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TOPPING TURBINE (103-JAT) ROTOR INSTABILITY IN 1150-STPD KELLOGG AMMONIA PLANTS

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Agrico Chemical Company operates three nominal 1043 MeTPD (1150 STPD) Kellogg Ammonia Plants in the U.S.. In the Kellogg Ammonia Plant, the synthesis gas compressor train consists of two DeLaval steam turbines in tandem, one back pressure and one condensing, driving two Dresser-Clark barrel type centrifugal compressors. The back pressure or topping turbine operates at 10,343 kPa (1500 psig) inlet / 3792 kPa (550 psig) exhaust steam conditions while the condensing turbine operates at 3792 kPa (550 psig) inlet / 14 kPa (4" Hg) absolute exhaust steam conditions. The normal operating speed range is 10,400 - 10,700 rpm with a combined driver output of 20,507 kW (27,500 hp).

In two of Agrico's three plants, instabilities in the rotor / bearing system have been an ongoing problem. On occasion plant rates, i.e. machine speed, have been restricted in order to limit the exhaust end shaft relative vibration on the 103-JAT to a maximum value of 89 μm (0.0035 in) peak to peak.

The purpose of this paper is to acquaint the reader with Agrico's experiences with exhaust end vibration and rotor instabilities on the 103-JAT topping turbine. The final conclusions arrived at in this paper were based on: 1) field acquired data both during steady state and transient conditions, 2) computer modeling of the rotor / bearing system, and 3) vibration data taken from a "control rotor" during a series of test runs in a high speed balancing machine from 0 to 110% of operating speed.

INTRODUCTION

A cross-sectional view of the 103-JAT is shown in Figure 1. The two stage rotor weighs 3585 N (806 lb_f) and consists of bucketed wheels that are integral to the shaft. The steam end bearing journal diameter is 126.75 mm (4.990 in) while the exhaust end bearing journal has a diameter of 101.45 mm (3.994 in). The bearing span and shaft overall lengths are 137.16 cm (54.5 in) and 196.22 cm (77.25 in) respectively.

The turbine is coupled at both ends. The steam end is coupled to the condensing turbine via a Koppers size 2-½ / 3 class ACCS / RM gear coupling. The half weight of the coupling assembly mounted on the steam end of the 103-JAT is 115.65 N (26 lb_f). At the exhaust end, the 103-JAT is coupled to the low pressure compressor through a Koppers size 4 class ACCS / RM gear coupling with a coupling half weight of 341.16 N (925 lb_f).

VIBRATION HISTORY

During the period from 1974 through 1979, exhaust end journal bearing inspections usually showed wiped pads in the lower half of the bearing with some evidence of pounding of the babbitt material noted. In addition, it was not uncommon to see evidence of damage to the babbitted bearing oil seal as well. A general opinion during this time was that the pads were being damaged during shutdown of the topping turbine. However, data taken during a shutdown and subsequent startup of the 103-JAT in January, 1979 showed that in reality the pads were being damaged during the constant acceleration of the syn gas machine from slow roll to minimum governor rpm.

Consequently, in early 1980 a modified startup procedure was implemented. The new startup procedure called for a step acceleration of the 103-JAT. The 103-JAT would be accelerated from: slow roll to 2000 rpm, 2000 to 4000 rpm, 4000 to 6000 rpm, and finally 6000 to 8000 rpm. After each 2000 rpm increment a mandatory five minute minimum hold was utilized in order to allow the bearing pad temperatures to stabilize. A maximum allowable pad temperature of 120°C (250°F) was established with a mandatory shutdown required at 130°C (265°F). By bringing the turbine up to minimum governor in this manner, it was hoped that the effects of temperature and alignment changes from slow roll to operating speed would be minimized. This procedure has successfully been utilized in all three of Agrico's Kellogg Ammonia Plants since 1980. During startup with the modified procedure, the maximum pad temperatures are approximately 93°C (200°F) with the shaft experiencing a maximum excursion of 50 to 76 μm (0.002 to 0.003 in) outside of the bearing set clearance.

