23-Se

DEVELOPMENT OF HIGH PRESSURE RATIO CENTRIFUGAL COMPRESSOR STAGE

S. Ramamurthy, S. Sankaranarayanan, K. Murugesan, D. Ramachar and R. Rajendran
Propulsion Division, NAL., Bangalore.

SUMMARY

A high-pressure ratio centrifugal compressor stage consisting of an impeller and a matched vaned diffuser were designed and tested in a closed circuit centrifugal compressor test facility. The detailed measurements were carried out both at subsonic and supersonic Mach numbers at impeller outlet. The stage characteristics showed achievement of compressor stage total to total pressure ratio of 6.5 with an efficiency of 85%. The vaned diffuser shows good static pressure recovery of the order of around 0.6 with total pressure loss less than 2% at design point. Some of the salient experimental results are discussed in this paper.

1. INTRODUCTION

Centrifugal compressors play an important role in small gas turbine engines, turbochargers, refrigeration plants, chemical and process industry. Use of centrifugal compressor stage in the development of small aeroengines have proven to be advantageous in achieving better specific weight as compared to axial stages. There is a growing tendency for design, modification and replacement of axial stages with a single stage centrifugal compressor in aero gas turbine engines.

With the above technology in mind, Propulsion Division of National Aerospace Laboratories had taken up the work on the indigenous development of high-pressure ratio centrifugal compressor stage. For this purpose a general-purpose design methodology for centrifugal compressor stage has been developed. Analytic functions have been newly evolved to define geometrical shapes for the impeller vanes. Numerical techniques and algorithms have been developed to transform the designed [1]vane geometry into a compatible form for machining the impeller in numerically controlled milling machine. Theoretical methods for flow analysis in the hub to shroud and blade to blade passages were developed and used for off-design performance prediction.

The initial work was started with a classical design of impeller. The impeller was machined using 3-axis numerically controlled machine in-house and tested in a Closed Circuit Compressor Test Rig [CLOCTER] for its performance. The tested impeller produces a total to total pressure ratio of 6 and an impeller total to total peak efficiency of 88 percent. Having successfully completed the design, development and experimental evaluation of a centrifugal impeller, a new compressor stage was designed keeping the same loading to get higher impeller performance and stage efficiency. The designed impeller is machined using five axis numerically controlled machine and the vane diffuser using 3-axis numerically controlled machine. The diffuser is suitably designed and matched to the impeller through an experimental iteration by varying the geometrical parameters. The compressor stage was assembled with suitable bearing systems in the closed circuit test facility. This compressor stage was experimentally evaluated for its performance under simulated conditions using Freon gas. The stage and the diffuser characteristics of this compressor are discussed in detail in this paper. The stage characteristics show that compressor stages total to total pressure ratio of 6.5 with an efficiency of around 85 % are achieved. At its optimum operating point the impeller efficiency is more than 90%. The vane diffuser shows a good static pressure recovery of the order of around 0.6 with total pressure loss less than 2% at design operating speed.

2 CENTRIFUGAL COMPRESSOR STAGE DESIGN

2.1. Aerodynamic Design

The centrifugal compressor stage was designed to meet the specific duty like mass flow rate and stage pressure ratio. The aerodynamic design of the centrifugal compressor is based mainly on an earlier developed methodology (Ref.1) taking into account considerations for diffuser design (Ref.2) and performance appraisals of past indigenous designs (Ref.3,4). The approach and details of the design methodology works on an iterative procedure. The two components of the stage comprising of impeller and vane diffuser were designed iteratively to get the required stage pressure ratio and the stage efficiency. The aerodynamic design of the impeller is based on the jet and wake model with loss correlation's supported by experimental results. The centrifugal impeller blade profiles were generated using an analytic equation proven to have an advantage in blade loading distribution. The semi-vaneless portion of diffuser after the impeller was configured to decelerate the supersonic absolute Mach number smoothly to unity at throat. The inlet condition to the vane diffuser was estimated by marching from impeller outlet to diffuser inlet using mass, momentum and energy equations. The vanes of the radial diffuser were designed to get the required pressure recovery for a given value of aspect ratio and length to width ratio. The leading edge was suitably configured to accommodate incidence variations over the operating range of the impeller. The profile of the vane diffuser from leading edge to the throat follows a special design..

