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# Influences of Manufacturing Tolerances and Surface Roughness of Blades on the Performance of Turbines

*The costs of the manufacturing of turbine blades are very dependent on the manufacturing tolerances and the quality of the surface. A good performance of a turbine needs a certain smoothness of the surface and small tolerances. For an optimisation of the costs it is necessary to know the influences of roughness and tolerances on the performance of a turbine. In our institute measurements were carried out with thinned and thickened blades mounted in a turbine, which represents different manufacturing tolerances. In addition measurements on a turbine with roughened blades were done. From these measurements a conclusion on the aerodynamical and thermodynamical behavior of the turbine is obtained. The paper gives a summary of the measurements and shows how the performance of a turbine is affected by the roughness and the profile tolerances.*

## Introduction

The manufacturing costs of turbine blades are to a considerable extent dependent on the following factors:

1 quantity,
2 production method,
3 material,
4 blade geometry (profile shape, twist etc.)
5 blade fastening,
6 blade size,
7 surface quality and
8 tolerances.

This paper is only going to deal with the last two points, namely the influence of production inaccuracy and surface roughness on the characteristics of turbines.

The flow surfaces of blades in flow machines have the function of directing and guiding the working medium. If they are to perform their function properly, a certain surface quality or minimum surface roughness is called for. As a result of the manufacturing process and technical expenditure involved turbine blades are produced with a surface roughness which in most cases becomes even greater during operation of the turbo machine as a result of erosion, corrosion, and contamination. In addition to the permissible surface roughness, the production of turbine blades must also observe certain tolerances with regard to the shape of the blade profiles. Deviations from the specified profile can occur in a relatively uniform form, e.g., through wear by the cutter during copy-milling, through wandering of the die during forging or through uncontrollable congealing during casting.

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## Definition of Roughness and Tolerances

An assessment of the manufacturing accuracy of a turbine blade is aided by a knowledge of the contour tolerance, the thickness tolerance, the chord length tolerance, the twist tolerance, and the location tolerance. In Fig. 1 the tolerances are shown on a profile section. The contour tolerance $a$ is the permissible vertical deviation of the profile contour from the specified profile which is shown in bold print. It is presupposed that the profile is not wavy or exhibits kinks in curvature through deviations within the contour tolerance. The thickness tolerance $b = d_1 - d$ as the difference between the profile thicknesses $d_1$ of the actual profile and $d$ of the specified profile gives the allowed tolerances from the profile thickness; it is of particular significance for the trailing edge thickness $D$ and hence for the efficiency of turbo machines. The chord length tolerance $\Delta l = l_1 - l$ signifies a permissible inaccuracy in the length of the profile chords, with $l_1$ as the chord length of the actual and $l$ that of the specified profile. The chord length tolerance is important for the discharge flow direction of the working medium. The twist error $\Delta\beta_s$ is a value for the deviation of each profile section from the preset stagger angle $\beta_s$, which is measured between the profile tangents and the cascade front [1].[1] The foregoing tolerances have an effect on the losses and on the discharge

[1] Numbers in brackets designate References at end of paper.

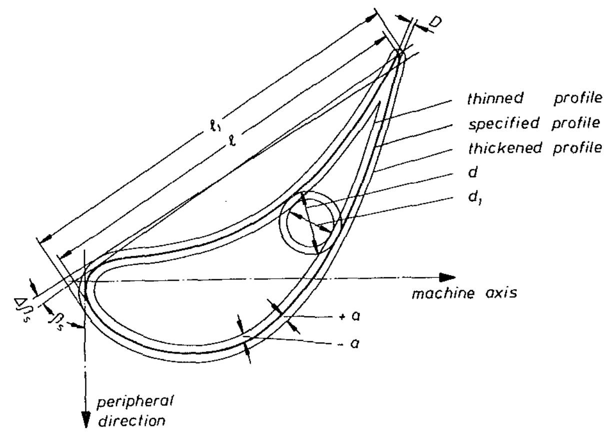


**Fig. 1 Definition of the manufacturing tolerances: $a$ = contour tolerance; $D$ = trailing edge thickness; $d$, $d_1$ = profile thickness; $l$, $l_1$ = blade chord; $\beta_s$ = stagger angle; $\Delta\beta_s$ = twist tolerance**

flow direction, which in turn influences the mass flow, the enthalpy drop and the efficiency of the turbine. The tolerances and the roughness also exert an influence on the mechanical behavior of a blade (e.g., vibrations and strength), but this is not going to be discussed here.

For twisted blades the profiles are fixed in several paraxial sections, for which location tolerances are to be given. The blade contour is interpolated between the individual sections. The more a blade is twisted and the smaller the tolerance areas are the closer the paraxial sections must be before one can assume the same profile accuracies for those sections which are not measured. In general the chosen distance between sections is between 15 and 40 mm, according to requirements. The profile contours are measured with bladetesting devices whereby contour gauges are pushed against the profile and the deviations from the specified profile are measured by feeler gauges and according to the light-gap procedure. Other possibilities exist using optical or scanning methods. With the latter procedures the location tolerance of the profile sections at the blade root is very important. The measuring procedures are not going to be discussed here.

Normally the coordinates of profiles for a two-dimensional reference system are given and adopted without adjustment for the manufacture of the blades. However, since a plane cascade strictly speaking represents the development of a cylinder section, manufacture would really have to take into account the curvature of the section's surface; this influence, however, is so small as to be negligible.

The quality of the surface is defined as the height of the peaks, the structure of the roughness and the waviness. In workshop practice, however, only a roughness value is prescribed. In Germany this is usually the greatest height of the roughness peaks, "the roughness height $R_t$." In the English-speaking countries, the center-line-average value CLA or arithmetical average AA is more common.