Typical exhaust end vertical and horizontal[1] spectra from the shaft proximity probes are shown in Figures 3 and 4. Strong ½ x and 1 x components, 24.4 μm (0.00096 in) and 63.5 μm (0.0025 in) respectively, are readily apparent with other minor multiples of ½ x rpm also noted. During the period from 1980 to the present, some important points have been noted:

1. The 1 x component begins to rapidly climb at speeds in excess of 10,200 rpm.
2. The ½ x component tends to reduce in magnitude as more load is shifted to the 103-JAT from the condensing turbine.
3. The ½ x component tends to increase in magnitude with increasing bearing clearance.
4. If the bearing clearance exceeds 0.203 mm (0.008 in), load shifting from the condensing turbine to the 103-JAT has little or no effect. However, removing load from the 103-JAT <u>always</u> made the ½ x component increase in magnitude.
5. Within reasonable limits, imposed hot misalignment of the 103-JAT decreased the ½ x component's magnitude.
6. The apparent translational balance resonance speed range is from approximately 5200 to 8000 rpm with the resonance peak at approximately 6800 rpm (figures 4 and 6).

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[1]Probe orientation on the 103-JAT is as follows: Facing the turbine from the steam end, with 0° coincidental with the positive x-axis on a Cartesian Coordinate System, and all angles measured in a CCW direction from 0°, vertical = 135°, horizontal = 45°.

7. The ends of the rotor are in phase (305° and 277° respectively) at the resonance amplitude at 6800 rpm (figures 5 and 7).

8. Between 8200 and 9200 rpm, the exhaust end suddenly goes out of phase with the steam end (figure 7).

9. Between 10,200 and 10,300 rpm the 1 x amplitude increases from 10.16 μm (0.0004 in) to 38.1 μm (0.0015 in) at the steam end and from 2.54 μm (0.0001 in) to 25.40 μm (0.001 in) at the exhaust end (figures 5 and 6). Also within this speed range, the ½ x component suddenly appears at approximately 12.70 μm (0.0005 in) at both ends of the rotor.

10. From 10,300 rpm to 10,560 rpm the exhaust end vibration increases to 23.88 μm (0.00094 in) at ½ x and 66.80 μm (0.00263 in) at 1 x (figure 4).

ROTOR DYNAMICS ANALYSIS

The rotor / bearing system of the topping turbine was mathematically modeled as a series of lumped mass stations connected by elastic elements. Each mass station contains the mass and inertia properties for the discrete rotor segment that it represents. Included in the model, along with the mass station information, are the bearing support locations and the speed dependent properties of the oil film.[1]

Figure 8 is a plot of the first three undamped critical speeds of the 103-JAT versus a nominal range of bearing stiffness values. Also plotted on this critical speed map are the bearing stiffness versus speed curves for the steam and exhaust ends. One can readily see that three out of the four principal stiffness curves intersect the third mode at the rotor speed of 10,500 rpm. Thus within the normal operating speed range, excitation of the third critical speed should be expected, especially in the horizontal plane. Tables 1 and 2 are a brief summarization of the data presented in figure 8.

So as not to readily excite the third mode within the normal operating speed range, the design of the bearings was modified. The main parameters of the redesign were:

1. The bearing stiffness would not be coincidental with the third mode at operating speed.

2. The bearing stiffness would be more nearly uniform between the steam and exhaust end bearings.

3. The bearing damping characteristics would be improved at operating speed.

The redesigned bearings were fabricated by Centritech Corporation and are designated internally by Agrico as the MOD-2 design. The MOD-2 bearing differs from the DeLaval standard bearings as listed in the following.

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[1]The computer modeling of the rotor was performed by Centritech Corporation of Houston, Texas.

STEAM END

	MOD-2	DeLaval
Pad Length	3.81 cm (1.50 in)	6.03 cm (2.373 in)
L/D	0.30	0.48
Preload	0.33	0.00
Orientation	Load Between Pads	Load On Pad
Adjustable Bearing Clearance	Yes	No
External Pressure Pad	Yes	No

EXHAUST END

	MOD-2	DeLaval
Pad Length	3.81 cm (1.50 in)	4.13 cm (1.625 in)
L/D	0.38	0.41
Preload	0.25	0.00
Orientation	Load Between Pad	Load On Pad
Adjustable Bearing Clearance	Yes	No
External Pressure Pad	Yes	No

The external pressure pads in the top half of both of the bearings were added in order to provide a secure fit between the bearings and turbine case during all phases of operation and thus reduce the possibility of generating a ½ x vibration component.

The stiffness and damping coefficients for the MOD-2 bearing are summarized in Table 3 and 4 and depicted on the critical speed map as shown in figure 9. The principal stiffness curves all intersect the third mode in the 11,700 to 12,000 rpm range thus reducing the possibility of exciting this mode within the normal operating speed range of the turbine. Figures 10 through 12 depict the anticipated rotor mode shapes assuming an average support stiffness of 5.25×10^5 N/cm (3.00×10^5 lb_f/in).

