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Thermal Influences in Gas Turbine Transients — Effects of Changes in Compressor Characteristics

N. R. L. MACCALLUM

Senior Lecturer,
Dept. of Mechanical Engineering,
University of Glasgow,
Glasgow, Scotland

During transients of axial-flow gas turbines, the characteristics of the compressor are altered. The changes in these characteristics (excluding surge line changes) have been related to transient heat transfer parameters, and these relations have been incorporated in a program for predicting the transient response of a single-shaft aero gas turbine. The effect of the change in compressor characteristics has been examined in accelerations using two alternative acceleration fuel schedules. When the fuel is scheduled on compressor delivery pressure alone, there is no increase in predicted acceleration times. When the fuel is scheduled on shaft speed alone, the predicted acceleration times are increased by about 5 to 6 percent.

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ABSTRACT

During transients of axial-flow gas turbines, the characteristics of the compressor are altered. The changes in these characteristics (excluding surge line changes) have been related to transient heat transfer parameters, and these relations have been incorporated in a program for predicting the transient response of a single-shaft aero gas turbine. The effect of the change in compressor characteristics has been examined in accelerations using two alternative acceleration fuel schedules. When the fuel is scheduled on compressor delivery pressure alone, there is no increase in predicted acceleration times. When the fuel is scheduled on shaft speed alone, the predicted acceleration times are increased by about 5 to 6 per cent.

NOMENCLATURE

A = flow area
c = chord
c_p = specific heat at constant pressure
F = ratio of heat transfer to fluid to work transfer from fluid in an element of a compressor or turbine
h = heat transfer coefficient
k = thermal confuctivity of fluid
ℓ = distance from leading edge
m = index of non-adiabatic compression or expansion
M = mass flow rate of fluid
N = rotational speed of shaft
P = pressure
Pr = Prandtl Number
Q = heat flux to fluid in compressor or turbine
T = temperature
y = blade height
γ = isentropic index
η = small-stage, or polytropic efficiency
μ = viscosity

Subscripts

1,2 = inlet to, outlet from compressor
ave = average
lam = laminar
th = throat
turb = turbulent

INTRODUCTION

It is important that reliable methods are developed for the prediction of the transient behaviour of gas turbines. For example, one wants to be able to predict the speed response and the thrust response of an aero gas turbine when a given acceleration fuel schedule is applied.

The earliest programs for the prediction of the transient performance used equilibrium characteristics for the components, and ignored heat transfer effects. However these simple programs seriously underpredicted the times required for the speed and thrust responses - Thomson (1)[1] quotes underpredictions of 20 to 30 per cent for acceleration times.

The discrepancies between these simple predictions and the observed acceleration times have been attributed to one or more of the following factors:

(a) heat absorptions in the compressor(s) and turbine(s)

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

(b) heat absorption in the combustion chamber metal
(c) incorrect tip clearances in compressor(s) and turbine(s) during transient
(d) incorrect seal clearances during transient
(e) lag in combustion process.

Fawke and Saravanamuttoo (2), after making reasonable simplifying assumptions, have included factor (a). Thomson (1) makes an estimate of the fuel flow equivalent to the difference between the increased fuel flow and the observed increases in kinetic energy and exhaust energy flux. This is then deducted from the fuel flow in the acceleration schedule used in the prediction program. Thomson also indicates methods of predicting heat transfer to combustion chambers, tip clearance changes and combustion lag - factors (b), (c) and (e). Maccallum (3) has illustrated how seal clearance changes in transients may be predicted. Excessive openings of critical seals may cause significant increases in cooling air flows, which will result in proportionately less gas passing through the higher pressure stages of the turbine. Also, the effects of these flows, if they return to the main flow in the turbine, may have to be considered (4) to (8).

One factor which has not previously been considered is:

(f) the change in the compressor characteristic due to the transient heat transfer.

A theoretical investigation of this aspect is reported in the present paper. In addition, there is an assessment of the heat transfer correlations, and of the simplifying geometric assumptions that have previously been used.

