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ANALYSIS OF HIGH-SPEED MULTI BEARING SPINDLE SYSTEM

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Abstract: - One of the most critical components of any high speed spindle design is the bearing system. Our design requirements state that the spindle must provide high rotational speed, transfer torque and power to the cutting tool, and be capable of reasonable loading and life. Deflection of the spindle system depends on the applied load and stiffness of the bearing. Stiffness of the bearing in turn depends on radial reaction at the bearing and location of the bearing. Estimation of the accurate bearing stiffness plays equal importance in finding deflection of the spindle system. In conventional method the stiffness of the bearing is computed by approximately radial reaction, the obtained stiffness is assumed to be constant and computed stiffness input in to the model. A new method is such required for considering precise inter dependence of spindle shaft deflection and bearing stiffness. The objective of the project is to develop a method iteratively to compute the appropriate spindle deflection, bearing reaction and bearing stiffness by varying required bearing number and by varying the span bearing span.

Keywords:- *Spindle, Stiffness of bearing, Radial reaction, Torque, Deflection.*

I. INTRODUCTION

In modern applications, there is a growing demand for significantly higher machine tool capabilities. Since the main objective of high power machining is to increase the metal removal rate (MRR) by adopting more aggressive cutting condition that may cause chatter vibration and Spindle failure, the machine tools must have higher rigidity, stability and reliability. Among all the component of a machine tool, the spindle system is the most critical part, since its dynamic properties directly affect the cutting ability of the whole machine tool.

Figure 1.1 Spindle system

A typical pulley-drive spindle system for millings is shows in Figure 1.1 consists of the tool, tool holder, spindle shaft, bearings, pulley, clamping unit, and the housing attached to the machine tool. Traditionally the spindle stiffness is computed by modeling the spindle shaft as a beam with varying cross section, and supported in multiple flexible spring supports. The lacuna with this method is that the stiffness of the bearing, which is being inputted, is in actual practice not constant, but

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that it self depends on the reaction at the support bearing. This method can lead over designed bearings. Bearing stiffness depends on the structure of the bearing, such as size of the bearing balls, curvature of the bearing rings, contact angle, preload, deformation of the spindle shaft and housing, thermal expansion and spindle speeds. The dynamics of the spindle system changes with the cutting force and spindle speed, especially when the preload is low. All these elements lead to a non linear problem for the spindle system.

Hence, accurately predicting the stiffness of the machine tool spindle system is the key issue in the design of spindles, enabling designer to predict the performance of spindles before they are manufactured. The objective of this work is to develop a general method for predicting deflection of a spindle system, so that designer can check the performance of the spindle system and improve the machine tool design before it is physically manufactured. This general model can be used to predict the reaction load stiffness of bearings and bearing life.

II. LITERATURE SURVEY

Kang et al, presented both static and dynamic analyses to examine the necessary integrated procedures for the design of spindle-bearing systems, with modeling and analysis of these systems being based on the finite element method. Within this study, the effects of design parameters on static and dynamic performance of spindle-bearing systems are analyzed in order to establish the requirement fordesign modifications, and the paper proposes a number of examples, along with a set of guidelines for the design of machine tool spindles.[1]

Osamu maeda et al, presented an expert spindle design system strategy which is based on the efficient utilization of past design experience, the laws of machine design, dynamics and metal cutting mechanics. The paper provides a set of fuzzy design rules, which lead to an interactive and automatic design of spindle drive configurations. The arrangement of bearings is optimized using the Sequential Quadratic Programming (SQP) method. In this study accurate stiffness of spindle system is not computed which may lead to improper designed bearings. After computing accurate stiffness the above method can be used to optimize bearing span.[2]