2.2 Proto Type Vs Model Impeller

The designed impeller is appropriately scaled by keeping the aerodynamic parameters same, so that tests could be carried out under simulated conditions in freon. The prototype impeller is scaled to the required size to give the same work coefficient as that of a prototype in air. The diffuser is also appropriately scaled to match with the scaled impeller. The scaled model compressor could be tested over its full range of operation in the test rig. Appropriate formulations were used to account for density and ratio of specific heat variations between freon and air.

3. CENTRIFUGAL COMPRESSOR STAGE MACHINING

The designed impeller was machined using 5-axis BOSTOMATIC numerically controlled milling machine. Using an in-house developed programme, large numbers of co-ordinates were generated for the pressure and suction surface of the blade along hub and shroud from overall geometry. These co-ordinates were used as input data for 5-axis numerically controlled milling. A lisp programme uses these co-ordinates in the AutoCAD environment to generate the 3-D model of the impeller.

The vane diffuser co-ordinates compatible to 3-axis Deckle Machine was generated from an inbuilt software. These co-ordinates were directly feed to the numerically controlled machine through an IBM personnel computer for machining the blades. Provision is made to set the diffuser to different stagger angles to get the required area ratio and aspect ratio. The 3-D model of the compressor stage generated using Auto-CAD is shown in Figure-1.

4. STAGE PERFORMANCE EVALUATION

4.1 Cloctcr Test Facility

The schematic layout of the facility CLOCTER is presented in Figure-2. The compressor was rotated at a desired speed by an electomechanically coupled twin DC motor system. Thyristor control with feed back for the DC motors ensured maintenance of the speed to an accuracy of one percent. The compressor and the DC motor were connected together through a step-up gear box (1:6). An electronic torquemeter coupled in between the gearbox and the compressor was used to measure compressor speed and input power. A gate valve provided in the closed circuit was used to vary the massflow rate through the compressor. A heat exchanger in the closed circuit was used to ensure steady inlet flow conditions. An orifice plate in the inlet duct was used for mass flow measurement. Cooling water system in the circuit pumps cold water into the heat exchangers to cool the hot gas and hot lubricating oil. Lubricating system in the circuit is used to pump lubricating oil to the compressor bearing and gearbox to dissipate the generated heat. The test compressor is experimentally evaluated for its performance in closed circuit facility using Freon to get its wide operating range of both design and off-design points.

4.2 Data Acquisition System

The measurements were carried out using an on-line data acquisition system HP-3497A. The data acquisition system is connected to a dedicated IBM computer. Analogue signals from pressure transducers and thermocouples were sequentially scanned during a test sequence. Global parameters like total pressure, static pressure, total temperature at inlet and outlet of impeller and diffuser, speed and power input to the compressor were measured using this on-line data acquisition system using a computer Aided Testing Software developed in-house (Ref. 10).

Total pressure measurements were carried out at impeller inlet, impeller outlet, vane diffuser throat, vane diffuser outlet and volute outlet. The total pressure at impeller inlet and volute outlet were measured using a single sensor pitot probe. The total pressure at impeller outlet, diffuser throat and diffuser outlet were measured using special combination probes. The static pressures at different locations were measured by providing small static pressure holes on the casing. The static pressures at the diffuser throat plane and along the diffuser channel were also measured. The total temperatures at different measurement planes were measured using Chromel-Alumel thermocouples A calibrated orifice-meter was used to measure the massflow rate through the compressor. The compressor speed and power input were measured using an electronic torquemeter. During experiments the Freon concentration was continuously monitored. This concentration value is used in the analysis of experimental results. In addition to the above measurements the bearing oil temperatures and casing vibrations were also recorded.

4.3 Performance Characteristics

The measurements were carried out for various mass flow rates and compressor speeds. The flow being supersonic at impeller outlet, the total pressure measurements through a pitot probe need to be corroborated otherwise. Such a need arises due to the fluctuations in the flow angle from the impeller and the sensitivity of the probe. The total pressure at impeller outlet was alternatively estimated from two other aspects. One was by assuming a slip from empirical correlation's and the other was by evaluating Euler work done by the impeller from total temperature rise.