CONTROL ROTOR TEST

A prototype version of the MOD-2 bearing was tested at DeLaval's repair shop in Houston, Texas. In order to duplicate as nearly as possible the actual rotor configuration, component balance coupling hubs were mounted on the control rotor. The control rotor was first run in the DeLaval bearings to establish a baseline (figure 13). The DeLaval bearings were replaced with the MOD-2 bearings and the control rotor was run for a second time (figure 14). Figures 15 and 16 compare the DeLaval and MOD-2 bearings on the same plot.

FIELD STARTUP

The MOD-2 bearings were installed in the 103-JAT topping turbine and field tested during the startup of the Ammonia Plant on May 15, 1985. Figures 17 and 18 are waterfall plots of the steam end vertical and horizontal probes. Figures 19 and 20 are waterfall plots of the exhaust end vertical and horizontal probes. In the 6000-6500 rpm range the additional harmonics in figure 18 and the frequency shift in figure 20 were caused by clipping of the input signal during the tape recording process.

The first observed critical speed occurred between 6200 and 6630 as summarized below:

STEAM END

Location	RPM	1 x (peak to peak)
61V (vertical)	6628	85.85 μm (0.00338 in)
63H (horizontal)	6628	95.25 μm (0.00375 in)

EXHAUST END

Location	RPM	1 x (peak to peak)
64V (vertical)	6224	73.15 μm (0.00288 in)
66H (horizontal)	6224	103.19 μm (0.00406 in)

A comparison of steady state data between the DeLaval and the MOD-2 bearings is listed below:

Location	DeLaval			MOD-2		
	rpm	$\frac{1}{2}$ x	1 x	rpm	$\frac{1}{2}$ x	1 x
61V	10,560	10.16 μm (0.00040 in)	15.24 μm (0.00060 in)	10,600	---	35.56 μm (0.0014 in)
63H	10,560	13.72 μm (0.00054 in)	32.51 μm (0.00128 in)	10,600	---	50.80 μm (0.0020 in)
64V	10,560	25.15 μm (0.00099 in)	59.18 μm (0.00233 in)	10,600	---	20.32 μm (0.0008 in)
66H	10,560	23.88 μm (0.00094 in)	66.80 μm (0.00263 in)	10,600	---	35.56 μm (0.0014 in)

CONCLUSIONS

The rotor dynamics study showed that the normal operating speed range was coincidental with the third undamped critical speed. Analysis of existing startup data supported the theory that the normal operating speed range falls within the domain of a resonance response region. In addition, analysis of startup and steady state data strongly suggested that the $\frac{1}{2}$ x mechanism was primarily due to loose bearings in the turbine case.

The MOD-2 bearings were offered as a possible solution to these two instability mechanisms. With the MOD-2 bearings in operation, the $\frac{1}{2}$ x instability is no longer present at either end of the turbine. At the same time, the overall exhaust end vibration levels have been reduced, on the average, by 68%, while the overall steam end vibration levels have remained approximately the same.

TABLE 1

ORIGINAL FIVE SHOE
TILT PAD BEARING COEFFICIENTS

STEAM END

	SPEED (RPM)				
STIFFNESS (lb_f/in):	3719	6203	9869	10,500	12,374
K_{XX}	3.79×10^6	4.81×10^6	5.65×10^6	5.71×10^6	5.96×10^6
K_{YY}	3.57×10^5	2.78×10^5	2.18×10^5	2.11×10^5	1.93×10^5
DAMPING (lb_f-sec/in):					
C_{XX}	4.09×10^2	3.87×10^2	3.69×10^2	3.68×10^2	3.62×10^2
C_{YY}	1.07×10^3	6.98×10^2	5.11×10^2	5.01×10^2	4.55×10^2

EXHAUST END

	SPEED (RPM)				
STIFFNESS (lb_f/in):	3246	6378	8513	10,500	11,112
K_{XX}	2.81×10^6	4.34×10^6	5.11×10^6	5.68×10^6	5.85×10^6
K_{YY}	7.45×10^5	5.48×10^5	4.79×10^5	4.33×10^5	4.22×10^5
DAMPING (lb_f-sec/in):					
C_{XX}	2.51×10^2	2.34×10^2	2.26×10^2	2.21×10^2	2.19×10^2
C_{YY}	1.54×10^3	7.87×10^2	6.01×10^2	5.02×10^2	4.75×10^2