Jin Kyung Jin Kyung Chio and Daigil Lee adopts the design and manufacturing methods as well as static and dynamic characteristics of carbon fiber-epoxy composite spindle bearing system were investigated using analytical and experimental methods to improve the performance of the spindle-bearing system. As the carbon fiber epoxy composite material has excellent properties for structures, owing to its high specific modulus, high damping and low thermal expansion, the vibration and thermal characteristics of spindles will be improved if the proper design and manufacturing methods for the composite spindle are developed. Varying cross section of multi stepped shaft manufacturing, computing accurate stiffness, radial load and deflection is not discussed. [3]

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III. MODELING AND ANALYSIS

3.1. Modeling and Analysis of Two Bearing Spindle System

A beam held in two support is statically determinate system the reaction can be computed in go. Without the need for iterative procedure is explained in this article. Fig 3.1 shows model of two bearing spindle system, the bearing parameters and spindle shaft dimensions are listed below.

Bearing parameters

Both A and B are angular contact ball bearing and are having same dimensions
Outer diameters (OD) – 90 MM
Inner diameters (ID) – 50 mm
Number of row of rolling elements (I) – 1
Contact angles (α) – 15 degree

Spindle shaft dimensions:

Outer diameter (OD)- 50 mm
Inner diameter (ID) – 5mm
Length of the spindle shaft – 140 mm
Young's modules – 98100 N/mm^2

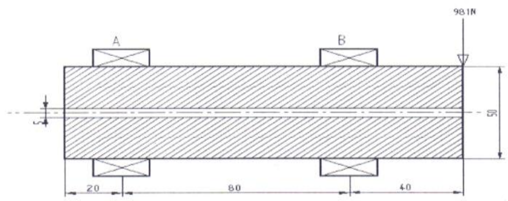

Figure 3.1 Model of two bearing spindle system

Figure 3.2 shows the FEM model of the spindle shaft for computing reactions at two bearings support with simply supported boundary conditions. A load of 981 N is applied at the spindle nose.

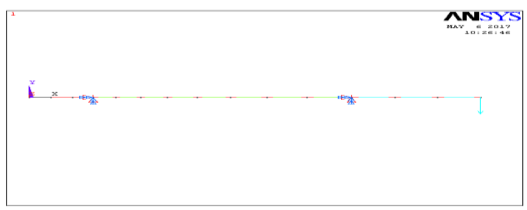

Figure 3.2 FEM of the spindle shaft with two simply supported boundary conditions

Table 3.1 radial reaction values at two bearings support for applied load and simply supported boundary conditions.

Table 3.1 Value of Fr

Node number	Fr
1	-490.5
2	1471.5

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The dimensions of the rolling elements can be calculated by following formula [8].

1. Rolling element diameter D_W= q1 (D-d) mm (2.1)
2. No. of rolling element Z = q2 [D+d/Dw] (2.2)

Table 3.2 Value of q1 & q2

Sl. No	Type of bearing	q1		q2	
1	Angular contact bearing	From	To	From	To
		0.25	0.32	1.40	1.24

Table 3.3 Equations for Calculating the Elastic Deformation of the Bearing and Rolling Element Load

Type of bearing	Loading condition δa=0
Angular contact ball bearing	$\delta r = (0.002/ \cos \alpha)\ ^3\sqrt{}\ (Q^2/Dw)$
Rolling element load	$Q= 5\ Fr / (\ IZ. \cos \alpha)$

In case of bearing transmitting both axial and radial loads the loading condition under which δa = -0 occurs when Fa=1.25 Fr tan α for other condition and data on dimension of the bearing element reference may be made to bearing manufacturer, δr = 0, where Fr = 0.

Figure 3.3 shows FEM model of the above spindle system taking in to consideration the bearings. The bearings are modeled here as 3D elements (MATRIX27). A load of 981N is applied at the spindle nose.

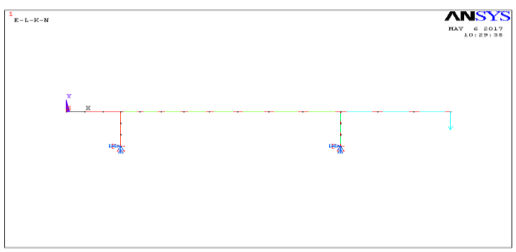

Figure 3.3 FEM Model of spindle system with two bearings as 3D spring element.

