EFFECT OF TURBINE DESIGN PARAMETERS ON LEAKAGE AND OUTLET LOSSES

M.A. Awwad

Mechanical Engineering Department, Faculty of Engineering,
Alexandria University, Alexandria Egypt.

ABSTRACT

A mathematical model is introduced for a staggered labyrinth sealing system which is commonly used in turbines in order to estimate the leakage losses, and to investigate the effect of different design parameters on these losses. A new expression is also deduced for the outlet loss coefficient and the stage loss coefficient due to leakage, based on the sum of the mixing losses, power loss due to mass flow rate reduction through the moving blades, and the change of absolute flow direction at moving blade entrance. The results showed that these losses depend on the stage degree of reaction and the load factor.

NOMENCLATURE

A_c	Clearance area	m^2
C	Absolute Velocity	m/s
D	Diameter	m.
di	Enthalpy drop	J/Kg
h	Blade height	m.
L_c	Leakage Loss Coefficient	...
L_o	Outlet Loss Coefficient	....
Lu	Specific power	Watt/kg.
$\dot{m}$	Mass flow rate	Kg/s
p	Static pressure	N/m^2
r	Degree of reaction	...
S	Rdial distance	m
T	Temperature	K°
v	Specific volume	m^3/kg
Z	Number of labyrinth	...
α	Absolute angle	degree
λ	Stage load factor ($Lu^2/2$)	...
μ	Labyrinth coefficient of discharge	...
ϕ	Mass flow factor (Cax/u)	...
ρ	Density	Kg/m^3

SUBSCRIPTS

1 Fixed blade inlet
2 Moving blade inlet
3 Moving blade outlet
0 After mixing
ax In axial direction
t Total
u In peripheral direction
′ For fixed blades
″ For moving baldes

INTRODUCTION

Despite the considerable progress in the development of advanced sealing techniques such as gas-film seals, labyrinths will continue to be the most widely used sealing element in jet engines as well as in stationary turbomachines. Simplicity and compact design, reliability, and operational characteristics especially high temperature resistance are some of its major advantages.

Many attempts have been made for describing the leakage losses through labyrinth seals [1], [2], and [3]. Reasonably good results are generally obtained using empirical relations. However, the new generation of jet engines, with drastically reduced fuel consumption require more accurate predictions, since the reduction of losses leads directly to an improvement of the engine efficiency. Furthermore compressor characteristics and pumping are dependent on the sealing. Due to experimental requirements, leakage measurements are usually carried out with scaled-up models. This is necessary to reduce errors in determining the geometry, especially the radial clearance. With an engine clearance of 0,1 mm, for example, a deviation of 0,01 mm., would lead to an error of 10%. The scaling effect on leakage losses was studied by "Witting", [4].

In cases of high pressure ratio compressors, since the specific volume of gas is reduced considerably at exit,the blade height is relatively small and the compressor performance will be inherently deteriorated by leakage through the clearance. The same condition also occurs at the inlet of a turbine with high pressure ratio.

According to the literature, the variation of compressor performance due to tip clearance is expressed in different ways, and the empirical coefficients of these equations vary widely depending upon the blade geometry, and the operating conditions. This means that the physics of pressure loss due to tip clearance is not yet well understood. Seeno. [5] has studied the effect of tip leakage on the pressure losses and flow field distortion induced by tip clearance of centrifugal and axial compressors. His measurements of the relative velocity distributions between and behind the blades revealed that leakage flow rolled up forming a vortex between the blades. Accordingly, the velocity distribution was distorted within a layer of thickness about ten times the tip clearance.

In order to understand the behaviour of blades in the distorted casing boundary layer due to leakage, a considerable effort has been made for measuring the details of flow between the blades which is influenced by the tip clearance [6,9]. By rotating various types of probes together with the impeller, the relative velocity distributions between and behind the blades could be measured. Lakshminarayana [8] performed measurements in the flow field in the annulus wall and tip region of a compressor rotor, using a triaxial hot-wire probe rotating with the rotor. The flow was surveyed across the entire passage at five axial locations, and at six radial locations inside the passage. The measurements indicated that the leakage flow starts beyond a quarter-chord and tends to roll up farther away from the suction surface than that observed in cascades. Substantial velocities and radial inward velocities are observed in this region. The annulus wall boundary layer is well behaved up to half a chord, beyond which interactions with the leakage flow produce complex profiles.

The effect of tip clearance on the casing wall boundary layer was investigated by Hunter [10]. Detailed measurements were made of the casing wall boundary layer development across a large-scale, low-speed axial compressor rotor blade row. The downstream boundary layer was found to thicken as the rotor loading and blade-end clearance were increased, with fluid tending to accumulate towards the pressure side of the passage, and the tip clearance had a deleterious effect upon the performance of the compressor.

