2001 유체기계 연구개발 발표회 논문집, pp. 315 ~ 321
한국과학기술원(KAIST) 기계공학동, 2001. 11. 30 ~ 12. 1

Case History for Reduction of Shaft Vibration in a Steam Turbine

In Chul Kim* ,Seung Bong Kim**, Jae Won Jung***, Seung Min Kim****

Key Word : turbine, rotor, bearing, vibration, alignment

ABSTRACT

The shaft system of turbine is composed of rotating shaft, blades, bearings which support the shaft, packing seal which prevent the leakage of steam, and couplings which connect the shaft. Shaft system component failure, incorrect assemblage or deflection by unexpected forces causes vibration problem. And every turbine has its own characteristics in dynamic response.

In this paper we propose the three-bearing supported type rotor which is real equipment and being operated this time as commercial operation. From 1996 it has a high vibration problem and there are many kinds of trial to solve this problem. In resent outage we performed a special diagnosis and carried out appropriate work. We would like to introduce and explain about this case history.

1. introduction

Recently combined cycle steam turbines are widely using in the power plants owing to their outstanding thermal efficiency. However, high-speed rotating machines, such as combined cycle steam turbines, experience excessive shaft vibration which results in serious damage and economic losses. Such shaft vibration is mainly caused by the followings: resonance, unbalance, misalignment, and instability. Various alignment maintenance techniques have been offered to reduce the vibration magnitude as well as to improve the reliability of the rotating system.

This paper describes an update of bearing alignment method using load cell as well as laser alignment equipment, which are executed in the 75MW combined steam turbine whose rotational speed is 3,600 rpm. A rotor, which consists of high, intermediate and low pressure steam turbines, is supported by three bearings which is called as three-bearing supported-type rotor.

High vibration about 160㎛ peak to peak has been occurred at the first bearing of this unit since the beginning of the operation. It is almost impossible to adjust the bearing of the three-bearing supported-type rotor, in both the vertical and horizontal directions simultaneously, by using the traditional shaft alignment technique.

An updated alignment method proposed in the present study uses the laser equipment for measuring the bearing positions as well as load cells for measuring the bearing reaction forces. And then both the vertical and the horizontal positions are finally adjusted by the consideration of the mutual influences between the bearing positions and the reaction forces.

* : 한전기공(주), ickim@kps.co.kr
** : 한전기공(주), sbkim2@kps.co.kr
*** : 한전기공(주), jwjung@kps.co.kr
**** : 한전기공(주), kps0070@kps.co.kr

2. Shaft Vibration Before Outage

2.1 Specification of steam turbine

This combined cycle steam turbine will be discussed in this paper is characterized by a high pressure rotor with two bearings and a intermediate-low pressure rotor coupled to it with only one bearing which is located at the generator end of this low pressure rotor. The turbine end of the low pressure rotor is carried by the No.2 bearing located at the generator end of the high pressure rotor, and its schematic diagram is shown in Fig. 1.

In this reason, the dynamic characteristics of this turbine are different from that of the typical two-bearing supported turbine.

Table. 1 Specification of Steam Turbine in this project.

Type		Tandem-Compound, One-Rotor-One-Bearing
Rated Output		75MW
RPM	TBN	3600rpm
	GEN	3600rpm
Bearing Specification	No.1	5 Tilting Pad
	No.2	Shortened Elliptical
	Thrust	Kingsbury
	No.3	4 Tilting Pad
No. of Stage	HP	9단
	IP	8단
	LP	5단

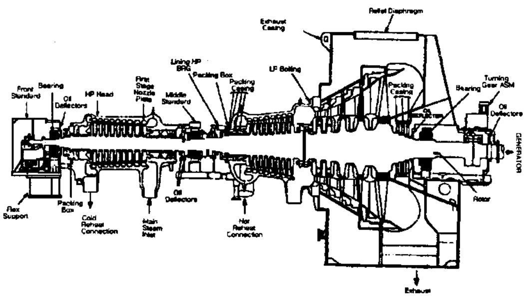
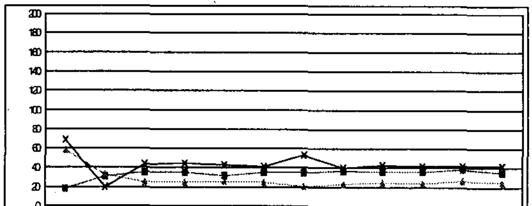