HEAT TRANSFER IN COMPRESSORS AND TURBINES

Heat Transfer Correlations

For flows in compressors and turbines, Fawke and Saravanamuttoo (2) have used the Colburn Equation

$$\frac{hy}{k} = 0.023 \left(\frac{My}{A\mu}\right)^{0.8} (Pr)^{0.4} \qquad (1)$$

This equation is valid for developed flow. However the flow over the aerofoil of a blade in a compressor or turbine will be far from developed. A new boundary layer has to be started at the leading edge of each blade. Consequently it might be more reasonable, say for the compressor, to use the flat plate correlations for developing laminar and turbulent layers. Thus for a laminar layer of length ℓ, the average heat transfer coefficient is given by

$$h_{lam} = 0.664\ k\ (Pr)^{0.333} \left(\frac{M}{A\mu\ell}\right)^{0.5} \qquad (2)$$

and for a turbulent boundary layer

$$h_{turb} = 0.037 \left(\frac{M}{A}\right) c_p \left(\frac{M\ell}{A\mu}\right)^{-0.2} (Pr)^{-0.667} \qquad (3)$$

In a compressor, it might be assumed that the boundary layer on the pressure surface was turbulent throughout its length, while that on the suction surface was initially laminar, becoming turbulent. An average heat transfer coefficient might then be given by

$$h_{ave} = 0.25\ h_{lam} + 0.75\ h_{turb} \qquad (4)$$

Comparing the predictions of equations (1) and (4) indicates that the values of coefficient given by equation (1) are typically at least 30 per cent lower than those given by equation (4).

The above expressions have ignored the influence of turbulence in the main flow. Turbulence intensities are known to be high (8 to 10 per cent). Two recent studies on typical turbine (not compressor) blades show that such levels increase the average heat transfer coefficient. Brown and Burton (9) indicate a 60 per cent increase in average heat transfer coefficient for an increase in turbulence intensity from 1.8 to 8.6 per cent (intensity defined by root mean square of fluctuations, normalised by free stream velocity). Bayley and Milligan (10) show increases of similar nature but depending on the frequency of the turbulence fluctuations. The range of increase in coefficients was from 20 per cent to 160 per cent for turbulence intensities varying between 14 and 48 per cent. It is suggested therefore that a reasonable practice to adopt in calculating heat transfer coefficients in a compressor is to use equation (4) with the coefficient increased by 60 per cent. This practice has been adopted in the present work.

For turbines, fortunately there is more experimental data available giving average coefficients on blades. For example Halls (11) quotes

$$\frac{hc}{k} = 0.235 \left(\frac{Mc}{A_{th}\mu}\right)^{0.64} \qquad (5)$$

It is suggested that where experimental correlations of this nature are available, these should be used. In the present work, heat transfer coefficients were calculated from the above equation. It is interesting to note that the predictions of equation (4), when applied to typical turbine blades, give coefficients which have to be increased by about 80 per cent to bring them into line with the experimental correlation of equation (5). This magnitude of increase provides support for the increase of 60 per cent recommended for the compressor.

Representation of Blades during Temperature Transients

Several models have been suggested (12) for representation of blades during temperature transients. Each of these is adequate for speed transients, and so it is recommended that nothing more sophisticated than a simple "unfinned" model is required, in which the aerofoils and platforms are represented by separate plates of equivalent total surface areas and thermal capacities. To make some allowance for the thermal contact between platforms and the devices on which they are mounted - disks or casings - in the present work the platform thermal capacities have been increased by 50 per cent.

CHANGES IN COMPRESSOR CHARACTERISTICS DUE TO HEAT TRANSFER

Having established heat transfer rates, two effects are now examined. The first results from changes in the development of the boundary layers on the aerofoils, particularly on the suction surfaces. The second is due simply to the alteration in density of the air at a plane, resulting from the heat transfer from, or to, the air in the section of the compressor up to that plane. This results in an alteration in the local ratio of axial velocity to blade speed.

Effect of heat transfer on Boundary Layers

In general terms, heat transfer from a wall to a boundary layer increases the rate at which the layer develops. The effect is more severe in the presence of an adverse pressure gradient. The transition region is moved upstream and, in cases where the boundary layer separates, heat transfer generally accelerates the process (13, 14). This advancement of the separation point was observed experimentally, and was also predicted by a procedure based on a modified momentum integral equation.

These findings have been applied (15) to the flow in an axial air compressor. It is suggested there that heat transfer from the blades to the air has little effect on the development of the layer on the pressure surfaces. However on the suction surfaces the prediction method shows under certain conditions a significant increase in the displacement thickness in the vicinity of the trailing edge when the heat transfer is to the air. This is illustrated qualitatively in Fig. 1. In this case the angle of departure of the flow on the suction surface will be increased (angles measured from the axial direction) due to the more rapid displacement thickness development. It was

