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A STUDY OF THE PREDICTION OF TIP AND SEAL CLEARANCES AND THEIR EFFECTS IN GAS TURBINE TRANSIENTS

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ABSTRACT

Clearances of compressor and turbine blade tips and seals alter during and following speed transients. These changes affect the performance of the components and hence of the engine.

This paper describes models for the prediction of blade tip clearance changes and seal clearance changes. These models have been applied to the H.P. Compressor and H.P. Turbine and to two seals controlling air flows in a Two-spool Bypass Engine. The predicted acceleration rates appear to be more influenced by the changes in the seal clearances than by the tip clearance changes.

The increases in computing time in the engine transient program which result from inclusion of the model are acceptable.

NOMENCLATURE

A = aspect ratio
C = clearance
C_p = specific heat at constant pressure
D = diameter
E = Young's Modulus
g = acceleration due to local gravity
Gr = Grashof Number ($= \frac{l^3 \theta \alpha g}{\nu}$)
h = local heat transfer coefficient
H = height
k = fluid thermal conductivity
l = characteristic length, blade chord
L = axial length of stage
M = mass
Nu_b = average Nusselt Number for blade (= hl/k)
Nu_r = local Nusselt Number on disc (= hR/k)
Nu_{ic} = Nusselt Number for inner surface of casing ($= h(D_o - D_i)/k$)
Pr = Prandtl Number
R = radius
Re_r = local rotational Reynolds Number ($= \omega R^2/\nu$)
Re_b = Reynolds Number over blade ($= ul/\nu$)
Re_{ic} = Reynolds Number for inner surface of casing (see Eq. 4)
T = absolute temperature
u = average gas velocity
V = volume of disc section
Va = axial velocity of fluid
Vt = blade tip tangential velocity
α = coefficient of cubical expansion
β_m = mean air angle ($\tan \beta_m = \frac{1}{2}(\tan \beta_i + \tan \beta_o)$)
Δ = difference or change
ε = effectiveness
η = efficiency
θ = temperature difference
λ = non-dimensional clearance ($= C/H_b$)
ν = kinematic viscosity
ψ = blade loading ($= 2C_p \Delta T_s/Vt^2$)
ω = angular velocity

Subscripts

av = average
b = blade
bt = blade tip
c = cooling air
d = disc
g = gas
h = hub of blade
i = inner, or inlet
j = j'th section
m = mean
mf = final metal (temperature)
o = outer, or outlet
r = at a radius
s = stage

1. INTRODUCTION

It is desirable to develop accurate methods for predicting the transient behaviour of gas turbines. The earliest programs for such predictions used equilibrium characteristics of the components and ignored alterations in these, such as due to heat absorption or tip clearance changes. Thomson (1)[1] quotes underpredictions of acceleration time of 20 to 30 per cent as compared to the real engine, when using these simplified procedures.

The influence of the non-adiabatic flow (a direct heat transfer effect) has subsequently been studied for a Single-spool Engine (2) and for a Two-spool Bypass Engine (3).

Alterations in tip clearances cause changes in component efficiencies. This can be regarded as an indirect effect of heat transfer, although tip clearances are also influenced by centrifugal and pressure effects. Another indirect effect of heat transfer lies in the response of seals which control cooling air flows.

The objects of the present paper are to develop models for representing tip clearance changes and to use these models to predict the effect of tip clearance changes during transients on the performance of the various components and hence of the engine. The models used in this work have been described briefly in a previous paper (4). More detailed descriptions are given in this paper. The transient program for the Two-spool Bypass Engine (3) has been extended to include these models so as to enable the calculation of the desired effects on engine transient response.

2. DETAILED DESCRIPTION OF THE METHOD

Gas turbines have to operate over a range of steady-running conditions. The mechanical and thermal loadings of the components vary, producing small changes in the dimensions in both the radial and axial directions. These changes can result in relative movements between rotating and stationary components, thus causing varying tip clearances and off-design seal mass flows. During accelerations and decelerations of the engine these movements can be modified by delays in the temperature responses.

In this paper, the radial changes only are considered. For much of this section the H.P. compressor of a typical two-spool bypass engine has been selected to demonstrate how the representation of a multi-stage turbo-machine can be simplified, still retaining reasonable accuracy.