If the peaks are small, i.e., enveloped in the laminar sublayer, then there is no influence on the skin friction. The surface acts hydraulically smooth. If the peaks protrude from the laminar sublayer then the exchange of impulses from turbulent pulsations on these rough areas becomes greater, the shearing stress increases, and additional loss occurs. The turbulent pulsations are not just dependent on roughness height but also to a large degree on the structure of the roughness. There is a discrepancy, therefore, between the flow roughness and the manufacturing roughness. The correlations are very complicated [2].

For the flow effect, therefore, predominantly idealized and simplified forms of roughness were investigated, such as

corrugated roughness
ribbed-type roughness elements and
idealized sand roughnesses.

For the roughness of the blade surface one uses the equivalent sand roughness $k_s$. In the range of the square resistance law—i.e., a fully turbulent boundary layer and the total pressure loss $\Delta p_{tot} \sim w^2$ with $w$ as the velocity of the flow medium—it produces the same resistance coefficient as the surface under consideration. Strictly speaking the sand roughness only applies to a certain form of roughness which consists of tightly packed granules of sand of the grain size $k_s$. If the resistance coefficients determined for the sand roughness are to be referred to other forms of roughness then one must know which roughness height (e.g., the roughness height $R_t$ or the center-line-average value CLA) characteristic for the form of roughness concerned is equivalent to the sand grain size $k_s$, so that both forms of roughness produce the same resistance. For a milled surface where the flow is perpendicular to the milling grooves the milling roughness equivalent to the sand roughness is

$$R_t = 2.56\ k_s \tag{1}$$

That means that the height of the milling roughness is, at the same resistance coefficient, 2.56 times the size of the sand grains. At the same roughness height the milled surface acts smoother than a surface roughened by sand. If the flow over the milled surface is in the direction of the milling grooves then the milling roughness equivalent to the sand roughness is

$$R_t = 5\ k_s \tag{2}$$

Just a slight deviation of the flow direction from the direction of the milling grooves by 10 deg makes the equivalent roughness drop to the value of 2.56 $k_s$ [3]. For mechanically manufactured surfaces the German standard DIN 4767 correlates the roughness height $R_t$ to the center-line-average value CLA. On the basis of the values in the standard we have the relation

## Nomenclature

$a$ = contour tolerance, mm
AA = arithmetical average, $\mu$m
CLA = center-line-average value, $\mu$m
$D$ = trailing edge thickness, mm
$d$, $d_1$ = profile thickness, mm
$e$ = throat, mm
$H$ = shape factor of boundary layer
$k_s$ = sand roughness, $\mu$m
$l$, $l_1$ = profile length, mm
$\dot{m}$ = mass flow, kg/s
$n$ = speed, r.p.m.
$P$ = output, W
$p$ = pressure, bar
$R$ = ideal gas constant, J/(kg K)
$R_t$ = roughness height, $\mu$m
Re = Reynolds number
$\mathrm{Re}_\vartheta$ = boundary layer Reynolds number
$T$ = temperature, K
$t$ = pitch, mm
$w$ = flow velocity, m/s
$\beta_1$ = inlet angle
$\beta_2$ = discharge angle
$\beta_s$ = stagger angle
$\delta$ = displacement thickness, mm
$\delta_u$ = turning angle
$\Delta h$ = enthalpy drop, J/kg
$\Delta h_s$ = isentropic enthalpy drop, J/kg
$\Delta p_{tot}$ = total pressure loss, bar
$\Delta\beta_s$ = twist tolerance
$\Delta\eta$ = efficiency drop
$\epsilon$ = trailing edge coefficient
$\zeta_D$ = trailing edge loss coefficient
$\eta(T)$ = dynamic viscosity, Ns/m$^2$
$\eta$, $\eta_e$ = efficiency
$\vartheta$ = momentum loss thickness, mm
$\nu$ = kinematical viscosity, m$^2$/s
$\rho$ = density, kg/m$^3$

**Subscript**

0 = design

$$\mathrm{CLA} \approx 0.14\, R_t^{1.14} \tag{3}$$

For such calculations, however, one must usually reckon on a margin of error of 50 percent or more. Equation (3) is valid when CLA and $R_t$ are expressed in $\mu$m. According to our own measurements on areas with an emery grain surface, equation (3) applies to this kind of surface too; it can be assumed as valid for a variety of surfaces. For mechanically produced surfaces it is suggested that the center-line-average value be converted into the sand roughness height $k_s$ according to the relation

$$k_s \approx 2.19\, \mathrm{CLA}^{0.877} \tag{4}$$

derived from equations (1) and (3).

## Consequences of Roughness and Profile Inaccuracy

The properties of a blade cascade—essentially the loss and the discharge flow direction—can be significantly influenced by surface roughness and profile inaccuracy, which in turn can change the characteristic values of a turbine such as efficiency, mass flow and enthalpy drop. In order to be able to classify and understand the processes we must first visualize the actual physical process which takes place during flow in a cascade.

The losses in a plane blade cascade arise through friction on the blade surface, through separation zones and, in certain circumstances, through shock waves at supersonic velocities. The second characteristic for assessing a cascade, the discharge angle, is largely determined by the cascade geometry and therefore by the profile accuracy in manufacture.

The shock losses are excluded from our considerations since in the normal working range they occur relatively seldom. The loss of separation, which arises right at the inlet of the cascade owing to extremely faulty flow, is only of significance at part-load conditions. More important is the separation loss which occurs at the trailing edge of the profile, the so-called trailing edge loss, because in general the flow cannot follow the curvature of the trailing edge. The losses are thus mainly composed of the portions of pure boundary layer friction on the blade surface and that of the trailing edge separation.

