Design and Performance Prediction of an Axial Flow Compressor

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ABSTRACT.

The main objective of this paper is to create a method for designing and studying the performance of a multistage axial flow compressor. A mathematical methodology based on aerothermodynamics is used to study the on /off design performance of the compressor. Performance curves are obtained by changing the performance parameters in terms of design parameters (diffusion factor, solidity, Mach number, and inlet flow angle). Results show the great effect of diffusion factor on increasing efficiency than that of solidity, also the effect of both (diffusion factor and solidity) in increasing the amount of compression and efficiency of the compressor. Higher efficiency was found at the mean line between the root and tip of the blade. Best lift to drag ratio is found at inlet flow angle of (55^{o}).

Keywords: **Axial compressor, diffusion factor, solidity, inletflow angle, cascade aerodynamics.**

1. INTRODUSTION.

In an axial flow compressor, air is compressed while continuing its original direction of flow. From inlet to exit the air flows along an axial path. An axial flow compressor has two basic elements a rotor and a stator. The rotor has blades that are fixed on a spindle. These blades impel air rearward in the same way a propeller does. The rotor turns at a high speed and impels the air through a series of stages; high velocity airflow is produced. After the air is impelled by the rotor blades, it goes through the stator blades; they are fixed and act as diffusers at each stage. They partially convert high velocity air into high pressure .Each rotor / stator pair is a compressor stage. Each consecutive compressor stage compresses the air even more. The number of stages is determined by the amount of air and total pressure rise required .The greater the number of stages, the higher the compression ratio [1]. Many theoretical investigations had been carried out to study the design and performance of an axial flow compressor. *Metes, p.j.* [2] searched the optimization of design process of an axial flow compressor for different applications; it is carried out on the data of 15 stages. This type of compressor; covered a wide range of stage pressure ratios and Mach numbers by using numerical optimization. Algorithms to one and two dimensional calculation methods that used to show the compressor performance depend on variable stators. An improvement in this compressor efficiency and geometry (at on design and off -design point) is attainable. *Mc Glumpy and Fai* [3] studied and developed the design of axial flow rotor cascade by using a smaller airfoils (splitter vanes) placed circumferentially between the main blades of compressor rotor to improve the overall performance due to pressure ratio and efficiency . These splitter vanes improve the flow control in the axial flow during operating at off-design case and also reduce the number of stages in compressors. *Flack* [4] created a method to model an axial flow compressor. The calculation is based on common thermodynamics and aerodynamics principles in a mean stream line analyses. In this study the stage load coefficient is selected by changing it and keeping the other parameters constant,,hence, the pressure ratio will vary .This is an iterative process until the pressure ratio is converged. The purposes of this paper is to find better correlations and method to model a compressor that will result in fewer hours of fine tuning them in advanced fluid dynamic programs and hence same time and not to mention money .The primary objective of this study is to obtain the initial design, performance, and also by reflecting their parameters in terms of prototypes the aerodynamic of the compressor cascade.

2. THEORETICAL ANALYSIS.

Fig. (1) shows the compressor stage nomenclature .The behavior of the stage can be computed in terms of main design parameters ($\propto_\circ$, D, $M_\circ$, γ, σ, and $\emptyset_c$)
The following assumptions were used:
Two – dimensional flow, Constant mean radius, and repeating row, repeating airfoil cascade geometry

2.1. General Solution.

A number of stages (n) of repeating row stages maybe placed in succession so they are written in form $\pi_s^n \geq \pi_c$
The important aerodynamic parameters are:

-Flow deflection

The flow deflection is one of the important parameters that reveal the amount of diffusion without any significant loss.

$$\delta_\alpha = f(D, \sigma, \alpha_\circ, \alpha_1)$$

Where δ_α gives the blade turn angle, the exit flow angle is α_1 [5]

$$\cos \alpha_1 = \frac{\Gamma + \sqrt{\Gamma^2 - \left[\Gamma^2 + {}^1/_{(4\sigma^2)}\right]\left[1 - {}^1/_{4\sigma^2}\right]}}{\left[\Gamma^2 + {}^1/_{(4\sigma^2)}\right]} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1) \qquad (1)$$

Where the circulation Γ can be calculated as:

$$\Gamma = \frac{1+D}{\cos\alpha_\circ} + \frac{\tan\alpha_\circ}{2\sigma}$$

-Stage efficiency

Several efficiencies are used to compare the performance of compressor stage designs . The most commonly used is the stage efficiency (η_s) ; it's defined as ratio of the ideal work per unit mass to the actual work per unit mass between the same total pressures . It can be calculated as: [5]

$$\eta_s = \frac{\left(\frac{1-\phi_c}{\cos^2\alpha_\circ}\right) - 1/\cos^2\alpha_1}{\frac{1}{\cos^2\alpha_\circ} - \frac{1}{\cos^2\alpha_1}} \quad \ldots\ldots\ldots\ldots\ldots.\ldots (2) \qquad (2)$$

While the stage pressure ratio may be written as:

$$\pi_s = 1 + \left[\frac{1-\emptyset_c}{\cos^2\propto_\circ} - \frac{1}{\cos^2\alpha_1}\right] \frac{\gamma M_\circ^2 \cos^2\propto_\circ}{\left[1 + \frac{\gamma-1}{2} M_\circ\right]^{\gamma/\gamma-1}} \qquad (3)$$

The stage temperature ratio can be determined by: [5]

$$\tau_s = 1 + \left[\frac{\pi_s - 1}{\eta_s}\right]^{\gamma-1/\gamma} \qquad (4)$$

-Diffusion factor

The blade loading is usually measured by the diffusion factor. This relates the peak velocity on the suction surface of the blade to the velocity at the trailing edge. The boundary layer is separated because of the large static pressure rise causing by the growing of boundary layer on the blade suction surface. The design of axial flow compressor is limited to diffusion factor ≤ 0.6. Total pressure loss increases with diffusion factor above 0.6. Afactor of 0.4 is a value that can eliminate the danger of separation. [5, 6, 7].