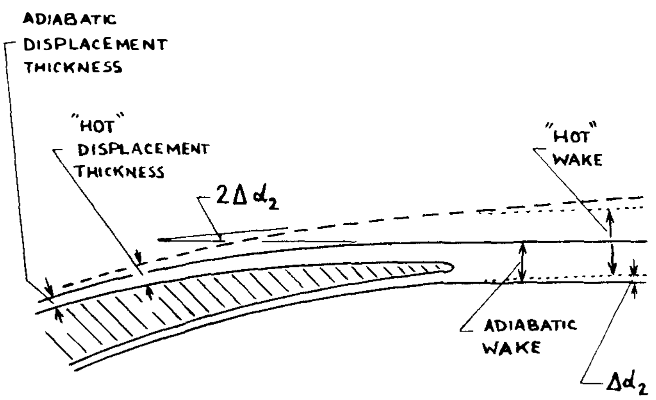

Fig. 1 The effect of heat transfer from a "hot" compressor blade on boundary layer development (Ref.15)

then assumed that the averaged increase in the leaving angle from a "hot" blade will be one half of the increase in the inclination of the line through the boundary of the displacement thickness on the suction surface. Thus for a "hot" blade there will be a reduced deflection across the blade row. If the blade is "cold" there is the possibility of increased deflection. These changes are predicted to take place only if the flow is near to separating, i.e. when there is significant positive incidence. In reference (15) these effects were incorporated in a program for predicting the compressor characteristics. This program also assumed that the wake losses in a non-adiabatic situation corresponded to the losses in the adiabatic case where the incidence to a blade row produced the same average leaving angle as in the non-adiabatic case. It was predicted that these boundary layer changes cause the constant speed pressure characteristics of the compressor to be displaced, and also cause movement of the surge line.

Effect of density change due to heat transfer

This density change due to heat transfer alters the ratio of axial velocity to blade speed, and hence alters the working points of the stages. The effect this has on the compressor constant speed pressure characteristics and surge line has been predicted in references (16) and (15).

Changes in Compressor Characteristics

In the present work, a sixteen-stage axial flow compressor of a single-shaft aero gas turbine, of maximum pressure ratio 9.5, was selected for studying these effects.

The thermal response of the compressor to an acceleration was first predicted, each blade row of the compressor being treated individually. The temperature differences between the air and the aerofoil surfaces, and the heat fluxes, were then included in the program for predicting the performance of the compressor. Predicted characteristics are shown on Figs. 2, 3 and 4 for the conditions existing during the acceleration when the compressor is at 85, 90 and 95 per cent respectively of the maximum speed. The predicted characteristics in which both boundary layer changes and density changes due to heat transfer are accounted for are represented by chain dotted lines. The corresponding characteristics when only the boundary layer changes are allowed for are indicated by the dashed lines and the adiabatic characteristics are shown by the solid lines. It is seen that in all cases there is a shift of the speed line, the major contributor to this displacement being the density change resulting from heat transfer. The predicted adiabatic surge line is shown, as are the predicted surge terminations of the non-adiabatic characteristics - in the present work the definition adopted for the location of the surge position is the point where the pressure characteristic has a maximum.

Predicted characteristics at speeds of 85 and 90 per cent during decelerations from 100 per cent speed are also shown on Figs. 2 and 3.

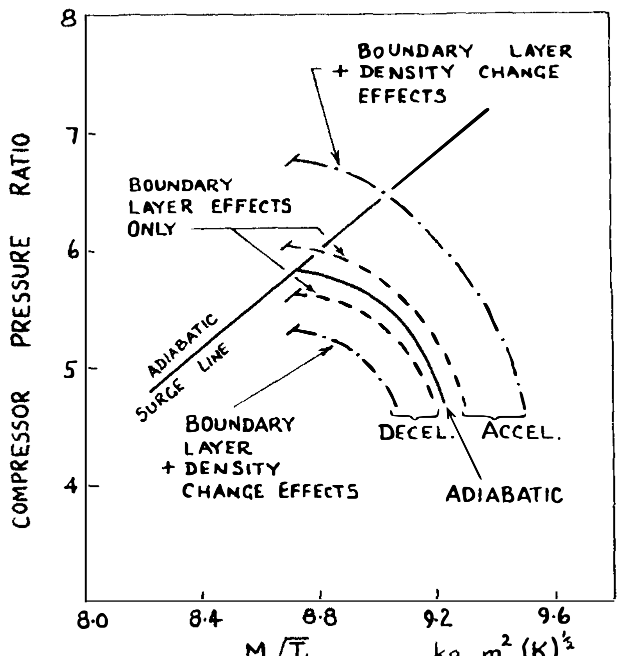

Fig. 2 Predicted effects of heat transfer on compressor characteristics during acceleration and deceleration at sea level - $N/\sqrt{T_1}$ = 0.85 x take-off $N/\sqrt{T_1}$

The movements of the surge lines are important, and one case has been discussed in some detail in