Table 3.4 Analytically Computed Values of Radial Reaction, Radial Displacement and Radial Stiffness of Two Bearings Spindle system.

Iteration	Node	F_r (N)	Bearing	Bearing Parameters						D_w (mm)	Z	δ_r (mm)	Kr (N/mm)
				OD (mm)	ID (mm)	q1	q2	I	A				
1	1	-490.5	A	90	50	0.285	1.32	1	15	11.4	16	0.0058	84568.96
	2	1471.5	B	90	50	0.285	1.32	1	15	11.4	16	0.0122	120614.75
2	1	-490.5	A	90	50	0.285	1.32	1	15	11.4	16	0.0058	84568.96
	2	1471.5	B	90	50	0.285	1.32	1	15	11.4	16	0.0122	120614.75

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The above table shows the computed bearing radial reactions with out and with bearing parameters remains unchanged. As such, the need for iterative procedure does not arise in two bearing spindle system. As the reactions at bearing do not change, iterative procedure required to compute the bearing reactions.

3.2. Modelling and Analysis of Three Bearing Spindle System

A beam supported in three support becomes statically indeterminate system, the reaction cannot be computed in one go. As such there is a need for iterative procedure to compute exact reactions at bearing, stiffness and deflection of spindle system. The bearing parameters and spindle shaft dimensions are listed below both A and B and C are angular contact ball bearing and are having same dimensions

Outer diameters (OD) – 90 mm
Inner diameters (ID) – 50 mm
Number of row of rolling elements (I) – 1
Contact angles (α) – 15 degree

Spindle shaft dimensions:
Outer diameter (OD)- 50 mm
Inner diameter (ID) – 5mm
Length of the spindle shaft – 140 mm
Young's modules – 98100 N/mm^2

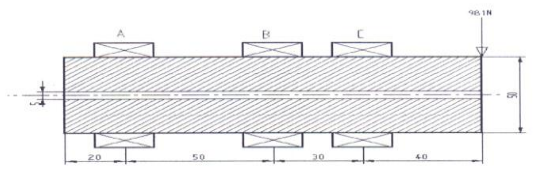

Figure 3.4 Model of three bearing spindle system

3.2.1. Analytical approach

In analytical approach first the reactions are computed for simply supported condition. The obtained reactions are used to compute stiffness of bearing through long hand calculations. Computed stiffness is given to bearing by 3D spring element [MATRIX27]. Figure 3.5 shows the FEM model of the spindle shaft for computing reactions at three bearings support with simply supported boundary conditions. A load of 981 N is applied at spindle nose.

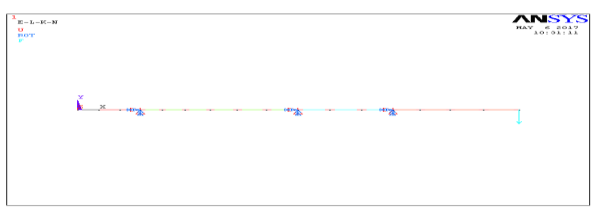

Figure 3.5 FEM Model of the spindle shaft with three simply supported boundary conditions

Table 3.5 Radial reaction values at three bearings support for applied load & supported boundary conditions

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Node number	Fr
1	-162.15
2	-875.54
3	2018.70

Figure 3.6 shows FEM model of the above spindle system taking in to considerations the bearings. The bearings are modelled here as 3D spring element [MATRIX27]. A load of 981 N is applied at the spindle nose.

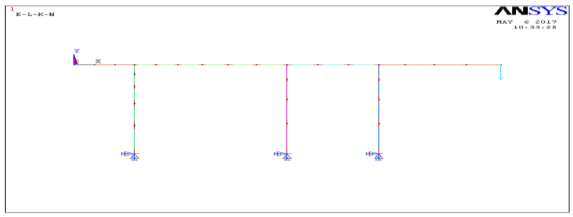

Figure 3.6 FEM Model of spindle system with three bearings as 3D spring element.

