PERFORMANCE TESTING AND R & D WORK ON CENTRIFUGAL COMPRESSOR IN CLOSED CIRCUIT TEST RIGS

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1. ABSTRACT :

The utility and application of the closed circuit test rigs in carrying out R and D work on Centrifugal compressors are discussed in detail. The performance improvements of a medium specific speed compressor stage obtained by testing in a closed circuit test facility with air as the working medium is given as an example of the utility of such rigs. This paper also explains the sailent features of the versatile closed circuit test facility coming up at the Propulsion Division, N.A.L., Bangalore. This facility is compared with similar facilities elsewhere existing at various industries and Universities.

2. INTRODUCTION:

There has been a substantial progress and increase in application of centrifugal compressors for the past ten years, serving the chemical, petroleum and natural gas industries. Centrifugal compressors used in these industries come in diverse specifications with regard to number of stages, capacity, pressure levels and gas handled. The ultimate aim for any compressor manufacturer is to deliver efficient, reliable compressor with reasonably good surge margin. This can be achieved only when aerodynamic performance are experimentally evaluated. In most of the compressor industries in order to ensure and improve the performance of multi-stage machines, single stage configurations which are representative of a very wide family of stages are tested under simulated condition. This kind of testing and related R and D work on compressors are increasingly carried out in Closed Circuit Test Rigs to gain certain advantages.

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3a. CLOSED CIRCUIT TEST RIG :

Figure (1) shows the schematic arrangement of a closed circuit test rig, used for testing single stage compressors. The pipe forms a closed loop connecting the discharge to inlet. The essential elements are the heat exchanger, valve, the orifice plate and a bypass line. Unlike the open system the closed-loop provide great flexibility in the manipulation of inlet pressure and inlet temperature which in turn control the inlet density. Hence power requirement of given machine can be decreased by reducing the density level of the medium. The effect of Reynolds number and Mach number on compressor performance can be studied. By using mixture of gases the value of γ, i.e. the ratio of specific heats of any gas can be simulated in the closed circuit test facility with non-dimensional similarity parameters unchanged. From Figure-(2), it may be noted that upto the tip Mach number of 0.8, which is the upper limit for most of the industrial centrifugal compressors, there is no marked variation of fluid outlet angle for different gases (R -1). So when heavy gas like Freon-12 is used as working medium, the velocity of sound in the medium is comparatively low (approximately half the velocity of sound in air) and hence simulation is possible under lower shaft speed. This to some extent solves the high speed bearing and stress problems of the test rotors, which can be made now quickly in available material. Another advantage is the volume of the test rig is very small and any selected gas can be filled depending upon the designers choice in small quantity. Also the flow inside the loop has less fluctuations and precise aerodynamic measurement is possible. Such features improve the accuracy of measurement and extend the scope of useful testing for beyond that of open system, which can be used only with atmospheric air. The noise level of closed loop system is much lower than open loop system.

Number in paranthesis with prefix R designate references at the end of the paper.

To get wider operating range of the compressor at low pressure ratios, it becomes necessary to have lower system losses. For this purpose a bypass line shown in Figure-(1) is used. A part of the mass flow will be bypassed the heatexchanger, where the pressure losses were thought to be quite comparable to the total system pressure losses.

3. b PERFORMANCE TESTING AND R & D WORK ON COMPRESSORS :

The ultimate aim of industrial stage testing is evaluating the aerodynamic performance of the compressor at design and off design flow rates to present the results in a simple and selectable form. Later on these can be used for designing multi-stage machines. If the design point efficiency is low or the off design conditions are well below the expected level, improvement methods have to be applied out of experience by either altering the design philosophy or by improving component efficiency through changes in parameters like impeller inlet blade shape, vane shape, diffuser entry,........etc. A series of design and experimental exercise will help in finding optimum solutions and correlations thus avoiding time consuming repetitive experimental work in the later stages.