The effect of tip-clearance on the performance of nozzle blades of radial turbines has also been investigated [11,12]. They found that the effective turning angle of a nozzle becomes smaller and the pressure loss becomes larger as the tip clearance of a turbine-nozzle increases. In another work [13] Wagner demonstrated that the main contributions to total pressure loss, blockage, and the distortion of the static pressure field are due to tip leakage and the hub corner stall. An experimental program was conducted [14] in a highly loaded, single stage, low speed research compressor that featured variations, in tip clearances, shroud wall roughness, and stage loading level. The results showed that tip clearances and stage loading levels exerted a very strong influence on casing boundary layer growth.

The two-parts paper [15,16] outlines a new methodology for predecting and minimizing tip flows, and focuses on the control of tip leakage through minimization of the discharge coefficient to control the normal leakage flow component. Minimization of the discharge coefficient was achieved through viscous analysis and was supported by discharge-rig testing. The analysis for the cross-flow used a stream function vorticity formulation. Support testing was conducted with a water table discharge rig in which tip-coolant discharge could also be simulated. Experimental and numerical tip-leakage results are obtained, based on a discharge coefficient parameter for five different tip configurations.

The pressure loss based on the tip clearance was also studied by Senoo and Ishida [17].They found that this pressure loss consists of the pressure loss induced by the leakage flow through the clearance and the pressure loss for supporting fluid against the pressure gradient in the channels and in the thin annular clearance space between the shroud and the impeller. Investigatins of a tip clearance cascade in a water analogy rig was carried out by Graham [18], he studied in his work the tip clearance flow region of high pressure axial turbine blades for small gas turbine engines in a water flow cascade. He found that the unloading is the major feature of the pressure field at both tip and mid span, and is intimately connected with scraping effects and the behaviour of the clearance vortex.

The blade to blade variation of relative stagnation pressure losses in the tip region inside the rotor of a single-stage axial flow compressor was investigated by Lakshminarayana [19]. Performance testing and detailed flow measurements were made in an axial compressor rotor with various tip clearances [20].The experiments were conducted on the condition of the same incidence angle at mid span. Thus, the effect of tip clearance distinguished from that of incidence angle was investigated on the overall performance, workdone factor, blockage factor, and increases in displacement, momentum, and blade-force-deficit thickness of the casing wall boundary layer.

The outlet losses are one of the most important losses in turbomachines, since the absolute Mach number at the exit plane of the rotor is in the region of 0,3 to 0,4 [21]. Thus, there are significant outlet losses . These losses can be reduced by mounting a diffuser at the outlet of the turbine, but this is associated with other losses and thus, not all outlet kinetic energy can be converted into potential energy.

In the present work a new method is introduced to claculate the leakage and outlet losses in terms of stage design parameters. The method of calculating the leakage losses is based on the calcualtion of the losses in the total pressure caused by mixing of the formed eddies, in addition to the losses due to decreasing the mass flowing through the rotor, and the change of absolute inlet angle due to leaking mass.

Description of Leakage Model

In any one of gas turbine stages the major part of the gas passes through the blades, while the leaking mass passes through the annulus of the tip clearance, being accelerated in the meridional direction, conserving the inlet angular momentum, and the two parts of the flow mix together again at the blade exit. Figure (1) shows a diagramatic sketch for this model with labyrinth sealing system. The relatively large space between two adjacent orfices decelerates the flow through the generated eddies and the kinetic energy is converted into heat energy. This process is represented on the i-s diagram Figure (2)] by a Fanno line.

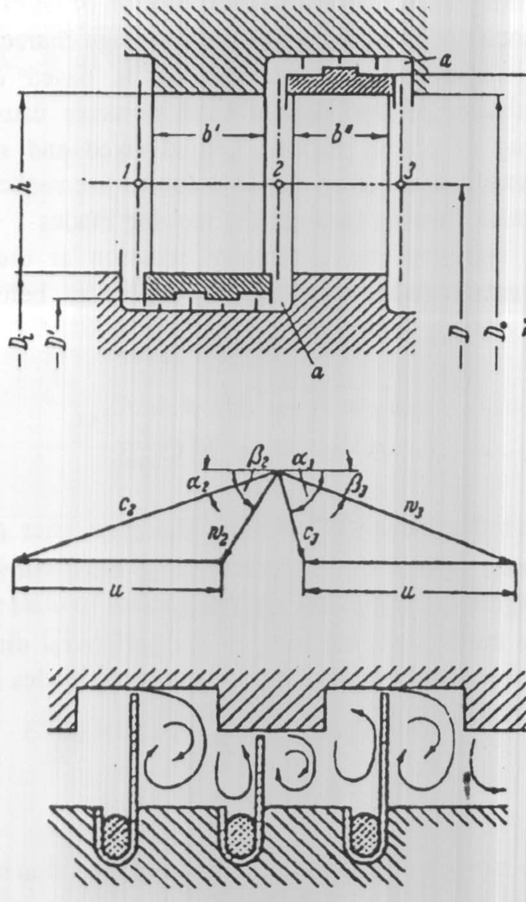

Figure 1. Axial turbine stage with staggered labyrinth.

