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New Features in the Design of Axial-Flow Compressors with Tandem Blades

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A multitude of examinations and studies of compressor blades with 100 percent reaction have shown that tandem blades can advantageously be used in stationary industrial axial-flow compressors, designed for applications in the chemical, iron and steel industries. Because there are no limitations for axial-flow compressors with tandem blades, this type of compressor can operate up to maximum pressure ratios of 10 for suction volume flow of about 600,000 m^3/h. In particular, the handling of light gases, such as helium, leads also to large sizes. Based on measurements taken on a four-stage experimental compressor, criteria are developed for the design of axial-flow compressors with tandem blades. The basic concept of a multi-stage industrial compressor provides for the combined arrangement of compressor stages of single and tandem-cascade design. This permits an optimum performance to be achieved at a considerable reduction of the constructional expenditure.

INTRODUCTION

Since the development of slotted aerofoils in the early twenties, the distribution of the total flow deflection over several partial aerofoils has been basically known. It was obvious to apply slotted aerofoil configurations of this type also to blade profiles in a cascade arrangement in the form of so-called tandem cascades.

The slotted-aerofoil or tandem-cascade effect is based on the fact that when a relatively large amount of flow deflection is divided between two single cascades, a new boundary layer is formed at the profiles of the downstream cascade, which largely prevents or at least delays premature flow separation. Therefore, relatively high pressure coefficients can be realized at good efficiencies by the selection of suitable tandem cascades for the blading of axial-flow compressors. Thus the specific energy transformation in axial compressors can be considerably increased.

Clarifying the flow conditions in tandem blade cascades has been the subject of many studies and papers. The most important references are quoted in [1].

The attempt to adapt slotted aerofoils to the blade cascades of a turbomachine was made at a rather late date [2 to 7]. Marcinowski [2] was the first to investigate a deceleration tandem cascade in a single-stage blower. He was able to lay down various data about the optimum tandem-cascade arrangement for the cascades of sheet blades which he examined. Sheets [3] achieved a maximum efficiency of 0.96 at a pressure coefficient of 0.95 with tandem blades in a single-stage axial blower, by utilizing the laminar effect on the blades of the first cascade. Furthermore, Linnemann [4] and Ihlenfeld [5] studied single-stage blowers with tandem cascades for the stator and rotor blades at comparatively low tip speeds. Linnemann's investigations showed that in comparison to single cascades, tandem cascades feature a better efficiency and a more favourable location of the flow separation line. Ihlenfeld [5] ascertained a favourable effect by tandem cascades on the secondary and clearance losses so that the compressor efficiency can be improved also in this regard.

Towards the end of the sixties experimental studies were performed on several single-stage axial-flow compressors with tandem blades under the sponsorship of NASA [6, 7]. Keenan and Bartok [6] used a slotted stator blade consisting of multiple circular arc profiles with inlet flow Mach numbers of 1.1. A design pressure ratio of 1.45 was achieved.

In contrast, Burger and Keenan [7] performed measurements on a single-stage axial-flow compressor without inlet guide vanes. The stage consisted of a tandem blade rotor with downstream stator blade row. The stage efficiency achieved at a stage pressure ratio of 1.93 was 83.1 %.

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All studies and papers published so far were concerned with tandem blades in single-stage axial-flow compressors with a reaction of the blading of around 50 %. The fact that despite the available results tandem cascades have not so far been more widely used in multi-stage axial compressors is certainly due to the reaction of 50 %. At this particular degree of reaction, the tandem-cascade configuration achieves only a relatively small improvement of the aerodynamic characteristics. As the fixation of the blade roots requires some further expenditure, it has so far not been economically rewarding to use tandem cascades for a degree of reaction of 50 %.

Therefore, the Institute for Turbomachinery of the University of Hanover, West-Germany, has since 1970 systematically studied the aplication of tandem cascades in multi-stage axial-flow compressors with 100 % reaction blading. The main criteria of the examined blades were the following:

- design according to free vortex conditions
- homogeneous repetitive blading
- inlet flow Mach number M_1 of 0.85 in the design point
- economic blade root fixation
- reduced overall length
- reduced number of blades

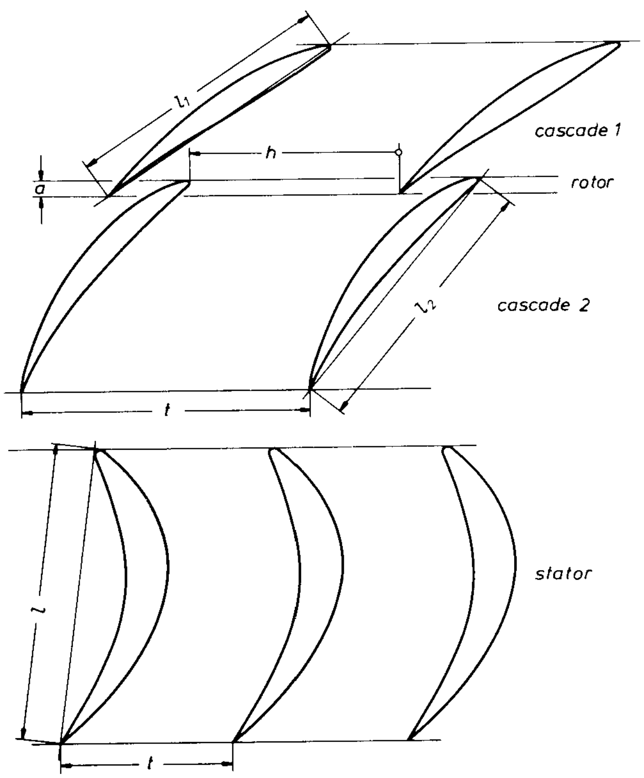


Fig. 1 Tandem-cascade stage (mean section)

a axial displacement
h displacement in peripheral direction
l length of chord
t pitch

The most important results of the individual steps of development for a multi-stage axial-flow compressor with tandem blading will be discussed in the following. In conclusion, a basic concept will be presented for an industrial compressor.

APPLICATION OF TANDEM BLADES IN AXIAL-FLOW COMPRESSORS

The aim is to reduce the overall length and the number of blades as against axial-flow compressors of conventional single-cascade design. At a comparatively high reaction of, say, between 80 and 100 %, this can be achieved by greatly decelerating the flow in the tandem blade rotor, which consists of two decelerating single cascades in series, viz. cascade No. 1 and cascade No. 2 (Fig. 1).

In the first cascade the relative velocity w_1 is reduced to w_{21}, and in the second cascade finally to the value of w_2 (Fig. 2, top). Downstream the tandem blade rotor the flow is then again deflected from the absolute velocity c_2 to c_1 by impulse-type cascades of a higher aerodynamic loading, or with the aid of single-cascade stator blades of low deceleration (Fig. 2, top).