The friction loss of a blade is only affected very slightly by the usual manufacturing inaccuracies, it is the roughness which is of great influence. At the beginning in the upstream stagnation point of a cascade flow the boundary layer is laminar. With further progress along the contour the laminar boundary layer becomes unstable and veers into the turbulent current. The turbulent flow causes higher losses than a laminar one. The roughness of the surface has two effects: it pushes the transition point of the boundary layer forwards, as can be seen from Fig. 2, and increases the friction in the subsequent turbulent boundary layer. The figure shows the boundary layer Reynolds number plotted against the shape factor $H$ of the boundary layer, where the roughness referred to the momentum loss thickness $\vartheta$ is shown as a parameter. The shape factor characterizes the velocity profile of the layer near to the wall, it is the ratio of the displacement thickness $\delta$ to the momentum loss thickness $\vartheta$ of the boundary layer, i.e., $H = \delta/\vartheta$. The Reynolds number is formed from the velocity in the undisturbed flow, the momentum loss thickness at the transition point in the case of laminar flow, and the kinematic viscosity. One can see that at the same boundary layer condition ($H$ = const. and $\vartheta$ = const.) the transition is more likely to occur the greater the roughness. The full curves originate from measurements [4, 5], the dotted part is a theoretical calculation for smooth surfaces [6].

The turbulent flow downstream the transition point has the smallest loss at hydraulically smooth conditions. The admissible sand roughness for this is a function of the Reynolds number formed with the chord length $l$. For the relative roughness the upper limit is given as

$$(k_s/l)_{\mathrm{adm}} \leq 100/\mathrm{Re} \tag{5}$$

With the Reynolds number

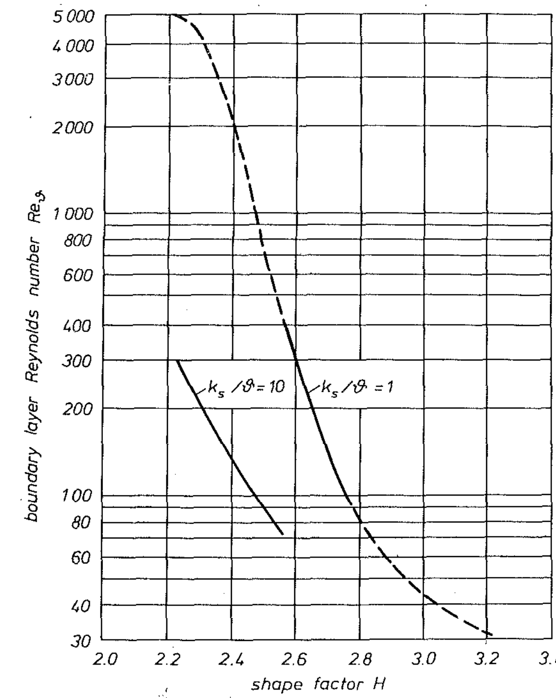


**Fig. 2 Transition point with sandgrain roughness: —— transition with rough surface; – – – transition with smooth surface and a high turbulence**

$$\mathrm{Re} = wl/\nu \tag{6}$$

one obtains

$$k_{s\,\mathrm{adm}} \leq 100\ \nu/w \tag{7}$$

One can see from equation (7) that the admissible roughness increases in proportion to the kinematic viscosity $\nu$ and in inverse proportion to the flow velocity $w$. The kinematic viscosity

$$\nu = \eta(T)/\rho = \eta(T)RT/p \tag{8}$$

achieves its lowest values at high pressures $p$ and high temperatures $T$. $\eta(T)$ is the dynamic viscosity, $\rho$ the density and $R$ the gas constant.

In a steam turbine changes occur, for example

the *temperature* from 550 to 25°C, i.e., $(550 + 273)/(25 + 273) \approx 3$

the *pressure* from 300 to 0.03 bar, i.e., $300/0.03 = 10\,000$

For the given temperature range $\eta(T)$ changes by about the factor 3, so that at the same velocity at the turbine inlet the permissible roughness of the blades would have to be 1 000 times lower than at the outlet. As a result, turbo machines for high pressures require good surfaces and are particularly sensitive to manufacturing faults and erosive and corrosive influences. With careful manufacturing, milling roughnesses of 3 to 10 $\mu$m are achieved (CLA = 0.5 to 1.9 $\mu$m).

Above the admissible roughness the loss increases sharply, as Fig. 3 shows. Here the efficiency drop is plotted against the relative roughness $k_s/l$. The efficiency drop $\Delta\eta$ means in this instance the difference between efficiencies when the blade surface is smooth and when it is rough. The permissible roughness in the former case is $k_s/l \approx 0.2 \cdot 10^{-3}$. Under investigation were the mean sections (Section 3) of the turbine blading shown in Fig. 4.

What has been said so far has referred to completely rough blades. With only partly rough blade surfaces the greatest effect on losses is to be observed for blades which are rough in the rear part of the suction side. In the front region the influence is less. In general it can be said that the roughness is more noticeable when the flow is retarded.