2.1 Thermal Effects

2.1.1 Model for Blade Tip Movement. The movement of the blade tip depends on the responses of the disc and of the blade.

Considering first the disc (Fig. 1), some faces are rotating adjacent to stationary faces, while other faces form walls of what are effectively rotating chambers. With regard to faces adjacent to stationary walls, several investigators (for example, (5) - (7)) have examined the heat transfer coefficients and/or the drag characteristics. The variables studied included the radial inflow or outflow of coolant and the influence of a shroud. It was decided to base the heat transfer expressions to be used in the present work on experimentally derived correlations. Unfortunately in none of the experimental work did the conditions attain those in the engine, particularly with regard to the rotational

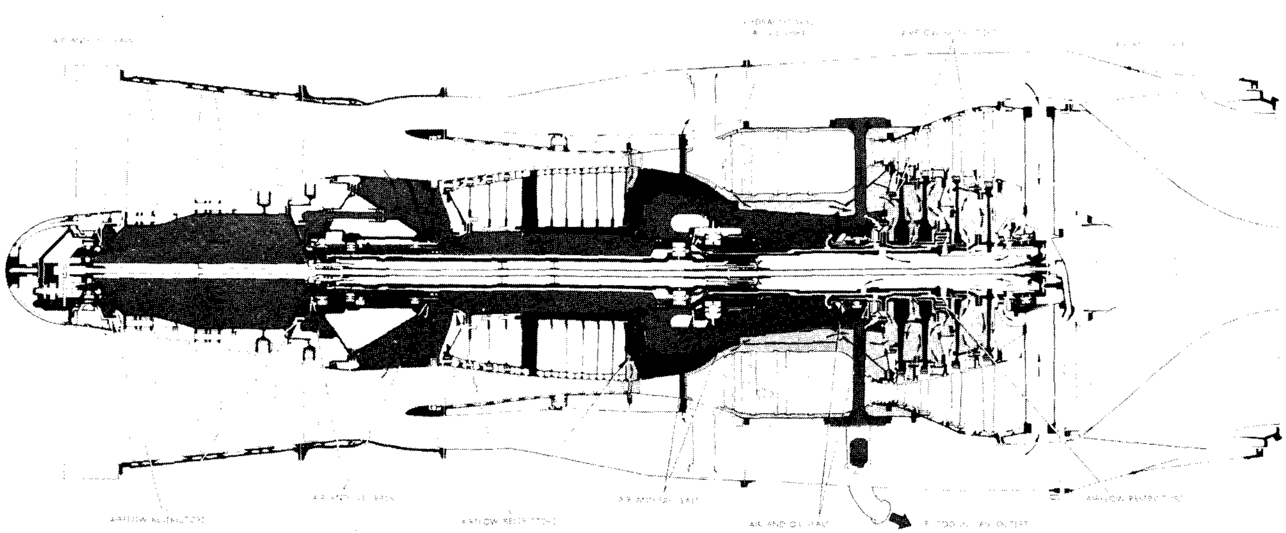


FIG. 1 GENERAL ARRANGEMENT OF TWO-SPOOL BYPASS ENGINE

1. Underlined numbers in brackets designate references at end of paper.

Reynolds Number - the highest value in experiments being 4×10^6 as compared with typically 12×10^6 in the engine. At the highest Reynolds Number of the experimental tests, with typical gap distances between the rotating and stationary discs and with typical shroud clearance and typical outflow of coolant the drag coefficient and heat transfer coefficient were found to be about 30 per cent higher than those for a free rotating disc. The free rotating disc correlation can be extended to Reynolds Numbers of 12×10^6. Assuming in the typical engine situation with an adjacent stationary face, a shroud and a coolant mass flow, that the coefficients are still 30 per cent higher than for a free disc, then the correlation for local Nusselt Number is (8):

$$Nu_r = 0.0253\ (Re_r)^{0.8} \qquad (1)$$

This correlation is not greatly sensitive to shroud clearance or coolant mass flow changes - doubling the coolant mass flow increases the Nusselt Number by 10 per cent, while doubling the shroud clearance lowers it by 5 per cent. It is suggested that for the present work Eq. (1) gives an adequate general correlation.

Considering now the rotating faces which form the walls of rotating chambers, the heat transfer mechanism is effectively natural convection in a high gravity field, the value of the gravitational acceleration being a function of both the rotational speed and the local radius. For this situation the following equation is suggested:

$$Nu_r = 0.12\ (Gr\ .\ Pr)^{0.33} \qquad (2)$$

The above correlations have previously been used (8) to predict the temperature distributions in a turbine disc both during transients and in steady-running conditions. The steady-running temperature distributions were satisfactorily compared with results from thermal paint observations, predicted temperatures generally being within 15 deg. K of the observed values.

The present work is aimed at producing simple models to describe the movements of blade tips, casings etc. It would be preferable if the rather complicated shape of a compressor or turbine disc could be simplified. It is proposed that a deep disc rotor arrangement can be represented by three components - a thick hub, a thin diaphragm and an outer section or rim (Fig. 2). Comparisons between this grossly simplified model and the rigorous finite element transient (9) conduction analysis are given later in this paper.

