EFFECT OF BLADE STAGGER ANGLE ON THE VIBRATION CHARACTERISTICS OF AXIAL FAN IN TURBOMACHINARY SYSTEM BY FINITE ELEMENT METHOD USING ANSYS

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ABSTRACT

This paper discusses the dynamic behavior (natural frequencies, modes shapes) of the axial fan. The fan blade stagger angle are adjustable. Three angles 20^o, 35^o and 60^o are studied. Three dimensional finite elements with six degrees of freedom per node are employed to idealize the axial fan. ANSYS12 program is performed that takes the variables of the fan blade system. The obtained results reported the tenth structural natural frequencies and mode shapes which are based upon the behavior of fan, it can be shown that the natural frequency of the fan increased with increasing mode number and the amplitude decreases as the stagger angle increases. The results shows that the maximum amplitude occurred at stagger angle of 20° and have less error ratio.

تأثير زاويةِ تمايلِ النصلِ على خصائصِ إهتزازَ المروحةِ المحوريةِ في انظمةِ الماكنة النفاثة بطريقةِ العناصرِ المحددةِ باستخدام برنامج الأنسز

أثير زكي محسن /الكلية التقنية / المسيب – قسم هندسة المضخات

الخلاصة

تم دراسة السلوكَ الديناميكيَ (ترددات طبيعية، أشكال أنماطِ) للمروحةِ المحوريةِ. حيث ان زاوية تمايلِ نصلِ المروحةَ كَانتْ قابلة للتعديلَ. ثلاث زوايا 20 °، 35 ° و60 ° تم دراستها. حيث استخدمت طريقة العناصر المحدودة الثلاثية الأبعاد بستّة مِنْ درجاتِ الحريةِ لكلّ عقدةِ لتَمْثيل المروحة المحوريةِ. تم اعداد برنامج بلغة برنامج الانسز نسخة 12 لدراسة متغيّراتَ نظامِ نصلِ المروحة. بينت النَتائِجُ المستحصلة عن الترددات الطبيعيةِ العشرةِ وأشكالِ الانماطِ المقابلة لها حيث انها مستندة على سلوكِ المروحةِ، بأنّ الترددَ الطبيعيَ للمروحةِ زادَ كلما ازداد عددِ النمطِ وتتناقصِ السعة عند تزايد زاوية التمايلَ. وقد تم استنتاج ان السعة القصوى تحدث في زاويةِ تمايلِ 20 ° ولَها أقل نسبة خطأ.

INTRODUCTION

The subject of axial fan system and it is behavior was studied by many authors; several investigators have been studied the vibration of blade, fan, rotor, bearing and casing. Corsini et al,1999, presented a simplified model for the three dimension blade – to – blade flow developing in the rotor that was based on the numerical simulation of rotor flow. Vogt and Frasson,2002, studied the aerodynamic coupling in a single blade row environment by means of the aerodynamic influence coefficient technique. The aerostatic of a cascade can thus be determined by having only one blade oscillating and measuring the response on all neighbor blades. Franklyn Kelecy,2002, carried out a recent study on the ability of computational fluid dynamic CFD simulation to predict accurately fan performance over a wide range of flow rates. Roy et al,2005, aimed to improve the understanding of the effects of sweep on the losses by experimentally studying the (30°) parabolic forward swept blade in an axial flow compressor stage.

Most of the forgoing published work gave a greet deal to the dynamic response to a part from a rotary system theoretically and/or experimentally. Hence, it becomes essential to study the vibration characteristics of an axial fan system at difference stagger angle. This study is identified theoretically by using finite element method via ANSYS12 software which is studied the modal analysis of an axial fan, the dynamic characteristics of fan is studied using finite element method via ANSYS12 software. The fan structure is discretized using 4 node tetrahedral solid finite element having three displacements and three rotations as degrees of freedom per node. The eigenvalues and eigenvectors are obtained. The results of the modal analysis is presented as contour plots on the deformed configuration of the fan.

AXIAL FAN

The purpose of the study is to investigate vibration characteristics of an axial fan. The assembly of the photograph of the turbomachinary system is shown in **Fig.1.** it can be shown that a turbo-machine device which is important in many industries carrying the axial fan .

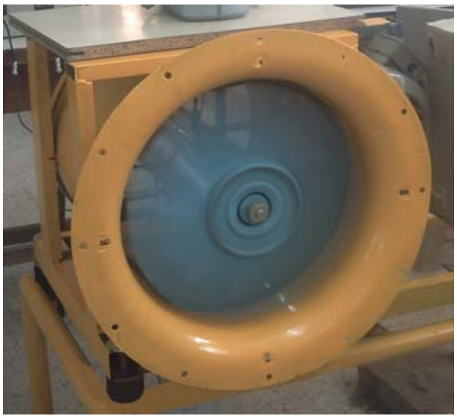

Fig.1. Photograph of the turbomachine system.(Wafa,2007)

The axial fan used in this test rig is usually used in turbomachinary devices, such as power plant exhaust gas towers, steel plant, pharmaceutical, fertilizer plant, oil / gas production (processing and transportation), chemical process, petrochemical / refinery, ……etc. (Fan and fan system,2004).

The study's fan consists of (7) blades aerofoil section (NACA 5309), each blade span is (114.35) mm , root chord is (86.64) mm , tip chord (46) mm , and maximum thickness of blade is (6) mm , which is located at 30 % from the leading edge(Franklyn Kelecy,2002), as shown in **Fig.2.**

Fig.2. The photograph of the fan .

The fan blade material is (AL-Cast–Alloy) which density (2650) Kg/m³ and a young modules equal to (76) GPa (Fan and fan system,2004), the rotor disk of fan is (150.75) mm diameter. The stagger angles of the blade can be changed during the test with angles (20°, 35° and 60°) .

