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EXPERIMENTAL DATA ANALYSIS OF FIVE MOORE 10000 CLASS FANS USED IN AIR COOLED HEAT EXCHANGERS

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ABSTRACT

Air-cooled heat exchanger that included the Moore 10000 class fans and tube bundle are used to cool the sour gas in Salman gas production platform in Iran. The objective of these fans is to produce enough pressure to overcome the friction through the bundle and proper air volume to cool the sour gas. Even if Number of blades give the rough estimation of produced pressure but blade angle is an important parameter to optimum the number of blades and to prepare an laminar flow through the blades to avoid the sever vibration. Effect of blade angle on static pressure, efficiency and flow rate for five models of these fans has been illustrated here. With increasing the blade angle the Static pressure rise and reach to maximum value, after that more increased in blade angle cause the static pressure start to drop off sharply. It is because of turbulent effect of flow on the blade that called the staling or burbling flow. Cyclic nature of this staling flow causes the severe vibration on blade. The flow rate and power consumption will increase while the blade angle increases. Howell criteria give an upper estimation of nearly 15-20 percent for losses of energy. For model 2 and model 3 that have four number of blades so lower relative to other models, deviation of Howell energy losses relative to real (measured) energy losses are high so Howell criteria is not applicable to axial fan with low number of blades. With increase in number of blades the Howell energy losses will close to real value. Model 2 with seven numbers of blades shows this fact. Looking through the variation of flow coefficient and stage loading factor with blade angle it shows the maximum efficiency nearly in flow coefficient of 0.35 and loading factor of 0.25 will happen.

Keywords: axial flow fans, losses, energy, efficiency, blade angle, static pressure.

1. INTRODUCTION

Moore 10000 class fans are the axial flow fans that are used in air coolers to cool the sour gas in Salman gas production platform in Iran. The experimental data from the factory test and performance test of these equipments have been analyzed here. The main objective of these fans is to produce enough pressure to overcome the friction through the bundle and to prepare proper air volume to cool the sour gas. Number of blades give rough estimation of produced pressure but blade angle is important parameter to optimum the number of blades and to prepare an laminar flow through the blades to avoid the sever vibration. The main purpose of this paper is to determine effect of changing blade angle on static pressure, flow rate, efficiency and energy losses in these fans and comparing these energy losses with cascade test result of Hawell [1] (1942). These type of fans have a high space-chord ratio so simplified theoretical method based on isolated aerofoil theory is often used, Wallis 1961 [4]. Attempts have been made to extend the scope of isolated aerofoil theory to less widely space blade by the introduction of an interference factor, Weing, 1935 and Wislicenus, 1947 [2]. Van Niekerk also used the blade elementary theory to design the duct fans, 1958 [3]. Here we are intending to analyze this important experimental data to have a basis for judgment of the data that will be obtained from theoretical or numerical modeling.

1.1 Fan parts and datasheets

At this section is tried to describe the fan parts and general view of fan coolers. These coolers consist of two major parts, tube bundle on the top and fan inside the plenum. Fans produce the air volume with enough pressure that passes through the tube bundle and cause to cool the sour gas inside the tube.

Figure-1. General view of air cooled heat exchanger, the tube bundles are on top and the fans are inside the plenum.

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Figure-2. General view of plenum with blades inside.

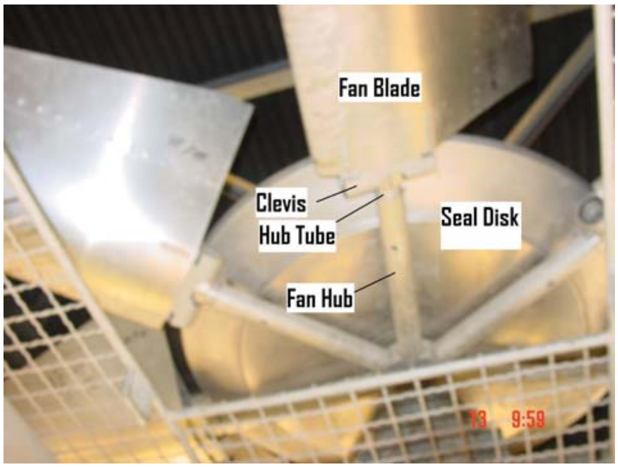

Figure-3. Major parts of fans.

Figure-3 shows major parts of fans like, air seal, fan hub, hub tube, clevis etc.

For better imagination of performance and size of fans the fan datasheets have been come in Table-1.

Table-1. Fan datasheet for five models of Moore fans used in Salman gas production platform, Iran.