$D = \frac{V_{max} - V_e}{V_i}$ i=inlet, e=exit,

The diffusion factor can be written in terms of flow angles: [5]

$$D = \left[1 - \frac{\cos\alpha_0}{\cos\alpha_1}\right] + \left[\frac{\tan\alpha_\circ - \tan\alpha_1}{2\sigma}\right]\cos\alpha_\circ \tag{5}$$

- Flow coefficient

For modern axial flow compressors of gas turbine engines the flow coefficients are in the range (0.45 -0.55) at the mean radius [6].

$$\phi = \frac{V_a}{U} = \frac{V_a}{\omega r} \text{ ……… (6)} \tag{6}$$

or in terms of flow angles (α_0,α_1)

$$\Phi = \left[\frac{\tan\alpha_\circ - \tan\alpha_1}{\tan\alpha_\circ + \tan\alpha_1}\right]$$

-Stage loading

The ratio of the stage work to rotor speed squared is called the stage loading and is defined as:

$$\psi = \frac{c_p\,\Delta T_1}{U^2} = \frac{c_p\,\Delta T_1}{(\omega r)^2} \tag{7}$$

The stag loading coefficients have a range (0.3-0.35) at the mean radius in modern axial flow compressors. [6]

- Flow path

The flow path in initial design is based on the mean radius .The annulus area is based on the flow properties (T_t ,P_t , Mach number ,and flow angles) and the total mass flow rate **.**

$$A = \frac{m\sqrt{T_1}}{P_1\ COS\alpha MFP(M)} \tag{8}$$

From which tip, mean, and hub radius may be calculated. where MFP is the mass flow parameter.

$$MFP = \frac{m\sqrt{T_1}}{P_1(\ COS\alpha_\circ)}$$

A conservative analysis can be performed by using the largest annulus area for any stage and Taking $\mathbf{A_T/A_H}$ =1.0. [5]

- The centrifugal stress (σ_c)

The force acting on an airfoil at cross section and any radius must restrain the centrifugal force on all the material beyond it **Fig. (2).**The root of the airfoil must experience the greatest force.

$$\sigma_C = \frac{\rho\,\omega^2 A}{4\pi}\left[1+\frac{A_T}{A_H}\right] \tag{9}$$

Where A =flow path area= $\pi(r_T^2 - r_H^2)$
While, the rotational speed is given by[6]

$$\omega = \sqrt{\frac{4\pi\sigma_C}{\rho A\left(1+\frac{A_T}{A_H}\right)}} \tag{10}$$

Where $\frac{A_T}{A_H} = 1$
To reduce the centrifugal stress by using more light weight materials (e.g., Titanium with a density of about 4600 kg/ m^3) [6].

2.2. Cascade Force Analysis.

An airfoil produces a lifting force that acts at right angles to the air stream and a dragging force that acts in the same direction as the air stream .Forces acting on the cascade are shown in **Fig. (3).**

- lift force (L)

The lift force arises because the speed at which the displaced air moves over the top of the airfoil (and over the top of the attached boundary layer) is greater than the speed at which it moves over the bottom. The pressure acting on the airfoil from below is therefor greater than the pressure from above. The design of airfoil has a critical effect on the magnitude of the lift force. [9]
If a mean flow direction is defined as:

$$\tan\alpha_m = \frac{\tan\alpha_0 + \tan\alpha_1}{2} \tag{11}$$

Then the lift force on the cascade perpendicular to that direction [10,11] is:-

$$L = \rho s {V_a}^2 \sec\alpha_m\,(\tan\alpha_0 - \tan\alpha_1) - s\Delta P_O sin\,\alpha_m$$

$$C_L = 2\left(\frac{s}{c}\right)\cos\alpha_m\,(\tan\alpha_0 - \tan\alpha_1) \tag{12}$$

-Drag force (D)

The drag coefficient is a measure of the loss of energy transfer with the useful task of producing lift .The maximum energy transfer implies the largest possible fluid deflection or lift coefficient, while maximum efficiency requires the lowest possible Loss of pressure or drag coefficient .From the cascade test results, the profile losses through compressor blading of the same geometry may be estimated. These losses are estimated by using the following drag coefficients, for the annulus walls loss [8]

And for the secondary loss,

$$C_{Ds} = 0.018{C_L}^2$$

Where s, h are the blade spacing and blade height respectively.

The drag force is the resultant of the viscous forces produced at the surface of the blade. So the overall drag coefficient can be written as:[12]

$$C_D = C_{DP} + 0.02\ \frac{s}{h} + 0.018\ C_L^2 \tag{13}$$

C_{DP} = Profile drag coefficient and it is in the range (0.018 to 0.025) [8, 11, 13].

Then the total pressure losses coefficient can be written as:

$$\omega_c = \frac{C_D}{\left[\left(\frac{S}{C}\right)\frac{\cos^3\alpha_m}{\cos^2\alpha_0}\right]} \tag{14}$$

This enables the total pressure loss coefficient for the blade row to be determined [13].

- Static pressure rise coefficient (C_{pr}) **:**

The static pressure rise across a compressor cascade may be expressed in terms of the inlet dynamic head and the inlet and discharge flow angles [10].

$$p_1 - p_0 = \frac{1}{2}\rho(V_0^2 - V_1^2) = \frac{1}{2}\rho V_0^2\left[1 - \frac{\cos^3\alpha_0}{\cos^2\alpha_1}\right]$$

A static pressure rise coefficient for the cascade may be defined as:

$$C_{pr} = 1 - \frac{\cos^2\alpha_0}{\cos^2\alpha_1} \tag{15}$$

If a total pressure loss through the cascade,ΔP_0 , is introduced , then

$$C_{pr} = \left[1 - \frac{\cos^2\alpha_0}{\cos^2\alpha_1}\right] - \omega_c \tag{16}$$

-Blade Efficiency (η_b)

The efficiency of the blade row (η_b) is defined as the ratio of the actual pressure rise to the theoretical pressure rise and can be written as [13].

$$\eta_b = 1 - \frac{\omega_c}{\left[1 - \frac{\cos^2\alpha_0}{\cos^2\alpha_1}\right]} \tag{17}$$

-Compressor Cascade Efficiency (η_D)

The efficiency of a compressor cascade can be obtained in terms of drag, lift coefficients and the mean flow angle [13].

(18)

-Aspect Ratio (AR)

As in the case of axial flow compressors the designer has a considerable freedom to choose the number of blades in a blade row. A large number of blades leads to a shorter compressor, and vice versa. As the number of blades in rotors and stators is changed, there will be proportional changes in the excitation frequencies from blade wakes, and nonlinear changes in blade natural frequencies, thus allowing the designer to use the choice of number of blades as a principal method of avoidance of critical excitation. [15]

The aspect ratio is defined as:

$$\mathrm{AR} = \frac{\mathrm{h}}{\mathrm{c}} \tag{19}$$

Where ***h*** is the blade height and ***c*** is the chord . To specify the number of blades, suitable limit for aspect ratio and solidity must be assumed. The spacing s can be obtained from the definition of solidity

$$\sigma = \frac{c}{s} \tag{20}$$

The number of blades is simply the circumference of the mean line divided by the blade spacing, rounded up to the next integer [6, 13]

$$\eta = \frac{(2\pi * r_m)}{s} \tag{21}$$

3. THE COMPUTER PROGRAM.

For the design and performance prediction of the AFc,in the sequence mentioned, many design parameters were selected. This makes compressor design both fascinating and perplexing. [16]

The computer program is capable of carrying out the initial design of the axial flow compressor, and prediction of its performance. Also itis used to simulate the on-design, off-design, optimum performance, and to perform the useful calculations of cascade aerodynamics and performance. To calculate the off-design performance of the compressor, there is an iterative procedure having several trials carried out to ensure that all compressor variables are consistent with the "handle" values.

