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Turbocharging and Turbochargers

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TURBOCHARGING AND TURBOCHARGERS

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Evolution and outlook of turbochargers for vehicle engines

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1 INTRODUCTION

Turbocharging of engines for on-road vehicles, construction/earth moving and agricultural equipment, and for the small engines in the industrial, marine, and aircraft markets, had its serious start of development around year 1950.

At that time, turbocharging of the large engines for marine, industrial and locomotives was already well established because of the benefits of primarily increased power density and secondarily improved specific fuel consumption. The turbocharger designs for these large engines were predominantly of the following type:

- o Centrifugal compressors.
- o Axial flow turbines with turbine housings of both divided and undivided type.
- o Water cooled hot sections.
- o Mostly outboard bearing arrangement, straddling both wheels.
- o Mostly self-contained lubrication systems.
- o Rolling contact or plain bearings.

The turbocharger frame sizes covered mainly diesel engines from approximately 800HP per turbocharger and up to the very large units for the largest marine diesels and were developed and manufactured by both the engine companies and a very few independent suppliers at relatively small production rates.

Around year 1950, some vehicle manufacturers foresaw the need for higher output engines and got seriously involved in evaluating turbocharging as a means to achieve that goal. A few of their engines were already supercharged with mechanically-driven Roots blowers. Their own expertise in turbomachines was, in most cases, limited to waterpumps and hydrodynamic transmissions. Their evaluation of the existing types of design and sources of supply led them to new candidate sources for developing and manufacturing more appropriate types of designs for the vehicle markets.

Next, I will first give an overview of the evolution of the vehicle turbocharger design for the commercial diesel markets; second, address the application to the passenger cars; and third, take a glance at the future.

2 COMMERCIAL VEHICLE ENGINE TURBOCHARGER DESIGN

The early, major drive for this development was the need for increased power density and reduced specific fuel consumption.

Some pioneering engine companies and a few newcomers in the turbocharger field, in a close and very beneficial cooperative effort, pooled their resources and arrived fairly quickly at the following design concept:

- o Double overhung wheels to bearings arrangement with a centerhousing containing all bearings and seals.
- o Radial flow wheels for both compressor and turbine.
- o Plain bearings, pressure lubricated by engine oil system.
- o No water cooling.

This design concept promised to reach the objectives of small package size, weight, and cost while having a good potential for development of the necessary level of aerodynamic performance, mechanical ruggedness, durability, and reliability.

These early design choices appear to have been good decisions since they are still utilized today, some thirty years later. Other alternate choices were, of course, pursued to varying extents. They, however, disappeared or never reached production status.

Within the frame of this design concept, the vehicle turbocharger evolved into a very simple, but sophisticated product as a consequence of strong competition among suppliers and of advances in technology in a variety of disciplines. It was rather the cumulative effects of many detail advancements than some breakthroughs which characterized the progress made over the years.

2.1 Radial Inward Flow (Centripetal) Turbines

Although radial flow machines have been used in pumps, compressors, and hydraulic turbines for a long time prior to 1950, it was surprising that radial inward flow turbines for compressible fluids stayed an orphan until the late 1940's when their introduction into small gas turbine engines really demonstrated their capabilities. They were a key item for the success of vehicle turbochargers due to their ruggedness while maintaining good efficiency and because the wheels could be economically produced as single piece castings by the investment process. The single piece design is free from major stress risers and made it feasible to use high temperature casting alloys which had only small elongation properties. This was in the days when I was told that a turbine for a turbocharger better be capable of digesting pieces of piston rings and piston lands!

The early designs of turbine wheels were of the full disc type. They gradually disappeared in favor of deeply scalloped, or almost star-type shapes, which reduced transient thermal stresses and moment of inertia.

Friction welding the turbine wheel to its shaft became an almost universal solution and replaced the early mechanical, brazed, or electron-beam weld joints.

Turbine Nozzles: The early traditional vaned nozzle design was mostly gradually replaced by vaneless volute turbine housings with only a modest loss of efficiency in the higher specific speed versions of turbines. This resulted in smaller diameter and lighter turbine housings and eliminated the installed cost of vaned nozzle rings. The vaneless housing design, however, shifted the wheel blade exitation frequencies and increased their intensity which required increased stiffness of the blades, particularly for the high specific speed wheel trims.

Divided Exhaust Systems: The early turbocharged vehicle engines had undivided exhaust systems and, therefore, undivided turbine housings. The advantages of divided exhaust systems are pulse separation for better gas-air exchange in the cylinders and pulse energy conservation and utilization in the turbine for increased boost pressure at low engine speed and faster turbocharger acceleration. As these advantages became more evident to the engine designers the turbine housings had to be divided too. The housing division first utilized was of the sector (Buchi) type as conventionally used in the large axial flow turbine turbochargers. However, for the radial turbine, the meridional division featuring two separate 360° volutes is more efficient and has become the industry standard, at least for four and six cylinder engines.