TABLE 2

UNDAMPED CRITICAL SPEEDS
EXISTING TURBINE ROTOR
ORIGINAL BEARINGS

	HORIZONTAL PLANE	VERTICAL PLANE
FIRST CRITICAL	BETWEEN 940 AND 1200 RPM	BETWEEN 4500 AND 5600 RPM
SECOND CRITICAL	BETWEEN 1600 AND 2100 RPM	BETWEEN 6200 AND 7900 RPM
THIRD CRITICAL	9800 RPM	BETWEEN 10,000 AND 11,000 RPM

ORIGINAL PAGE IS OF POOR QUALITY

TABLE 3

CENTRITECH MOD-2
TILT PAD BEARING COEFFICIENTS

STEAM END

	SPEED (RPM)				
STIFFNESS (lb_f/in):	3408	6289	7811	10,500	12,899
K_{XX}	2.87×10^5	3.88×10^5	4.12×10^5	4.39×10^5	4.91×10^5
K_{YY}	4.50×10^5	5.37×10^5	5.37×10^5	5.27×10^5	5.78×10^5
DAMPING (lb_f-sec/in):					
C_{XX}	8.35×10^2	7.66×10^2	7.06×10^2	6.27×10^2	5.70×10^2
C_{YY}	10.60×10^2	8.21×10^2	8.21×10^2	7.20×10^2	6.71×10^2

EXHAUST END

	SPEED (RPM)				
STIFFNESS (lb_f/in):	3353	5912	8928	10,500	11,349
K_{XX}	3.44×10^5	3.16×10^5	3.43×10^5	4.71×10^5	3.91×10^5
K_{YY}	6.14×10^5	5.10×10^5	4.91×10^5	6.13×10^5	5.24×10^5
DAMPING (lb_f-sec/in):					
C_{XX}	11.50×10^2	8.48×10^2	8.14×10^2	7.59×10^2	7.29×10^2
C_{YY}	16.70×10^2	10.80×10^2	9.95×10^2	8.85×10^2	8.25×10^2

TABLE 4

UNDAMPED CRITICAL SPEEDS
EXISTING TURBINE ROTOR
CENTRITECH MOD-2 BEARINGS

	HORIZONTAL PLANE	VERTICAL PLANE
FIRST CRITICAL	BETWEEN 4550 AND 4700 RPM	BETWEEN 5600 AND 5700 RPM
SECOND CRITICAL	BETWEEN 6700 AND 7600 RPM	BETWEEN 8200 AND 8400 RPM
THIRD CRITICAL	BETWEEN 11,700 AND 11,800 RPM	12,000 RPM