To perform the design, performance, and cascade aerodynamics the program was built and written in Fortran power station to simalate the sequential aero-thermodynamic processes.

3.1. Cascade Aerodynamics Module.

The aerodynamic of flow through compressors can be handled in two ways:

1-isolated airfoil, in this case the blades are rather widely pitched, which will be called low solidity and each blade behaves as a single airfoil.

2-cascade of airfoil: in the cascade theory the design is based upon wind- tunnel tests [17].This model is characterized by its adaptation to evaluate all the aerodynamic and performance parameters due to the conventional input data for classical performance calculations. **Fig**. **(4)** shows the cascade aerodynamic design procedure.

4.RESULTS AND DISCUSSION.

Modeling of the axial flow compressor may be reflected in many informative results that can reveal the brilliance of the design, performance, and development of the Compressor. The results describe the envelope of the compressor and its operation within the limitations of the design and performance. **Figs. (5,6 ,and7)** present the behavior of the compressor stage for the entire range of the design parameters.

The stage efficiency increases with inlet flow angle. Higher solidity results in lower rotational blade speed and consequently higher stage loading and hence higher efficiency .Closer blade spacing means that the blades can operate over a wider range of inlet flow angles without separation **Fig.(5).** Variation in stage loading with inlet flow angle and solidity in the range of (0.9 to 1.1) is presented in **Fig.(6).**It shows that for a certain value of inlet flow angle, as solidity increases the stage loading increases too. **Fig. (7)** shows the stage pressure ratio dependence on inlet flow angle and diffusion factor in the range of (0.4 to 0.55).Higher values of diffusion factors result in higher stage pressure ratios as they give higher stage loading.

The mathematical model is written such that the performance parameters are functions of the input choices. The diffusion factor is always chosen in the design stage calculation.This research is adapted to choose different values of the diffusion factor for rotor and stator and it is also calculated for different sectors along the blade height .These calculations are drawn in **Fig. (8).**

An interesting generalized map is illustrated in **Fig**. **(9).**The results show that a peak value of the key performance parameter (stage efficiency) was found slightly increasing at the mean line. That is why the on-design calculation was made on this path. The variations in stage loading and flow coefficient are also included in this figure. **Fig**.**(10)** shows the dependence of stage efficiency on the location along the blade height, also it shows that the maximum value is attained at the mean radius.

Fig.(11) shows the influence of inlet flow Mach number on the drag coefficient and its gradient with blade height .It shows that the drag increases with the increment of Mach number. It also shows that the drag near the hub is higher than that at the tip because of the increase in skin friction which leads to higher skin friction change. **Fig**. **(12)** shows the effect of blade row opening on the lift coefficient along the blade height. Lower values of radius ratio lead to higher lift coefficient .For a given radius ratio, higher values of solidity means low opening which results in lower lifting force. **Fig**. **(13)** shows that the increase in the inlet flow angle causes increase in the drag coefficient due to diffusion. Also the increment in solidity causes the lift force to decrease, consequently the resulting loss due to secondary flow leads to decrease in the value of drag coefficient. **Fig. (14**) shows the total pressure loss coefficient increases with the increment of inlet flow angle and solidity.

The maximum range of blade efficiency is achieved when solidity is around (0.9) and the value of drag coefficient is about (0.034). Beyond these values the blade efficiency decreases while the drag coefficient increases, and the solidity increases too, **Fig**. **(15).** The maximum range of blade efficiency is achieved when the value of lift coefficient is higher than (1.2), also the blade efficiency decreases with the increase in solidity, **Fig. (16).**The given loss is based on a set of given values of up to (0.04), also the blade efficiency increases with the increase in spacing (s\c). This value represents the reciprocal of solidity coefficient with increment of the calculated loss coefficient in terms of the design constraints, the blade efficiency decreases, **Fig**. **(17).** The best efficiency for the compressor cascade is found at the values of the inletflow angle of around (55°). When the values of the air inletflow angle decreases the efficiency decreases due to the stall that may issue with the boundary layer, **Fig. (18).**

5. CONCLUSIONS.

1) A value of inlet flow angle of about 55° can be considered as optimum as it gives flow coefficient in the range preferred in modern compressors at the given solidity and the stage loading. A compromise could be done to satisfy reasonable values of flow coefficient and stage loading within the constraints.
2) An interplay performed between the main design parameters (diffusion factor, solidity) and inlet flow angle could characterize the design technique of the axial flow compressor.
3) The reduction in the spacing between the blades and the increment in the total surface area cause an increase in the loss of pressure or drag due to skin friction, also at lower solidity awide spacing between blades happen.
4) The best design value for the construction of cascade of the compressor is at the inlet flow angle which ranges between (55° – 58°) ,and also by giving reasonable efficiency and low drag to lift ratio . This range is in line with most modern designs.

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Nomenclature

Symbol	Definition	Unit
A	**Annulus area**	m^2
	Tip to hub annulus area	
c	**Blade chord**	m
C_D	**Drag coefficient**	
C_{Da}	**Drag coefficient due to annulus walls losses**	
C_{Dp}	**Profile drag coefficient from cascade tests**	
C_{Ds}	**Drag coefficient due to secondary losses**	
C_L	**Lift coefficient**	
C_{pr}	**Static pressure rise coefficient**	
C_p	**Specific heat at constant pressure**	kJ/kg.K
D	**Diffusion factor**	
DF	**Drag force**	N
h	**Blade height**	m
L	**Lift force**	N
M_0	**Inlet Mach number**	
m	**Mass flow rate**	kg/sec
MFP	**Mass flow parameter**	
n	**Number of stages**	
P_s	**Static pressure**	N/m^2
P_t	**Total pressure**	N/m^2
R	**Gas constant**	J.kg/K
r	**Radius**	m
r_H	**Hub radius**	m
r_m	**Mean radius**	m
r_T	**Tip radius**	m
s	**Space between blades**	m
s/c	**The space-to-chord ratio**	
T_s	**Static temperature**	K
T_t	**Total temperature**	K
U	**Rotor speed, mean wheel speed**	m/s
V_0	**Inlet velocity**	m/s
V_a	**Axial velocity**	m/s
V_m	**Mean velocity**	m/s

Greek symbols.

