THE 'COMBINE' HIGH TEMPERATURE GAS TURBINE

BY

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ABSTRACT

A combination of the functions of expansion, compression, and/or cooling within the same blading is described, enabling gas entry temperature to be substantially increased. The concept has been demonstrated by previous experimental work on a three-pass axial flow compressor, and provides a convenient means of intercooling, and waste heat recovery. Significantly greater reductions, in specific fuel consumption and engine size, are attainable than from conventional complex gas turbine cycles.

Introduction

It is well known that gas turbines are commonly used as prime movers for aircraft and marine propulsion, as well as for the generation of electrical power. It is also well understood that thermal efficiency and power can be raised by increasing the temperature ratio of the thermodynamic cycle, by raising the maximum temperature at entry to the turbine. Much effort has been directed to this end, by improvements in high temperature alloys and in methods of turbine cooling, by which means it is now possible to operate at gas temperatures some 300–400°C above that of the working levels for alloys currently in use.

Unlike the constant volume prime mover, therefore, in which the working components are exposed to both the hot expansion gas and the cold induced air, the constant pressure gas turbine components are separate, and the turbine is required to meet the maximum cycle temperature which therefore restricts its performance accordingly. However, if the functions of the two processes of compression and expansion can be combined in the same way as for the constant volume cycle, then temperatures at entry to the turbine can be increased.

In concept it would appear contradictory to apply the process of diffusion to an accelerating turbine passage, although use of a compressor in reverse as a turbine provides normal level of efficiency. For the same blading to function well as a turbine and as a compressor, whilst rotating in the same direction, however, requires a compromise design solution where any penalty in efficiency must be more than compensated by an increase in thermodynamic cycle performance. Furthermore the design must allow admission of the hot gases of combustion to the blading, alternately with the cold air of compression.

During the early 1970s a new type of compressor was introduced to meet a specific marine propulsion cryogenic requirement, for which there was no suitable type available. The design used the well-proven principles of multi-stage axial flow, now developed for the aero gas turbine, and the regenerative type which allows the flow to pass through each row of blading a number of times. The concept used the high performance axial flow method for the design, but in place of the full annulus the flow was directed through segmented passages within the annulus, each decreasing in width with increase in fluid density. The process, which is analogous to decreasing blade heights through each stage within the conventional axial flow multi-stage compressor, provided the means of achieving a number of 'multipass' stages in one. After passing through each segment the flow is directed over the tips of the blading and into an adjacent segment, passing when this is required through an intercooler. Patent applications[1] filed in 1972 were granted on the 7th January 1976, and a three-pass experimental machine was successfully tested in the following few years. The design is described in references 1 and 2.

The performance proved to be identical to that predicted by the conventional multi-stage axial flow design method used, which was to be expected provided that the circumferential segments were of sufficient width and the pressure gradients small enough to preserve adequate symmetry. There are, however, intrinsic losses involved in returning the flows after each pass to the adjacent segment.

The procedure of multipassing need not be confined to flows in axial flow compressors, however; indeed, provided that advantages are derived, the multipass axial flow turbine would be less of an aerodynamic performance challenge than the diffusion process of the compressor. There is every reason to expect therefore that the multipass axial flow turbine will at least equal this.

By combining the two functions in a single rotor, and allocating one or two segments to the high temperature turbine expansion, in parallel with the low temperature compressor flow segments, there are advantages of particular significance to the gas turbine as a prime mover. This is because the highly stressed blading and its rotor are now subjected to a much lower temperature than the maximum gas temperature, depending upon the ratio between the hot and cold segmental areas through which the rotor passes. These can, within fairly wide limits, be selected according to required working temperatures for the materials, no demands being made on compressor bleed air for conventional cooling. Furthermore, for prime movers where intercooling is acceptable the cooling capacity and reduction in working temperature is improved.

Emphasis has been placed on the need to reduce temperature level of the highly stressed components, because they present the most difficult problem by conventional means. However, the stator blades, nozzles and casings within the turbine segments, will be subjected to higher gas temperatures permitted by the lower mean rotor temperatures. This can be achieved by closed circuit heat transfer by forced convection between the hot nozzles and cold stator blades within each stator casing ring, using either high pressure gas or liquid metal as the medium. Such techniques in the highly stressed rotor and blading assemblies are not in use due to the practical difficulties, or for stator cooling since their temperature working levels are controlled by the rotor. It is however well documented for applications such as reactor heat transfer and cooling.

It will be apparent from the description so far that raising the maximum gas entry temperature to the turbine is by increasing the fuel/air ratio up to the desired level, the maximum for complete combustion being stoichiometric which is 0.068 corresponding to a temperature of approximately 3000°K. At this condition excess air to reduce gas inlet temperature is not required, which will reduce oxidation.

For aircraft jet propulsion the 'combine' principle would not be used for the whole of the thermodynamic cycle since work done is jet thrust, involving only a propelling nozzle, the only turbine work being that required to drive the compressor. Therefore the work of compression additional to the ram intake pressure can be performed by conventional means, that is by a multi-staged, axial flow compressor, leaving a minimum of work and pressure rise to be achieved by the compressor passes in the 'combine' high temperature gas turbine (HT/GT) unit. Its function is then to circulate air for cooling, thus allowing the increased gas temperature. For the pure jet application both an increase in specific thrust and fuel consumption result, whilst for the turbofan, heat exchange between the adjacent cooling air and turbine segments before by-pass mixing and expansion in the propelling nozzle is convenient, and provides a reduction in fuel consumption.

For power generation where specific fuel consumption is the main consideration, and where power is generated by means of a turbine the 'combine' principle would be required throughout the cycle in order to allow satisfactory working temperatures. It would also be advantageous to intercool the compressor passes, and recover waste heat from the turbine exhaust, all of which is convenient. Since specific power is also increased, plant size is reduced for a specified output, which would be of particular significance in marine applications.

It will be appreciated therefore that combining the functions of compression, cooling, and expansion within the same blading with the facility for waste heat recovery and intercooling, permits a much increased gas inlet temperature, resulting in substantially improved performance in terms of thrust, power and specific fuel consumption according to the application upon which the blading design depends. In general and for the jet engine, the turbine profile is preferred operating at negative incidence and at zero work, all of the pressure rise being by conventional means, flow being passed through the blading for cooling only. Where a pressure rise is required in addition to cooling, a zero camber design is the choice to enable the same blading to operate as a compressor or as a turbine, by change of incidence and deflection from stator settings. The compromise design cannot therefore be expected to equal that of the high performance axial flow compressor, and turbine blading, but it is shown that a very respectable performance can be achieved, the effect of which becomes of less signficance with increase in maximum gas temperature.