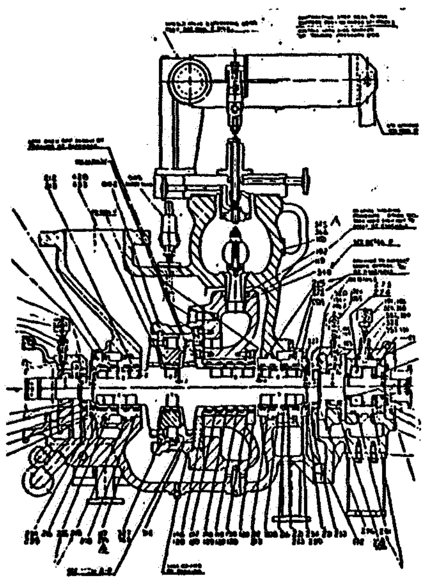


Figure 1

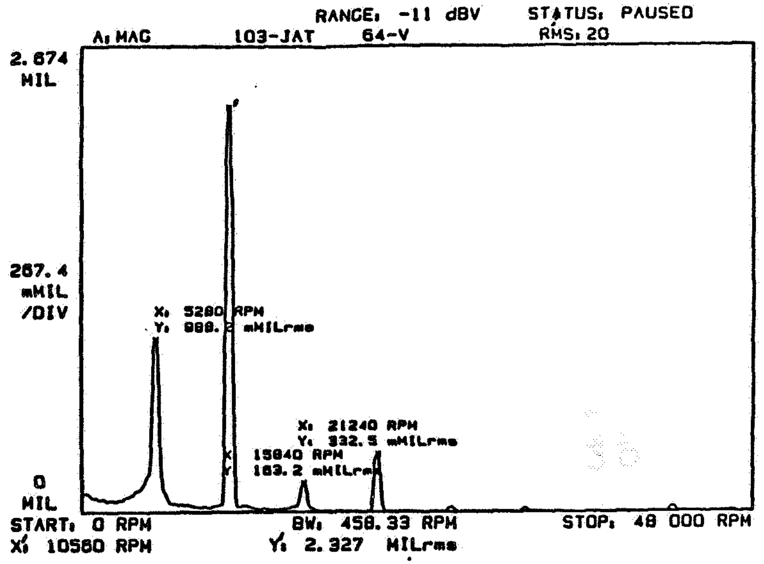


Figure 2

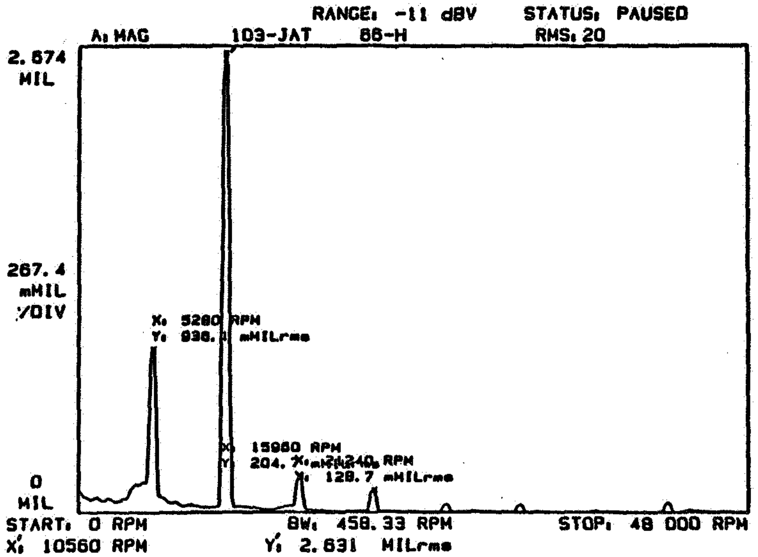


Figure 3

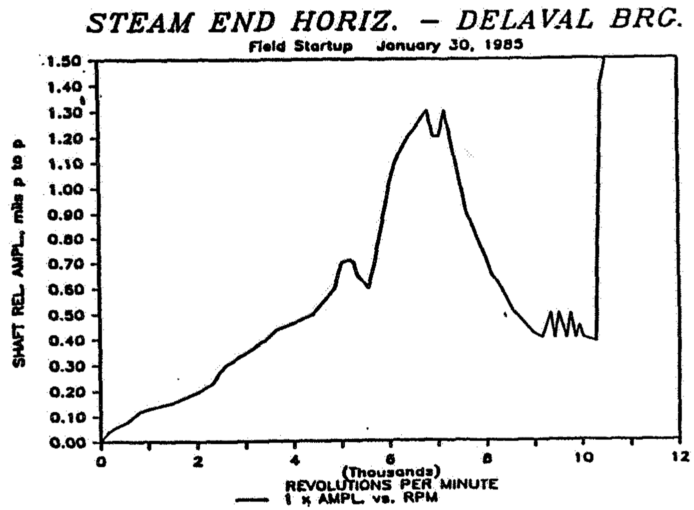
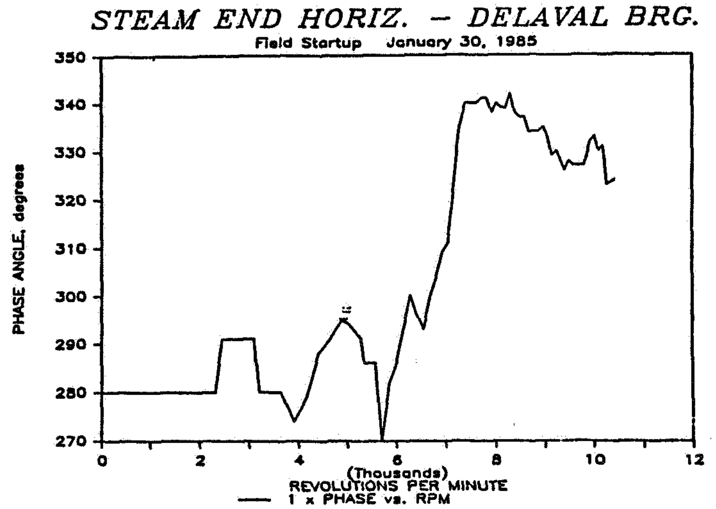