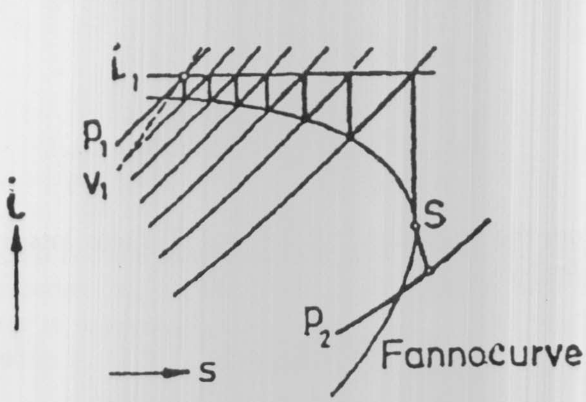

Figure 2. Expansion process through labyrinths represented on the i-s diagram.

Determination of the Leakage-Loss Coefficient "L_c"

In the present work a new expression of leakage loss is concluded in terms of the turbine stage characteristic design parameters. This method is based on the calculation of the loss in total pressure caused by mixing of eddies formed behind fixed and moving blades, in addition to the loss due to the reduction in the mass flowing through the moving blades

By applying the continuity equation at the axial clearance between fixed and moving blades, before and after mixing, then :

$$(\dot{m}-\dot{m}') = \rho . D_i . \pi . h . C_{2ax} \quad (1)$$

$$\dot{m} = \rho . D_i . \pi . h . C_{2axo} \quad (2)$$

Where the index "O" denotes the state after mixing process, while $\dot{m}'$ is the leaking mass flow rate through the fixed blade labyrinth.

The momentum equation in the peripheral direction before and after mixing behind the fixed blades can be written as:

$$(\dot{m}-\dot{m}') . C_{2u} = \dot{m} . C_{2uo} \quad (3)$$

The momentum equation in the axial direction may be written as :

$$(\dot{m}-\dot{m}') . C_{2ax} + P_2 . \pi . D . h = \dot{m} . C_{2axo} + P_{20} . \pi . D . h \quad (4)$$

(The previous equations are applied also at point 3)
The losses due to mixing process are given by:

$$\left.\begin{aligned} \Delta P_{t2} &= P_2 + \frac{\rho}{2} C_2^2 - P_{20} - \frac{\rho}{2} C_{20}^2 \\ \Delta P_{t3} &= P_3 + \frac{\rho}{2} C_3^2 - P_{30} - \frac{\rho}{2} C_{30}^2 \end{aligned}\right\} \quad (5)$$

From the equations (1,2,3,4 and 5) these losses are given by:

$$\left.\begin{aligned} \Delta P_{t2} &= 2\left(\frac{\dot{m}'}{\dot{m}}\right) . \left(\frac{\rho}{2}\right) . C_2^2 , \\ \Delta P_{t3} &= 2\left(\frac{\dot{m}''}{\dot{m}}\right) . \left(\frac{\rho}{2}\right) . C_3^2 , \end{aligned}\right\} \quad (6)$$

The flow angles are distorted due to leakage. The flow direction after mixing, α_{20}, α_{30} can be calculated from the above equations as :

$$\left.\begin{aligned} \frac{\tan \alpha_{20}}{\tan \alpha_2} &= \left(1 + 2\frac{\dot{m}'}{\dot{m}}\right) \\ \frac{\tan \alpha_{30}}{\tan \alpha_3} &= \left(1 + 2\frac{\dot{m}''}{\dot{m}}\right) \end{aligned}\right\} \quad (7)$$

The distortion of flow angles $\Delta \alpha_2$ and $\Delta\alpha_3$ can then be calculated as:

$$\Delta \alpha_2 = (\alpha_{20} - \alpha_2), \; \Delta\alpha_3 = (\alpha_{30} - \alpha_3)$$

The leakage stage loss coefficient "L_c" may be defined as:

$$L_c = \frac{\Delta P_{t2} + \Delta P_{t3}}{\rho . L_u} + \frac{\dot{m}''}{\dot{m}} \quad (8)$$

