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Design and Performance Analysis of Cooling Tower Axial Fan Using the Thin Airfoil Theory and CF D

تصميم وتحليل أداء المروحة المحورية لبرج التبريد باستخدام نظرية الجناح ذي المقاطع الرقيقة وديناميكا الموائع الحسابية

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KEYWORDS:
Axial Fan – Cooling Towers – Controlled Vortex – CFD – Matlab – Performance Curve – Turbogrid

***الملخص العربي:*-تم تحويل معادلات السريان الحاكمة في اتجاه السريان وفي الاتجاه الطولي للريش للمراوح المحورية لأبراج التبريد إلى لوغاريتم عددي وتم تحويل هذا اللوغاريتم إلى تطبيق بالماتلاب ذي واجهة رسومية بأقل عدد ممكن من المدخلات وذلك بهدف إخراج التصميم المطلوب وتحليل توزيع السرعات والضغوط باستخدام نظرية الجناح ذي المقاطع الرقيقة. تلك المراوح تتميز بعدم وجود ريش توجيه على الدخول أو الخروج وأنها وحيدة المرحلة وذات ضغوط منخفضة وكميات سريان كبيرة. أيضاً تم حل المعادلات الحاكمة لسريان الهواء في هذه المراوح باستخدام برنامج ANSYS CFX-16.0 ومقارنة نتائج التصميم الجديد بالموجود (الذي يتميز بالتصميم عند قطر متوسط) لمروحة برج تبريد الأمونيا بشركة الدلتا للأسمدة بطلخا - مصر، وذلك بهدف اقتراح تصميم جديد ذي كفاءة عالية يعطي كمية هواء أكبر أو يعطي ضغط أكبر عند نفس كمية الهواء ونفس استهلاك الطاقة مما يُمَكِن من تركيب حشو جديد ذي مقاومة أكبر ولكن يزيد من مساحة سطح التلامس بين الماء والهواء مما يزيد من كفاءة التبريد في الحالتين.**

***Abstract*— Numerical algorithms for the flow of cooling tower axial fans in the flow wise direction and span-wise direction have been transformed and solved by the Matlab application with a graphical user interface to obtain the optimum axial fan design. The analyses of the velocity and pressure distributions were achieved using the thin airfoil theory. The fan has a single rotor with no inlet or outlet guide vanes and produces a large amount of flow rate and a small pressure rise. Then, the air flow governing equations for the new design were solved using the commercial computational fluid dynamics (CFD) code ANSYS CFX-16.0. A comparison between the new design and an existed design (usually a mean line design approach) of the ammonia cooling tower fan in El-Delta for Fertilizers Company in Talkha City, Egypt was done. The efficiency of the new design is higher than that of the existed one. The new designed fan is capable to deliver more air flow or the same flow rate at higher pressures with the same power requirement. This enables the installation of new high resistance fill, which increases the area of contact between air and water to increase cooling efficiency.**

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I. INTRODUCTION

COOLING tower is an essential part of nearly every industrial plant. Recently, the increasing of the performance of new and existing cooling towers is of great interest. One of the methods of increasing cooling towers performance is to force the fan system to deliver more cooling air with the limiting factors of dimensions and power. The other method is to install higher resistance fill for the same flow rate and power which is the most effective approach. Whatever the choice, the geometry of the blades is fixed.

The studies of axial fans began in the twentieth century in USA and UK. These studies are concerned mainly with airfoil shape for certain Reynolds and Mach numbers, and finding ideal incidence and loading coefficients.

Moreover, the design and analysis of axial fans are usually a section or more in turbomachinery books and usually grouped with axial compressor in most textbooks, Dixon and Hall (2010), Wilson and Korakianitis (1998), Lewis (1996), and Csanady (1964). The previous studies dealt primarily with the mean line design and analysis, and radial equilibrium theory for the flow in the span-wise direction. Turner (1966) studied the performance of fans with low pressure by isolated airfoil theory and the performance of compressors with high pressure by cascade analysis and analyzed the results.

Wallis (1968) deduced an algorithm for predicting deviation angle and the axial fan design. Downie et al. (1993), Wallis (1983) and Wallis (1961) studied the performance of axial fans experimentally and theoretically.

Bruneau (1994) deduced a simple design approach, which was the foundation for many other studies that used his algorithm and verified it. He concluded a numerical algorithm for designing outlet blade angle at a discrete radial station that matches proposed simple vortex distribution at blade exit. This meridional flow plane (non-free vortex approach) is used the present study in the design program.

Lewis (1996) was a pioneer as he used Pascal programming language to design and analyze turbomachinery. Furthermore, he studied both flow in span-wise and flow wise. Also, the loading coefficient values, vortex flow distributions and optimum stagger angle were studied.

Aungier (2003) studied the design processes of an axial compressor. The popular blade profiles, which are used in both axial flow compressors and fans, were obtained.

Falck (2008) designed a multi-stage axial compressor using the mean line method by constructing a Matlab program with graphical user interface. This was an inspiration to do the same for axial flow fans used in cooling towers, but extended to three-dimensional design and showing some perfomance parameters also in the program interface.

Pascu (2009) introduced a simplified approach for axial fan design. The blade camber line was created by an algorithm that computes the camber line tangent to flow direction with a second order polynomial.

Aktürk and Camci (2010) discussed the tip leakage effect on axial fans performance. They installed several tip extensions with various shapes and choose the one which gives the least leakage and best performance.

Louw et al. (2012) reviewed and summarized meridional flow plane (non-free vortex) algorithm suggested by Bruneau (1994).