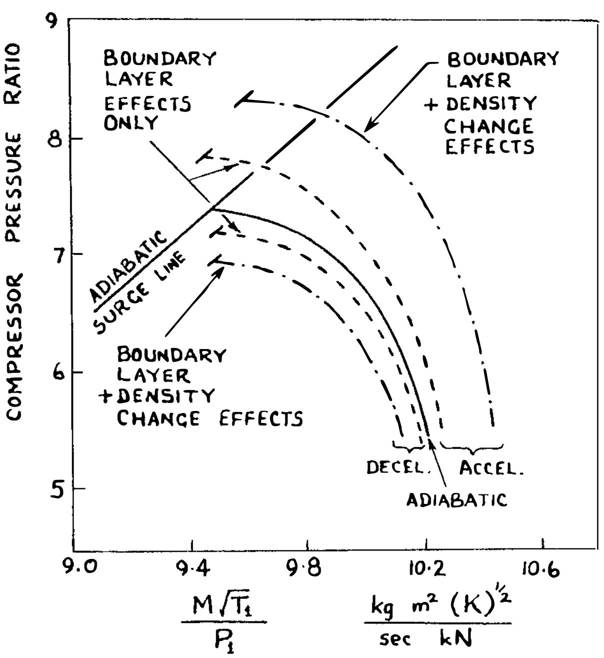

Fig. 3 Predicted effects of heat transfer on compressor characteristics during acceleration and deceleration at sea level - $N/\sqrt{T_1}$ 0.90 x take-off $N/\sqrt{T_1}$

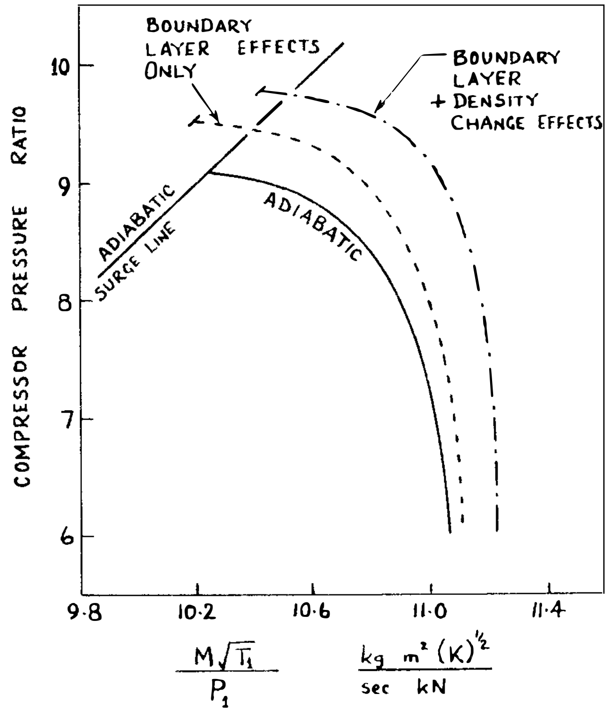

Fig. 4 Predicted effects of heat transfer on compressor characteristics during an acceleration at sea level - $N/\sqrt{T_1}$ 0.95 x take-off $N/\sqrt{T_1}$

reference (15). In the present paper this is not discussed further, apart from stating that during an acceleration of a "cold" engine the surge line is beneficially moved by the heat transfer. During a deceleration, deterioration of the surge line is predicted.

The movement of the constant speed line is important to the present investigation of transient response. Examination of the results shown in Figs. 2 to 4 suggests that the transient characteristics may be regarded as being similar to the equilibrium characteristics when the compressor is running adiabatically at a speed different by, say, ΔN from the actual speed, N, of the compressor. Thus the "aerodynamic" or effective speed of the compressor is $(N + \Delta N)$. The correlation of these "speed changes" with relevant transient parameters is discussed below.

RELATION OF CHANGES IN CONSTANT SPEED CHARACTERISTICS TO TRANSIENT PARAMETERS

The changes due to the two effects - boundary layer and density changes - are considered separately.

