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Improved Design Tool for Thermal Synchronous Instability

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ABSTRACT

The investigation of synchronous instability has been active since the 1990's and the number of machines experiencing this type of instability continues to rise. The initial design threshold speed criteria was developed and compared to available case studies during 1998 - 2001. Further advances were made from 2002 – 2013 in the prediction of the bearing journal thermal hot spot, advanced theory to predict mid-span thermal imbalance, plus a new instability threshold criterion, that is useful for both low speed and high speed machines. However, the interest in shaft overhang instability is dominant from a practical view-point. The new improved design procedure is explained and the resulting advanced computer program that automates the process will be outlined. A review of three machines will illustrate the new capability that has been developed to streamline the computation of the stability map required for the design of rotating machines operating on tilting pad bearings.

Keywords: synchronous, thermal, instability, Morton Effect

INTRODUCTION

The analysis of critical path high pressure rotating machinery must include the influence of the bearings and gas labyrinth seals for both synchronous imbalance response and damped critical speed analysis. The damped critical speed analysis is a check for non-synchronous instability which can result in high sub-synchronous vibration by self-excited excitation arising from hydrodynamic bearings, gas labyrinth seals, rotor internal friction, and aerodynamic excitation. These sources of instability have been considered in the design of high speed turbines, compressors, and pumps since the early 1980's. An additional source of high vibration in machinery with fluid-film hydrodynamic bearings was realized in the mid 1990's by Paul Morton [1, 2] and by 1997 the oil and gas industry machinery experts were beginning to discuss the new source of un-steady synchronous vibration. Normally synchronous vibration was known to be the result of a mass imbalance which was usually corrected by low speed balance and if required high speed balance. Imbalance usually produced a steady vibration at a fixed speed, The occurrence of a spiral vibration produced by a mid-span metal to metal rub was known to occur in large turbines. It was also known that a loose turbine disk could result in a growth in synchronous vibration. The fluid-film bearing created hot spot in the bearing journal results in a thermal bow of the shaft outboard of the journal. This quickly became known as the Morton Effect but the prediction of when it would occur was not available to designers.

A field problem reported by Faulkner, Strong and Kirk [3, 4] in 1997 was the earliest US documented case of a machine that had both a loose turbine disk causing a high vibration and then a Morton Effect after the disk was redesigned to have a tight fit to the shaft at speed. This vibration would start increasing as the speed increased and drop out quickly when the speed reduced. The vibration is shown in Figure 1 where the speed is increased and decreased twice to produce the two synchronous increasing amplitude pulsating vibrations plotted versus time. It takes about 15-20 minutes to get the growth up to point where the speed is reduced in each case, the second higher speed condition being worse than the first lower speed condition. Figure 1 is an overlay of the horizontal and vertical probes at the turbine bearing location.

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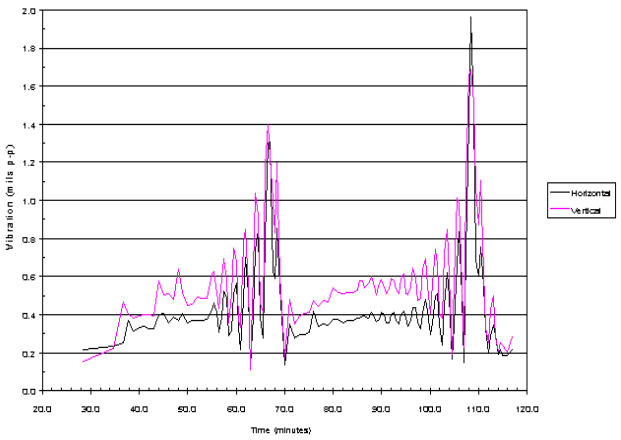

Figure 1 Morton Effect Vibration on the turbine end journal bearing [3, 4].

