The Turbomachinery of the British Leyland 2S/350/R Engine

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This paper describes the design and development of the compressor and turbines of the British Leyland 2S/350/R truck engine. The design of these components has to meet the cycle requirements for efficiency and operating range on the one hand and have the ability to withstand the steady and vibratory stresses on the other. In addition the design has been influenced both by manufacturing methods and by the fact that the engine has to operate in an automotive environment. The paper discusses how these often conflicting requirements have been satisfied in the latest version of the engine.


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INTRODUCTION

The concept of the British Leyland 2S/350/R truck engine is based on the earlier experimental Rover car engines (1, 2) and, as a result, uses a similar cycle, a low pressure ratio two-shaft design with a ceramic regenerator. To keep cost to a minimum, the same basic turbomachinery layout was adopted, namely a single-stage centrifugal compressor driven by a single-stage turbine followed by a single-stage power turbine. The only difference is that on this engine, the compressor turbine is axial rather than radial flow. Also, to save cost, the main components are, as far as possible, simple one-piece castings or forgings with integral blades. A diagram showing an outline of the turbomachinery is given in Fig. 1. A general description of the latest version of the engine is given in reference (3). While, in this paper, we will discuss in more detail the bladed components.

CYCLE CONSIDERATIONS

The required power for a truck engine lay between 350 and 400 hp, and the cycle had to be chosen to give a fuel consumption both at full power and over the operating range comparable with a diesel engine. After development, the engine has to achieve a life of up to 12,000 hr, and, times for road transport applications, a quick response to throttle movements is necessary, the inertia of the compressor and compressor turbine rotors must be as low as possible. The performance targets for the compressor and turbines, as for all other engine components, come directly from the cycle calculations. The main features are given in the following.

Pressure Ratio

This was chosen as 4.1 which is close to the theoretical optimum for the chosen turbine inlet temperature, likely component efficiencies, and a 90 percent thermal ratio heat exchanger. However, it was recognized that any increase in pressure ratio as the compressor was developed would be welcome for two reasons.

Firstly, the curves of specific fuel consumption against pressure ratio are fairly flat in the region of the optimum, but this optimum rises with lower thermal ratio. Secondly, part-load fuel consumption is improved if a higher than optimum design pressure ratio is accepted.

Efficiency (at Design Point)

1. Compressor: 85 percent total to total
2. Compressor turbine: 85 percent total to total
3. Power turbine: 82 percent total to static (87 percent total to total).

These values were based on a combination of previous experience and the best computational methods available to us.

Part-Load Operation

A simple two-shaft engine has a unique running line in that each compressor speed has a corresponding turbine inlet temperature and air mass flow at which the engine can run in equilibrium. Even with high heat exchanger thermal ratio, the specific fuel consumption increases rapidly at reduced power. If, however, the power turbine nozzles are closed at reduced power in theory, any turbine inlet temperature can be used at each compressor speed. If this is kept high at part load and the regenerator used to return the exhaust heat to the cycle, the part-load fuel consumption is greatly improved. It was decided to control the engine to run at a constant gas temperature between the compressor and power turbines (T6 in British Leyland notation) from full power down to at least 20 percent.

1 Numbers in parentheses designate references at end of paper.

2 This was originally set at 1280 K (1007 °C), but, in view of the engine life target was subsequently revised to 1287 K (1014 °C).
power. The compressor flow range between surge and choke must be sufficient to accept the reduced mass flow at compressor speeds below design which the higher turbine inlet temperature requires. Also, to get the maximum benefit, the turbines have to maintain as much of their design point efficiency as possible at lower speeds and inlet pressures.

**Compressor**

The compressor is based on two previous series of successful compressors; namely, the Hover compressors described in reference (1, 2) and the Austin 250 compressors described in reference (4). A design point one-dimensional analysis is given in Appendix 2 to show the operating conditions of each component.

Fig. 2 shows the characteristics obtained on our compressor rig after initial development. This rig consists of a 500-hp d-c motor driving the compressor through two step-up gearboxes. The inlet air is drawn from outside the building through a British Standard venturi meter and discharged from the test compressor through a variable throttle. Pressure and temperature measurements are made to evaluate total-to-total pressure ratio, mass flow, and compressor temperature rise. The methods used comply as closely as possible with those recommended by Dimock (5); three of the main points are given as follows.

