IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 13, Issue 1 Ver. III (Jan. - Feb. 2016), PP 12-19
www.iosrjournals.org

Design And Analysis Of High Speed Milling Spindle For Minimum Deflection

Suhas D. Upase[1], M. C. Swami[2]

[1] *(M.E. Machine Design Student of Mechanical Department, M. S. Bidve Engineering College, Latur,Maharashtra,India.)*

[2] *(Faculty of Mechanical Department, M. S. Bidve Engineering College, Latur, Maharashtra, India.)*

[3] *(Faculty of Mechanical Department, G. N. D. Engineering College, Bidar, Karnataka, India.)*

Abstract: *In this paper high speed milling spindle design of 16000rpm with minimum static deflection has been determined by varying bearing stiffness, span length between bearings and front bearing inner diameter. In present work intially we design different diameter spindles like ϕ55, ϕ60, ϕ65 and differnet bearing stiffness of respective diamenter by keeping 15^0contact angle . Further we evaluate the deflection of the spindle nose, static analysis and optimum bearing span length by theoretical analysis and compaire with ANSYS results. The result shows that the span length and bearing siffness has significant effect on spindle deflection.*

Keywords: *High speed spindle, Static deflection, Bearing stiffness,Ansys software .*

I. Introduction

The main mechanical component in machining centers is spindle. The spindle rotates at high speed with holding different cutter, to machines a material assembled to machine tool table. Finishing quality of workpiece and productivity directely affected by static and dynamic stffness of the spindle. The bearing arrangement is important factor for spindle design. In machine tools spindles are rotating drive shafts that act as axes for cutting tools or to hold cutting instruments.High speed spindles have important component of any type of high machining process whether in CNC, tool machining centers and oyher process components. Motorized spindle have been developed to achive high speed rotation, which equipped with bulit in motor as a part of spindle for eliminating the power transmission devices like gears, belts.

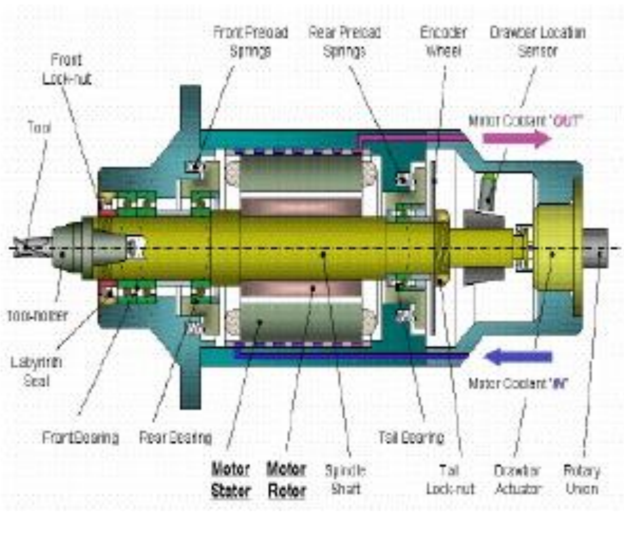

Fig. 1 High frequency milling spindle arrangements [7]

II. Static Stiffness Analysis.

2.1. Introduction

Static stiffness is one of the important performances of machine tools. Hence it must be correctly defined and related parameters of spindle must be properly determined. Since power loss will be there in terms of mechanical effciency, but as compared to conventional belt drive system where around 20-30%, here it has been reduced to 10-15%.

Hence the available power at the spindle is assumed to be around 8.5 to 9 KW. Stiffness of the spindle is defined as its ability to resist deflection under the action of cutting force.[11]

As shown in fig. 2 when a force exerted on the spindle nose is P, the displacement at the spindle nose in the same direction as that of P is '**δ**' and correlation stiffness is defined as **K = P/δ**

Using carbide cutting tool, cutting speed to cut thestainless steel material is 120 m/min.[9]

The cutting force P can be calculated from the unit powerconcept referring to the following equations:[7]

$P = 6120\, N_m / V$ kgf ---------(Eqn.-1)

where, N_m=power of the motor at the spindle=9 KW=9×10^3 W
V =cutting speed =120 m/min=2 m/sec