Symbol	Definition	Unit

α	**Inlet flow angle**	degree
α	**Mean flow angle**	degree
α	**Exit flow angle**	degree
Γ	**Circulation**	m^2/s
γ	**Specific heat ratio**	
δ	**Flow deflection**	degree
η	**Blade efficiency**	**%**
η	**Compressor efficiency**	**%**
η	**Compressor cascade Blade efficiency**	**%**
η	**Stage efficiency**	**%**
π	**Compressor pressure ratio**	
π	**Stage pressure ratio**	
ρ	**Density**	Kg/m^3
σ	**Solidity**	
σ	**Centrifugal stress**	N/m^2
τ	**Stage temperature ratio**	
ϕ	**Flow coefficient**	
ζ	**Pressure loss**	
ψ	**Stage loading**	
ω	**Rotational speed**	r.p.m(rad/sec)
ω	**Total Pressure loss coefficient**	

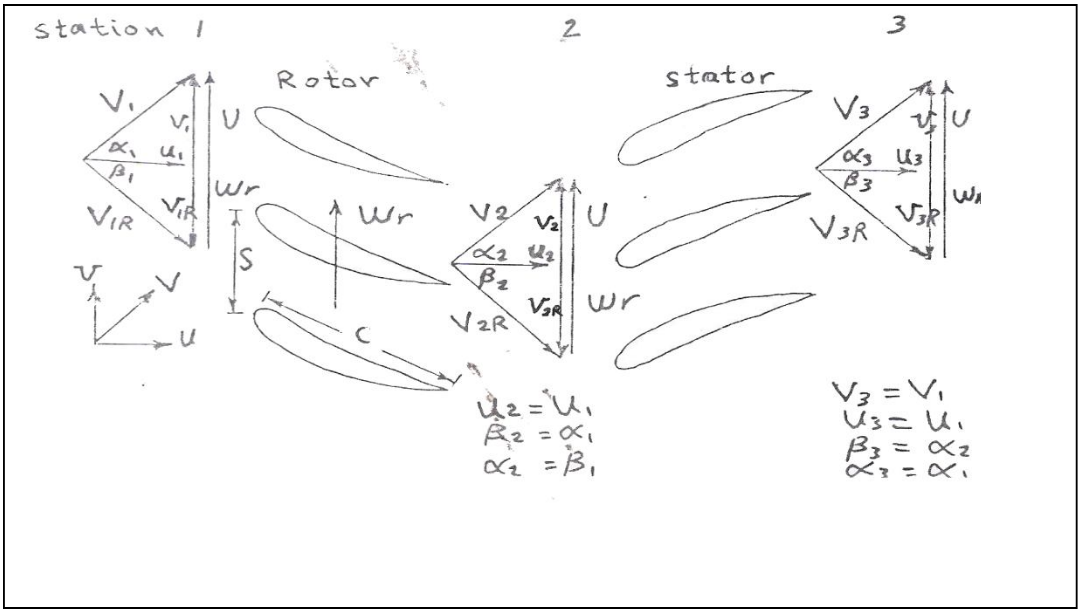

Figure (1): Repeating row compressor stage nomenclature [5].

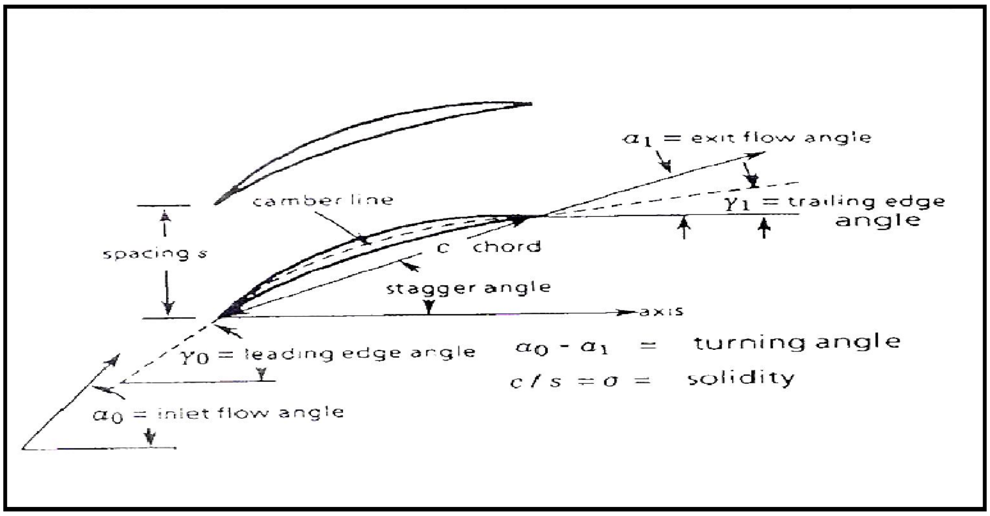

Figure (2): cascade Aircoil Nomencalture [5].

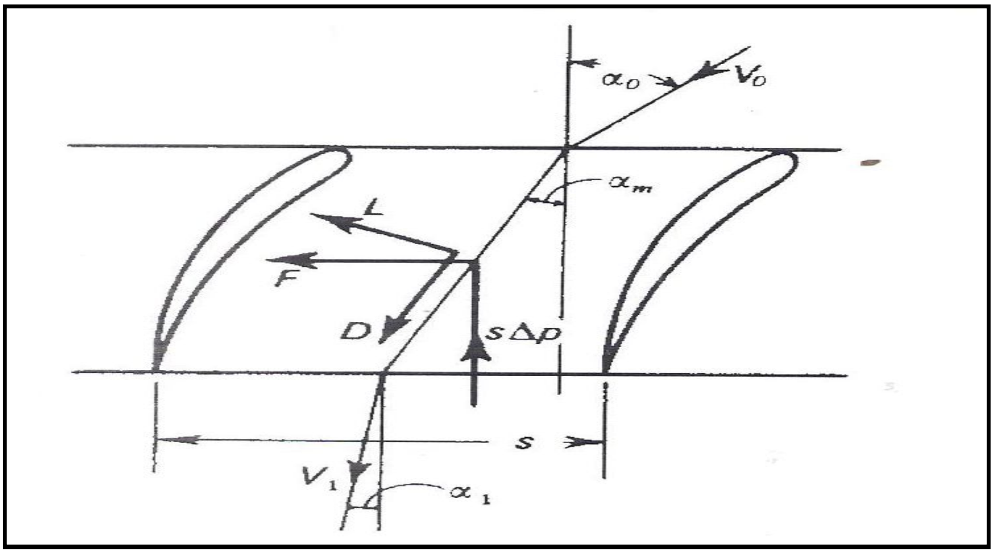

Figure (3): Applied and Effective Forces Acting on Cascade [13].