Turbine Controls: Early turbocharger/engine performance matching studies revealed the difficulty to obtain the desired vehicle engine torque versus speed characteristic while avoiding overboost and excessive firing pressures in the high speed range. This is a consequence of the flow characteristic of the turbine which is not ideal for the positive displacement type engine flow characteristic. Early attempts to solve this inherent incompatibility by means of the well known (practiced in hydraulic turbines) pivotable nozzle vane design were not successful and way ahead of their time. There were other much more important problems in the turbocharger which had to be solved first. In the applications where turbine controls were an absolute requirement, i.e, aircraft engines and natural gas engines, the thermodynamically less efficient but mechanically much simpler turbine bypass valve or wastegate became the accepted solution to achieve the desired variable turbine flow characteristic.

A large number of inventions to achieve the same result without bypassing the turbine were conceived, however, without sufficient benefits for commercial vehicles. Also, attempts to achieve similar effects by applying controls to the compressor were bound to fail because they did not attack the problem at its origin.

In general, the vehicle, engine, and turbocharger engineers learned to live with this situation and to minimize the compromise by optimizing turbocharger performance at lower engine speeds and by accepting the decay in turbocharger efficiency at the upper end of the engine speed range. Only more recently wastegates have also been introduced into some truck applications.

2.2 Compressors

Some of the early vehicle turbochargers featured backward curved compressor wheel blades with their inherent wide operating range and high peak efficiency. They, however, suffered from limited tip speed capability. When the engine manufacturers pushed for higher boost pressures, the backward curvature disappeared in favor of the lower stress 90° blading.

The introduction and development of the rubber pattern/plaster mold aluminum wheel casting process was one of the most significant contributors to the turbocharger progress. It resulted in low cost, high fidelity wheels with excellent finished-shape cast aerodynamic surfaces, including leading edges. Material and process development, in conjunction with finite element stress analysis techniques, made it possible to reintroduce moderately backward curved designs. Nevertheless, in spite of all the progress, cast aluminum compressor wheel disc/hub fatigue remains a limiting factor for the highest boost pressure cyclic load applications.

Diffusers: With the exception of some special cases which use vaned diffusors, the vaneless diffusor combined with volute-type compressor housings became predominant because of the wider flow range. With it, the high frequency aerodynamic noise greatly disappeared.

Performance: The development of sophisticated computer codes, driven by the gas turbine industry, greatly benefited the aerodynamic performance. Also, novel time and cost effective N.C. manufacturing methods for development hardware contributed to quick experimental evaluation of variations in aero and stress designs.

2.3 Bearings And Rotor Dynamics

Although rolling contact bearings were successfully used in small gas turbine engines, they were not cost-competitive and not durable and reliable enough for the prevailing environmental conditions in vehicle turbochargers. Engine oil pressure lubricated plain bearings became the norm. Most of the journal bearing designs feature floating or semi-floating bushings with double oil films. A lot of intelligent development, much of it of an experimental nature only, in conjunction with improvements in the engine lube system resulted in fairly forgiving bearing/lube systems with acceptable durability, reliabilty and functional performance.

Their high load capacity made it feasible to eliminate the traditionally used group balancing operation of the total rotating assembly which greatly simplified the turbocharger assembly in the factory and in field overhaul. It also had sufficient margin for extending the turbocharger frame sizes down to the smaller vehicle engines. It was, furthermore, possible to live with a balancing accuracy which was essentially within the existing technology level of the balancing equipment industry.

Nevertheless, when analyzing field returned units, one cannot avoid coming to the conclusion that there must be opportunities for further progress.

2.4 Transient Engine Torque Response

The turbocharger has a very nice feature. It is self-regulating, to some extent, without use of any controls. At steady state part - BMEP, it does not unnecessarily oversupercharge the engine as an engine-driven supercharger would. The consequence of this feature is, of course, the "turbo lag" which is most pronounced at low engine speeds, and increases with increasing engine rated power. Many paths were pursued to minimize this effect.

Reduction in turbocharger rotor moment of inertia, or more accurately, reduction of the kinetic energy of rotation was achieved by the development of units with high specific speed (small wheel diameters), way above what is common in small gas turbines at the expense of somewhat reduced component efficiencies. More optimum stress designs of wheels which best utilize every element made a further contribution.

