Performance of a Highly-Loaded HP Compressor

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ABSTRACT

A 4-stage axial research compressor, which is representative of the rearmost stages of a highly-loaded military or civil compression system, has been designed and tested at RAE Pyestock. The compressor is of large scale, with extended inter-row gaps, to facilitate the acquisition of detailed aerodynamic data. The unit performed well on test, exceeding its design pressure ratio of 4.0, and achieving a peak polytropic efficiency at design speed of 89%. Flow profiles obtained from area traversing at stator exits are presented and discussed. The measured performance is compared with an S1-S2 calculation incorporating an inviscid-viscous blade-to-blade method.

INTRODUCTION

The main aim of the compressor research programme at RAE Pyestock is to develop improved aerodynamic design and analysis methods for axial flow compressors, so that higher levels of performance can reliably be achieved. This methods development is complemented by a sequence of compressor design, manufacture, test and analysis; improved designs can then be produced and the sequence repeated. For core compressors, the process is centred around a new multistage axial research unit, designated C147. This compressor is representative of the rearmost stages of a highly-loaded military or civil compression system. However, it is much larger than an engine unit, in order to allow detailed investigations of the internal flow, and its design provides maximum freedom for future reblading. The initial 4-stage version has a design pressure ratio of 4.0, and a front stage can be added in future builds to provide a 5-stage variant with a pressure ratio of 6.4. Conceptually two further stages could be added on the front to produce a 7-stage compressor with a pressure ratio of about 18.

This paper describes the aerodynamic design of the initial build of C147, which was undertaken using conventional blade profiles and correlation methods. Overall performance characteristics and area traverses of the internal flow are then presented and discussed. Finally a post-test analysis is described, using a simplified version of the RAE S1-S2 calculation system (Calvert and Ginder, 1985), which can form the basis of an improved design system.

COMPRESSOR DESIGN

Design concept

The principal design parameters for the 4 and 5-stage configurations of C147 are specified in Table 1. It can be seen that the levels of exit Mach number, hub speed and hub/tip ratio are within current engine limits. Therefore the increased duty of C147 relative to current engine compressors has to be achieved by higher aerodynamic loading. Both the 4 and 5-stage versions were designed, but only the 4-stage unit was manufactured and tested for build 1 - the zero stage can be added for subsequent builds. A sectional drawing of C147 is shown in Fig 1.

The stagewise distributions of $V_a/U$ (axial velocity/mean blade speed), $\Delta h/\rho U^2$ (enthalpy rise/mean blade speed$^2$), pressure ratio and inlet flow angle at mid height are given in Fig 2. The axial velocity was kept as high as possible over the front stages in order to ease the loading. It was then reduced rapidly in the last two stages where the annulus is nearly parallel in order to drop the Mach number at exit to the design value. The loading parameter $\Delta h/\rho U^2$ follows the same trend as the axial velocity, with the last stage significantly off-loaded to compensate for the lower inlet velocity and increased diffusion. The axial work distribution is similar, since the mean blade speed is nearly constant, and this results in stage pressure ratios varying from 1.61 for the zero stage to 1.28 for stage 4. The radial work distributions were chosen to produce constant radial total pressure profiles at exit from each stage. The stage inlet flow angles for the front stages were chosen to limit the stator hub Mach number to about 0.8. The inlet angles increase in the rear stages in order to...
reduce the stator duties, and to balance the inlet Mach numbers and flow deflections for the rotors and stators. The residual swirl at exit from stator 4 is removed by an outlet guide vane. The stagger of the inlet guide vane for the 5-stage version is remotely variable, and those of stator 0 and the 4-stage IGV are settable. The IGVs and stators of the 4-stage version are unshrouded.

The overall mechanical design of the unit was undertaken at RAE Pyestock. A particular feature is the main rotor assembly; the complete drum with discs was machined from a single titanium forging. A zero-stage disc can be bolted to the front of this drum for the 5-stage version. The unit was designed for running tip clearances at design condition of 1% of blade height for the rotors and stators. Measurements taken from proximity probes during test indicated that a mean rotor clearance of 1.3% was in fact achieved.

Particular attention was paid in the design of the compressor to maximise its value as a flexible research vehicle for the development and validation of blading design methods. The unit is much larger than typical engine HP compressors (typically twice linear scale) with larger than usual axial gaps (23 mm or about 50% chord) to enable extensive instrumentation to be inserted with minimal disruption to the flow. Provision is made in the casing for laser viewing.