Substituting from eq.(6) into eq.(8) the loss coefficient "L_c" can be obtained in terms of $\frac{\dot{m}'}{\dot{m}}, \frac{\dot{m}''}{\dot{m}}$

Determination of Leaking Mass Flow Rate:

There are two known methods for calculating the leaking mass flow rate. The first one depends on constructing the fanno line for a certain labyrinth inlet conditions (P_1, T_1, C_1), by assuming a starting value of mass flow rate and then calculating the corresponding static back pressure "P_2". If this pressure is not in coincidence with the actual back pressure, another value of mass flow rate is to be assumed.

The iteration must be repeated until the calculated pressure becomes equal to the assumed pressure. This method is time expensive and not adequate for determination of "L_c". The second method is suggested by "STODOLA" (as explained in [1]), who treated this problem as an isentropic expansion in series of nozzles with a small pressure ratio. STODOLA found that the leaking mass can be expressed as:

$\dot{m}_c = \mu . A_c . \sqrt{\frac{P_1}{v_1}} . \sqrt{1/(Z-1+\phi^2 max)}$ where μ is equal to the discharge coefficient of tip clearance area A_c, Z is the number of labyrinths and ϕ_{max} $= \left[\frac{2}{\gamma+1}\right]^{\frac{1}{\gamma-1}} . \sqrt{\frac{2\gamma}{\gamma+1}}$ and is known as maximum mass flow function.

Following the same steps suggested by STODOLA we can get $\dot{m}_c$ as a function of: $\frac{P_2}{P_1}$, Z.

The outlet velocity from each labyrinth "C" $= \sqrt{2\Delta i}$ where Δi is equal to the enthalpy drop in one labyrinth, and because the pressure drop across one labyrinth is small, then $\Delta i = v.\Delta P$, and considering the flow through the labyrinth as a throttling process then $P_1 v_1 = P.V =$ Const., and then:

$(\frac{\dot{m}_c}{\mu . A_c})^2 = \frac{P.\Delta P}{P_1 . v_1}$, and for a number of labyrinths equal to "Z" (for the fixed blades)and, when ΔP tends to zero then:

$$\int_1^2 -P.dP = \frac{z'}{2} P_1 . v_1 (\frac{\dot{m}'}{\mu . A_c})^2$$

$$\dot{m}'_c = \mu' . A'_c . \sqrt{\frac{P_1}{v_1}} . \sqrt{\frac{1}{Z'}[1-(\frac{P_2}{P_1})^2]} \qquad (9)$$

The outlet velocity from the last labyrinth C′ can be expressed as:

$$C' = \sqrt{\frac{v_1}{Z'}(\frac{P_1^2 - P_2^2}{P_1})} \cong \sqrt{\frac{2v_1}{Z'}(P_1 - P_2)} \qquad (10)$$

(Since the stage pressure drop in a multi stage axial turbine (P_1-P_3) is small). By comparing C′ with the outlet velocity of an orifice having the same pressure drop, it can be found that $C' = \frac{1}{\sqrt{Z'}}$ of the velocity of this orifice. By using the definitions of load parameter "λ" flow parameter ϕ, and the stage degree of reaction "r" and for a small stage pressure drop, then:

$$L_u = di = v.dp, = v. (P_1 - P_3), \lambda = \frac{L_u}{u^2/2}$$

$$\lambda = \frac{P_1 - P_3}{\rho . u^2/2}, \phi^2 = \frac{C^2 ax}{u^2}, r = (\frac{P_2 - P_3}{P_1 - P_3}), \text{then}$$

$$\left.\begin{aligned} \frac{C'}{C_{2ax}} &= \frac{1}{\phi}\frac{\lambda}{2}\sqrt{\frac{\lambda}{Z'}(1-r_i)} \\ \frac{C''}{C_2 ax} &= \frac{1}{\phi}\sqrt{\frac{\lambda}{Z''}(1-r_o)} \end{aligned}\right\} \qquad (11)$$

where r_i,r_o are the degree of reactions at the inner and outer diameters of the blades, respectively.

The ratio of leaking masses can then be calculated as:

$$\left.\begin{aligned} (\frac{\dot{m}'}{\dot{m}}) &= \frac{\mu'}{\sqrt{Z'}} . \frac{D'}{D_m} . \frac{S'}{h} . \frac{1}{\phi} . \sqrt{\lambda(1-r_i)} \\ (\frac{\dot{m}''}{\dot{m}}) &= \frac{\mu''}{\sqrt{Z''}} . \frac{D''}{D_m} . \frac{S''}{h} . \frac{1}{\phi} . \sqrt{\lambda . r_o} \end{aligned}\right\} \qquad (12)$$

where S′ and S″ are the radial clearance distances,μ' and μ'' are the coefficient of the discharge for fixed and moving blades, respectively.