FINITE ELEMENT EQUATION

The element equations of the undamped dynamic system can be expressed in general form: $[M]\{\ddot{\delta}_e\} + [K_e]\{\delta_e\} = \{F_{e(t)}\}$ (1)

where : $[K_e] = \int_{vol} [B]^t \cdot [D] \cdot [B] \, dvol$

(2)

the analysis assembles all individual element equations to provide stiffness equations for the entire structure or mathematically

$$[M]\{\ddot{\delta}\} + [K]\{\delta\} = \{F_{(t)}\} \quad (3)$$

Where: $[K] = \sum_{i=1}^{M} [K_e]$ (4)

$$[M] = \sum_{i=1}^{M} [M_e] \quad (5)$$

EIGENVALUE SOLUTION

Finite element method is used to analyze of the structural dynamic problems, an algebraic eigenvalue problem is obtained as stated in Eq.(9). For most engineering problems, [K] and [M] will be symmetric matrices of order n (Erik,1990). The solution of the eigenvalue problem in Eq.(9) is sought in the evaluation of an element stiffness matrix or in the calculation of the condition number of a structure stiffness matrix (Klaus,1996).

$$[M]\{\ddot{\delta}\} + [K]\{\delta\} = 0$$

(6)

Pre-multiplying by $[M]^{-1}$

$$-[I]\omega^2\{\delta\} + [D_o]\{\delta\} = 0$$

(7)

Where:

$[D_o] = [M]^{-1}[K]$ (8)

$$\lambda = \omega^2$$

$$[[D_o] - \lambda[I]]\{\delta\} = 0 \quad (9)$$

For non trivial $|[D_o] - \lambda[I]| = 0$ (10)

EIGENVECTOR SOLUTION

Finite element method is applied to find mode shape for a system. It can be express in the following equation:

$$[C] = [[D_o] - \lambda[I]] \quad (11)$$

$$[C]^{-1} = \frac{Adj[C]}{|C|} \quad (12)$$

Pre-multiplying Eq.(12) by $|C|[C]$:

$$|C| = [C]Adj[C] \quad (13)$$

$$\left|[D_o] - \lambda[I]\right| = \left[[D_o] - \lambda[I]\right]Adj\left[[D_o] - \lambda[I]\right]$$

If λ is one of the eigenvalues then :

$$\left|[D_o] - \lambda_i[I]\right| = \left[[D_o] - \lambda_i[I]\right]Adj\left[[D_o] - \lambda[I]\right] \quad (14)$$

The left side of the previous equation becomes zero hence:

$$0 = \left[[D_o] - \lambda_i[I]\right] Adj\left[[D_o] - \lambda_i[I]\right]$$

$$\{\delta\}_i = Adj\left[[D_o] - \lambda_i[I]\right]$$

$$0 = \left[[D_o] - \lambda_i[I]\right]\{\delta\}_i \quad (15)$$

hence the system equation can be written in the form :

$$[M]\{\ddot{\delta}\} = \{F\} - [K]\{\delta\} = \{F\} - \{F\}^{int} = \{F\}^{residual} \quad (16)$$

$$\{F\}^{int} = [K]\{\delta\}$$

(17)

$$\{\ddot{\delta}\} = [M]^{-1}\{F\}^{residual} \quad (18)$$

In practice , the above equation does not usually require solving of the matrix equation , since lumped masses are usually used which forms a diagonal mass matrix (Mario Paz,1990).

The solution to Eq.(18) is thus trivial , and the matrix equation is the set of independent equations for each degree of freedom i as follows:

$$\delta_i = \frac{f_i^{residual}}{m_i} \quad (19)$$

MODEL GENERATION BY ANSYS12

The ultimate purpose of a finite element analysis is to re-create mathematically the behavior of an actual engineering system(Saeed Mouveni,1999). In other words, the analysis must be an accurate mathematical model of a physical prototype (Tim Langlais,1999).In the broadest sense, the model comprises all the nodes, elements, material properties, real constants, boundary conditions and the other features that used to represent the physical system.

The two Dimension model is done by drawing and dragging to get three dimension model then meshing with element soild72 (Marimuthu,2007).

Solid 72 is a four-node tetrahedral element , with each node having three translational degrees of freedom in the nodal x– , y– , and z–directions , as well as rotations about the nodal in x–, y– , and z– directions , as shown in **Fig.3.** (ANSYS manual,2009 and Training manual,2010)

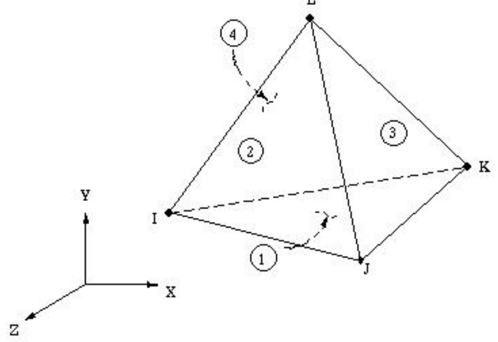

Fig.3. The Solid72 Element

Procedure is presented for modeling of axial fan system by ANSYS12 software by using solid-modeling approach method. By drawing disk and create keypoints to airfoil, these keypoints get from DESIGN FOIL program V6.02 , by entering the constant parameter that represent :

i- Max camber HEIGHT as percentage of chord.
ii- Max camber LOCATION as percentage of chord.
iii- Max AIRFOIL thickness as percentage of chord .

In our studied the airfoil for NACA5309 is used. **Fig.4.** show the complete fan.

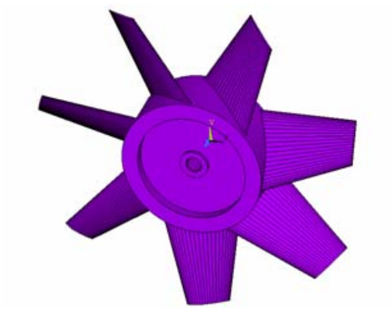

Fig.4. Complete fan

In order to get actual angle of attack (Hani, 2010) , the transformation coordinates system is needed as in **Fig.5.** From this figure the following equations can be get :