Figure 4



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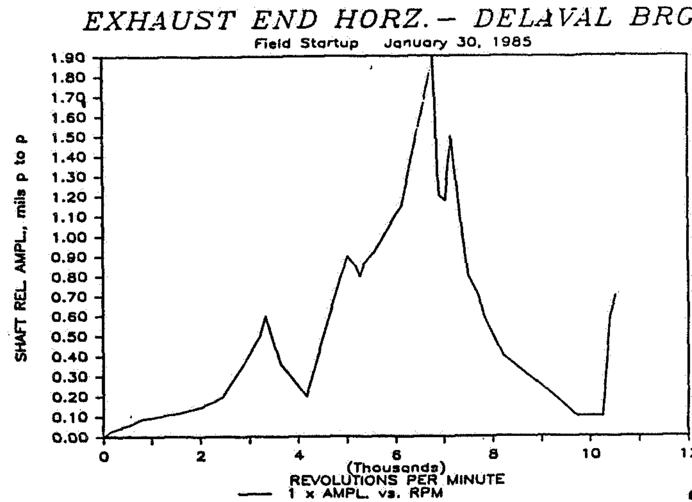

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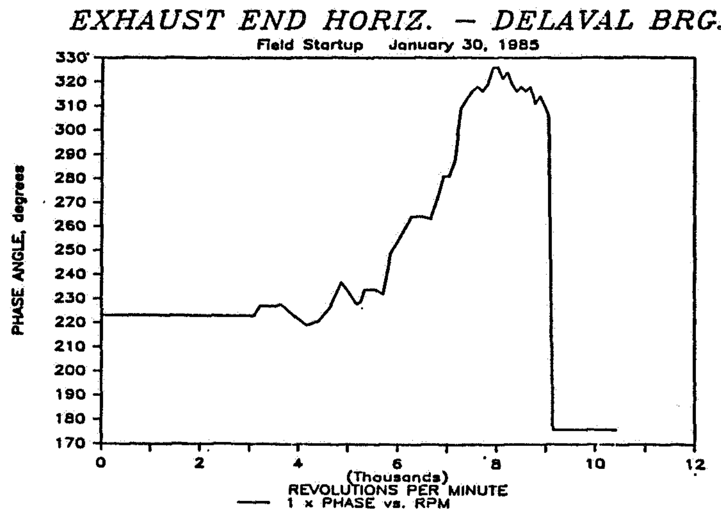


Figure 7

DE LAVAL 103 JAT STEAM TURBINE

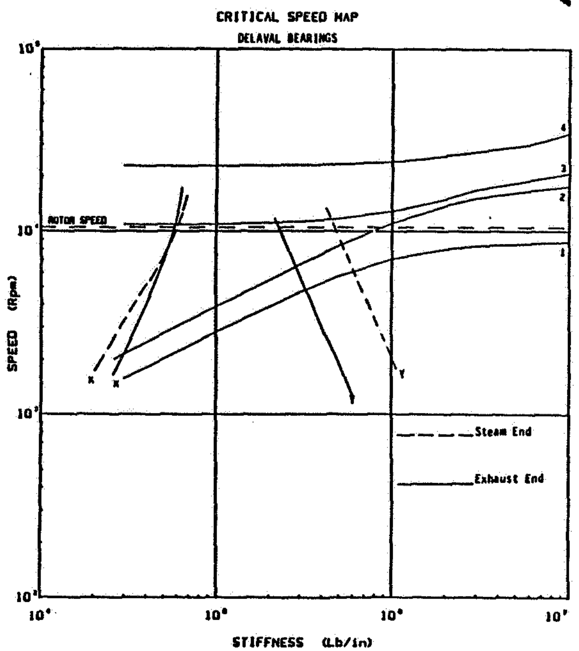
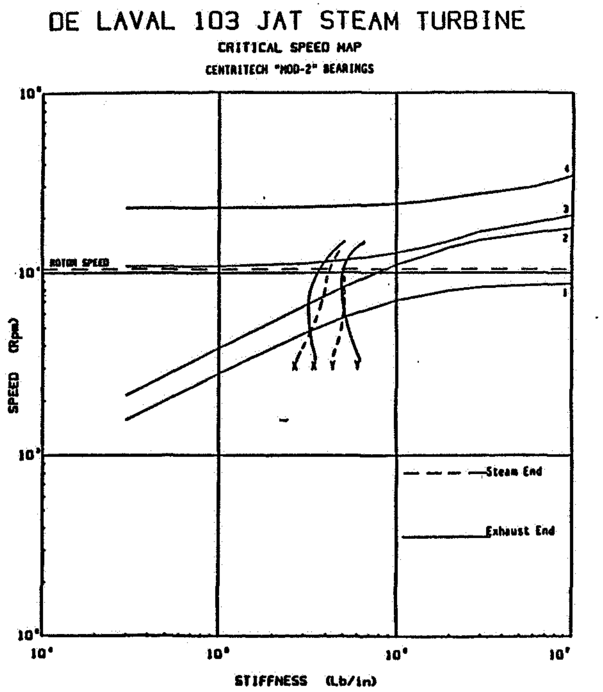