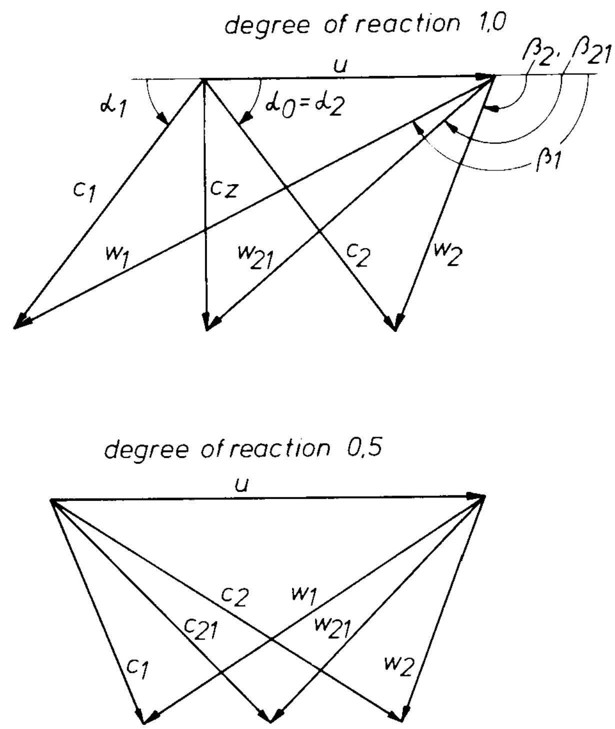


Fig. 2 Velocity diagrams of axial-flow compressors for degree of reaction of 50 and 100 %

c_z axial velocity
u circumferential velocity
w_1, c_2 inlet velocity
w_2, c_1 outlet velocity
w_{21} outlet velocity cascade 1
inlet velocity cascade 2
α_2, β_1 inlet angle
α_1, β_2 outlet angle

If, with a reaction of 100 %, each compressor stage consists of a tandem blade rotor and an associate deflecting stator blade row (Fig. 1), one stator blade row is saved in comparison with a configuration having two single-cascade stages at the same delivery head (Fig. 3). As every other stator blade row is eliminated, the bladed rotor length can be reduced by 30 % and the number of blades by 40 %, at the same aerodynamic loading of the impeller profiles. Considering that there are more stator blades than rotor blades, the saving of stator blades has a greater percentage effect on the total blading. The reduction of the constructional expenditure is caused by the increase of the aerodynamic loading of the stator blades which at a high degree of reaction have a comparatively low deflection of flow.

A comparison of the velocity diagrams of compressors with 50 and 100 % reaction for the same head shows that in a 50 % reaction machine the stator blade rows must also consists of tandem cascades because of the symmetry of the velocity diagrams (Fig. 2, bottom). In this way, the pressure coefficients can be increased by between 5 and 10 % [4, 5], but due to the aerodynamic loading capacity of the decelerating stator blades it is not possible to save one blade row in every two stages. It is thus obvious that a major reduction of the constructional expenditure by the use of tandem blades can only be achieved at a comparatively high degree of reaction.

INVESTIGATIONS OF TANDEM-CASCADE BLADING

Optimization of tandem rotor blades

To ensure a favourable performance for an axial-flow compressor with 100 % reaction, the tandem blades should be of an almost

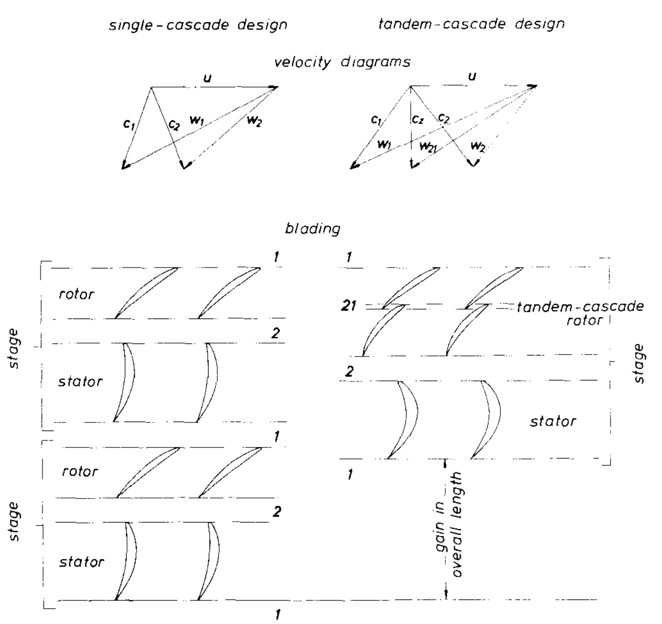


Fig. 3 Comparison of single-cascade and tandem-cascade design

fully optimum cascade configuration over their height in all design sections. The optimum cascade configuration is understood to be the arrangement defined by a particular relative cascade spacing a/l_1 and a particular relative cascade displacement h/t (Fig. 1), with minimum profile losses.

In [8] it was demonstrated for a four-stage experimental compressor how a tandem rotor blade can be optimized over the blade height with a suitable calculation method.

The compressor stages consist of a tandem blade rotor and a single-cascade stator blade row each (Fig. 4). The axial-flow compressor is rated for the handling of air, with the following data:

pressure ratio	π_{tot}	= 2.5
mass flow rate	$\dot{m}$	= 6.6 kg/s
speed	n	= 11,000 rpm
number of stages	z	= 4
power at coupling	P	= 672 kW

The tandem blades consist of NACA-profiles series 65.

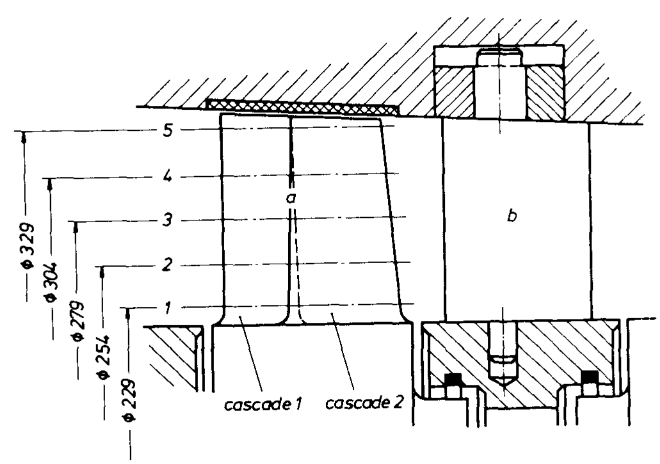


Fig. 4 Compressor stage

a tandem cascade rotor
b stator

The tandem blade to be optimized was designed in five sections (Tables 1 and 2). The definition of the lift coefficient c_a^* is the same which is used for the above mentioned NACA-profiles.

When optimizing a tandem-cascade rotor blade, one starts from the lowermost section of the blade, i.e. section No. 1. For this section the optimum cascade configuration is determined by means of the applied calculation method. The boundary condition assumed here is that the profile centres of gravity of the rotor blade sections of both cascades must be located on radius vectors in order to prevent additional bending stresses in the blade as a result of centrifugal forces. The radius vectors extend through the centreline of the rotor (Fig. 5).

Table 1 Profile and cascade data tandem cascade rotor blade cascade 1

Sections	Dimen-sion	1	2	3	4	5
r	mm	114.5	127.0	139.5	152.0	164.5
t	mm	34.25	38.0	41.8	45.4	49.25
ℓ	mm	35.0	36.0	37.0	38.0	39.0
t/ℓ	-	0.978	1.056	1.130	1.195	1.263
β_1	deg	155.5	155.0	155.5	156.7	158.6
β_{21}	deg	141.0	143.8	146.5	148.6	150.5
$\Delta\beta_1$	deg	14.5	11.2	9.0	8.1	8.1
c_a^*	-	1.06	0.81	0.71	0.68	0.72
β_s	deg	144.0	145.4	147.1	148.8	150.6
d/ℓ	-	0.100	0.094	0.088	0.082	0.076

Table 2 Profile and cascade data tandem cascade rotor blade cascade 2

Sections	Dimen-sion	1	2	3	4	5
r	mm	114.5	127.0	139.5	152.0	164.5
t	mm	34.25	38.0	41.8	45.4	49.25
ℓ	mm	35.0	36.5	38.0	39.5	41.0
t/ℓ	-	0.978	1.041	1.100	1.150	1.201
β_{21}	deg	141.0	143.8	146.5	148.6	150.5
β_2	deg	109.0	119.4	127.8	134.1	139.0
$\Delta\beta_2$	deg	32.0	24.4	18.7	14.5	11.5
c_a^*	-	2.12	1.70	1.37	1.11	0.92
β_s	deg	121.6	128.0	133.3	137.6	141.0
d/ℓ	-	0.100	0.094	0.088	0.082	0.076

Using the selected relative cascade spacing a/l_1, pressure distribution and boundary layer calculations are performed for the individual sections by application of the method detailed in [8]. The results of the optimization calculations for all five sections of the tandem rotor blade are illustrated in Fig. 6, which for the optimum cascade spacings a/l_{1min} of sections 1 to 5 determined as above, indicates the relative deflection $\Delta\beta/\Delta\beta_\infty$ and the relative loss coefficient ζ/ζ_∞ as a function of the relative cascade displacement h/t. The relative loss coefficient for the compressor profiles represents the total pressure loss of cascades 1 and 2 referred to the dynamic pressure of the inlet flow. The optimization calculation is based on a Reynolds number Re_1 of 3.5×10^5. The lower part of Fig. 6 shows the correlation between the individual design sections and the relative cascade displacement h/t. For the individual sections it becomes obvious from the chart in terms of the relative loss coefficient ζ/ζ_∞ that all design sections feature an optimum cascade configuration with a minimum of losses. The reduction of profile losses as compared to corresponding single cascades with an indefinitely large axial spacing decreases from 24 % for section No. 1 to 12 % for section No. 5.