Table 3.6 Analytically Computed Values of Radial Reaction, Radial Displacement

Iteration	F_r (N)	Bearings	Bearing Parameters						D_w (mm)	Z	δ_r (mm)	Kr (N/mm)
			OD (mm)	ID (mm)	q1	q2	i	α				
1	-162.5 -875.4 2018.7	A B C	90 90 90	50 50 50	0.283 0.283 0.283	1.32 1.32 1.32	1 1 1	15 15 15	11.4 11.4 11.4	16 16 16	0.0028 0.0086 0.0131	57707.91 101218.69 134380.03
2	-363.23 199.3 1346.7	A B C	90 90 90	50 50 50	0.283 0.283 0.283	1.32 1.32 1.32	1 1 1	15 15 15	11.4 11.4 11.4	16 16 16	0.0064 0.0032 0.0115	87362.07 61526.16 117230.78
3	-380.83 241.03 1320.7	A B C	90 90 90	50 50 50	0.283 0.283 0.283	1.32 1.32 1.32	1 1 1	15 15 15	11.4 11.4 11.4	16 16 16	0.0065 0.0036 0.0113	88160.31 63805.28 116842.2
4	-386.4 283.84 1311.5	A B C	90 90 90	50 50 50	0.283 0.283 0.283	1.32 1.32 1.32	1 1 1	15 15 15	11.4 11.4 11.4	16 16 16	0.0066 0.0038 0.0113	88380.66 66916.63 116209.33

The Table 3.6 shows the output of the analytical method, the table consists of radial reaction F,, rolling element diameter Dw, number of rolling element Z, radial displacement 6, and radial stiffness Kr. It can be seen that there is a drastic variation of the bearings radial reaction from iteration 1 to iteration 2. To get accurate radial load, radial displacement and radial stiffness iterative procedure is adopted till the radial load values converge.

IV. RESULTS AND DISCUSSION

4.1. Computation and Radial Reaction of Bearings

Precise estimation of radial reaction, radial stiffness and radial deflections of multi-bearing spindle system using APDL approach is disused in this chapter. In first iteration, the bearing supports are

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modeled as rigid elements. In the other words bearing supports are modeled as rigid radial constraint and free rotation. Considering the radial reaction at the supports, the approximate stiffness is computed iteratively. Figure 4.1 shows spindle shaft with simply supported conditions to compute radial reaction at bearing supports.

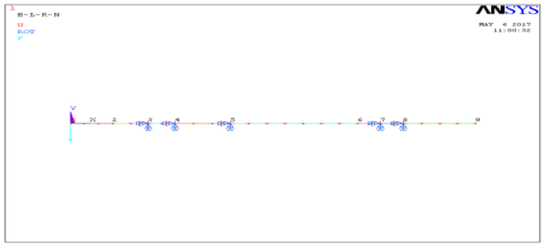

Figure 4.1 FEM model of the spindle shaft with simply supported boundary conditions

Table 4.1 shows the output of the iterative procedure. The table shows radial reaction Fr, rolling element diameter Dw, number of rolling elements Z, radial displacement Or and radial stiffness Kr. It can be seen that in iteration 5 the radial reaction at bearing C by considering bearing parameter is 948.03N, where as in iteration 6 the value is 949.21N. It shows a nearer convergence, the results of iterative procedure output are corresponding to 0.1% convergence.