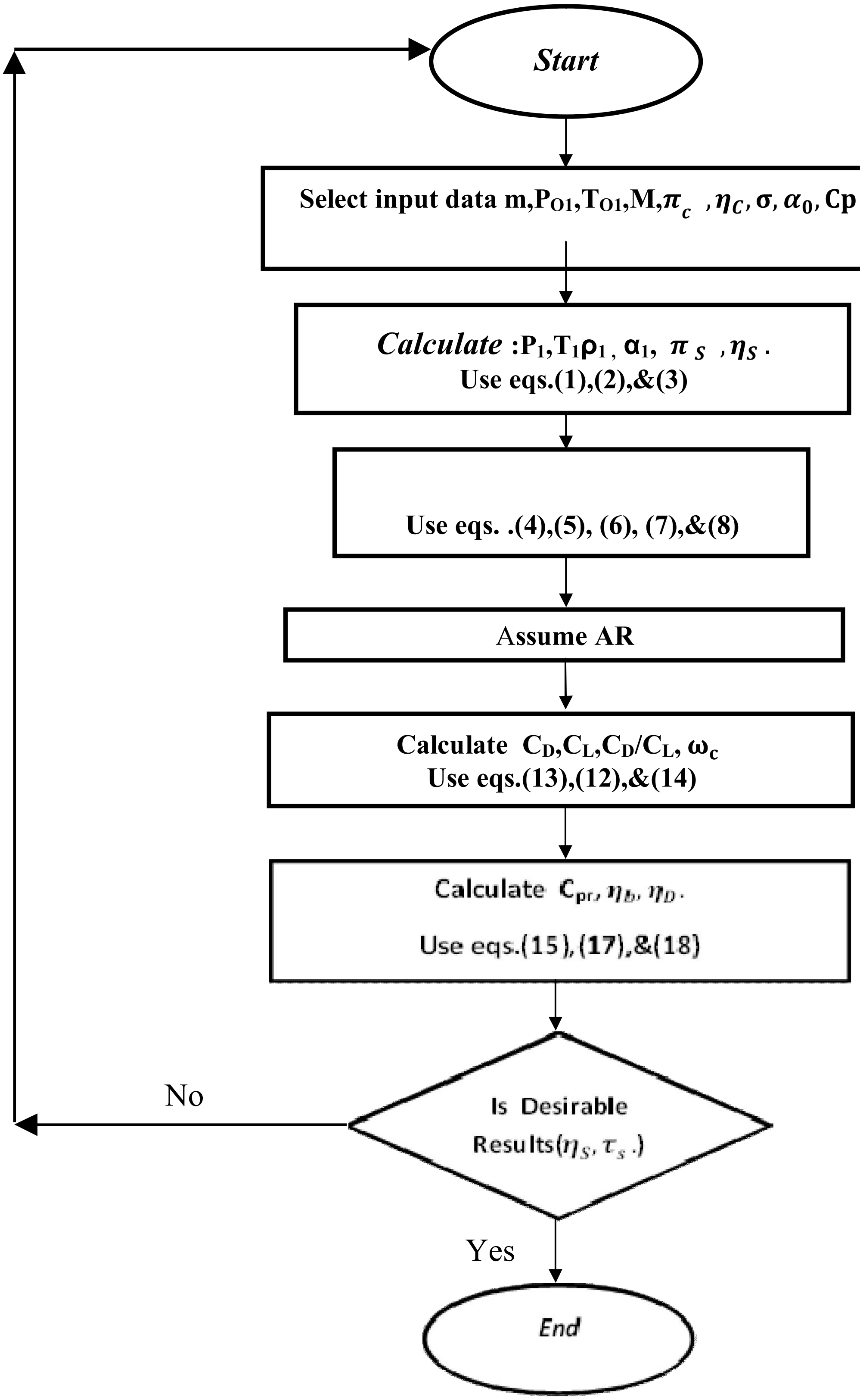

Figure (4): Cascade Aerodynamic Design Procedure.

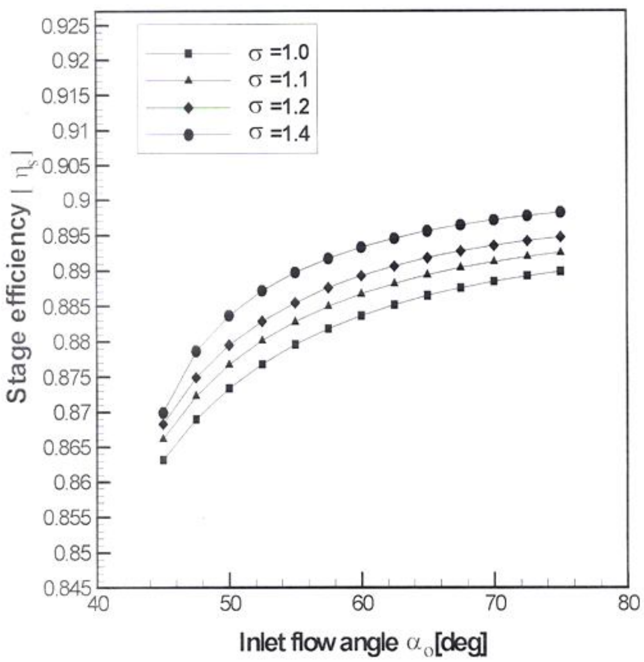

Figure (5): Variation in stage Efficiency with Inlet flow Angle for Different solidity.

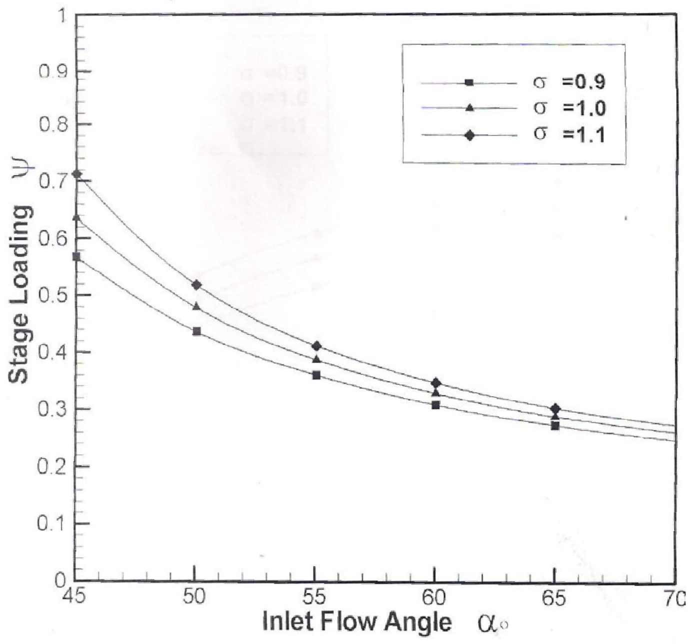

Figure (6): influence of Solidity on Stage loading.