With regard to the response of turbine blades, for simplicity, uncooled blades will be considered initially. A suitable correlation for heat transfer between the gas stream and turbine blade surfaces has been given by Halls (10):

$$Nu_b = 0.235\ (Re_b)^{0.64} \qquad (3)$$

For compressor blades, one approach has been to adopt a weighted average between laminar and turbulent boundary layers developing on flat plates (11). The results of this method have been found to be within five per cent of the results obtained by applying Halls' turbine correlation to the compressor blades. Therefore for convenience

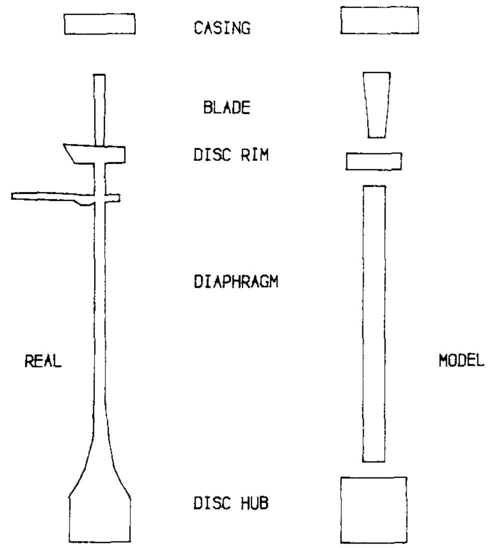


FIG. 2

REPRESENTATION OF A TYPICAL STAGE.

in the present work, which is intended to produce a model applicable to both compressor and turbine tip movements, Halls' correlation is used for all blades.

2.1.2 Model for Casing Movement. The casing structures can be complex, with inner and outer surfaces subjected to gases or air at differing temperatures and pressures, moving with differing velocities. In addition there may be spaces within the casing through which certain air or gas flows pass. At the present stage of development of the method, the effects of these cavities have been ignored, and the casings treated as solid walls.

In order to estimate the heat transfer coefficient at the internal surface of the casing, the approach used has been to regard this as a cylinder in which a smaller cylindrical shape is rotating. The heat transfer coefficient in this case is found from (12):

$$Nu_{ic} = 0.015[1 + 2.3\ (D_o + D_i)/L]\ [D_o/D_i]^{0.45} \times [Re_{ic}]^{0.8}[Pr]^{0.33} \qquad (4)$$

In this case Re_{ic} is given by:

$$Re_{ic} = (Va_m^2 + Vt^2/2)\ (D_o - D_i)/\nu$$

In applying Eq. (4) to the present work, the linear dimension, L, used was the axial length of the blade pair, it being assumed that the end-wall boundary layer is effectively restarted at each blade pair.

For the outer surface of the casing, the expression used was that for a developing turbulent boundary layer on a flat plate.

2.2 Mechanical Effects

The mechanical effects cause growth due to the change in the "pull" of the blades and of the disc sections. Pressure difference changes across a section may also cause growth, as may the effect of accelerating, or decelerating, the disc.

2.2.1 Disc. For simplicity in developing the model, it was assumed that the disc could be represented by the same three rings as in the thermal model. The predicted growths have been compared with those given by a finite element analysis. It was found that a multiplying factor of 1.3 had to be applied to the predictions of the three-ring model to bring them into agreement. (In the use of this three-ring model under the varying thermal and mechanical conditions, continuity of radial dimension at the interface was maintained.)

Disc distortion in the angular direction during a transient was calculated, but the resulting radial movement was negligible.

2.2.2 Blades. The blades are assumed to be rods of uniform cross-sectional area with or without an added shroud. The procedure was compared with calculations of elongation using the finite element method and the results were in good agreement.

2.2.3 Casing. With regard to the casing, the only relevant mechanical loading considered is the pressure change during the transient and its effect is found to be very small, about one per cent of the total movement of the rotor due to mechanical effects.

2.3 Clearance Movements

Once all these effects on the dimensions of the rotor stage or component are calculated, they are added to obtain the resulting change in the clearance.

All material, gas and air properties are calculated as functions of the temperature of the metal or fluid. In the prediction procedure it is of course possible to change the material in any section very easily.

The methods described in the preceding sections have been applied to the 12 stages of the H.P. Compressor of a Two-spool Bypass Engine. The Engine is illustrated in Fig. 1. The rotor system uses deep discs and these are surrounded by air drawn from the fifth and sixth stages. After circulating round these discs this air discharges to the bypass duct. The outer sections of the rotor discs, adjacent to the blade platforms, are effectively in contact with the air passing through the compressor. The predicted blade tip clearance movements for Stages 1, 6 and 11 during and following an acceleration from idle speed at sea level, static, are illustrated in Fig. 3. Looking at the clearance paths at the beginning of the transient, there is a sharp decrease in tip clearance due to mechanical effects, and the fast thermal growth of the blades. The thermal response of the casing is slower than that of the blades. Much of this takes place after the speed transient is completed, producing an increase in tip clearance. The disc has the slowest response and the final slow decrease in clearance is due to the slow expansion of the disc, which takes five minutes or more.