![FIG 1 C147 COMPRESSOR](https://proceedings.asmedigitalcollection.asme.org/doi/abs/10.1115/GT2018-76763)
windows. In addition, the fixed blade rows are inserted in circumferential slots on relatively long platforms so that numbers and chords can be changed as part of subsequent aerodynamic redesigns. Although essential to enable the compressor to fulfil its research role, the large axial gaps reduce the area contraction across the blade rows, and hence increase the diffusion and aerodynamic loading. However, this should be offset by the higher efficiency attainable from this large-scale rig, compared with an engine-size rig tested at atmospheric inlet conditions. Chordal Reynolds number for the first stage rotor at the design condition is about 1 × 10^6.

Aerodynamic design methods

The required flow vectors at the design point were determined using a conventional streamline curvature method with calculating planes at the inlet and outlet of each row. Allowance was made for the annulus wall boundary layers by using an effective flow annulus. This was defined using blockage factors which reduced by 1/2 per blade row from 0.97 at IGV inlet (0.98 for the 2-stage version) to 0.94 at rotor 3 exit, and then remained constant over the last four blade rows.

The choice of blading correlations was based on a survey of existing methods, which indicated that the most successful loss correlation for a range of core compressors was one developed from a stage-stacking multistage prediction method (Howell and Calvert, 1978). This correlation predicted a polytropic efficiency of over 90% for a large unit such as C147. Whilst this is a reasonable target ultimately, it was decided that the assumed design value should be lower to avoid the risk that the optimum axial matching of the compressor would only be achieved at speeds above 100%. Therefore the predicted losses were increased arbitrarily to give a design target efficiency of 88%. The radial variation of loss gave levels at hub and tip which were approximately 60% higher than at mid height. This was considered a reasonable compromise between the larger variations which occur in front stages and the flatter distributions observed in the rear stages of multistage compressors where spanwise mixing effects (not modelled by the present throughflow method) are important. The modified Kaisly correlation (Thake, 1970) was used to set the exit blade angles; this predicted deviation angles to within ±2° for the same range of core compressors. A number of criteria were used to guide the choice of blade incidence and pitch/chord ratio to ensure adequate operating range for the sections. The ratio of the blade thickness area to the critical area for the design point flow was taken as the indicator of choke margin. Stall margin was assessed using conventional stalling and separation parameters.

Particular care was taken for the C147 blading to apply the design rules in a consistent manner, and to ensure that the radial and axial variations of the blade parameters were smooth.

Detailed blade design

For the rotor blades double circular arc (DCA) profiles were chosen, since the tip relative Mach numbers are all in the range 0.65 (rotor 4) to 1.09 (rotor 0). The choke margin for rotor 4 was set at 7% and for the following rotor rows to at least 12% higher than the previous rotor. Adequate stall margins were generally obtained with near-zero incidence angles to the blade camber line, though higher values were needed near the hubs of rotor 0 and 1 to provide adequate choke margin. Rotor diffusion factors are greater than 0.5 throughout and approach 0.6 at the hub. For the sections at mid-blade height, camber angles vary between 30° and 40°, with the maximum blade thickness specified as 5-6% of chord. Pitch/chord ratios are about 0.6.

For stators 0-3, where the peak inlet Mach number ranges from 0.62 to 0.69, DCA profiles were also chosen. The Inlet Mach numbers for stator 4 and the OGV are below 0.6 and 0.7 thickness profiles (on circular arc camber lines) were adopted for these rows, as for the IGVs. The stator choke margins were set to be greater than for the preceding rotor row whilst preserving adequate stall margins. The resulting incidence angles fall from 0° for the first four stages to -2° for stator 4; zero incidence was chosen for the OGVs to limit the amount of blade camber. As for the rotors, diffusion factors are high with most values exceeding 0.5. Apart from the IGVs and double row OGV, camber is about 40° with a 7% thickness/chord ratio for the sections at mid-span. The corresponding pitch/chord ratios are about 1.3 for the adjustable IGV and 0.6-0.7 for the stator rows.
Test installation and instrumentation

C147 was installed and tested on the main compressor test facility at RAE Pyestock. The drive system, comprising a 100MW steam turbine, allowed the 4-stage compressor to be tested throughout its operating envelope at nominally atmospheric inlet conditions. Mass flow was measured from a calibrated venturi nozzle. Overall pressure and temperature rise were determined from Pitot tubes and thermocouples mounted within the duct, the delivery instrumentation comprising 12 equi-spaced rakes with 5 radial measuring stations for both pressure and temperature. Area-weighted values were used to define compressor measuring stations for both pressure and temperature.

Overall performance

The overall performance calibration of the 4-stage compressor is plotted in Fig 3 for non-dimensional operating speeds between 70% and 100% of design. The design target performance is also indicated together with the area traverse operating condition. Prolonged operation at speeds between 83% and 95% was prevented by excessive vibration levels in the transmission system.