The conclusion from the bearing design study given in Table 1 was that by reducing the load capacity of the bearing, the operating eccentricity ratio would be 0.57. An amplitude of 76 micron (3 mil-pp) would be required to cause the thermal instability. The condition for instability was thought to be an orbit enclosing the bearing center, hence the column of max allowable amplitude in Table 1.

Table 1 Results of bearing design study [3, 4]

diametral clearance mm (mils)	pocket depth mm (mils)	eccen-tricity ratio (dim)	shaft center to bearing center mm(mils)	allowable amplitude, peak-to-peak mm(mils)
0.127 (5)	0.304 (12)	0.070	0.004 (0.175)	0.010 (0.4)
0.178 (7)	0.304 (12)	0.200	0.018 (0.70)	0.038 (1.5)
0.203 (8)	0.254 (10)	0.265	0.027 (1.06)	0.055 (2.2)
0.203 (8 mil)	0.406 (16)	0.220	0.022 (0.88)	0.051 (2.0)
0.203 (8)	0.304 (12)	0.250	0.025 (1.00)	0.055 (2.2)
0.229 (9)	0.304 (12)	0.300	0.034 (1.35)	0.071 (2.8)
0.127 (5)	0.304 (12) 12.7 x1.27 (0.5x.05in) undercut	0.570	0.036 (1.43)	0.076 (3.0)

Starting soon after this Morton Effect experience, the industry sponsored research group at Virginia Tech would start work on a project to produce a design tool to predict when a rotor shaft would have the Morton Effect. The desire of the sponsors was that only the standard vibration analysis computer program results were to be used to make the prediction.

The results of the first phase of the Morton Effect project produced a program consisting of both Fortran and Matlab code to make the prediction and plot the results. The process consisted of calculating the forced response of a trial weight placed at the overhang cg for speeds from low speed to above design speed. The response orbit amplitude and phase detail at the bearing of concern was part of the input for the Morton code. The overhang equivalent mass, the cg of the shaft from the centerline of the bearing to the shaft end, journal length and diameter, the average oil temperature, and oil viscosity were needed. The Morton program was written to compute the thermal bend and overhang thermal imbalance, then the combined mechanical and thermal imbalance. The total imbalance was plotted as a function of speed and the threshold value was 50 % above the API acceptable unbalance level which produced a force equal to 10% of the total rotor weight. The threshold was considered to be when the total imbalance had increased to a value 50% larger than the mechanical imbalance [5, 6].

The results were compared to the available field and test stand information in the literature [7 - 13]. The steady state forced response analysis was by the VT-FAST PC-based computer program [14]. The threshold condition selected produced predictions that agreed with the available test data. The program was a research tool and not acceptable as a design code for industry use due to the mix of program languages and the amount of data that needed to be collected per case.

The annual meeting discussion of machines with Morton Effect continued as a major high point of each meeting until the industry funding ended in 2012. In addition the program was converted to a Visual Basic pre and post processor which helped to build the necessary input and to plot the results of the program. The program was provided to the sponsors but the required data necessary for the analysis was still not easily produced since most companies have their own forced response analysis and most have different phase nomenclature that made it more difficult to correctly use the Morton analysis.

CURRENT DESIGN PROCESS

The current program for Morton Effect, VTMAP, is a very good program if it is used on a regular basis. Unfortunately the occurrence of Morton Effect is not a daily or even monthly event so the use of the program can be a challenge to master.

The process described in the Introduction for the current program is shown as a flow chart in Figure 2. The data file required for execution of the program is the output of a response program and a bearing analysis program plus various other parameters. The files are combined into a specific format required for the input file. For each variation of the parameters, all the programs must be re-run, data collected and put into the input format.

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NEW DESIGN PROCESS

The new design process is shown in the flow chart of Figure 3. The rotor model is prepared and the bearing model is specified for each bearing in the rotor system. The program Dyrobes [15, 16] is then executed and a screen for additional parameters is presented to the user.