Fig. 1 Schematic diagram of One-Rotor-One-Bearing



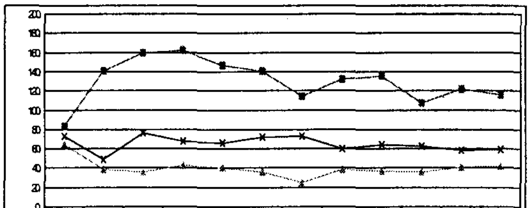
	95-11-28 (70MW)	96-04-08 (77MW)	00-07-27 (71MW)	00-08-11 (70MW)	00-08-21 (70MW)	00-09-01 (71MW)	00-09-15 (65MW)	00-09-22 (68MW)	00-10-12 (65MW)	00-11-06 (67MW)	00-11-15 (70MW)	00-11-27 (75MW)
No.1	18	31	35	35	32	35	35	37	35	35	39	35
No.2	58	33	25	25	26	25	21	23	25	24	27	25
No.3	69	20	44	45	43	41	53	40	43	42	42	42

Fig.2 Direct Vibration Value of Each Bearing(Xabs)

2.2 Vibration at No.1 Bearing

According to the investigation of vibration trend for several years, shaft vibration at No.1 bearing was changed from 35㎛p-p at 25% load up to maximum 140㎛p-p at 100% load after major outage in 1996. In spite of continuous maintenance action,

Vibration value has not changed and been kept in quite excessive state before the last January outage. Figure 2 and 3 shows the shaft vibration(X-absolute and Y-absolute) trends for several years at Nos.1, 2 and 3 Bearings in this steam turbine



	95-11-28 (70MW)	96-04-08 (77MW)	00-07-27 (71MW)	00-08-11 (70MW)	00-08-21 (70MW)	00-09-01 (71MW)	00-09-15 (65MW)	00-09-22 (68MW)	00-10-12 (68MW)	00-11-06 (67MW)	00-11-15 (70MW)	00-11-27 (75MW)
No.1	83	141	160	162	147	141	114	132	136	107	122	116
No.2	64	39	36	43	40	36	25	39	37	35	41	42
No.3	73	49	76	68	66	72	73	60	64	63	58	59

Fig.3 Direct Vibration Value of Each Bearing(Yabs)

From the analysis of shaft centerline trend plot measured at No.1 bearing before this outage. The shaft centerline moved to the right side in No.1 bearing by 0.15mm and vibration was increased followed on this movement, as the power load was increased over 40MW.

3. Measurement and Trouble Analysis

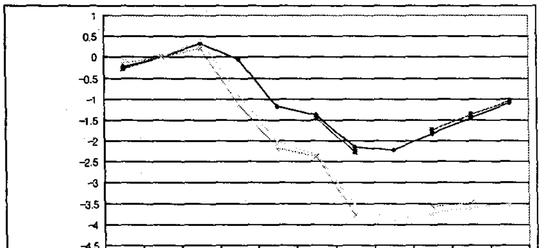
Several kinds of measurements for troubleshooting were carried out as follows.

- Measurement of the vertical level changes at horizontal joints by laser equipment for foundation deformation
- Measurement of the vertical level of the rotor center at Nos. 1, 2 & 3 bearing oil bore by laser equipment
- Measurement of bearing reaction by load cell Coupling alignment
- Measurement of rotor position at bearing oil bores
- Alignment of casings & diaphragms by laser equipment
- Measurement of radial clearance between rotating & stationary parts

3.1 Vertical level changes at horizontal joints

Some turbine foundation and supporting system could have been deformed or subsided gradually by turbine self-weight and thermal stress especially for several years after construction depending on operation condition of the unit. No.1 Bearing is on the front standard supported by the flexible support. According to design criteria, turbine loads (29,750 Kg) are divided into 1,341 Kg at No.1 bearing, 9,954 Kg at No.2 and 18,455 Kg at No.3 under coupled condition.