Therefore,
P=6120*9/**120 =459 kgf**
P= 4502.8 N

III. High Speed Millingspindle

3.1. Features of milling high speed spindle to be designed

The spindle rotates at 16000 rpm with power rating 9 KW. Spindle is mounted on ball bearing of15^0 angular contact. In analysis the stiffness of outer diameter of spindle is near about inner diameter of bearing**.**

3.2 Rigidity analysis by analytical (Determination of spindle deflection)

Unit deflection of the spindle due to its own bending under the cutting force and also deflection caused due to elasticity of bearing**.**

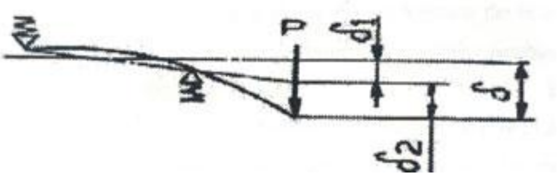

Fig. 2 Deflection of the spindle nose

The deflection of the spindle nose 'δ' for an unloaded spindle due to load P is given by [5]

$$\delta = \delta_1 + \delta_2$$

$$\delta = P\left[\frac{1}{S_A}\left(\frac{a+L}{L}\right)^2 + \frac{1}{S_B}\left(\frac{a}{L}\right)^2 + \frac{a^2}{3E}\left(\frac{L}{I_L} + \frac{a}{I_a}\right)\right]$$

------- (Eqn. 2)

Where, $\delta 1$= Deflection due to radial yielding of the bearings in mm
δ_2 = Deflection due to elastic bending of the spindle in mm
a = Length of overhang in mm
L = Bearing span in mm
E = Young's modulus of Spindle material in N/mm^2
P = Cutting force in N
S_A = Stiffness of the front bearing N/mm
S_B = Stiffness of the rear bearing in N/mm
I_L = Second moment of area of the shaft at the span in mm^4
I_a = Second moment of area of the shaft at the overhang in mm^4

3.3 Arrangements of Bearing

Two different bearing arrangements 'A' and 'B' are considered for the design. Angular contact ball bearings with 15° contact angle are used with medium preloading. The bearing arrangement 'A', with duplex bearing set at the front, in which a pair of angular contact ball bearings are arranged in back-to-back and the duplex bearing set at the rear mounted in back-to-back fashion. The bearing arrangement 'B', with triplet bearing set at the front, in which one pair of angular contact ball bearings are arranged in tandem with respect to each other and back-to-back with respect to a single angular contact ball bearing and the duplex bearing set at the rear mounted in back-to-back fashion. The triplet bearing and duplex bearing sets are shown in fig. 3.

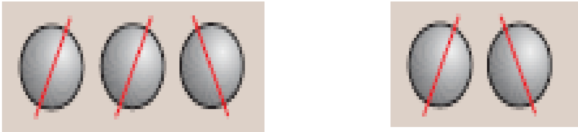

Fig. 3 bearing arrangements (a) Triplet bearing, (b) Duplex bearing [8]

The stiffness can be calculated for 15^0 contact angle [2], for medium preloading and for ϕ55, ϕ60, ϕ65 mm diameters bearings. From the FAG Spindle Bearing catalogue: [6]

For Triplet arrangement, Stiffness K = 1.36*6*C_a

For Duplex arrangement, Stiffness K = 6*C_a

Where C_a = stiffness value of the bearing in N/µm

1) For dia. 60 mm front triplet bearing , axial rigidity (axial stiffness value) for medium preloading condition is C_a =71.6 N/µm. [8]

K= 1.36*6*71.6= 582.64 N/µm.

2) For dia. 55 mm front duplex bearing , axial rigidity (axial stiffness value) for medium preloading condition is C_a =67.2 N/µm. [8]

K= 6*67.2= 403.2 N/µm

Specifications, size, & stiffness of different bearings is given and calculated by FAG Spindle Bearings is shown in the table

Table: 1Specifications, size, & stiffness of different bearings[8]