'Combine' HT/GT Blade Cooling

The Rotor and Blading

The mean working temperature reached by the rotor and blading will depend primarily upon the number of 'cold' air passes, to that of the high temperature gas and, as there is generally only a segment for expansion, it will depend upon the number of air passes and on the fluid properties, which are temperature related. The literature[4,5,6] indicates that when the heat fluxes in the air and gas segments balance, then the differences between blading and air, and gas and blading temperatures, are related to the ratios of mass flow and Prandtl Number as follows:

$$(T_b - T_a)/(T_g - T_b) = (1 - Pr_a)/(1 - Pr_g).\ (Pr_g/Pr_a)^{0.4}\ .\ \dot{m}_g/\dot{m}_a$$

which is approximately equal to

$$(1 - Pr_a)/(1 - Pr_g)\ .\ \dot{m}_g/\dot{m}_a$$

where a = air; b = blade; g = gas; $\dot{m}$= mass flow; Pr = Prandtl no.; and T = temperature, °K. This is derived from the specific heat (C_p) and Pr relationship on the basis of the kinetic theory for ideal gases where C_p is proportional to $Pr/(1 - Pr)$ and the approximation that Nusselt number $(Nu) = CR_e$, not the familiar $Nu = CR_e^{0.8}$ (where R_e = Reynolds number). For a high gas temperature of say 2500°K, and air temperature from the compressor of 700°K., then for fluid properties calculated at the mean bulk temperatures, Pr_a is 0.7, and Pr_g is 0.76. Therefore when $\dot{m}_g/\dot{m}_a = 0.5$ the rotor blade temperature is 1390°K.

The Stators and Nozzles

Although turbine nozzles in the expansion segment or segments are not subjected to the same stress levels as the rotor, they will be exposed to high gas temperatures, so that cooling will be required. Since stator and nozzle blades are an integral part of the same stator ring, a convenient and simple well-proven means of heat transfer from the hot to the cold source through the casings and blades is available. This is by liquid or gas convection as the medium, methods which are in general use for reactor cooling and heat transfer, but which have found little use for blade cooling due to the difficulties in channelling the fluid from the highly stressed rotor to a source of heat rejection through a rotating assembly. As the 'combine' deals with this problem by an alternative means, nozzle cooling by the method described which is a known technology will not be further elaborated upon. However, as for the rotor, a heat balance between the hot and cold flows is required but with the freedom to change relative surface areas to some extent by solidity of stator vanes to that of the nozzles. In all other respects the relationship between the blade, gas, and air difference in temperatures is the same as for the rotor. For a limiting maximum Mach number within

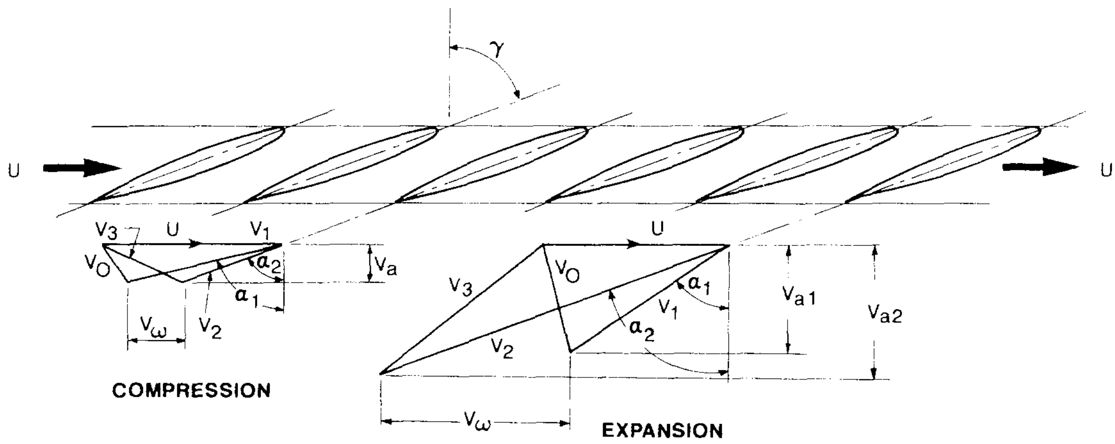

FIG. 1—ROTOR BLADING AND VECTORS OF THE 'COMBINE' HIGH TEMPERATURE GAS TURBINE. WORK $= UV_{\omega} = UV_a(\tan \alpha_1 - \tan \alpha_2)$. COMPRESSION +VE; EXPANSION −VE

U: blade speed
V: fluid velocity
V_a: axial fluid velocity
V_{ω}: whirl velocity
α: fluid angle
γ: blade stagger angle

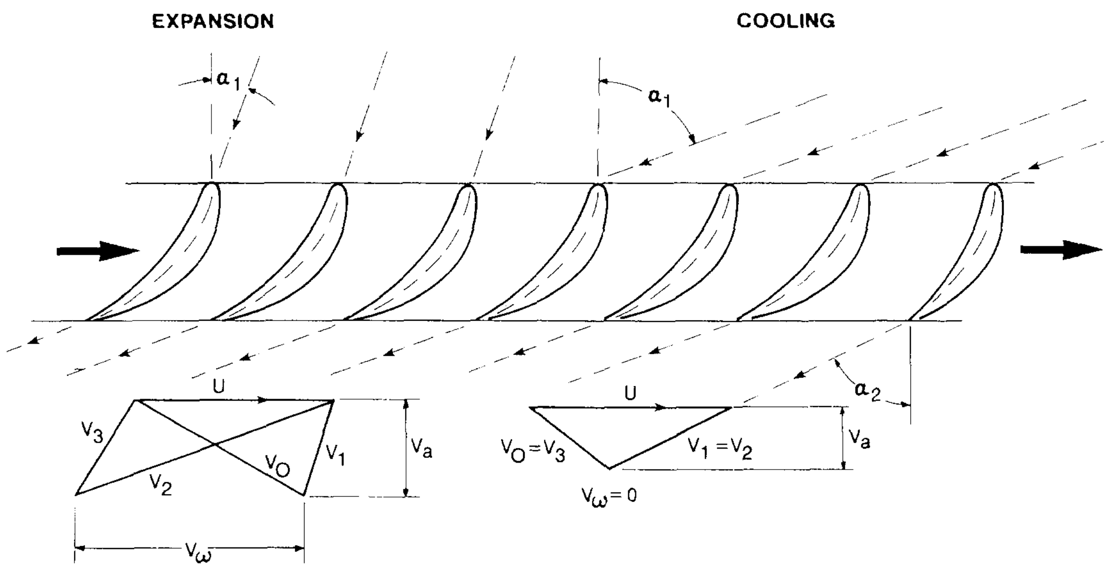

FIG. 2—EFFECT OF FLUID OUTLET ANGLE ON ZERO CAMBER BLADING DESIGN PERFORMANCE (DERIVED FROM REFS. 8 AND 9). MAXIMUM MACH NUMBERS $M_{n1} = M_{n2} = 0.8$