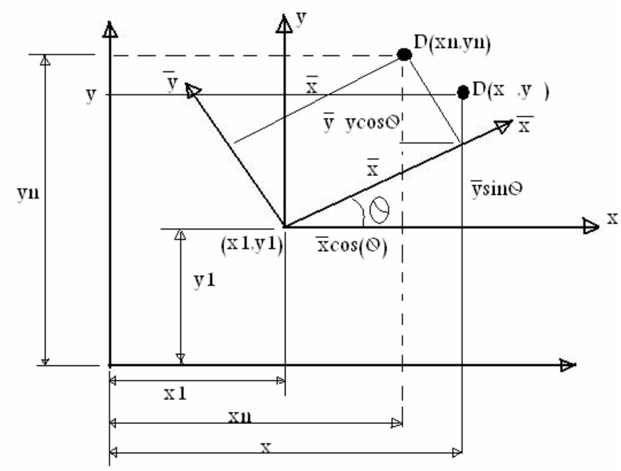

Fig.5. transformation coordinates

From **Fig.5.** it can be deduced that

$$\bar{x} = x - x_1 \quad (20)$$

$$\bar{y} = y - y_1 \quad (21)$$

$$xn = x_1 + \bar{x}\cos\theta - \bar{y}\sin\theta \quad (22)$$

$$yn = y_1 + \bar{x}\sin\theta - \bar{y}\cos\theta \qquad (23)$$

Thus the fan model in case of blade stagger angles 20° , 35° and 60° will be as in **Fig.6**

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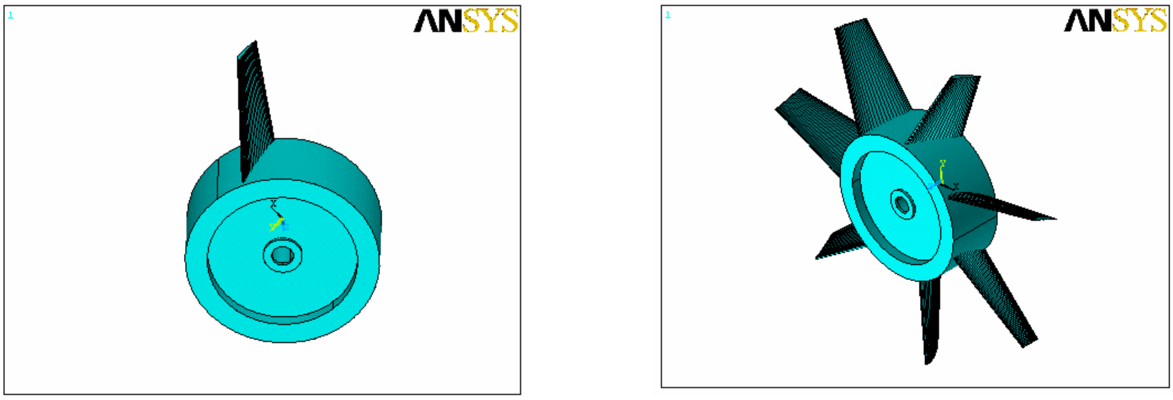
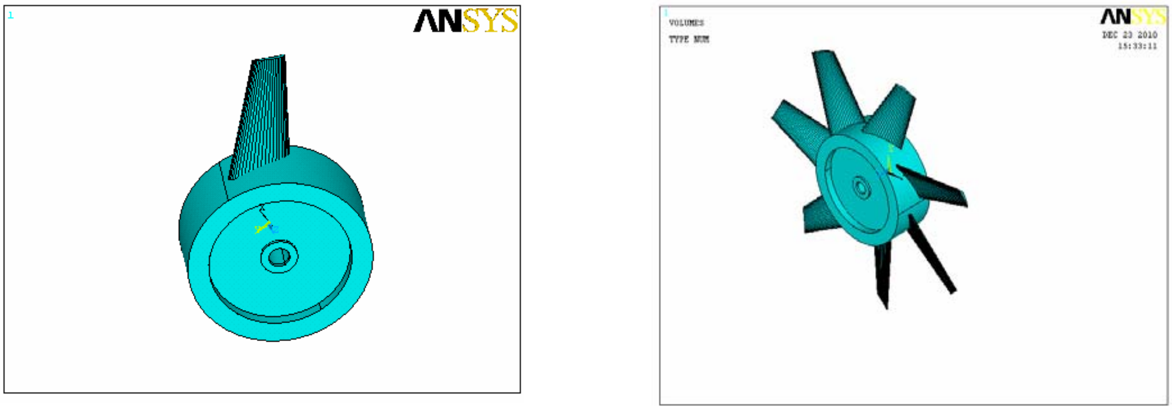
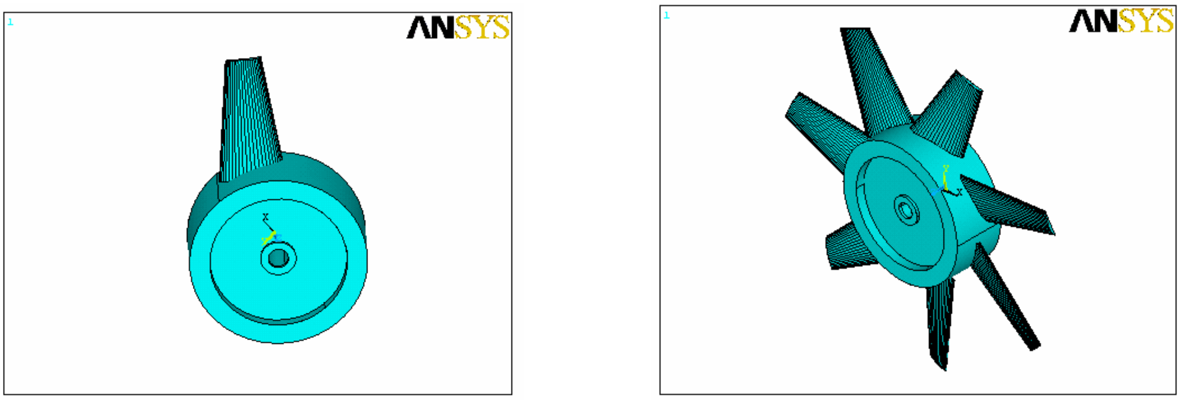


Stagger Angle = 20°



Stagger Angle=35 °



Stagger Angle = 60°

Fig.6. Fan model at different blade stagger angle

Fan is also meshing by using tetrahedral element (SOLID72) and it can be controlled by using element size mesh. **Fig.7.** shows the mesh of the fan.