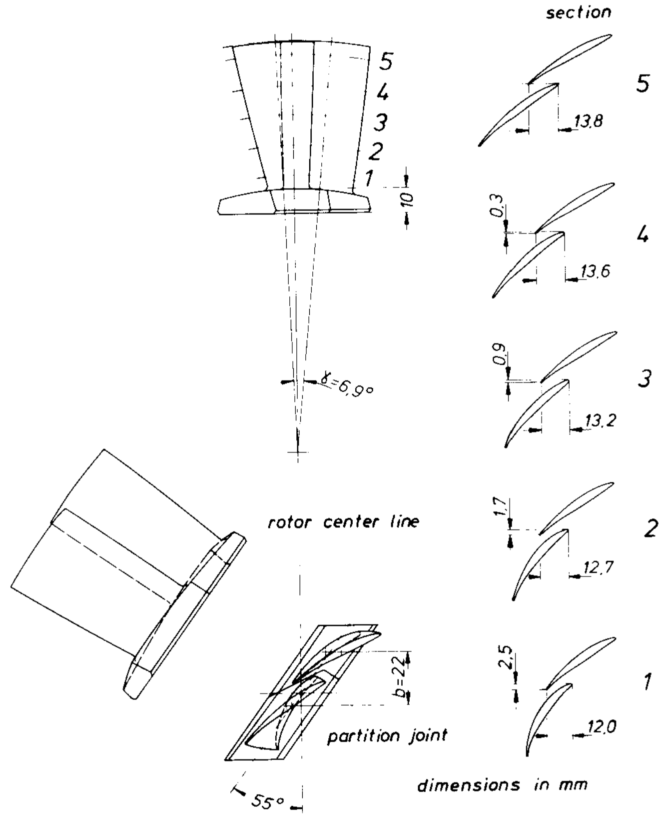


Fig. 5 Tandem rotor blade

Cascade examinations [1] for sections 1, 3 and 5 have confirmed the theoretically optimal cascade configuration. According to wind tunnel measurements the reduction in profile losses on account of the tandem-cascade effect is between 16 and 22 %. Accordingly, the results of the optimization calculation largely agree with the measurements.

Fig. 7 shows the tandem rotor blade designed for the above experimental compressor. For the actual implementation of the tandem rotor blade, a blade root fixation was developed which permits cascades No. 1 and No. 2 to be closely spaced. It was assumed that the blade buckets of both cascades are seated in each case on separate blade roots. Dovetail roots were selected. Both blade roots had a suitable split joint (Figs. 5 and 7) which made it possible to achieve an axial overlap of the two cascades.

In Fig. 8 we see two interferograms for the middle section of a rotor blade (section 3) with an optimum cascade configuration. The interferograms were taken by a four-panel interferometer with an infinitely large interference band width. According to the applied method, the stagnation point is the reference point for evaluation. When looking in flow direction at the points where the bands meet the profile contour, starting in each case from the leading edge

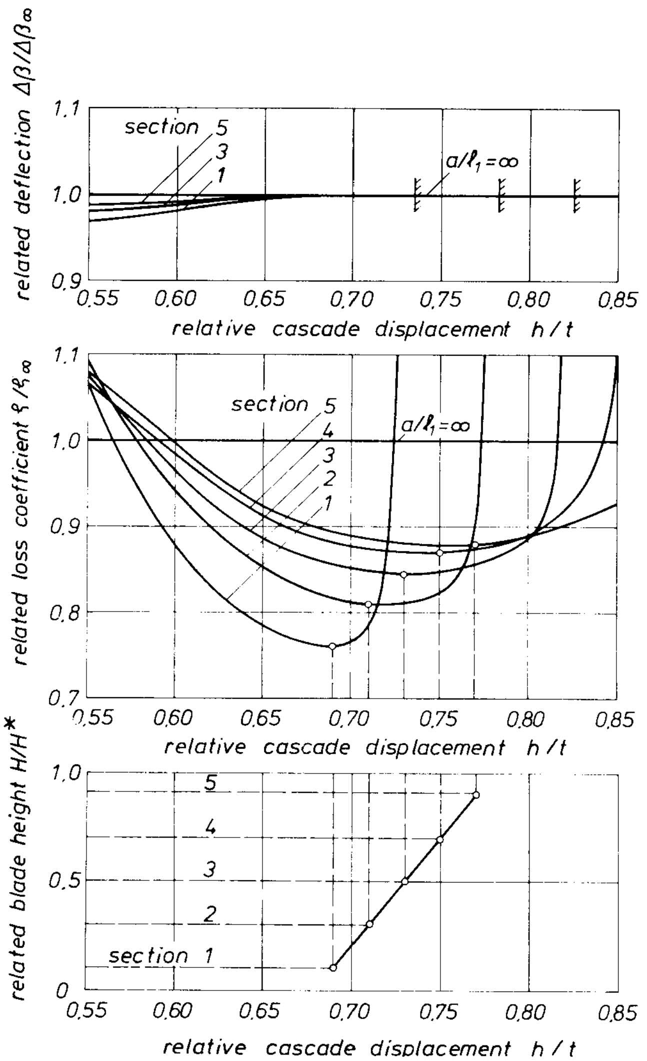


Fig. 6 Optimum cascade configuration for a tandem rotor blade

Section 1	$a/\ell_{1min} = -0.071$
Section 2	$a/\ell_{1min} = -0.047$
Section 3	$a/\ell_{1min} = -0.024$
Section 4	$a/\ell_{1min} = -0.008$
Section 5	$a/\ell_{1min} = 0$

stagnation point of the profiles, one can see that the density and the flow velocity change from band to band. With increasing number of bands, the flow velocity rises. A comparison of two interferograms recorded for inlet flow Mach numbers to cascade No. 1 of 0.36 and 0.5 shows (Fig. 8) that the tandem-cascade flow is influenced by the Mach number mainly inside the cascade No. 1 only, though outside the gap area. As can be further seen, the flow in the gap has almost the same configuration and is thus largely independent of the Mach number.

