DIFFICULTIES ENCOUNTERED FOR PRESSURE AND TEMPERATURE MEASUREMENTS IN HIGH PERFORMANCE COMPRESSORS

Jacques PAULON

Office National d'Etudes et de Recherches Aérospatiales
29 avenue de la Division Leclerc
92320 CHATILLON - France

DIFFICULTIES ENCOUNTERED FOR PRESSURE AND TEMPERATURE MEASUREMENTS IN HIGH PERFORMANCE COMPRESSORS

Jacques PAULON

Office National d'Etudes et de Recherches Aêrospatiales
29 avenue de la Division Leclerc
92320 CHATILLON - France

Abstract

Flow traverses made in different sections of axial compressors are very useful in order to obtain a detailed understanding of the flow. Generally the measurements are redundant and it is possible to control the validity of some of them. In the case of two advanced supersonic axial flow compressors, we bring to light the difficulty of measuring static pressure and temperature in some sections. Nevertheless it is possible to validate a data reduction method.

1 - INTRODUCTION

Most measurements made during compressor research or development testing aim at overall performance determination. The usual ones are :

- mass flow rate, by mean of nozzles or diaphragms,
- pressure ratio, by means of total pressure rates upstream and downstream of the compressor,
- total temperature rise, by means of temperature rates also upstream and downstream.

All these measurements give the performances of the compressor and are very important for the user. But, for the designer, such measurements are not sufficient, since they give no means for improving the performance of the compressor. For that purpose, a detailed knowledge of the flow field is necessary.

Apparently it seems quite easy to obtain such information, by using pressure probes, temperature probes, yaw probes that have been tested in wind tunnel. Wall pressure measurements with miniaturized pressure transducers can also be made. It becomes apparent that the experimental data are redundant and the flow fields that can be derived from these data are quite different, according to the kind of measurement results that was used.

Two test compressors were used to clarify this problem, both of them quite advanced and typical, thus, of the most difficult cases that can be encountered.

2 - DESCRIPTION AND INSTRUMENTATION OF THE TEST COMPRESSORS

The two advanced transonic compressors have a highly converging channel. The compressor I is driven in air at reduced inlet pressure and atmospheric outlet (Fig. 1a) ; the compressor II is driven in freon in a loop circuit (Fig. 1b).

The compressor I has been tested in two configurations :

- industrial type, with a gap between rotor and stator too small to allow radial traverses (Fig. 2, version I),

- research type, with a large axial gap between rotor and stator (Fig. 2, version II) that allows radial traverses at the rotor exit.

The figures 2 and 3 show the flow paths and the instrumentation station of the two compressors.

Besides, the tables I and II summarize the measurements made in each compressor and state whether the measurements seem reliable or not (in the two cases, the flow tangential velocity at the exit of the stator is very small).

3 - DATA REDUCTION DOWNSTREAM OF A ROTOR

Equations used for data reduction are well known :

- generalized radial equilibrium equation,
- mass flow conservation equation,
- Mach number equation,
- total temperature rise equation.

But neither the meridional slope ε of the streamline and the radius of curvature R_m of the meridional streamline were measured and guesses have to be made concerning their radial variations.

In addition, the measured value of the direction with axis α'_2 of the flow seems to be unreliable, since it is expected that the stator may interfere with the directional probe. Therefore assumptions have to be made in radial variations of these parameters and it is aimed to obtain radial distributions of static pressure that matches the inner and outer wall pressure measurements and a mass flow that corresponds to the one measured upstream.

(i) Assumptions on meridional streamline slope

An obvious linear radial distributions has been tested :

$$\varepsilon = \varepsilon_i \frac{r_o - r}{r_o - r_i}$$

(r_o = outer radius and r_i = inner radius) with the geometrical value ε_i = 5° and with also ε_i = 10°.

Solving the radial equilibrium equation with a linear law for the curvature radius,

$$R_m = 0{,}35 \frac{r - r_i}{r_o - r_i} + 0{,}15$$

we found that the pressure distribution along the radius is practically not affected by the choice of ε_i (Fig. 4).