To test each stage of a multi-stage industrial centrifugal compressor in a closed-circuit test facility, the stage should have a configuration shown in Figure-(3), consisting of inlet bend, impeller, semivaneless space, diffuser and return channels. In Figure-(4) a typical ψ Vs ϕ graph plotted after experiment is given as an illustration (R - 2). The vane diffuser stage has same radius ratio as vaneless diffuser. The vaned diffuser stage shows four percent higher efficiency than the vaneless diffuser stage with small reduction in operating range. To stress the R and D need on compressor, the performance of a low specific speed, stage is given in Figure-(5). It can be seen from this figure that between the vaned and vaneless diffuser combinations, the vaned diffuser avoided the onset of surge at lower flow coefficients. But the stage efficiency with both combinations, was found only around 50 percent. Another setting of the vaned diffuser (22°) was tried with the compressor; which improved the efficiency albiet marginally. Subsequently, a new design method for the stage was undertaken (R -3) in order to improve the efficiency of this low specific speed stage. This is a typical example of R and D work using such test rigs.

3. C CLOSED-CIRCUIT TEST RIG OF N.A.L. :

A versatile closed circuit test rig (CLOCTER) where Industrial Turbocharger, Gas Turbine and Refrigeration compressors can be tested is being built at the Propulsion Division of N.A.L., Bangalore. Schematically the rig is represented in Figure-(6) and the specifications are given in Table-I. The experimental apparatus consists of a Thyristor control D.C. drive, Step up gear box, Torque meter, Test Compressor and Closed Circuit Piping. A heat exchanger and a throttle valve are included in the rig to get the desired inlet conditions. The compressor shaft is fitted with high speed 'Sealol' shaft seal to prevent any leakage of working medium through shaft clearances. A novel technique was developed to monitor the concentration of medium inside the loop continuously during experiments. The concentration was indirectly estimated by measuring the accoustic velocity in the medium.

Performance of the compressor will be obtained by measuring total and static pressures at Inlet and outlet of the compressor using total pressure probes and wall tappings. Total temperature being measured using chromel-Alumel thermocouple. The velocity distribution at inlet and outlet of the impeller will be measured using three hole yaw probe. The mass flow rates through the circuit will be estimated using orifice meter by measuring the differential pressure using pressure transducer. A computer (HP 3054 A) will be used for fast data Acquisition and reduction.

Closed circuit test facilities available at various Industries and Universities elsewhere are given in Table-II. It is observed from this table, the power used for the drive motor in the industrial test facilities were quite large, where the compressor manufacturers intended to test the prototype compressors for guarantee tests or to get mere overall performance. Whereas Research Laboratories and Universities concentrating more on R & D work on model compressors chose lower power due to its restiriction.

Compared to the test facilities available at Universities and Laboratories, the test facility at N.A.L. has slightly higher power for the drive motor. This was chosen to have greater flexibility in carrying out R and D work on various types of compressors ranging from industrial to small prototype gas turbine compressors. A similar closed circuit test facility is available at B.H.E.L., Hyderabad, where the industrial centrifugal compressors are tested at design speed for guarantee tests, which is quite different from the R & D work carried out at N.A.L.

4. CONCLUSIONS :

A closed circuit test rig for testing Industrial, Refrigeration, Turbocharger and Gas Turbine compressors will be built at Propulsion Division, N.A.L., Bangalore. This facility helps to test the centrifugal impellers at reduced power using heavy gas like Freon-12. This also enables us to study the effects of Reynolds number and Mach number on compressor performance. Initially compressors will be tested both in Air and Freon-12 mediums. The performance deviations in both cases due to Reynolds number and Mach number were suitably correlated with the experimental data obtained. In latter stages High Pressure ratio air compressors will be tested in Freon-12 medium to reduce the power required and the performance thus obtained will be suitably corrected using the earlier correlations to get the performance in Air. Thus eliminating the impeller stress and high speed bearing problems to some extent.

5. ACKNOWLEDGEMENT :

The authors thank UNDP/UNESCO for financial assistance given to build the closed circuit test rig at N.A.L., Bangalore and deputing two of N.A.L. Scientists abroad to get training in closed circuit compressor research. The authors also thank Dr. P.A Paranjpe, Head, Propulsion Division, N.A.L. (Project Director, UNDP) for his valuable suggestions and in coordinating the project to commission the closed circuit test rig at N.A.L.

