Intelligent Over-vibration Protection Modeling & Simulation For Three Phase Induction Motor

Abstract—This paper presents an intelligent over-vibration simulation and protection preference for three phase medium and high power induction motor. Three phase induction motor is widely used in industry because they are rugged, reliable and economical.Monitoring and over-vibration protection of machines can avoid consequential damage and avoid its consequences.The effect of three main different abnormal operating cases which are rotor unbalance,shaft coupling misalignment and torque pulsations on the rotor vibration are examined.The results are collected and evaluated for motor status. Rotor unbalance and shaft coupling misalignment will cause radial vibrations,while torque pulsations will cause torsional vibrations.The vibration analysis and simulation is calculated using matlab(m-file)to produce the different vibration waves.Fuzzy logic technique is utilized in determining machine vibration source ,and if vibration displacement (r.m.s) at bearing exceeds limits,then fuzzy will feed a relay to trip machine.The fuzzy is programmed by the output of the simulation(m-file). A case study of 5100 kW motor was used in rotor vibration modeling,rotor stiffness and damping values were calculated. Fuzzy logic as an artificial intelligence technique can be used as a vibration monitoring and protection tool.

Keywords— *Induction motor ,Vibration modeling,Fuzzy*

I. INTRODUCTION

Vibration monitoring is part of machine condition monitoring,which is used to identify a significant change which is indicative of a developing a fault,further this data can be used in predictive maintenance,this allows maintenance to be scheduled to avoid consequential damage and avoid its consequences.This technique is normally used for rotating equipments. A complete study was done for a motor, starting from degree of freedom for motor ,showing three common vibration causes using simplified mathematical representation,collecting motor data from vendor catalogue,calculating rotor stiffness and damping values,comparing results with international standards for setting vibration limits as IEC-60034 and ISO 1940-1,last step using fuzzy logic for reading input and stating machine status(healthy or not)and to feed a relay to trip motor above certain value. Researcher’s recent works in this field are : unbalance compensator to control rotor unbalance mass [1,2 and 6],current and vibration signals to identify machine status [3,5 and 7], control strategy to minimize torque ripple by considering time harmonics,and space harmonics[4],excitation forces modeling[8,9 and 10], fuzzy vibration controller based upon displacement and vibration of single degree of freedom[12].The objective of this paper is to calculate vibration magnitudes and directions for rotor unbalance,shaft coupling misalignment and torque pulsations,and if vibration waves produced by each have unique characteristic,then a fuzzy controller-selected as an artificial intelligent technique-can map inputs and take actions. In this work,section 2 deals with degree of freedom for different abnormal operating cases.section 3 explains fuzzy logic as part of vibration monitoring and protection. Section 4 shows motor data used in calculations. Section 5 shows results. Section 6 is the conclusion.

II. DEGREE OF FREEDOM MODELING

Degree of freedom is the number of independent motions that are allowed to the body. It is shown-later in this section-that rotor unbalance and torque pulsations are 1^{st} degree of freedom systems,while shaft coupling misalignment is 2^{nd} degree of freedom system.

2.1 Shaft unbalance

This type of forced vibration as defined by [8] ”This occurs when a rotating structure does not possess perfect symmetry in its mass distribution”.

The unbalance excitation force acting at RPM frequency[8]

$$F=M.\omega^2.e \qquad (1)$$

Where
M = rotor weight(kg).
ω = rotor angular speed (rad./sec).
e=eccentricity of Rotor center of gravity (meter).

Shaft unbalance can be represented with unbalanced mass, m,with an eccentricity(deviation of unbalanced mass from center of rotation),rotates with an angular speed , ω,as shown in figure 1.