Vad (2013) studied the effect of combining controlled vortex design and sweep along radius. It is found that the tip leakage and skin friction losses were reduced.

Also, Masi and Lazzaretto (2015) increased the performance efficiency by applying sweep along radius for low hub-tip ratios axial flow fans.

Now, every cooling tower fan manufacturer has its own selection program for its own fan series that helps designers to select the appropriate fan shape.

The objective of the present study is to create a Matlab code which is able to design and analyze a one stage axial fan that is used in cooling towers and has no inlet or outlet guide vanes. The design is made in the span wise direction using non-free vortex and in the flow direction using nonstandard airfoil series. The analysis is done using the thin airfoil theory. The Computational Fluid Dynamics (CFD) code Ansys CFX (Ansys Inc., 2015) is performed to the deduced design to verify the efficient design.

II. Design and Analysis

The assumptions used in design and analysis processes are as follows:

1- The fan has a hub and tip shroud with no inlet or outlet guide vanes.

2- The blade is made up of a series of two-dimensional sections along the span. At each section, the flow is two-dimensional.

A. Simplified Three-Dimensional Analysis

Wu (1952) suggested the model shown in Figure (1), where the surfaces S-1 and S-2 are not two-dimensional exactly but twist with flow and blades.

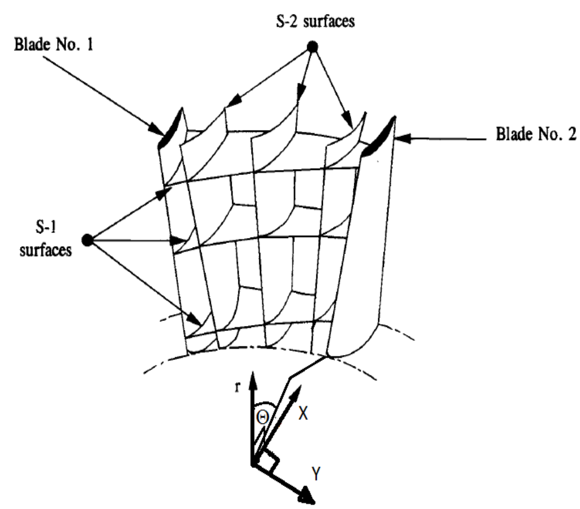

Figure (1) S-1 and S-2 stream surface according to Wu (1952)

Figure (1) also shows the axial (X), circumferential (Y) and radial (r) directions. Also, it defines the angle (Θ) used to define sweep along radius.

The usual simplified analysis following Louw et al. (2012) is done assuming discrete series of two-dimensional flow layers at S-1 surfaces or radial station.

B. Inlet and Outlet Flow Angles

Assuming shock free inlet conditions with no incidence angle, the inlet velocity can be computed as shown in Figure (2);

$$C_1 = C_{x1} = Q/A \qquad (1)$$

And it is constant for all radial stations and vortex indices.

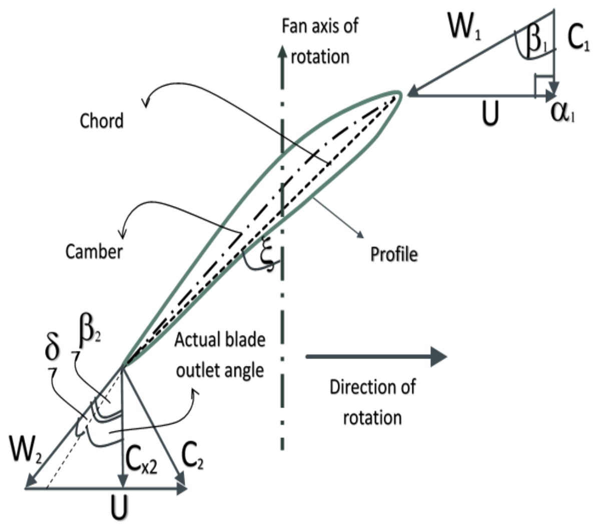

Figure (2) Cross-section of a fan blade showing the velocity vectors and angles for a rotor-only arrangement

To compute the outlet flow conditions which vary along a radius according to the radial equilibrium theory, two constraints must be satisfied:

1) Required flow rate and hence axial velocity distribution should be governed by the radial equilibrium condition.

2) Required pressure rise is governed by Euler's equation.

These two constraints are implemented through two iteration loops. The inner loop finds the outlet axial flow velocity, which gives the required flow rate by the following equation, Louw et al. (2012):

$$C_{x2}(r) = \sqrt{2\Omega a(r^{n+1} - {r_h}^{n+1}) - R(r)a^2(n+1) + {C_{x2}}^2(r_h)} \quad (2)$$

where

$$R(r) = ln\left(\frac{r}{r_h}\right)^2 \quad \text{for} \quad n = 0$$

$$R(r) = \frac{r^{2n} - {r_h}^{2n}}{n} \quad \text{for} \quad n \neq 0$$

This loop continues until C_{x2} matches required flow rate from the relation

$$m^{\cdot} = 2\pi\rho \int_{rh}^{rt} C_{x2}(r).rdr \approx 2\pi\rho \sum_{rs=1}^{n_{rs}} C_{x2}r \quad (3)$$