A number of features of the overall compressor performance are worthy of comment. Peak polytropic efficiency at design speed was 1% above the target at 89%. The design flow and pressure ratio were also exceeded by 2.7% and 3.5% respectively (on a constant throttle line through the design point). The unit demonstrated considerable flow range, even at high speed, with a surge margin of about 25% for the test point at the design speed and pressure ratio.

Throughout the speed range tested, the compressor was most efficient at conditions corresponding to an operating surge margin of 15-20%.

As would be expected from the overall characteristics, the fixed inter-stage instrumentation indicates that the machine is generally well matched, with each stage operating clear of stall at design. At high speed, rotor 1 appears to limit the flow.

Area traverse measurements

Traverse measurements were taken at design speed at a condition within 1% of peak efficiency as indicated on Fig 3. Surge margin at this point was in excess of 20%. Area traverses of yaw angle and total pressure (corrected to standard atmospheric inlet conditions) are presented in Fig 4, taken using a cobra probe positioned downstream of the second stage stator. The arrow indicates the direction of hub rotation. (The indicated flow angle variations through the wake should be discounted, since nulling the cobra probe to determine flow angle is clearly susceptible to error in a highly-sheared wake region.)

In general the passage-to-passage repeatability is encouraging, especially in the free-stream region of the flow. The predominant variation of flow properties occurs in the end-wall and wake regions as might be expected. The pressure traverses indicate regions of high shear near the end-walls, extending to 3-10% of the passage height. The stator wakes are quite broad, extending over about 20% of the blade pitch. These wakes broaden appreciably towards the outer wall, particularly on the suction surface side. The blade itself is thicker in this region and, in addition, there is no tip clearance or rotating end-wall to off-load the end of the blade. There are quite strong radial variations of flow angle and total pressure which differ from both the design intent and the post-test S1-S2 analysis. Near the outer wall, the decrease in flow angle provides evidence of migration from the pressure to the suction surfaces, as would be anticipated from classical secondary flow theory. Near the hub, the annulus boundary layer becomes highly skewed due to the rotating end-wall, though this effect is constrained near the blade surfaces. The total pressure results show evidence of quite complicated local flow phenomena in this region.

In summary, the dominant features of the flow are the end-wall boundary layers, blade wakes and radial variations of total pressure and flow angle. All of these can, in principle, be modelled by a quasi-3D S1-S2 flow analysis of the type described below.

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*Surge margin is defined as

\[ \text{Surge PR} = \frac{\text{Surge PR} - \text{Operating PR}}{\text{Operating PR} - 1} \]

where the surge pressure ratio is taken to be the value at the same mass flow as the operating point.
The RAE S1-S2 calculation system

Considerable effort has been devoted at RAE Pyestock over recent years to the setting up and validation of an S1-S2 calculation system. This was not available to assist with the design of the initial build of CI47. However, it has been applied in a post-test analysis mode to add to our understanding of the measured results. This exercise has also acted as a further validation of the system for use in the design of multistage core compressors.

The S1-S2 approach uses separate treatments to calculate flow in the hub-to-tip or meridional (S2) plane, and on a series of blade-to-blade (S1) stream surfaces. The axisymmetric stream surfaces used for the S1 analysis are defined by the S2 calculation, and the blade performance data for the S2 calculation are part of the S1 solution. Thus, by iterating between them, a converged quasi-3D solution for the whole flow field can be obtained. The RAE method has been described in detail elsewhere (Calvert and Ginder, 1985). It incorporates an interactive inviscid-viscous S1 treatment (Calvert, 1982) which predicts blade section deviation angle, and profile and shock loss. End losses, arising from secondary flows, annulus wall boundary-layers and tip clearance, are not predicted from the S1-S2 treatment currently adopted. Therefore extra loss must be specified as an input. This loss, together with annulus blockage, represent the only empirical inputs required to the S1-S2 calculation system. Since the present S2 program does not include a spanwise mixing mechanism, it is necessary to distribute the end losses over the whole blade span. Analysis of a number of multistage core compressors at RAE has led to a rule which typically gives total loss coefficients which are twice profile loss at mid-span and rise by about 65% towards the blade ends. A blockage allowance which increases by \( f \% \) per blade row for the first three stages was adopted, with an inlet blockage of 2% to the first stage rotor.

For application to multistage core compressors, a simplified version of the S2 technique described by Ginder (1984) is currently adopted. It is considered that the extra complication of having calculating planes within the blade rows is not generally necessary for a core compressor where the hub/tip ratios are fairly high and the blade rows operate mainly subsonically and unchoked. Therefore the program was run in the 'ductflow' mode, with calculating planes placed at the blade leading and trailing edges only. Linear variations of stream-surface radius and thickness were assumed within the blade rows for the S1 calculations.