6. REFERENCES :

1. Industrial Centrifugal Compressors, V.K.I. Lecture Series- 95, Vol. II, 1977.
2. A Short report on the design of a radial compressor stage working in low specific speed. Report submitted to Fluid dynamic Laboratory, Sulzer Brothers Ltd., Switzerland, 1978.
3. Aerodynamic performance of a centrifugal compressor with Vaned Diffusers by Y.Yoshinaga; I.Gyobu; H.Mishina; F.Koseki, and H.Nishida Journal of Fluids Engineering. Transactions of the ASME. DECEMBER 1980, Vol. 102, PP :486-493.

Table - I

SPECIFICATION OF CLOCTER : AT N. A. L.

Type :	Horizontal closed circuit
AIMS	To test Industrial, Gas Turbine, Turbocharger, Refrigeration C.F. compressors
Working Medium	Air, Freon-12, Freon-12 + Argon.
Motor Power Output (KW)	375
Compressor Max. Speed (RPM)	18,000
Impeller Max. Tip Dimameter (MM)	525
Flow coefficient $= \dfrac{\dot{m}/\rho_{01}}{\frac{\pi}{4} D_2^2 U_2}$	0.1145 (Typical For Industrial Comp
Head coeff. $= \dfrac{\Delta H_i}{\frac{U_2^2}{2}}$	1.09 (Typical For industrial comp.)

TABLE - II

A BRIEF SURVEY OF CENTREFUGAL COMPRESSOR RESEARCH OF VARIOUS INDUSTRIES AND UNIVERSITIES

Place	Sulzer 1	Sulzer 2	Nuovopignone	Aachen University	Onera 1	Onera 2	I. H. I.	Hitachi 1	Hitachi 2	Kyushu University	E. P. C.	Creare	N.A.L.
	Winter thur Switzerland	Winter thur Switzerland	Florence Italy	West Germany	Paris France	Paris France	Japan	Japan	Japan	Japan	Japan	U.S.A.	India
Test rig Type	Closed circuit Horizontal	Closed circuit Vertical	Closed circuit Horizontal	Closed circuit Horizontal Axial flow Compressors	Closedcircuit Horizontal	Closedcircuit Vertical	Closed circuit Vertical	Closed circuit Horizontal	Closed circuit Vertical	Closedcircuit Vertical	Closedcircuit Horizontal	Closed circuit Vertical	Closed Circuit Horizontal
Working Medium	Air, Freon-12	Air, Freon12, CO_2	Freon-114 Freon+Air Mixture	Freon-12 + Air	Freon-114	Freon-114	Freon-12 Air, CO_2	Freon-12 And Air	Air, CO_2 N_2, He	Freon-12	Freon-12, CO_2, Freon-22 N_2	Freon + Argon	Freon-12 Air
Motor Power out put (kw)	125	800	1500	1430	1500	40	200	300	550	220	700 (A.C)	145	375
Compressor Max. speed (R.P.M.)	18,000	14,000	16,000	15,000	9,500	14,000	17,000	25,500	20,000	20,000	6,000	32,000	18,000
Impeller Max. Tip Diameter (mm)	315	500	800	200	450	600	280	250	Variable	280	350	120	525
Flow Coefficient $\phi = \frac{m_1}{D_2^2 V_2 \rho_{01}}$	0·01-0·028	0·10	$\phi_2 = \frac{C_{n2}}{V_2} = 0{\cdot}11$	0·74	–	–	0·021	0·1–0·2	–	0·018	0·297	$\dot{m} = 0{\cdot}773$ kg/sec.	$\dot{m}$=6kg/s