By substituting from equations (6),(12) into equation (8), the leakage loss coefficient "L_c" is obtained as:

$$L_c = 2 \frac{\mu'}{\sqrt{Z}} . \frac{D'}{D_m} . \frac{S'}{h} . \frac{\phi}{\lambda} . \frac{1}{\sin^2\alpha_2} . \sqrt{\lambda(1-r_i)} +$$

$$2 \frac{\mu''}{\sqrt{Z''}} \frac{D''}{D_m} \frac{S''}{h} \sqrt{\lambda . r_o} \left[\frac{\phi^2}{\lambda} . \frac{1}{Sin^2\alpha_3} + \frac{1}{2}\right] \qquad (13)$$

Since it is usual in turbine design to assume the degree of reaction "r_m" at the blade mean diameter then it is better to express "L_c" in terms of "r_m". The relation between r_i, r_o & r_m is found to be:

$$(1\text{-}r_i) = \frac{(1-r_m)}{(1-h/D_m)^2}, r_o = 1 - \frac{1-r_m}{(1+h/D_m)^2}$$

Substituting these values in eq. (13) and assuming that $\mu' = \mu'' = \mu$, $Z = Z' = Z'' = Z, = S' = S'' = S$, $\frac{D'}{D_i} = \frac{D''}{D_o} = 1$

Therefore

$$L_o = \frac{2\mu}{\sqrt{Z}} \cdot \frac{S}{h} \cdot \frac{\phi}{\sqrt{\lambda}} \left\{ \frac{1}{\mathrm{Sin}^2 . \alpha_2} \cdot \sqrt{1-r_m} + \left(\frac{1}{\mathrm{Sin}^2 \alpha_3} + \frac{\lambda}{2\phi^2} \right) \cdot \sqrt{(1+h/D_m)^2 - 1 + r_m} \right\} \quad (14)$$

and

$$\left. \begin{aligned} \frac{\dot{m}'}{\dot{m}} &= \frac{\mu}{\sqrt{Z}} \cdot \frac{S}{h} \cdot \frac{1}{\phi} \cdot \sqrt{\lambda(1-r_m)} \\ \frac{\dot{m}''}{\dot{m}} &= \frac{\mu}{\sqrt{Z}} \cdot \frac{S}{h} \cdot \frac{1}{\phi} \cdot \sqrt{\lambda\{(1+h/D_m)^2 - 1 + r_m\}} \end{aligned} \right\} \quad (15)$$

The relation between α_2 , α_3 is found as:

$$\frac{\tan \alpha_3}{\tan \alpha_2} = \frac{\lambda/4 + 1 - r_m}{\lambda/4 - 1 + r_m} \quad (16)$$

The value of "α_2" is usually chosen for best efficiency and ranges between 18° → 20° [3]. The value of: $\frac{S}{D_m}$ can be taken after Traupel [3] as 0,002 then $\frac{S}{h} = \frac{0,,002}{h/D_m}$ and the value of "μ" can be assumed = 0,8 [1].

For the sake of investigations, taking: α_2 = 20°, Z= 8, h/D_m = 0,15 then different graphs can be constructed which reveal the effect of the turbine design parameters on the leakage losses.

RESULTS, DISCUSSION AND CONCLUSIONS:

Equations (14) and (15) are represented in Figure (3), with the degree of reaction "r" as the varying parameter and the workdone factor "λ" as the independent parameter .

We notice that the ratio $\frac{\dot{m}'}{\dot{m}}$ always increases with "r", starting with the minimum value of $\frac{\dot{m}'}{\dot{m}}$ and tends to reach a maximum value which is nearly equal to that of all other workdone factor, but at a greater degree o reaction. For "λ" = 2 the ratio $\frac{\dot{m}'}{\dot{m}}$ increase gradually with "r", and reaches its maximum value a r = 0,6 and after that begins to decrease with "r" bu with a greater negative gradient than its increasin gradient. For "λ"= 3, $\frac{\dot{m}'}{\dot{m}}$ increases slowly till i reaches its maximum at r = 0,3 and after that i decreases slowly in a narrow region of "r", after whic it begins to decrease more rapidly than at "λ" = 2 For "λ" = 4,5,6 the fall in the value of $\frac{\dot{m}'}{\dot{m}}$ with "r starts with nearly the same maximum value at r=zero The value of $\frac{\dot{m}'}{\dot{m}}$ tends to zero as "r" approache unity, eq. (14), while the vlaue $\frac{\dot{m}''}{\dot{m}}$ has a definit value according to eq. (15) for r=1. The ratio $\frac{\dot{m}''}{\dot{m}}$ increases rapidly with "r" and tends to infinite whe "r" tends to unity, Figure (3), but the value of $\frac{\dot{m}''}{\dot{m}}$ has a certain value eq. (15).

