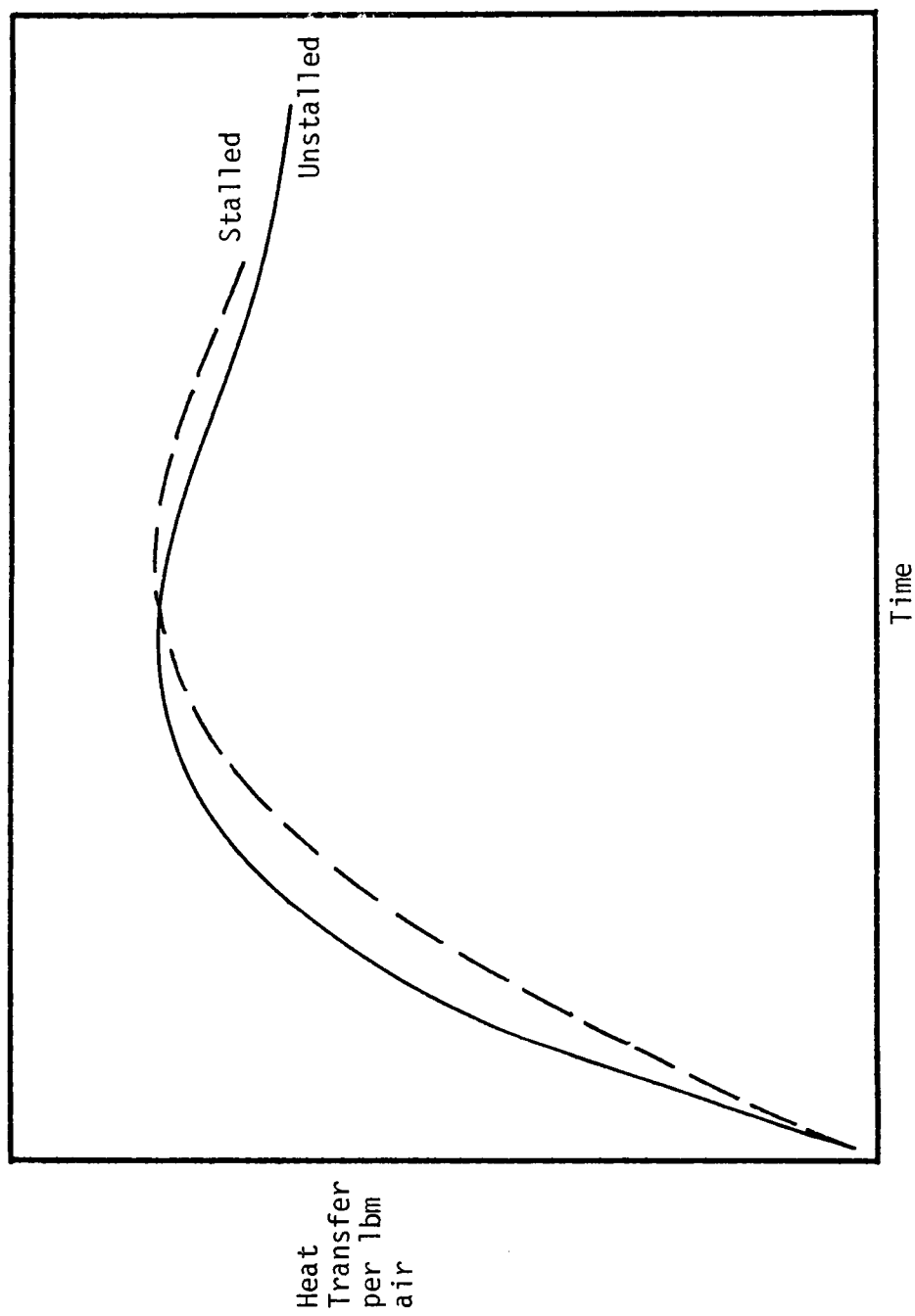


(b)

Figure 22 (continued)



(c)  
Figure 22 (continued)



(d)

Figure 22 (continued)