The outlet total pressure is obtained from the mean wall static pressure in the 5-in. (127-mm) dia delivery pipe plus a mean velocity head calculated from continuity. This is probably less than the average of a total pressure traverse.

The thermocouples measuring temperature at inlet and outlet are unshielded which due to the velocity difference means that the measured temperature rise could be 0.5°C low. In calculating the adiabatic efficiency, the specific heat of air corresponding to the average temperature in the compressor is used. For pressure ratio values over 4.1, this gives a value of efficiency 0.8 percent lower than if the specific heat of the inlet air is used.

It is not necessary to discuss in detail the changes tried on the rig, but instead we are devoting the following paragraph to our experience.
with and present thoughts on each component in the flow path.

**Intake**

The compressor has a straight inlet pipe which receives air from the truck filter boxes through a standard venturi meter contraction. In the center of the duct, the rotor ends with a smooth bullet-shaped spinner. These ensure that the rotor receives air axially with the wall boundary layer as thin as possible. A good quality inlet flow is essential for high performance.

**Rotating Guide Vane (or Inlet Guide Vane)**

This is the next important component from the performance point of view. We design for uniform axial inlet velocity, and Appendix 2 shows that the required turning varies from about 40 deg at the root to about 60 deg at the tip with Mach numbers of 0.96 to 0.87, respectively. It is not claimed that axial cascade data is directly applicable to centrifugal compressors, but it is interesting to note that while the root conditions are within the well-known limits established by Howell (6), the tip aerofoils must run stalled. This leads to strong vibration excitation on the blades. It also follows that the air passing through the lower part of the annulus can be diffused most efficiently into the impeller. However, at the hub diameter, the root of 17 blades, thick enough to be acceptable from a stress and vibration point of view, have to be accommodated and yet leave a well-shaped diffusing passage between them.

We have been able to keep the nominal tensile stress at the blade roots down to a fairly modest 9.5 tons/sq in., but the thin blades have rather low natural vibration frequencies. It is necessary to ensure that these do not coincide with an integral order of rotational speed near the design speed. Fig. 5 shows an interference diagram for our current rotating guide vane. It will be seen that the excitation orders are low integral multiples of speed, e.g., at design, orders 4 and 5 are near the resonant frequency. Also, small changes in the blade natural frequency cause a large variation in the interference speeds. We have found that if we run near a resonance at high speed where Mach numbers and, hence, excitation levels are high, the position of the operating point relative to surge has a strong effect on the magnitude of the blade vibration stresses. For example, if we run on a resonance, we might get 3 tons/sq in. alternating stress, if we now move close to surge, the stress may double to 6 tons/sq in., and at surge itself, this will become 12 tons/sq in. This is only maintained for a short time, but at a frequency of 2 KHz, the fatigue damage accumulates quickly. This problem of avoiding damaging resonances and yet keeping a good diffusing root passage is very difficult to solve, but the value of doing so cannot be overemphasized. In our experience, a design which has a poor root passage, drives more flow to the higher Mach number tip and definitely reduces efficiency regardless of the quality of the downstream components.

The guide vane is made as a stainless-steel precision casting which is resistant to corrosion and erosion and also has sufficient strength to resist workshop damage. Since it has a much smaller diameter than the impeller the use of steel does not significantly increase the gas generator inertia.

**Impeller**

This is an Ealuminium 55-32 aluminum alloy close forging which is machined on its outside profile only and not in the air passages. From inertia considerations, a material of greater density than aluminium would be unacceptable, and the forged material withstands the high blade stress better than castings available to us at present. The hub and shroud profiles are designed for steady diffusion through the impeller, and there are 17 full vanes and 17 extra half vanes or splitter vanes in the radial part only. Testing without the splitter vanes showed an equal efficiency and similar flow range, but the increased slip reduced the temperature rise and, hence, the pressure ratio for a given tip speed.

**Vaneless Space**

Because this is an inherently low efficiency diffuser, we believe it should be kept to the minimum necessary to fulfill its two main functions. Firstly, it reduces the Mach number of the air entering the diffuser, and secondly it allows the impeller wakes to mix and thus reduces vibration excitation on the impeller. Ten percent of the impeller radius is adequate.