Figure 8



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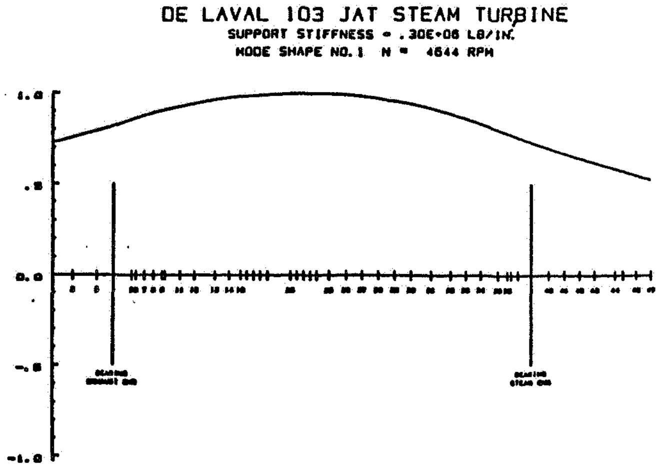
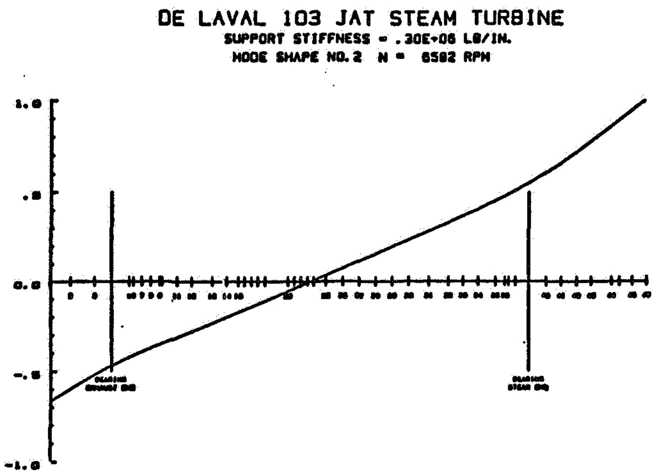


Figure 10



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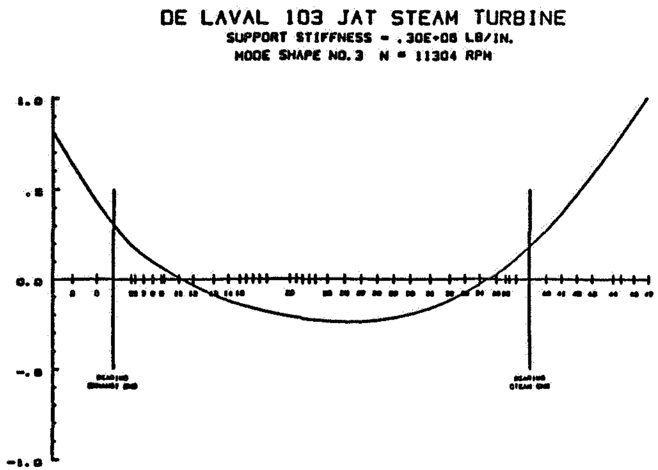