As experience has shown that greater gaps are presumably better than smaller gaps, an optimum relative cascade displacement $(h/t)_{opt}$ was selceted for section No. 1 of the tandem-cascade rotor blade in Fig. 5,

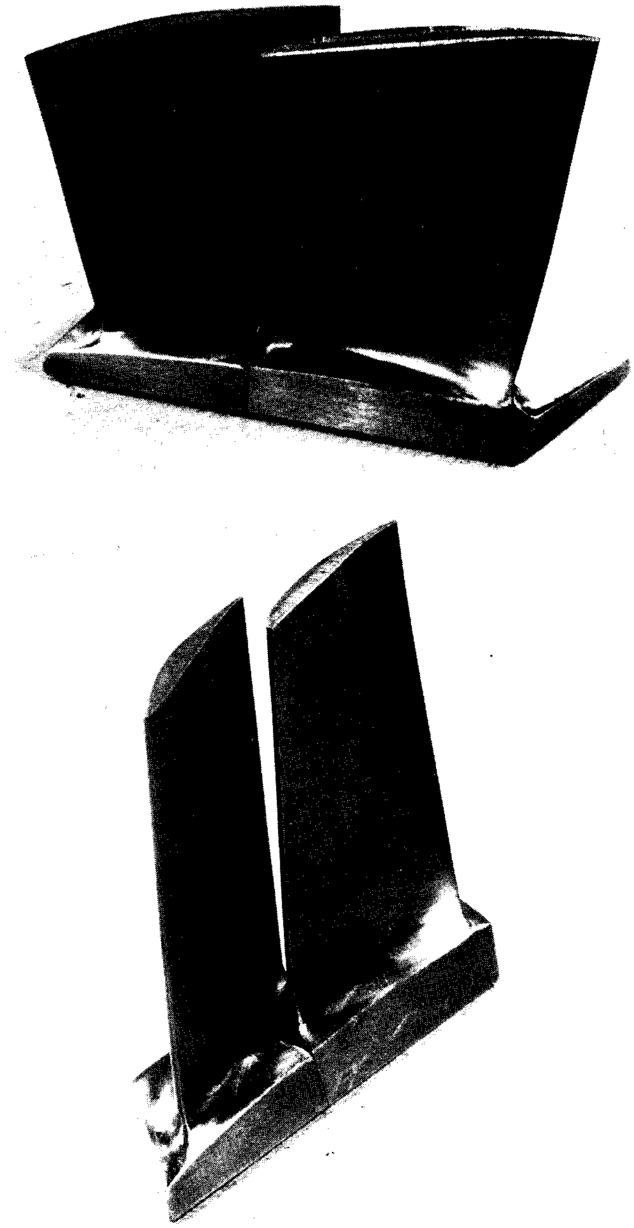
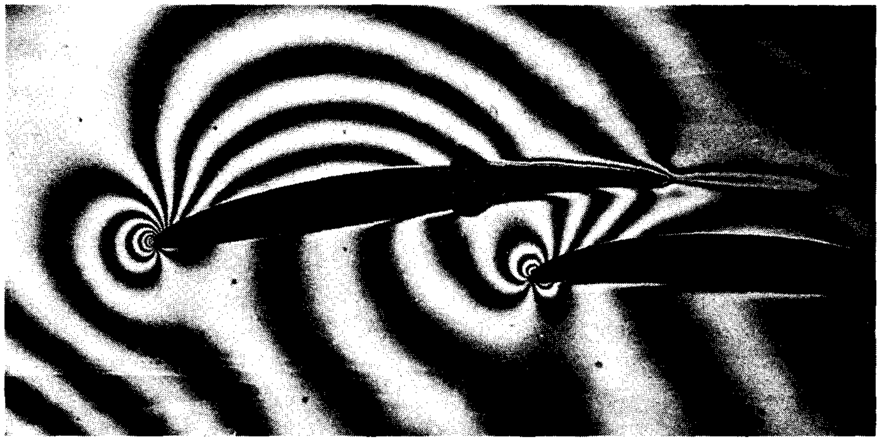


Fig. 7 Tandem rotor blade of an experimental compressor

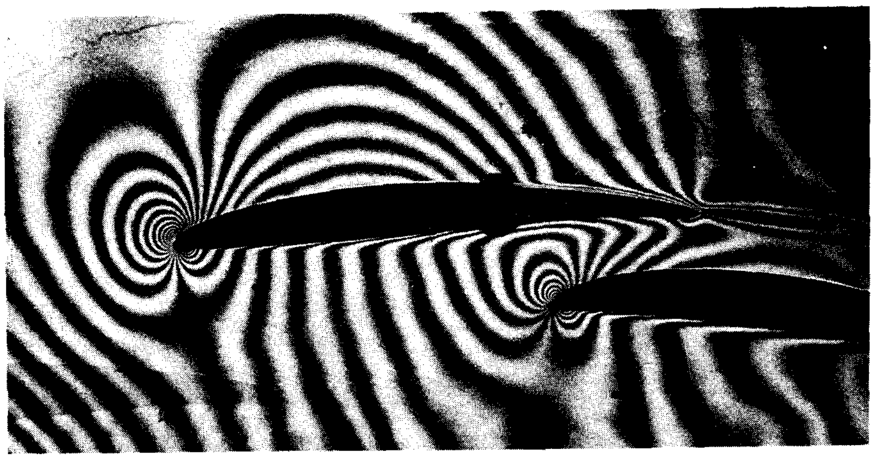
which is smaller by 5 % points than the corresponding value determined from Fig. 6. The boundary condition of all profile centres of gravity having to be arranged on radius vectors was satisfied. Accordingly, tandem rotor blades can also be used for comparatively high tip speeds such as occur in industrial axial-flow compressors.

Design and off-design performance of stator blades

The stator blade rows of the examined compressor blading consist of impulse cascades subjected to an aerodynamically higher load. Depending upon the profile section over the blade height, the deflection in the design point varies between 65 and 85 degrees.



$M_1 = 0{,}36$



$M_1 = 0{,}5$

Fig. 8 Interferograms for the mean section of a tandem rotor blade

β_1 = 155.5 deg, a/ℓ_1 = -0.024 h/t = 0.73

The deflection angles are thus ranging between the values for impulse-type turbines and deflection cascades subjected to a relatively low load, with deflections between 25 and 45 degrees, such as are used in conventional axial-flow compressors with 100 % reaction. As no examination results were yet available for impulse cascades in the area of medium deflection angles, Bammert and Ahmadi [9] performed specific investigations for this type of cascade.

Apart from their different aerodynamic loading in turbines and axial-flow compressors, deflection cascades show a different performance in off-design operation of these machines. As can be seen from the velocity diagrams for an impulse-type stage, the inlet flow to the impulse cascade approaches it from the suction side in off-design conditions. Flow separation will then start on the pressure side. On the other hand, the inlet flow to the stator-blade cascade of an axial-flow compressor with 100 % reaction in off-design conditions approaches it from the pressure side. Accordingly, flow separation will start on the suction side. Therefore, it can be assumed that impulse cascades in axial-flow compressors are bound to react more sensitively to flow separation under off-design conditions than in the case of impulse-type turbines.

The examinations conducted by Bammert and Ahmadi [9] were performed at the hub (A), mean (B), and tip sections (C) of a stator blade from the above experimental compressor (Fig. 9). The design date for the impulse blade are compiled in Table 3. Pressure distributions and cascade characteristics were determined in the cascade wind tunnel of the Institute for Turbomachinery for cascades A, B, and C. Fig. 10 shows the measured cascade characteristics for the three sections at an inlet flow Mach number of 0.2.

In Fig. 10 the loss coefficient ζ_v and outlet angle α_1 is plotted as a function of the inlet angle. The loss coefficient ζ_v is defined by

$$\zeta_v = 1 - \frac{c_1^2}{c_{1s}^2} \qquad (1)$$

In equation (1) c_1 and c_{1s} represent the outlet flow velocities for a flow with and without losses, respectively. The inlet angle was varied in the area between pressure and suction-side flow separation. In the design point, the loss coefficients are 0.038, 0.029 and 0.023 for cascades A, B, and C. The determined outlet angles were 45.0, 51.9, and 55.9 degrees.