(ii) Assumptions on meridional curvature

The radial distribution of curvature seems to be more critical, since it is aimed to check both the wall static pressure and the measured mass flow rate. As indicated on figure 5a, very different laws of curvature have been used ; they are represented by relations such as :

$$R_m = \left(R_{m_o} - R_{m_i}\right) \left(\frac{r - r_i}{r_o - r_i}\right)^n + R_{m_i}$$

The corresponding pressure distributions deduced from the indicated equations with boundary conditions and using the radial distributions of total pressure P_{t_2} downstream of the rotor and total temperature T_{t_3} downstream of the stator (we will see later why we don't use the total temperature T_{t_2} downstream of the rotor) are presented on the figure 5b.

It is noted that pressure distribution that satisfy the wall pressure conditions as well as the mass flow conservation condition is obtained with at least two radial distributions of curvature (laws 3 and 5).

It is frustrating to notice that similar results are obtained with different distributions of curvature. It must then be concluded that data reduction of experimental tests can give reliable radial distribution of static pressure and, consequently, radial distributions of Mach number velocity and direction of the flow (with axis), but no insight to the actual stage of the streamline.

(iii) Comparison between measured and computed parameters

It might be interesting to compare the computed values of some parameters to their measured values.

Figure 6 shows this comparison for the static pressure p_2 at the outlet of the rotor, and there is a large discrepancy between measured and computed values. Therefore, the probe has been calibrated in wind tunnel and it is a very good probe for transonic measurements. An explanation must be sought in the interference between the probe and the stator blades and in the unsteadyness of the flow coming from the rotor.

Figure 7 shows the comparison between measured and calculated flow angle α_2 downstream of the rotor. The same conclusion applies.

As indicated previously, the measurements of static pressure and flow angle downstream of a rotor are unreliable.

Similar comparisons were made on compressor II and figure 8 also shows some discrepancy between measured and calculated flow angle.

Another interesting comparison can be made on this compressor II, in which total temperature T_{t2} was measured downstream of the rotor and at the exit of the stator T_{t3} . Figure 9 compares the values of total temperature rise measured upstream and downstream of the stator ; this figure was plotted after correcting the radial positions of the downstream total temperature by taking into account mass flow conservation along individual stream surfaces.

Again one is striked by the difference in shape of the two distributions and the difference of level near the tip. One explanation may be that the total temperature downstream of the rotor is strongly time dependent and the averaging process in the probe is not the mass flow weighted averaging that is used in the stop outlet temperature averaging.

This explains why, as indicated previously, the measurement of total temperature downstream of a rotor is unreliable and why one used the total temperature T_{t3} , measured at the outlet of the stator, for the data reduction downstream of a rotor.

4 - CONCLUSION

Analysis of experimental data obtained on advanced transonic compressors shows the difficulty of describing the flow field inside a compressor, as long as conventional pressure and temperature probes only will be used.

More advanced techniques, such as laser velocimetry, are necessary in order to validate the results obtained with conventional systems. Meanwhile this it seems that, using total pressure radial distribution downstream of a rotor and total temperature radial distribution downstream of a stator, it is possible to obtain reasonably correct radial distributions of static pressure and flow angle between a rotor and a stator.

These computations need the knowledge of the mass flow and wall static pressure and it seems that these two parameters can be obtained with much accuracy.

TABLE I

Measurements in compressor I

		Station (See fig 2)	Small axial	Large axial	Reliable	Unreliable
Total pressure (traverses)		Inlet to the compressor B/O	P_{t_0}	P_{t_0}	+	
		Downstream of rotor F	—	P_{t_2}	+	
		Downstream of stator D	P_{t_3}	—	+	
Static pressure	Traverses	Downstream of rotor F	—	P_2		+
		Downstream of stator D	P_3	—	+	
	Wall pressure	Inlet of compressor A/O	P_0	P_0	+	
		Inlet of rotor 1 (*)	P_1	P_1	+	
		Outlet of rotor 2 (*)	P_2	P_2 (**)	+	
		Outlet of stator 3 (*)	P_3 (***)	P_3 (**)	+	
Total Temperature	Traverses	Outlet of rotor	—	T_{t_2}		+
		Outlet of stator	T_{t_3}	—	+	
	Rakes	Upstream of compressor	T_{t_0}	T_{t_0}	+	
		Far downstream	$T_{t_3^*}$	$T_{t_3^*}$	+	
Flow angle	Traverses	Downstream of rotor	—	α_2		+
		Downstream of stator	α_3	—	+	