The introduction of divided exhaust systems, as already mentioned earlier, provided a significant improvement in engine torque response and became an industry standard. Turbocompounding, with the compound turbine as the low pressure expansion stage, also significantly reduces the turbo lag and flattens the full load boost pressure versus speed curve. Development and use of turbine flow controls are in the early stages.

Many other systems, which can be characterized as power transfer systems, have been conceived and tried. They use other than engine exhaust gas energy to directly or indirectly help to accelerate the turbocharger and/or also to raise bottom end boost pressure during steady state at low engine speed. The sky is almost the limit for what can be conceived. A few of these ideas continue to be pursued on an experimental basis.

The most drastic approach has, of course, been the bypassing of the engine and operating the turbocharger as a gas turbine with its own combustor.

2.5 Competitive Systems

In the commercial diesel markets, turbocharging has become the only industry standard for increasing the power density. At times, I thought that the Comprex had the potential to become a serious contender. Maybe that belief was influenced by the fact that I spent my early professional years near the birthplace of the Comprex and that I was intrigued by the gas dynamical elegance of its concept.

3 TURBOCHARGERS FOR PASSENGER CARS

After a short, and before its time (1961, 1962), limited production, turbocharging of both gasoline and diesel cars became quite serious in the late seventies. It certainly represented a major milestone. The initial driver was car fuel consumption legislation in the United States which forced car manufacturers to eliminate the large displacement engines. In order to retain the high performance of the top of the line segment of the market, turbocharging found its entry.

Passenger car turbocharging set new requirements/challenges. The major ones are:

- o Higher peak exhaust temperature up to about 1000°C.
- o Broader flow rate operating range.
- o Engines more sensitive to peak pressures, requiring controls.
- o Higher torque responsiveness.
- o Miniaturization of turbo machines.
- o Lower load factors.
- o Less noise.
- o Tighter packaging.
- o Larger production rates.
- o Lower cost.
- o Consumer market.
- o More lax practices of operation, service maintenance.

A lot of progress has been made towards meeting these requirements. I will only touch on a few major ones.

The high exhaust temperature applications necessitated the use of more expensive alloy turbine housings and, in some cases, heat damming and even watercooling of bearing housings.

Wastegate turbine flow controls, integrated into the turbine housing and pneumatically actuated by compressor discharge pressure, became the most commonly used means of limiting boost pressure. Some integration of the turbo controls rationale with engine controls has been achieved in some applications.

More efficient ways of controlling boost pressure are being explored experimentally. Progress is not coming easily.

A large effort has been put in the development of ceramic turbine wheels for two major reasons. First, further reduction of turbo lag and second, reduction of weight and thereby cost of high alloy turbine housings as a consequence of the easier containment requirement in case of accidental wheel burst. It is hoped that these efforts will pay off in a few years.

Turbocharger-caused noise: This phenomenon was not expected and caught the industry by surprise. It is not the traditional aerodynamic noise of turbomachines and has its root in the rotor dynamics area. It turned out to be of a rather tenacious nature. The predominant frequency of the noise spectrum coincides with the rotational speed of the turbocharger. The noise has the character of a tone which stands out to the human ear from the more grey type of general background noise in a car. The turbocharger housings forced-vibrate, although with very small amplitudes only, and emit noise by themselves as well as transmit vibrations to other parts in the power plant and car which, in turn, become noise emitters. The industry has been able to reduce the problem to a large extent. However, it still is a headache to achieve 100% success in production.

Competitive Systems: The industry has always, and still does, investigate and develop other systems with the objectives of lower cost and better bottom end steady state and transient engine torque. The major contenders are the Comprex and mechanically driven superchargers. The many different concepts of superchargers of the positive displacement type as well as their drives and the, unfortunately, necessary controls have their own challenges also and, so far, have not yet had an impact on the market.

One of the most significant consequences of passenger car turbocharging was the emergence of additional turbocharger manufacturers and the tremendous increase of the staff of product engineers and manufacturing engineers, most of them devoted to advance the turbocharging/supercharging technology of vehicle engines.

4 OUTLOOK

Turbocharging, and in a broader sense, airsupply systems for IC engines, continue to be still a fascinating task not only for the engineers specialized in the many disciplines, but even more for the business managers who have to allocate the resources and direct the most appropriate efforts at the right time. Legislation, evolutionary developments, and major new technologies will influence reflectively both the engine and transmission side as well as the airsupply system side of the drive-trains. Close cooperation between them will continue to be very beneficial while maintaining the necessary competition and will result in best integrated total systems.

Let me point out a few areas in which advancements can be expected or should be actively sought.