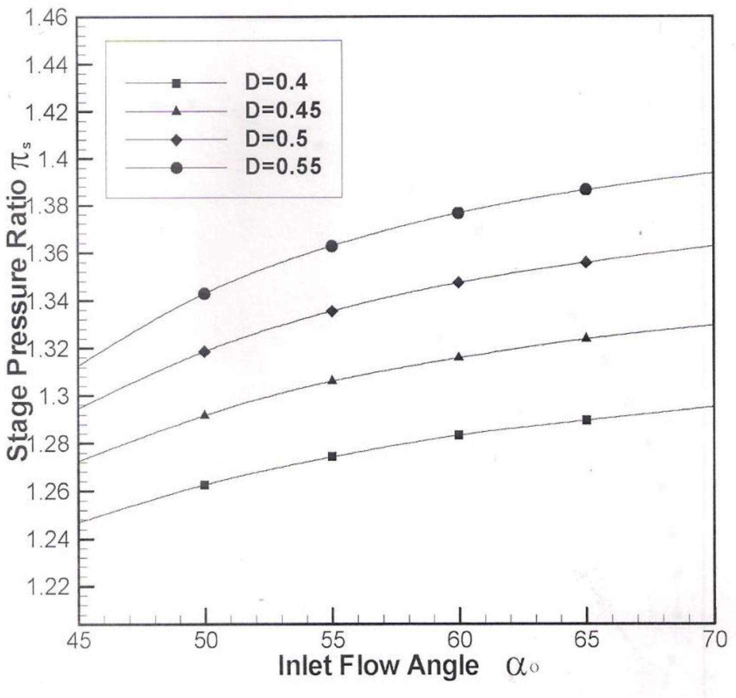

Figure (7) : Variation in stage Pressure ratio with Intel flow Angle for different diffusion factor.

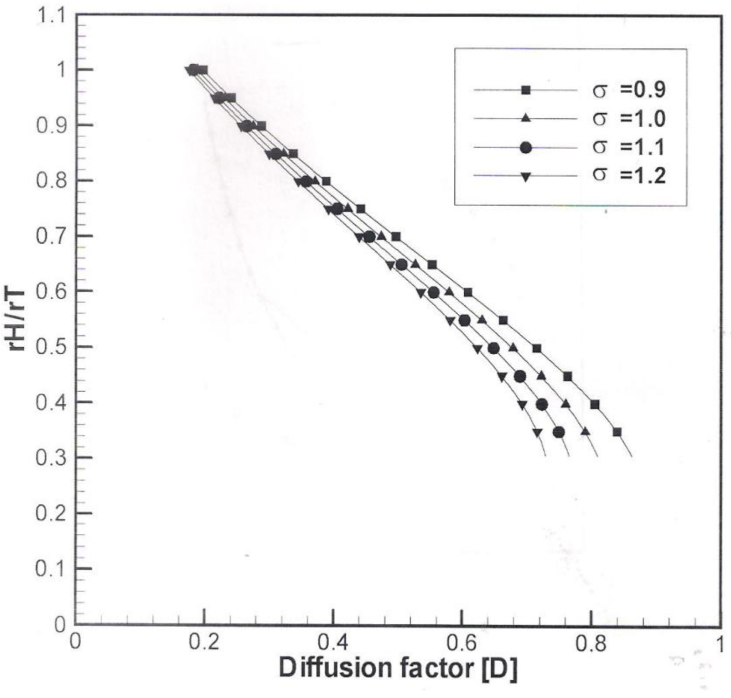

Figure (8): Variation in Hub Radius to tip radius Ratio with diffusion Factor at Different Solidity.

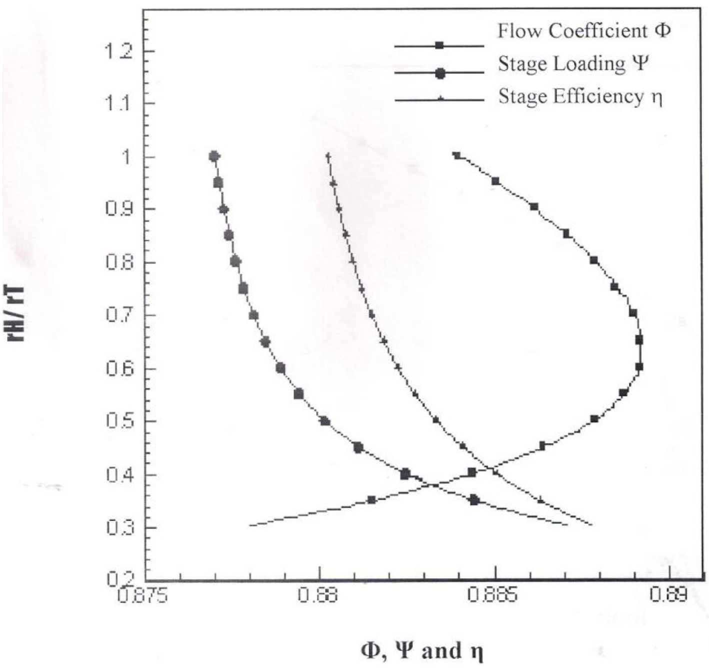

Figure (9): Variation in stage Efficiency stage Loading and flow coefficient with Hub Radius to Tip Radius Ratio.

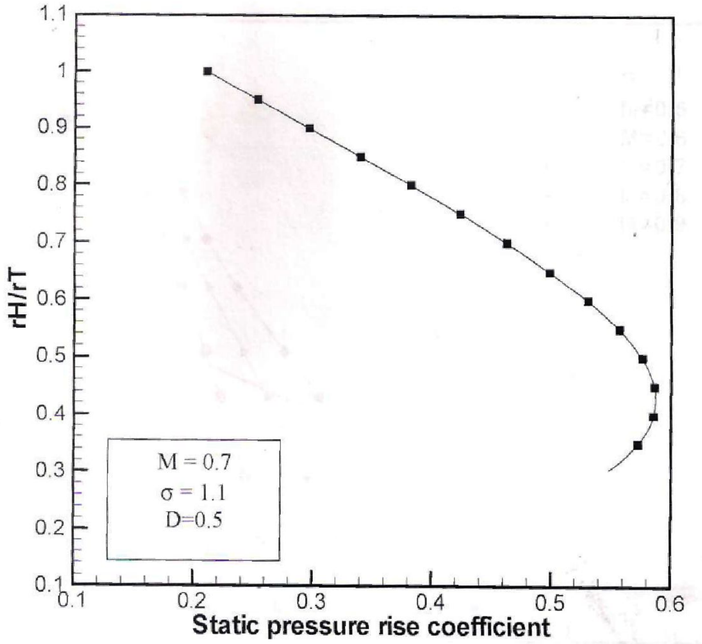

Figure (10): Variation in static Pressure rise coefficient along the Blade Height.