The outer loop finds the constant 'a' in the vortex distribution equation:

$$C_{u2} = ar^n \quad (4)$$

The calculations of 'a' continues through the inner and outer loops until it finds the appropriate value of 'a' that matches the required total pressure rise p_0 through the following relations; Louw et al. (2012)

$$Work = w(r) = \Omega r C_{u2}(\mathrm{r}) \quad (5)$$

$$Power = P_{av} = 2\pi\rho \int_{rh}^{rt} C_{x2}(r)w(r).rdr \approx \sum_{rs=1}^{n_{rs}} w_{rs} m^{\cdot}_{rs} \quad (6)$$

$$p_0 = P_{av}/Q \quad (7)$$

By knowing C_{x2}, U and C_{u2}, the outlet flow angle β_2 can be calculated. Then, the chord length is computed by inserting the loading factor constraint (Howell expression) mentioned in Wallis (1968)

$$C_L = 2\,[\cos(\beta_1)/\cos(\beta_2)\,]^{2.75} \quad (8)$$

To get the chord length L, Louw et al. (2012) deduced the following equations;

$$L = 2S(C_{u2}/C_{xm})\cos(\beta_m)\,/\,C_L \quad (9)$$

$$\tan(\beta_m) = \tan[0.5(\beta_1+\beta_2)] \quad (10)$$

$$C_{xm} = 0.5(C_{x1}+C_{x2}) \quad (11)$$

$$S = 2\pi r/n_b \quad (12)$$

The above angles are the flow angles; the blade outlet angles are calculated by allowing for deviation. From Figure (2),

$$\text{Actual blade outlet angle} = \beta_2 - \delta \quad (13)$$

$$Where\ \delta = m(\beta_1 - \beta_2)\sqrt{\frac{S}{L}} \quad (14)$$

where m is replaced by a value of "2" to give a value of δ in the range of 6 to 12 degrees as recommended by Wallis (1968).

C. Outlet Vortex Index Optimization

Various vortex index values (from −1 to 0) were iterated to get the design outlet flow angle and loading distribution coefficient along radial stations. The one with most degree of uniform loading characterized by small deflection $(\beta_1 - \beta_2)$ was chosen.

D. Camber Line and Profile

A good camber line shape should meet the criteria of shock free entry, continuous, and tangent to flow direction.

Following Pascu (2009), calculations were done based on the profile angle $\bar{\beta}$, which represents the angle between the tangent to flow and circumferential direction instead of flow angle β

$$\bar{\beta} = 90° - \beta \quad (15)$$

The computation of the blade shape was carried out considering the Cartesian system coordinates and is depicted in Figure (3).

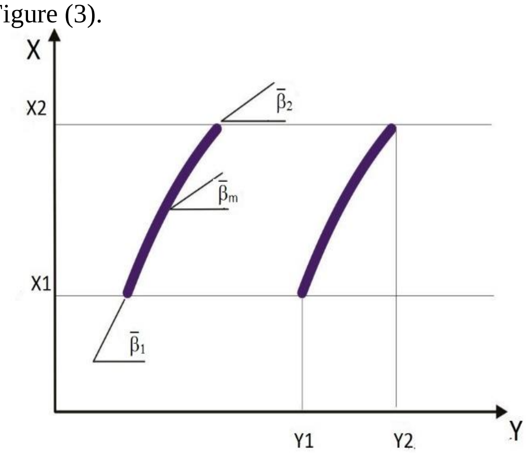

Figure (3) Blade camber line calculation

The indicated angles $\bar{\beta}_1$, $\bar{\beta}_m$ and $\bar{\beta}_2$ presented in Figure (3) are the inlet, the intermediate and the outlet profile angles, respectively. Each of the three stations on the profile is characterized by corresponding coordinates (X, Y). X and Y represent the axial direction and circumferential direction, respectively.

The blade shape at a specified section can be computed by relating the Y coordinate and angle distribution $\bar{\beta}(Y)$, Figure (3), since

$$\tan \bar{\beta}(Y) = \Delta X/\Delta Y \qquad (16)$$

The distribution is assumed parabolic because this assumption is appropriate for ducted axial fans as recommended by Pascu (2009).

Considering such a distribution,

$$\bar{\beta}(Y) = C_1 X^2 + C_2 X + C_3 \qquad (17)$$

Since both the inlet and outlet conditions are fixed, the blade shape computation is carried out with only one degree of freedom given by ($\bar{\beta}_{mid}$, Y_{mid}). The geometry is iterated until the axial chord length constraint is matched.

E. Thin Airfoil Theory

Following Turner (1966), the isolated thin airfoil theory should be applied instead of cascade relations for low pressure fans.

Following Csanady (1964), the pressure distribution on the upper and lower airfoil surfaces along with the pressure coefficient (C_P) was computed using the thin airfoil theory.

$$C_P = 1 - \left(\frac{W}{W_{inf}}\right)^2 \qquad (18)$$

$$\frac{W}{W_{inf}} = \frac{1}{\sqrt{1+\left(\frac{dZ}{dX}\right)^2}}\left(1 + \frac{u}{W_{inf}} \pm \frac{\gamma(\theta)}{2W_{inf}}\right) \qquad (19)$$

where the negative sign in Equation (19) is used for pressure on the lower surface and positive sign used for suction or upper surface. γ and u are the vorticity distribution representing the camber line and the induced velocity due to thickness, respectively. Csanady (1964) deduced the following equations for the vorticity distribution.

$$\frac{\gamma(\theta)}{W_{inf}} = 2\left[A_0 \frac{1+cos\theta}{sin\theta} + \sum_{n=1}^{\infty} A_n \sin(n\theta)\right] \qquad (20)$$

$$A_0 = \alpha - \frac{1}{\pi}\int_0^{\pi} \frac{dY}{dX}\, d\theta \qquad (21)^{1}$$

$$A_n = \frac{2}{\pi}\int_0^{\pi} \frac{dY}{dX} \cos(n\theta)\, d\theta \qquad (22)$$