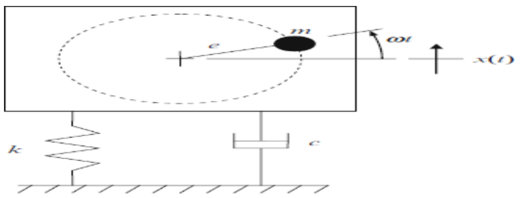


Figure 1.Single degree of freedom model with rotating unbalance[8]

Rotor vibrations are only in one direction,$x(t)$,radial direction-as we have only one force(unbalance excitation force) acting on one mass(rotor)-thus this type of system is (1st degree of freedom),Hence vibration calculations for this case relied on only first degree of freedom case.

Vibration displacement for 1st degree of freedom systems are dependent on rotor angular speed,rotor weight, rotor stiffness and damping values as shown in equation(2)

$$(X / F) = 1 / (-\omega^2 .M + i .\omega.c + k) \qquad (2)$$

Where
$i = \sqrt{(-1)}$
c = rotor damping(N-s/m)
k = rotor stiffness (N/m)
X = vibration displacement (m)
F =vibration force (N)

2.2 Shaft coupling misalignment

This type of forced vibration happens when two or more equipments are coupled,but their shafts coupling are misaligned.Coupling is used to allow a certain degree of misalignment. As shown in figure(2) two machine shaft centerlines,Z1 and Z2,which are misaligned both vertically($\Delta Y1$ and $\Delta Y2$) and horizontally ($\Delta X1$ and $\Delta X2$).Three Moments(MX, MY and MZ)and three force(FX, FY and FZ)which the coupling exerts on the machine's shaft are shown.Z1 is the axis of the driving machine,that(+)rotation is applied as shown and that rotation is in same direction as applied force.

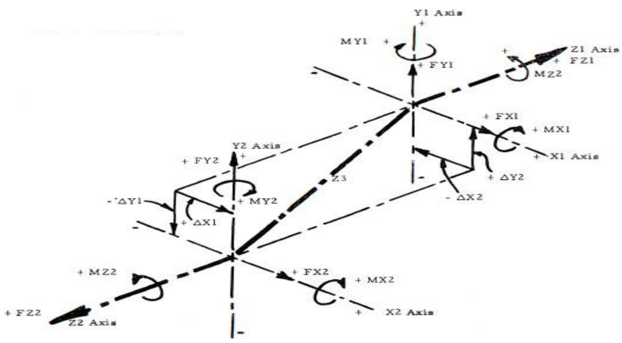


Figure 2.Coupling coordinate system[10]

Horizontal shaft misalignment vibration forces and moments[9]

$$FX1 = (-MY1-MY2)/Z3 \qquad (3)$$
$$MX1 = Tq.\sin\Theta1 + Kb.\ \Phi1 \qquad (4)$$
$$MX2 = Tq\ .\sin\Theta2 - Kb.\ \Phi2 \qquad (5)$$

Where
Tq = Torque(N.M).
$\Theta1$, $\Phi1$, $\Theta2$ and $\Phi2$ = misalignment angles.
Z3= distance between couplings centers of articulation.
Kb=Flexure coupling bending spring rate per diaphragm or per disk pack,(Lb-In/Deg).

Vertical shaft misalignment vibration forces and moments[9]

$$FY1=(MX1+MX2)/Z3 \qquad (6)$$
$$MY1 = Tq.\sin\Phi 1 - Kb.\ \Theta1 \qquad (7)$$
$$MY2= Tq.\ \sin\Phi2 + Kb.\ \Theta2 \qquad (8)$$
$$\Theta1=\text{Arc}\sin(\Delta X1 / Z3) \qquad (9)$$
$$\Phi1= \text{Arc}\sin(\Delta Y1 / Z3) \qquad (10)$$
$$\Theta2= \text{Arc}\sin(\Delta X2 / Z3) \qquad (11)$$
$$\Phi2= \text{Arc}\sin(\Delta Y2 / Z3) \qquad (12)$$

Where
$\Delta X1$ = Driver maximum horizontal misalignment ,$\Delta Y1$= Driver maximum vertical misalignment ,$\Delta X2$ = Driven maximum horizontal misalignment , $\Delta Y2$= Driven maximum vertical misalignment.