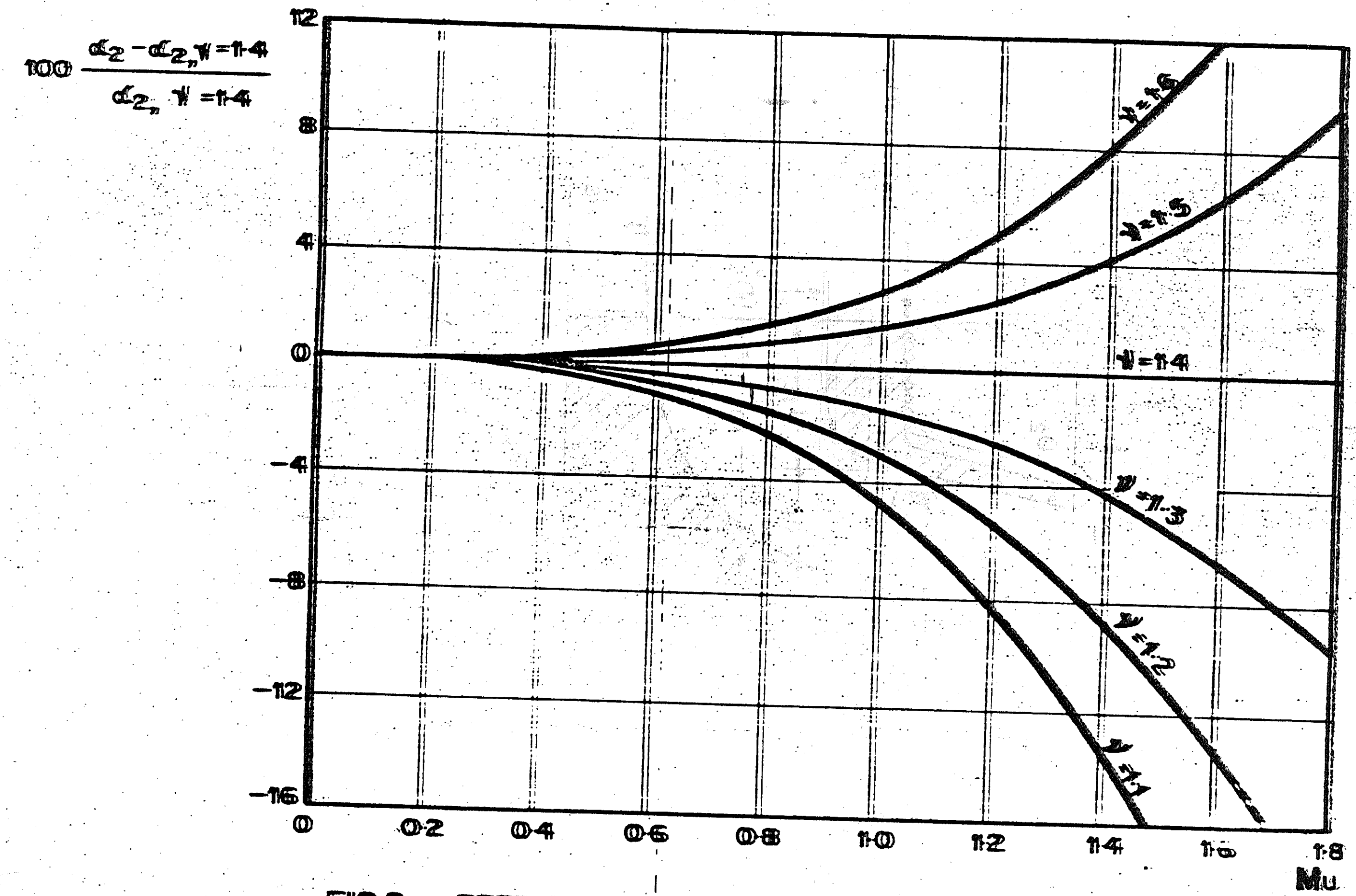