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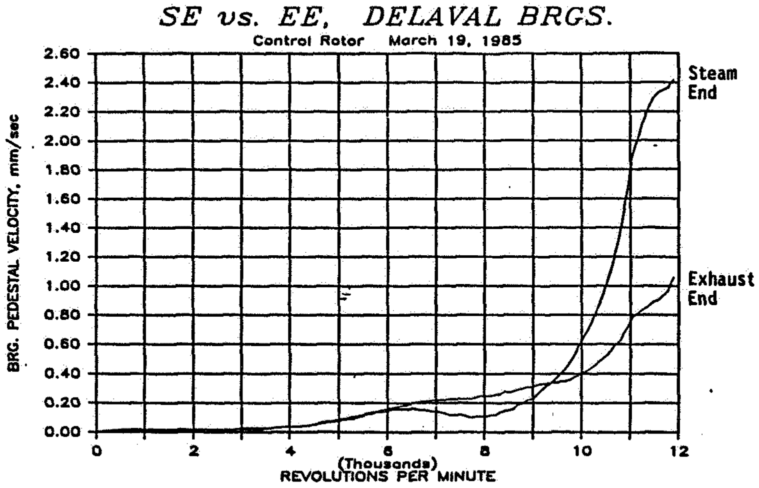
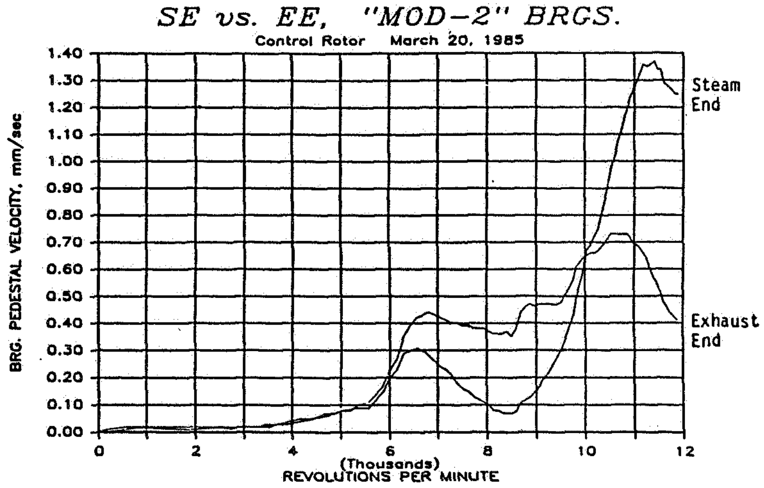


Figure 13



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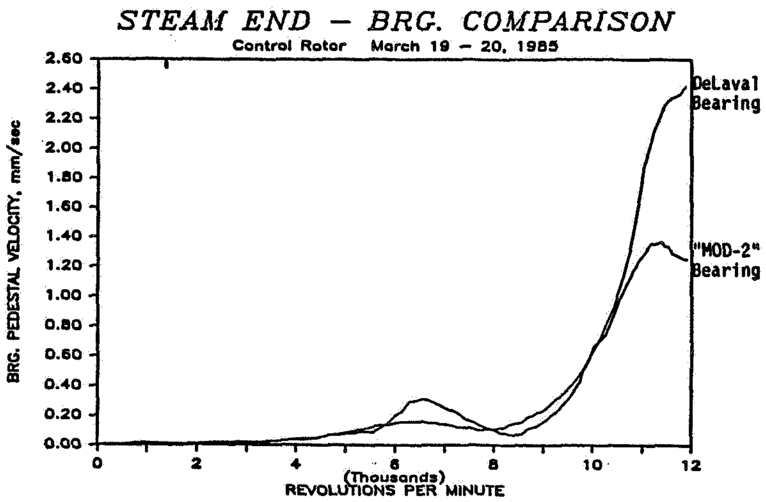

Figure 15

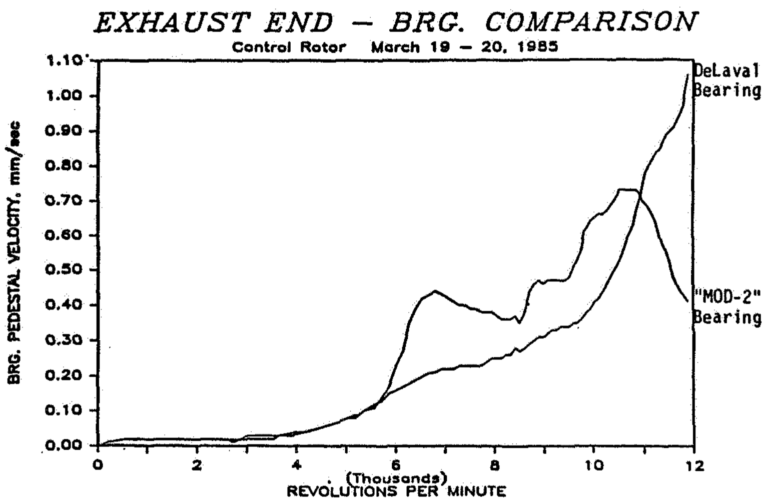

Figure 16