- Because of the periodic flow character of engines, turbocharger aerodynamical know-how should be further developed for the fluctuating and always "off-design point" instantaneous modes of operation.
- Plain bearings in these small high-speed machines, depending on the feeding schemes of the oil films, do operate temporarily and locally in a two-phase (air/gas and solid oil) mode. Better understanding of these physical mechanisms will help to arrive at bearing/rotor dynamical systems with improved durability, less friction and noise.
- Re-examination of old conceived/tried but abandoned concepts in light of advanced or new technologies or other changed circumstances may be beneficial.
- Reduced tolerance manufacturing methods and precise removal or addition of very small amounts of balancing material would help in the effective miniaturization of turbomachines.
- Innovations in small casting process technology and designs for further automation in manufacturing will reduce cost.
- Advances in design, material/process technology of compressor wheels will help to reach higher boost pressure commercial engines.

In conclusion, I feel that turbocharging of commercial diesels will continue to increase its share and that turbocharging and supercharging will fight for their own shares in the top of the line passenger cars and in the related light duty commercial vehicle market, and be strongly influenced by fuel prices, legislation and new technologies.

Experimental investigation of the fluid dynamics of alternative turbocharger compressor volutes

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SYNOPSIS The diffusion system of turbocharger compressors is usually composed of three components, a vaneless diffuser, a collecting volute, and a discharge conical diffuser. The usual practice is to diffuse fully in the vaneless diffuser with the volute acting as a collector only. This arrangement is extremely inflexible particularly when considering techniques that may be available for extending the stable operating range of the compressor.

Two vaneless diffuser/volute systems have been designed and manufactured as alternatives to the standard arrangement. The first design transferred part of the diffusion process to a conical diffuser downstream of the volute, whilst the second design allowed diffusion to take place in the collecting volute. Experimental results are presented in the form of detailed pressure measurements in the system and as overall compressor characteristics in order to compare the performance of the alternative designs. The overall performance of the prototype volute compares favourably with the standard design despite the significant reduction in vaneless diffuser radius ratio and the use of full tongues in the new volutes.

1 INTRODUCTION

Current developments of turbochargers to Diesel engine power plants of trucks and buses include the application of variable geometry turbines. It has been shown (1,2) that through the application of variable geometry, in which the swallowing capacity of the turbine can be adjusted, optimum turbocharger matching for torque back up, transient response and low specific fuel consumption can be achieved over the full operating range of the engine. However, it has also been shown that if the full potential of these developments is to be realised the development of the associated compressor with wide flow range to give adequate surge margin will be necessary.

Over the past decade substantial effort has been applied to the design of turbocharger impellers in order to improve both their operating range and efficiency. The diffuser system, usually consisting of a vaneless diffuser followed by a collecting volute, has remained largely unchanged.

The diffusion process in radial vaneless diffusers of centrifugal compressors is fundamentally different from that in straight conical diffusers. This difference arises due to the extremely high degree of swirl at inlet to the diffuser. Diffusion is achieved by reducing the tangential component of velocity, this being accomplished due to the conservation of angular momentum, the tangential component of velocity reducing in direct proportion to the increasing radius. Therefore, the larger the radius ratio of the diffuser the greater is the pressure recovery ideally attainable. Brown and Bradshaw (3) showed that no improvement in performance was obtained by increasing the radius ratio from 1.8 to 3.0, the long logarithmic flow path through the diffuser leading to large friction losses. Direct reduction of the radial component of velocity by increasing the cross sectional area of the vaneless diffuser only has a very small effect upon the total diffusion process. Equally attempts to improve diffuser performance by employing techniques commonly used in conical diffusers, i.e. boundary layer suction or energisation, to ensure full utilisation of the area ratio available will only yield limited improvements.

The vaneless diffuser is, therefore, extremely inflexible in that conventional techniques used for improving conical and rectangular diffuser performance are not applicable. In order to seek improvements in diffuser performance and flow stability, a study was instigated into the complete diffuser system, consisting of a vaneless diffuser, collecting volute, and a discharge conical diffuser. Due to the inflexibility of the vaneless diffuser the study has been aimed at transferring part of the diffusion process from the vaneless diffuser to the collecting volute and/or the discharge conical diffuser. The main problem that this will generate is the efficient collection of high velocity fluid in the volute prior to diffusion in a conical diffuser, or if diffusion is allowed in the volute itself, the development of a peripheral pressure gradient around the impeller. On the other hand, Abdelhamid (4) has shown that diffuser flow stability is improved by reducing the vaneless diffuser radius ratio. He also demonstrated that the application of throttle rings at the vaneless diffuser discharge also improved the flow stability. It may, therefore, be possible to obtain similar improvements by suitable vaneless diffuser and volute design.