	Model 1	**Model 2**	**Model 3**	**Model 4**	**Model 5**
Fan diameter (mm)	**3658**	**3353**	**3048**	**2743**	**3962**
Number of blades	6	7	4	4	6
Seal disk dia (mm)	2654	2512	1102	993	2871
Static pressure needed (Pa)	152	245	115	187	184
Flow rate needed (m^3/sec)	108.2	76	68.1	44.8	93.7
Speed (RPM)	265	280	319	354	245
Power absorbed (kW)	27.4	27.3	13.2	12.8	27.5
Blade angle (Degree)	24.9	22.8	24.5	22.4	19.6
Tip clearance (mm)	10	8	8	7	10

Clevis is used to adjust blade angle. To adjust the blade angle loose the hub tube bolt enough to allow the Clevis to be turned. See Figure-4.

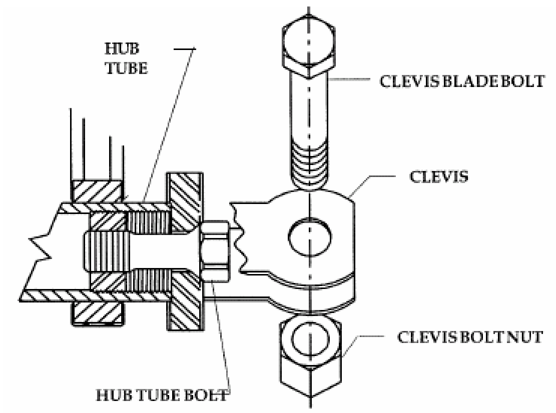

Figure-4. Parts of fan for adjusting the blade angle.

For measurement of blade angle, place the protractor level on the flat upper or lower surface of the Clevis as shown in Figure-5.

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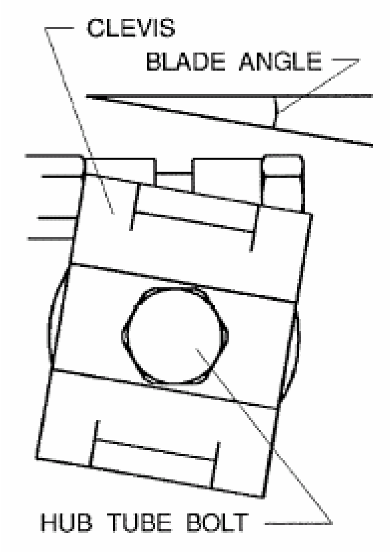

Figure-5. Clevis and blade angle measurement.

2. METHODOLOGY

2.1 Blade load angle measurement

To check the blade load a complete field run test have been done on the fans. The blade angle has been adjusted to an angle of approximately half that called out on the specification or measured on the unit. The draft gauge is connected to as quiescent a spot in the plenum as possible, preferably in the corner of the plenum ahead of the fan. The fans are start and recorded on the chart provided the blade angle and the static pressure indicated. The blade angle is advanced by one or two degrees and these data are recorded again. The blade angle is increased and followed the procedure until the motor is fully loaded.

For modell class fan, Figure-6 shows the blade angle versus static pressure and power consumption. It will be noted that the static pressure will be consistently increasing with increased blade angle until the blade loading reaches maximum.

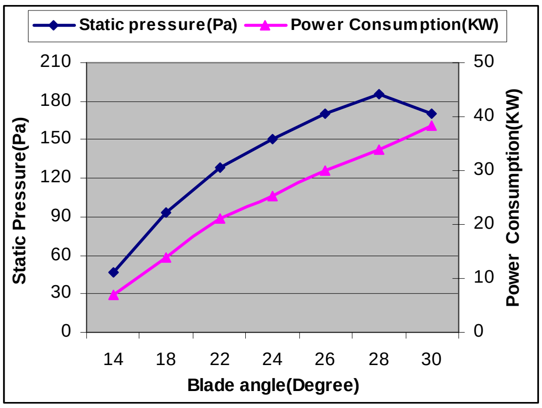

Figure-6. Static pressure and power consumption versus blade angle.

2.2 Flow measurement

The measurement of the air flow was carried out by using the anemometer to measure the velocity in various locations along the blade length and then multiply the average velocity by the measured area. Following sketch showing positions for air flow measurements at fan inlet.