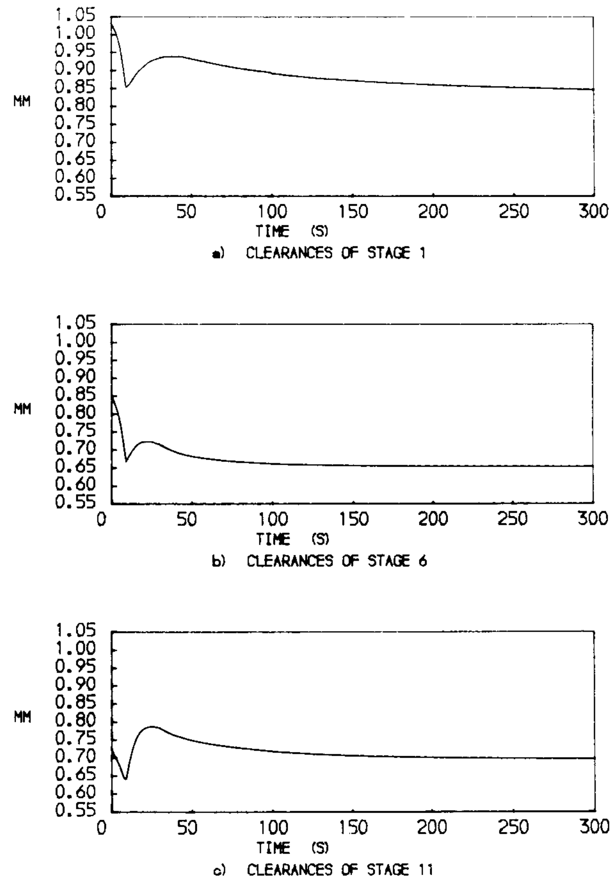


FIG. 3

PREDICTED BLADE TIP CLEARANCES OF THE TWO SPOOL ENGINE H.P. COMPRESSOR (STAGES 1, 6 AND 11) DURING AND FOLLOWING A SEA LEVEL ACCELERATION

It is interesting to notice that for Stage 1 the clearance of the end of the thermal transient is smaller than that at the end of the speed transient, for Stage 6 they are similar, while for Stage 11 the opposite to Stage 1 is the case. This is due to the effect of the air inside the shaft, this, as stated above, being drawn from between the fifth and sixth stages of the Compressor. Therefore the air inside the shaft is hotter than the core air for Stage 1, the two air streams are at similar temperatures for Stage 6 and the core air is hotter than the shaft cooling air at Stage 11. The cooling air dominates the expansion of the disc, whereas the core air dominates the expansion of the blades and the casing. This explains the different end-points of the clearance paths as compared to the clearance at the end of the speed transient.

To indicate the relative rates of thermal response of the various components, the time constants for the major components of Stage 5 are shown in Tables 1 and 2 at three instants during the acceleration and three instants during a deceleration respectively. The heat transfer coefficient is the only parameter affecting this constant, since, for a given configuration, the geometry is fixed and the material property changes due to variations in temperature are small. It is seen that the changes in the time constants of the disc hub and the diaphragm follow a similar pattern. A small difference in the pattern of these two sections arises because in this arrangement the diaphragm consists of two surfaces perpendicular to the axis of rotation, while the hub has an additional surface parallel to the axis of rotation, the latter affecting the average magnitude of the heat transfer coefficient for this section. A different pattern can be observed in the time constants for the disc outer section and the blades. Here the two sections are subjected to the same fluid flow. Also it was found that although different correlations were used to evaluate the heat transfer coefficients, these were of similar magnitudes. This explains the similar behaviour of these two sections. The casing on the other hand is influenced by two fluid flows and so exhibits a pattern peculiar to itself.

Table 1.

Stage 5, Two-spool Bypass Engine-H.P. Compressor.

Time constants during an acceleration at sea level, static.

Time in transient	2s	6s	10s
Disc hub	108s	66s	37s
Disc diaphragm	38s	17.8s	9.6s
Disc outer section	12.7s	9.0s	5.5s
Blade	2.3s	1.6s	1.0s
Casing	10.4s	7.6s	4.8s

Table 2.

Stage 5, Two-spool Bypass Engine-H.P. Compressor.

Time constants during a deceleration at sea level, static.

Time in transient	3s	6s	10s
Disc hub	60s	55s	62s
Disc diaphragm	21s	20s	23s
Disc outer section	13.1s	15.0s	26.0s
Blade	2.0s	2.3s	3.5s
Casing	6.1s	4.9s	12.5s

3. SIMPLIFIED METHOD TO BE INCLUDED IN TRANSIENT PROGRAM

It would be excessively cumbersome to include each individual blade row of each compressor into the transient program for the engine. The use of single "equivalent" stages instead of complete components would be highly desirable. A single equivalent stage can give satisfactory heat transfer rates (3). A single equivalent stage has therefore been developed for the H.P. Compressor described in the previous section, with the aim of making simple predictions of tip clearance changes and associated efficiency changes. The single equivalent stage uses averaged dimensions of the 12 stages and averaged material properties of specific heat, thermal expansion coefficient, Young's modulus and density. A further simplification of the modelling of the disc is described below.