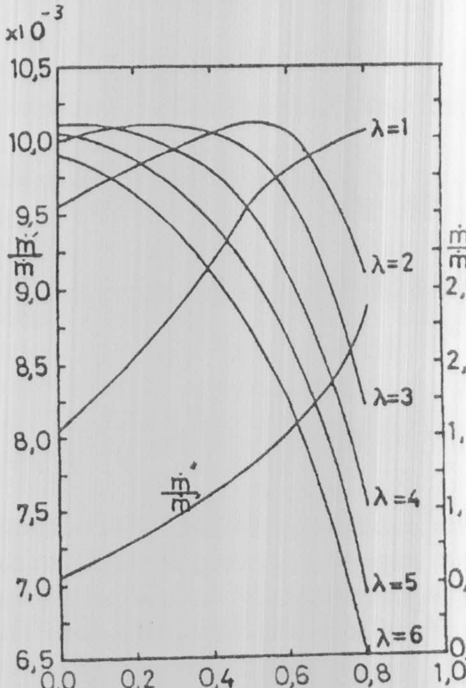

Figure 3. Effect of degree of reaction on the leaking mass ratio.

By representing $\frac{\dot{m}'}{\dot{m}}$ against the workdone factor "λ" and with the degree of reaction "r" as a separate parameter, Figure (4), it is noticed that for all degrees of reaction except that for r=0,8, the ratio $\frac{\dot{m}'}{\dot{m}}$ increases with "λ" to almost the same maximum value and after that it begins to decrease. The corresponding value of "λ" for maximum $\frac{\dot{m}'}{\dot{m}}$ decreases as "r" increases.

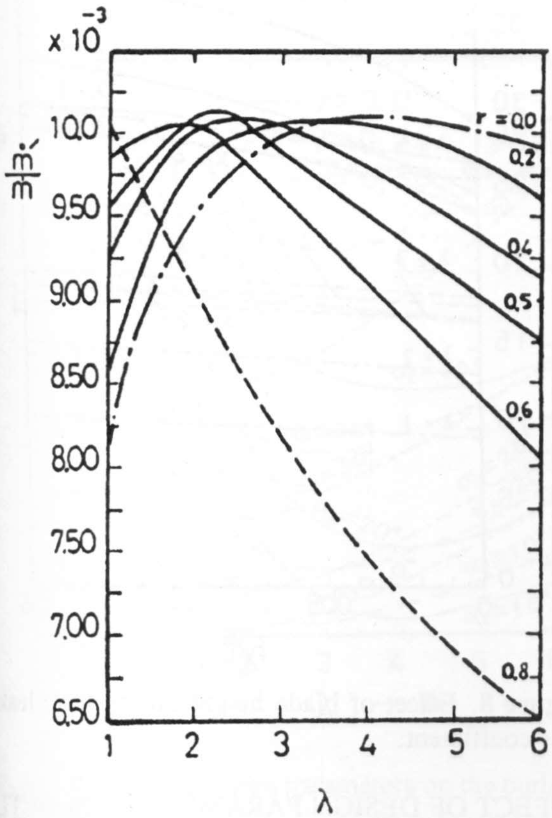

Figure 4. Effect of load factor on the leaking mass ratio.

For the degree of reaction r = 0,8, $\frac{\dot{m}'}{\dot{m}}$ always decreases with "λ" in its positive direction. It is expected that its deflection point lies in the negative region of "λ", which represents a compressor stage instead of a turbine, Figure (4).

Figure (5) represents the effect of degree of reaction "r" on the stage leakage loss coefficient "L_c", for different values of workdone factor "λ". The relation between the absolute outlet angle "α_3", and the flow ratio "ϕ" are also represented in the same Figure. It is noticed that "L_c" decreases firstly for all values of "λ", and after a certain minimum value it begins to increase. This minimum value increases with the increase of "λ", and also the value of "r" corresponding to the minimum value of "L_c" decreases as "λ" increases.