From the measured massflow rate, measured impeller outlet static pressure and slip condition based on Wisner (Ref. 11), impeller outlet velocity vector diagram parameters were calculated. With the process of transition from static to total assumed isentropic, the total pressure at impeller outlet (based on slip) was calculated. Another way of estimating this is to start with temperature measurement. Euler work was calculated from the difference of measured total temperature rise and the loss due to clearance and diskfriction. From Euler work the tangential component of absolute velocity was estimated. The meridional velocity calculated is calculated from the measured mass flow rate and outlet area. The resultant absolute flow velocity was calculated from the estimated tangential and meridional velocity. Static temperature at impeller outlet was estimated from energy equation. Thus from the measured static pressure and total temperature, the total pressure (based on Euler work) at impeller outlet was derived. The values of total pressure ratios estimated from Euler work and slip conditions agree reasonably well with the values of direct probe measurement upto impeller exit Mach numbers of around unity. At supersonic flow regimes though the measured total pressure using three-hole yaw probe at impeller outlet shows some deviations, total pressure ratio obtained from Euler work or from slip considerations is in agreement.

For a given speed the measured absolute flow angle increases with decrease in flow coefficient. The slope of the curve abruptly changes with change in Mach number from subsonic to supersonic. The value of impeller outlet flow angle at the designed flow rate is met in the present case as per design. To calculate the incidence at impeller inlet, tip relative flow angle was estimated from the inlet flow and rotational speed. The difference between the relative flow angle and the impeller blade angle would give the incidence at inlet. The measured absolute flow angle at impeller outlet was subtracted from the diffuser blade angle to get incidence to the diffuser. It is found that there is a single operating point where the incidence angles to both impeller and diffuser are zero. It is also found that the supersonic flow in the impeller pushes the diffuser incidence towards high positive incidence while the incidence to inducer remains small.

A vane diffuser is used following the impeller. In between the impeller and vane diffuser a small vaneless space was provided. The performances of stage were measured with various settings of the diffuser. Detailed stage measurements carried out are shown here only in one of the diffuser configuration. Measurements were carried out at diffuser throat and outlet using combination probe. The measured performance characteristics of the stage are shown in Figure-3 and Figure-4. A stage total to total pressure ratio of 5.5 with an efficiency of around 84 percent was achieved at design speed. The speed parameter values indicated in Figure-3 to 6 is rotational speed divided by factor, which is a product of 1000 and square root of normalised inlet total temperature to ambient temperature. The characteristics in Figure-3 indicate points very close to stall except for speed parameter of 1.65. The incidence angle to the inducer at the point of stall was found to be quite different for subsonic and supersonic flow. Hence the line joining the left end points of the characteristics shows a kink exactly at the point where the flow changes from subsonic to supersonic.

It is found that beyond a value of speed parameter 1.60, the absolute outlet velocity from the impeller is supersonic. There is a vaneless space of increasing area following the impeller outlet up to the diffuser leading edge. From the leading edge of the diffuser vanes up to the diffuser throat, there is semi vaneless portion of decreasing area. The supersonic flow coming out of the impeller can undergo a shock in the vaneless portion, turn subsonic and accelerate back to sonic at the diffuser throat. Alternatively an oblique shock anchored at the leading edge of the diffuser vanes may reduce the flow Mach number to be just sonic behind the shock, which is just near the throat. In this case the flow turning through the oblique shock for the designed supersonic Mach number should be commensurate with the curvature and blade angle setting provided at the leading edge of the diffuser vanes. When the interstage design matches this, pressure ratio exceeding 4 are achieved with a good efficiency. The pressure recovery coefficient in the vane diffuser is calculated by estimating the rise in total pressure and normalising with respect to impeller outlet dynamic head. Over a range of incidence the diffuser provides a good pressure recovery. At design point the diffuser has a pressure recovery of 0.6.(Figure-5).

From the measured total pressure at diffuser outlet and the total pressure at impeller outlet, the total pressure loss is calculated and normalised with respect to impeller outlet dynamic head. The percentage total pressure loss in the diffuser is plotted against incidence for various speeds in Figure-6. At optimum incidence the maximum total pressure loss is around 1.8%. With negative incidence to diffuser the loss in total pressure increases sharply.