Sl. No	Details	Type and Specification	Size Details	Stiffness (N/µm)	Load rating dynamic C (kN)	Load rating static Co (kN)	Attainable Speed in (r/min)
1A	Front bearing	HSS 7011 CT P4S Duplex	d=55mm D=90mm B=18mm	403.2	1860	1900	20000
2A	Front bearing	HSS 7012 CT P4S Duplex	d=60mm D=95mm B=18mm	428.4	1930	2000	19000
3A	Front bearing	HSS 7013 CT P4S Duplex	d=65mm D=100mm B=18mm	453.0	2000	2160	17000
1B	Front bearing	HSS 7011 CT P4S Triplet	d=55mm D=90mm B=18mm	548.3	1860	1900	20000
2B	Front bearing	HSS 7012 CT P4S Triplet	d=60mm D=95mm B=18mm	582.64	1930	2000	19000
3B	Front bearing	HSS 7013 CT P4S Triplet	d=65mm D=100mm B=18mm	616.08	2000	2160	17000
4R	Rear bearing	HSS 7011 CT P4S Duplex	d=55mm D=90mm B=18mm	403.2	1860	1900	20000

3.4 Optimum Bearing Span Length

The front bearing set should be positioned to minimize the overhang of the spindle nose. It is required to optimize the bearing spacing 'L_0' for maximum spindle stiffness. This requires examination of relative combinations to deflection, which arise from both bearing deflections and spindle bearing. By using bearings with a smaller cross section, larger spindle diameters can be used without changing the housing bore diameter or slightly increasing the housing bore diameter. This increases the bending rigidity of spindle and leads to increase in overall rigidity. Initially optimum bearing span is calculated using eq. 3for each configuration. Fig.4.shows the influence of spread on spindle deflection at point of load. Details of different bearing arrangements and calculated optimum bearing span length (L_0) are listed in table .

Equation to calculate Optimum Bearing Span Length is [11]

---------- $$L_0 = \left[6EI_L \left(\frac{1}{S_A} + \frac{1}{S_B} \right) + \left(\frac{6\,EI_a}{a\,S_A} \right) Q \right]^{1/3} \quad \text{(Eqn. 3)}$$

Where,

L_0 = Static optimum Bearing Span Length in mm

Q =Trial value for iterative determination of L_0 = 4a

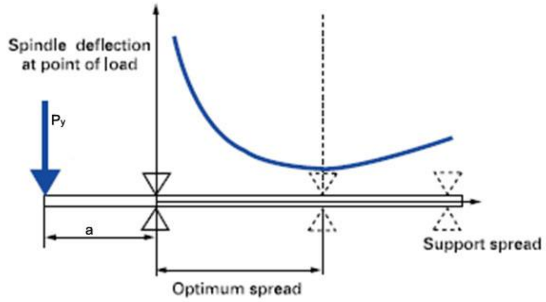

Fig. 4Influence of spread on spindle deflection at point of load

3.5. Calculation to find Optimum Bearing Span Length:

$$L_0 = \left[6EI_L \left(\frac{1}{S_A} + \frac{1}{S_B} \right) + \left(\frac{6\,EI_a}{a\,S_A} \right) Q \right]^{1/3}$$

E = Young's modulus = $2.1\text{x}10^5$ N/mm^2
S_A= Stiffness of front bearing = $403.2\text{x}10^3$ N/mm (for dia. 55mm)
S_B = Stiffness of rear bearing = $403.2\text{x}10^3$ N/mm (for dia. 55mm)
I_a = Second moment of area = it is calculated and found to be $379.84\text{x}10^3$ mm^4
I_L = Second moment of area of the shaft at the span in mm^4
= it is calculated and found to be $562.5\text{x}10^3$ N/mm
a = Length of overhang = 65 mm
Q = Trial value for iterative determination of L_0 = 4xa = 4x65 = 260
For First trial value take Q =260mm, therefore
L_1 = 202.12mm
For second trial value take Q =202.12mm, therefore
L2= 193.10mm
For third trial value take Q =193.10mm, therefore
L_3 = 191.62mm
For fourth trial value take Q =191.62mm, therefore
L_4 = 191.37mm=L_0

Table: 2 Details of different bearing arrangements with optimum bearing span length (L_0).