Published information on the performance of volutes is very limited. Brown and Bradshaw (3) studied a series of diffuser volutes with a mixed flow impeller but found that changes in volute geometry and surface conditions resulted in negligible differences in compressor performance. Current volute designs do not, however, correspond to those tested by Brown and Bradshaw.

Stiefel (5) studied the optimisation of impeller, vaneless diffuser and volute, and found that stable operation could be achieved to high pressure ratios by reducing the size of the volute by up to 30%.

2 ALTERNATIVE VOLUTE DESIGNS

In addition to the standard turbocharger vaneless diffuser and volute two alternative prototype diffuser/volutes were designed and manufactured. The design procedure was largely a geometric exercise carried out with the aid of the computer aided design package 'DUCT' which ultimately provided the necessary control tapes for machining purposes. The objective of the investigation was to transfer part of the diffusion to the collecting volute or a discharge conical diffuser; the exit radius of the vaneless diffuser was, therefore, arbitrarily reduced from a radius ratio of 1.625 to 1.4.

The first prototype diffuser/volute (P1) did not allow any diffusion in the volute, the diffusion taking place in an exhaust conical diffuser. The second prototype volute (P2) allowed diffusion to occur in the volute, the final discharge area being the same as that of the standard volute. For this second prototype volute the discharge radius of the vaneless diffuser was allowed to increase with azimuth angle to the full radius of the standard vaneless diffuser. The pressure rise with azimuth angle in the volute then being matched by a similar rise around the vaneless diffuser. The variation of area and radius to the centre of area of each volute is shown in Figs. 1 and 2 respectively. The standard volute has a tongue at 60° of azimuth angle, see Fig. 1 for definition of zero azimuth angle, which allows a recirculatory flow area of approximately 200 mm². For the prototype volutes the tongue commenced at the zero azimuth angle position and allowed no recirculatory flow area; the intention being to cut this tongue back for testing at a later date.

Downstream conical diffusers used were of 6.3 deg. included angle for the standard and P2 volutes, and 11 deg. for the P1 volute, all enlarging to the 5000 mm² area exit duct. The vaneless diffuser width was 5.16 mm at impeller exit (50.4 mm radius), converging at an angle of 12½ deg. to 5.1 mm and parallel from there to diffuser exit.

3 EXPERIMENTAL TEST FACILITY

Evaluation of the alternative volute designs was performed by comparing the overall compressor performance with that obtained with the standard volute. By detailed pressure measurements in the volute and vaneless diffuser, the effect of each design on the static pressure field in the vaneless diffuser and volute was studied. Since an integral part of the study was to be the transfer of diffusion downstream to a conical diffuser with expectations of improved performance of that diffuser due to swirl generated by the volute, traverses of the flow passage both upstream and downstream of the conical diffuser were included.

Two different impellers were available for the investigation, both having the same shroud profile and exit tip diameter, thus permitting complete interchangeability on the test rig. The radial impeller had six full blades and six splitter blades, while the backswept impeller had an exit blade angle of 15 degrees, eight full, and eight splitter blades.

The test compressor was part of a standard turbocharger assembly. The radial inflow turbine, which was directly connected to the test compressor, was driven by compressed air from a reciprocating compressor and exhausted to the atmosphere. The test compressor drew air from the atmosphere, and via a throttle valve and metering orifice returned the air to the atmosphere in an open loop. Performance results are presented for the radial impeller only unless noted otherwise.

Pressure tappings were spaced at intervals of 18 degrees of azimuth angle around the volute and through most of the vaneless diffuser. In order to better resolve pressure patterns in the vaneless diffuser around the tongue, the spacing was decreased to 9 degrees over a 72 degree sector. The position of this sector could be adjusted by rotation of the bearing housing relative to the volute to align it with the tongue position of the particular volute being tested.

Adequate coverage in the radial direction to enable inferences to be made about the applicability of free vortex assumptions in the vaneless diffuser and volute flow led to a distribution of between three and five rings of tappings in the diffuser (depending on which volute was used), and in the volute, at the outer, top and inner extremes of the volute cross sectional profile (see Fig. 3). In addition, it was possible with the standard volute to install tappings on either side of the diffuser exit lip (rings 6 and 10, Fig. 3).

Temperatures were measured with platinum resistance thermometers, readings being taken under computer control by sequential switching of the thermometers to a conditioning amplifier. The static pressures were read again under computer control by means of a single pressure transducer and a scanivalve. Both the pressure and temperature recordings were arranged to read a calibration value once in each operation cycle. No attempt was made to measure pressure fluctuations associated with stall and system surge.