**Diffuser and Collecting Scroll**

A target of 81 percent total to total compressor efficiency was quoted in the foregoing, but an outlet velocity was not specified. Since the velocity at the face of the heat exchanger disk is below 20 fps, the compressor outlet velocity head must be low enough to discard without serious penalty. We have arbitrarily chosen 150
fps as a suitable figure. In our experience, the best system to diffuse from 0.9 Mach number to this low velocity is an aerofoil diffuser followed by a carefully designed collecting scroll. At first sight, this seems illogical because we collect the air into a 7-in. (127-mm) dia pipe and then spread it again over the heat exchanger inlet area which is about 200 sq in. We definitely find this worthwhile however, in order to get good diffusion to 150 fps and, at the same time, maintain the uniform circumferential pressure field which is essential for the proper operation of both the diffuser and the impeller.

The aerofoil diffuser is designed as a cascade of 15 C3 aerofoils having the required deflection transformed from the axial compressor form into circular coordinates to suit the centrifugal compressor. Although this is often criticized because the equations used only apply to incompressible, inviscid two-dimensional flow, in practice, provided the deflection is below Hewlett's "nominal" value (6) for axial cascades, a wide range efficient diffuser results. The tails of the vanes are thickened on the pressure face from the basic C3 aerofoils to allow the front cover fixing bolts to pass through them. This does not significantly affect the performance.

Later Impeller Development

Fig. 2 shows that at 37,000 rpm, the design point pressure ratio is 4.83 at 62.5 percent adiabatic efficiency which comfortably exceeds the targets. However, the flow range is not sufficient for a constant TS engine. We suspected that any attempt to improve the range by different matching of the rotor and diffuser would either cost us design speed performance or have only a small effect. It became apparent that the most promising approach lay in departing from radial impeller blades and using back swept blades. These are blades which, in the radial part of the impeller, curve backward against the direction of rotation as in Fig. 4. This allows freedom to specify the relative outlet angle from the impeller instead of accepting the one automatically given by the radial vanes. This type of impeller is discussed by Speer (7) and the advantages for a given pressure ratio are given in the following.

1 Reduced diffuser inlet Mach number which should accept a larger range of incidence at high efficiency.
2. Increased degree of reaction
3. Reduced impeller diffusion due to the higher relative velocity for a given absolute velocity at outlet
4. Reduced aerodynamic loading (Cp GTE1/GTE2)
5. At constant speed, the temperature rise increases quite steeply as the outlet is throttled instead of staying fairly flat. This gives a stable impeller which will delay the onset or surging.

However, the stress is increased by two effects. Firstly, the tip speed increases for a given pressure ratio which raises the bore stress. Secondly, back swept vanes are subject to high centrifugal bending stress, whereas radial vanes are subject to tension only. From both cost and inertia considerations, we had to use aluminium, but we found we could retain RR 56 material, design for some increase in pressure ratio, and have the foregoing advantages provided we were satisfied with a medium value of back sweep angle and eliminated the splitter vanes. Work on the radial impeller had already shown that this was no great disadvantage. Since we did not wish to change the rotational speed, the extra tip speed to compensate for the back sweep and increase the pressure ratio was obtained by increasing the diameter from 9 in. (229 mm) to 10 in. (254 mm). Fig. 5 shows the compressor characteristic obtained. The design point pressure ratio is now 4.46 at 97 percent adiabatic efficiency. The flow range is sufficient for running at constant T6, and the running line is close to the peak efficiency point of 33 percent for most of the operating range.

THE COMPRESSOR TURBINE

Since the structural design of the compressor turbine rotor presents many difficulties and, to a large extent, dictates the design, we will discuss this first and then describe the design of the whole stage and its performance characteristics.

The major constraints on the design of this component were: (a) a turbine inlet temperature of 1340 K (1067°C) and (b) a small disk diameter to achieve low gasifier inertia. These two factors led to a blading design with low reaction high axial velocity and high aerodynamic loading as shown in Fig. 6(a).

Early difficulties with blade vibration were overcome by thickening the root of this blade, and the resulting performance loss was recovered by increasing the reaction of the blading as shown in Fig. 6(b).

Obviously, the blade metal temperature was increased by this change, but these factors offset the otherwise reduced creep life: (a) the
Turbine inlet temperature was reduced by 15°C as mentioned earlier, (b) compressor developments had dictated a drop in gasifier speed from 38,200 to 37,600 rpm and (c) the new blade with thicker root and stronger taper has lower stresses.