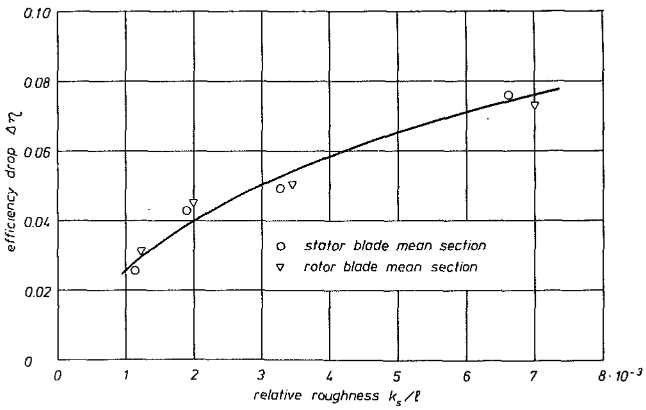


**Fig. 3 Efficiency drop in a two-dimensional cascade**

The separation loss at the trailing edge is mainly dependent on the thickness of the trailing edge. It is possible to show theoretically that a linear relationship exists between the trailing edge thickness and the trailing edge loss [5]. Thus a thickening of the trailing edge through utilization of the contour tolerance has a direct effect on the loss. The trailing edge loss is calculated according to

$$\zeta_D = \epsilon \frac{D}{t \sin \beta_2} \tag{9}$$

where $\epsilon$ means a coefficient which is shown in Fig. 5 plotted against the turning angle $\delta_u = 180° - \beta_1 - \beta_2$ of the cascade, $D$ is the trailing edge thickness, $t$ the pitch and $\beta_1$ and $\beta_2$ the inlet and outlet flow angles, respectively, a definition of which can be seen from Fig. 4. The graph in Fig. 5 is the result of a large number of measurements [7].

The second characteristic magnitude—the outlet flow angle of the plane cascade—is in general only influenced by the cascade geometry and the profile shape. With the CLA values of up to around 6 $\mu$m normally used in manufacture, the roughness is of minor importance. One can approximate the outlet flow angle by applying the sinus rule

$$\sin \beta_2 = e/t \tag{10}$$

where again $\beta_2$ is the outlet flow angle, $e$ the throat of the flow channel and $t$ the pitch of the blade cascade. From this it can be seen that if there is a manufacturing inaccuracy in the blade, the width of the flow channel is changed, and also as a result the throat and the outlet flow angle. In Fig. 6 the outlet flow angle $\beta_2$ is plotted against the contour tolerance $a/t$ referred to the pitch. The curves are based on measurements on two-dimensional cascades with various profiles [7, 8], which here represent blade cascades for a root, mean and tip section of a twisted rotor blade and the mean section of a guide blade appertaining to a turbine stage. Angle changes of around 4 deg can occur at a negative contour tolerance of only 1.25 percent, but this includes the influence of the simultaneous shortening of the chord length. A deviation from the specified chord length similarly influences the outlet flow angle. Fig. 7 shows the percentage change in the outlet flow angle plotted against the percentage shortening of the chord length according to cascade measurements. The curve rises sharply with increasing shortening of the chord length. The outlet flow angle is further affected by faulty stagger angles due to the twist error $\Delta\beta_s$. Here the change in the outlet flow angle is roughly equivalent to the twist error.

The loss coefficient and the outlet flow angle of the two-dimensional cascade also influence the characteristic magnitudes of a turbine namely the efficiency, the mass flow and the enthalpy drop. A precise calculation of the effects of surface roughness and manufacturing inaccuracies can be carried out by means of a three-dimensional flow computation [9]. Qualitatively this shows that a profile loss increased through surface roughness or through a thicker trailing edge diminishes the efficiency of a turbine.

Measurements on a turbine have shown that the drop in efficiency in the turbine is greater than in the corresponding plane cascades. For a constant isentropic enthalpy drop the mass flow decreases at lower efficiency; the same happens with a reduced outlet flow angle. With an increased outlet flow angle or at lower efficiency one must step up the mass flow if one is to achieve the same output of the turbine. In order to illustrate these effects quantitatively as well, the results are shown below of measurements which were performed on a multistage turbine [1, 10, 11].

## Results of Turbine Measurements

The effects of the contour tolerance (i.e., thinning or thickening of the profile) and the roughness of the blades on the efficiency,