Sl. No	Front bearings	Bearing Stiffness N/μm	Rear bearings	Bearing Stiffness N/μm	Attainable speed in rpm	Optimum bearing span length 'L_0' mm
'A'	Ø55 x 90 x 18 Duplex	403.2	Ø55 x 90 x 18 Duplex	403.2	20000	193.33
	Ø60 x 95 x 18 Duplex	428.4			19000	205.52
	Ø65 x 100 x18 Duplex	453.0			17000	222.39
'B'	Ø55 x 90 x 18 Triplet	548.3	Ø55 x 90 x 18 Duplex	403.2	20000	175.53
	Ø60 x 95 x 18 Triplet	582.64			19000	186.49
	Ø65 x 100 x18 Triplet	616.08			17000	199.85

Analylitical analysis has been carried to evaluate stiffness of the spindle and to optimize the design to have maximum spindle nose deflection. Varying the span length from 140 mm to 230 mm. This span length variation becomes important to accommodate the integral motor rotor. By considering overhang of the spindle is around 65 mm from front bearing center and spindle nose size BT-40 taper also the flange. Front bearingdiameter is varied from 55 to 65 mm to evaluate its role in imparting the stiffness to the spindle system.

3.5. Calculation of spindle deflection

$$\delta = P\left[\frac{1}{S_A}\left(\frac{a+L}{L}\right)^2 + \frac{1}{S_B}\left(\frac{a}{L}\right)^2 + \frac{a^2}{3E}\left(\frac{L}{I_L}+\frac{a}{I_a}\right)\right]$$ ------------(Eqn. 2)

Where,

a = Length of overhang in mm = 65 mm

L = Bearing span in mm = 140 mm

E = Young's modulus of Spindle material in N/mm^2 = $2.1x10^5$ N/mm^2

P = Cutting force in N =4502.8 N

S_A = Stiffness of the front bearing N/mm = $403.2x10^3$ N/mm (for dia. 55mm)

S_B = Stiffness of the rear bearing in N/mm = $403.2x10^3$ N/mm (for dia. 55mm)

I_L = Second moment of area of the shaft at the span in mm^4

= it is calculated and found to be $562.5x10^3$ mm^4

= π x $(60^4 – 35^4)/64$ = $562.5x10^3$ mm^4

I_a = Second moment of area of the shaft at the overhang in mm^4

Consider BT-40 Taper. Front taper side is ϕ44.45mm and rear taper is ϕ24.5mm.

Average Diameter at the mid of BT-40 Taper is = (44.45+24.5) / 2 = 34.475mm

I_a = Second moment of area = it is calculated for ϕ55, ϕ60 and ϕ65mm and found to be

= π x $(55^4 – 34.475^4)/64$ = $379.84x10^3$ mm^4

= π x $(60^4 – 34.475^4)/64$ = $566.83x10^3$ mm^4

= π x $(65^4 – 34.475^4)/64$ = $806.9x10^3$ mm^4

eq. 3.3➔ For bearing A type and front & rear dia. is 55mm then deflection is found to be

$\delta = 39.07x10^{-3}$ mm.

Variations of deflection and stiffness values for bearing arrangements ‘A’ and ‘B’ are given in following table.

Table: 3 Variations ofDeflections and Stiffness for bearing arrangements ‘A’ & ‘B’ .

Sr. No.	Front bearing dia.mm	Span Length Mm	Bearing arrangement ‘A’		Bearing arrangement ‘B’	
			Deflection δ µm	Stiffness K N/µm	Deflection δ µm	Stiffness K N/µm
1	55	140 to 230	39.07 to 36.81	115.2 to 122.3	32.70 to 31.91	137.5 to 141.1
2	60	140 to 230	35.93 to 34.00	125.3 to 132.4	29.97 to 29.43	150.2 to 155.0
3	65	140 to 230	33.67 to 32.03	133.7 to 140.5	27.99 to 27.70	160.8 to 162.5

Optimum deflection and stiffness values for bearing arrangements ‘A’ and ‘B’ are given in the table 4.

Table: 4 Optimum Deflections and Stiffness for bearing arrangements ‘A’, ‘B’.

Front bearing dia mm	A			B		
	Span length mm	Δ µm	K N/ µm	Span length Mm	Δ µm	K N/ µm
55	210	36.69	122.7	190	31.47	142.9
60	200	33.84	133.06	190	28.9	155.8
65	200	31.80	141.6	**180**	**27.08**	**166.2**

The diameter of the spindle between the bearings has more influence on the rigidity as is evident from the table3. This diameter could be varied to the extent of a maximum of 65 mm from the practical considerations. To accommodate the integral motor rotor, the span length is taken as 200 mm [Table 2]. At the 65 mm diameter & 200mm span length, the stiffness of the arrangement ‘B’ has a value of around 166.2 N/µm and the deflection of the spindle nose based on static analysis is around 27.08 micron. From the point of view of static analysis, bearing arrangement type ‘B’ is chosen as an optimum design.