Figure 3 shows shaft 2nd bending mode due to bending moments which the coupling exerts on the machine's shaft. 2nd bending mode is the oscillation of the extreme end-of-span mass bulk up and down in out-of-phase having one intermediate nodel point near mid-span and the two nodal points near the two main bearing.

Bending moment[10] = square root[(FX1. (hub length + disk pack centerline to shaft end) - $MY1)^2$ + (FY1. (hub length + disk pack centerline to shaft end) +$MX1)^2$] (13)

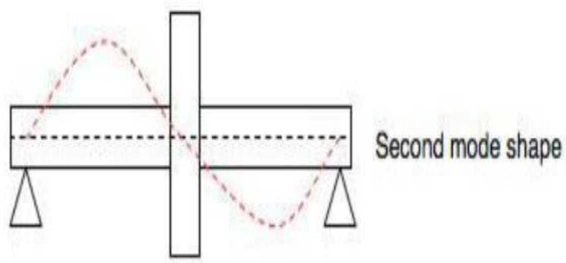


Figure 3.Second bending mode

2nd bending modes can be approximated as two masses moving in opposite directions,Hence our rotor can be modeled as two masses vibrating in two direction-,x1(t) and x2(t)-out of phase 180 º- , in radial direction,thus this type of system is (2nd degree of freedom).

Tranfer function for 2nd degree of freedom systems,used to obtain rotor vibration displacement in meter – obtaing X1 is sufficient as X1 = X2 ,but out of phase 180 º-is

$$(X / F) = [(-1 \,.\, \omega^2 \,.\, M2) + (i \,.\, \omega .\, c2) + k2)] / [(-1 \,.\, \omega^2 \,.\, M1+i \,.\, \omega.(c1+c2)+(k1+k2)).\ (-1 \,.\, \omega^2 \,.\, M2+i \,.\, \omega.c2+k2))-((-i.\, \omega.c2-k2).(-i.\, \omega.c2-k2)] \qquad (14)$$

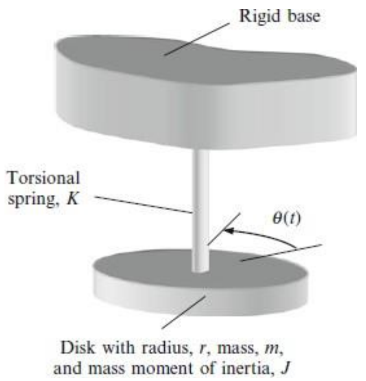
Where
M1 =mass 1 weight.(kg)
M2=mass 2 weight.(kg)
c1=mass 1 damping (N-s/m)
c2=mass 2 damping (N-s/m)
k1=mass 1 stiffness(N/m)
k2=mass 2 stiffness (N/m)

2.3 Torque pulsations

This type of forced vibration refers to periodic increase or decrease in output torque as the motor shaft rotates,this happens due to slight asymmetries in magnetic field generated by the motor winding.

The study is applied to uncoupled motor (i.e motor at run test,with no load attached).

As shown in figure(4),for single degree of freedom torsional system,the independent coordinate is the rotational angle , θ,rather than position.



Figure(4)Lumped parameter model for single degree of freedom torsional system.[8]

Vibration angular displacement for 1st degree of freedom systems are dependent on rotor angular speed, rotor mass moment of inertia, rotor torsional stiffness and damping values as shown in equation(15)

$$\theta/TP = 1 / (-\omega^2 .J + i .\omega.c + k) \qquad (15)$$

where

J = rotor mass moment of inertia in ($kg.m^2/rad$)
θ =vibration displacement (radian)
TP=torque pulsation(N.m)
K=torsional stiffness(N.m/rad)

III. FUZZY LOGIC FOR VIBRATION MONITORING AND MACHINE PROTECTION

Figure 5 shows that Fuzzy inference systems basic structure depends on three conceptual components, first component is the database which defines the membership function used in the fuzzy rules,second component are fuzzy rules which have already determined by rule base.,the third component is reasoning that used to perform the inference system procedures according to the fuzzy rules in order to produce the fuzzy output [11],