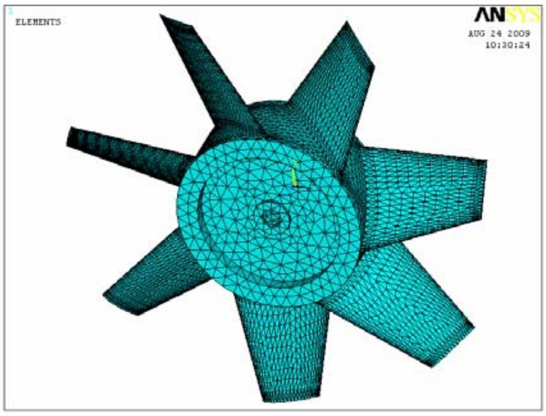


Fig.7. fan mesh

Hence the following is the program of modeling the vibration characteristics of an axial fan in APDL (ANSYS Parametric Design Language).

APDL PROGRAM FOR MODELING VIBRATION CHARACTERISTICS OF FAN

```
/view,1,1,1,1
/PREP7
ET,1,SOLID72
MP,EX,1,76E9
MP,NUXY,1,.3
MP,DENS,1,2650
CYL4,0,0,75.373E-3,360,8.5E-3,360,58E-3
CYL4,0,0,58.273E-3,360,8.5E-3,360,10E-3
VSBV,1,2
K,100,0,0,10E-3
CSKP,11,0,100,30,32
WPCSYS,1,11
CYL4,0,0,15E-3,360,8.5E-3,360,-3E-3
K,101,0,0,48E-3
CSKP,12,0,101,14,16
WPCSYS,1,12
CYL4,0,0,58.273E-3,360,8.5E-3,360,-10E-3
VSBV,3,2
K,102,0,0,-10E-3
CSKP,13,0,102,45,46
WPCSYS,1,13
CYL4,0,0,15E-3,360,8.5E-3,360,3E-3
K,500,0,0,-19E-3
```

```
K,501,0,6E-3,-19E-3
K,502,6E-2,0,-19E-3
CSKP,15,0,500,502,501
WPCSYS,1,15
CYL4,0,0,75.373E-3,360,8.5E-3,360
CSKP,14,0,65,68,2
WPCSYS,1,14
A=0.05989197448 :B=0.06323777403
T=35
X1=(1.000134*A)    :   Y1=0.000936*B   :   X2=(0.996248*A)
Y2=0.001907*B      :   X3=(0.984649*A) : Y3=0.004776*B
X4=(0.965502*A)    :   Y4=0.009410*B   :X5=(0.939085*A)
Y5=0.015599*B     :      X6=(0.905786*A)   : Y6=0.023068*B
…………………..   :   ……………………. : ………………
Y49=-0.000338*B  :   X50=(0.995866*A)    : Y50=-0.000784*B
X51=(0.500067)*A:    Y51=(.04744)*B
K,5100,X51,Y51
XN1=X51+(X1-X51)*COS(T)-(Y1-Y51)*SIN(T)
YN1=Y51+(X1-X51)*SIN(T)+(Y1-Y51)*COS(T)
K,10000,XN1,YN1
XN2=X51+(X2-X51)*COS(T)-(Y2-Y51)*SIN(T)
YN2=Y51+(X2-X51)*SIN(T)+(Y2-Y51)*COS(T)
K,20000,XN2,YN2
…………………………………………………
XN50=X51+(X50-X51)*COS(T)-(Y50-Y51)*SIN(T)
YN50=Y51+(X50-X51)*SIN(T)+(Y50-Y51)*COS(T)
K,5000,XN50,YN50
SPLINE,2800,2700,2600,250000,240000,230000
SPLINE,230000,220000,210000,200000,190000,180000
SPLINE,180000,170000,160000,150000,140000,130000
SPLINE,130000,120000,110000,100000,90000,80000
SPLINE,80000,70000,60000,50000,40000,30000
SPLINE,30000,20000
SPLINE,5000,4900,4800,4700,4600,4500
SPLINE,4500,4400,4300,4200,4100,4000
SPLINE,4000,3900,3800,3700,3600,3500
SPLINE,3500,3400,3300,3200,3100,3000
SPLINE,3000,2900,2800
ASKIN,114,115  :     ASKIN,113,116 :      ASKIN,112,117
ASKIN,111,118    :    ASKIN,110,119  :    ASKIN,109,120
ASKIN,108,121    :   ASKIN,107,122   :    ASKIN,106,123
……………………………………………………………
ASKIN,94,135    :   ASKIN,93,136     :   ASKIN,92,89
A,250000,2700,2600
AADD,26,27   :    AADD,28,29  :    AADD,30,31
…………………………………
AADD,31,26  :   AADD,28,27
AGEN,,26,,,,,7E-3,,,1
VEXT,26,,,,,114.96E-3,.5309325947,.5309325947,.5309325947
```

```
CSYS,1 :    VGEN,7,3,,,,51.42857143,,
VADD,ALL
SMRT,8   :    CM,_Y,VOLU   :   VSEL, , , ,     11
CM,_Y1,VOLU   :   CHKMSH,'VOLU'   : CMSEL,S,_Y
VMESH,_Y1
FINISH
/SOLU
ASEL,S,,,20  :   ASEL,A,,,726  :   ASEL,A,,,11
ASEL,A,,,21   :   ASEL,A,,,12  :  ASEL,A,,,727
NSLA,S,1
D,ALL,ALL
ALLSEL,ALL
RATIO=4*ATAN(1)/30
ANTYPE,MODAL
PSTRES,ON
NBF=20
MODOPT,LANB,NBF
/OUT,FANFAN
SOLVE
FINISH
```

RESULTS AND DISCUSSIONS

Free vibration analysis represents a study of the vibration characteristics of the rotor system such as natural frequency and mode shapes. The natural frequency and mode shapes of an axial fan is very important parameter in the design of a turbomachine system for dynamic loading conditions and minimization of machine failures. A detailed study is made using the formulation presented in this paper on the natural frequencies and mode shape levels of an axial fan. The free vibration characteristics have been investigated by ANSYS12 software.

The results report ten structural eigenvalues and eigenvectors based on the behavior of fan. **Table 1** explains the natural frequencies of the fan with blade stagger angle 20° ,35° and 60° respectively. **Figs. 8**, **9 and 10**. show mode shapes at 20°, 35° and 60° respectively.

Table 1 Natural frequency of axial fan.