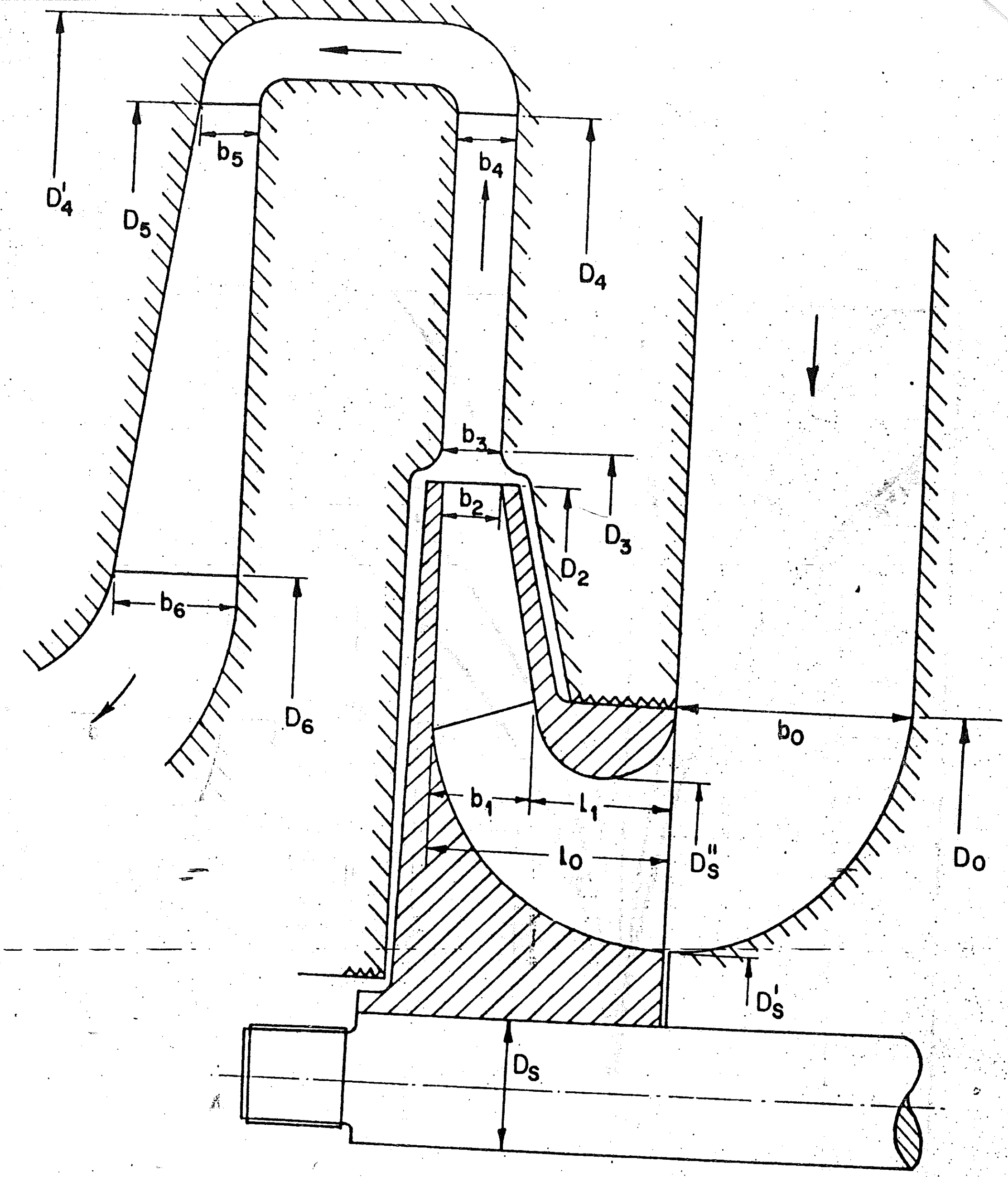