3.1. Thermal Growth

The subdivision of the equivalent disc as described in Section 2 is retained for the calculation of thermal effects. As in the previous more complex model the various rotor sections are assumed individually to have infinite thermal conductivity, material properties are assumed to be constant throughout the relevant temperature range, although air and gas properties are still calculated as a function of temperature. The process ensuring continuity of the radial dimension of the disc section interfaces however is eliminated. Instead an average disc temperature is found by using Eq. (5).

$$T_{av} = \Sigma V_j T_j / \Sigma V_j \tag{5}$$

The thermal expansion of the "characteristic" disc is thus obtained. It will be shown later that this approximation provides adequate accuracy.

With regard to the thermal growths of the blades and the casing, the method of calculation is the same as described in the previous section.

3.2 Centrifugal Growth

3.2.1 Disc. The centrifugal growth is simplified for this model by calculating the expansion of a disc of the same inner and outer radii and of a uniform thickness equal to the minimum diaphragm thickness. To check the validity of this simplification, growths were compared with those obtained from finite element analysis. It was found that the introduction of the same factor of 1.3 as in section 2.2.1 gave satisfactory agreement. The resulting expression for centrifugal growth of the disc is given in Eq. (6), where the first term in brackets represents the pull of the blades on the disc.

$$\Delta R_d = 1.3 R_o \omega^2 \left[\frac{\Sigma M_b}{l} \left(\frac{R_o^2 + R_i^2}{R_o^2 - R_i^2} \right) + \rho_d \left(R_i^2 (3+\sigma) + R_o^2 (1-\sigma) \right) \right] \tag{6}$$

where the characteristic length, l, is the diaphragm thickness.

In Section 2.2 pressure effects were found to be very small, so they are ignored in this simplified analysis.

3.2.2 Blades. The centrifugal growth of the blades is calculated as described in Section 2.2.2. Hence Eq. (7) is obtained from integral calculus.

$$\Delta H_b = (R_{bt}^3/3 + R_h^3/6 - R_{bt}^2 R_h/2) \rho_b \omega^2 / E_b \tag{7}$$

3.3 Efficiency Loss

The component being considered is simplified to a single-stage equivalent compressor or turbine (4). The tip clearance at that condition is estimated as indicated above, and the efficiency reduction relative to zero tip clearance is found from the relation (13):

$$\Delta\eta = \frac{0.7\lambda\psi}{\cos\beta_m}\left[1 + 10\left(\frac{\varphi\lambda A}{\cos\beta_m}\right)^{0.5}\right] \qquad (8)$$

As an illustration, the efficiency changes in the H.P. Compressor of the Two-spool Bypass Engine, relative to zero tip clearance, have been calculated during a sea level acceleration transient using the simplified procedure described above. The results, labelled "simplified" are shown in Fig. 4. For comparison, the predictions of the single equivalent stage model using the more precisely calculated disc expansion, as described in Paragraph 2.2.1, are shown alongside, marked "one stage". The efficiency changes that are predicted by treating all twelve stages individually are also given, labelled "engine". It can be seen that the results of the two simplified methods are in close agreement with the results from the stage by stage analysis, discrepancies for the "simplified" model never exceeding 0.3 per cent of efficiency. The "simplified" procedure has therefore been adopted as satisfactory for use in the engine transient program.

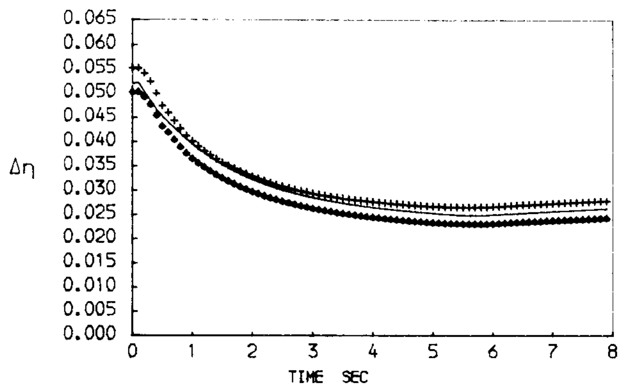


FIG. 4
EFFECTS OF CLEARANCE MOVEMENTS ON H.P. COMPRESSOR EFFICIENCY
EFFICIENCY LOSS COMPARED TO ZERO CLEARANCE
MODELS. ENGINE + ONE STAGE ◆ SIMPLIFIED—

The next step in the analysis is the calculation of the efficiency loss compared to the stabilised value of clearance at that particular speed and inlet conditions. Stabilised clearances at the desired conditions are evaluated using the same methods and hence stabilised values of effiency loss due to clearance openings are obtained. A difference in efficiency is found by subtracting the stabilised efficiency loss from the transient efficiency loss, and the efficiency of the component obtained from the characteristics may then be modified accordingly. This difference is shown in Fig. 5. It is seen that in this compressor, the changes in efficiency are very small.