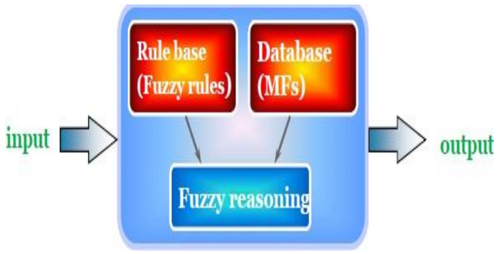


Figure 5. Fuzzy inference system structures

Figure 6 shows fuzzy membership theory,vibration input data can be classified by membership functions,to give machine status.

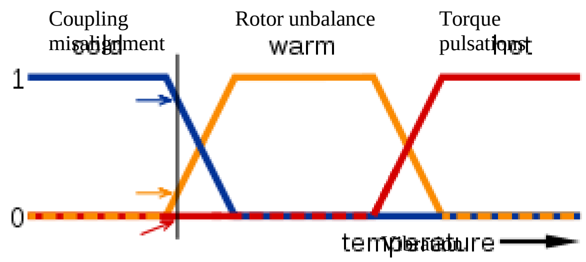


Figure 6 fuzzy working principle.

Input data to feed fuzzy controller- as proposed in this paper- are shown in figure 7,three inputs are used to set rules for vibration monitoring and protection,first input are vibration magnitudes which are calculated-shown in section V-,2nd and 3rd inputs are already calculated-from section II-will decide machine status.

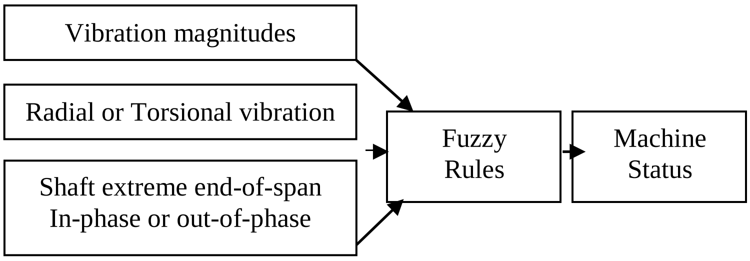


Figure 7. Fuzzy proposed inputs and expected outputs .

Flow chart- as proposed in this paper- for fuzzy controller work steps.

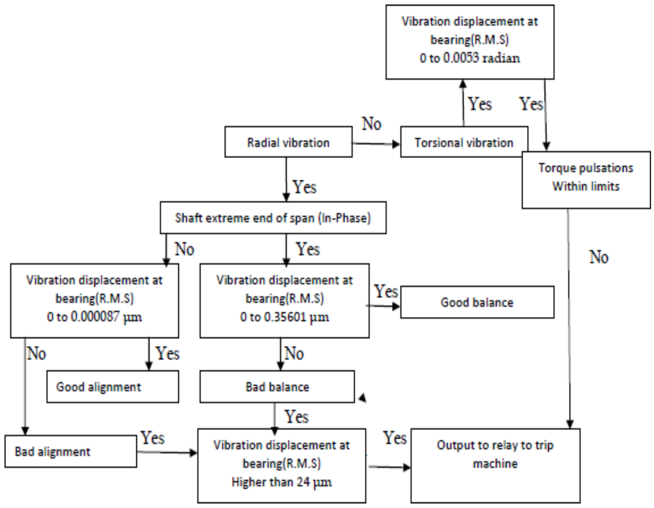


figure 7.flow chart of fuzzy rules.

A sample of fuzzy output is shown in figure 8,in this case vibration magnitude is 0.0591µm ,vibration direction(radial or torsional)were divided by membership functions-0 to 0.5 for radial and 0.5 to 1 for torsional- value of 0.239 means radial vibrations.
Vibration phase(Shaft extreme end-of-span
In-phase or out-of-phase
)were divided by membership functions-0 to 0.5 for in-phase and 0.5 to 1 for out-of-phase-in figure 8,value of 0.408 means in-phase.Machine status were divided by membership functions 0 to 6-corresponds to number of rows in graph 8- to cover following cases-good and bad balance, good and bad alignment,torque pulsations and machine trip, Output value is 0.501 which corresponds to good balance.