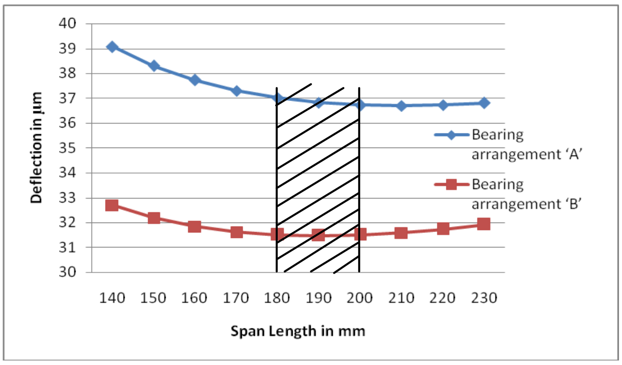

Fig. 5 Deflection at the nose for bearing arrangements (front bearing dia 55mm)

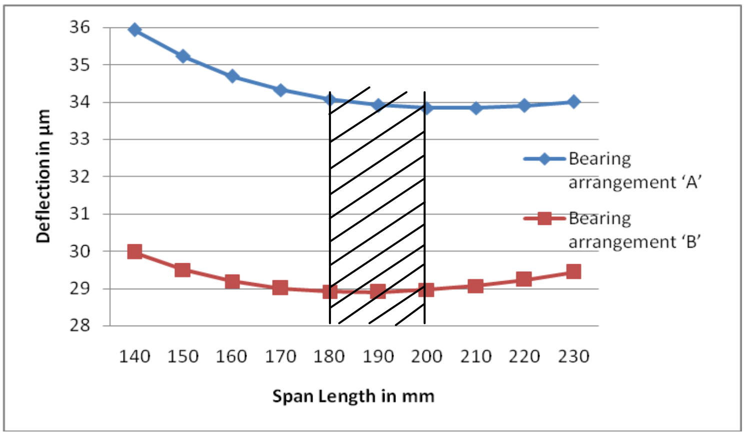

Fig. 6 Deflection at the nose for bearing arrangements (front bearing dia 60mm)

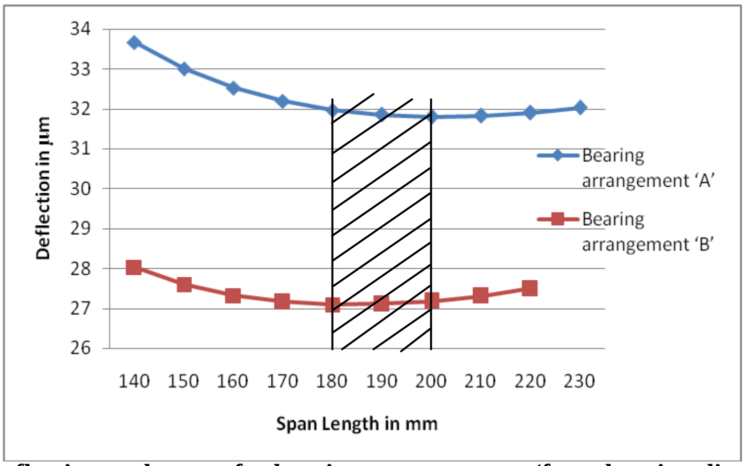

Fig. 7 Deflection at the nose for bearing arrangements (front bearing dia 65mm)

Fig. 5 to 7 shows the effect of bearing elasticity and spindle deflection on the overall rigidity.Variation of deflection and the stiffness when the bearing span is changed from 140 to 230 mm is not significant. This gives the designer flexibility in sizing the span to accommodate the integral induction motor [11]. As there is not much difference in the spindle stiffness in 180mm, 190mm, 200mm span length, to accommodate the integral motor 200mm span length will be considered

IV. Static Stiffness Analysis By Fem

4.1 Element Description: Beam Elements

There are several element types used for FE analysis such as solid, plane and beam elements. For the spindle analysis for the bearing spans optimization, beam elements are the most suitable because the shapes of the spindle and the cutting tool are roughly cylindrical. Besides a time consuming iteration method will be used for the optimization, so the calculation time for the dynamic properties in each iteration step must be as short as possible. In order to shorten the calculation time, the simplest elements (the beam elements) are the most suitable type.[11]

4.2 Static Stiffness Analysis

The geometric model is created in ANSYS. The finite element model is built using BEAM3 and COMBIN14 element[11].