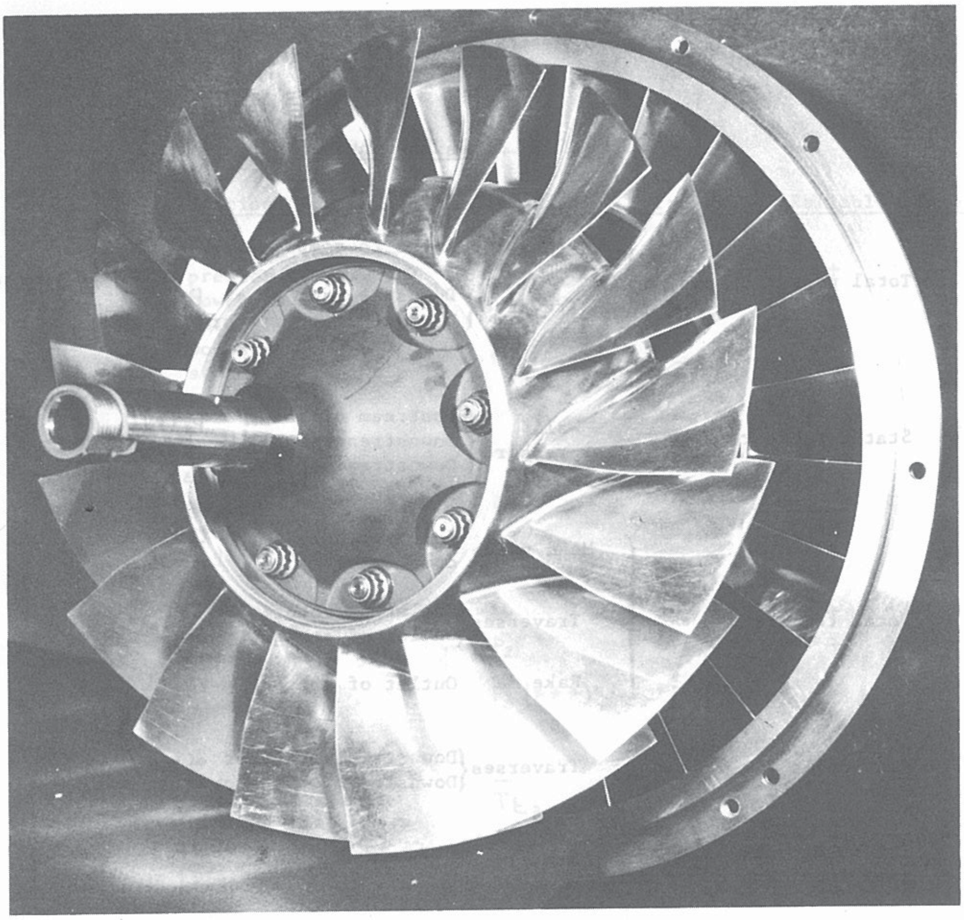
(*) Measurements made on inner and outer wall
(**) Measurements made at station (2) and also at station (2') in front of stator
(***) Measurements made also at station (3')