Throughout this development, the compressor turbine rotor has had 41 blades of approximately 0.6-in. (16 mm) root chord on a hub diameter of 8.5-in. (216 mm) sold.

The blade stress temperature and life distribution along the blade span are shown in Fig. 7, which refers to the current 50% reaction blading. The blades on these rotors have a low aspect ratio compared with usual aircraft practice, although some helicopter engines have similar rotors. The reasons for this include manufacturing constraints such as minimum cast thickness for trailing edges and general dimensional control of the profile. Also, experience has shown that a short rugged blade has good resistance to corrosion and erosion, and that many of the usual sources of loads and stresses are of minor importance, for example:

1. The axial and tangential gas bending
2. Torsion due to centrifugal unbalancing
3. Thermal stress due to chordwise temperature gradients.

It is noteworthy, however, that any offset of the blade centroid due to faulty stacking can cause high root bending stress; e.g., an offset of 0.03 in. (0.75 mm) in the blade nose from the root centroid causes bending stress of about half the nominal root stress.

It is questionable whether a simple design criterion of, say, 12,000 hr creep life at design point is valid for an automotive engine which spends much of its time at varying speeds and loads. In our experience, the medium term (1000/3000 hr) problems are fatigue, both vibration and thermal about which more will be discussed later. It is our view that until more experience is gained with these fatigue problems, it is reasonably safe to design for a nominal creep life to the design specification and then rate the engine according to experience. Even this involves using extrapolated data on material properties, a procedure that cannot rigorously be justified. As users of 713 LC3 material will know, the creep rupture life increases dramatically with even slight reductions in temperature at the stress levels typical of automotive designs. For example, at 7 tons/sq in. and 900°C, a drop in temperature of 15°C doubles the life and a drop of 50°C increases life by ten times.

The thermal fatigue problem of the compressor turbine is associated with the rotor disk rather than the blading. The existence of rim cracks on integral turbine rotors is familiar to most people in the small turbine world, but it is worth giving a brief explanation of the mechanism as we understand it. Fig. 8 shows a typical stress distribution in our compressor turbine disk, and it will be noted that there is a considerable compressive hoop stress at the rim. At design point, this may or may not be sufficient to yield the material, but on shutdown, the centrifugal tensile stress is reduced and the thermal compression increases to the yield point of the material. This situation is worse than it seems at first sight, because the elastic modulus (E) of the material is lower at the high-temperature rim. This means that the compressive stresses shown correspond to higher strain levels than if E were constant through the disk. On shutdown to a uniform, lower temperature, this high strain converts to a higher tensile stress corresponding to the higher value of E. Since the mass of material is relatively small compared to the bulk of the
Compressor turbine

In addition to the hot shutdown mechanism, there is the cold start case where high rim temperatures are achieved before the body of the disk heats up, and we estimate that compressive rim strains of about 0.3 percent occur 36 sec after start-up. Taken together, the cold start and hot shutdown sequence subjects the rotor to a severe thermal fatigue cycle which must be catered for in the design. It is difficult to specify what level of compressive stress should be aimed for to prevent rim cracking. Obviously, it should be less than the yield stress under any condition but also it must be below an unknown thermal fatigue threshold for the engine design life.

There are two ways to tackle the rim cracking problem; either keep temperature gradients and stresses below the "threshold of fatigue" or control the cracking which is otherwise inevitable. Many successful turbine designs use the first approach but are usually to be found in early low temperature engines, geared engines, or in the cooler stages of a multi-stage turbine. Also, the rotors are not usually made of cast materials. The alternative approach of controlling or containing the cracking is necessary for the successful application of high-temperature cast turbines in automotive engines. There are a number of detail methods, but the principle on which they all work is that the hot rim circumference is broken up by slots, pockets, or false blade roots so that the radius of the disk at which the circumference is continuous is at a region where temperature and hoop stress are within the capability of the material. Without such methods, rim cracking can be expected to appear within a few hundred hours of operation, but with the technique we have adopted, it is no longer regarded as a problem. An incidental advantage of such methods of construction is that the burst behavior of the rotor is by uniaxial tension somewhere above the continuous rim. The relatively small pieces thrown off are easily contained by the main engine casing. These arguments apply equally well to the power turbine, and, therefore, we do not regard it necessary to provide containment shields.