Fig. 4 represents the vertical level changes between 1998 and 2001 outages at horizontal joints of front & middle standards, Nos. 1, 2 and 3 bearings, High, Intermediate and Low Pressure Turbines. If the vertical level at No.1 bearing



	Front	No.1				No.2				No.3	
1998 Left Side	-0.23	0	0.33	-0.04	-1.17	-1.37	-2.14	-2.21	-1.82	-1.45	-1.09
1998 Right Side	-0.28	0				-1.46	-2.26		-1.73	-1.37	-1.03
2000 Left Side	-0.07	0	0.42	-0.88	-2.01	-2.32	-3.54	-3.91	-3.72	-3.59	-3.5
2000 Right Side	-0.13	0	0.22	-1.12	-2.17	-2.36	-3.77		-3.57	-3.48	

Fig. 4 Flange Level of Lower Casing

was unchanged, the level was changed down 0.93 mm at No.2 bearing and 2.13 mm at No.3 bearing respectively for two years.

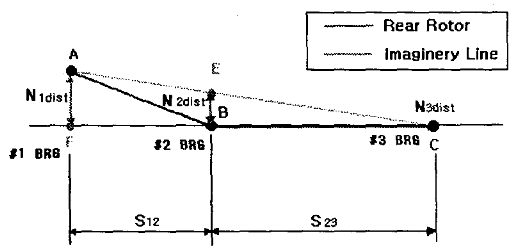
Three kinds of alignment method were carried out in this outage for bearing and shaft alignment.

3.2 Rotor Position check with bearing and without be

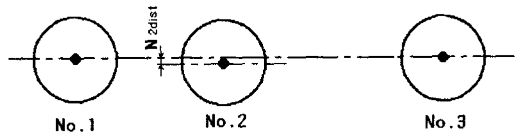
A first rotor position check was made and recorded at the oil deflector fit No.1 bearing under the rotor sitting in its lower bearing halves, and a second rotor position check was made under the lower half of No.1 bearing removed and rotor properly supported on a sling. For this check the rotor must not be moved from the position taken the first rotor position check. However the rotor was moved 0.22 mm from right to left direction.

3.3 Geometrical Alignment by Laser

Fig. 5 represents the design bearing elevation curve for steam turbine only offered by manufacture.



(a) Geometry View



(b) Circular View

Fig. 5 Schematic Diagram of Bearing Center Level

Table. 2 Dimension Value of Turbine System(Design)

Scalar Dimension	Geometry Dimension	Value(mm)
N_{1dist}	$\overline{AF}$	0.86
N_{2dist}	$\overline{EB}$	0.53
N_{3dist}	$\overline{CC}$	0
S_{12}	$\overline{AB}$	3195
S_{23}	$\overline{BC}$	5210

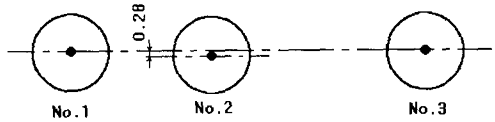


Fig. 6 Bearing Center Level in Disassembly

No.1 bearing can be set 0.86 mm above the projected laser line(BC), or No.2 bearing can be set 0.53 mm below the projected laser line(AC) to meet the design value.

The related dimensions for design criteria are shown in Table. 2.

Fig. 6 represents the vertical position of No.2 bearing bore with respect to the laser centerline(AC) projected from the rotor center at No.1 bearing oil bore to No.3 bearing oil bore.

As shown in Fig.6, N2dist was measured 0.28 mm when the laser centerline projected from No.1 bearing to No.3 bearing.