Mode No.	Natural freq. (Hz) when Stagger Ang. 20°	Natural freq. (Hz) when Stagger Ang. 35°	Natural freq. (Hz) when Stagger Ang. 60°
1	567.624	632.306	562.78
2	571.706	634.893	565.616
3	572.663	635.587	566.729
4	574.184	637.536	577.412
5	576.362	638.73	579.183
6	577.502	639.137	586.3
7	578.323	640.25	588.884
8	1885	1690	1014
9	2277	2130	2221
10	2283	2143	2249

It can be noticed from **Table 1**, that the natural frequencies for the fan with stagger angle 35° are more than the fan with angle 20° and 60° for the 1 to 7 mode shapes and less than 20° and more than 60° in the other mode shapes. Also it can be seen that the natural frequencies are increased with large range after the mode 7. Because the deformation of the fan at modes 8,9,10 is decreased and that caused large range in natural frequency.

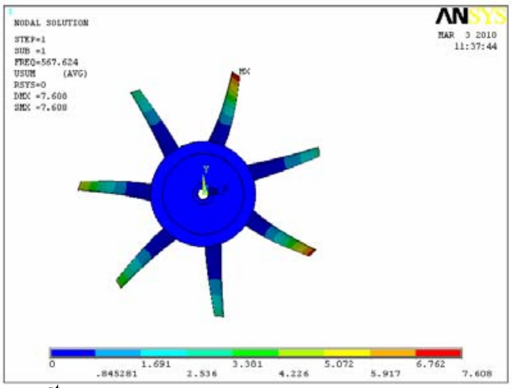
Table 2 Average Y- Amplitude of Vibration Versus Stagger angle for Experimental (Wafa,2007) and the present Finite lent results

Stagger angle	Y-amplitude (m) Experiential (Wafa,2007)	Y-amplitude (m) ANSYS	Discrepancy%
20°	$5.8*10^{-5}$	$5.9*10^{-5}$	1.69
35°	$4.9*10^{-5}$	$5.01*10^{-5}$	2.19
60°	$4.1*10^{-5}$	$4.3*10^{-5}$	4.65

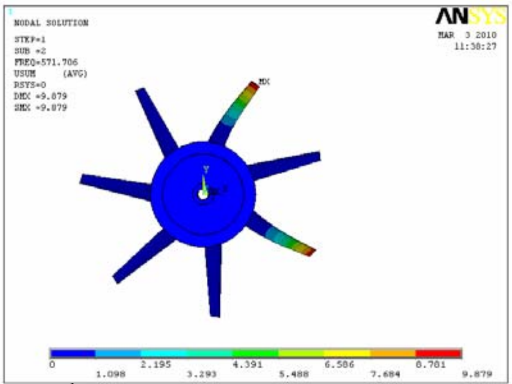
Table 3 Average X- Amplitude of Vibration Versus Stagger angle for Experimental (Wafa,2007) and the present Finite Element results

Stagger angle	X-amplitude (m) Experiential (Wafa,2007)	X-amplitude (m) ANSYS	Discrepancy%
20°	$5.1*10^{-5}$	$5.4*10^{-5}$	5.55
35°	$4.5*10^{-5}$	$4.9*10^{-5}$	8.16
60°	$3.9*10^{-5}$	$4.2*10^{-5}$	7.14

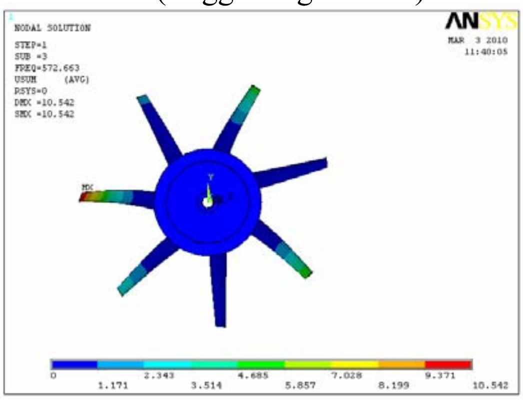
Tables 2 and 3 show that in general the average amplitude in x-direction is greater than that in y-direction and the amplitude decreases as the stagger angle increases . Maximum amplitude occurred at stagger angle of 20° both in x and y – directions and have the less error% i.e. 1.69 in y–direction and 5.55 in x– direction.

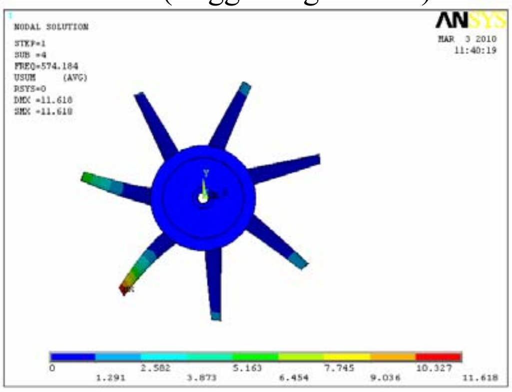


1st mode (stagger angle = 20°)

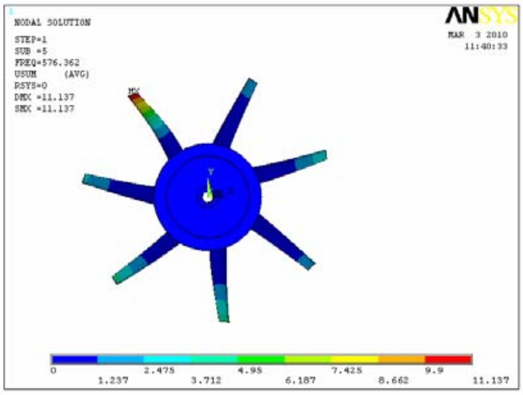


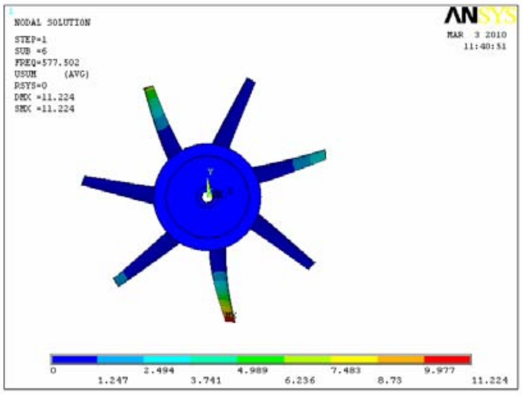
2nd mode (stagger angle = 20°)