We will now complete our discussion of the compressor turbine by describing the aerodynamic features and performance. Our earlier engines used a plenum chamber to turn the gas from the combustion chamber into the nozzle annulus. The nozzle vanes turned the gas from an axial direction to approximately 65 deg for entry into the rotor as shown in the velocity triangles (Fig. 6). We found this ducting arrangement rather inefficient, because the incidence on the nozzle blades varied round the annulus and there was not enough acceleration of the flow approaching the nozzle. These effects were clearly demonstrated visually by water flow tests in a full-sized perspex model of the duct and nozzle. The result of this poor flow was a non-uniform distribution circumferentially and thin annulus wall boundary layers. The high total pressure loss in the nozzle was obviously unwelcome, but the poor distribution also increased the losses in the rotor. The later design of the duct and nozzle shares the turning of the flow between a scroll or volute and the nozzle blades. The flow from the combustion chamber becomes the inlet tangential flow of the scroll and then follows a free vortex path down to the nozzle entry duct. From the entry lip onward, an axial velocity component is generated as the gas flows down this duct, and the annulus wall curvature is so specified that the acceleration is sufficient to keep the boundary layers thin. The choice of the turning distribution between the scroll and the blades is arbitrary, although it is controlled to some extent by the space available.

The main advantage of the System is that the initial part of the turning is done in the scroll without generating large secondary flows, while the turning at higher Mach numbers approaching the rotor is done by conventional, but high stagger, stator blades. The overall result is: (a) lower secondary flow which is generally accepted as being a large part of the loss in low aspect ratio turbines, and (b) a more uniform...
circumferential flow distribution giving the rotor and the subsequent power turbine nozzle a reasonably clean flow. The mean entry angle to the nozzle blades is 42 deg relative to the axial direction which means that the blades turn the flow a further 21 deg. This keeps the blades very lightly loaded, even though vibration considerations permit only 12 blades.

Fig. 9 shows the compressor turbine characteristics obtained. The mass flow at the design value of about 2:1 expansion ratio is correct for the engine and the efficiency is within 2 percent of stated target.

Current work to improve details of the scroll, nozzle, and rotor will enable us to reach the target of 85 percent.

THE POWER TURBINE

So far, the discussion on turbine design has been concerned with the compressor turbine which is reasonably easy to cover in a separate description. However, a description of the power turbine and its nozzle cannot be separated from the overall turbine design problem for automotive engines. To state the obvious, the automotive engine is a shaft power engine in which the power turbine is required to produce mechanical power with a minimum of aerodynamic losses. This means that the exhaust gas velocity leaving the stage must be as low as possible, typically 500 fps.

Fig. 10 Power turbine velocity triangles (mean radius)

Fig. 11 Power turbine characteristics (at one nozzle setting)

We acknowledge that recovery of pressure in an exhaust diffuser would benefit turbine efficiency, and the extensive work described in references (8, 9) is very relevant. However, our own work on the diffuser has not proved very fruitful, and so design policy has been toward a power turbine with low axial velocity.

There we have the nub of the automotive design problem: how is the high axial velocity (typically 900 fps) compressor turbine to be joined to the low axial velocity power turbine with a variable area nozzle in between? This problem has been well known for some time. Reference (10), for example, gives a very interesting analysis.

In the 28/350 engine, the connection is made with a two-row power turbine nozzle, and we will describe an intermediate stage of development. The first row of fixed blades turns the flow about 63 deg, while the second row is variable and turns the flow the final 10 deg or so depending on control system adjustment for engine matching temperature. The velocity diagrams for the power turbine are
The design and development of this two-row nozzle has presented many problems, but we believe that it has the ultimate advantage for the following reasons:

1. The tip clearance losses of these variable blades is less than with a single-row nozzle of variable blades.
2. There are no long diffusing duct in which flow is uncontrolled.
3. There are no nozzle support struts to add drag.

The power turbine is an integrally cast wheel in 713 H and has 37 blades of 1.6-in. (20-mm) chord on a root diameter of 7.48 in. (190 mm). The design speed stress and temperature distributions of the blading are shown in Fig. 12. Since this is an automotive engine which spends most of its time at speeds less than this, there is little danger of creep failure. Although the higher aspect ratio blades compared to the compressor turbine are more subject to untwisting torsional stress, our experience justifies using normal centrifugal stress as a basis for design.