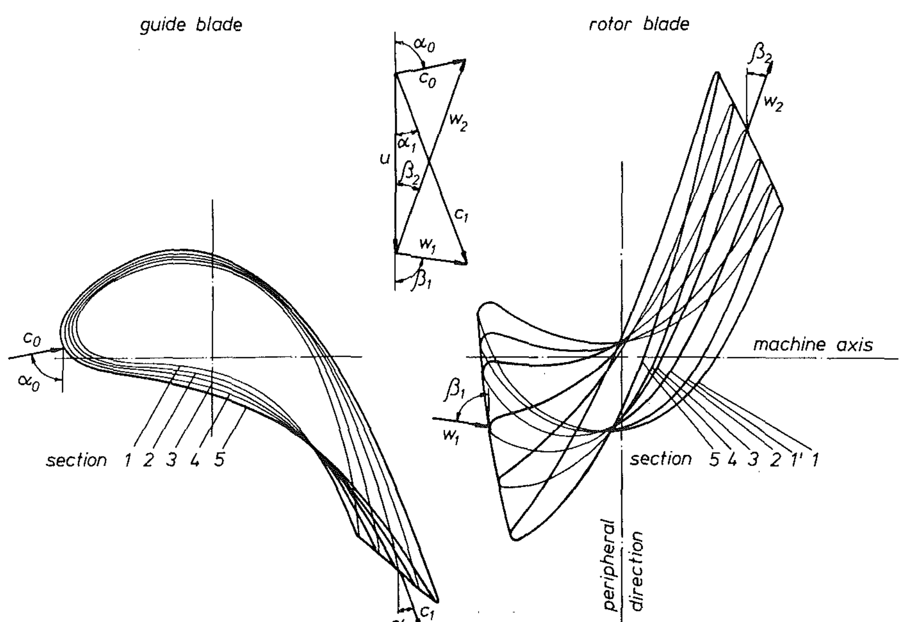


**Fig. 4 Superimposed profile sections of guide and rotor blade**

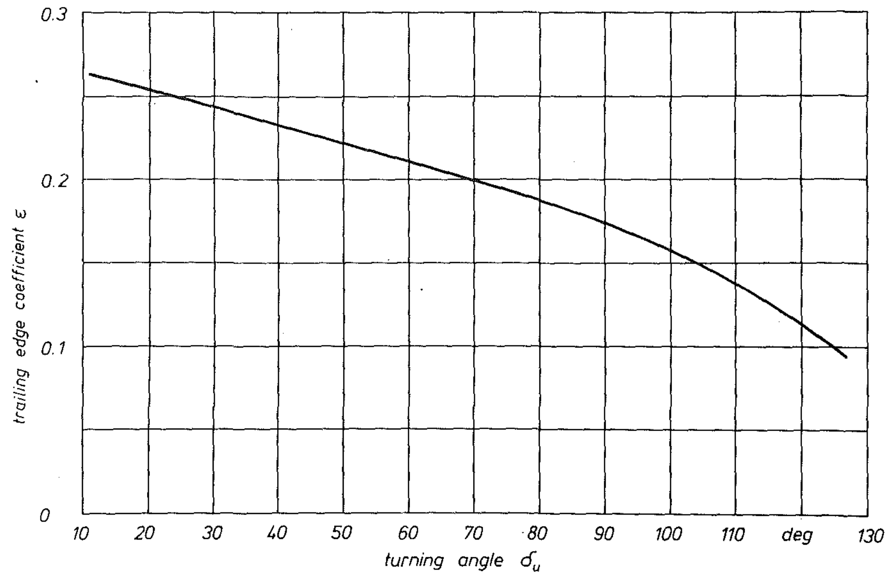


**Fig. 5 Trailing edge coefficient dependent on the turning angle**

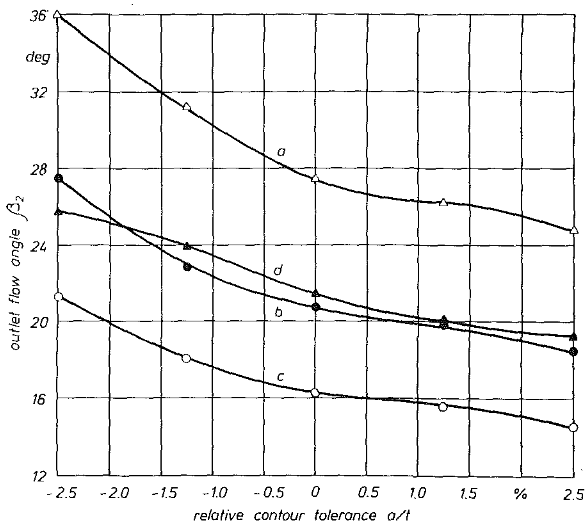


**Fig. 6 Outlet flow angle dependent on the contour tolerance: $a$ = rotor blade root section; $b$ = rotor blade mean section; $c$ = rotor blade tip section; $d$ = guide blade mean section**

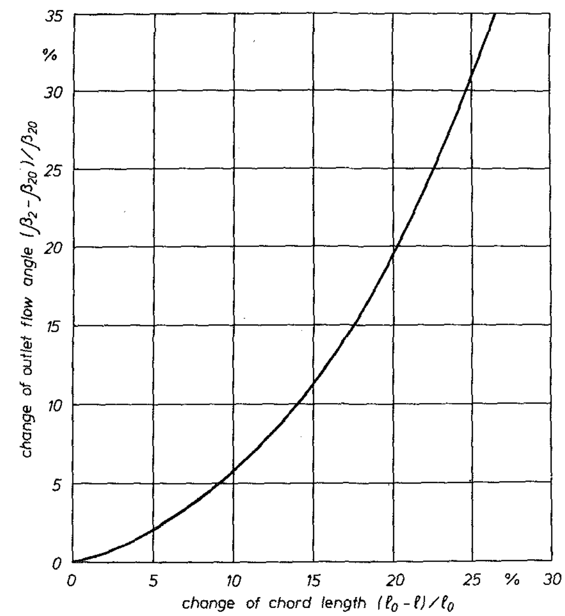


**Fig. 7 Outlet flow angle dependent on chord length: $\beta_2$ = outlet flow angle; $l$ = chord; subscript 0 = design**

the enthalpy drop and the mass flow were investigated on a 4-stage air turbine. For this purpose two turbine blading sets were manufactured. One blade set possesses for all four stages normal profiles in accordance with the design. In Fig. 4 the sections of the guide and rotor blades are shown superimposed on one another, the guide blades on the left, the rotor blades on the right. The blading is typical with 50 percent reaction in the middle section. The blades are twisted according to the potential vortex. The second set of blading with changed blades has, in the first two stages, thinned profiles thinner than the normal profile by a uniform 1 mm all the way round and vertical in relation to the profile contour and in the two final stages profiles 1 mm thicker all the way round. On one occasion the two blade sets were roughened by gluing on emery and measured in the turbine [11], and on another occasion, with the surface smooth, the influence of profile inaccuracies on the characteristic magnitudes of turbines was ascertained through a combination of thinned, normal, and thickened stages [1, 10]. With regard to the influence of the uniform contour tolerance, the next three Figs. 8–10 show the changes in mass flow, in enthalpy drop and in efficiency plotted against the relative contour tolerance. The pitch $t$ was chosen as the relative magnitude since the influence of the uniform contour tolerance is to a large extent dependent on the pitch. The design point $A$ was connected to each of the measuring points B by means of straight lines. The small letters indicate the change, i.e., where the tolerances occur. For curve $a$, a negative contour tolerance (thinning) was realized in the first stage of the 4-stage turbine; for curve $b$, stages 1 and 2 are thinned, while for $c$ stages 3 and 4 have positive contour tolerances (thickening); for curve $d$ only the fourth stage is thickened. In the turbine under investigation there is roughly the same enthalpy drop at the design point in each stage; one stage therefore represents 25 percent of the blading. For applying these results to other turbines it is suggested that the percentage of enthalpy drop in each stage be considered rather than the overall enthalpy drop.