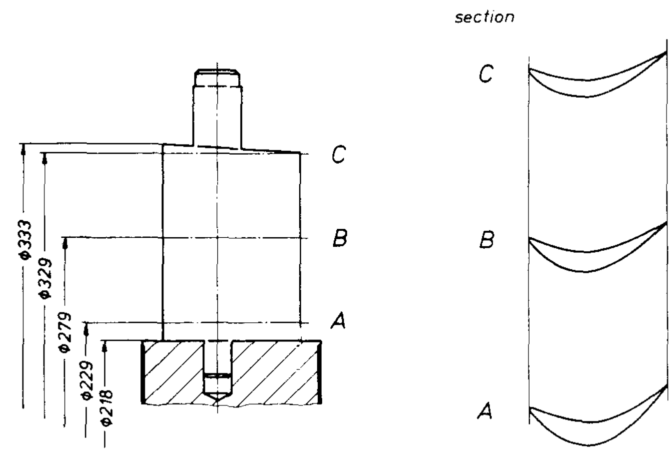


Fig. 9 Stator blade

A base section
B mean section
C tip section

The operating range is limited by the rise of the loss coefficient to 50 % above the minimum value. As can be seen from Fig. 10, the operating range in which there is no flow separation is 30 degrees for cascade A and 25 degrees fro cascades B and C. In the area without flow separation the outlet angle is largely independent of the flow inlet angle.

Overall performance of an experimental compressor

Upon conclusion of the examinations on the tandem rotor blades [1 and 8] and stator blades [9] the blading manufactured by M.A.N.-GHH Sterkrade, Oberhausen, was installed into a multi-stage experimental compressor (Figs. 11 and 12). First, the thermodynamic rating of the compressor was checked by operation in the rated duty point [10]. As can be seen from Fig. 13, the design point was nearly reached. At rated speed the delivery head achieved for the design mass flow of 6.6 kg/s was 1 % less than the design value. The total internal efficiency of the blading was measured as 83.7 %, which was 1.1 % points less than the theoretical design value. As it will be explained later the measurements have shown higher losses in the hub area of the rotor blades than predicted. Considering a more optimum design total internal efficiency of 86 % can be expected for the blading.

Table 3 Layout parameter of the impulse cascade

Index	Unit	A	B	C
Inlet flow angle α_0	deg	47.3	53.2	56.4
Outlet flow angle α_1	deg	48.6	54.2	58.3
Angle of flow deflection	deg	84.1	72.6	65.3
Stagger angle	deg	83.o	83.o	81.1
Pitch-chord ratio	-	o.47	o.58	o.68
Relative Thickness	-	o.16	o.14	o.12

When considering the comparatively high stage load, and in view of the fact that in terms of engineering new ground was stepped

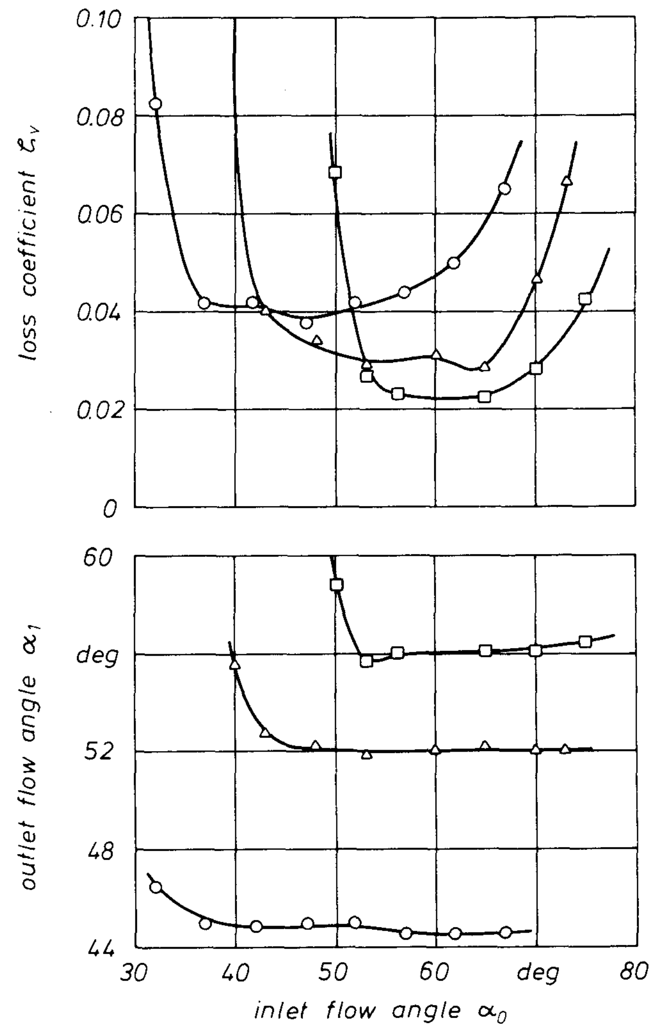


Fig. 10 Cascade characteristics stator blade $M_0 = 0.2$

impulse cascade A hub section
impulse cascade B mean section
impulse cascade C tip section

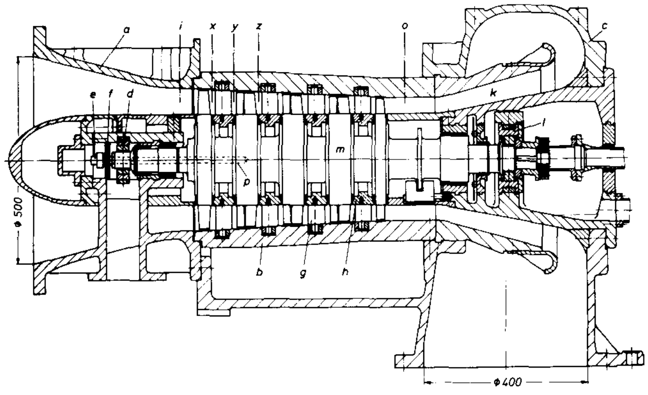


Fig. 11 Experimental compressor with tandem rotor blades

- a inlet casing
- b stator-blade carrier
- c outlet casing
- d journal bearing
- e co-rotational transmitter
- f receiving antenna
- g stator blade
- h shroud
- k diffuser
- l double thrust bearing
- m rotor with rotor blades
- p bore for transmitter line
- i,x,y, z,o measuring planes

on by the novel arrangement of tandem rotor blades and single-cascade stator blades, the coincidence between the theoretical values and the measured values is to be assessed as "good". This means that the thermodynamic design method has stood the test.

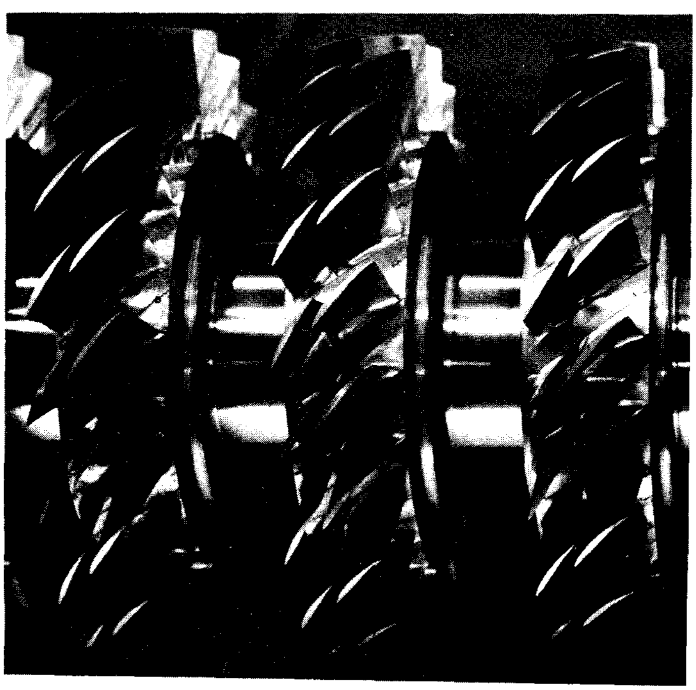


Fig. 12 Rotor of an experimental compressor

Some very important information about the performance of the tandem-cascade compressor is obtained from performance measurements with speed control (Fig. 13). Global performance measurements show the efficiency curve and the pressure ratio in the stable operating range. The performance curves illustrated in Fig. 13 were produced by scanning the flow upstream and downstream the blading (plane i and o. in Fig. 11). As can be seen from the upper diagram, the characteristic curves have a comparatively flat shape towards the maximum. This can be explained by the relatively high stage load. Another striking feature is that at rated speed the distance between the surge line and the design point is fairly small. The basic aim pursued in designing the blading was to