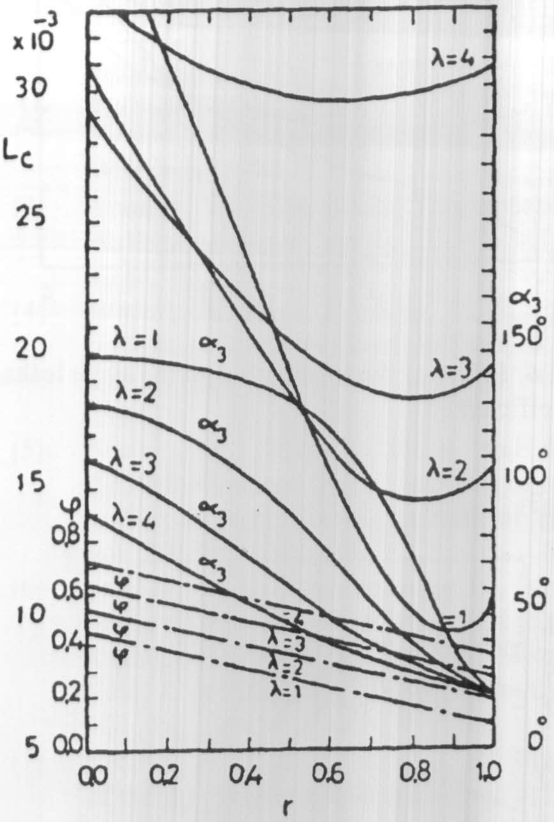

Figure 5. Effect of degree of reaction on the stage leakage loss coefficient.

The relation between "λ" and "L_c" is represented in Figure (6), with the degree of reaction as a separate parameter. It can be noticed that at zero degree of reaction the loss coefficient "L_c" decreases at first to a minimum value (at λ=2.5) & after that begins to increase. The relation for r = 0,2 shows the same tendency but is more flat. For r = 0,5 and 1 "L_c" increases always with "λ".The rise for r = 1 is steeper than that of r = 0.5.

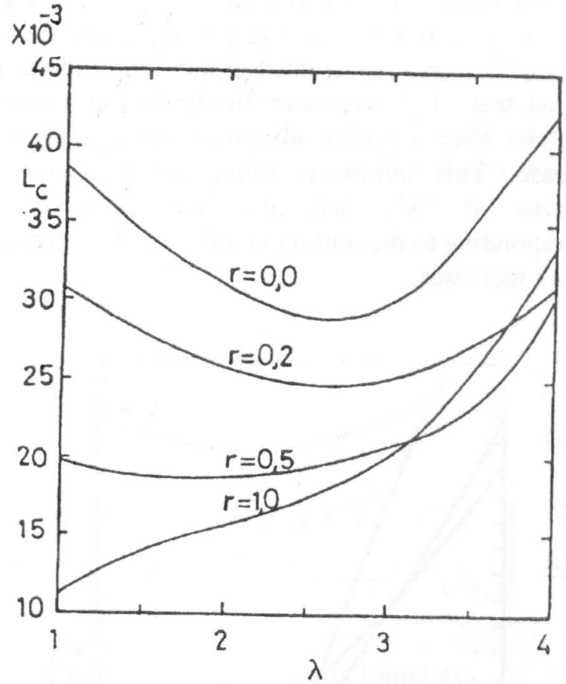

Figure 6. Effect of the load factor on the stage leakage loss coefficient.

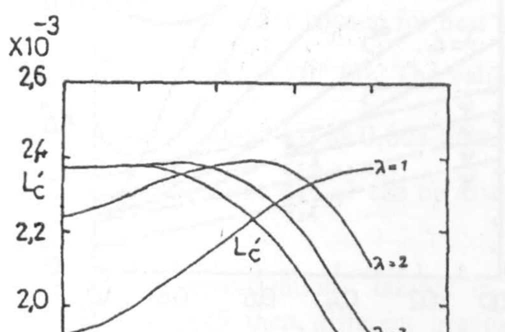

Figure 7. Effect of degree of reaction on flow deterioration and its leakage loss coefficient.

The additional losses due to flow deterioration caused by leakage "L_c'" is illustrated in Figure (7). The trend of the curves is similar to that of $\frac{\dot{m}'}{\dot{m}}$ represented in Figure (3), but with lower corresponding magnitudes. The change of inlet angle "$\Delta\alpha_2$" due to leakage is also illustrated in this figure, and it shows almost the same trend. The leakage loss coefficient varies linearly with h/D m for degrees of reactions of 0.5 and 1 while it is not the case for r = 0, 0 . Figure (8)