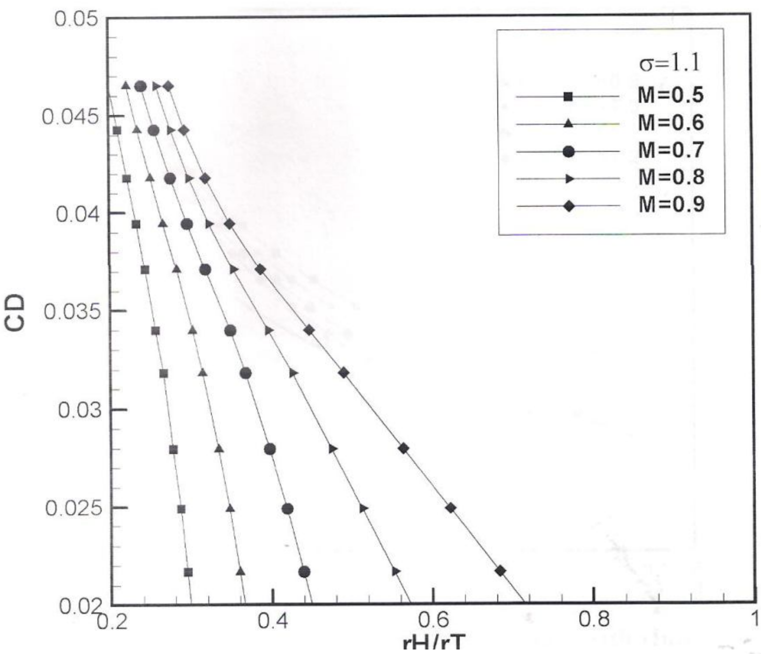

Figure (11): Variation in Drag coefficient With Hub radius to Tip Ratio at Different mach Number.

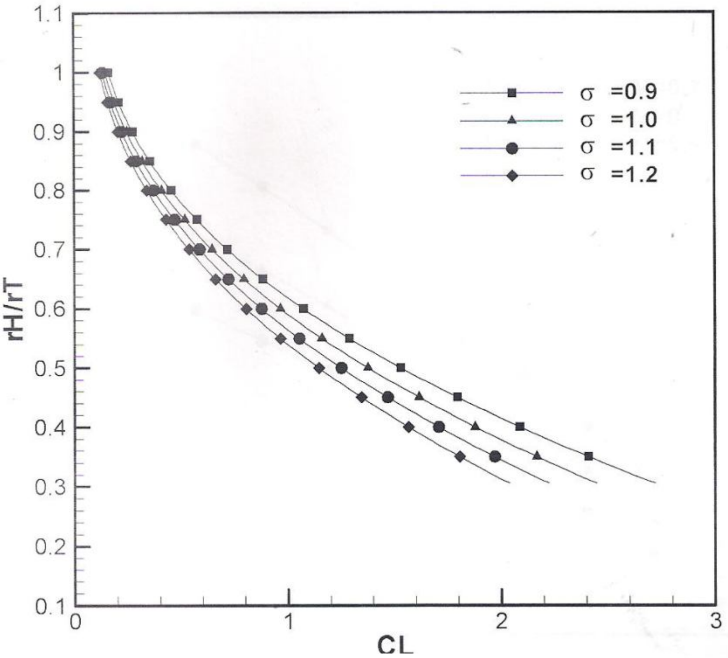

Figure (12): Variation in lift coefficient with Hub radius To Tip Radius ratio at Different solidity.

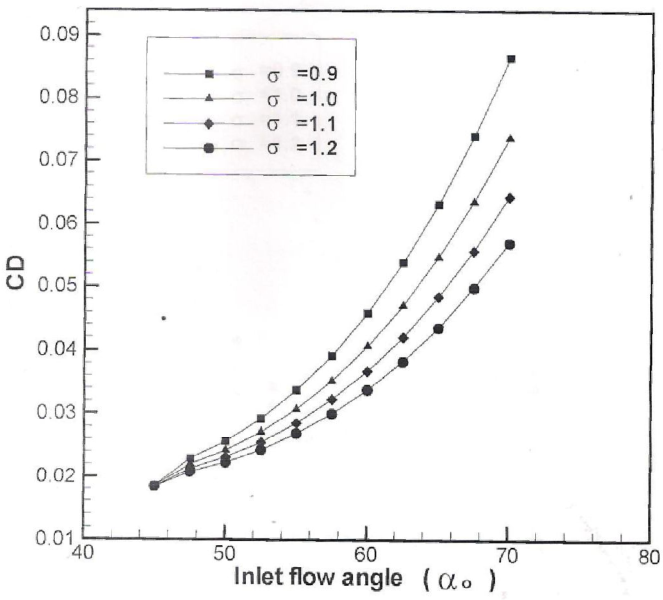

Figure (13): influence of solidity on Drag Coefficient.

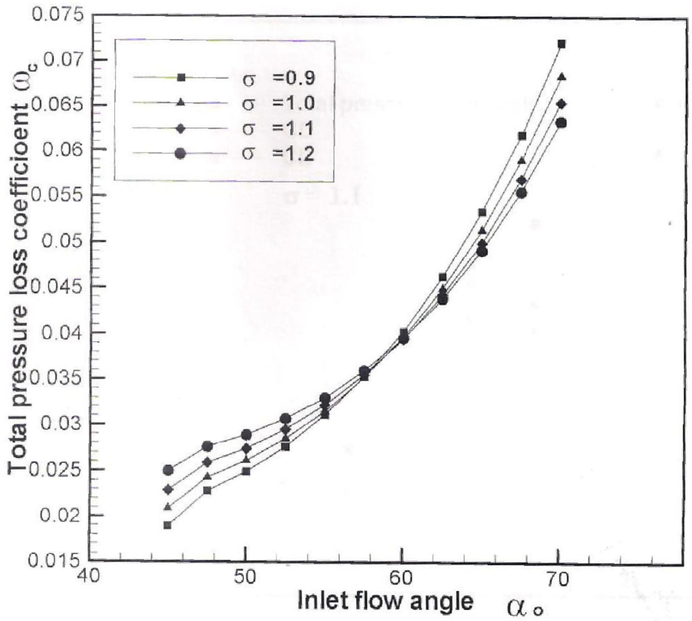

Figure (14): influence of inlet flow Angle and solidity on Total pressure loss coefficient.