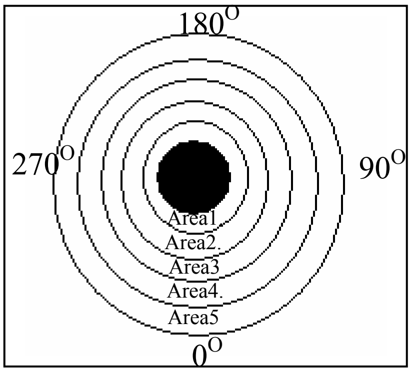

Figure-7. Schematic sketch to show how to measure the velocity.

The velocities were read at angles 45, 135, 225, 315 degrees. Measurement locations are major axes midpoint of five concentric equal areas annular. The area of each annuls equals the area of the fan-ring diameter minus the area of the seal disk divided by five, see the sketch.

3. DATA ANALYSIS AND RESULTS

The data from the field run test have been analyzed here in the following subsection:

3.1 Effect of blade angles on static pressure, power consumption and flow rate

With refer to Figure while the blade angle increases the static pressure increase up to maximum value. With more increased in blade angle the static pressure start to drop off sharply. So long as airflow over the blade is smooth and clings to the surface of the blade the little turbulent is present. With increasing the blade angle the air flow breakaway from the convex side of blade. This is known as stalling or burbling flow, since the air, instead of clinging to the blade breaks away near the leading edge and causes to make a turbulent flow on convex side of blade. When this occurs, the blade loses a large portion of its lift. Flow, however, will re-establish briefly and break again, the cycle being repeated continuously, resulting in a severe vibration throughout the fan as the flow alternately makes and breaks. With increasing the blade angle the power consumption and flow rate will increase but in stalling flow while the power consumption increase the increasing rate of the flow decrease and even in some point stop and remain fixed so it causes to reduce the efficiency in stalling flow. Figures 8, 9 and 10 show the change of static pressure, power consumption and flow rate with blade angle.

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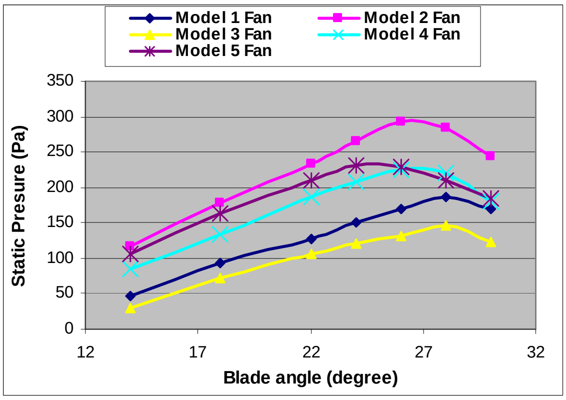

Figure-8. Static pressure vs. blade angle for different models of fans.

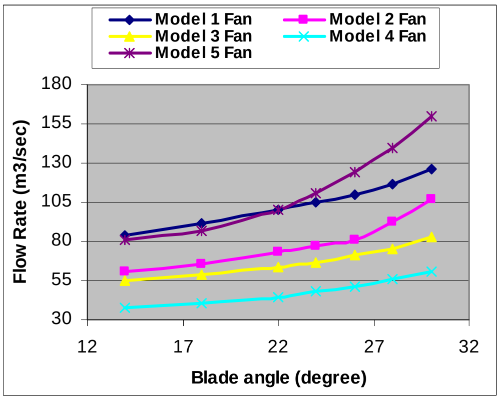

Figure-9. Flow rate vs. blade angle for different models of fans.

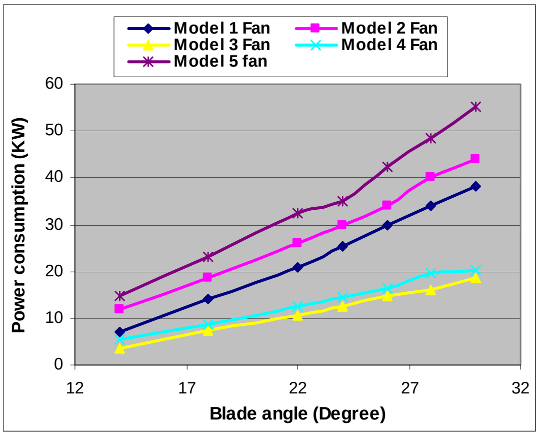

Figure-10. Power consumption vs. blade angle for different models of fans.

3.2 Effect of blade angle on efficiency

As we know change in kinematics energy is small so the input power to fan consumes to increase the pressure in reversible case. The efficiency can be defined as the power needed to produce the pressure change to power absorbed in real case that produce the same change in static pressure,

$$\eta = Q\Delta P / P_{ab} \quad (1)$$

η is the efficiency of fans; Q is volume flow rate, Δp is the change in static pressure and P_{ab} is the power absorbed in real case.