TABLE II

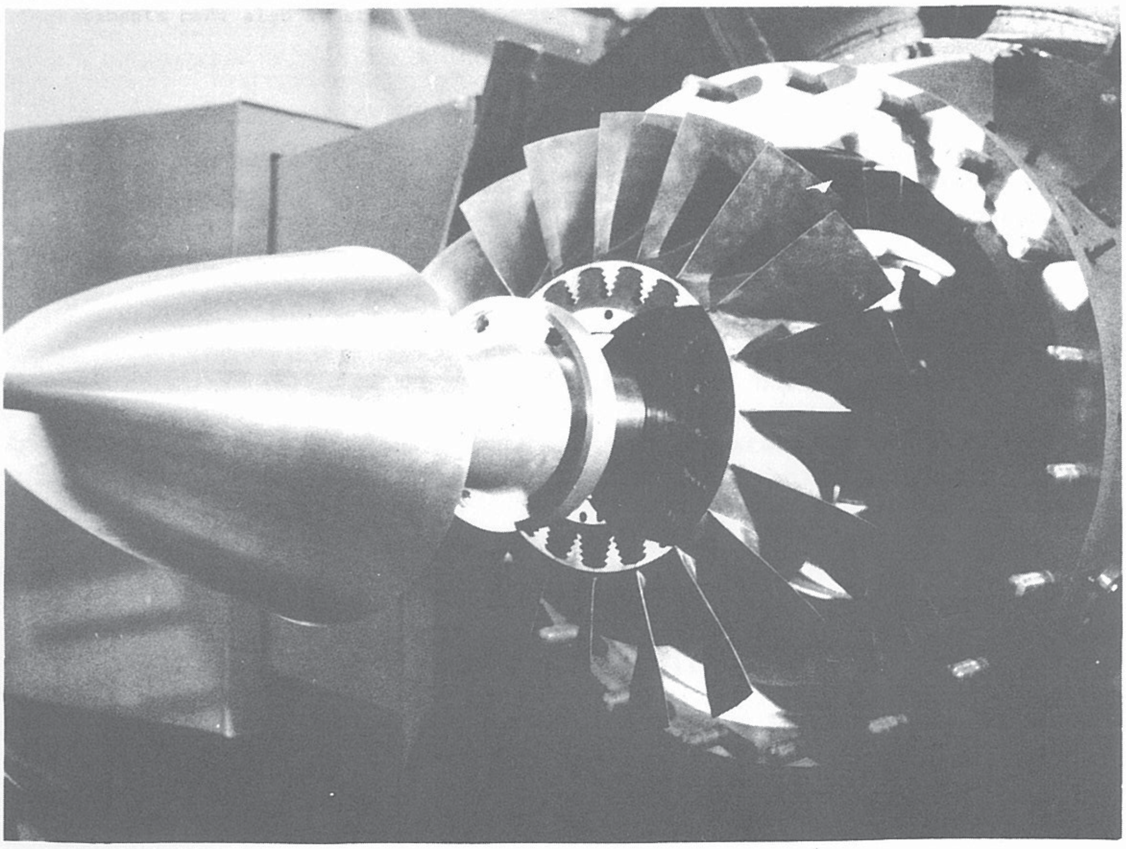
Measurements in compressor II

		Station (see fig 3)		Reliable	Unreliable
Total pressure	Mean value	Inlet to compressor	Pt_o	+	
	Traverse	Downstream of rotor	Pt_2	+	
		Downstream of stator	Pt_3	+	
Static pressure	Wall pressure	Upstream of rotor	p_1 (*)	+	
		Downstream of rotor	p_2 (**)	+	
		Downstream of stator	p_3 (**)	+	
Total temperature	Mean value	Inlet to the compressor	Tt_o	+	
	Traverses	Outlet or rotor	Tt_2		+
		Outlet of stator	Tt_3	+	
	Rake	Outlet of stator	Tt_3'	+	
Flow angle	Traverses	Downstream of rotor	α_2		+
		Downstream of stator	α_3	+	

(*) Outer wall only
(**) Inner and outer wall



1a) Air compressor (nr I)



1b) Freon compressor (nr II)