The influence of the uniform contour tolerance on the mass flow $\dot{m}$ can be seen from Fig. 8. Plotted against the relative contour tolerance is the percentage change in mass flow $(\dot{m} - \dot{m}_0)/\dot{m}_0$. The

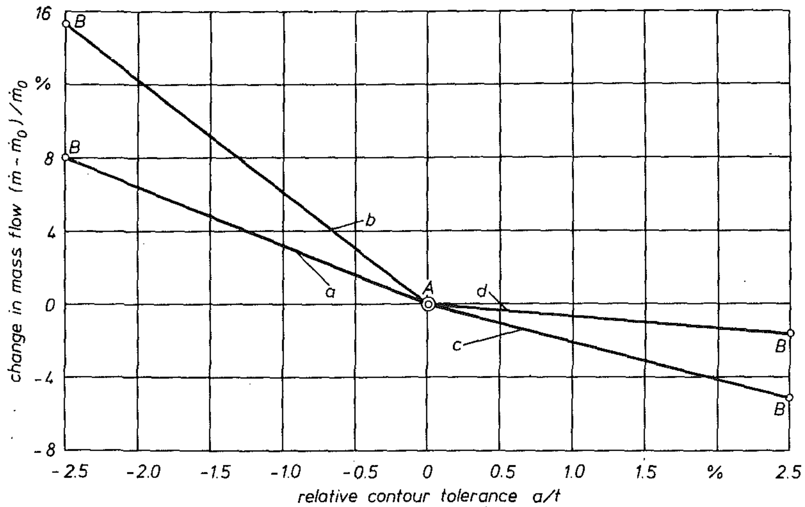


**Fig. 8** Mass flow dependent on the contour tolerance: $A$ = design point; $B$ = measuring points; $a$ = stage 1 (thinned); $b$ = stages 1 + 2 (thinned); $c$ = stages 3 + 4 (thickened); $d$ = stage 4 (thickened)

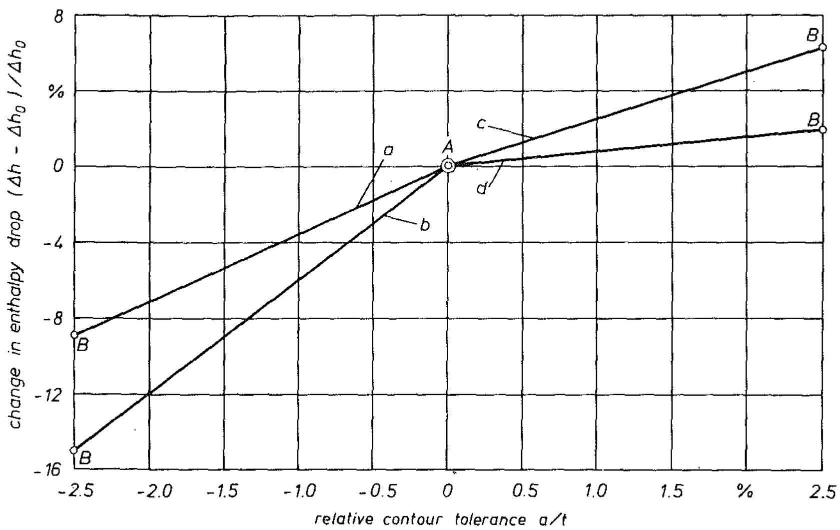


**Fig. 9** Enthalpy drop dependent on the contour tolerance; legend see Fig. 8

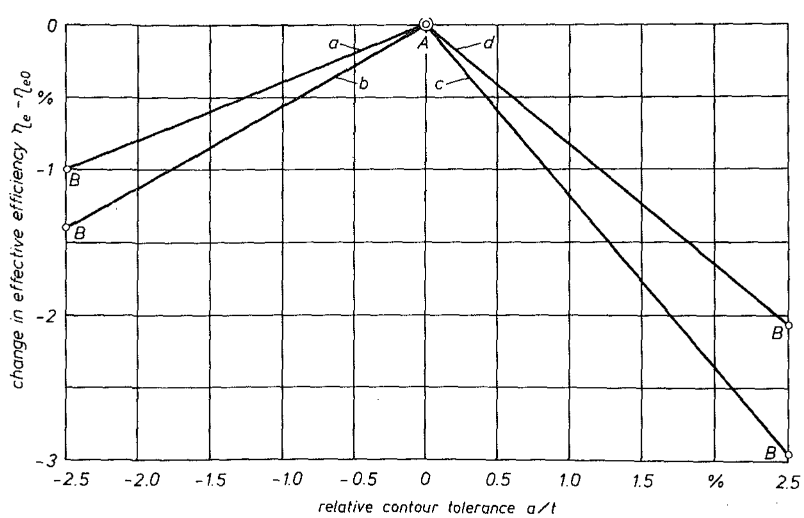


**Fig. 10** Efficiency dependent on the contour tolerance; legend see Fig. 8

*isentropic enthalpy drop* and the *speed* are kept *constant*. $\dot{m}_0$ is the rated mass flow. A uniform profile thinning in stage 1 of 0.5 percent (curve *a*) results in the mass flow increasing by 1.7 percent, whereas a uniform profile thickening in stage 4 of 0.5 percent (curve *d*) only brings about a 0.3 percent reduction in the mass flow. In the case of 0.5 percent thinning in stages 1 and 2 (curve *b*) the mass flow increases by 3 percent. A 0.5 percent thickening in stages 3 and 4 (curve *c*) reduces the flow by 1 percent. The main causes for this are the smaller outlet flow angles when the profiles are thicker and the larger angles when the profiles are thinner.