4 PRESENTATION AND DISCUSSION OF RESULTS

In terms of overall performance maps the results obtained with the standard and two prototype volutes are presented in Figs. 4 and 5. It is apparent from Figs. 4 and 5 that there are serious deficiencies in pressure head and high mass flow performance for the P1 volute compared to the standard volute. Overall stage maximum efficiency is reduced from 76.9% to 75.6%. This reduction is accompanied by the development of a downward gradient of efficiency (in the direction of increasing speed) over the whole length of the maximum efficiency line tested. Pressure ratio deficiencies relative to the standard volute also exhibit a speed dependent relation, varying from 0.01 at 40krev/min. to approx. 0.1 at 70 krev/min. These differences increase rapidly with mass flow beyond the 70% efficiency contour, thus reducing the operating range of the compressor.

These mass flow and impeller speed proportional drops in pressure ratio suggest a frictional loss increase for the P1 volute over the standard volute which correlates with the higher gas velocities expected with collection at a reduced vaneless diffuser exit radius. The change in gradient of efficiency along the maximum efficiency line further reinforces this conclusion. The surge line and maximum efficiency line are both moved to higher mass flows with further loss of operating range.

The P2 volute gave a small improvement in efficiency with the radial impeller, Fig.5, though with the backswept impeller no significant change was visible. Efficiency contours describe broader areas and maximum efficiency, although at a lower speed, is up by 1.1% over the standard volute with the radial impeller.

The pressure ratio results of the P2 volute show some of the characteristics of the P1 volute, dropping away on either side of the high mass flow 75% efficiency contour for both impellers relative to the standard volute, but with the drop on the surge side increasing with impeller speed. Over the central portion of the map and over the whole low flow part of the 40 krev/min. line up to the 70% efficiency contour the pressure ratio lines for both volutes are close. At high mass flows the pressure ratio falls off as with the P1 volute but in a less severe manner.

Surge performance is better with the standard volute. The P2 volute imposes a maximum penalty in terms of mass flow units of 0.85 for the radial impeller at 50 krev/min., and 0.65 for the backswept impeller at 63 krev/min. This reduces to between 0.1 and 0.3 for other speeds.

In order better to assess the characteristics of static pressure distribution through the diffusion/collection zone, and their relationship to the operation of the compressor, contours of constant standard deviation of relative static pressure (of static pressure rise to stage static pressure rise ratio, with azimuth angle), in the diffuser are presented on the performance maps of Figs. 6,7 and 8. The relationship for standard deviation was derived from a formula given by Bryant (6) for estimating the standard deviation from finite samples of infinite populations.

$$\frac{\sum_{1}^{n} \Delta\theta_i\, p_i^2 - \left(\sum_{1}^{n} \Delta\theta_i \cdot p_i\right)^2 / \sum_{1}^{n} \Delta\theta_i}{\frac{n-1}{n} \sum_{1}^{n} \Delta\theta_i}$$

For the case of constant intervals of azimuth angle between tappings, (such as in the volutes, or rings 6 and 10 in the standard volute), this reduces to

$$\frac{\sum_{1}^{n} p_i^2 - \left(\sum_{1}^{n} p_i\right)^2 / n}{(n-1)}$$

The general trend for all the results is for the contours of standard deviation to form regular, slightly radiating patterns with little or no curvature. There is a markedly greater gradient in the variation of standard deviation with mass flow towards choke than towards surge, while there is negligible gradient along the line of minimum standard deviation (marked as s minimum on the figures); i.e. for increase in speed, but following the s minimum line, the relative static pressure distribution with azimuth angle remains constant. Contours are approximately parallel from map to map, although the severity of pressure distortion, reflected by the magnitude of the standard deviation, does change with configuration. The backswept impeller shows a better match of minimum standard deviation to maximum efficiency loci than the radial impeller; this then corresponds with the better overall performance obtained with the backswept impellers. Comparison of the minima of standard deviation for each map, (the numbers adjacent to, or arrowed to rings marked on the s minimum line), shows that volute P2 produces the lowest minimum of .010 versus .012 and .022 for the P1 and standard volutes respectively. The close spacing of contours for the P1 volute is contrasted by the wider spacing of standard and P2 volutes, and reflects their relative mass flow ranges.

Lines of minimum standard deviation of pressure in the volute for the top ring (see Fig. 3) are also shown and the divergence of these lines from the diffuser s minimum lines for the P2 volute is a graphic demonstration of the diffusion occurring in the volute passage for that volute.