Static pressure measurements through sidewall tappings were extensively made across the diffuser throat as well as the downstream of diverging passage. Coupled with the measured total pressure at diffuser exit and the total pressure at diffuser inlet (or the impeller outlet), the wall static pressure measurements were very helpful in visualising the flow pattern as it diffuses through the vaned passage. Both flow velocity and angle were studied and used to characterise this pattern. The flow pattern was found to be significantly different for supersonic absolute flows at impeller outlet. In this case the flow undergoes shock compression before reaching sonic conditions at the throat. At well-matched conditions of design, it is found that the flow angle in the semi vaneless portion matches with the direction perpendicular to the throat and that variations across the throat are minimal.

Distribution of pressure recovery along the diffuser meridional direction was also estimated and studied. As expected substantial amount of recovery was seen ahead of the diffuser throat itself. Particularly in the case of supersonic flow, shock compression contributes a lot to this pressure recovery. Even downstream of the diffuser throat, in the passage of uniform divergence, the variation of pressure recovery along the meridional plane was found to be far from being linear.

5 CONCLUSIONS

This compressor stage was experimentally evaluated for its performance under simulated conditions of supersonic flow using freon gas. The stage characteristics of this compressor and the diffuser were measured in detail. The stage characteristics show that a compressor stage total to total pressure ratio exceeding of 5.5 with an efficiency of 84% was achieved. At this operating point the impeller efficiency is more than 92%. The vane diffuser shows a good static pressure recovery of the order of 0.6 with total pressure loss less than 1.8% at design operating speed.

6. ACKNOWLEDGEMENTS

Referees comments, which were useful, are gratefully acknowledged. Initial phase of this work was supported by AR & DB through the initiative of Propulsion Panel and a grant-in-aid project. Continued support and encouragement was given by Director NAL for further R&D work in this area of supersonic flow centrifugal compressors. This work was carried out at CLOCTER facility of Propulsion Division, N.A.L. The technical assistance given by Mr.V.Rangarajan, Scientist, Propulsion Division and workshop Staff members throughout the course of this project is gratefully acknowledged. The technical assistance given by Mr.A.D.Manoharan, Mr.A.Ramachandran and Mr.D.Satyarajan is acknowledged.

7. REFERENCES

1) S.Ramamurthy, S.Sankaranarayanan, K.Murugesan , "Aerodynamic Design of High Pressure Ratio Centrifugal Compressors". NAL-TM-PR-8602, 1986.

2) S.Ramamurthy, K.Murugesan "An Analytic Approach to D... ign of Centrifugal Impeller Geometry", Submitted to Jl. of Turbomachines for Publication, of ASME, Awaiting Acceptance.

3) M.R.Vanco, " Portion Program for Calculating Velocities in Meridional Plane of Turbomachine. I-Centrifugal Compressor". NASA TN-D-6701, 1972.

4) D.H.Wilkinson, "Calculation of Blade-to-Blade Flow in a Turbomachinery by streamline Curvature Method", ARC R&M 3704, 1972.

5) K.Murugesan, "Analysis of Francis Turbine Runner Blade Channel Flow", NAL-TM-PR-304, 1-78,1978.

6) S.Sankaranarayanan, S.Ramamurthy and K Murugesan, D.Ramachar, (Establishment of Closed Circuit Centrifugal Compressor Test Rig (CLOCTER) ", NAL-TM-PR-8604, 1986.

7) S.Ramamurthy, K.Murugesan and G.Geetha, "Development of Instrument for Measuring the Centrifugal of Gases in Two Component Mixtures", NAL-TM-PR-UNO-103 (102) / 1, November, 1981.

8) S.Ramamurthy , K.Murugesan and S.Sankaranarayanan, "Computer Aided Design of Centrifugal Impellers ", NAL-TM-PR-8603,1986.

9) S.Ramamurthy and K.Murugesan , "Vane Tweaking For Centrifugal Impellers ", AR&DB Seminar on Propulsion, August, 1990.

10) S.Sankaranarayanan, S.Ramamurthy and K.Murugesan , "Performance Evaluation of Centrifugal Compressors Using Computer Aided Testing (CAT) in CLOCTER Facility " , NAL-TM-PR-8505, 1985.

11) F.J.Wiesner, "A Review of Slip Factors for Centrifugal Impellers", Jl. Engg. Power, Trans, ASME, Vol. 87, 1965, P. 181.