Firstly, estimates were made of the changes in effective compressor speed, ΔN, corresponding to the displacements of the constant speed characteristics shown in Figs. 2, 3 and 4 due to the boundary layer effects. Grant (13) has shown that, for a given air temperature, changes in boundary layer development - for example movement of the separation point - are proportional to the temperature difference between the surface and the air. Therefore the changes in speed, ΔN, found above have been plotted, in normalised form, against this temperature difference, normalised by the average temperature in the compressor. These results are shown in Fig. 5. The correlation is not good, but bearing in mind that the movements due to boundary layer effects are small,

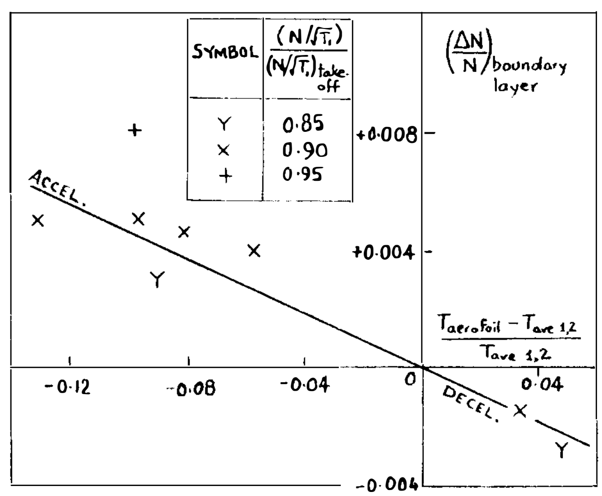

Fig. 5 Predicted changes in "effective" speed of compressor during transients due to boundary layer effects

the following relation, illustrated by the solid line on Fig. 5 was adopted -

$$\left(\frac{\Delta N}{N}\right)_{\text{boundary layer}} = -\,0.005\left(\frac{T_{\text{aerofoil}} - T_{\text{air}}}{T_{\text{ave }1,2}}\right) \qquad (6)$$

In a similar manner, estimates were made of the changes in effective compressor speed due to the density changes resulting from heat transfer. These are plotted on Fig. 6 against the heat flux per unit mass of air. A much better correlation is found for these changes, represented by the expression

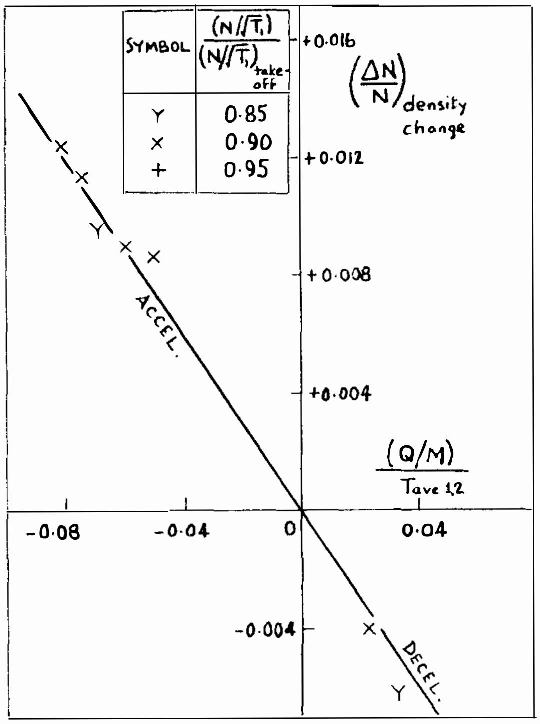

Fig. 6 Predicted changes in "effective" speed of compressor during transients due to density change resulting from heat transfer.

$$\left(\frac{\Delta N}{N}\right)_{density\ change} = -0.15 \left(\frac{Q}{M}\right) \frac{1}{T_{ave\ 1,2}} \qquad (7)$$

These expressions have been incorporated in the gas turbine transient program discussed later.

NON-ADIABATIC COMPRESSIONS AND EXPANSIONS

If the ratio, F, of the heat transfer to the air to the work transfer from the air in an element of a compressor remains constant along the compressor, then it has been shown (17) that the index of the actual compression path, m, is related to the isentropic index by

$$\frac{m-1}{m} = \frac{(1-F)(\gamma-1)}{\eta\,\gamma} \qquad (8)$$

The corresponding relation for turbines is

$$\frac{m-1}{m} = (1-F)\,\eta\,\frac{\gamma-1}{\gamma} \qquad (9)$$

This simple analysis has been included in the gas turbine transient program discussed later. The analysis produces results virtually identical to those given by the method adopted by Thomson (1).

SIMPLIFIED THERMAL REPRESENTATION OF COMPRESSORS AND TURBINES

Ideally, in temperature transients, each row of blades in the compressor, or turbine, should be treated separately as blade dimensions vary from row to row. However this greatly complicates the inclusion of thermal effects in gas turbine transient programs. It would be much more satisfactory if an adequate representation could be provided by a single representative row of blades, or failing this, say two or four representative rows. The row, or rows would have the appropriate total surface areas of aerofoils and platforms, and the appropriate thermal capacities. The accuracy of some form of simplified representation has been assessed in the present work by considering the acceleration of the sixteen-stage axial compressor previously discussed. The predicted total fluxes of heat, per unit mass of air, to the metal of the compressor are compared on Fig. 7 for the three cases of complete row by row representation, representation by four characteristic rows and representation by a single characteristic row. The four-row model is very satisfactory, discrepancies being typically less than 3 per cent. The single row representation is less accurate however and heat transfer rates are overpredicted, particularly during the last 2 seconds of the speed transient, when the discrepancies range from 10 to 15 per cent.