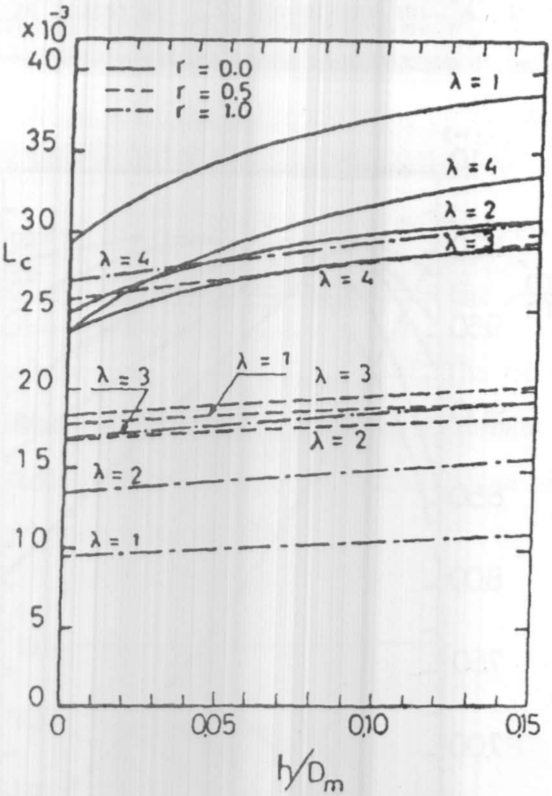

Figure 8. Effect of blade height on the stage leakage loss coefficient.

EFFECT OF DESIGN PARAMETERS ON OUTLET LOSSES

The outlet kinetic energy from the last stage of a turbomachine must be induced with the outlet losses if no diffuser is mounted at the exit of this stage. Refering the outlet losses to the specific work of the last stage, then we can get an expression for the outlet loss coefficient L_o where:

$$L_o = \frac{c_3^2}{2Lu} = \frac{Cax^2}{\lambda . u^2 Sin^2 \alpha_3} = \frac{1}{\lambda} . \frac{\phi^2}{Sin^2 \alpha_3} \qquad (17)$$

The relation between cot α_2, cot α_3, ϕ, and "r" can be expressed by:

$$\cot\alpha_2 = \frac{1}{\phi}(\frac{\lambda}{4}+1-r), \cot\alpha_3 = \frac{1}{\phi}(\frac{\lambda}{4}-1+r) \quad (18)$$

where α_2, α_3 are measured as shown in Figure (1), substituting with (18) in (17) then:

$$L_o = \frac{1}{\lambda \cdot \cos^2\alpha_2}(\frac{\lambda}{4}+1-r)^2 + r - 1 \quad (19)$$

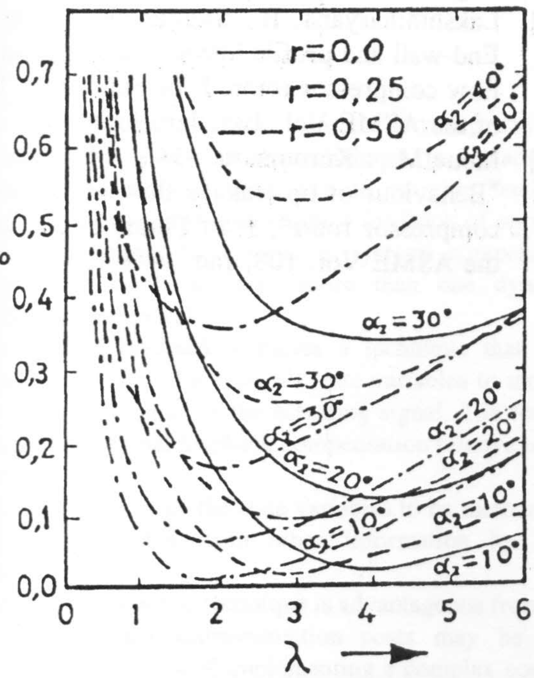

Figure 9. Effect of design parameters on the outlet loss coefficient.

The relation between "L_o" and "λ" is represented in Figure (9), with the degree of reaction "r" and the fixed blade outlet angle "α_2" as separate parameters. The obtained results show that the outlet loss coefficient decreases at first rapidly with "λ" for all angles and degrees of reactions until it reaches a certain minimum value then it begins to increase again. It is noticed that for greater α_2 the outlet loss coefficient is also greater for the same "λ", and "r" while it decreases as the degree of reaction increases for the same "λ" and" α_2. The minimum possible outlet losses can be achieved when the direction of absolute velocity from the moving blades of last stage coincides with the meridional direction, that means when $\alpha_3 = 90°$. From eq (18) we can get the relation between λ, r as $\lambda=4$ (1-r) which fulfills this conditions. This is obvious in Figure (9) where the minimum outlet loss coefficient for r = 0&0.25 and 0.5 correspond to λ= 4,3 and 2 respectively.

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