Figure 1 : Test compressors

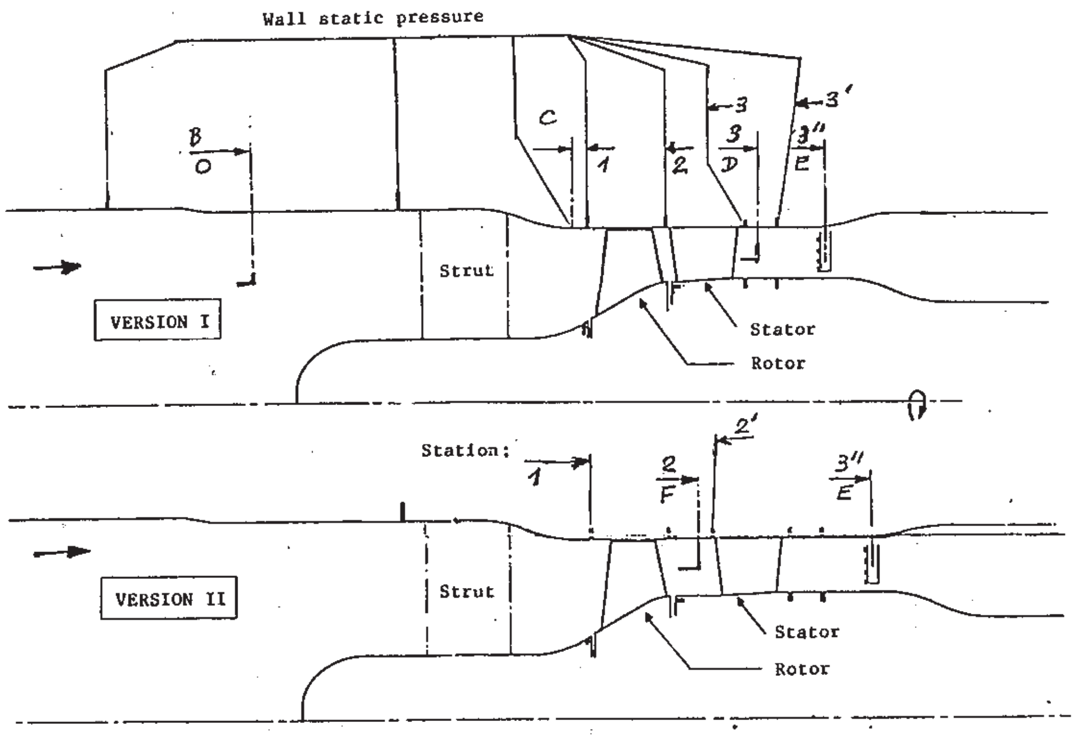


Figure 2 : Flow path and instrumentation stations
(Air compressor)

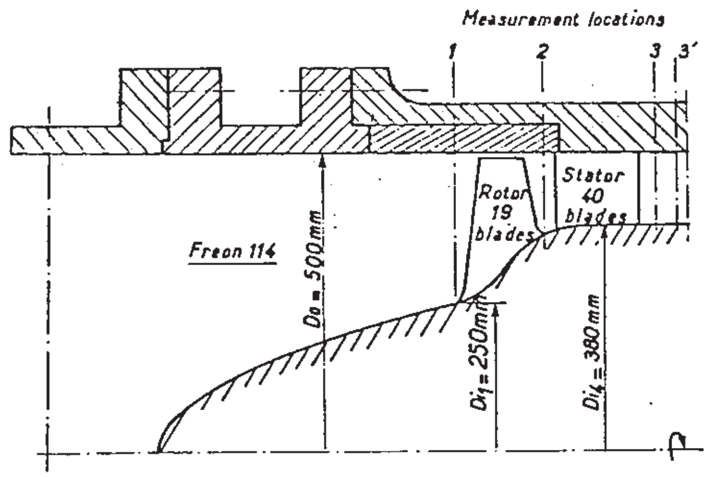


Figure 3 : Flow path and instrumentation stations
(Freon compressor)

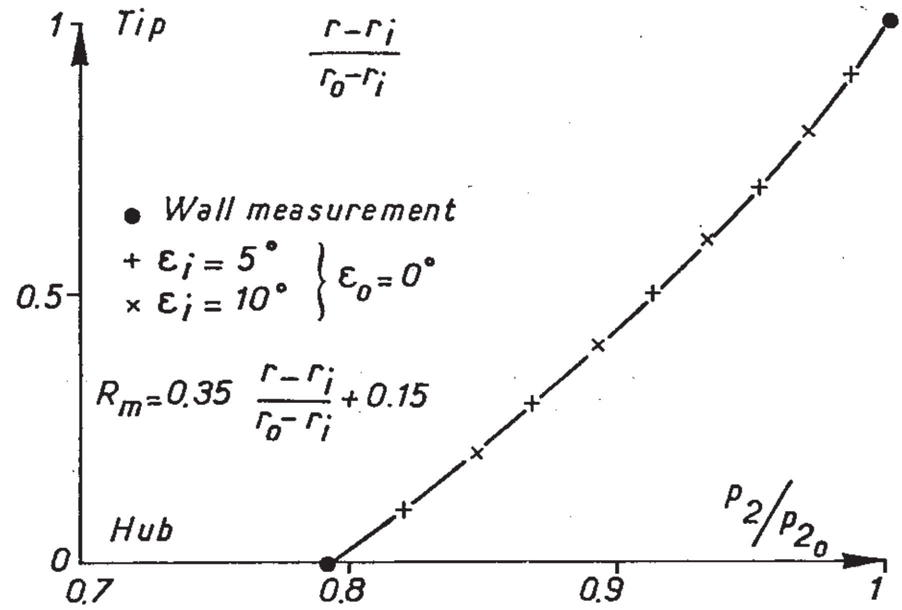


Figure 4 : Influence of streamlines slope on static pressure radial distribution (Air compressor)