The thickness distribution (Z) is always normal to the camber line curve and its drop-shaped profile is defined by Csanady (1964),

$$Z = \frac{8b}{3\sqrt{3}L}\left(1 - \frac{X}{L}\right)\sqrt{X(L-X)} \qquad (23)$$

$$\frac{dZ}{dX} = \frac{4^{b}/_{L}}{3\sqrt{3}}\frac{L-4X}{X}\sqrt{\frac{X}{L}\left(1-\frac{X}{L}\right)} \qquad (24)$$

$$\frac{u}{W_{inf}} = \frac{\frac{4b}{L}}{3\sqrt{3}}\left(3 - 4\frac{X}{L}\right) \qquad (25)$$

where $b = 2Z_{max}$ occurring at 25% of the chord length, which is a rough analytical approximation for the parametric nature of the NACA airfoil series profile.

(1)The integration by the angles corresponding to affected points and affecting points forming vortices along chord is nearly equal and all integration angles use θ as McCormick (1995) used.

The ideal incidence angle α_{ideal} for the certain camber line is given by:

$$\alpha_{ideal} = \frac{1}{\pi}\int_0^{\pi} \frac{dY}{dX}\, d\theta \qquad (26)$$

Also, the loading coefficient C_L ($L/0.5\rho W_{inf}^2$) can be calculated from the thin airfoil theory;

$$C_L = 2\pi A_0 + \pi A_1 \qquad (27)$$

All the above analytical equations including integration and derivative are converted to a numerical code by:

- Discretizing X, Y and Z domains through a finite resolution.
- Transforming X and Y coordinates, so that X aligns with chord line and the origin lies on the leading edge.
- Get θ values from X values by:

$$\theta = \cos^{-1}(1-2\,X/L) \qquad (28)$$

A Matlab code (Mathworks Inc. Matlab, 2015) was implemented to find the required thin airfoil performance parameters.

Obviously, this theory is only an approximation, especially the value of loading coefficient. This value is not accurate enough and further refinement using the vortex panel method by dividing camber in a number of panel sections, but it is used here just for finding the optimum angle of attack and finding an approximation for the pressure distribution and loading coefficient.

F. Optimum Stagger Angle

The last stage in designing fan blades is to find the optimum stagger angle at every radial station by iterating for various stagger angles at every radial station until the angle of attack α_{att} from the assumed stagger angle generates an ideal flow angle from thin airfoil theory $\alpha_{att} = \alpha_{ideal}$

$$\alpha_{att} = \beta_{inf} - \xi \qquad (29)$$

$$\beta_{inf} = \tan^{-1}[0.5*(\tan\beta_1 + \tan\beta_2)] \qquad (30)$$

which is the classical definition of mean stream direction used by Csanady (1964) to define main free stream velocity (W_{inf}) and hence, angle of attack (α_{att}).

G. Program Modules Summary

A new design and analysis code, which is divided into seven function modules, is implemented in Matlab by the authors as follows:

1. Main function: handles the user interface and collects design parameters from the user. Then, calls the design functions and text output function and draws the results on a graph built in the user interface window.
2. InitAngArbVort function: calls InitAng function for each outlet swirl distribution and chooses the one, which satisfies the best uniform chord length through radial stations.
3. InitAng function: computes the inlet conditions from required flow rate and shock free entry, and the outlet conditions from required pressure rise and outlet swirl distribution for the specific vortex index.
4. Profile3D function: generates the profile at each radial station by calling Profile2D function and

ThinairfoilPerform function and chooses the stagger angle for ideal incidence.

5. Profile2D function: generates profiles for the required inlet and outlet blade angles and the required stagger, which satisfies the required blade chord.

6. ThinairfoilPerform function: computes the ideal loading, ideal loading angle, and pressure coefficient based on relative main free stream flow velocity.

Out function: stacks the airfoil on the center of the rectangle that bounds first radial station and outputs the geometry of the designed fan to a text file.

III. MATH

To verify the Matlab code, an existed axial fan of ammonia cooling tower fan in El-Delta for Fertilizers Company in Talkha City (Dakahlia, Egypt) was studied and an upgrade of its design is implemented to increase the efficiency (i.e. more pressure drop for the same flow rate and power consumption constraints).

A. Existed Fan Description

The existed cooling tower fan has 581 m^3/s at 175 Pa of total pressure (100 Pa of static pressure and 75 Pa of wasted dynamic pressure computed from the average exit velocity). The fan has outlet velocity recovery stack, while the fan brake power is 138.4 kW and motor power is 160 kW.

According to Hudson Corporation (2000), the velocity recovery stack converts up to 70% of the dynamic energy at exit plane to useful work, which saves more than 10% of consumed power. When analyzing without the stack, it is assumed that the aerodynamic power for no recovery stack equals the total power with the stack.

The existed fan has 8.5 m outer diameter and 2.42 m inner diameter. It has 8 blades with a constant chord length of 0.407 m and a stagger angle value of 20° measured from circumferential direction, which is constant along the blade length. The profile is straight at upper (pressure) surface and curved at lower (suction) surface with a maximum thickness of 8% chord.

B. Hub-Tip Ratio Optimization

Figure (4) shows the relation between the dynamic pressure and the required power versus hub-tip ratio, *Xh*, for the studied case deduced from various runs of the Matlab code.