C_p: specific heat
U: blade speed
V: fluid velocity
V_{ω}: whirl velocity
ΔT: temperature drop/rise
η: isentropic blading efficiency

the blading, generally the rotor, the ratio of the widths of the 'cold' air segments to those of the gas is approximately:

$$W_a/W_g = (T_a/T_g)^{1/2} \cdot \dot{m}_a/\dot{m}_g$$

where W = width of air or gas segment. As an example, for a cooling system at the conditions selected in the previous paragraph, the segmental width ratio $W_a/W_g = 1.058$, widths being nearly equal.

'Combine' HT/GT Blading Performance

Conventional axial flow compressor blading is concave in the direction of rotation, and the turbine blading is convex. Therefore for a compromise blade designed to operate as a compressor and as a turbine zero camber would be the optimum and probably the only choice. For applications where expansion and cooling are the only requirements, a turbine profile would be preferred due to its higher efficiency, and work capacity. Blading performance comparisons are shown in FIGS. 1 to 5., from which the penalty for the use of the compromised blading of zero camber is evident. For high performance therefore, it would be preferable to use conventional turbine profiles cooled by air supplied by means of a high performance Primary Compressor; however when the performance penalty can be accepted the multipass compressor can be used to advantage.

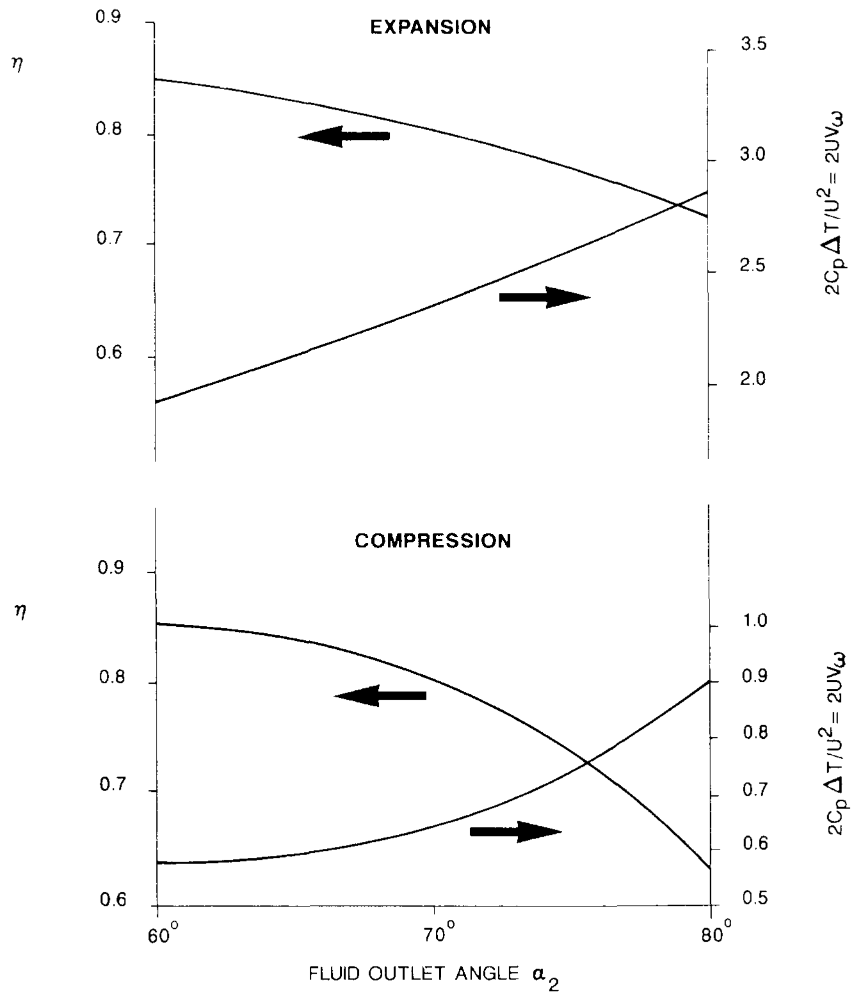

FIG. 3—'COMBINE' ROTOR BLADING AND VECTORS FOR EXPANSION AND COOLING.
WORK = $UV_\omega = UV_a(\tan\alpha_1 - \tan\alpha_2)$

U: blade speed V_ω: whirl velocity
V: fluid velocity α: fluid angle
V_a: axial fluid velocity

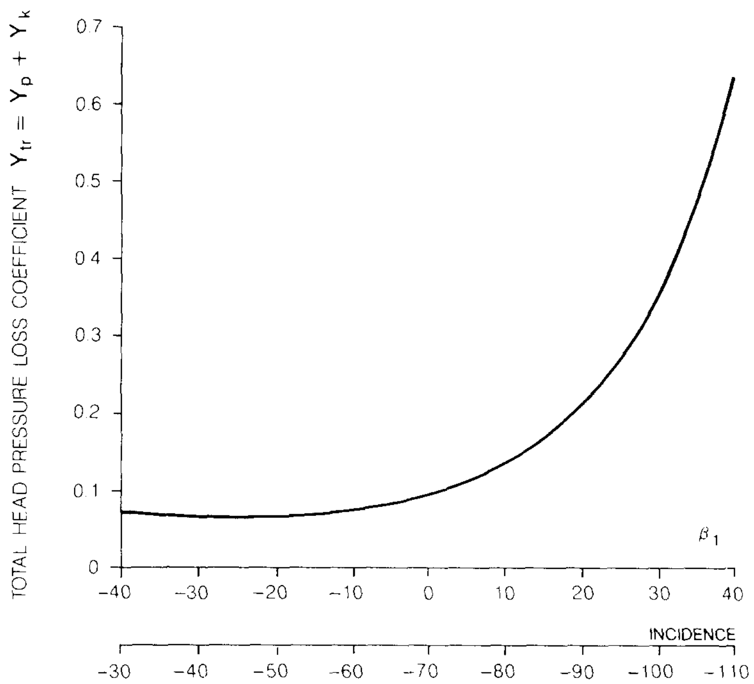

FIG. 4—'COMBINE' TURBINE BLADING LOSS COEFFICIENTS (FROM REF.8) PITCH/CHORD RATIO + 0.75.
$\alpha_1 = \alpha_2 = -70°$. ZERO LIFT (BLADE COOLING ONLY)
INCIDENCE $= \alpha_2 - \beta_1$
α: fluid angle
β: blade angle