Consider that, for the P2 volute, when the static pressure at a given radius in the diffuser is constant with azimuth angle, (s minimum), the pressure in the volute will be increasing with azimuth angle, since diffusion is occurring in the volute passage. Thus the standard deviation of pressure in the volute passage will not be a minimum. Now, in general, for both volute and diffuser, as mass flow increases, the gradient of pressure with azimuth angle tends towards the negative, (see Fig. 9). So for a minimum standard deviation of volute pressure for the P2 volute, the positive gradient due to diffusion in the volute would have to be offset by a shift to a higher mass flow.

5 VOLUTE EXPERIMENTAL STATIC PRESSURE DISTRIBUTION

Figs. 9,10 and 11 show pressure distribution in the three volutes at outer, top and inner rings; and at choke, s minimum, and surge operating points at 50 krev/min. for the radial impeller or 52 krev/min. for the backswept impeller. It will be noticed that at choke for the standard volute, pressures at outer, top, and inner rings would appear to coincide at an azimuth angle of 150 degrees, (although inner ring tappings for the standard volute start only at an azimuth angle of 180 degrees). Corresponding curves for the prototype volutes show a marked pressure difference between the inner ring and top ring throughout the curved axis portion of the volute passage, though top and outer pressures coincide at around 60 degrees azimuth angle.

In the standard volute, wall tappings were additionally provided near diffuser exit, both on the shroud surface and in the volute just upstream of the merging of volute and diffuser flows (see Fig. 3). Fig. 12 shows comparisons, at three different compressor operating points, of pressure in the standard volute at ring 10 (the back of the diffuser exit lip) with ring 8 (the top of the volute). Pressure at the back of the lip is below that at the top, more especially at high mass flow, and is very similar to that at the inner ring, Fig. 9.

Similar plots for the prototype volutes do not show such a large disparity of pressures between the top of the volute and locations of similar radius on the diffuser hub surface. Fig. 13 shows a comparison for the P1 volute between top and ring 4 pressures. In this case ring 4 is at a larger radius from machine axis than the top position, reflected in the higher pressures shown for ring 4.

Fig. 14 compares top and ring 5 pressures for volute P2 and shows a slightly higher pressure at ring 5 over the full range of azimuth angle. Ring 5 is at a larger radius than the top ring up to 300 degrees azimuth angle, after which, due to the spiral plan of the volute passage centre lines, the top ring is at a larger radius.

5 CONCLUSIONS

The performance of a turbocharger compressor with two prototype diffuser-volute systems has been compared to the performance with the standard turbocharger volute system. The two prototype volutes are substantial modifications to the standard design with a significant reduction in vaneless diffuser radius ratio and, in the case of the second prototype, diffusion being permitted in the volute passage.

The successful transfer of diffusion from the vaneless diffuser to the volute in the case of the P2 volute, and to the downstream conical diffuser in the case of the P1 volute has been demonstrated. The transfer in both cases has not itself led to any directly attributable performance penalties. However, collection at a smaller diffuser exit radius does lead to losses thought to arise from the increased friction effects of the higher volute flow velocity. At peak efficiency these amounted to approximately 1% for the P1 volute, but at higher mass flows both prototype volutes showed an increasing susceptibility to this effect as compared with the standard volute. Considering the significant changes in volute design the comparison, particularly of the P2 volute, with the standard design is encouraging, although no improvements in performance in terms of pressure ratio and surge margin are shown. The prototype volutes were designed with tongues which prevented any recirculating flows in the volute passage; the standard volute in contrast had the tongue cut back thereby allowing a signficant recirculatory flow area. The effect of the tongues upon the pressure variation with azimuth angle has been demonstrated and it is intended to further test these volutes with the tongues cut back. The prototype volutes have been manufactured in three parts so that design modifications can be readily carried out.

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REFERENCES

(1) WALLACE, F.J. and ZIARATI, M.R. 'Variable geometry turbocharging for transport engines'. Paper C38/82 Turbochargers and Turbocharging Conference. I.Mech.E. London 1982.

(2) FLAXINGTON, D. and SZCZUPAK, D.T. 'Variable area radial inflow turbine'. Paper C36/82 Turbochargers and Turbocharging Conference. I.Mech.E. London 1982.

(3) BROWN, W.B. and BRADSHAW, G.R. 'Design and performance of family of diffusing scrolls with mixed flow impeller and vaneless diffuser'. NACA Report 936 1947.

(4) ABDELHAMID, A.N. 'Control of self-excited flow oscillations in vaneless diffuser of centrifugal compression systems'. ASME Paper No. 82-GT-188.

(5) STIEFEL, W. 'Experience in development of radial compressors'. Von Karmen Institute for Fluid Mechanics Lecture, Series 50, May 72.

(6) BRYANT, E.C. 'Statistical Analysis'. McGraw-Hill 1966, 2nd Edition.