From Figure (4), it is obvious that smaller hub diameter is necessary to reduce power and waste pressure to the atmosphere.

The hub-tip ratio is a controlling parameter in the Matlab code to obtain best design. The user chooses the lowest inner diameter with the better uniform loading factors along radius deduced from the thin airfoil theory. It is shown in Figure (5) that the ratio $Xh = 0.3$ is very close to that of the existed design.

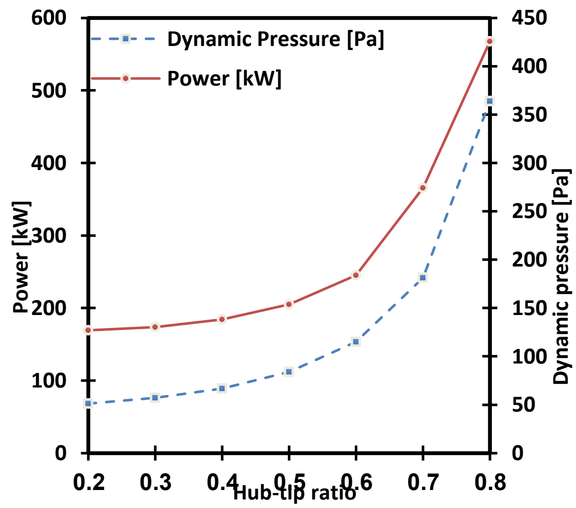

Figure (4) Hub-tip ratio effect on power and dynamic pressure

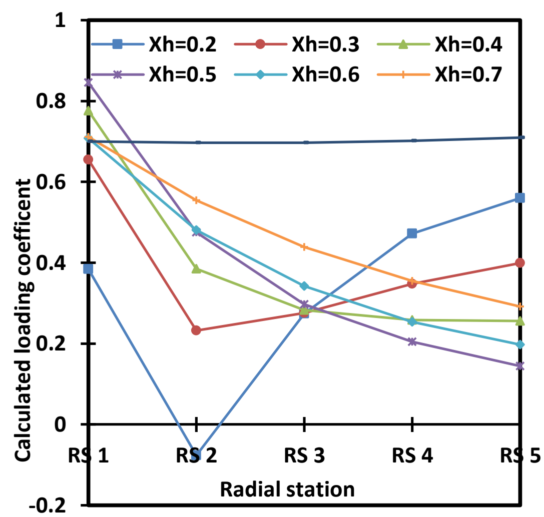

Figure (5) Hub-tip ratio effect on loading distribution across radius

C. New Design

Flow rate, speed, density and outer diameter are the basic design parameters. The only left parameter is the inner diameter which is assumed and then checked for power and loading distribution until getting uniform loading distribution with low power consumption. Inner diameter was chosen as described in the previous section. The chosen inner diameter is 2.5 m.

Matlab design code gives fan geometry in an output text file that contains the blade camber line definition at five radial stations. Only inner and outer radial stations are shown in Table (1) and for practical considerations only these two stations are taken.

TABLE (1)
NEW DESIGN FAN CAMBER LINE DEFINITIONS

Geometric Aspect	Radial Station 1	Radial Station 2
Radius, r (m)	1.25	4.25
Chord Length, L (m)	0.615	0.615
Stagger Angle, ξ (°)	49.8	75.9
□ at beginning (°)	□9.02	□1.01
Axial Position 1, X1 (m)	□0.1985	□0.0751
Axial Position mid, Xmid (m)	0.0	0.0
Axial Position 2, X2 (m)	0.1985	0.0751
□1 (°)	57.81	79.51
□mid (°)	50.24	76.24
□2 (°)	35.24	61.90

IV. CFD VERIFICATION

Before simulating the new design to get its performance parameters, the existed fan was simulated and compared with its actual results in the field.

The axial fan geometry was drawn in Bladegen (Ansys Inc., 2015) with the same dimensions of the existed one. The geometry is shown in Figure (6), which clarifies the constant profile and stagger angle designed at mean line.

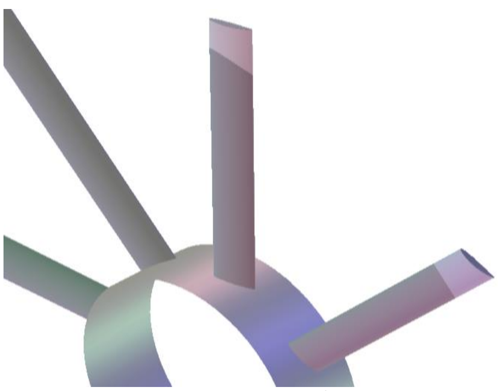

Figure (6) Bladegen model of the existed design

This geometry was imported to Ansys Turbogrid (Ansys Inc., 2015) and meshed as shown in Figure (7).

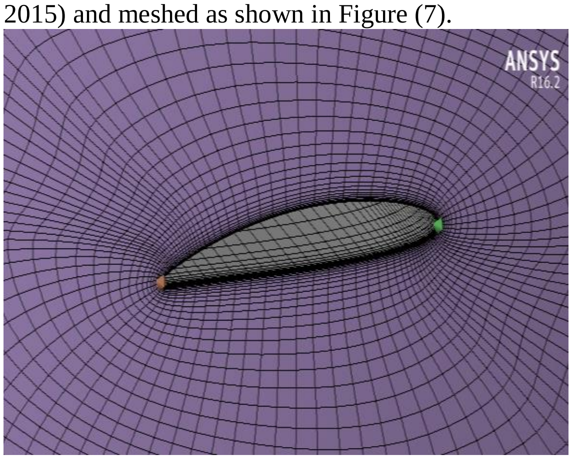

Figure (7) Mesh generated by Ansys Turbogrid of the existed design

This mesh was analyzed using Ansys CFX (Ansys Inc., 2015) with total inlet pressure and static outlet pressure boundary conditions. Total inlet pressure represents system resistance and static outlet pressure represents atmospheric pressure.