12) S.Ramamurthy, S.Sankaranarayanan, K.Murugesan and D.Ramachar, "Performance Evaluation of Centrifugal Compressors", NAL-TM-PR-8605, 1986.

13) S.Ramamurthy, K.Murugesan and S.Sankaranarayanan, "Theoretical Model for the Performance Evaluation of Centrifugal Compressors". NAL-TM-PR-8705, 1987.

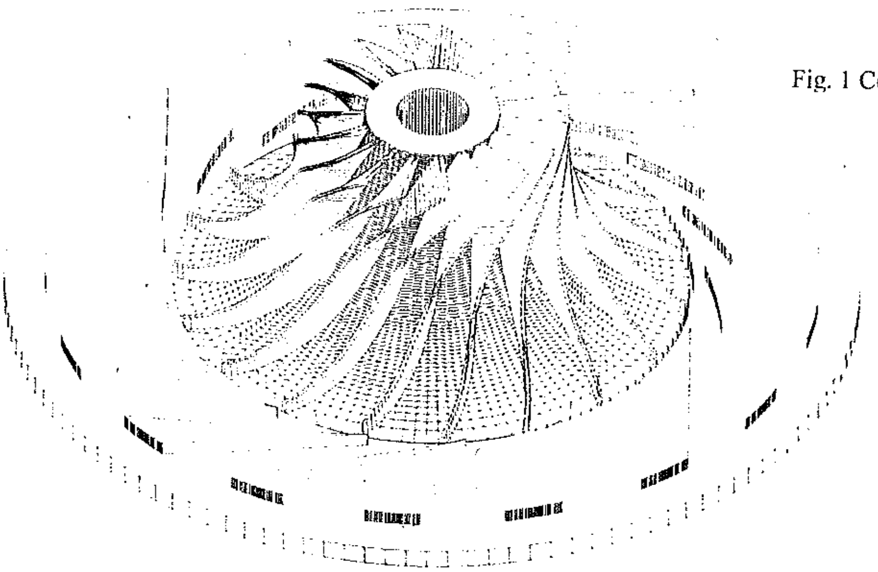
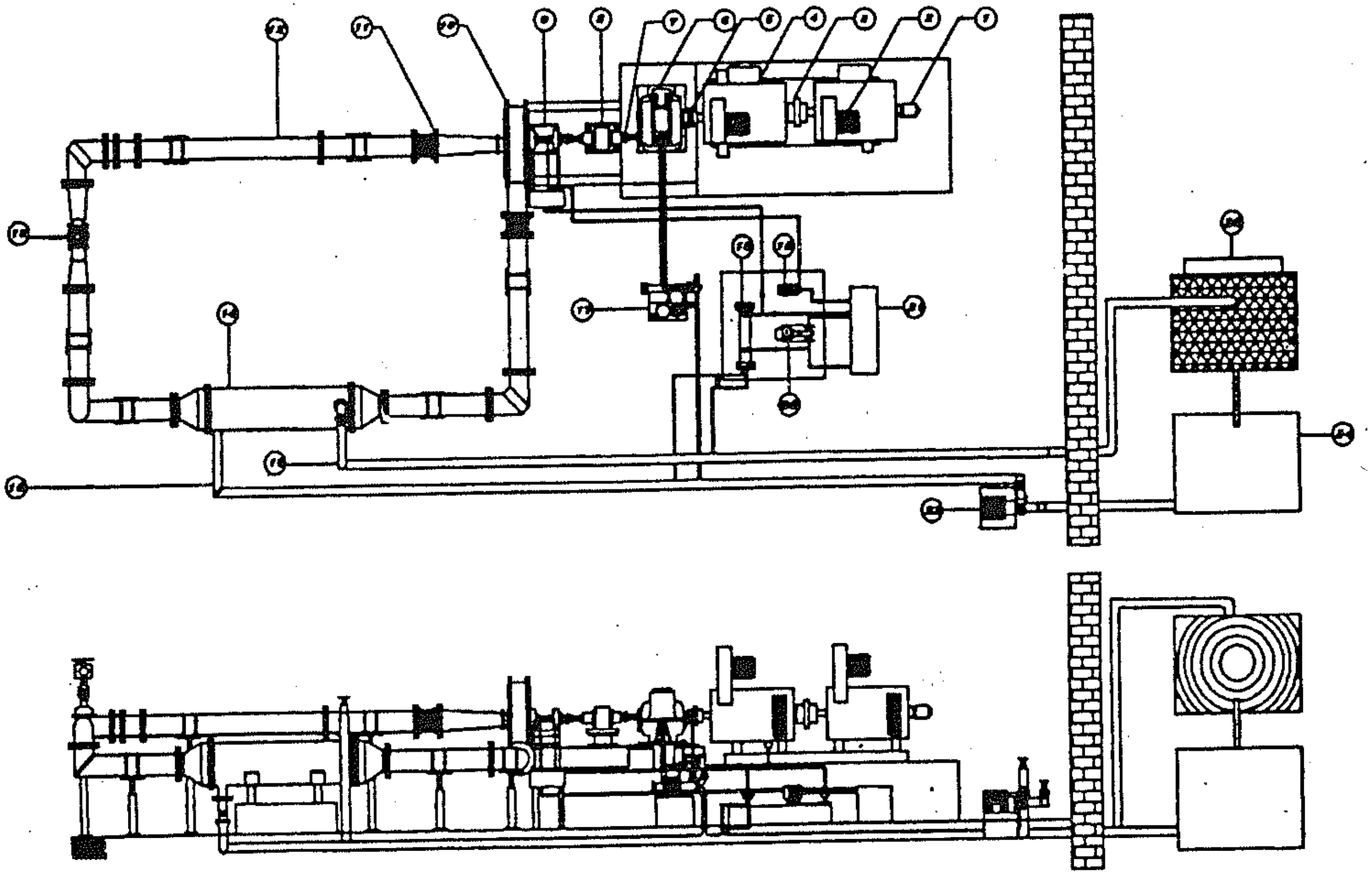