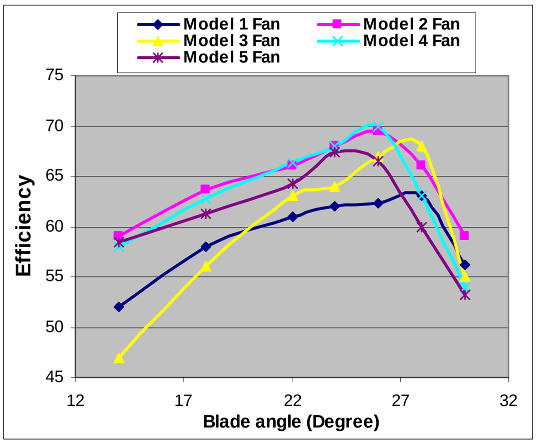

Figure-11. Efficiency vs. blade angle for different models of fans.

We can see from Figure-11 that the maximum efficiency obtains in the range of 22 to 26 degree of blade angle.

3.3 Energy losses and loss coefficient

A real fluid crossing the fan blades experiences losses in energy or total pressure due to skin friction, turbulent effects and other effects, so

$$P_{ab} = Q\Delta P + Q\Delta P_{loss} \quad (2)$$

So the pressure loss can be written as

$$\frac{\Delta P_{loss}}{\rho} = \frac{P_{ab}}{\rho Q} - \frac{\Delta P}{\rho} \quad (3)$$

A non dimensional form of pressure loss often is useful in presenting the results of fans data

$$\xi = \frac{\Delta P_{loss}}{0.5 \rho W_1{}^2} \quad (4)$$

W_1 is entry relative velocity.

3.3.1 Loss analysis, Howell correlation

Here we try to estimate the total pressure loss coefficient via Howell (1954) correlation which divided the total loss into three categories,

a) Profile losses on the blade surface
b) Skin friction losses on annulus walls
c) Secondary losses that means all losses not included in profile and skin friction losses.

We know that drag coefficient is obtained from

$$C_D = C_{Dp} + C_{Da} + C_{Ds} \quad (5)$$

If we use the empirical values for each part of drag coefficient, Howell [1]

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$$C_{Da} = 0.02S / H \qquad (6)$$

$$C_{Ds} = 0.018{C_L}^2 \qquad (7)$$

Where S is space between blades and H is blade height.
Correlation for lift and drag coefficient are [1]

$$C_L = 2.0\frac{S}{L}Cos\beta_m(\tan\beta_1 - \tan\beta_2) \qquad (8)$$

$$C_D = \frac{S}{L}(\xi)\frac{Cos^3\beta_m}{Cos^2\beta_2} \qquad (9)$$

$$\tan g\beta_m = 1/2(\tan g\beta_1 + \tan g\beta_2) \qquad (10)$$

C_{Dp} is calculate if we use the cascade test result, Howell 1942 [1] to determine the ξ_p and put this value in correlation (9).
S is chord length and is calculate by

$$S = \pi D / n \qquad (11)$$

To calculate the angles and relative velocity consider the Figure bellow

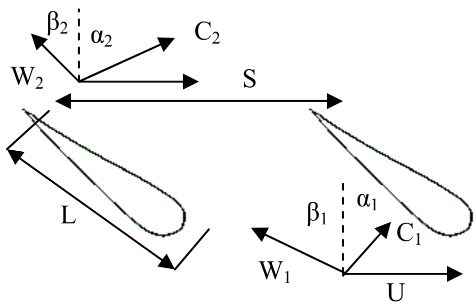

Figure-12. Typical velocity diagram for each fan.

C_1, W_1 are the inlet absolute relative velocity respectively, C_2 and W_2 are the absolute and relative exit velocities. U is the rotor velocity.
From the velocity diagram it is obvious that the

$$U = R_{ave}.\omega \qquad (12)$$

$$R_{ave} = (R_1 + R_2)/2 \qquad (13)$$

$$\tan\alpha_1 = \tan\beta_1 - \frac{U}{Cx} \qquad (14)$$

$$Cx = \frac{Q}{A} \qquad (15)$$

$$\tan\alpha_2 = \frac{P_{ab}}{\rho Q\, Cx} + \tan\alpha_1 \qquad (16)$$

$$\tan\beta_2 = \tan\alpha_2 + \frac{U}{Cx} \qquad (17)$$

Q is the volumetric flow rate, A is entrance area, P_{ab} is the fan power and C_x is axial velocity.
The C_x, U, P_{ab} and β_1 are the known parameter and the unknown parameters can be identified through the above equations.