Iteration	NODE	F_r (N)	Bearing	Bearing parameter OD (mm)	ID (mm)	Q1	q2	i	α	D_w (mm)	Z	δr (mm)	Kr (N/mm)
1	1	21324.9	A	150	85	0.285	1.32	1	15	18.52	17	0.0618	344592.24
	2	-3098.74	B	150	85	0.285	1.32	1	15	18.52	17	0.0238	213874.38
	3	-3785.44	C	150	85	0.285	1.32	1	15	18.52	17	0.0259	223072.21
	4	-840.71	D	125	70	0.285	1.32	1	15	15.67	16	0.0075	110966.01
	5	210.22	E	125	70	0.285	1.32	1	15	15.67	16	0.0030	69894.28
2	1	9041.38	A	150	85	0.285	1.32	1	15	18.52	17	0.0349	258872.93
	2	3618.81	B	150	85	0.285	1.32	1	15	18.52	17	0.0189	190778.99
	3	594.78	C	150	85	0.285	1.32	1	15	18.52	17	0.0056	104505.43
	4	-2043.81	D	125	70	0.285	1.32	1	15	15.67	16	0.0137	149159.13
	5	-1401.16	E	125	70	0.285	1.32	1	15	15.67	16	0.0106	131519.13
3	1	8239.13	A	150	85	0.285	1.32	1	15	18.52	17	0.0328	251187.69
	2	4269.8	B	150	85	0.285	1.32	1	15	18.52	17	0.0211	201594.12
	3	839.93	C	150	85	0.285	1.32	1	15	18.52	17	0.0071	117244.11
	4	-1799.4	D	125	70	0.285	1.32	1	15	15.67	16	0.0125	142956.71
	5	-1760.11	E	125	70	0.285	1.32	1	15	15.67	16	0.0124	141907.63
4	1	8001.82	A	150	85	0.285	1.32	1	15	18.52	17	0.0321	248545.2
	2	4485.13	B	150	85	0.285	1.32	1	15	18.52	17	0.0218	204927.36
	3	920.07	C	150	85	0.285	1.32	1	15	18.52	17	0.0076	120861.16
	4	-1717.73	D	125	70	0.285	1.32	1	15	15.67	16	0.0122	140739.0
	5	-1879.30	E	125	70	0.285	1.32	1	15	15.67	16	0.0129	145042.8
5	1	7920.10	A	150	85	0.285	1.32	1	15	18.52	17	0.0319	247696.41
	2	4555.36	B	150	85	0.285	1.32		15	18.52	17	0.0221	205989.49
	3	948.03	C	150	85	0.285	1.32	1	15	18.52	17	0.0077	121835.40
	4	-1696.04	D	125	70	0.285	1.32	1	15	15.67	16	0.0120	140031.07
	5	-1917.16	E	125	70	0.285	1.32	1	15	15.67	16	0.0131	146008.80
6	1	7894.40	A	150	85	0.285	1.32	1	15	18.52	17	0.0319	247479.40
	2	4378.03	B	150	85	0.285	1.32	1	15	18.52	17	0.0221	208086.27
	3	949.21	C	150	85	0.285	1.32	1	15	18.52	17	0.0077	122002.42
	4	-1682.61	D	125	70	0.285	1.32	1	15	15.67	16	0.0121	139062.67
	5	-1929.03	E	125	70	0.285	1.32	1	15	15.67	16	0.0131	148382.92

Figure 4.2a and Figure 4.2b shows the final deflection of the spindle system considering things, after computing the converged radial reaction by iterative procedure approach deflection of the system is computed. The deflection at nose for cutting load of 9810N is 0.067528 mm.

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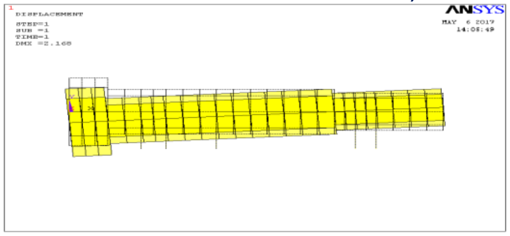

Figure 4.2a Deflection of the spindle system

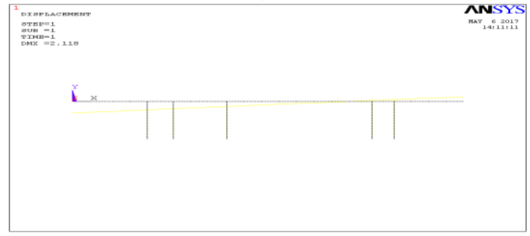

Figure 4.2b Deflection of the spindle system

After computing the converged radial load by APDL approach, the radial reaction is used in computing precise bearing life of the spindle system. This helps in proper designing of the spindle system.