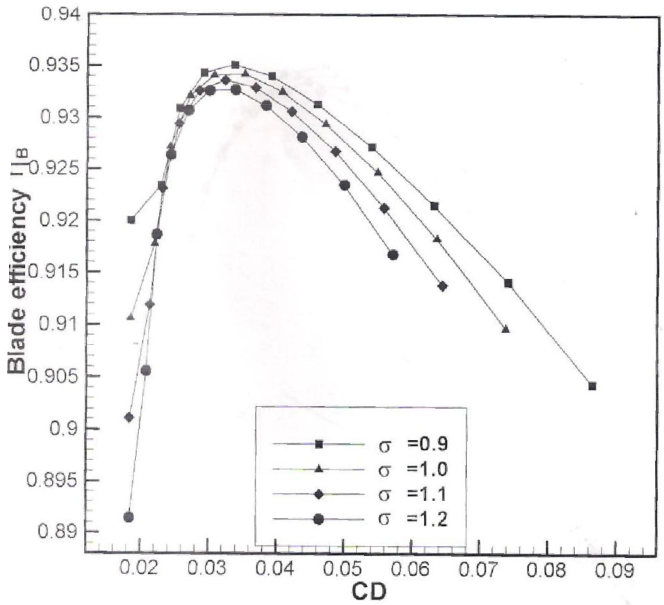

Figure (15): Variation in Blade Efficiency with Drag coefficient at Different solidity.

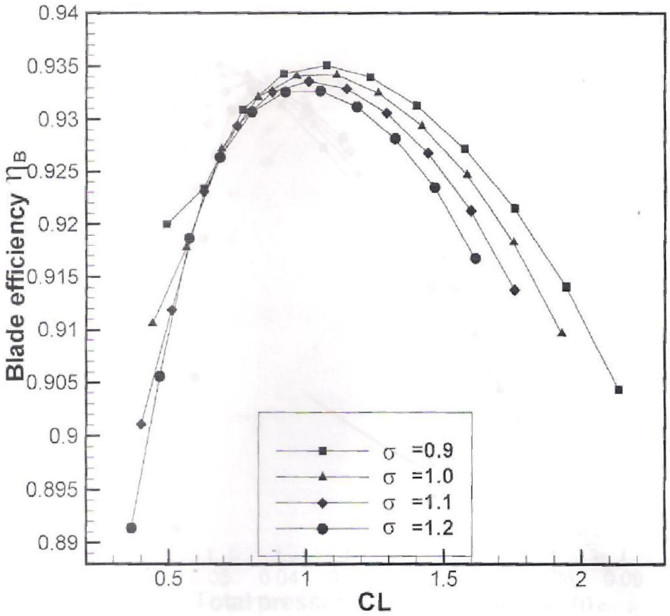

Figure (16): Variation in Blade Efficiency with lift coefficient for Different solidity.

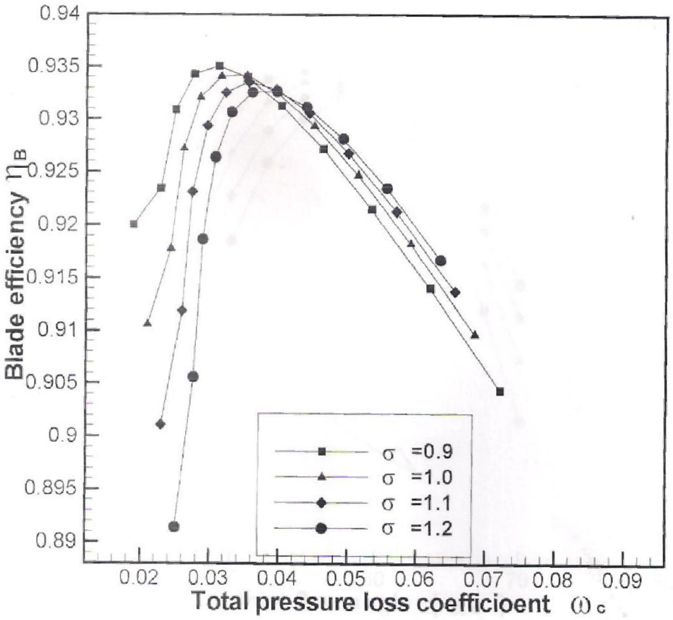

Figure (17): Variation in Blade Efficiency with Total pressure loss coefficient at Different solidity.

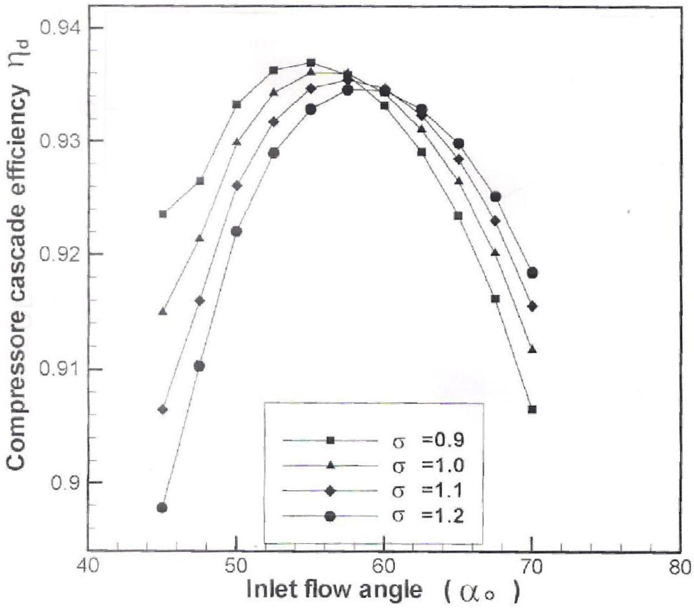

Figure (18): influence of Solidity on Compressor Cascade Efficiency.

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التصميم والأداء المتوقع للضاغطة المحورية

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الخلاصة.

الهدف الرئيسي هو ايجاد طريقة لاجراء تصميم و دراسة اداء للضاغطة المحورية متعددة المراحل , وبأستخدام منهجية رياضية تعتمد الديناميكية الحرارية و الهوائية لاستنباط الاداء التصميمي للضاغظة . تم الحصول على أفضل منحنيات الاداء بتغير المحددات الادائية بدلالة المحددات التصميمية وأظهرت النتائج التأثير الكبير لعامل الانتشار على زيادة الكفاءةوكان هذا العامل اكبر من عامل الصلابة، كذلك تأثير كل من عامل الانتشار والصلابة على زيادة الكفاءة ونسبة الانضغاط . وجدت ان اعلى كفاءة تقع ضمن خط متوسط البعد بين جذر وحافة الريشة احسن نسبة للرفع الى الكبح تم الحصول عليها عند زاوية دخول الهواء الى الريشة حوالي (55°).

الكلمات الرئيسية:الضاغط المحوري،معامل الإنتشار،الصلابة،زاويةالجريان الداخلية،ديناميكيةصف الريش.