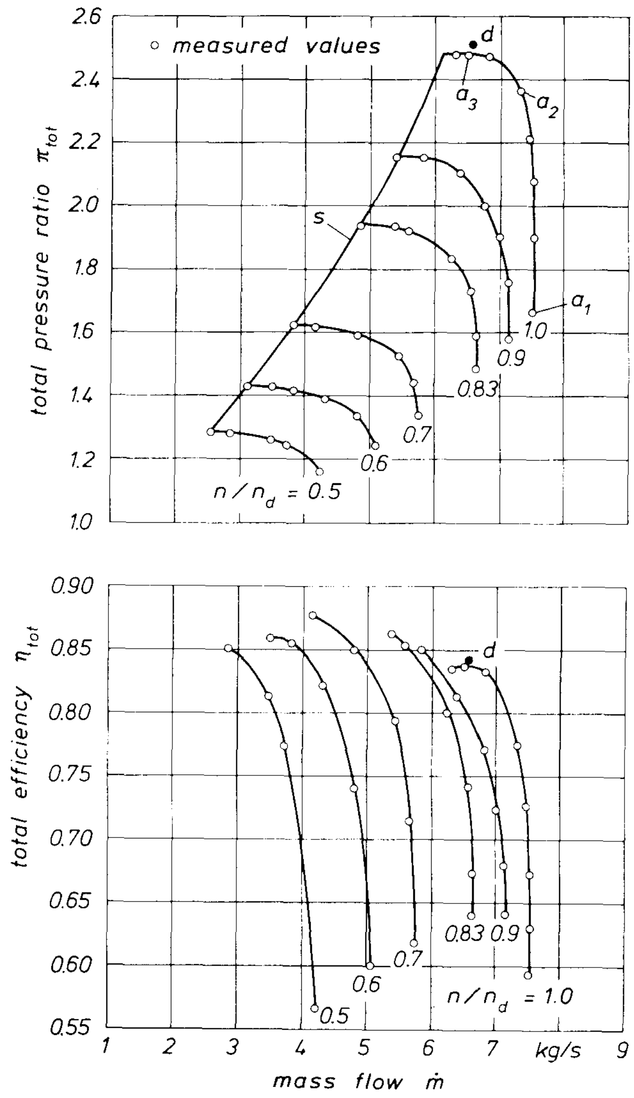


Fig. 13 Performance curves of experimental compressor

- d design point
- s surge line
- n/n_d speed ratio
- a_1, a_2, a_3 operating points

achieve the same delivery head with a tandem-cascade stage as in two single-cascade stages of conventional design. This led to fairly high aerodynamic loads, especially in the hub area of cascade No. 2 of the tandem blade rotor. Apparently, this also had its effects on the surge line. Therefore, it is to be recommended for industrial compressors to reduce the aerodynamic load of the tandem rotor blades by approximately 10 to 15 % as against the experimental compressor. In this way one would certainly re-establish a somewhat steeper performance curve.

For constant speeds the efficiency curves reach a maximum near the surge line. This is also explained by the fact that on account of the aerodynamic load the surge line is located at relatively large suction volumes.

Scanning of flow

To permit the performance of the experimental compressor to be analysed more precisely, the flow was scanned, amongst others, upstream and downstream the first stator blade row as well as upstream and downstream the first tandem blade rotor over the blade height [10]. Flow scanning was performed for the duty point a_3 (Fig. 13) which corresponded to the design point. Apart from the radial distribution of pressures and temperatures, scanning also supplies data of the flow angle and the efficiencies, so that the performance of the rotor and stator can be examined in detail.

Fig. 14 shows the arrangement of the measuring planes as well as the location and number of the measuring radii R_1 to R_{10}. Flow scanning downstream the stator blade row was performed radially and peripherally. For taking the scanning measurements, wedge-shaped probes were used, such as are described in [11].

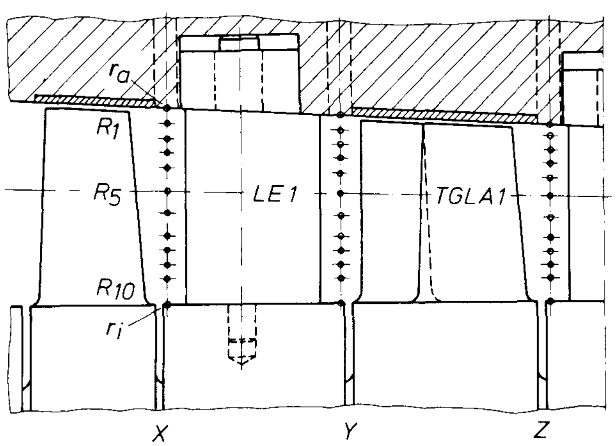


Fig. 14 Measuring planes for scanning in the experimental compressor

LE 1 stator blade
TGLA1 first tandem rotor blade

The results of the scanning measurements are plotted in the next figures (Figs. 14 to 18), as a function of the relative radius in line with equation

$$r' = \frac{r - r_i}{r_a - r_i} \tag{2}$$

where r is the respective measuring radius between the hub (r_i) and the outer wall contour (r_a).

In Fig. 15 we see a comparison of the measured flow angles and the design angles relating to the stator blades. The top diagram shows that the medium approaches the stator blade row almost over the full height too much from the suction side by 4 degree, which causes a certain flow acceleration. Measurements and design are in good agreement for the outlet flow angle, except for the areas near the wall.

Fig. 16 shows the efficiency of the stator blade η' based on the definition

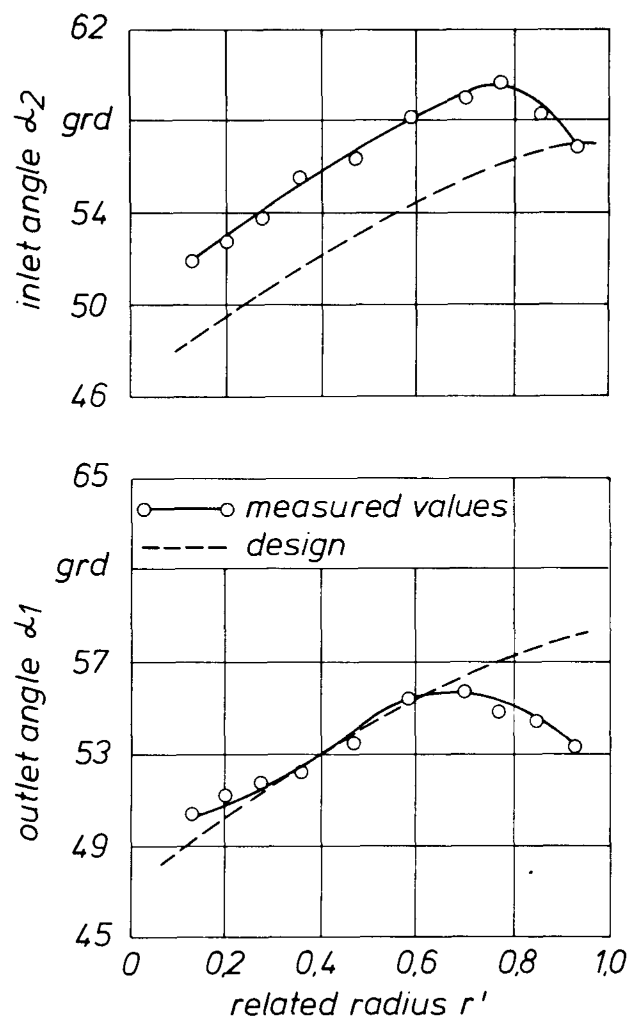


Fig. 15 Flow angles stator blade LE1 (design point)

$$\eta' = \frac{c_1^2}{c_{1s}^2} \qquad (3)$$

Index "s" indicates the isentropic status. Comparatively good stator efficiencies were ascertained in the middle area of the mean flow. On account of secondary and wall losses, these deteriorate by a relatively large degree towards the hub and tip.