Calculation of Bearing Life

Calculation of Bearing Life Table 4.2

Bearing	Fr (N)	F_a, (N)	X	Y	F (N)	C (N)	Lh hours
A	7894.40	2452.5	1	0	7916.47	95600	61658.93
B	4578.03	2452.5	1	0	4573.22	95600	316167.65
C	949.21	2452.5	0.5	0.26	1112.25	95600	22046829.8
D	-1682.61	0	1	0	1699.87	71500	2283498.10
E	-1929.03	0	1	0	1929.03	71500	1513455.85

Natural frequency

The first two natural frequencies are 685.05 and 970.85Hz. The spindle in the machine runs at Maximum speed of 5000rpm. These works to a working frequency of 5000/60 = 83.33Hz. Now the working frequency and the system natural frequency 685.05Hz are quite for away. As such no resonance is likely to occur.

Analysis of the Spindle System with Bearing Span Variation

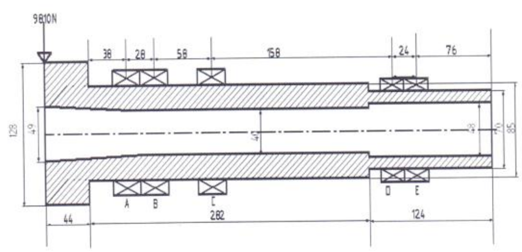

Figure 4.3 Model OD spindle system with bearing span variation

Table 4.4 shows the iteratively computed deflections values at spindle nose by APDL approach for bearing span variation.

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Varying bearing span (mm)	Stiffness of spindle system (N/mm)
28	0.06600
38	0.06670
48	0.06720
58	0.06752
68	0.06774
78	0.06785
88	0.06791
98	0.06792
108	0.06791
118	0.06790
128	0.06787
138	0.06782
148	0.06776
158	0.06767
168	0.06759

Figure 4.4 shows the plot of variation of deflection at spindle nose of spindle system by varying span between bearing b & C. with increase in the span from 58 mm to 168 mm the deflection at spindle nose gradually increases. Further by increasing the span from 58 mm to 28 mm the deflection at spindle nose decreases. The minimum deflections of 0.06600 mm is computed at bearing span of 28 mm.

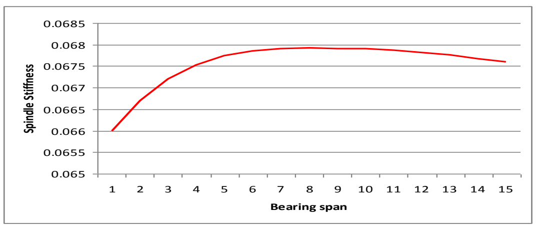

Figure 4.4 Deflections at spindle nose by varying bearing

Table 4.5 Stiffness of Spindle System by Varying Bearing

Varying bearing span (mm)	Stiffness of spindle system (N/mm)
28	148636.36
38	147076.46
48	145982.14
58	145290.28
68	144818.42
78	144583.64
88	144455.89
98	144434.62
108	144455.89
118	144477.17
128	144541.03
138	144647.51
148	144775.67
158	144968.22
168	145139.81

Figure 4.5 shows plot of variation for stiffness of spindle system by varying span between Bearing B and C. With increase in the span from 58 mm to 168 mm the stiffness of the spindle System decreases. Further by decreasing the span from 58 mm to 28 mm the stiffness of the Spindle system increases. The maximum stiffness of 148636.36 N/mm is computed at bearing Span of 28 mm.