FIG.2 EFFECT OF ν ON DEFFUSER ENTRY ANGLE α_2



DIA.	MM	WIDTH	MM	OTHER DETAILS
D_0		b_0		D_s =
D_1		b_1		D'_s =
D_2		b_2		D'_s =
D_3		b_3		l_0 =
D_4, D'_4		b_4		l_1 =
D_5		b_5		
D_6		b_6		

FIG.3 STAGE CONFIGURATION OF CENTRIFUGAL COMPRESSOR

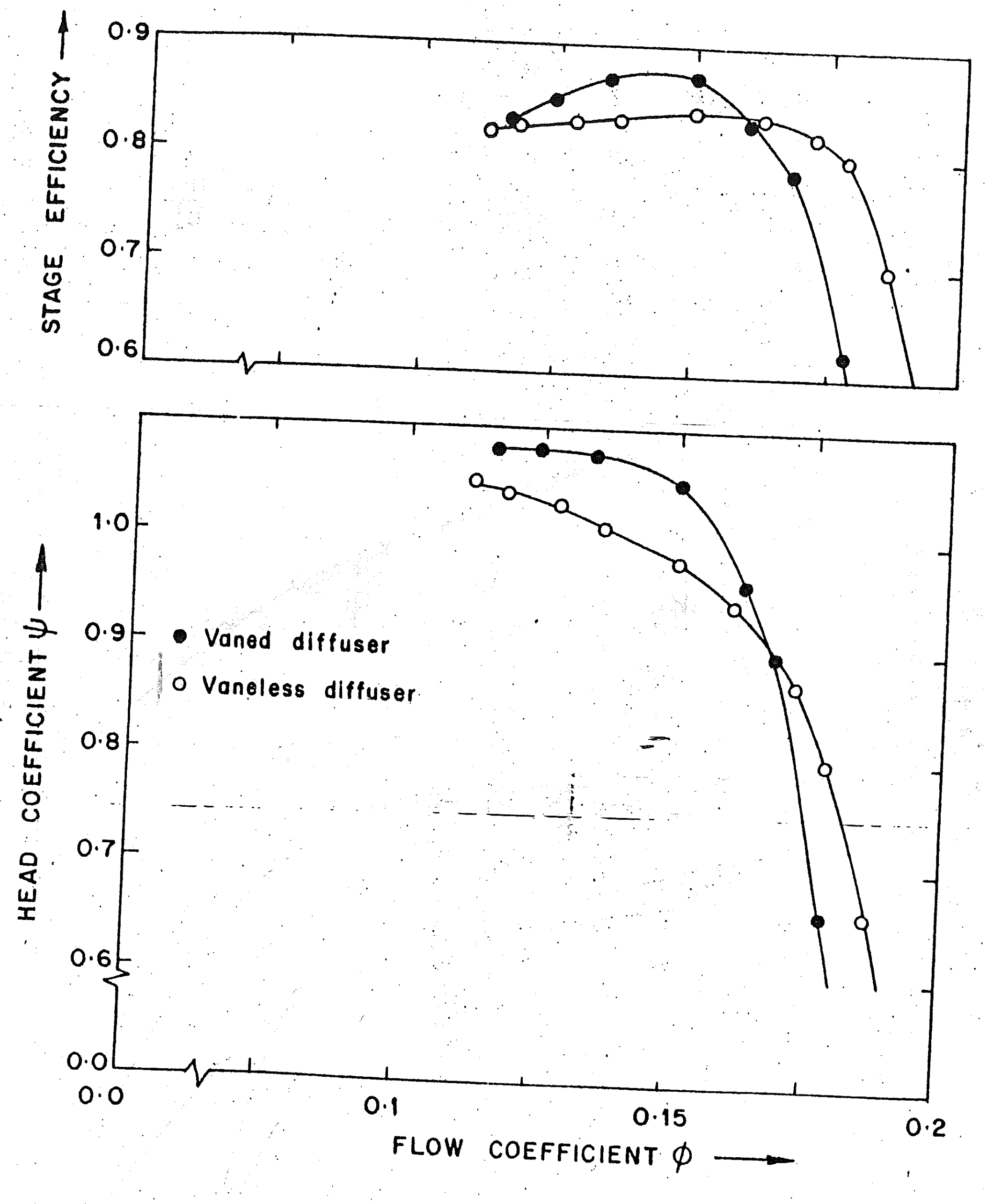


FIG. 4 PERFORMANCE CHARACTERISTICS OF VANELESS AND VANED DIFFUSER

WORKING MEDIUM — AIR $M_{u2} = 0.8$

IMPELLER — $D_2 = 315$ mm $\frac{b_2}{D_2} = 0.01068$

$b_2 = 3.35$ mm $e = 3.0°$

$\beta_2 = 57°$

DIFFUSER α_2 = 18° ——— △ ———

22° ——— ⊙ ———

VANELESS ——⊡ ———

$M_{u2} = \frac{u_2}{\sqrt{\gamma R T_{01}}}$ $\phi_1^* = \frac{\dot{m}_1}{\rho_1^* D_2^2 \; U_2}$ $\mu_y^* = \frac{H_{is}}{u_2^2}$