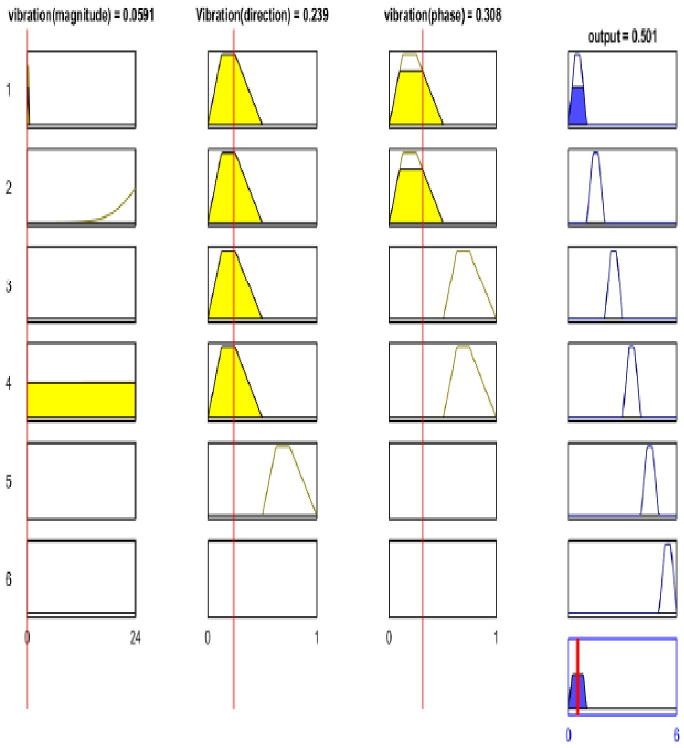


figure 8.fuzzy system output sample.

IV. MOTOR DATA

The machine under examination is a three phase induction motor,4-pole,horizontal mounting.

Few important features of motor rotor :which affect rotor resistance to vibration forces,were not mentioned in vendor catalogue,which are rotor stiffness and damping values.
The next step was to calculate these values.

A. Through calculations

Transverse stiffness for flexible shaft,which is used in radial vibration calculations-i.e shaft unbalance and shaft coupling misalignment cases.

$$k(eff) = (48.E.I) / L^{3.} \qquad (16)$$

Where

E = young's modulus (N/m^2)

I=area polar moment (m^4)

L=span of the shaft(meter)

$$\text{Damping ratio} = c / [2 . \sqrt{(k.M)}] \ (17)$$

Torsional stiffness for flexible shaft,which is used in torsional vibration calculations-i.e torque pulsations case.

$$k \ (N.M/rad) = (G * I)/ L \qquad (18)$$

Where

G = shear modulus(N/m^2)

As shown in table 1,Model number siemens (1SB4 636-4JX90-Z)-is chosen as it is a high power motor-following parameters are used in obtaining experimental results,as shown in section V.

Parameter	unit	value
Output power	kW	5100
Speed	Radian/second	156.2
Rotor weight	kg	3510
Shaft torque of rated load	N.m	32644
Young's modulus	N/m^2	$200 * 10^9$
Area polar moment	m^4	0.005
Shaft mass moment of inertia	$kg.m^2$	186
Shaft density	kg/m^3	7870
Span of shaft	m	2.135
Damping ratio for steel rotor shaft assembly	Dimensionless	0.0175
Flexure coupling bending spring rate per diaphragm or per disk pack	N.m/rad.	2578.05
Distance between couplings centers of articulation	m	0.698
Hub length	m	0.12
Disk pack centerline to shaft end	m	0.00127
Shaft shear modulus	N/m^2	$80 * 10^9$
Shaft polar second moment of area	m^4	0.0000784

Table 1.machine data

V. RESULTS

Vibration displacements (r.m.s) at machine bearing for the 3 vibration sources were calculated and plotted against motor speed(+/- 10% variation)refer to figures(9 and 10).studing good to worst cases scenarios were considered as follow :

5.1 Shaft unbalance
Eccentricities examined are from zero-perfect balanced motor- to 0.00273 meter -which is the value which will make vibration displacement at bearing(r.m.s) = 24μm,which is the limit –for uncoupled motors - of maximum vibration magnitude for shaft height greater than 280 mm as per IEC60034.