Whatever the choice, some loss in blading performance will arise from blade cooling, although at high negative incidence turbine pressure loss coefficients remain low (FIG. 4) and at maximum Mach numbers does not represent a significant loss in effective efficiency of the compressor and can be reduced if required by increasing W_a accordingly. In this way cooling losses can be reduced to below 2% per pass; and although no bled cooling air is required, turning losses will occur through each of the return ducts, which however will be small, due to the reduced air velocities emerging from the blading; and by the use of good design for turning[7]. However the summary of all these must allow a satisfactory 'combine' performance overall not to detract to any large extent from the increased cycle thermodynamic efficiency. It will be seen from FIGS. 2, 4 and 5 (obtained from refs. 8 and 9) that in terms of blading efficiency the penalty is modest, and its significance reduces with increase in maximum permissible temperatures in the cycle. As a simple illustration, performances for generation of electrical power at conventional levels of blading efficiency and cycle temperatures are compared in FIG. 6 with the 'combine' performances at the permissible higher maximum gas temperatures and much reduced blading efficiencies, assuming a heat exchanger thermal ratio of 75% and intercooling. The blading efficiencies selected are arbitrary in order to illustrate effect rather than performance of the 'combine' unit, which will be dependent upon application as a prime mover. A detailed estimation of losses and their effect on efficiency of the primary compressor is shown in Appendices A and B, which indicate a penalty of between 3% and 5% isentropic.

Boundary Conditions between Segments

It will be apparent that, between each flow segment, operation of the blading changes, particularly between the air and gas segments and also between air passes when used for compression and cooling. In the development of the Multipass Axial Flow Compressor[2] it was this unknown aspect which gave rise

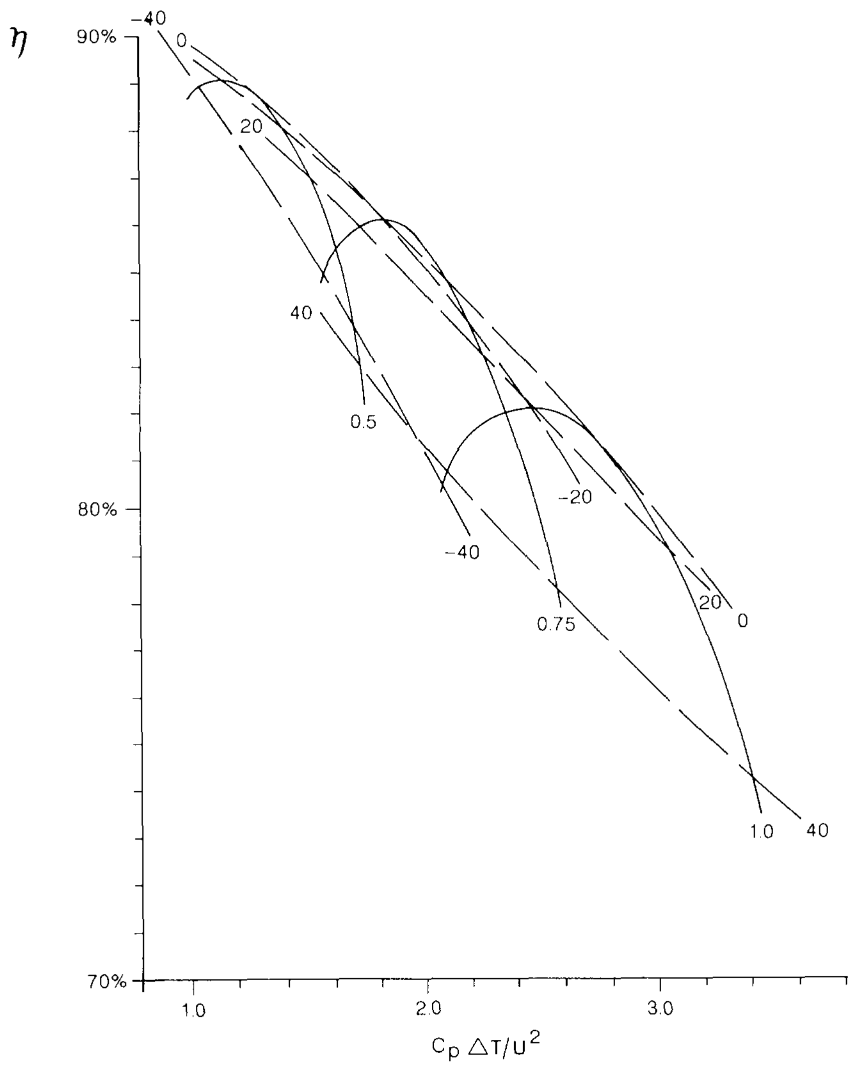

FIG. 5—'COMBINE' SINGLE PASS TURBINE PERFORMANCE FOR $\alpha_2 = -70°$ (ZERO INCIDENCE). $\alpha_1 = \beta_1$ MARKED ON CURVES FOR FLOW COEFFICIENTS V_a/U OF 0.5, 0.75 AND 1.0

C_p: specific heat = 1.34 kJ/kgK
U: blade speed
V_a: axial fluid velocity
α: fluid angle
β: blade angle
η: isentropic efficiency

to the greatest concern, as investigation was not possible without building the machine. Since the requirement for the original cryogenic helium compressor specification had been changed, it was decided to rebuild for use with air as a three-pass experimental machine for general duties where compact silent operation was important. The design was based upon current practice[9], assuming axial symmetry, but with a means of rematching passes, should this be required. In the event, this was not required, the performance being identical to axisymmetrical design, upon which it was based. Certainly, the high stagger rotor blade setting to ensure minimum widths between wide angle segments minimized circumferential effects; it is to be recommended for optimum performance, and will be essential for boundaries between air and gas passes due to the pressure differences. This can always be arranged so that inleak is from 'cold' to 'hot', and minimized as for the experimental compressor, and by change in stator settings to meet the change in duty. However, this aspect will require the same attention as for the compressor duty, although less of a challenge.