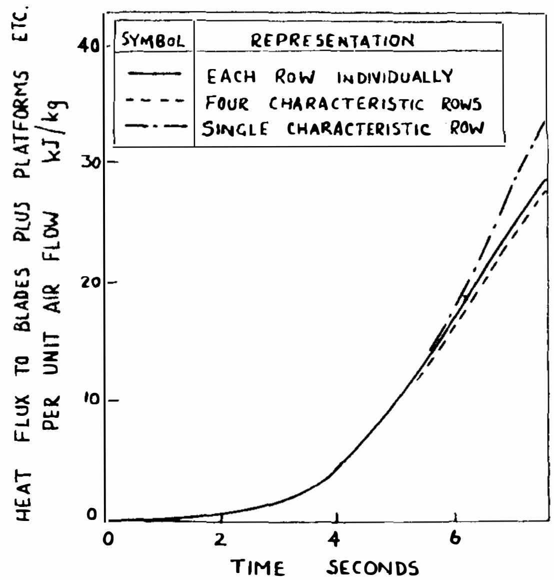

Fig. 7 Validity of simplified representation of a sixteen-stage axial compressor in a temperature transient - predicted thermal response to an acceleration at sea level.

In view however of the other uncertainties in the inclusion of heat transfer in gas turbine transient prediction procedures, it is considered that the single blade row representation is adequate for most purposes. The transient program used below utilises single-row representation.

PREDICTED EFFECTS OF HEAT TRANSFER IN GAS TURBINE TRANSIENT RESPONSES.

A program was prepared for predicting the transient response of a single-spool aero gas turbine engine. The engine used the sixteen-stage axial compressor whose transient characteristics have been analysed in the previous paragraphs.

The speed transient program incorporated the facilities discussed above for including non-adiabatic compressions and expansions and also allowed for the displacement of the constant speed lines on the compressor's characteristics. Single-row representation was used for calculating the heat transfer rates in the compressor and turbine.

The transient response has been calculated for accelerations at sea level using two alternative acceleration fuel schedules.

In the first the fuel flow is a function of the compressor delivery pressure, as illustrated in Fig. 8,

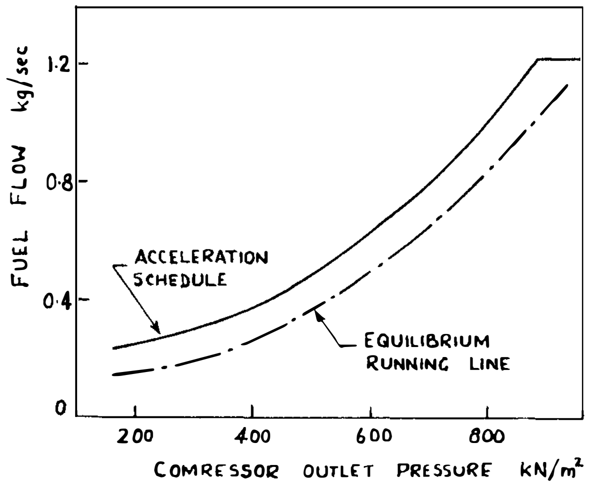

Fig. 8 Acceleration fuel schedule "A" - function of compressor outlet pressure.

and in the second the fuel flow is a function of the engine shaft-speed as given in Fig. 9.

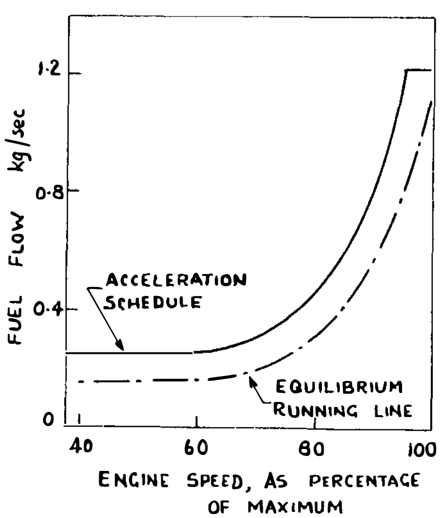

Fig. 9 Acceleration fuel flow schedule "B" - function of engine speed, for acceleration at sea level.