**POWER TURBINE VIBRATIONS**

Unlike the bladed components on the gasifier, the power turbine has extra duties to perform, namely, operation between zero and over 100 percent speed and operation with var-
able nozzle angles. Because of the wide speed range, it is practically impossible to find a combination of nozzle blades and rotor blade frequency which avoids interference somewhere in the range, and, because of the reverse nozzle operation, the aerodynamic excitation can be severe because of the heavy wakes in the flow. One has the choice of selecting a large number of nozzle vanes to keep the lower vibration modes at low power turbine speeds or damping the blade vibration by suitable rotor construction, such as shrouds, separate blades, or wire lacing. In view of the manufacturing cost of turbine rotors, the choice reduces to doing whatever possible with the nozzle which is the major source of excitation.

The magnitude of the excitation may be estimated by calculating the component of lift force in the direction of blade flaps. At design condition, this force results in an estimated root bending stress of 560 psi. If we assume that as the blade passes the wake from the nozzle the lift force disappears, then we have a first flap bending excitation varying between zero and something greater than 560 psi. Our own experimental work and the theoretical work of others, e.g., Evans (11) and Armstrong (17), has shown the complicated nature of rotor vibrations added to which are effects of very low levels of damping. Typical values of damping expressed as "q factor" are about 10 to the power of three, so that without some mechanism of aerodynamic damping, the bending fatigue stresses of the power turbine blade would be very high. In practice, the worst condition occurs in reverse nozzles where the flow is a series of jets with large dead zones in between.

We have carried out blade vibration investigations under actual engine running conditions by attaching strain gages to the blades. In Appendix 1, a description of the strain measurement system is given. A typical test result is shown conveniently in Fig. 13, which is a diagram of a so-called "2 mod" storage oscilloscope display where the intensity of the spot gives an indication of the magnitude of a resonance.

There are a number of interesting points as follows:

1. The order lines 30 and 22 corresponding to the 30 variable vanes and the 22 fixed vanes are clearly seen.
2. There are a number of other order lines not easily explained.
3. There is an eighth-order line which is the difference between 30 and 22.
4. The first flap mode (1F) can be clearly seen as a horizontal line with resonances in the speed range.
5. The alternating stress levels of the most intense resonances are between 10,000 and 50,000 psi.
6. There are a number of so-called blade resonances at slightly different frequencies which are due to the fact that we have not just one blade vibrating in isolation but a complete rotor with a strain gage on one of many blades. The problems of "aliasing" and multiple resonances of bladed disks are discussed by Evans (11).

It is possible to make estimates of what might be a tolerable level of first flap vibration, but, in the case of this torsional mode, it is difficult to say what constitutes the effective stress amplitude as far as fatigue life is concerned. Our experience so far has been that the 1F mode gives the most trouble and we arrange to avoid engine operation where this mode is excited at high speeds.

While most of the vibration discussion has been concerned with the power turbine, it is possible to draw some general points which are relevant to all the bladed components. We have found that the main source of excitation is aerodynamic rather than mechanical, e.g., bearings, gears, gussets, or similar possibilities. Problems start with the lowest mode of blade or vane vibration, and one must look first at the source of excitation for this mode.
In the case of the compressor turbine, the golden rule is to ensure that first flap and nozzle blade passing frequency do not interfere anywhere in the running range. The consequences of hitting this vibration at, say, 10 kHz are a rapid accumulation of fatigue damage with obvious results.

In addition to hot strain gage testing in an engine, we carry out bench testing of bladed components to determine modes and frequencies. We use holography extensively, and without it, we would find it difficult to be sure of what we are looking at, and it is our practice to use the latest techniques available [19]. A typical hologram is shown in Fig. 14. These pictures serve two main purposes: [a] to help identify the true 1P or 1P mode and not be confused by one of the many disk modes at similar frequencies; and [b] to help identify the blade in question where to locate strain gages for the hot engine tests.

We see a continuing development of our understanding of integral rotor vibrations, while the overall aim is to go beyond the troubleshooting stage to estimates of fatigue life in service.