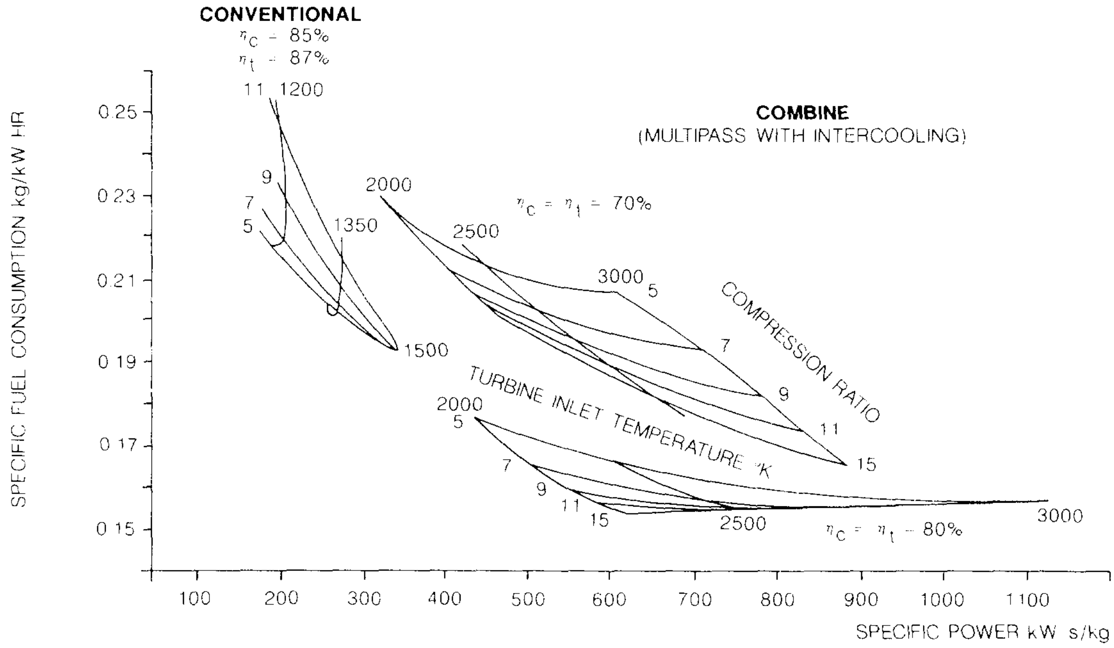

FIG. 6—PERFORMANCE COMPARISON BETWEEN 'COMBINE' AND CONVENTIONAL GAS TURBINE. HEAT EXCHANGER EFFECTIVENESS = 75%

c: compressor
t: turbine
η: isentropic efficiency

'Combine' HT/GT Power Generation and Propulsion

The advantages of raising gas temperature at entry to the turbine is well understood and have been the subject of much work on improvement in materials and blade cooling. However, as with research on blade aerodynamics, the reward for effort decreases since the major advance in all areas has already been made. There is a limit to the amount of compressor air which can be directed through internal passages in highly stressed blading. At the present

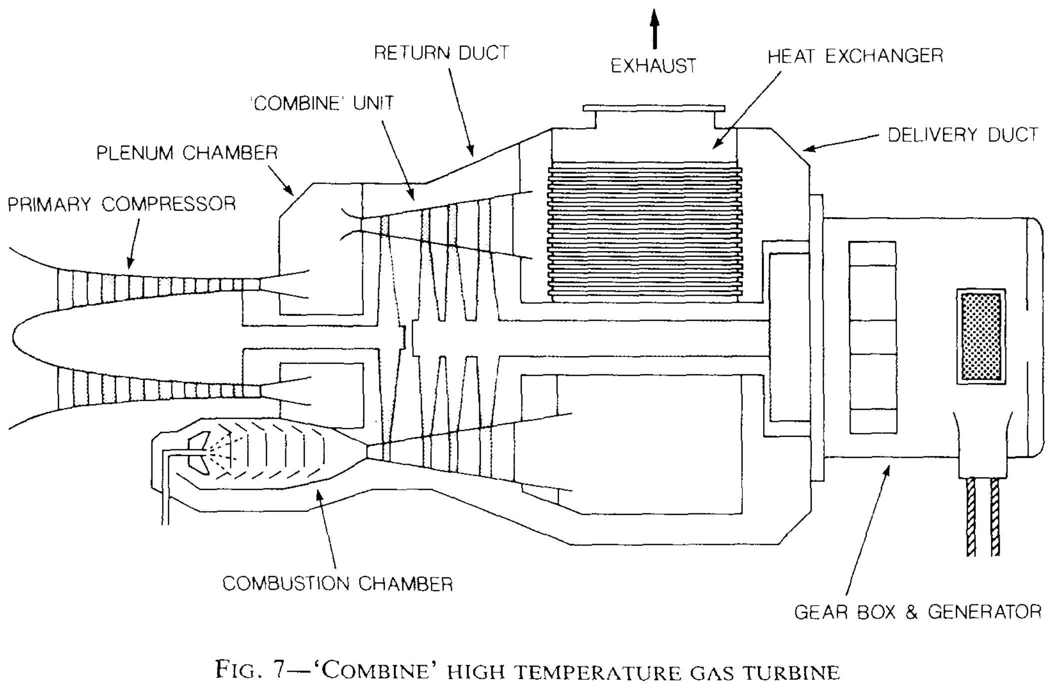

FIG. 7—'COMBINE' HIGH TEMPERATURE GAS TURBINE

time blade temperatures can be reduced by between 200°C to 300°C which, using current alloys and a air mass flow of 1½–2% of the total for each blade row[3], will permit a gas temperature of 1650°K. To raise this to 2000°K or more would by conventional means be a formidable task, and the demands on engine air and blade shapes unacceptable. It would appear therefore that if further significant advances are to be made, either an alternative in cooling or super materials are required. The impetus behind higher gas inlet temperature has been provided by the aero engine industry with the requirement for increased specific thrust and reduced size of engine. For the generation of shaft power, less emphasis is placed on this, and much more on specific fuel consumption, so that waste heat recovery, which has found little favour for aircraft propulsion engines, would be of greater attraction. This is largely due to the difficulty of ducting the air and gas to a common source of transfer of waste heat, and to the