The predicted responses to the acceleration fuel schedule based on pressure are given in Fig. 10. Two starting speeds are considered - 38 per cent and 50 per cent of maximum speed. Fig. 10 shows the predicted responses for the following assumed situations:

(a) adiabatic compression and expansion,
(b) with heat transfer to compressor, but ignoring movement of characteristic,

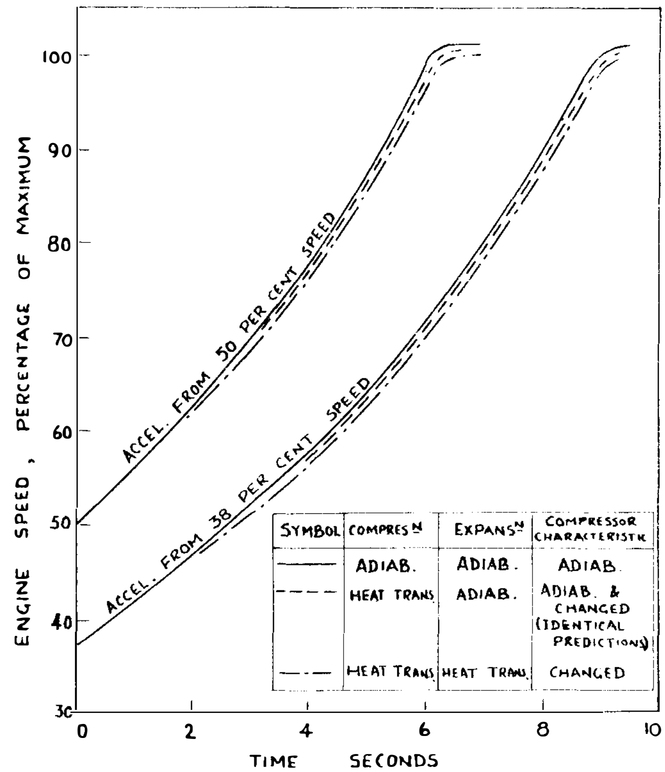

Fig. 10 Predicted accelerations at sea level using fuel schedule "A" - function of compressor outlet pressure.

(c) with heat transfer to compressor, and including movement of characteristic,
(d) with heat transfer to compressor and to turbine, and movement of compressor characteristic accounted for.

The results show that the inclusion of heat transfer causes a slight lengthening of the predicted acceleration times, but this amounts to only about 5 per cent. Inclusion of the change in compressor characteristics produces no alteration in the predicted acceleration times.

The predicted responses to the acceleration fuel schedule based on shaft speed are given in Fig. 11. Inclusion of heat transfer in the compressor and turbine again leads to an increase in the predicted acceleration times, though the change is only about 3 to 4 per cent. However there is a significant change when the alteration in compressor characteristics is introduced. This increases the predicted acceleration times by about a further 5 to 6 per cent.

The magnitudes of the changes in predicted acceleration times due to the inclusion of heat transfer in the compressor and turbine (but ignoring compressor characteristic change) are similar to those reported by Fawke and Saravanamuttoo (2)

CONCLUSIONS

Transient heat transfers alter the characteristics of axial flow compressors. The changes in characteristics can be approximated by using the concept of changes in effective speed, and these changes can be correlated to heat transfer parameters.

The techniques have been applied to a single-shaft aero gas turbine. When the gas turbine uses an acceleration fuel schedule which is a function solely of compressor delivery pressure, inclusion of the

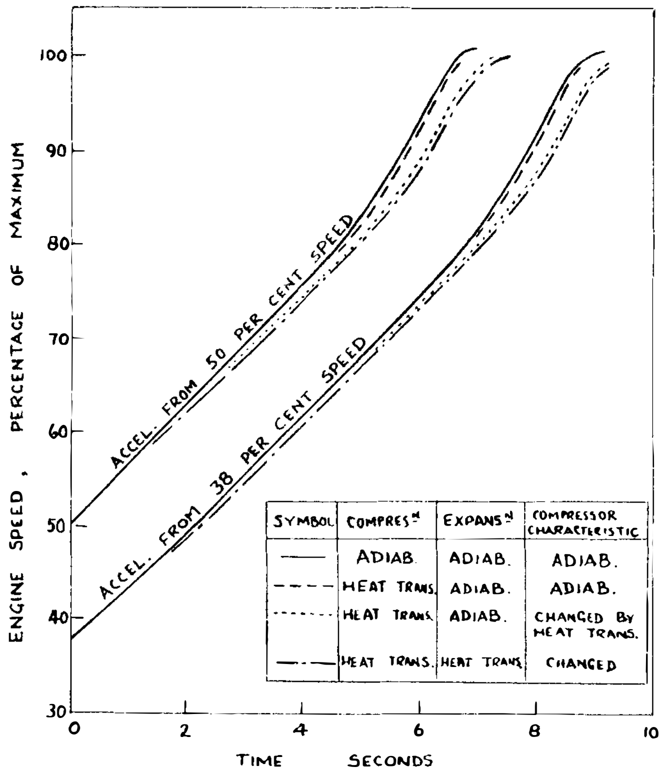

Fig. 11 Predicted accelerations at sea level using fuel schedule "B" - function of engine speed.