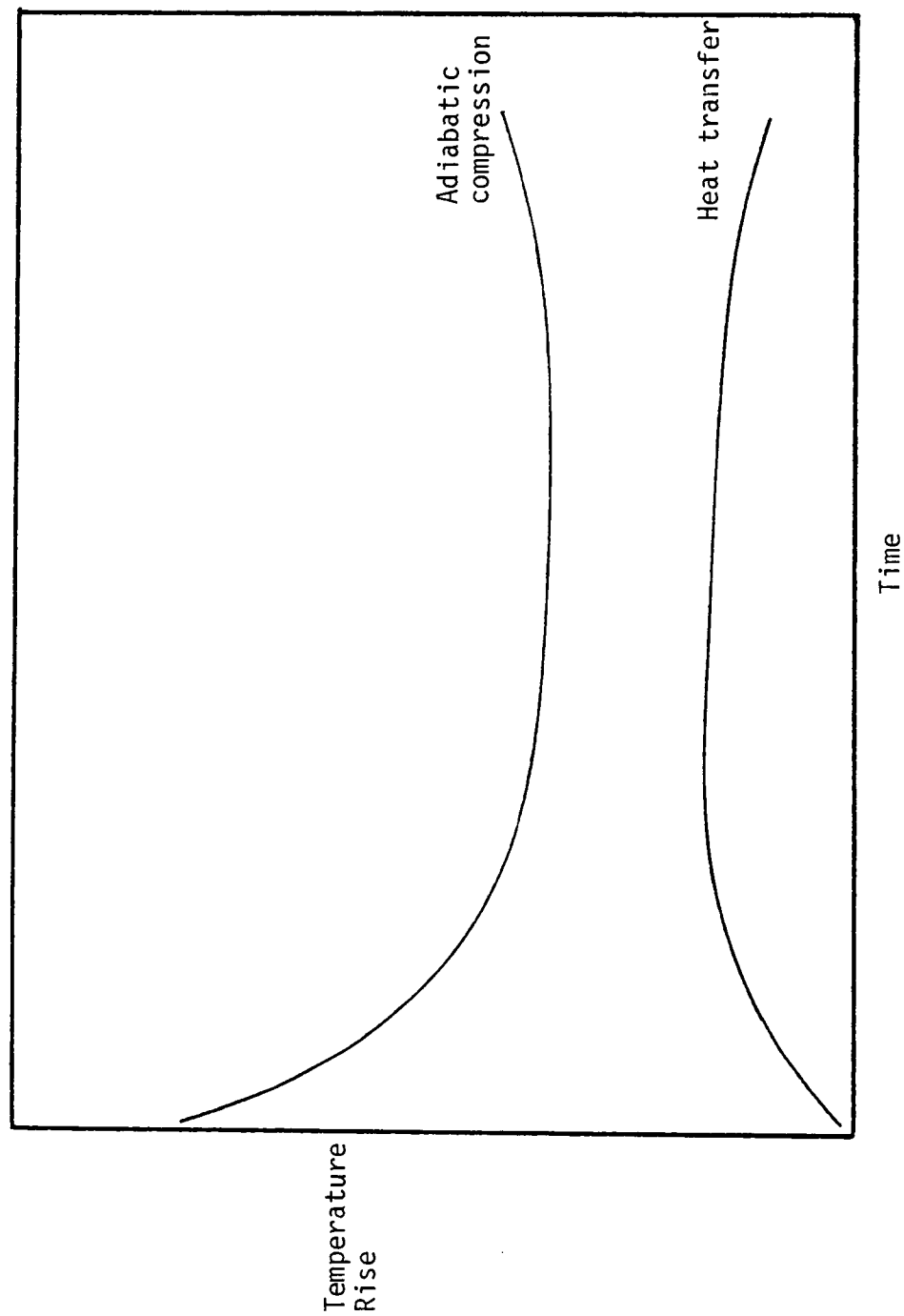


Figure 23. Typical temperature rise through a compressor due to adiabatic compression and heat transfer.

compressor decrease. However, the temperature rise due to heat transfer will increase as the speed and pressure ratio drop. At its maximum point the heat transfer accounts for about 25-35% of the temperature rise through the compressor.

The transient heat transfer rate can be affected by many things. From equation (32) the heat transfer rate is a function of the mass flow, the specific heat at constant pressure, and the difference in adiabatic and actual exit temperatures. These parameters are also dependent on several things. The flight mach number and altitude of the test will influence the mass flow of the machine. The engine control system, which controls the power level setting, influences the rotational speed and compressor exit pressure. This can affect the mass flow and the compressor exit temperature. The test presented in this investigation represents only one mach number and altitude condition (0.8 mach number and 20K feet altitude) from two different engines with two different control systems. Practically speaking, the ratio of the mass flow of the air to the mass of the compressor hot metal parts has a definite effect on the heat transfer rate. A larger engine that contains more mass in contact with the air flow will have a greater heat storage capacity. Furthermore, an engine with a greater mass flow rate will have a greater heat transfer rate.

The accuracy of this approach depends on the accuracy of the mass flow calculations, the correlation between pressure and temperature, and the evaluation of the adiabatic compressor efficiency, which can be inaccurate during thermal transients. It is estimated that this analysis produces plus or minus 10% accuracy in the calculated

heat transfer rates. The uncertainty is larger if the raw engine test data are scattered.

## CHAPTER V

### RESULTS AND CONCLUSIONS

The literature search at the initiation of this investigation proved to be rather extensive, covering domestic as well as foreign sources. The results confirmed that there is a significant influence of transient heat transfer on the occurrence of compressor instability. The effect of heat transfer from the compressor metal to the airstream during a deceleration was felt in two ways. First, there was a general reduction in air density due to "bulk" heat transfer effects, and second, there was an influence on the growth and shape of the boundary layers on the annulus and blades. The analytical methods used in this investigation concluded that the combined effects of "bulk" heat transfer and the boundary layer growth resulted in the reduction of the surge margin and a loss of compressor performance.

The evaluation of the magnitude of the stored thermal energy in the blades alone revealed that there was a significant amount of energy released from the compressor metal during a thermal transient that can possibly affect the compressor stability as described in the preceding paragraph.

The preliminary results from the analysis of actual engine data from "bodie" stability tests showed higher heat transfer rates for a test that encountered compressor instabilities. These preliminary results were very significant but not conclusive due to the assumptions

taken during the analysis procedure and the limited number of transients analyzed. Although the intended results were achieved, more tests need to be investigated to support the conclusions.

An interesting result was achieved, though. The amount of stored thermal energy contained in just the blades was about 2500 Btu's. This was, as explained before, supposed to be released fairly quickly. Upon further investigation of the quantitative analysis results, it was shown that the calculated amount of energy released during the first 3-4 seconds of the transient compared with the calculated amount of stored thermal energy within the blades. Since the blades will release their stored energy rather quickly, it can be concluded that this analysis procedure is relatively accurate and can be used for future investigations.

## CHAPTER VI

### RECOMMENDATIONS

This investigation has confirmed the importance of the influence of transient heat transfer on compressor stability. Short term plans should include the study of more engine stability tests to verify the preliminary results. Long-term plans should include: the theoretical analysis of individual blade heat transfer rates and how they correlate to cascade stability; and, the experimental investigation of the influence of blade heat transfer rates on the occurrence of cascade stall.

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## VITA

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