This kind of result might come from the deformation of No.1 or No.3 bearing, or from the rising of No.2 bearing.

However it might be almost impossible for No.2 bearing to rise, and also it could be unreasonable for No.1 bearing to deform, if the considerable low load of No.1 bearing compare to other bearings and the structure of the turbine system are taken into consideration. Therefore it is very reasonable that No. 3 bearing must have been down.

Table. 3 Dimension Value of Turbine System(1st Measuring)

Scalar Dimension	Geometry Dimension	Value(㎜)
N_{1dist}	$\overline{AF}$	0.46
N_{2dist}	$\overline{EB}$	0.28
N_{3dist}	$\overline{CC}$	0

Table. 4 Results of No.3 Bearing Increase(Upward 0.65㎜)

Scalar Dimension	Geometry Dimension	Value(㎜)
N_{1dist}	$\overline{AF}$	0.85
N_{2dist}	$\overline{EB}$	0.53
Movement	$\overline{CC}$	0.65 ↑

Table.5 Specification of Load Cell

Bearing	Scale of measuring	Name of product, Maker
No.1	3 ton	Hydraulic Scale, KEIKI
No.2,3	2~60 ton	CastonⅢ, CAS

According to maintenance history for this unit, shims of the generator feet have been removed to adjust the coupling misalignment which might have come from No. 3 bearing down. Basically No.3 bearing should be up to solve the problem fundamentally.

Following equations (1) and (2) indicates changes of relative elevation at No.1 and No.2 bearing when No.3 bearing is adjusted.

$$N1dist : (S12 + S23) = N2dist : S23 \quad (1)$$
$$N1dist : S12 = N3dist : S23 \quad (2)$$

From the above equation we obtain 0.27 mm which is No.3 bearing movement to reach design elevation.

3.4 Measurement of bearing reaction by load cell

During initial installation of the turbine, the shaft reaction should be measured for possible future reference. Oil whipping may occur when bearing loads are reduced because of elevation differences between adjacent bearings, particularly between the turbine and the generator. Design value of bearing reaction is shown in Table 6.

The shaft reaction or weight can be measured by suspending a load cell from the crane hook so that it will indicate the weight of the shaft when the support provided by the lower half of the bearing is removed.

After dial indicators are installed on the top and bottom

Table.6 Design Value of Bearing Load

구 분	No.1	No.2	No.3
Design(Kg)	1,341	9,954	18,445

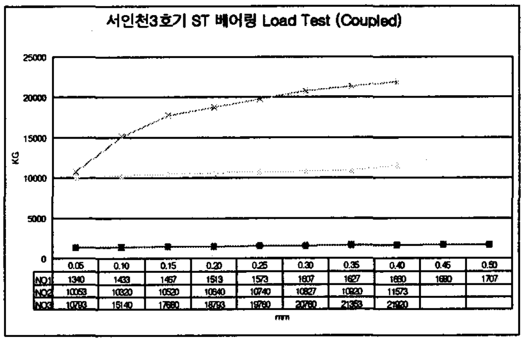


Fig 7 Coupled Bearing Load Data

Table 7 Load linear equation

Bearing	Equation
NO1	771 x + 1348
NO2	3490 x + 9913
NO3	17074 x + 15348

of the journal, raise the shaft gradually with the chain fall recording the load cell readings every 0.05mm of shaft movement. The shaft should be raised until the lack of clearance in adjacent parts prevent further movement, and then lower the shaft gradually to the initial position, recording the load cell readings every 0.05mm of shaft movement.

Fig. 7 represents the graph which plotted load cell readings versus shaft position from the measured data after inserting shims in generator feet.

The point at which the plotted line intercepts the initial rotor position level line is the bearing reaction and should correspond to the factory stated shaft load of 18,445 kg at No.3 bearing. The average slope of the curve is the stiffness gradient which indicates how much the bearing loading is changed for a given change in bearing elevation.