Figure-13 shows the energy losses for model one fan that calculated by existed data and compare with energy losses using Howell correlation.

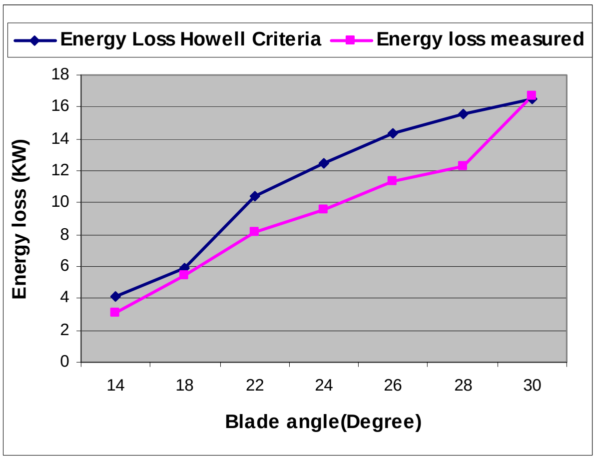

Figure-13. Energy losses calculated by existed data and Howell correlation.

Two other parameters that can be used in design of axial fans are flow coefficient (φ) and stage loading factor (ψ). With using following correlations, values of flow coefficient and stage loading factor can be obtained from,

$$\phi = \frac{C_x}{U} \qquad (18)$$

$$\psi = \frac{P_{ab}}{\rho Q U^2} \qquad (19)$$

In Figure-14 to find the values of φ and ψ in maximum efficiency the variations of flow coefficient, stage loading factor and efficiency have come in one chart.

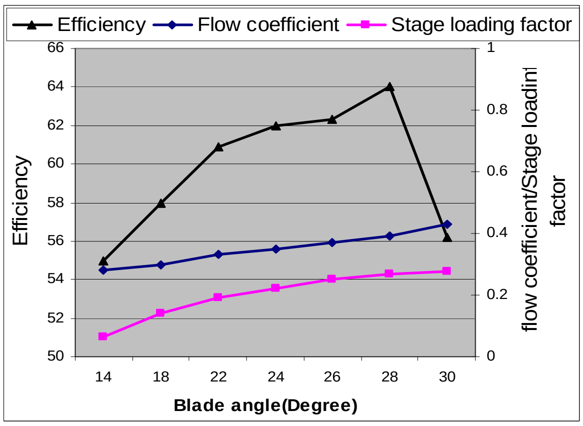

Figure-14. Variation of flow coefficient and loading factor with blade angle.

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Table-2. Values of real energy losses and calculated energy losses by Howell correlation for different models of fans.

Energy loss / Blade angle	Model 2		Model 3		Model 4		Model 5	
	E.M (kW)	**E.H** (kW)	**E.M** (kW)	**E.H** (kW)	**E.M** (kW)	**E.H** (kW)	**E.M** (kW)	**E.H** (kW)
14^o	3.6	4.8	1.8	2.2	2.3	3.7	6.1	7.3
18^o	7.5	6.7	3.3	5.3	3.3	7.05	8.9	15.08
22^o	10.2	8.8	3.8	8.4	4.2	9	11.6	18.6
24^o	11.4	9.6	4.5	9.6	4.6	9.5	11.7	18.7
26^o	12.3	10.4	4.6	10.4	4.8	10.1	14.2	19.7
28^o	12.7	13.6	5.1	11.3	7.2	10.7	19.4	19.9
30^o	12.3	17.8	8.3	11.7	9.3	10.9	25.8	19.3

E.M and E.H are respectively real (measured) energy loses and calculated energy losses by Howell correlation.

4. CONCLUSIONS

With refer to efficiency chart it can be conclude that maximum efficiency for these five model fans will occurred between blade angle ranges of 22 to 26 degree. The stalling flow that should be strongly avoided in axial Fans will start from blade angle 27 degree.

Energy losses through the blades by Howell correlation for fans with six blade number and more give an upper estimation of nearly 15-20 percent for losses of energy. For models 2 and model 3 that have four number of blades so lower relative to other models, deviation of Howell energy losses relative to real (measured) energy losses are high so Howell criteria is not applicable to axial fan with low number of blades. With increase in number of blades the Howell energy losses will close to real value. Model 2 with seven numbers of blades shows this fact (Table-2).

Looking through the variation of flow coefficient and stage loading factor with blade angle, Figure-14, it shows that the maximum efficiency in flow coefficient of 0.35 and loading factor of 0.25 will happen in these fans.

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