The flow angles measured in the design point for the tandem blade rotor are represented in Fig. 17. The radial pattern of the measured flow inlet angle approximately corresponds to the design values. As compared to the design, the inlet flow to the tandem blade rotor drifts too much to the suction side by approximately 2 degrees in the area of the mean flow.

In consideration of a homogeneous blading design, a mean inlet flow angle distribution was selected for all tandem impellers; this, however, in the same sense as measured, deviates from the theoretical inlet flow angle by approximately 1 degree. The inlet flow angle deviation then still remaining is assumed to result from the increase of the axial velocity in the stator blade row.

As can be seen from Fig. 17, the measured outlet flow angle partly deviates by a considerable margin from the design angle. In the outer channel half, the deflection is up to 4 degrees greater, whereas towards the hub it is up to 3 degrees smaller than the design value. This can be interpreted as suggesting a beginning of flow separation in the hub area, which causes the flow to be displaced towards the outside. Obviously, this is also confirmed by the radial distribution of the tandem blade rotor is defined by the relation

$$\eta'' = \frac{w_2^2}{w_{2s}^2} \qquad (4)$$

As can be seen, the tandem rotor efficiency declines very strongly towards the hub, which is assumed to be due to local flow separation. Based on these measurements it can be said that the selected aerodynamic loading in the rotor hub area with a deflection of 46.5 degrees was too high.

Checking the outlet flow angle by re-

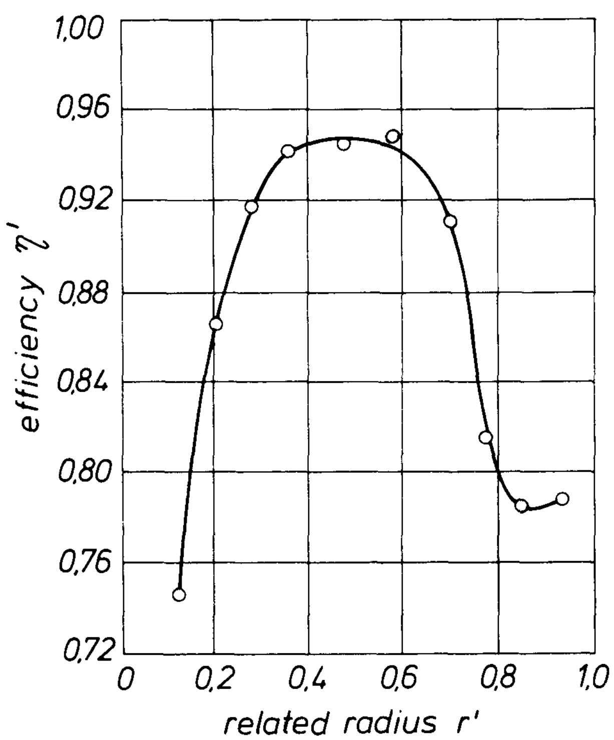


Fig. 16 Efficiency stator blade LE1 (design point)

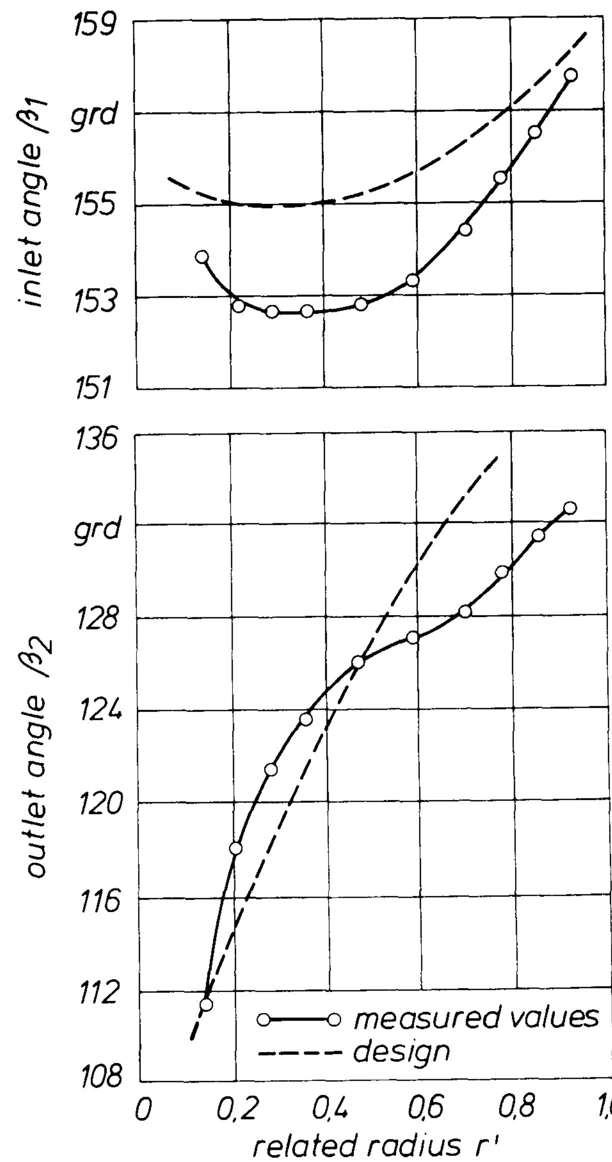


Fig. 17 Flow angles tandem rotor blade TGLA1 (design point)

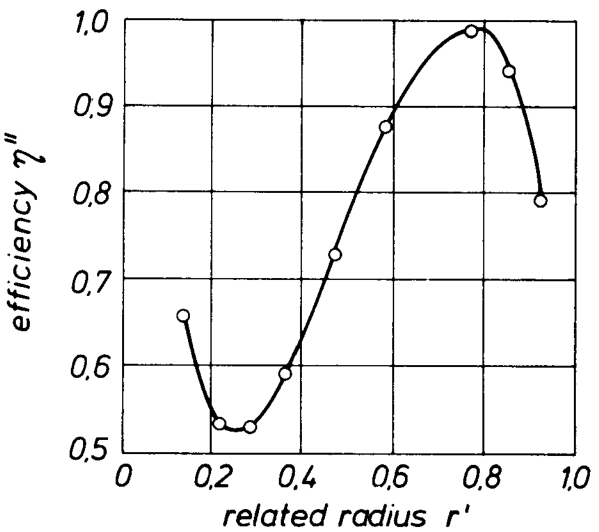


Fig. 18 Efficiency tandem rotor blade TGLA1 (design point)

ferring to the generally known criteria has shown that there is no phenomenon of flow stagnation at the hub in the design point. However, if throttling is continued, the possibility of flow stagnation near the hub cannot be fully excluded.

BASIC CONCEPT OF INDUSTRIAL COMPRESSORS WITH TANDEM-CASCADE BLADING

Blading design

For the industrial application of a homogeneous tandem-cascade blading the following basic principles can be derived from the multitude of examinations and studies that have been performed on the four-stage tandem-cascade axial-flow compressor described above [10] :

1. To permit a compressor stage to be built which consists of tandem rotor blades and single-cascade stator blades, the degree of reaction should be in the range between 90 and 100 %.

2. To avoid high aerodynamic loads in the tandem-blade root section as well as the stagnation of flow at the hub, the free vortex distribution should be disregarded when designing the blading in the hub area.