It is possible to make linear the above graph as shown in Table 7.The bearing reaction at this state is when x is zero at each linear equation.

Comparing to the design value shown in Table 6, the load at No.3 bearing is lower by 3097 Kg than design load, but on the other hand there is a little bit difference at No.1 and No.2. No.3 bearing should be up by 0.22mm to correspond to the design load.

3.5 Coupling Alignment

This section is intended for determining the bearing movement calculating from the difference between design and measured value in parallel(rim) and angular(face) at the turbine-generator coupling.

According to the bearing elevation and catenary curve, faces are open 0.08 mm at top and the generator half of the coupling is 0.38 mm below the turbine half. As shown in Fig. 8 bottom, the coupling alignment measurement was carried out, its value found that rim is 0.10 mm, face is 0.08 mm.

This kind of result might come from the deformation of No.3 bearing, or from the rising of generator. However it might be almost impossible for the generator to rise, therefore it is very reasonable that No. 3 bearing must have been down.

Most of rotating machines without rotating system with gearboxes have design criteria whose rim is zero, face is zero.

Horizontal design criteria and as found measured values are shown in Fig. 8. Appropriate bearing or generator feet should be adjusted to correspond to the horizontal design criteria by the typical method. At this time, radial clearance between rotating and stationary parts should be reviewed simultaneously before any movement.

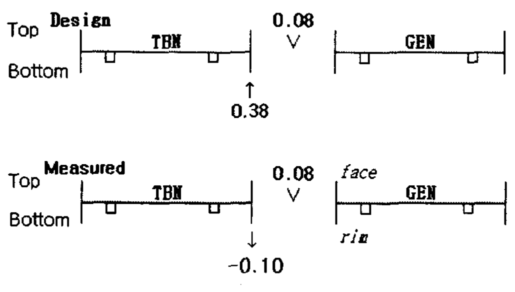


Fig.8 TBN-GEN Alignment Data(Top-Bottom)

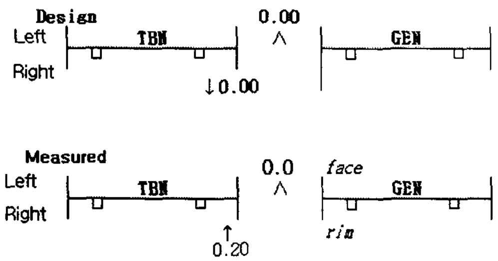


Fig.9 TBN-GEN Alignment Data(Left-Right)

Table 8 Results of Movement Calculation and Expected Alignment Value

Brg No.	Movement		Expected Alignment Value			
			Vertical		Horizontal	
	Vertical	Horizontal	Face	Rim	Face	Rim
No.1	0		0.08 mm	0.37 mm	0 mm	0 mm
No.2	0					
No.3	↑0.30					
No.4	0	0.2				
No.5	↑0.30	0.2				

Results of movement calculation for corresponding to the horizontal & vertical design criteria are shown in Table 8.

3.6 Maintenance Action Summary

- Raised low pressure turbine casing rear by 0.5mm and lowered No.3 bearing by 0.30mm. Consequently the absolute movement of No.3 bearing is 0.20mm upward.
- Moved high pressure turbine casing front and rear to the right side by 0.50mm and 0.60mm respectively.
- Moved No.1 bearing to the right side by 0.22mm.
- Suitable shims were proportionately placed under the generator feet to be raised 0.30mm at No.5 bearing position, and moved generator front and rear to the right side 0.20mm each

4. Result of Operation

The vibration and operating data obtained during startup and normal operation are usually used for verifying whether the trouble analysis and the maintenance action taken before operation are correct or not. Fig.10 & 11 represent absolute vibration(X-abs, Y-abs) at No.1 through No.5 bearing when the unit is unloaded and 75MW load condition.