For this type of rotors under study-parts of process plant machines-maximum allowable eccentricity is 0.0000403m-as per ISO 1940-1 - vibration displacement(R.M.S)at rotor center = 0.71202 μm,while at bearing(1st bending mode)=0.35601 μm.

Hence based on above results,an alarm to be generated if vibration displacement(R.M.S)at bearing exceeds 0.35601 μm . Machine to be tripped if vibration displacement(R.M.S)at bearing reachs 24μm.

5.2 Shaft coupling misalignment
Driver maximum horizontal and vertical misalignment ratio to distance between couplings centers of articulation examined are from zero-perfect aligned couplings- to 0.001,as vendor catalogue specifies maximum horizontal and vertical misalignment=0.00008 m.

Hence based on above, at maximum horizontal and vertical misalignment = 0.00008 m (as per vendor data), vibration displacement(R.M.S) at bearing(2nd bending mode)= 0.000087 μm,Misalignment will not reach alarm limits,above maximum horizontal and vertical misalignments-which is normal ,as misalignment is a static force,which causes only fatigue to shaft-but is not a major source of vibration.

5.3 Torque pulsations

Torque pulsations examined are from zero to 30 % of full load torque,as the maximum allowed torque pulsations for reciprocating type compressors per API 618 Standard Requirements, is 25% of full load torque. At maximum allowed torque pulsations = 32644 x 25% =+/- 8161 N.m . Vibration displacement(R.M.S) at bearing= 0.0053 radian.Theoritcally we can change radian to meter = radian x rotor radius ,so fuzzy controller to compare vibration values with same units(meter).

VI. CONCLUSION

Vibration monitoring is one of the key elements to monitor machines conditions,hence this paper explores vibration waves characteristics for different vibration sources,which the fuzzy system maps it into outputs. A relation between motor speed and different vibration waves are plotted,studing good to worst cases scenarios are developed for each case. Maximum allowable eccentricity for case study is calculated,alarm set point for rotor out of balance is calculated.Calculations shows that misalignment is not a major source of vibration.Vibration displacement for maximum torque