Two meshes were solved and compared as shown in Table (2). It is clear that there is no need for more refinement.

TABLE (2)
COMPARISON BETWEEN TWO MESH SIZES FOR EXISTED DESIGN

Mesh Size (Nodes)	Flow Rate (m3/s)	Power (kW)	Total Efficiency (%)
930,000	561	139.1	82.0
1,500,000	552	132.4	82.3

The results agree with the actual known conditions to a good degree of accuracy (for flow rate and pressure error was ~5%). Now, the numerical model is verified and it can be applied to the new designed fan model.

Firstly, the high resolution method was used but the solver prompts reverse flow error and the solution did not converge. This region of reverse flow is the main disadvantage of the usual mean line design.

To get the actual conditions, the solver was restarted using upwind method, which converged with no errors.

Only total to total efficiency is considered here. Fan total to total efficiency equals the actual total pressure rise divided by theoretical total pressure rise.

$$\eta_{00} = (\Delta p_{00})_{\text{actual}}/(\Delta p_{00})_{\text{theoretical}} \qquad (31)$$

It has been assumed that η_{00} = 80% in the design Matlab code to account for more pressure rise when designing for new fan. It has not been deduced for new design in the Matlab code. It has been deduced only from CFX simulation.

V. RESULTS AND DISCUSSION

Axial fans used in cooling tower usually have large outer diameters to draw most possible flow rate and small inner diameter to have small exit velocity for the same static pressure. In other words, not all fans have pressure recovery devices and the fan output is the total pressure.

Now, it is obvious that smaller hub diameter is necessary to reduce waste power and pressure to the atmosphere. This makes a challenge for popular mean line design that is attractive for ease of analysis and manufacture, but has the following disadvantages for small hub diameters:

- Non-uniform loading across the span, which may lead to the extreme case of reverse flow close to the fan hub.
- Non-uniform loading across the span which leads to the reduction of the performance of the fan as some areas do not contribute to pressure rise or have small share.

The proposed design has the following main characteristics:

- Controlled vortex distribution,
- Free blade profile.

A. *Proposed Design Simulation*

Only the inner and the outer stations were inserted to Ansys Bladegen manually to model the new design shown in Figure (8).

The thickness distribution according to Pascu (2009) for low pressure fans does not have great effect on efficiency.

NACA four digits airfoil with 10% maximum thickness at 75% chord was chosen in the present simulations.

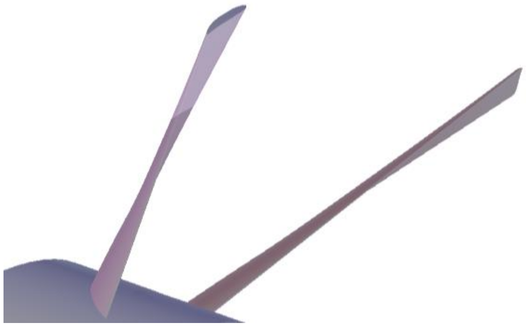

Figure (8) Bladegen model of the new design for constant chord and variable stagger angle

This shape was meshed as before by Ansys Turbogrid to obtain a new mesh with 970,000 nodes mesh and 1,500,000 nodes mesh. Then, the model is solved by Ansys CFX using a high resolution method which converges successfully (no reverse flow).

Total inlet pressure was used as the changing parameter to get the performance curves like existed fan simulation.

The changes in the flow rate value for the two mesh cases were small (570.5 and 573.1 m^3/s) as shown in Table (3). So, there is no need for further refinements.

TABLE (3)
COMPARISON BETWEEN TWO MESH SIZES AND DESIGN VALUES FOR NEW DESIGN

Mesh Size (Nodes)	Flow Rate (m3/s)	Power (kW)	Total Efficiency (%)
970,000	570.5	131.3	89.7
1,500,000	573.1	131.2	90.0
Design Program Values	581.0	142.1	80.0 (Assumed)

Figures (9), (10) and (11) are comparisons between the existed design and new design. It is clear from the results that the two simulations have the same trend results, but with increments in total efficiency and flow rate and decrement in power consumption for the new design.

The efficiency at operating point has been raised from 82.3% to 90% in the new designed fan. There is about 7% increase in flow rate and efficiency along all simulation range. Therefore, more flow rate with more pressure difference is obtained by less consumed power.