3. In consideration of a relatively wide stable operating range, the aerodynamic load of the tandem-cascade stage should not exceed the 1.7-fold of the load of conventional single-cascade stages. The aerodynamic design of the individual rotor blade profiles can be based on the diffusion factors developed by Lieblein.

4. The design of the tandem rotor blade should consider a uniform distribution of the total delivery head over the two blades in order to achieve minimum loss coefficients for the tandem blade. The profile families to be used for the blades are optional. Axial overlap of cascades No. 1 and No. 2 is not absolutely necessary, so that basically all blade roots which can be axially inserted, such as pinned, fir-tree, and dovetail roots, are eligible for the fixation of tandem blades.

   Fig. 19 shows the optimum cascade configurations of a tandem rotor blade for industrial application. As can be seen, the profile losses for a tandem arrangement are by approximately 13 to 22 % lower than the values of comparable single cascades.

   When designing tandem rotor blades one must consider that the profile centres of gravity of all blade sections are located on radius vectors passing through the rotor axis, thus avoiding additional bending stresses in the blade bucket caused by centrifugal forces.

5. The stator blades should have thick leading edges with a relatively wide insensitively range.

   If possible, cylindrical stator blades should be used for reasons of economy in manufacture.

6. The danger of soiling of the blades, which would cause the gaps to be filled, does not exist. This risk is excluded because there is always flow acceleration in the gap, preventing dirt particles from depositing.

Design features of an industrial compressor

The last illustration, Fig. 20, shows the possible design of a multi-stage axial-flow compressor with tandem cascades. In this case an industrial axial compressor with 100 % reaction has been selected which is followed by a centrifugal end stage.

As a tandem-cascade stage with 100 % reaction blading is bound to feature elatively high inlet Mach numbers, it is recommended to arrange single-cascade stages upstream the tandem-cascade stages. The advantage inherent in this is that a degree of reaction other than 100 % could be adopted for the first stages, in this way further increasing the mass flow rate referred to the inlet area, while maintaining the max. inlet Mach number. Furthermore, this basic concept permits adjustable stator blades to be fitted to the first stages if guide vane control is required.

Tandem-cascade stages have flatter performance curves than conventional single-cascade stages. However, as relatively steep performance curves are desirable in many cases, the combined use of stage groups of single-cascade or tandem-cascade design yields an optimum performance with a considerable reduction in the constructional expenditure.

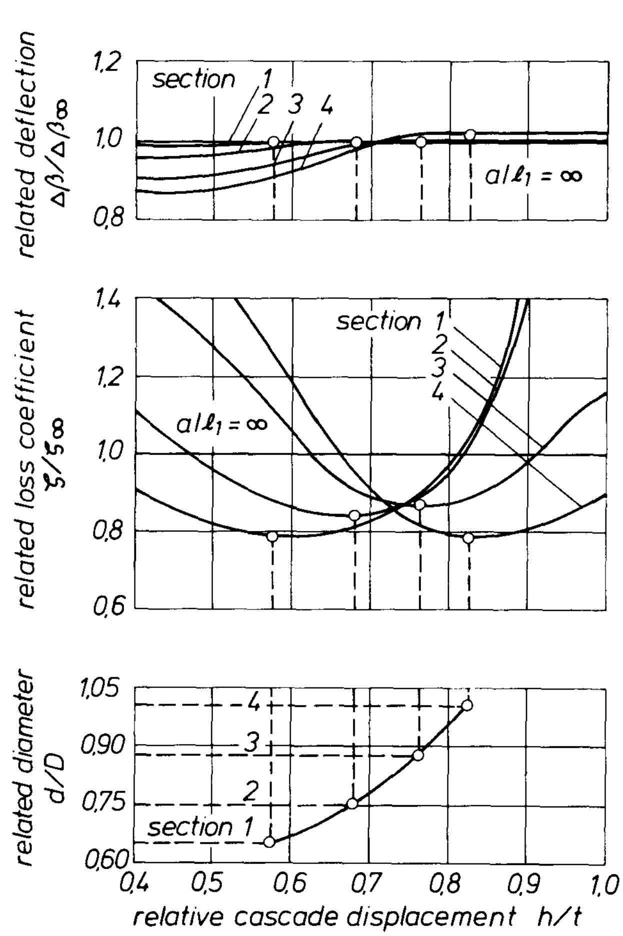


Fig. 19 Optimum cascade configuration for a tandem rotor blade

Section 1 $a/\ell_{1opt} = +0.029$
Section 2 $a/\ell_{1opt} = +0.064$
Section 3 $a/\ell_{1opt} = +0.107$
Section 4 $a/\ell_{1opt} = +0.150$

When introducing axial-flow compressors of tandem-cascade design, one can work on the premises that the individual constituent parts of the blading, such as deceleration and deflection cascades, are conventional components which have proved effective in a great number of applications. Therefore, the use of tandem cascades in industrial axial-flow compressors does not represent a too excessive step in engineering development. Faultless and unproblematic operation of the experimental compressor has shown that additional risks for the operational reliability of a tandem-cascade compressor are not to be anticipated.

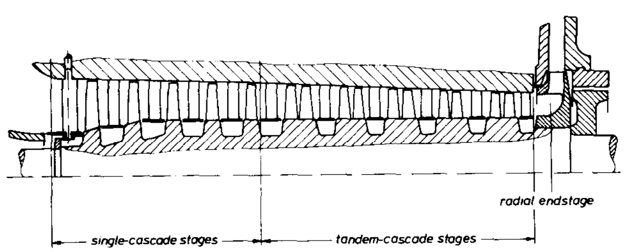


Fig. 20 Industrial compressor with tandem-cascade stages

SUMMARY

Since the development of the slotted aerofoil, the use of this particular configuration for blade profiles in the form of so-called tandem cascades has been an obvious alternative. Employing tandem-cascade blades in axial-flow compressors has been an attempt that was made at a relatively late date.

The fact that tandem cascades have not as yet been installed in axial-flow compressors to a greater extent can be attributed to the previously examined degree of reaction, which used to be 50 %, where only minor improvements were achieved in the aerodynamic properties.

For this reason, the present paper reports about examinations and studies made on a 100 % reaction blading. Theoretical and experimental cascade investigations have shown that tandem rotor blades can be optimized over the blade height to achieve minimum flow losses. The optimization done for the design point of different cascades has not taken into account the three-dimensional effects coming from the hub and the wall. As compared to conventional single cascades, a tandem-cascade configuration is able to reduce the flow losses by approximately 18 %.

Single-cascade stator blade rows consisting of impulse cascades with a medium deflection angle show a more sensitive reaction to flow separation in off-design operation than impulse cascades employed in impulse-type turbines. The operating range in which there was no flow separation on the examined stator blades was between 25 and 30 degrees.

The thermodynamic design was checked by measurements on a four-stage experimental compressor. The design point was almost fully reached. At design volume conditions, the delivery head fell short of the rated value by 1 %. The total efficiency of the blading was 83.7 %.

To permit a more accurate analysis of the experimental compressors performance, the flow was scanned radially downstream the stator blade row and downstream the tandem blade rotor.

The scanning measurements showed that with the chosen relatively high aerodynamic loading of the tandem rotor blade, the flow in the tandem rotor was displaced from the hub towards the stator-blade carrier due to the three-dimensional effects near the hub. The basic principles derived from the measurements taken will permit the reliable rating of tandem-cascade axial-flow compressors for

industrial applications in the future. In conclusion, the basic concept of a multi-stage axial-flow compressor is explained which is designed for application in the chemical as well as the iron and steel industries. The next stage of development should be the use of single-cascade stages of a relative low degree of reaction together with downstream tandem-cascade stages with a reaction in the range of 100 %. This will yield an optimum performance in terms of the characteristics and surge line, while considerably reducing the constructional expenditure.

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