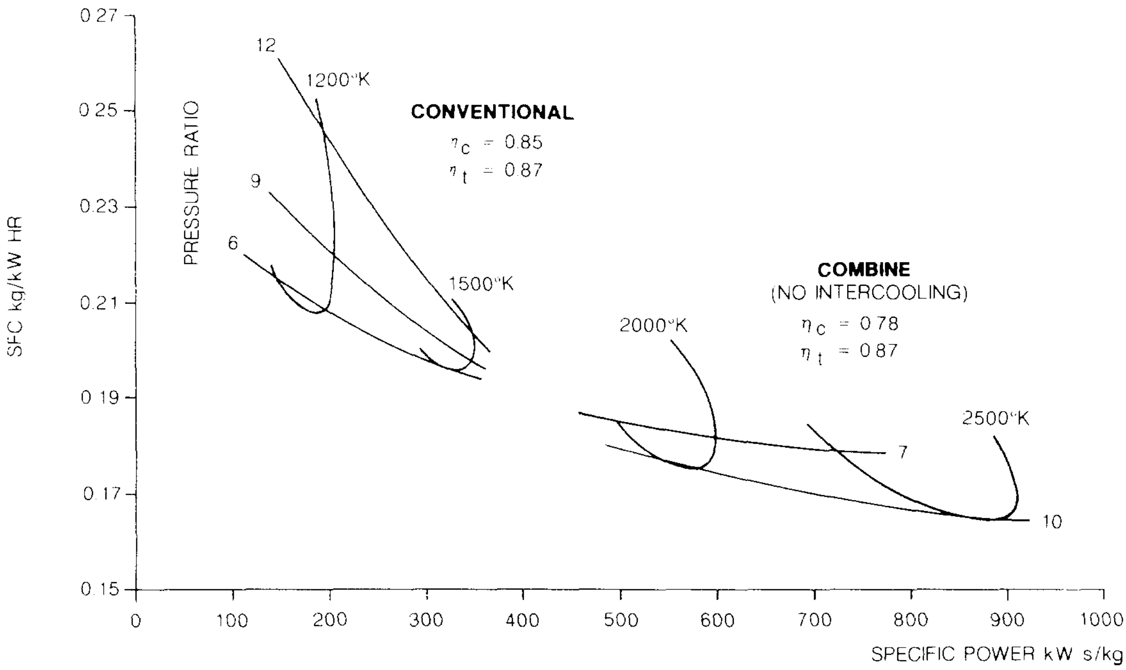

FIG. 8—POWER GENERATOR PERFORMANCE COMPARISON AT DESIGN POINT. HEAT EXCHANGER EFFECTIVENESS = 75%
c: compressor
t: turbine
η: isentropic efficiency

weight and size of the installation. With the 'combine' unit however, both air and gas occupy adjacent segments in parallel, and are well situated for heat exchange, which for the generation of shaft power would have the advantage of convenience in layout, and performance, as illustrated in FIGS. 7 and 8. The same advantages derive for the turbofan aero engine, however, in that space is not a particular restraint within the by-pass duct and fuel consumption is important. The thermodynamic cycle would be as for the power generation application illustrated in FIG. 7, with the electrical generator and shaft power being replaced by a propelling nozzle as shown in FIG. 9; heat exchange and mixing with by-pass air being performed before expansion. Performance comparisons for two subsonic speed conditions in FIGS. 10 and 11 between the conventional and a 'combine' unit are shown, due allowances being made for reduced compression and expansion efficiencies, and heat exchanger pressure losses, by means of the current data[3,4,8,9].

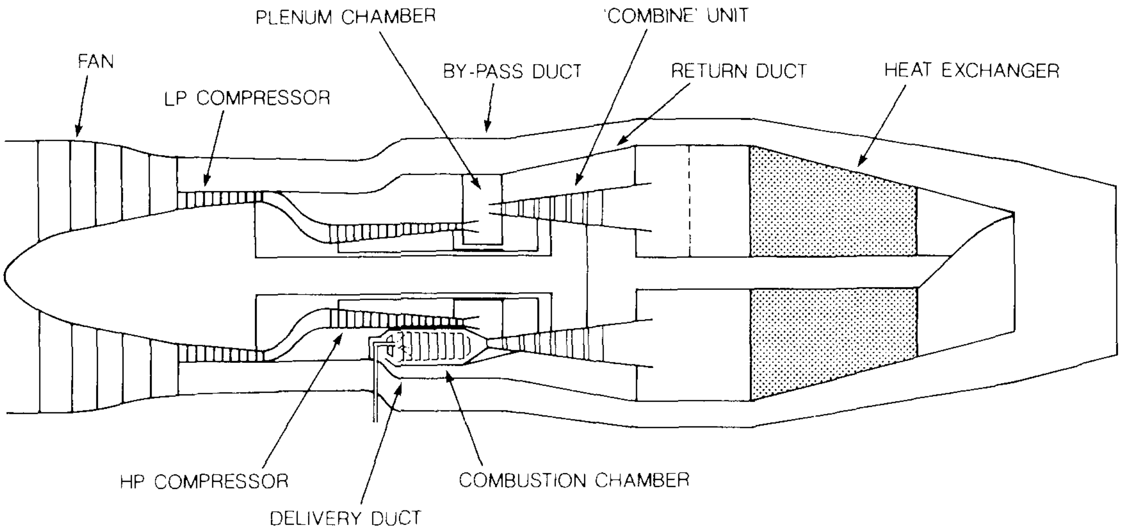

FIG. 9—LAYOUT OF 'COMBINE' TURBOFAN HIGH TEMPERATURE GAS TURBINE WITH HEAT EXCHANGE

Development

Prototype development of the Multipass Axial Flow Compressor gave credence to the hypothesis that conventional axial flow blading on a single rotor can operate satisfactorily through aerodynamic and circumferential pressure modes, behaving as a axisymmetric design provided that:

(*a*) segments are of sufficient width in terms of blade pitch to preserve adequate symmetry, which is related to the blade loading and pressure gradient;

(*b*) the flow does not stall, as the time and distance that is required circumferentially to re-establish flows in each segment is prohibitive;

(*c*) widths swept by the rotor blading should be minimized by the use of high stagger blading of maximum solidity.

By these means conventional axial flow design procedures without modification[8] can be used, and the same performance is to be expected. Segmental expansion through the same blading is not sensitive to variations in symmetry, blade loading, and change in operational mode between passes, and no effects were observed due to inleak from high to lower pressure passes. However, pressure gradients were small, and insufficient to measure any difference in flow between passes. In the 'combine' unit, pressures between the air and gas segments will be greater, increasing with expansion through the gas segment. However, by minimizing widths from high to low pressure swept by the rotor blading it is estimated an inleak of air to gas will be contained to within the levels bled for blade cooling in conventional engines. However, as for tests on the prototype Multistage Axial Flow Compressor, investigation of the transient conditions between segments will be required for the exact performance effects to be established. As has been seen optimum blading performance at high gas inlet temperatures is obtained at maximum efflux angles and rotor stagger, a necessary feature for minimum blade widths, so that no compromise is required in this respect.