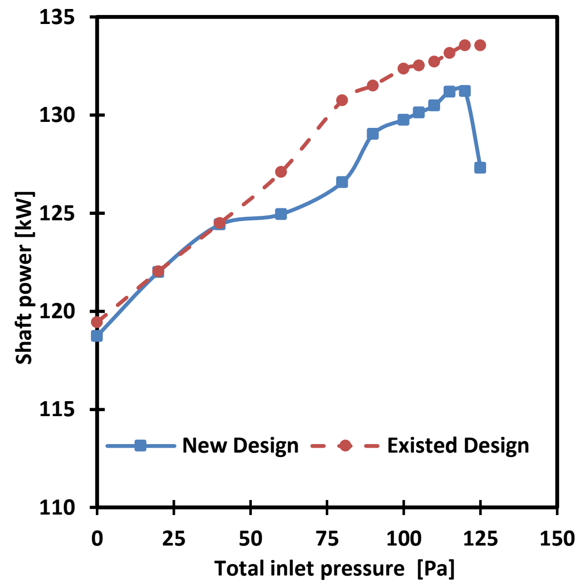

Figure (9) Flow rate vs. total inlet pressure for existed and new designs

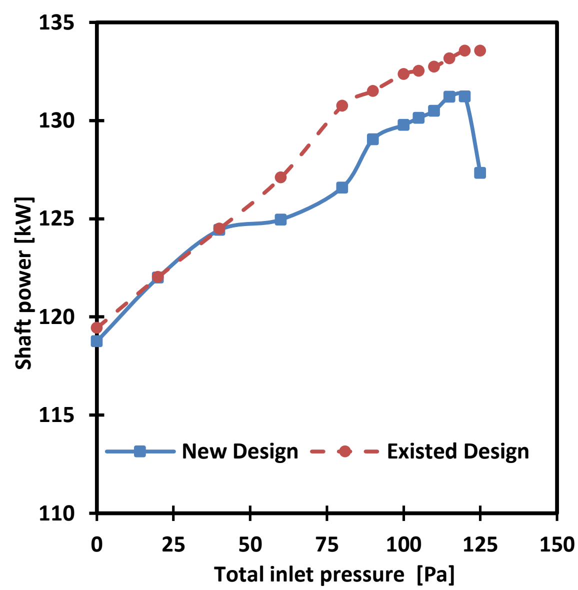

Figure (10) Shaft power vs. total inlet pressure for existed and new designs

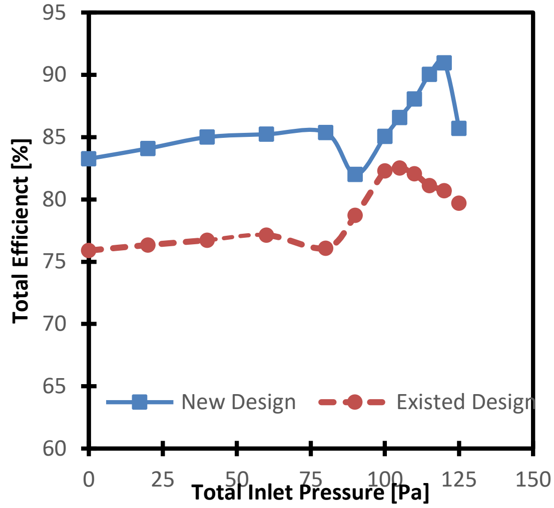

Figure (11) Total efficiency vs. total inlet pressure for existed and new designs

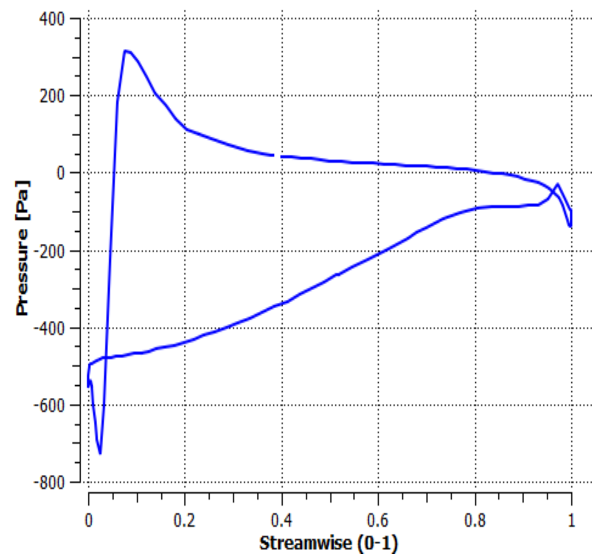

Figure (12) Blade loading at 20% span for existed design (Small pressure share for inner radii)

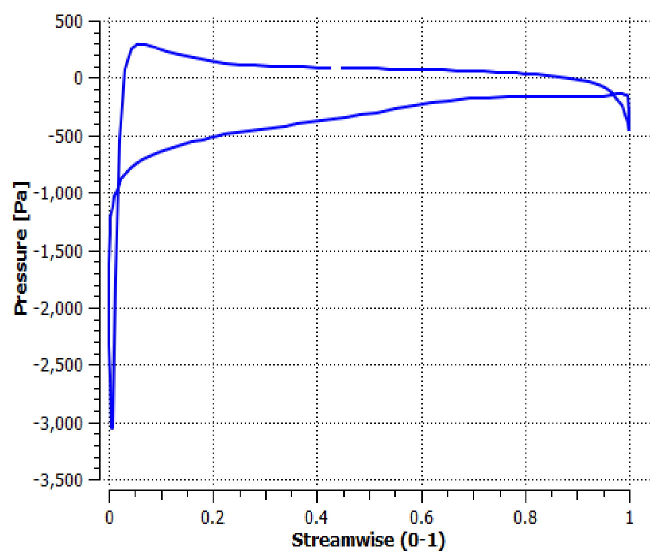

Figure (13) Blade loading at 20% span for new design (High pressure share for inner radii)

B. Controlled Vortex Distribution

To overcome the problem of mean line design (non-uniform loading distribution leading to reverse flow at small hubs); a controlled vortex distribution at blade exit section was applied. This is complicated to solve analytically.

The usual way is to solve numerically by taking discrete radial stations and apply cascade or isolate airfoil relations at these stations and link these stations with radial equilibrium theory and assuming simple vortex distribution as in Equation (4).