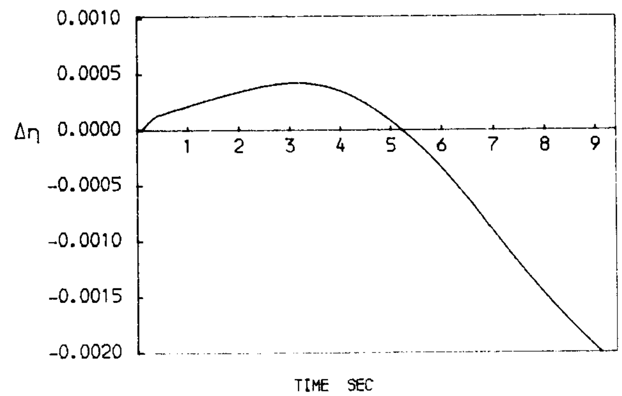


FIG. 5
EFFECTS OF CLEARANCE MOVEMENTS ON COMPRESSOR EFFICIENCY
EFFICIENCY CHANGE DUE TO DIFFERENCE BETWEEN
TRANSIENT AND STABILISED CLEARANCES

4. APPLICATION TO THE TWO-SPOOL BYPASS ENGINE

4,1 H.P. Compressor

The transient program of reference (3) has been extended to include the models described. By means of these the clearance of the "simplified" equivalent single-stage of the H.P. Compressor is calculated at each time step both in the transient and for steady running at that speed and compressor conditions. The clearances during the transient predicted by this simplified model are shown in Fig. 6. The predictions of the single-stage equivalent model using the more precise expansion (Paragraph 2.2.1) are also shown on this Figure (labelled "one stage"), confirming that the "simplified" model (which the engine transient program uses) and the "one stage" model are in good agreement.

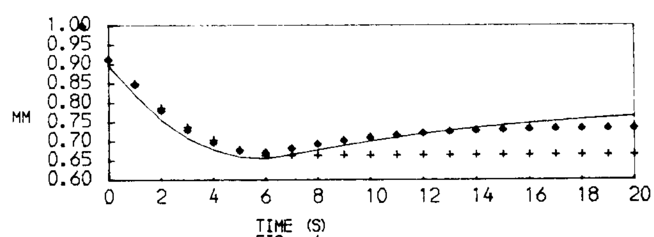


FIG. 6
EQUIVALENT STAGE H.P. COMPRESSOR CLEARANCES
TRANSIENT CLEARANCES —ONE STAGE ◆ SIMPLIFIED MODEL
+ SIMPLIFIED MODEL STEADY RUNNING CLEARANCES

The modified efficiency is then used for calculating the next incremental acceleration in the transient. As indicated on Fig. 5, the changes in efficiency of the compressor of this engine are very small, and on average give improved efficiencies. Their effects on the acceleration are slight, reducing acceleration times by less than 1 per cent.

4.2 H.P. Turbine

The models for the blade/disc arrangement can accommodate both shrouded and unshrouded blades - in the former case the centrifugal expansions are greater due to the weight of the shroud. Also, the thermal model for the blades can incorporate the cooling arrangement, using the simple relation:

$$T_{mf} = T_g - \varepsilon \ (T_g - T_c) \qquad (9)$$

where T_{mf} is the final metal temperature if that transient condition were maintained (this is the temperature which can be regarded as driving the heat transfer) and ε is the cooling "effectiveness" (a value of 0.6 being typical for a blade cooled by inner passageways).

Predicted tip clearances in the H.P. Turbine during a sea level acceleration are shown in Fig. 7. The predictions for two alternative blading arrangements are presented. In the first, the blades are unshrouded and uncooled (labelled "simple") while in the second they are both shrouded and cooled (labelled "S & C"). Both arrangements start with the same cold clearance. It can be seen that the additional expansion produced by the weight of the shroud is approximately balanced by the reduction in thermal expansion that results from cooling the blades.