The reduction of the enthalpy drop $\Delta h$ of the turbine has the opposite effect to that of the mass flow. From Fig. 9, which shows the percentage change in effective enthalpy drop in the turbine plotted against the contour tolerance, it can be seen that a uniform 0.5 percent profile thinning in stage 1 (curve a) causes a 1.8 percent reduction in the enthalpy drop, as opposed to an increase of only 0.4 percent in the case of a 0.5 percent thickening (curve *d*) in stage 4. The curves shown are applicable for *constant mass flow* and *constant speed*. A 0.5 percent thinning in stages 1 and 2 (curve *b*) results in a 3.1 percent reduction in the enthalpy drop. A 0.5 percent thickening in stages 3 and 4 (curve *c*) increases the enthalpy drop by 1.2 percent. From the last two graphs it becomes clear that with regard to the mass flow and reduction in enthalpy drop a negative contour tolerance (thinning) has a greater effect than a positive one (thickening). With efficiency the situation is different.

The efficiency is impaired not so much by the outlet flow angle as by the additional losses due to trailing edges being thickened when profile thickening occurs (positive contour tolerance). Fig. 10 shows the influence of the uniform contour tolerance on the effective turbine efficiency $\eta_e$ at *rated speed* and *constant enthalpy drop*. It can be seen from the graph that a uniform profile thinning in stage 1 of 0.5 percent (curve *a*) effects a drop in efficiency of 0.2 percent, while a uniform profile thickening in stage 4 of 0.5 percent (curve *d*) results in a drop of 0.41 percent. For a 0.5 percent thinning in stages 1 and 2 (curve *b*) the drop in efficiency is 0.28 percent. A thickening of 0.5 percent in stages 3 and 4 (curve *c*) results in a drop of 0.59 percent.

The effect of roughness on the characteristic magnitudes of the turbine are shown in Figs. 11 and 12. Fig. 11 shows the efficiency referred to the efficiency at the nominal point, which is $\eta_{e0} = 94$ percent, plotted against the relative roughness $k_s/l$. The graph is valid for rated speed and constant isentropic enthalpy drop. The graph is the result of measurements which were carried out on a turbine with a rough surface pasted on. Since this pasting on made the profiles thicker, the portion attributable to this thickening was discounted, and the result was this curve which shows the efficiency of a turbine as a function of the surface roughness with no other influences. Up to a relative roughness $k_s/l = 0.2 \cdot 10^{-3}$ the efficiency remains constant, since at a Reynolds number of $5 \cdot 10^5$ the blading must be viewed up to this limit as hydraulically smooth. The efficiency drops after this and at $k_s/l = 1 \cdot 10^{-3}$ it is only 0.95, referred to the nominal value. The drop in efficiency with increasing roughness continues almost linearly, only from $k_s/l = 9 \cdot 10^{-3}$ onwards does it begin to fall sharply again. This is due to the fact that, because of the great losses the mass flow at the constant isentropic enthalpy drop is similarly starkly reduced and the velocity triangles are distorted, which causes unfavorable flow to the profiles and consequently leads to increased flow losses.

How the mass flow changes with the relative roughness of the blade surface is shown in Fig. 12. The speed and the isentropic enthalpy drop are again constant. The effects of thickening due to the pasted-on roughness on the loss coefficient and the outlet flow angle, and hence on the mass flow, were eliminated, so that this curve shows the mass flow as affected by a mere fall-off in surface quality without any other influences. At a relative roughness $k_s/l \approx 1 \cdot 10^{-3}$ the mass flow, at constant isentropic drop, falls by 1 percent to 0.99, at $k_s/l = 8 \cdot 10^{-3}$ by 3 percent to 0.97. In the range of $k_s/l$ from $(0.2 \text{ to } 2) \cdot 10^{-3}$ the curve falls relatively sharply. This proceeds roughly parallel with the decrease in efficiency in this range. Up to $k_s/l = 8 \cdot 10^{-3}$ the decrease in efficiency is roughly a linear function of the roughness, above that the drop is sharp again. For $k_s/l = 10.6 \cdot 10^{-3}$ the resulting mass flow is 6 percent smaller than with a smooth blading. The renewed sharper drop in mass flow for this roughness range is the result of the change in the velocity triangles and thus in the inlet flow conditions of the blade rows; it is thus caused by the same factors which bring about the decline in efficiency.

The output of a turbine is determined by the mass flow and the effective enthalpy drop according to the equation

$$P = \dot{m}\,\eta_e \cdot \Delta h_s. \tag{11}$$

At constant isentropic drop $\Delta h_s$, with regard to the power output the effects of the roughness on the efficiency and the mass flow in Figs. 11 and 12 are to be multiplied. Hence a relative roughness $k_s/l = 1 \cdot 10^{-3}$ results, if one considers purely the influence of the roughness, in an output of only $0.99 \cdot 0.95 = 94$ percent. For $k_s/l = 10.6 \cdot 10^{-3}$ the relative efficiency is 0.88 and the mass flow 0.94, which represents a loss in power output of 17.3 percent. From these figures it becomes clear that a considerable part of a turbine's output can be lost through roughness.