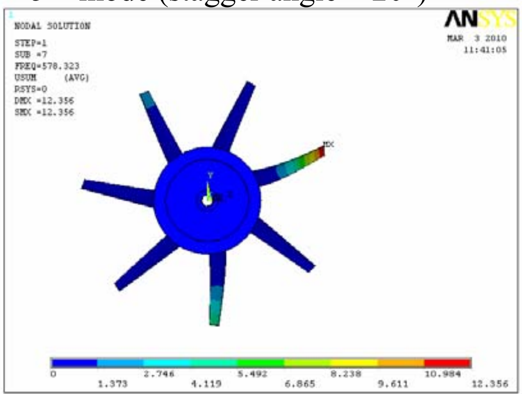


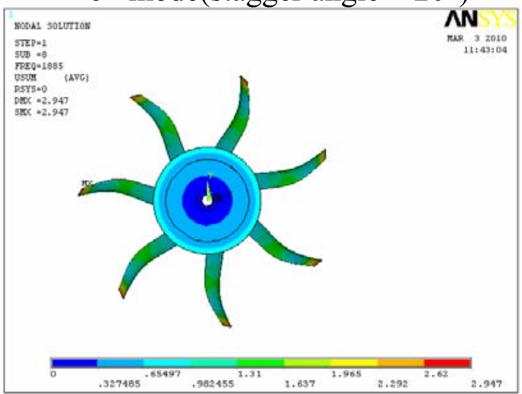
3rd mode(stagger angle = 20°) 4th mode(stagger angle = 20°)



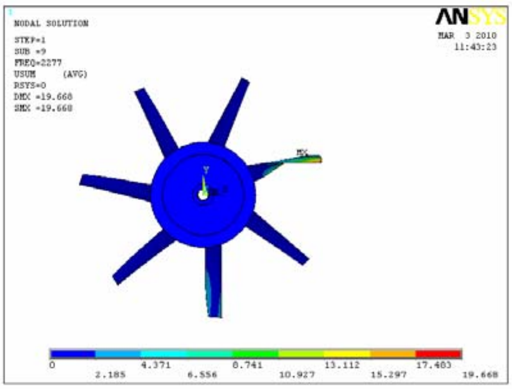


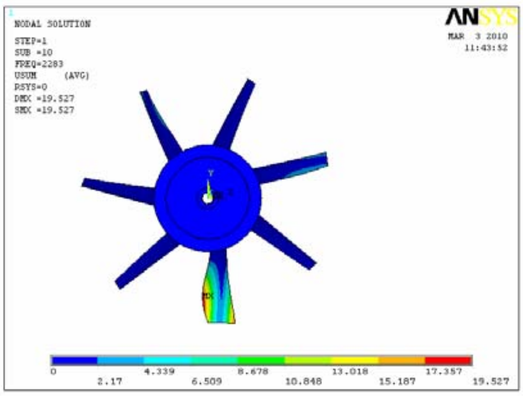
5th mode (stagger angle = 20°) 6th mode(stagger angle = 20°)





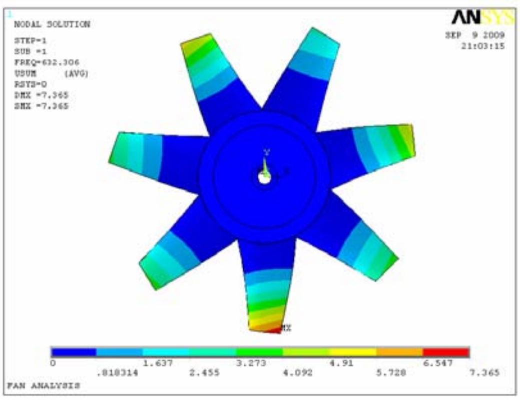
7th mode (stagger angle = 20°) 8th mode(stagger angle = 20°

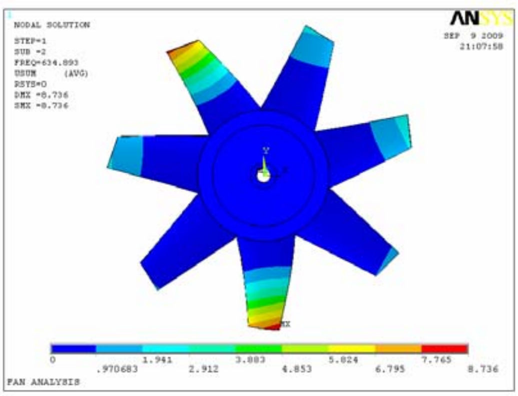




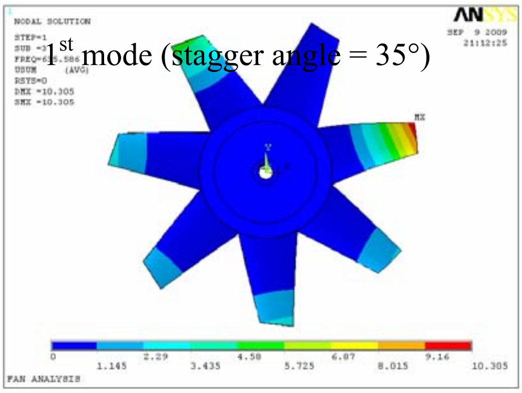
9th mode (stagger angle = 20°) 10th mode(stagger angle = 20°)

Fig.8. Mode shapes of fan with stagger angle = 20°

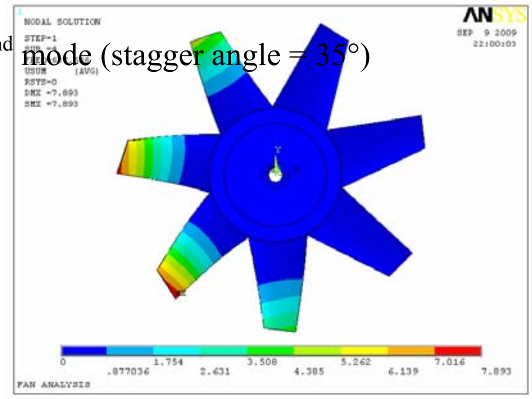




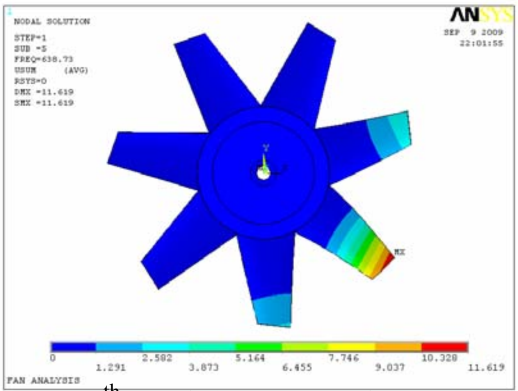
1st mode (stagger angle = 35°)