S1-S2 calculations for multistage compressors are run by specifying the values of inlet total pressure and temperature and of exit mass flow function. The S2 program then calculates the mass flow rate and overall pressure ratio. The specification of exit mass flow function is analogous to setting the exit throttle valve on the test rig and avoids the difficulties that can occur if either inlet mass flow or overall pressure ratio is specified. For example, the specification of inlet mass flow is clearly unsatisfactory when the compressor characteristic is vertical due to choking in a front blade row.

**FIG 4 AREA TRAVERSE MEASUREMENTS AT STATOR 2 EXIT**
Performance predictions

The S1-S2 analysis at the design speed traverse condition matches the measured overall mass flow and pressure ratio almost exactly. It also gives good agreement with the stage pressure ratio distribution indicated by the traverse results. The predicted overall polytropic efficiency is about 1% low, but could be corrected relatively simply by reducing the additional loss term applied, without changing the S1-S2 solution significantly. However, this would be at the expense of modifying the extra loss 'rule' that has been shown to give a good fit to the overall performance for a range of other multistage core compressors.

The radial variations of stage efficiency derived from the stator exit area traverses of total pressure and temperature are compared in Fig 5 with those predicted by the S1-S2 analysis. When interpreting these data, it should be recognised that considerable uncertainties arise in estimating stage efficiencies from traverse measurements, particularly for the rear stage where the temperature rise is small. Nevertheless, clear trends can be discerned.

The predicted stage efficiency levels and radial profiles remain broadly similar throughout the machine, reflecting the use of the relatively simple loss model described above. The experimental measurements show the first two stages to be more efficient than predicted, while the latter stages are lower. It would appear that the S1-S2 loss model needs better allowance for the effects of high hub/tip ratio rear stages. More importantly, the measured radial efficiency variations show a clear progression through the machine, from a severe profile for stage 1, reflecting loss concentrated near the end-walls, to much flatter profiles for the rear stages*. This effect shows the impact of spanwise mixing, which progressively distributes the end-wall losses across the blade span, and eventually leads to the 'repeating stage' situation for multistage core compressors.

The present loss model could clearly be modified to include appropriate stage-by-stage variations to match the experimental data. However, a better approach would be to make the loss model represent generated losses only, and to include a spanwise mixing mechanism in the throughflow calculations which, by redistributing loss, should enable the measured radial variations of temperature and pressure throughout the compressor to be modelled more closely. The spanwise mixing models suggested by Adkins and Smith (1982) and Gallimore and Cumpsty (1986) are obvious candidates. The efficiency profile for stage 1, which is least affected by mixing, could provide a model for generated loss. This indicates values over the mid-half of the span which are broadly in line with predicted profile losses, with end losses confined mainly to the inner and outer 25%. This pattern of loss could in principle apply throughout the machine given suitable levels of mixing.

Fig 5 summarises the computed blade-to-blade performance for the second stage stator at mid-height. The results are typical of those predicted for the DGA blade sections at this operating condition, and confirm generally good axial matching. In particular, the effective incidence angle is close to the optimum for the section, with no extremes of loading at the blade leading edge. The predicted deviation angles are about 1° less than the design intent, but there are no major discrepancies apparent. Typical deviation levels are over 10° due to suction surface boundary layer separation at about 80% chord for all blade rows (except from the IGVs). This separation is apparent both from the broad boundary layer and wake indicated on the Mach number contour plot, and also from an examination of the development of the boundary layer displacement thickness (t*). The experimental evidence of broad stator wakes at this operating condition has already been noted.

The ability of the RAE S1-S2 system to provide a detailed model of the internal compressor flows and to produce good agreement with the measured stage and overall performance is a promising basis for future work. Development is clearly needed to improve prediction of the radial variations of flow properties. However, even without this, the system provides a framework which will allow blade profiles to be tailored for the required duty. Such profiles should have significantly reduced boundary-layer separation compared with the conventional profiles used in build 1 of C147, and thus allow higher performance levels to be achieved.
CONCLUSIONS

A new multistage research compressor, designated C147, has been designed and tested at RAE Pyestock to investigate the potential of highly-loaded rear compressor stages. Particular attention was paid in the design of the unit to facilitate the acquisition of detailed inter-stage measurements, and also to provide flexibility for future reblading. The initial 4-stage build was designed using conventional blade sections with correlations for loss and deviation. The measured performance was very encouraging. The design pressure ratio of 4.0 was exceeded, with a peak polytropic efficiency at design speed of 89%. The compressor demonstrated considerable flow range, with peak efficiency achieved along a working line with 15-20% surge margin.

A post-test S1-S2 analysis at design speed has been shown to predict well the overall performance of the compressor, and also to give useful information on detailed blade flows. This system provides the framework to improve the performance of future builds of C147 by tailoring the blade profiles to control the boundary-layer development. It may be possible to improve the prediction of the radial variation of flow properties by the inclusion of spanwise mixing effects.

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REFERENCES


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