Fig. 1 Centrifugal Compressor Stage



1. TACHO GENERATOR
2. COOLING FAN
3. RIGID COUPLING
4. D.C.DRIVE MOTORS
5. GEARED COUPLING
6. GEAR BOX
7. FLEXIBLE DIPHARGM
8. TORQUE METER
9. BEAR'G SHAFT ASSEMBLY
10. VOLUTE
11. EXPANSION JOINT
12. CLOSED CIRCUIT LOOP
13. GATE VALVE
14. HEAT EXCHANGER
15. COLD WATER INLET
16. HOT WATER OUTLET
17. GEAR BOX LUB. SYSTEM
18. LUB. OIL SUCTION PUMP
19. LUB. HEAT EXCHANGER
20. LUB. OIL DELIVERY PUMP
21. LUB. OIL TANK
22. COOLING WATER PUMP
23. COOLING TOWER
24. COOLING WATER TANK

Fig. 2 CLOSED CIRCUIT CENTRIFUGAL COMPRESSOR TEST RIG

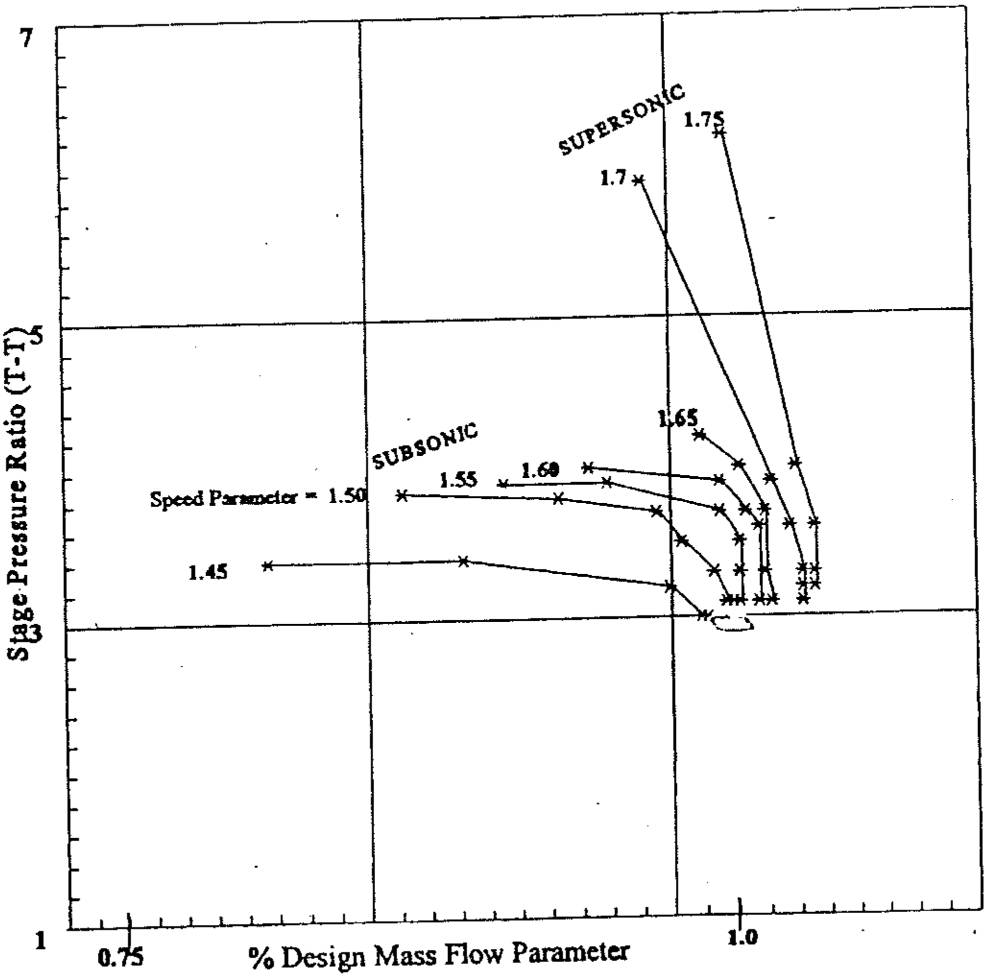