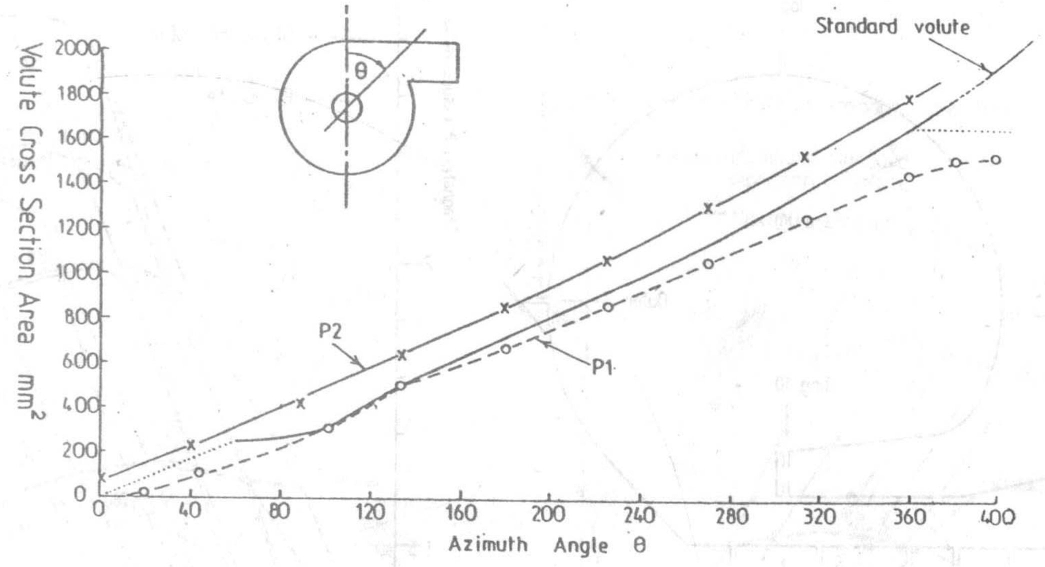

Fig 1 Variation of passage area around standard and prototype volutes

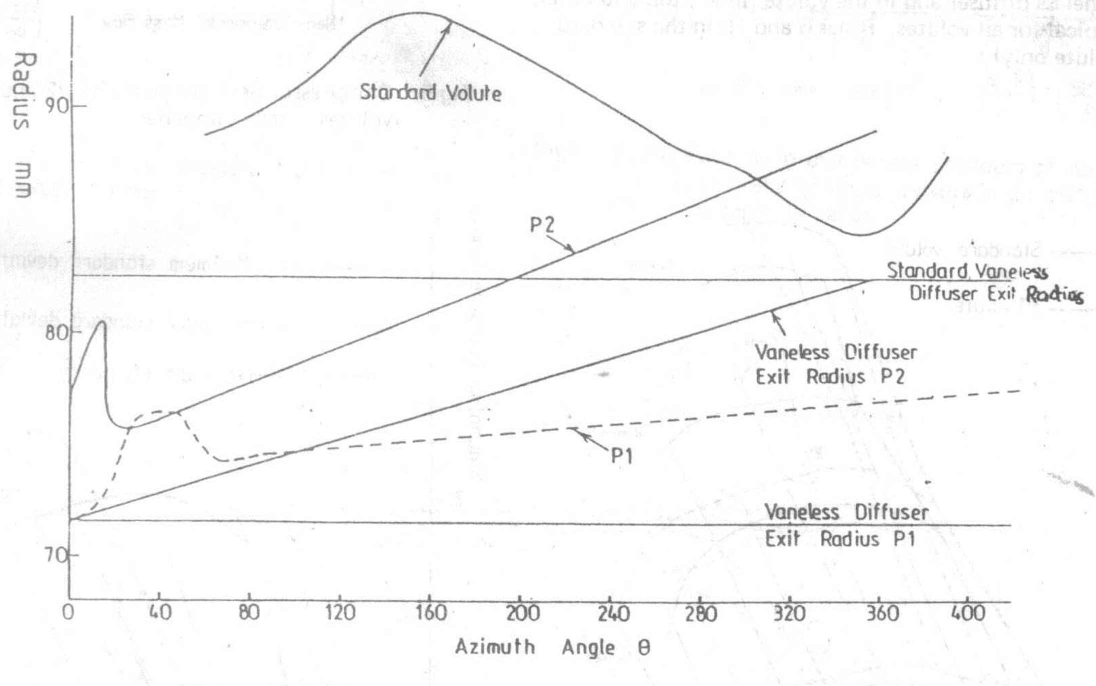

Fig 2 Variation of radius to section centroid around standard and prototype volutes