SUMMARY AND CONCLUSIONS

A description of the present compressor, compressor turbine, and power turbine of the British Leyland truck engine has been given covering aspects of performance, stress, and vibration. We have sometimes found it necessary to accept some compromise on one aspect to reach an acceptable standard on another. Careful rig work on both performance and vibration is essential, and the best results have been achieved where we have been able to substitute one improved component at a time, e.g., using the backshroud impeller with the already well-developed rotating guide vane, diffuser, and scroll.

We will conclude by suggesting where we think further performance improvements will lie.

COMRESSOR

The present flow range is acceptable, but we see the potential to raise the efficiency to 83 percent at the present design speed and the maximum pressure ratio to over 5:1. However, we shall give priority to retaining the high efficiencies in the pressure ratio range between 3 and 4:1, which is the important part of the operating range.

COMPRESSOR TURBINE

We expect to raise the efficiency 1 percent above our present target, i.e., to 87 percent at full load, and maintain higher efficiency at the lower speed and pressure ratio values corresponding to part load. It may well be possible, as our truck operating experience accumulates to raise the turbine inlet temperature by up to 20°C.

POWER TURBINE

We believe that future developments will raise the efficiency to between 83 and 84 percent (total to static) and, at the same time, improve the resistance to vibration particularly when the variable nozzles are turned toward reverse.

ACKNOWLEDGMENT

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APPENDIX I

VIBRATION MEASUREMENT SYSTEM

Fig. 15 shows the arrangement of the vibration strain measuring system for the power turbine. For tests on the compressor and compressor turbine, the slip ring unit is mounted at the front of the engine.

The system is based on four channels of information from strain gages on four separate blades spread around the rotor to ensure obtaining maximum vibration levels. The four channels being nominally identical give some redundancy to cover failures during the test; such events are not unknown. The unbonded wire gages are made by B.H. and the five-channel slip ring unit is our own development based on water-lubricated gold rings and brass brushes. With careful assembly and running-in, we achieve very low noise levels for this kind of work and signal-to-noise ratios of better than 20:1 at 0.1 percent strain signal.—usual; this varies considerably according to speed and condition of lubricating water. There
is nothing special about the amplifiers, but the tape recorder is an Ampex 1300 operating at 60 ips. We monitor the recorded signal via the playback heads using a four-channel CRO so that we are sure that the signals are being recorded and we also have a qualitative check on test progress. A problem in the early days of hot strain gage testing was gage life, but recent work has improved gage life on the power turbine to over 2 hr.

Need speed control has been found to be vital during engine vibration testing. Unless the rate of change of speed is kept well below 100 rpm/sec, then a depression of the vibration levels can be expected and so give misleading results. We have had little trouble with gasifier tests because the speed range is fairly narrow, 10,000 to 20,000 rpm, and we use a special ramped fuel/speed controller. But in the case of the power turbine, the test speed range is from say, 3000 to 37,000 rpm, which is comparatively large and cannot be easily controlled smoothly in the time available. The tape recorder at 60 ips allows about 10 min. per test, and there is little time for coping with a poor speed control. Therefore, interpretation of results must take into account any rapid changes in speed. It is of interest that a rate of speed change of 9000 rev/min. (33.3 rpm/sec) has been recommended by Armstrong and Stevenson [1].

To analyze the data recorded on the tape, we have several approaches, all of which we use to varying degrees depending on what we wish to investigate.

1. The well-established method of UV paper traces is used to examine closely a particular resonance and to obtain a rapid and reasonably accurate estimate of vibration level.
2. The tracking filter method is used where we know what excitation orders are operating and we wish to look at behavior through the speed range along a particular order line. The method has the advantage that good estimates of strain level can be made quickly. It is of course, the most valuable way to get an overall view is the Z mod diagram which is simply an interference diagram as in Fig. 15. By using log and linear modes of modulating the brightness of 'Z' of the CRO display, a good idea is quickly obtained as to where trouble can be expected. From then on, a more detailed analysis at various speeds or frequency bands can be carried out as desired.

The combination of these three analysis methods gives corroboration on strain levels and, together with the information of four channels, gives reasonable confidence in the test results and allows us to make design modification having a fairly clear expectation of the outcome.