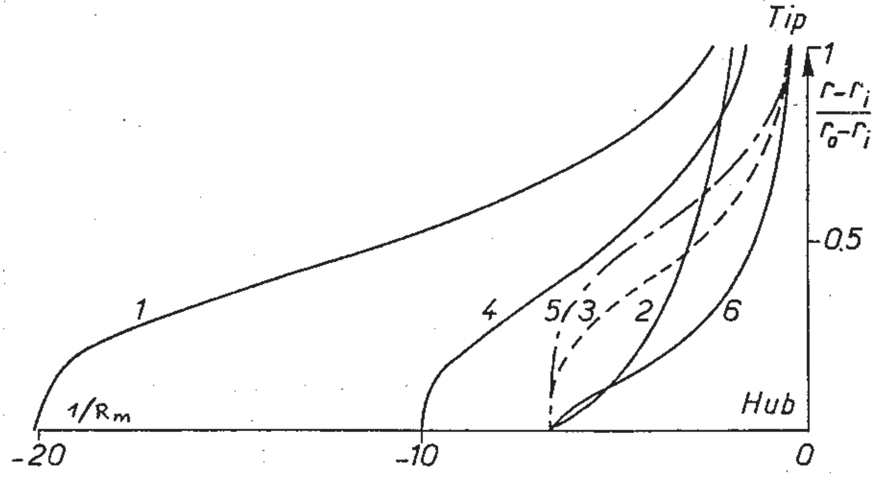


Figure 5a : Streamlines meridional curvature laws used for measurements interpretation

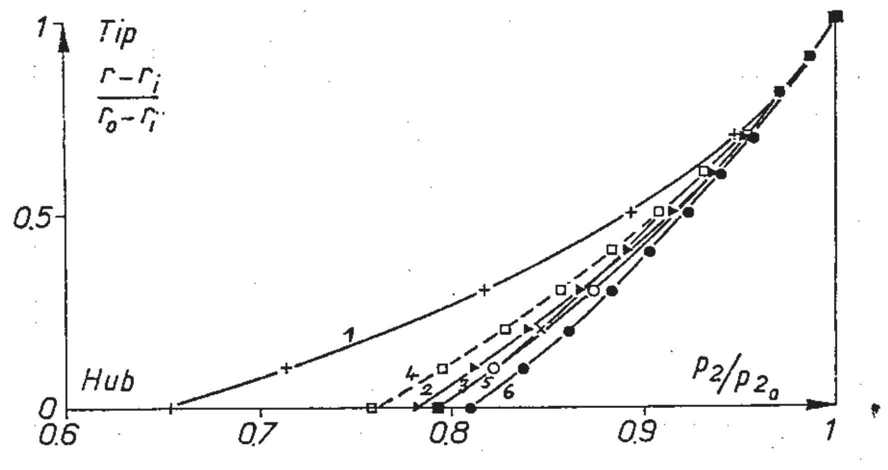


Figure 5b : Influence of streamlines meridional curvature on static pressure radial distribution at rotor exit (Air compressor)