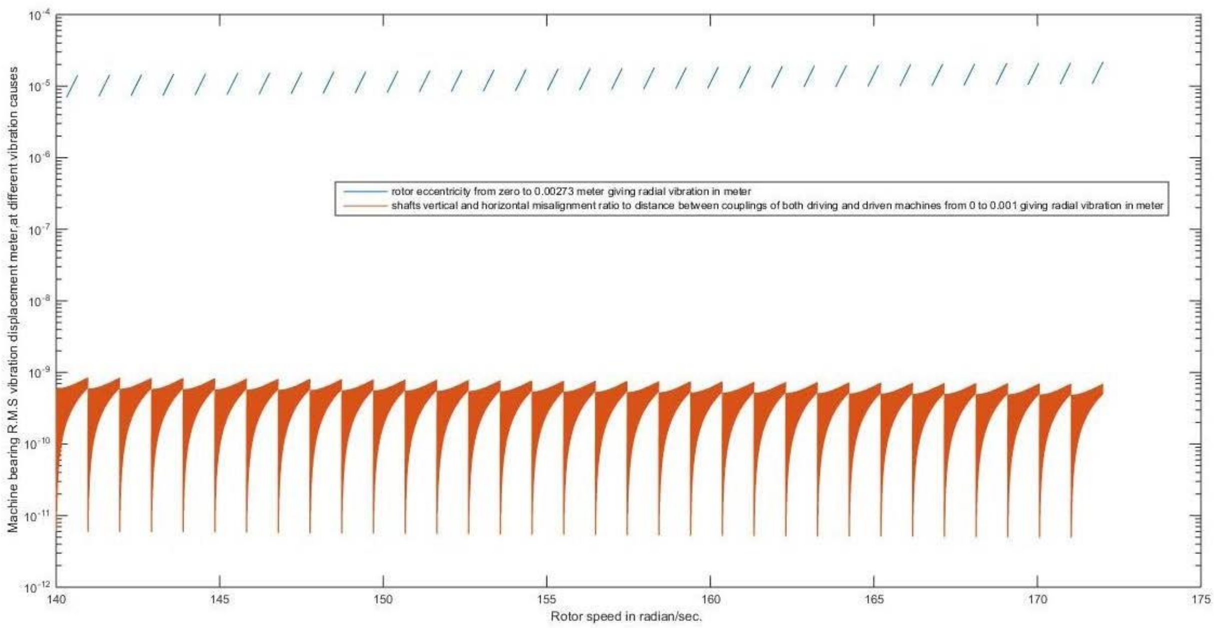


Figure 9 rotor speed in rad./sec.(X-axis) against machine bearing radial vibration displacement in meter (r.m.s) for shaft coupling misalignment(brown lines) and shaft unbalance(blue lines).

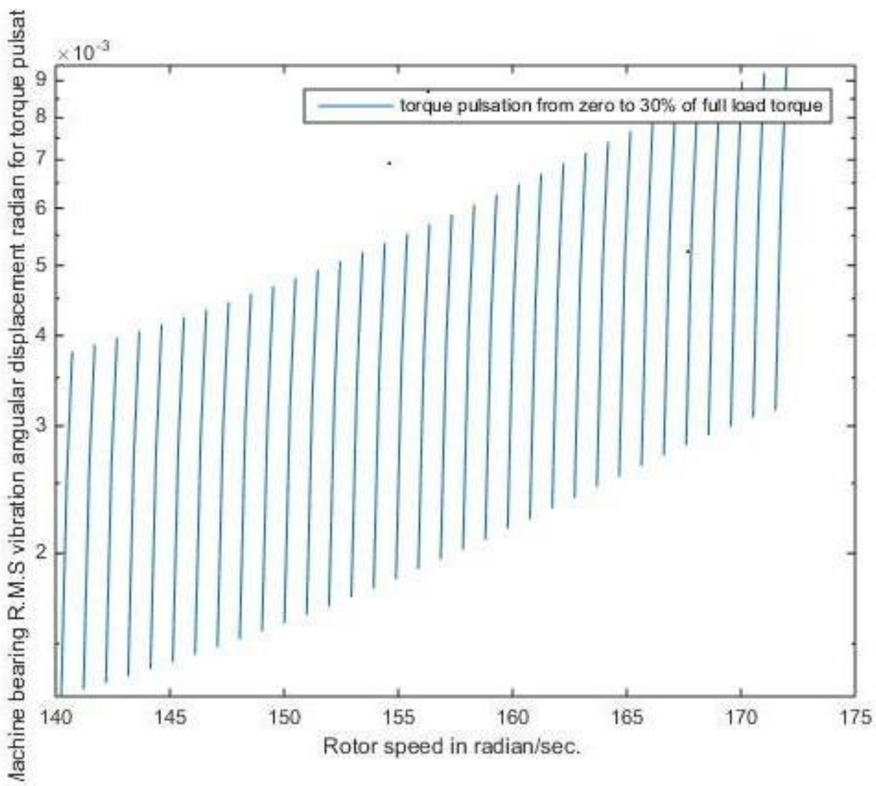


Figure 10 rotor speed in rad./sec.(X-axis) against machine bearing torsional vibration displacement in radian (r.m.s) for torque pulsations.

pulsation allowable is calculated.It is shown that fuzzy logic as an artificial intelligence technique can be used as a vibration monitoring and protection tool.

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