Cooling air pressure losses through the blading can be assessed in detail from well proven data[8] (FIG. 4) so that little further work will be required, other than to ensure velocities are kept sufficiently low. At maximum Mach number for the blading, the pressure losses through each pass through four stages of rotor and stator blading are reflected as a 2% to 3% loss in primary compressor isentropic efficiency. For most cooling requirements, no more than two passes are required so that the penalty for raising maximum gas temperature to 2500°K for a blading temperature of 1390°K can be accurately predicted. There will also be secondary losses due to turning between passes which are intrinsically low due to the fall in velocity through the blading, and the increased area of ducting for the return flow (FIG. 7). Design data for internal duct flows optimum geometries are used, a highly reliable source being the BHRA handbook[7], so that whilst well-proven guide lines will be present for preliminary design for blading and ducting, testing will be necessary as for the prototype Multipass Axial Compressor to substantiate predictions. In the

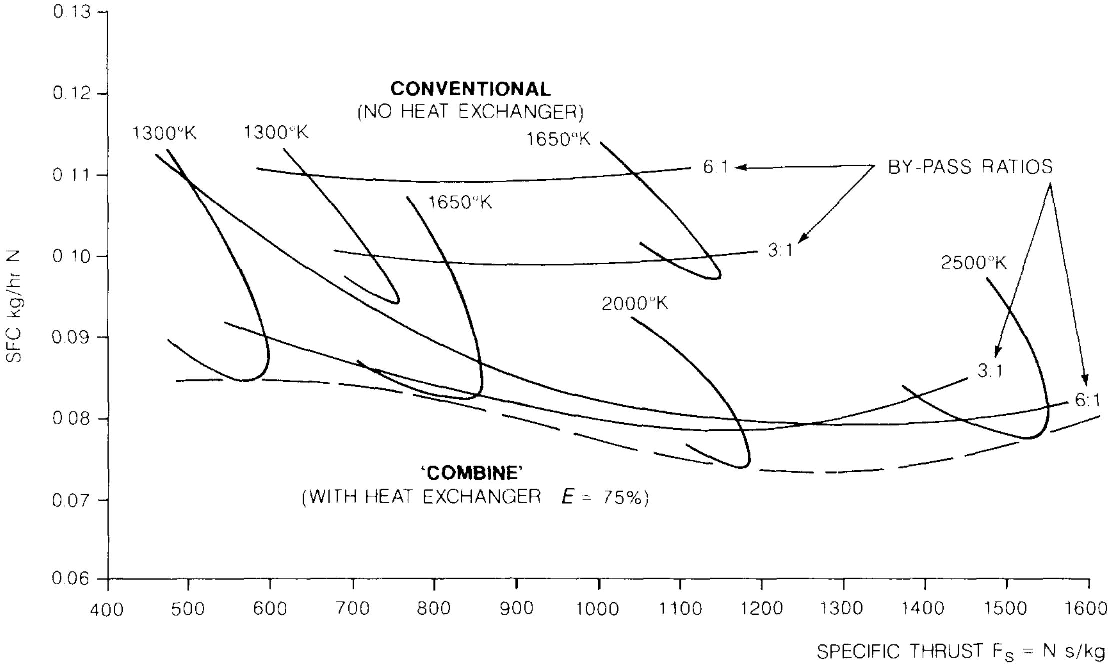

FIG. 10—TURBOFAN PERFORMANCE COMPARISON AT 10000 m ALTITUDE AND MACH NO. 0.75 (300 m/s). DESIGN POINT PRESSURE RATIO = 15:1
E: effectiveness

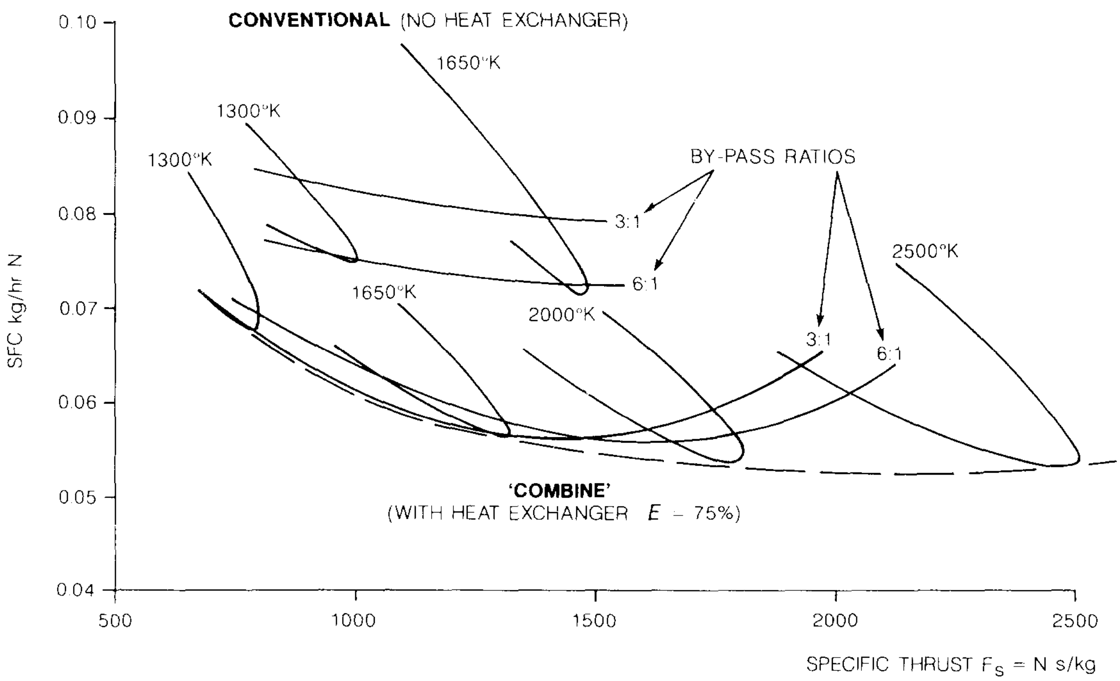

FIG. 11—TURBOFAN PERFORMANCE COMPARISON AT 5000 m ALTITUDE AND MACH NO. 0.50 (200 m/s). DESIGN POINT PRESSURE RATIO = 15:1

meantime, performance estimates for the 'combine' unit for power generation and for propulsion can be undertaken with confidence, being shown in FIGS. 8, 10 and 11.

Since all of the performance predictions are based upon the use of a Primary Compressor (FIG. 7), conventional axial flow performance is used as the datum, allowance then being made for cooling, duct and plenum chamber losses, arising from the 'combine' principle, on its efficiency which reflects on the high temperature cycle output and fuel consumption. The efficiency of expansion in the turbine segment is predicted from ref. 8 and shown in FIG. 5, no allowances being made for the effects of air inleak, which from estimates of quantity will be no more than for bleed air cooling in conventional engines. Allowing for this, then, a direct comparison by a neglect of both is valid.