η_y^* = STAGE EFFICIENCY $\mu_o = \frac{\mu_y^*}{\eta_y^*}$

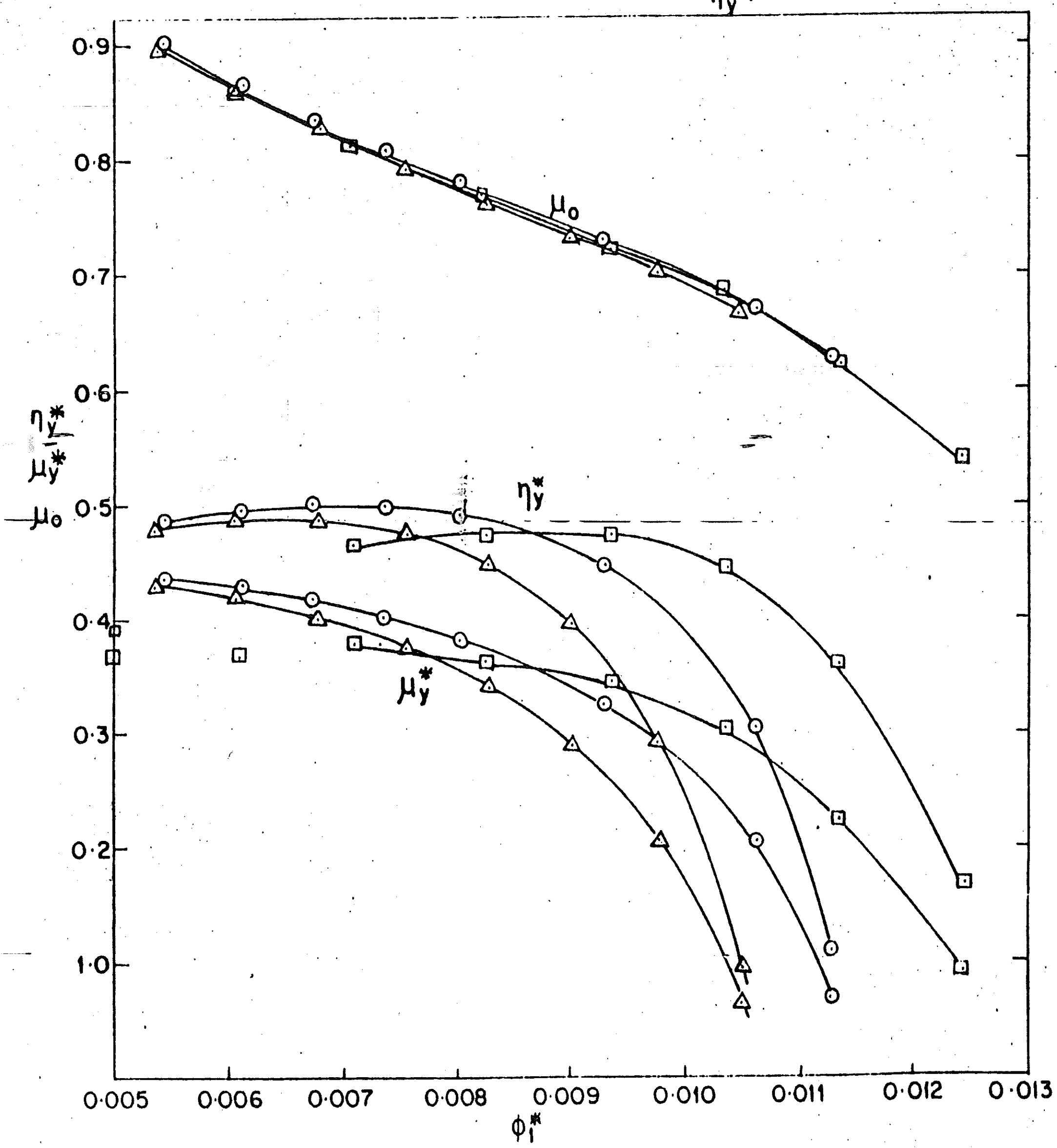
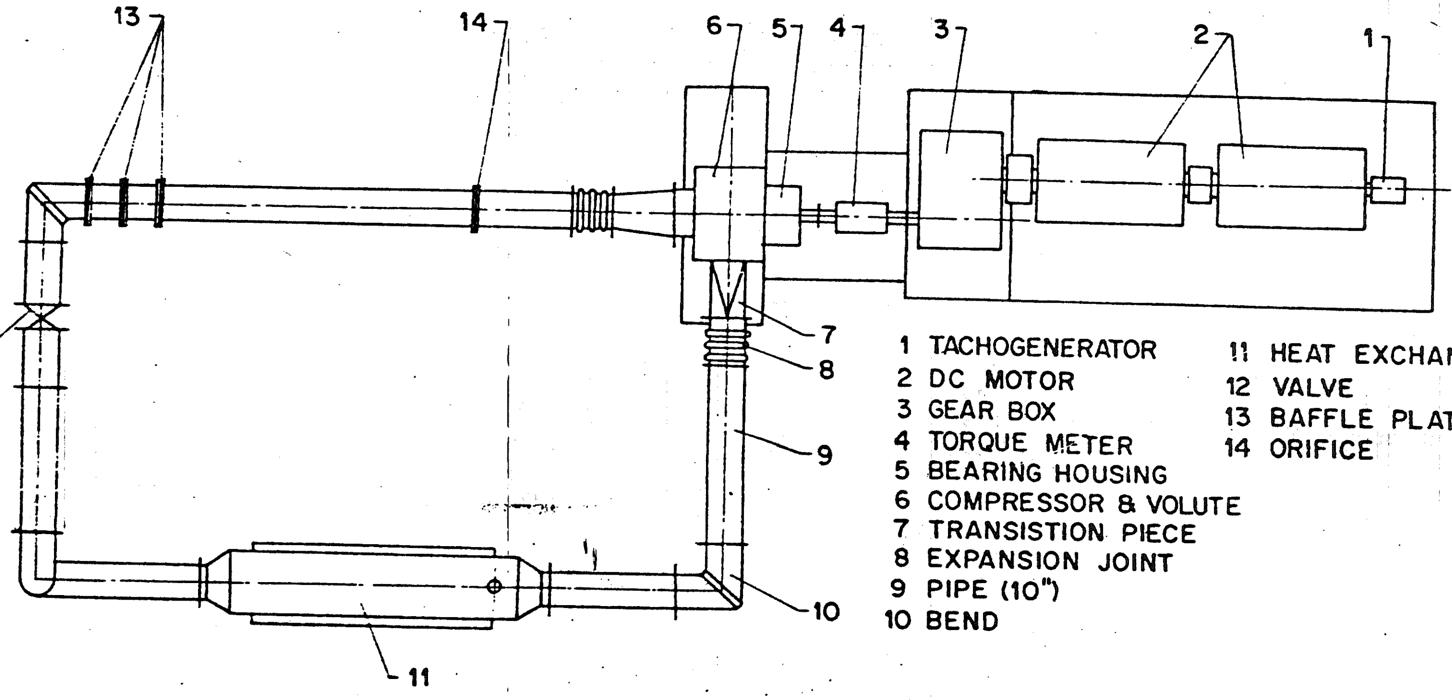


FIG. 5 CHARACTERISTICS OF CENTRIFUGAL COMPRESSOR



1 TACHOGENERATOR
2 DC MOTOR
3 GEAR BOX
4 TORQUE METER
5 BEARING HOUSING
6 COMPRESSOR & VOLUTE
7 TRANSISTION PIECE
8 EXPANSION JOINT
9 PIPE (10")
10 BEND
11 HEAT EXCHANGER
12 VALVE
13 BAFFLE PLATES
14 ORIFICE

FIG. 6 CLOSED CIRCUIT TEST FACILITY