Comparing to the absolute vibration data before and after outage shown in Fig.2 & 3 and Fig 10 & 11, the max vibration 141㎛pp at No.1 bearing was remarkably

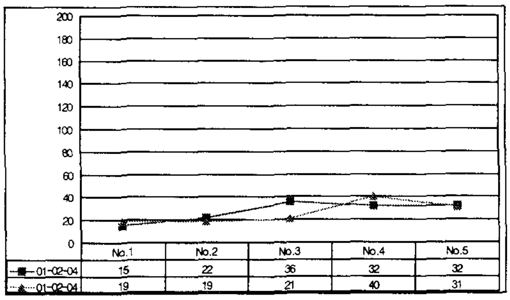


Fig.10 Absolute Vibration(Xabs) at Bearings after maintenance

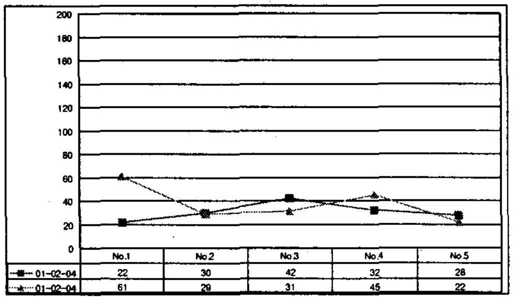


Fig.11 Absolute Vibration(Yabs) at Bearings after maintenance

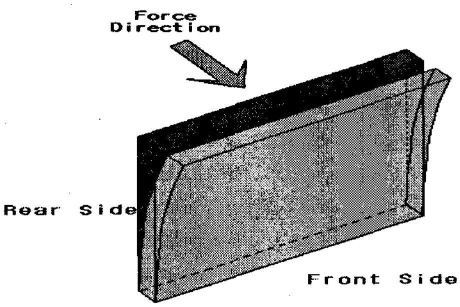


Fig.12 Axial Deformation of Front Standard

improved to 39 ㎛pp.

5. Measurement of strain by axial expansion u gage at front standard

In this section the stress change during startup on the flexible plate below the front standard is measured and analyzed for examining whether the axial expansion is unbalance or not, because the unbalanced axial expansion of the front standard could be the cause of the high vibration at No.1 bearing.

If the turbine casing is expanded by thermal growth, the flexible plate is bent as shown in Fig.12.

Ten strain gages were installed on one of the three plates. As shown in Fig.13, four strain gages(①②③④) were vertically and horizontally installed on the front of the plate, four(⑤⑥⑦⑧) were on the rear and two(⑨⑩) were vertically on the side.

5.1 Vertical strain of the flexible plate by casing exp

Fig.14 represents the vertical strain trend from two front strain gages(②④) and two rear gages(⑥⑧). Any abnormal symptom was not found in vertical strain from the fact that front gages loaded in bending compression, rear ones in bending tension and their similar increasing & decreasing rate.