compressor characteristic alteration in the engine transient program produces no change in the predicted acceleration times. However when the gas turbine uses an acceleration fuel schedule which is a function solely of shaft speed, the predicted acceleration times are lengthened by 5 to 6 per cent by allowing for the change in the compressor characteristics. In a deceleration, with the fuel scheduled on the shaft speed, the predicted deceleration time would be extended.

Heat transfer correlations have been compared and recommendations are made.

Simplified thermal representations of a compressor have been assessed, and a single-blade representation is adequate for most purposes.

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REFERENCES

1 Thomson, B., "Basic Transient Effects of Aero Gas Turbines," AGARD Conference Proceedings 151, Feb. 1975.

2 Fawke, A.J., and Saravanamuttoo, H.I.H., "Digital Computer Simulation of the Dynamic Response of a Twin-Speed Turbofan with Mixed Exhausts, "Aeronautical Journal, Vol. 77, Sept. 1973, pp. 471-478.

3 Maccallum, N.R.L., "Transient Expansion of the Components of an Air Seal on a Gas Turbine Disc," SAE Paper 770974, 1977.

4,5 Shrivastava, K.D., and Maccallum, N.R.L., "The Effect of a Transversely Injected Stream on the Flow Through Turbine Cascades: Part I - Flow Effects, and Part II - Performance Changes," ASME Papers Nos. 77-GT-87 and 77-GT-88.

6 Aburwin, B.A., and Maccallum, N.R.L., "The Effect of a Transversely Injected Stream on the Flow Through Turbine Cascades: Part III - Influence of Aspect Ratio," to be published in Journal of Engineering for Power.

7 Bindon, J.P., Aburwin, B.A., and Maccallum, N.R.L., "Comparison of Transverse Injection Effects in Annular and in Straight Turbine Cascades", Paper to ASME Gas Turbine Conference, 1979.

8 Aburwin, B.A., and Maccallum, N.R.L., "Vortex Effects resulting from Transverse Injection in Turbine Cascades, and attempts at their Reduction," Paper to ASME Gas Turbine Conference, 1979

9 Brown, A., and Burton, R.C., "The Effects of Free-Stream Turbulence Intensity and Velocity Distribution on Heat Transfer to Curved Surfaces," ASME Paper No. 77-GT-48.

10 Bayley, F.J., and Milligan, R.W., "The Effect of Free-Stream Turbulence upon Heat Transfer to Turbine Blading," AGARD, Propulsion and Energetics Panel Meeting, Ankara, September, 1977.

11 Halls, G.A., "Air Cooling of Turbine Blades and Vanes," Lecture to AGARD, Varenna, Italy, May 1967.

12 Maccallum, N.R.L., "Models for the Representation of Turbomachine Blades during Temperature Transients," ASME Paper No. 76-GT-23.

13 Grant, A.D., "The Effect of Heat Transfer on Boundary Layer Stability, Ph.D. Thesis, University of Glasgow, 1973.

14 Grant, A.D., "The Effect of Heat Transfer on Boundary Layer Stability in Axial Flow Compressors," Heat and Fluid Flow in Steam and Gas Turbine Plant, Instn. mech. Engrs., London, 1974, pp 252-258.

15 Maccallum, N.R.L., and Grant, A.D., "The Effect of Boundary Layer Changes due to Transient Heat Transfer on the Performance of an Axial Flow Air Compressor," SAE Paper 770284, 1977

16 Maccallum N.R.L., "Effects of 'Bulk' Heat Transfers in Aircraft Gas Turbines on Compressor Surge Margins," Heat and Fluid Flow in Steam and Gas Turbine Plant, Instn. mech. Engrs, London, 1974, pp. 94-100.

17 Maccallum, N.R.L., "The Performance of Turbo-Jet Engines during the 'Thermal Soak' Transient," Proceedings of the Instn. mech. Engrs, London, Vol. 184, Part 3G(II), 1969, pp. 23-29