The popular vortex distribution is the free vortex where $n=-1$, which is a step forward toward the advanced design, but it has the disadvantages of very large chord line at hub compared to that at tip if it is required to have uniform loading along radial stations, (Lewis, 1996).

The vortex distribution index is optimized to have the uniform loading across all radial stations as well as uniform chord lengths suggested by empirical design relations.

This should happen automatically and the criterion used is the difference between inlet and outlet blade angles at the hub and tip radial stations only for ease of meshing and manufacture. These two stations are considered when modeling the fan, but the choice of only two radial stations causes non-uniform loading coefficient calculated from thin airfoil theory as shown in Figure (5). So, when verifying blade loading at all radial stations, the hub diameter is changed manually to get the most degree of uniform loading. It is found that the uniform loading is achieved at small hub-tip ratios. Of course, high hub-tip ratio cannot be chosen as this needs more power and very low hub-tip ratio gives a negative loading coefficient for the chosen vortex distribution.

Figures (12) and (13) (part of the automatic fan report generated by CFX) shows the difference between loading at 20% of span for existed and new design where it is seen that more pressure difference is achieved at the inner radii for the new design. This is our goal, that is to distribute pressure rise at all radial stations, not only outer radii like existed mean line design.

C. Free Blade Profile

The free blade profile means that the blade camber line is computed analytically with no usage of published airfoil data to allow smooth and efficient flow deflection. Also, stagger angle is chosen to satisfy the ideal angle of attack from thin airfoil theory. Figure (14) shows a sample of an output profile generated by the Matlab code.

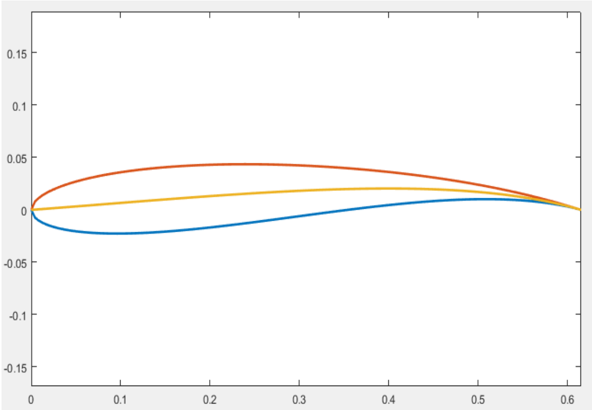

Figure (14) Fan blade profile generated by Matlab code, (dimensions in meter)

VI. Conclusion

Designing of a new cooling tower axial fan has been performed. Also, the analysis processes in the flow wise plane and the meridional plane (in the radial direction) were done.

The flow in meridional plane is governed by the radial equilibrium theory. Moreover, designing at mean radius, assuming constant flow conditions is not the best choice for low hub-tip ratios like those in large cooling tower fans and even free vortex design is impractical, as it requires large chord length variations along radius. The solution is the optimization of controlled vortex design to find vortex index, which matches most uniform chord lengths along radius.

The blade shape in the flow wise plane or cascade plane is chosen to efficiently deflect the flow from inlet angle to outlet angle. Therefore, camber line shape is deduced assuming second order polynomial variation of the angle between the camber and the circumferential direction, which simplifies the new design and does the job of efficiently guiding flow. This analysis is done via thin airfoil theory.

This methodology proved to have high efficiency when using the CFX study for the deduced design. The new designed fan is capable to deliver more air flow or the same flow rate at higher pressures with the same power requirement.

Nomenclature

Symbols:

A	Area [m^2]
A_0	Constant
A_1	Constant
A_n	Constant
a	Swirl coefficient
b	Blade maximum thickness, $b = 2Z_{max}$ [m]
C	Absolute velocity [m/s]
C_D	Drag coefficient
C_L	Loading coefficient
C_P	Pressure coefficient
L	Chord [m]
m	Constant determined experimentally
$\dot{m}$	Mass flow rate [kg/s]
n	Swirl exponent
n_b	Number of blades
P	Power [W]
p	Pressure [Pa]
Q	Volume flow rate [m^3/s]
R	Dimensionless radius
r	Radius [m]
S	Blade spacing (distance between airfoil leading edges, perpendicular to the axis of rotation) [m]
U	Circumferential velocity of fan blade [m/s]
u	Induced velocity due to thickness [m/s]
W	Relative velocity [m/s]
w	Work [J/kg]
X	Blade axial position
Xh	Hub-tip ratio
Y	Blade circumferential position
Z	Blade thickness distribution around camber line

Greek Letters:

α	Absolute angle [°]
β	Relative flow angle [°]
$\bar{\beta}$	β complementary angle [°]
γ	Vorticity distribution representing camber line
δ	Outlet deviation angle [°]
η	Efficiency
Θ	Angle from arbitrary radius to the projection in the (r, Y) plane [°]
θ	Coordinate dimension conversion, $\theta = \cos^{-1}(1-2X/L)$
ξ	Stagger angle [°]
ρ	Density [kg/m^3]
Ω	Rotational speed [rad/s]

Subscripts:

0	Total condition
00	Total condition to total condition
1	Inlet condition
2	Outlet condition
av	Average value in meridional plane
att	Attack
h	Hub
inf	Mean free stream direction
m	Mean value in cascade plane
max	Maximum
mid	Mid distance between inlet and outlet
rs	Radial station
t	Tip
U	Circumferential (swirl) direction
X	Axial direction

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