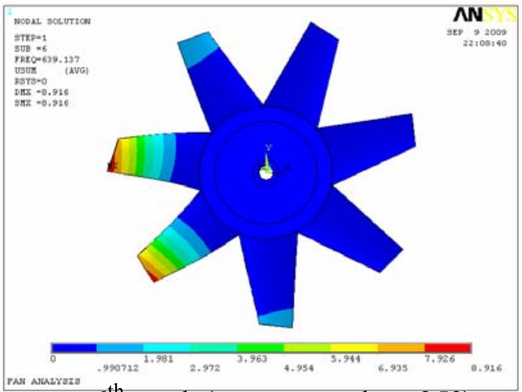
2nd mode (stagger angle = 35°)



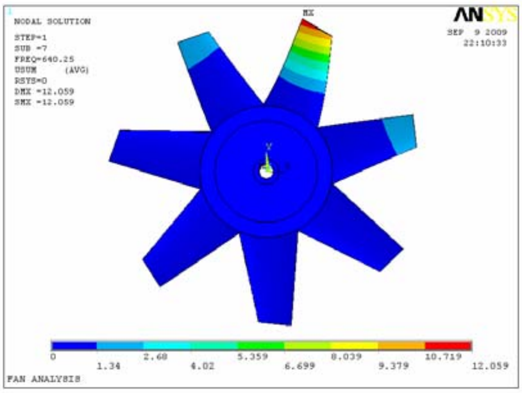
3rd mode (stagger angle = 35°)



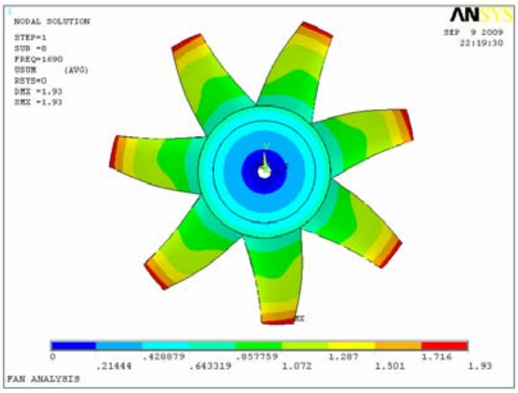
4th mode(stagger angle = 35°)



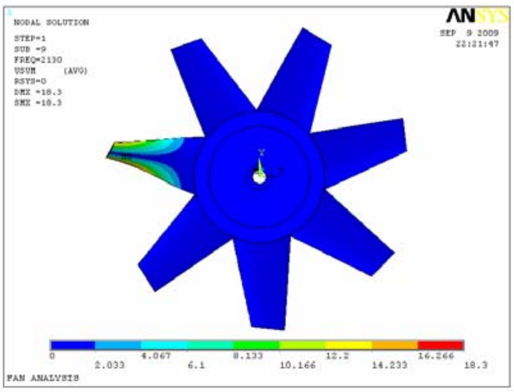
5th mode(stagger angle = 35°)



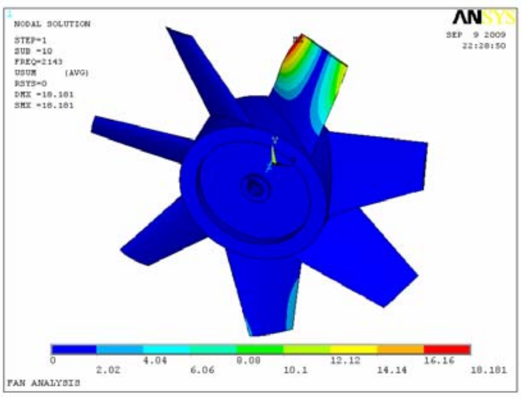
6th mode(stagger angle = 35°)



7th mode (stagger angle = 35°)



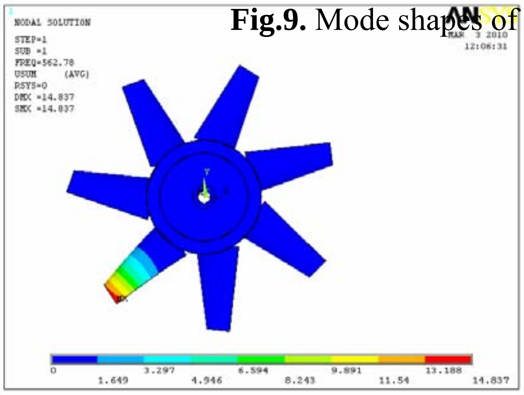
8th mode(stagger angle = 35°)

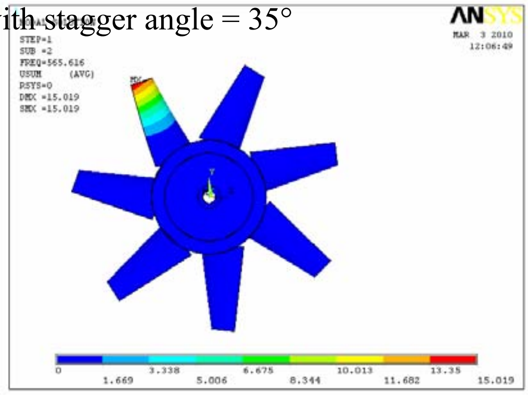


9th mode(stagger angle = 35°)

10th mode(stagger angle = 35°)