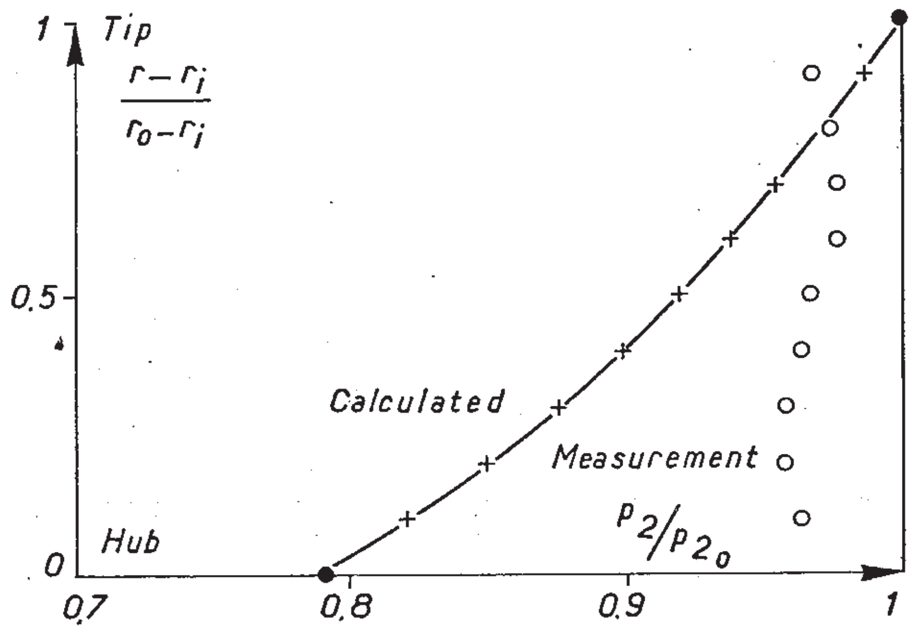


Figure 6 : Comparison of measured and calculated static pressure at rotor exit

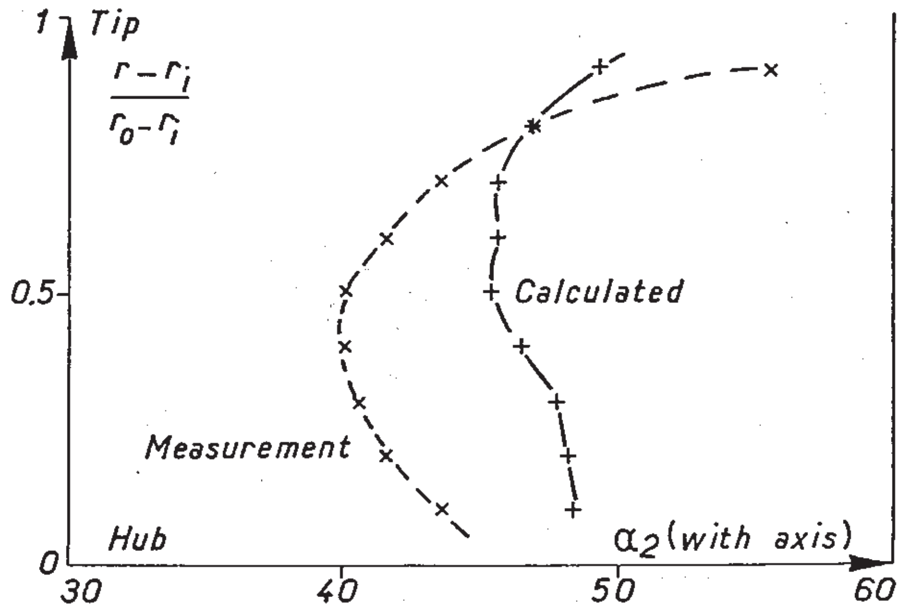


Figure 7 : Comparison of measured and calculated absolute angles at rotor exit (Air compressor)

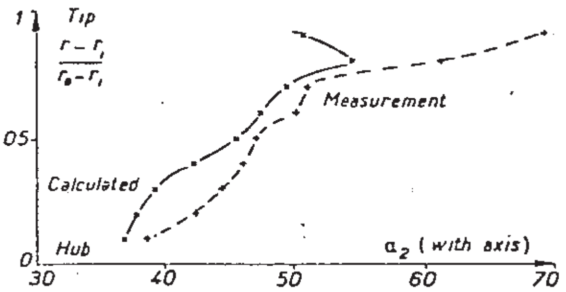


Figure 8 : Comparison of measured and calculated absolute angles at rotor exit (Freon compressor)

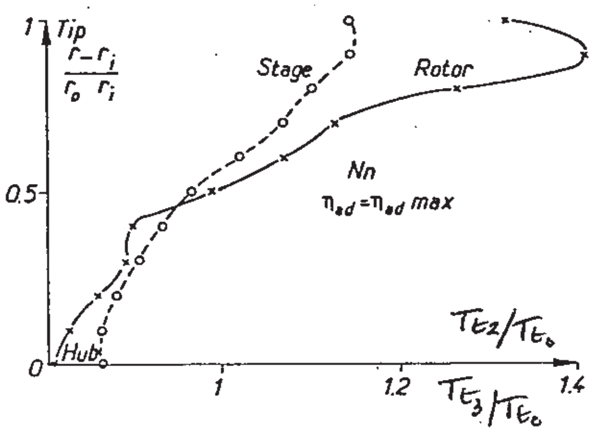


Figure 9 : Comparison of total temperature increase at rotor and stage exits (Freon compressor)