Fig. 3 Centrifugal Compressor Stage Pressure Ratio

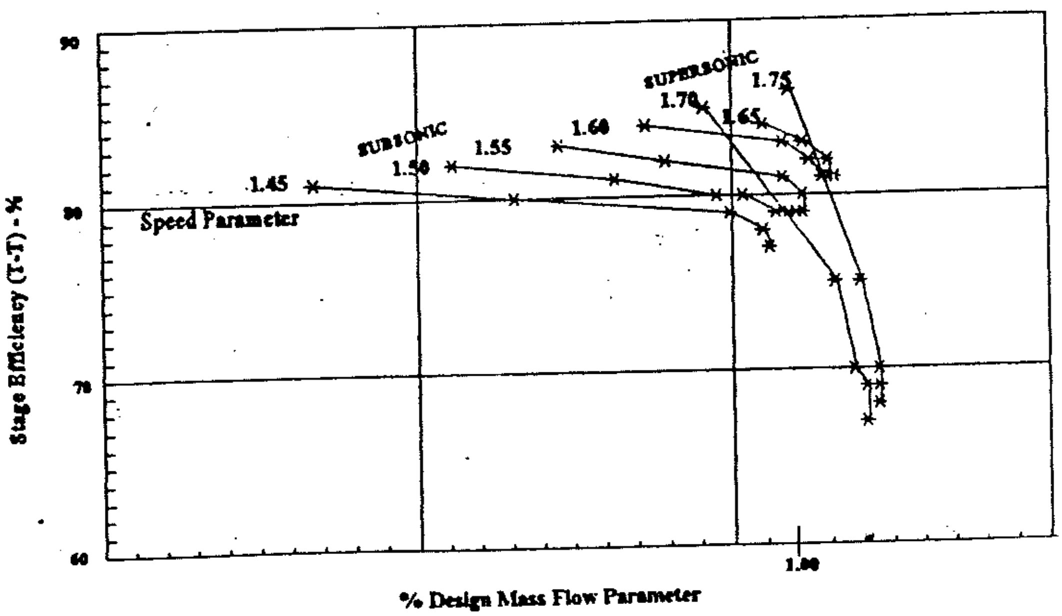


Fig. 4 Stage Efficiency Characteristics

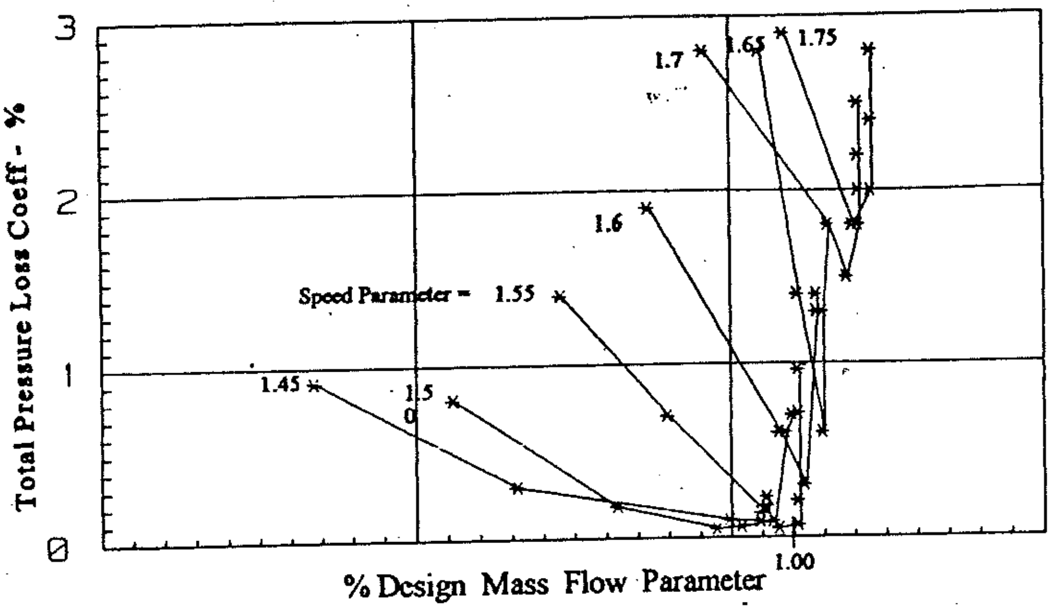


Fig.5 Diffuser Total Pressure Loss Coefficient

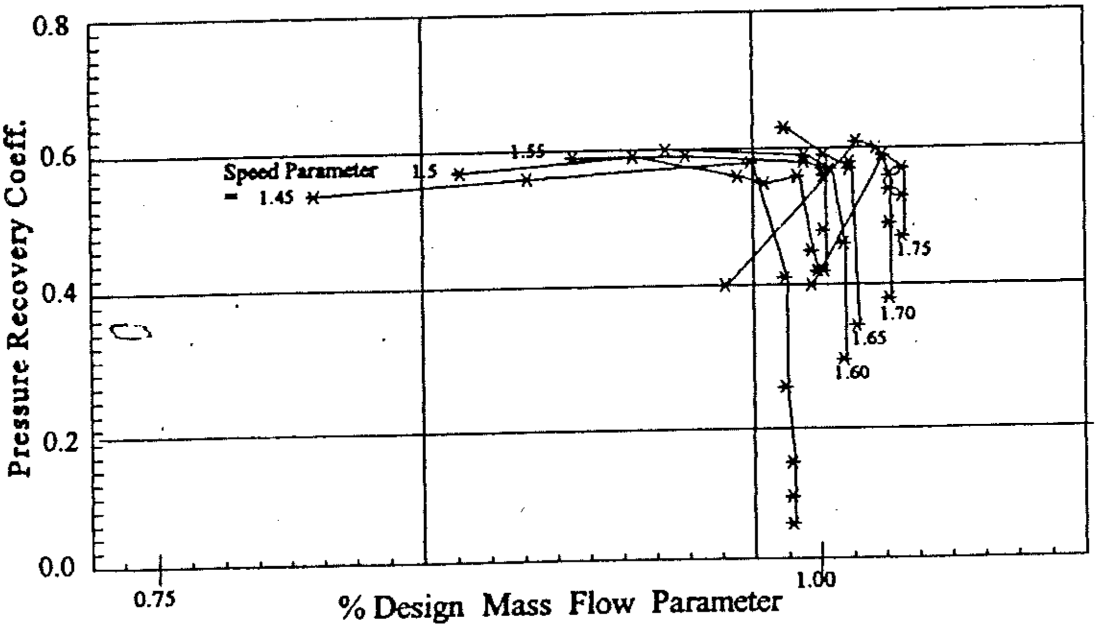


Fig. 6 Diffuser Pressure Recovery Coefficient