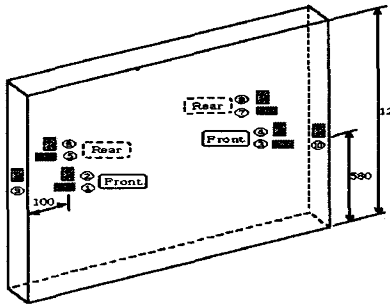


Fig.13 Attached Point of Strain Gauge

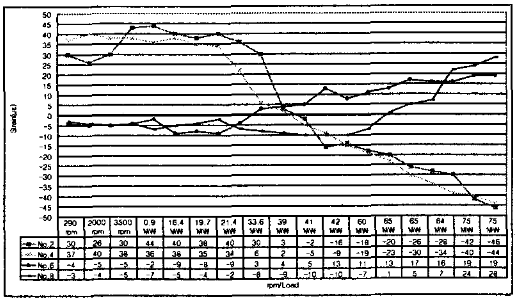


Fig.14 No.2,4,6,8 Strain Transformation

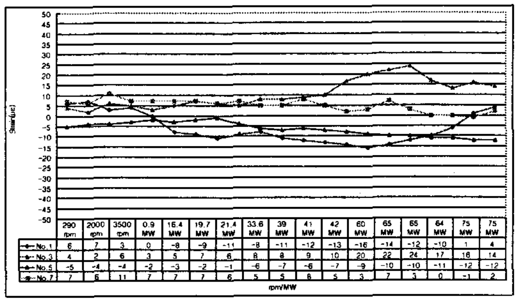


Fig.15 No.1,3,5,7 Strain Transformation

5.2 Horizontal strain of the flexible plate by casing expansion

As shown in Fig.15, on the whole there was no big difference in horizontal strain. During unit load changed from 28MW to 33.6MW, strain value at ⑩ was suddenly dropped from 12 to 27 with plumps sound. It is highly recommended to inspect sliding keys, pipes and etc in the next outage, because this result could come from sudden removal of restriction at sliding keys or pipes.

However it has no influence upon unit operation and vibration after the sudden drop of strain.

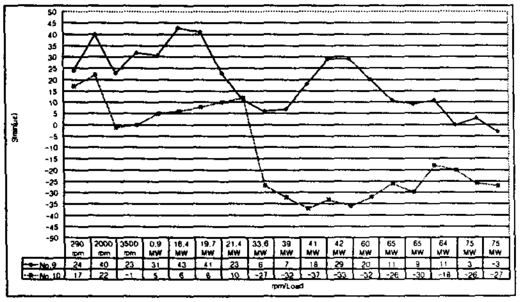


Fig.17 No.9,10 Strain Transformation

6. Conclusion

The bearing reactions are highly dependent upon the vertical bearing alignment. If improper vertical alignment is made, then the bearings may be either excessively loaded, or could become completely unloaded, which could lead to self-excited whirl instability. To minimize shaft stress and vibration, the bearings should be arranged along a catenary curve.

This paper shows that the high vibration problem at No.1 bearing has been solved by all-around troubles-hooting and related optimum maintenance action as follow.

- Measured vertical level changes at horizontal joints and verified intermediate-low pressure turbine 2.13mm down
- Measured rotor position with and without bearing, and moved 0.22mm to the right.
- Carried out casing and diaphragm alignment by laser.
- Carried out geometrical bearing alignment by laser and verified that No.3 bearing should be up 0.27mm.
- Measured bearing reaction by load cell under coupled condition and verified that No.3 bearing should be up 0.22mm.
- Carried out typical coupling alignment and calculated movement
- Measured strain on the flexible plate during startup using strain gage and verified normal axial expansion without a second sudden drop of side strain

The max vibration was 141μm peak to peak at No.1 bearing before outage was remarkably improved to 39μm peak to peak, comparing to the absolute vibration data before and after outage.

References

(1) John Piotrowski "Shaft Alignment Handbook 2nd Ed." 1986.

(2) Hamar Laser Inc."Hamar Laser-711 User's Manual", 1994

(3) Ronald bradshaw, "The mechanics of shaft alignment",

(4) GE Power system, "Couplings & Shaft, Steam Turbine-Generator Maintenance Training, Vol.ⅦA", 1995

(5) James L. Cox & Leon G. Wilde, "Alignment of Turbomachinery, Sawyer's Turbomach -inery Maintenance Handbook Vol.Ⅲ", Turbomachinery International, 1980

(6) J.P. Holman, Experimental Methods for Engineers, 6th Ed., 1994

(7) Albert W. Forrest, Jr., and Richard F. Lavasky, "Shaft Alignment using Strain Gages", Marine Technology, pp. 276ì 284, 1981

(8) James W. Dally, Experimental Stress Analysis, 1991

(9) Measurement Group Tech Note TN-504-1, Strain Gage Thermal Output and Gage Factor Variation with Temperature, 1993

(10) I.C.Kim, S.B.Kim, J.W.Jung, and S.M.Kim, "Technical Report for Reducing vibration in Seo-Inchon #2 Steam Turbine", Korea Plant Service & Engineering, 2001