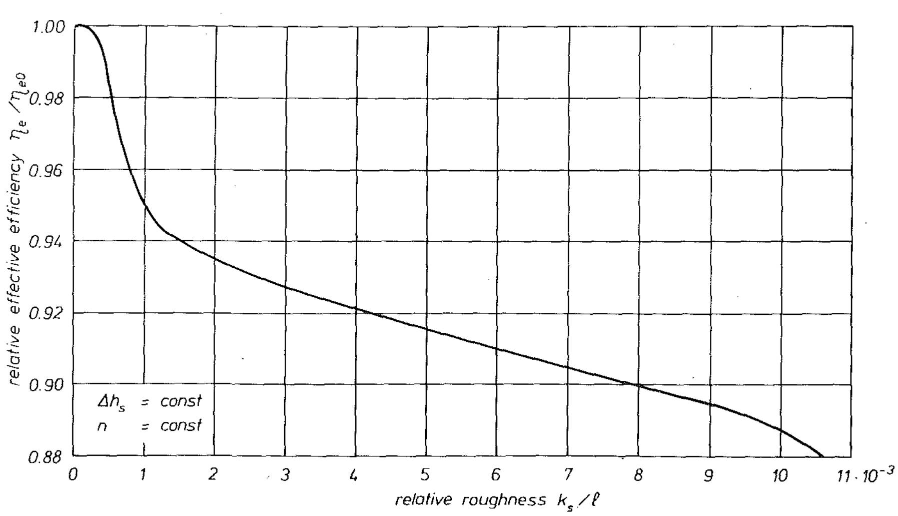


**Fig. 11 Efficiency dependent on surface roughness: $\Delta h_s$ = isentropic enthalpy drop; $n$ = speed**

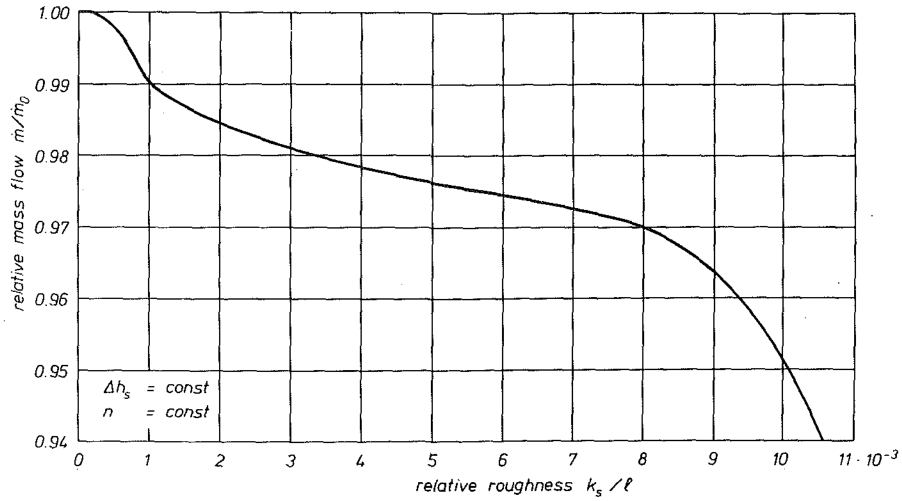


**Fig. 12 Mass flow dependent on surface roughness; legend see Fig. 11**

## Example

Using an example we are now going to give a brief illustration of how to cope with these requirements. The example chosen is the guide blade of a gas turbine. The profile is similar to the guide blade profile, shown in Fig. 4, of the turbine under investigation. The blade is cylindrical and is manufactured by precision casting. The chord length is 70 mm and the pitch ratio 0.75. The blades are due to be installed in a gas turbine of approx. 55 MW. The Reynolds number in the turbine is at normal operation, around $7 \cdot 10^5$.

First to be considered is the roughness. With the given chord length and Reynolds number we obtain a permissible sand roughness of $k_{s\ \mathrm{adm}} = 10\ \mu\mathrm{m}$ (equation (5)). If one uses the structure of a milled surface as a basis then from equation (4) we obtain a permissible CLA value of 5.6 $\mu$m, which is easy to realize.

If during the manufacture of a blade the chord length is changed by 0.6 mm, then the change in chord length by this amount (about 0.8 percent) according to Fig. 7 effects a 0.06 deg increase in the outlet flow angle, which is just as insignificant as a concomitant drop in efficiency.

A twist error of 0.5 deg is already critical in an outlet flow angle of 25 deg and must be considered as the absolute upper limit, since an error of this order in the outlet flow angle *of all the guide wheels* brings about a deviation of almost 2 percent from the design output. In the case of a contour tolerance of 0.25 mm, which corresponds to 0.48 percent of the pitch, the effects on the turbine can be very great. In the most unfavorable case the deviations in the channel cross section can be ±0.5 mm, i.e., with a *uniform* thickening of the profiles by 0.25 mm the channel becomes 0.5 mm narrower and in the case of thinning, 0.5 mm wider. According to equation (10) the channel cross section becoming 0.5 mm narrower corresponds to an approx. 0.5 deg decrease in the outlet flow angle. In the case of the channel widening by 0.5 mm the outlet flow angle is around 0.5 deg larger than according to the design. Calculations of the flow revealed that an increase in the outlet flow angle—i.e., in the case of thinning—in *all the guide wheels* of 0.5 deg leads to a 1 MW drop in the power output of the turbine, which in the case of an output of 55 MW is almost 2 percent. The efficiency falls by 0.25 percent when one uses the measuring results of the air turbine (Fig. 10) and converts them to this example. In the case of thickening (positive tolerance) the efficiency drops by 0.5 percent. In the light of these figures a smaller contour tolerance would appear to be desirable for this example.

The example shows that when the Reynolds numbers are not too high the flow requirements imposed on blade manufacture with regard to *surface roughness* are not too difficult to fulfill. With *tolerances* the effects can be more far-reaching. The chord length tolerance is of little importance here. The twist errors and the contour tolerance, which can alter the outlet flow angle considerably, are of the greatest influence for the example illustrated and the ones chosen should be of appropriate accuracy.

## Summary

The influence of uniform manufacturing tolerances and of surface roughness of the blades on the characteristic magnitudes of a plane cascade have been described and verified by measurement results. Tests similarly enabled one to determine the influences on mass flow, efficiency and output which can arise in a turbine. The example of a gas turbine blade shows what the effects of tolerances and surface roughness can be.

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