APPENDIX 2

COMPRESSOR DESIGN DATA

The results of a simple one-dimensional calculation through the compressor are given in the following to show the operating conditions of the components. The method follows that used by Cheshire [15], so we separate the power input factor (p) and the slip factor (s).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet pressure</td>
<td>14.7 psi, 300 K</td>
</tr>
<tr>
<td>Mass flow</td>
<td>3.9 lb/sec</td>
</tr>
<tr>
<td>Rotational speed</td>
<td>37,000 rpm</td>
</tr>
<tr>
<td>Impeller tip diameter</td>
<td>229 mm (9.016 in.)</td>
</tr>
<tr>
<td>Rotating guide vane tip diameter</td>
<td>133 mm (5.236 in.)</td>
</tr>
<tr>
<td>Rotating guide vane hub diameter</td>
<td>57 mm (2.224 in.)</td>
</tr>
<tr>
<td>Diffuser inlet diameter</td>
<td>252 mm (9.921 in.)</td>
</tr>
<tr>
<td>Diffuser exit diameter</td>
<td>270 mm (10.627 in.)</td>
</tr>
</tbody>
</table>

The axial velocity, assumed uniform, of 474.9 fps gives the following conditions at the rotating guide vane.

<table>
<thead>
<tr>
<th>Diameter</th>
<th>Air Angle, deg</th>
<th>Relative Mach No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tip, 133 mm</td>
<td>60.7</td>
<td>0.866</td>
</tr>
<tr>
<td>Above fillet radius, 64 mm</td>
<td>49.6</td>
<td>0.559</td>
</tr>
</tbody>
</table>

assumed that resonances occur at integral orders of engine speed, but there is little evidence to doubt this.
3. The real-time analysis method is used in various ways to extract information in different forms from the recorded data. If we are investigating a new blading arrangement in the engine, in this design, the rotating guide vane tip diameter is very close to that producing the minimum Mach number for the mass flow, rotational speed, and hub diameter.
1. Impeller total temperature rise:

\[ \Delta T = 1.65 \times 0.383 \times 146 \times 0.89 \times 11 \times 62 \times 8 \times 68 \times 32.2 X 400 \times 0.242 \]

2. Compressor adiabatic (or isentropic) efficiency = 0.875

3. Total pressure ratio = 4.20

To find the air conditions in the impeller and diffuser, some assumptions on the distribution of losses have to be made. It is accurate enough for this purpose to use the concept of polytropic or small stage efficiency to relate the density everywhere to the inlet density and the static temperature ratio (16). Applying this gives the results tabulated in the following.

<table>
<thead>
<tr>
<th>Absolute Angle (to tangential), deg</th>
<th>Relative Angle (to radial), deg</th>
<th>Absolute Velocity, fps</th>
<th>Absolute Mach No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impeller outlet</td>
<td>15.9</td>
<td>22.4</td>
<td>1594.9</td>
</tr>
<tr>
<td>Diffuser inlet</td>
<td>13.6</td>
<td>--</td>
<td>1119.1</td>
</tr>
<tr>
<td>Diffuser outlet</td>
<td>24</td>
<td>--</td>
<td>361.4</td>
</tr>
</tbody>
</table>

There are three points to be made from these figures:

1. The vaneless space reduces the diffuser inlet Mach number from over unity to approximately 0.9.
2. The diffuser exit velocity is much too high to discard, emphasizing the importance of the scroll.
3. It can be seen that the scroll automatically results in a relative angle for the air leaving the impeller of 22.4 deg from the radial direction against rotation. By using backward swept vanes the designer can control this angle to any increased value desired.

In the data for this section, the compressor dimensions are given in millimeters with the inch equivalents. This is because our drawings are metric, but our calculations use the "foot, pound, second" system.

APPENDIX 3

NOTES ON TURBINE RIG

In the sections on the turbines, performance characteristics were given Figs. 9 and 11. This section briefly describes the rig used and the methods of measurement.

The rig consists of the turbine being tested coupled through a reduction gearbox to a dynamometer. It is supplied with air at about 150°F from a slave compressor. To obtain turbine characteristic measurements of inlet total pressure and temperature, air mass flow, outlet pressure, and output power are required. In the case of the power turbine, we are interested in total to static efficiency so that the required outlet pressure is the mean static at the inner and outer walls. In practice, we find that this static pressure does not differ from that in the ducting leading to the heat exchanger. Therefore, the total to static efficiency includes all the flow losses as far as the heat exchanger disk surface. In the case of the compressor turbine, we assess total to total efficiency from...

REFERENCES