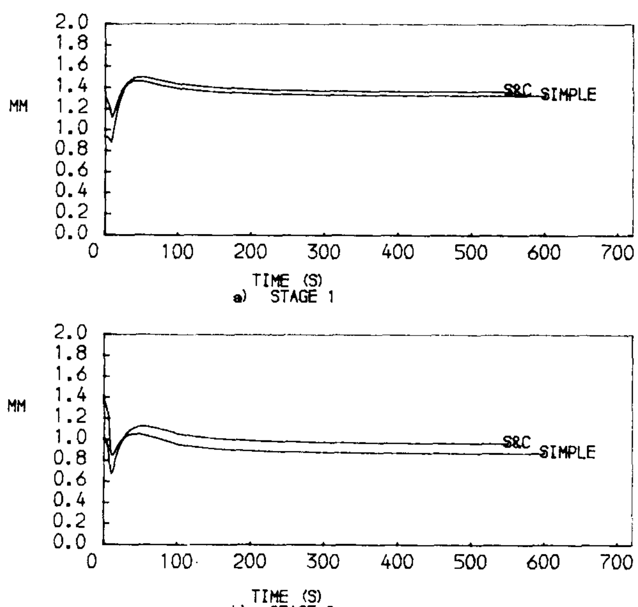


FIG. 7
CLEARANCES OF A TWO SPOOL BYPASS ENGINE H.P. TURBINE
SIMPLE, SIMPLE BLADE S&C, SHROUDED AND COOLED BLADE

In the real Engine the blades and nozzle guide vanes are shrouded and the early rows are cooled. For this scheme, the tip clearance movements have been determined for each row. These movements were transferred into efficiency changes using the correlation of Eq. (8), with the suggested modification (14) of reducing the efficiency changes by 80 per cent for shrouded nozzle guide vanes. It is estimated that a smaller reduction, say 50 per cent, should be applied to the efficiency changes for shrouded blades. The resulting efficiency values during the transient are only slightly different from the corresponding steady-running values, the difference being in the direction of improved efficiency. Inclusion in the transient program produced very small reductions in acceleration times - less than 1 per cent. None the less, in view of the assumptions that have been made in the study of the turbine, further work should be carried out to improve the analysis.

In view of the small effects resulting from inclusion of transient tip clearance changes in the H.P. Compressor and H.P. Turbine, it was considered unnecessary to make similar analyses in the L.P. Compressor and L.P. Turbine, which are subjected to less rigorous temperature changes.

4.3 Procedure for Estimation of Seal Clearance Movements

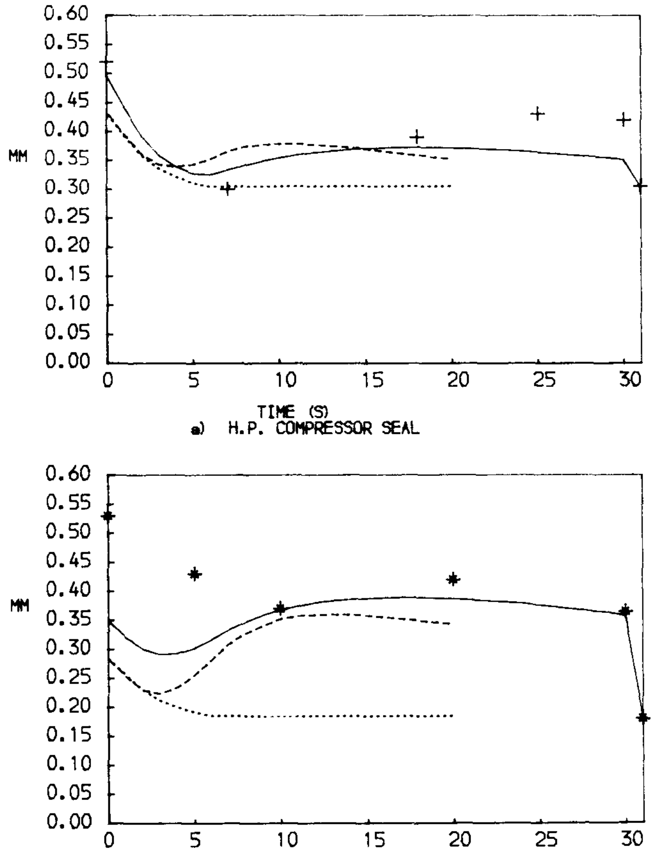
The methods and model used for tip clearance can be adapted for making estimates of seal clearance movements during transients. In the Two-spool Bypass Engine two important seals are the H.P. Compressor 12th Stage Outer Seal and the H.P. Cooling Air Seal on the H.P. 1 Turbine Disc.

4.3.1 H.P. Compressor 12th Stage Outer Seal. The movements of this Seal have been studied previously by Lim (15) using finite difference methods. The predictions of Lim and of the present methods are compared on Fig. 8(a). The agreement is sufficiently close to allow the present model to be used for seal clearance predictions during transients. It is seen that during most of the acceleration speed transient, seal clearances exceed their maximum speed stabilised values by more than 30 per cent.

4.3.2 H.P. Cooling Air Seal on H.P.1 Turbine Disc. The movements of this Seal have been studied previously by finite difference methods (8). The results of reference (8) are compared to the present methods in Fig. 8(b). It is seen that the seal clearances during the acceleration speed transient exceed the maximum speed stabilised clearance by about 100 per cent.