4.2.1 Material properties

For the static analysis of the spindle, the following propertiesare used:[11]

- Modulus of elasticity =210 GPa
- Poisson'sratio =0.27
- Density =7.8×10^{-6} kg/mm^3

4.2.2 Boundary Conditions

The spindle is modeled using BEAM3 and COMBIN14 elements. A tangential force of**4502.8 N** is applied at the spindle nose as shown in fig. 8.

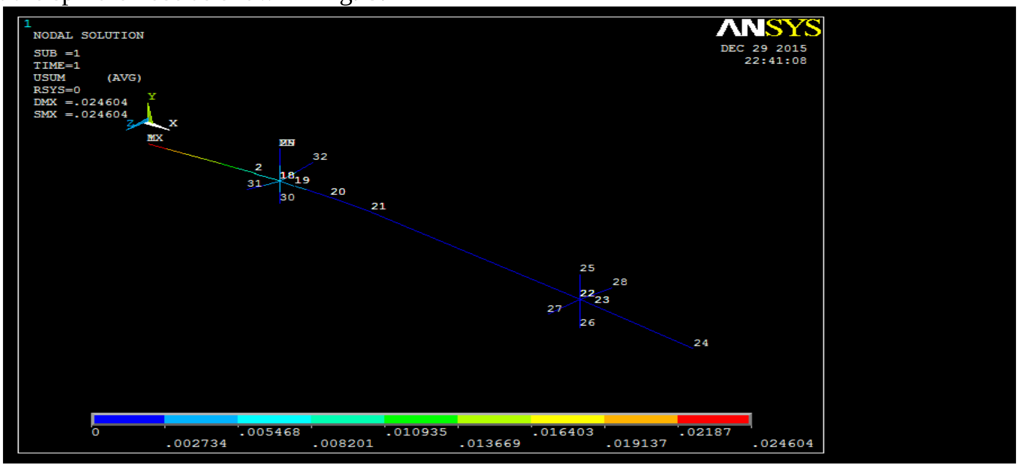

Fig.8 FEM showing Boundary Condition[11]

Table: 5 Comparison of Theoretical & FEA values for deflection and stiffness at the spindle nose whenspan length= 200 mm with'B' type bearing arrangement.[11]

Span length Mm	THEORETICAL		ANSYS	
	Deflection at spindle nose µm	Stiffness N/µm	Deflection at spindle nose µm	Stiffness N/µm
170	27.14	165.9	23.72	189.8
180	27.08	166.2	23.94	188.0
190	27.07	166.3	24.24	185.7
200	27.14	165.9	24.60	183.0

4.3 Static Results

The deflection at the spindle nose in Y direction is computed for various configurations and results are obtained for different diameters of spindle at the front bearing with 'B' type arrangement and span length of 200 mm as shown in fig. 9.

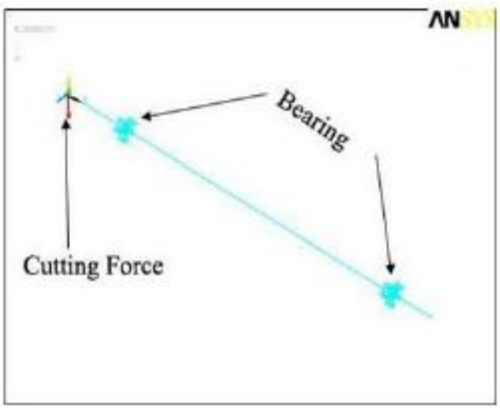

Fig.9Deflection at the spindle nose (front bearing Ø65mm, span length =200 mm)

Table:6Comparison of Theoretical & FEA values for deflection and stiffness at the spindle nose when the s pan length various from 170 mm to 200 mm for the front bearing with diameter of 65 mm with 'B' type Bearing arrangement [11]