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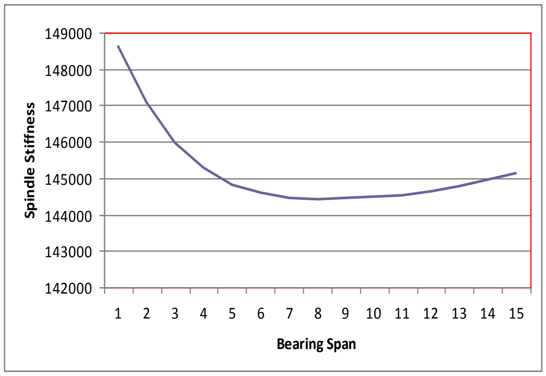

Figure 4.5 Stiffness at tool point by varying bearing span

Natural frequency

The first two natural frequencies by varying span between bearing B and C at 28 mm are 686.78 and 985.77 Hz.The spindle in the machine runs at maximum speed of 5000rpm.These works to a working frequency of 5000/60 = 83.33Hz. Now the working frequency and the system natural Frequency 686.78Hz are quite for away. As such no resonance is likely to occur. Table 6.7 shows comparisons of the bearing life for actual bearing span and bearing life for improved bearing span.

Table 4.6 Comparisons of bearing life for actual and improved bearing span

Bearings	Bearing Life			
	Redial load at actual bearing Span (N)	Bearing life for actual bearing span (hrs)	Radial load optimized bearing span (N)	Bearing life for Improved bearing span (hrs)
A	7894.40	61658.93	7351.22	76361.61
B	4578.03	316167.65	4096.44	441299.66
C	949.21	22046829.8	2044.40	3213448.13
D	-1682.61	2283498.10	-1734.50	2084625.95
E	-1929.03	1513455.85	-1947.77	1467155.86

Figure 4.6 shows plot of bearing life for actual bearing span and bearing life for improved bearing span. With the improved bearing span the life of bearing A has increased from 61658.93 hrs to 76361.61hrs.Further the variation of bearing life between bearings has decreased.

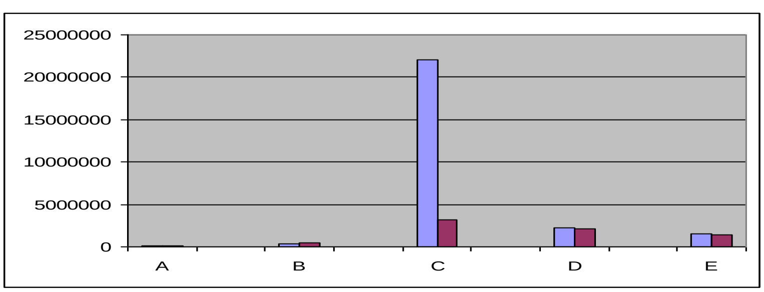

Figure 4.6 Bearing life for actual bearing span and bearing life for improved bearing span

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V. CONCLUSION AND FUTURE WORK

5.1. Conclusion

A methodology has been developed to iteratively compute appropriate radial reaction, radial stiffness and radial deflection of the multi bearing spindle system .The entire iterative process is Carried out using APDL. Exact deflection and stiffness at the tool point is computed. The reliability of APDL program was validated for there bearing spindle system with the analytical computed results of 1.19% accuracy. A case study was then conducted on CNC milling machine spindle system for which with the conventional method the radial reaction at bearings are 21324.97, -5098.74, -5785.44, -840.71, 210.22N. By varying the bearing span between two middle bearings from 58 mm to 28 mm deflection of the spindle system is reduced from 0.06752 mm to 0.06600 mm. The life of the bearings improved from 61658.93 to 76361.61hrs, 316167.65 to 441299.66 hrs, 22046829.8 to 3213448.13 hrs, 2283498.10 to 2084625.95 hrs, 1513455 to 1467155.86 hrs.

5.2. Scope For Future Work

1. The deflection of the spindle can be reduced by replacing the existing bearings with higher stiffness bearings.
2. The multi-stepped spindle shaft made of carbon steel can be replaced by composite shaft can increase spindle stiffness.
3. The program can be furthered to take of other bearing types.

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