Some development for internal closed circuit cooling of the nozzles through the stators will also be necessary, using heat transfer and flow data available from the open literature; this may be by means of work in parallel with the rotating assembly, which ideally would be concerned initially with cold flow investigations.

Conclusions

The 'combine' principle is the integration of well-proven work in a number of disciplines, particularly the axial flow compressor[9] and turbine[8], and the Multipass Axial Flow Compressor, which as a result of further work and investigations combined the two. It is shown that advantages are to be gained both in specific power or thrust, and thus engine size, and, particularly, in specific fuel consumption. These are well-known advantages to be gained by increase in the temperature of the thermodynamic cycle as shown in TABLE I.

TABLE I—*Comparison of specific fuel consumption and specific power (for generators) or specific thrust (for turbofans), for conventional and 'combine' gas turbines, using data from Figs. 8 and 10*

System	*GENERATION*		*TURBOFAN*	
	Specific Fuel Consumption kg/kW h	*Specific Power* kW s/kg	*Specific Fuel Consumption* kg/kW h	*Specific Thrust* N s/kg
T_{max}	1500°K		1300°K	
Conventional	0.195	350	0.095	750
T_{max}	2500°K		2000°K	
'Combine'	0.165	850	0.075	1200
difference	−15%	2.4:1	−21%	1.6:1

Experience gained in the development of the three-pass experimental axial flow compressor illustrated the advantages of maximum widths and minimum number of passes for reliable performance prediction in using the well-established high efficiency axisymmetric design data for the blading. For these reasons and since cooling requires two air passes, in general, to that of the high temperature gas pass, an equal area three-pass unit is the optimum for the gas, air, and blading temperature requirements met in the high performance engine using the 'combine' cycle. This also minimizes the cooling losses and effect on efficiency of the primary compressor, this being the penalty for the higher gas entry temperature. It is of interest to observe that to reduce the conventional 'complex' gas turbine fuel consumption to that of the 'combine' HT/GT would require the compressor and turbine isentropic efficiencies to be raised to 95%. Since this, and signficant advances by internal blade cooling and improved materials are unlikely, external cooling, using all of the compressor air, provides the means by which the specific fuel consumption and the engine installation size can be considerably reduced.

Whilst emphasis is placed on the performance advantages, the 'combine' principle can be equally well applied to a low cost simple system in which economy in fuel consumption and increase in specific power is of less importance than the capital cost of the prime mover. Such applications would be for small generating stand-by plant where low cost overrides other considerations. Thermodynamic efficiency and high gas temperatures would not be required, emphasis being placed on standard grades of material and low manufacturing costs. Such an arrangement would not justify a primary high performance compressor so that all of the pressure rise would be by means of the passes in the 'combine' unit. By this means standard or even low grade rotor materials may be used at current turbine gas inlet temperatures, at the price of a modest performance penalty.

Further Work

As for the three-pass experimental multipass axial flow compressor, it will be necessary to build and test an experimental unit to the specification of a current requirement to substantiate performance predictions. As with most new turbine designs, cold flow tests to provide information from a relatively inexpensive assembly will be required, without involving main engine components, which are expensive, and would be a complication to the initial work on the unit. In parallel with this, design of a stator ring based upon AEA heat transfer data and technology should be undertaken.

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APPENDIX A—ESTIMATION OF 'COMBINE' COOLING LOSSES (Rotor Blade $T_b = 1350°K$)

PR (opt. for minimum sfc)	20	15	10	FIG. 8
sfc ratio*	0.84	0.85	0.89	FIG. 8
Specific power* ratio	3.0	2.40	1.70	FIG. 8
Stages 'n'	9	6	5	FIG. 5
Passes 'N'	3	2	1	page 000
AR (blading)	12:1	9:1	6:1	PR (optimum)
T_g °K (gas)	2735	2380	1932	page 000
Y_r (rotor)	0.10	0.10	0.10	Ref. 8, pp. 8 & 9
Y_s (stator)	0.06	0.06	0.06	Ref. 8, pp. 8 & 9
Y_t (turning)	0.70	0.70	0.70	Ref. 7, p. 152
Y_{pl} (plenum)	1.0	1.0	1.0	Ref. 7, p. 62
$Y_{r+s}.n + Y_t$	2.14	1.66	1.50	Pass loss coefficient
$(Y_{r+s}.n + Y_t)N = Y_{Tot}$.	6.42	3.32	1.50	Total loss coefficient

* Relative to conventional cycle $T_g = 1500°K$ (FIG. 8)
AR (blading): annulus area ratio across each pass
PR: pressure ratio overall
Y: loss coefficient (proportion of dynamic head)

APPENDIX B—ESTIMATION OF 'COMBINE' COOLING LOSSES (Equal Pass Areas Max.Mach No. 0.8 $U_b = 300$ m/s)

$(V_a/U_b)_g$ inlet	0.86	0.80	0.72	FIG. 5
$(V_1$ m/s$)_g$ inlet	754	701	631	$V_1 = V_a/\cos\alpha_1$
$(V_2/T^{1/2})_g$ $M_{n2} = 0.8$	14.4	14.4	14.4	Compressible flow data
Q_a/Q_g	0.53	0.55	0.57	Compressible flow data
$(V_1/T^{1/2})_a$	5.80	6.10	6.40	Compressible flow data
$(V_1$ m/s$)_a$	161	162	160	Pass inlet
$(V_1$ m/s$)_a$	13	18	27	Pass outlet
$(\overline{V}_1$ m/s$)_a$	87	90	94	Pass mean
$\overline{V}_1^2/2K_p$ °K	3.73	4.0	4.36	Mean dynamic head
$\overline{V}_1^2/2K_p$. Y_{Tot}. °K	23.9	13.3	6.54	Total loss
$V_1^2/2K_p$. Y_{pl} (plenum)	5.6	5.6	5.6	V_1 at entry to plenum
Total cooling loss	29.5	18.9	12.14	'COMBINE' unit
Efficiency ratio*	0.96	0.95	0.96	
Efficiency**	0.80	0.81	0.82	

*Relative to conventional cycle $T_g = 1500°K$
**Effective primary compressor isentropic with cooling losses
$K_p = C_p . 10^3$
Q: non-dimensional flow parameter M. $T^{1/2}$/A. P
suffices a, g and b refer to air, gas and blading respectively
U_b: blade speed
V_a: axial velocity V_1: blade inlet velocity V_2: blade outlet velocity
η_c: primary compressor isentropic efficiency before cooling = 0.85