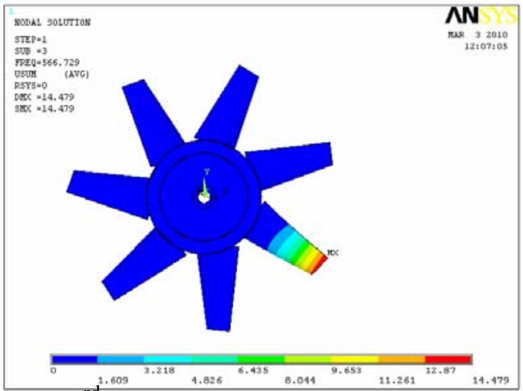
Fig.9. Mode shapes of fan with stagger angle = 35°

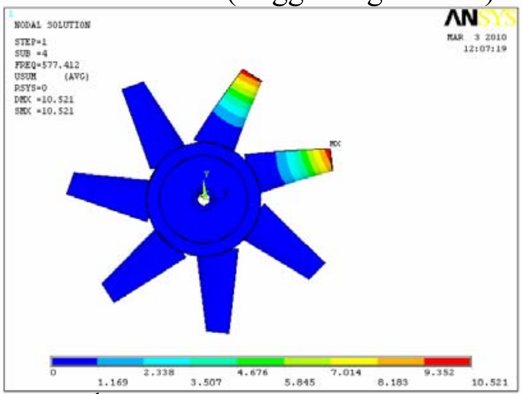




1st mode(stagger angle = 60°)

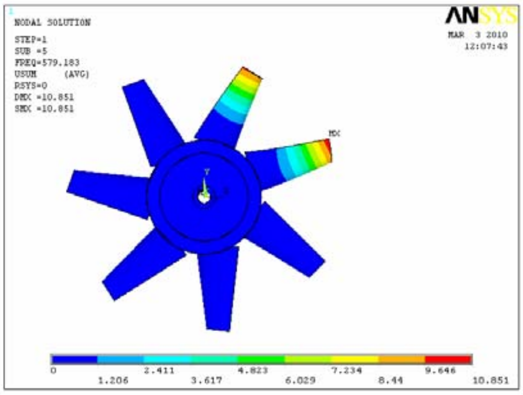
2nd mode(stagger angle = 60°)

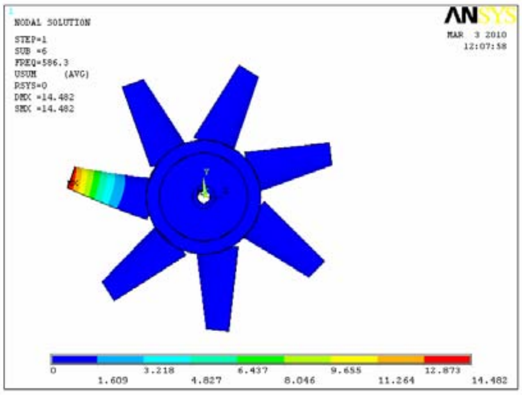




3rd mode(stagger angle = 60°)

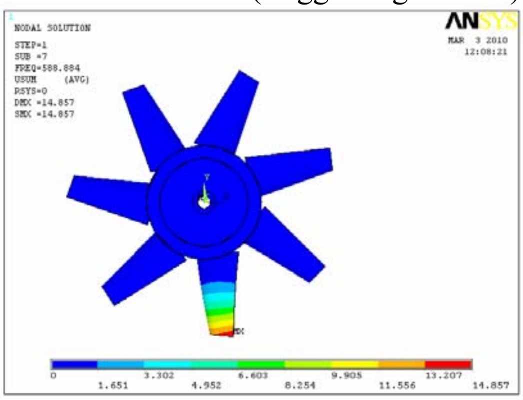
4th mode(stagger angle = 60°)

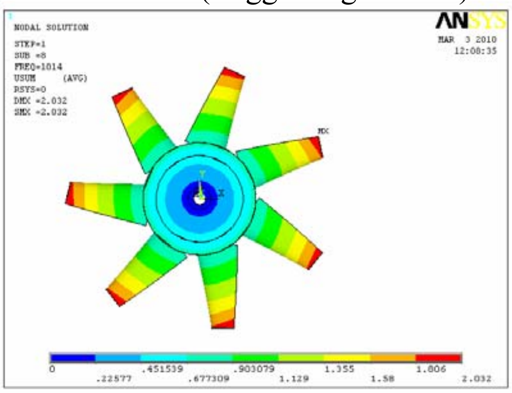




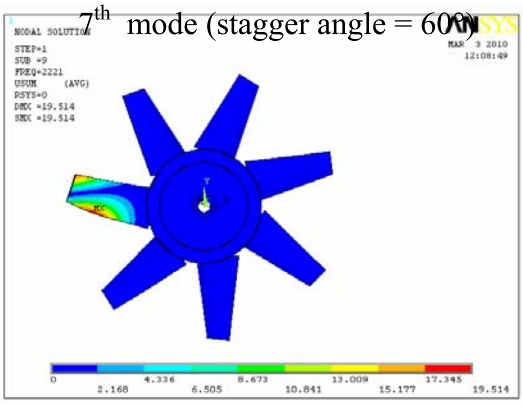
5th mode(stagger angle = 60°)

6th mode(stagger angle = 60°)

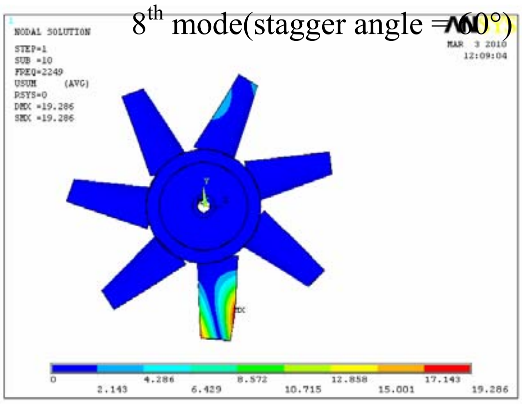




7th mode (stagger angle = 60°)



8th mode(stagger angle = 60°)



9th mode (stagger angle = 60°) 10th mode(stagger angle = 60°)

Fig.10. Mode shapes of fan with stagger angle = 60°

CONCLUSIONS

It can be concluded that the natural frequency of the fan increased with increasing mode number and the amplitude decreases as the stagger angle increases. The maximum amplitude occurred at stagger angle of 20° and have less error.

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SYMBOLS

[K[	stiffness matrix
[M]	mass matrix
ω	natural frequency
$\{\delta\}$	displacement vector
λ	eigenvalues