4.3.3 Effects of Seal Clearance Movements in Transients. In the early programs for predicting the acceleration rates of gas turbines it would probably be assumed that cooling and bleed air flows remained a constant fraction of the core air flow during the transient. However it has been illustrated that seal clearances during the speed transient can be very much higher than the maximum speed or the design clearances. Consequently these cooling and bleed flows, expressed as fractions, will exceed the design fractions.

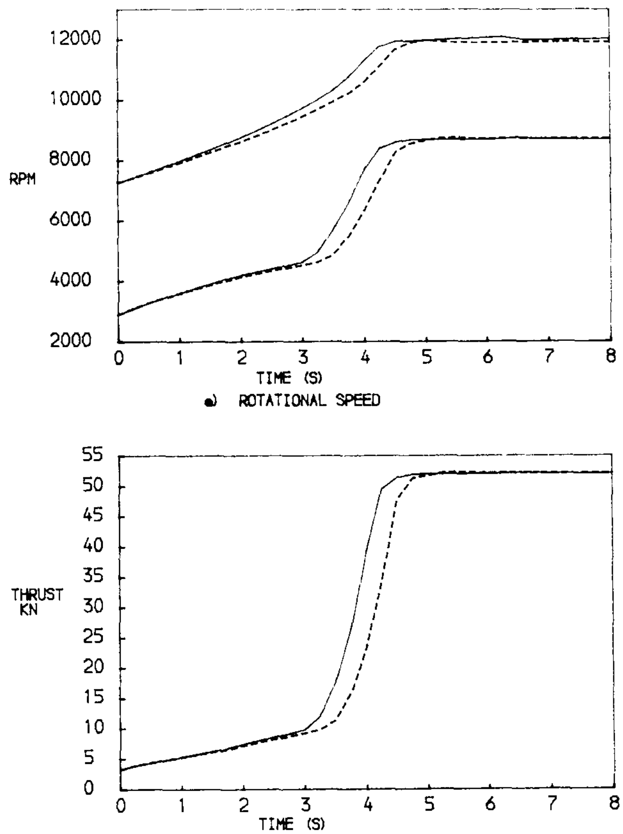
Allowances for the clearance movements of the H.P. Compressor 12th Stage Outer Seal and of the H.P. Cooling Air Seal on the H.P. 1 Turbine Disc have been included in the engine transient program, and the results for a sea level acceleration are illustrated in Fig. 9. For the acceleration, the fuel flow, as a non-dimensional group, was scheduled as a function of the H.P. Compressor pressure ratio. It is seen that the additional loss of air through these seals during the transient causes a significant increase - about 6 per cent - in the acceleration times.



a) H.P. COMPRESSOR SEAL

b) H.P. TURBINE SEAL

FIG. 8
CLEARANCES OF TWO SEALS OF THE TWO SPOOL BYPASS ENGINE
WITH MIXED EXHAUSTS DURING AN ACCELERATION
—FULL MODEL TRANSIENT CLEARANCES
SIMPLE MODEL CLEARANCES, TRANSIENT - -, STABILISED···
ANALYSIS OF, REF. 15 +, REF. 8 ◆



a) ROTATIONAL SPEED

b) THRUST

FIG. 9

EFFECTS OF SEAL CLEARANCES ON THE PREDICTED PERFORMANCE
OF A TWO SPOOL BYPASS ENGINE DURING AN ACCELERATION
—NO SEAL EFFECTS -- EFFECT OF THE TWO SEALS

4.4 Decelerations

In decelerations the "heat soak" period is much longer because the reduced rotational speed at the end of the transient results in much lower values of the heat transfer coefficients. All the components were studied in both accelerations and decelerations, and no rubs were encountered.

4.5 Control of Tip Clearances during Transients

When considering, for example, the transient response of the components (disc, blade, casing) of the H.P. Compressor, it is realised that the responses of the discs would have been different, hence the transient tip clearances would have been different, if the air in which the discs rotate had been drawn from, say, a later stage of the H.P. Compressor than at present. This realisation offers the designer another method of controlling tip clearance changes.

4.6 Computing Times

The increase in computing times is not large for the inclusion of the model described in a typical engine transient program. For example, computing times were about 15 per cent longer when the model was used to calculate and apply clearance paths in three different components during an engine transient.

5. CONCLUSIONS

A simple model has been developed for calculating clearances of seals and blade tips, and for determining their effects on engine response during a transient. It also enables the study of the effects of changing materials in the engine and of altering the pattern of cooling. The procedure has been used to show that changing the cooling of compressor discs, for example, alters the tip clearance response.

Seal clearances have a larger effect than compressor blade tip clearances on the transient performance of the Two-spool Bypass Engine considered.

Further work is being carried out to assess more accurately the turbine tip clearances and their effects during a transient.

The increase in computing time is not large, amounting to about 15 per cent when the model was used to calculate and apply clearance paths in three components during the transient.

6. ACKNOWLEDGEMENT

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