Diameter at Front bearing Mm	THEORETICAL		ANSYS	
	Deflection at spindle nose	Stiffness N/µm	Deflection at spindle nose	Stiffness N/µm
55	31.50	142.8	25.07	179.6
60	28.96	155.4	24.19	186.1
65	27.14	165.9	24.60	183.0

The deflection at the spindle nose in Y direction and the stiffness vlaues obtained through theoretical calculations and ANSYS are given in the table.5 and 6. The diamenter could be varied to the extent of a maximum of 65 mm from the practical considerations. The stiffness obtained by ANSYS of the optimized spindle with given configuration is around 183.0 N/ µm at the span length of 200 mm.

In this we consider Triplet bearing set arrangement 'B' for the front end of the spindle, in which one pair of angular contact ball bearings are arranged in tandem with respect to each other and back-to-back with respect to a single angular contact ball bearing. Tandem bearing pair will carry both radial and axial loads equally shared. [11]

4.4 Analysis of Results

The deflection of the spindle nose in Y direction is function of bearing stiffness, span length and diameter of the spindle bearing for a given overhang. The deflection of the spindle nose in Y direction affects the machining accuracy.[11]

V. Conclusions

- Span length between bearing is varied from 140 to 230 mm which gives flexibility in sizing the span to accommodate the integral induction motor.
- From results it is clear that Rigidity of the spindle is depends on the diameter of the spindle between the bearing. So from practical considerations of bearing speed, the diameter of the spindle varied from 55 to 65 mm maximum and result shows that 65mm diameter spindle for the arrangement 'B' has getting a stiffness value 165.9 N/µm and 27.14 mm is the deflection of the spindle nose based on static analysis for above arrangement. Therefore bearing arrangement type 'B' is chosen as an optimum design from static analysis results.
- The effect of front bearing stiffness is on overall stiffness, hence locate higher siffness bearings at the front.
- Rigidity analysis was carried out by using ANSYS and the result gives good correlation between theoretical calculation and ANSYS result.

References.

[1]. ANSYS Basic Analysis Procedures Guide, 002184, Aug2005, www.ansys.com.Book of Spindles, www.dynospindles.com.
[2]. CMTI Hand Book, Machine Tool Design Hand Book Tata McGraw-Hill Company Limited, 1983,421-588
[3]. Daniel P. Soroka, Hardinge Inc. One Hardinge Dr.Elmira,NY 14902 Hard Turning and the Machine Tool.
[4]. dsoroka@hardinge.com
[5]. Dr.Sinan Badrawy, Dynamic Modeling and Analysis ofMotorized Milling Spindles, Moore Nanotechnology Systems,LLC, 2003.
[6]. FAG Spindle Bearings Catalogue, AC 41 130/7 EA ·March 2008, www.fag.com.
[7]. HMT, Production Technology, Tata McGraw HillPublication, 1980, 7th Print, page 186-198.
[8]. INA-FAG BEARINX®-online Spindle Calculation-SCHAEFFLER GROUP Industrial.
[9]. Lipka I, Harkany, Determination of the optimum bearingdistance of cantilever shafts with two supports, Mach. Tool. Research Transactions, 45 (1964) 65-80.
[10]. Machine Manual, VMC 1000, The Mysore Kirloskercompany, Harihar.
[11]. Anandkumar Telang, Static stiffness analysis of high frequency milling spindle, eISSN:2319-1163/ pISSN:2321-7308
[12]. Modal analysis using ANSYS,www.mece.ualberta.ca/tutorials/ansys.
[13]. Momir Sarenac, Stiffness of Machine Tool Spindle as amainfactor for treatment accuracy-UDC 62-113,624.046,Mechanical Engineering Vol.1, No 6, 1999, pp. 665 – 674. Spindle Basics, 2008, www.spindlesworld.com.
[14]. Technical Specifications for IBAG-Motor Spindles,IBAG Switzerland AG, 1999, www.ibag.ch.
[15]. TIMKEN, Machine Tool Spindle, 1M-07-02 No. 5911,www.timken.com.
[16]. Yung C. Shin, Analysis of bearing configuration effectson high speed spindles, International Journal of Machine Tools & Manufacture, 44 (2004) 347–364.83, 421-588.