When the additional information is specified, the program combines all the required input for TMAP [17], executes the program and allows the user to preview the resulting program output. The stability plot is available for viewing and the design process can continue by making changes to the rotor model and the bearing model as required.

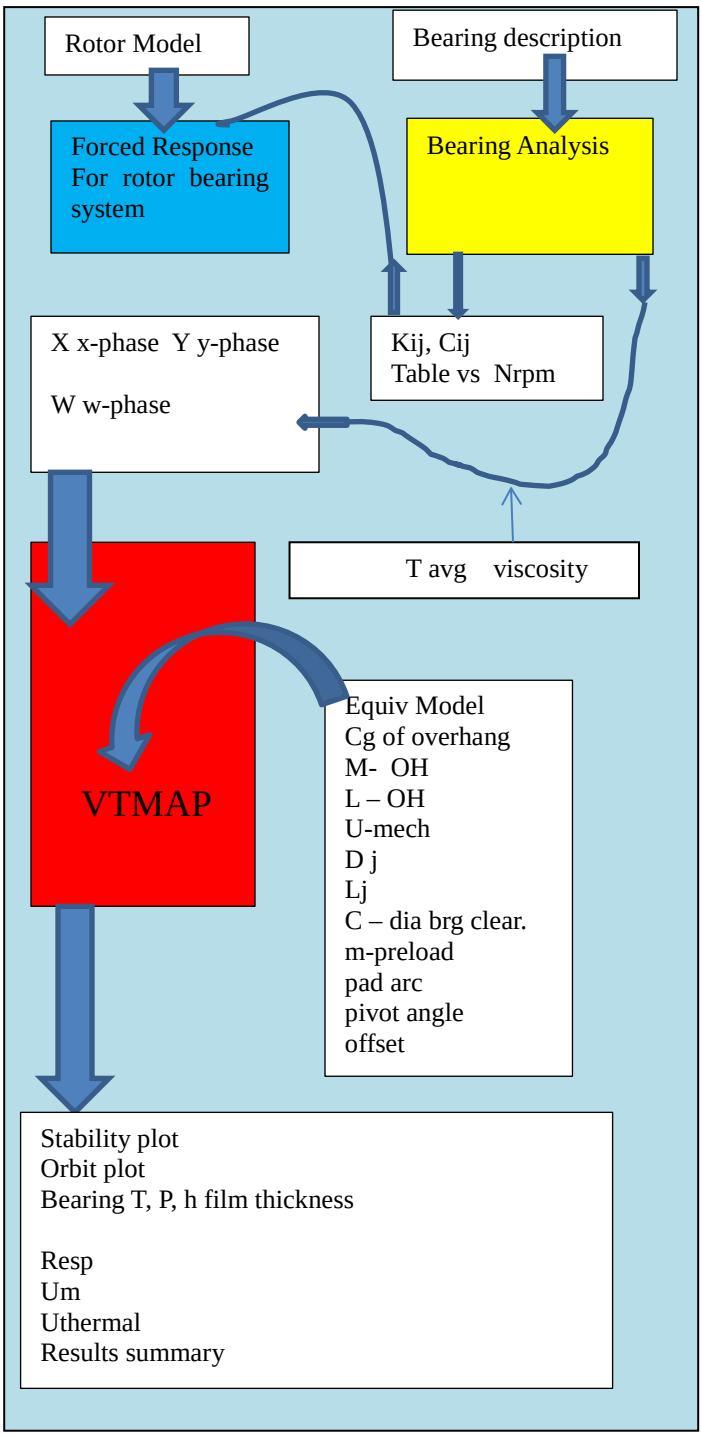

Figure 2 Flow chart of the current Morton Analysis

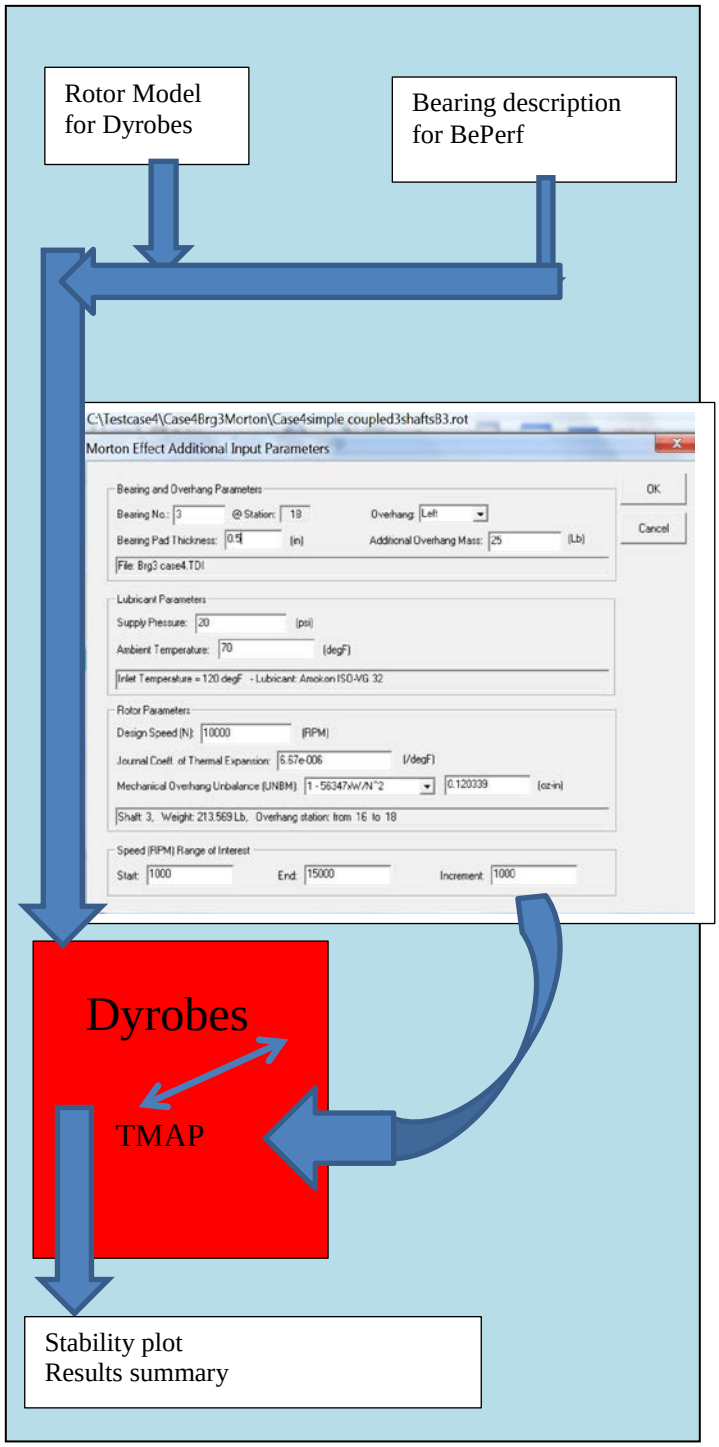

Figure 3 Flow chart of the new Morton Analysis

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When all design changes are complete, Dyrobes is run again and all the information is prepared by the program for the variation of the design. The data collection and formatting is all automatically prepared and TMAP is executed.

EXAMPLE 1

The new design tool will be utilized to evaluate a typical steam turbine for a Morton Effect potential. The first example rotor, Figure 4 is a large turbine model taken from the example folder of Dyrobes. A four pad tilting design was slected for this very heavy machine, 1.69E5 N (38E3 lb) total weight. The design speed is not given but a likely speed would be around 3600 rpm or maybe as high as 6000 rpm. A design speed of 5000 rpm was selected for this example stability check. The bearings shown in Figures 5 and 6 for loads of 8.9E4(20,000) and 7.56E4 N (17,000 lb) for bearing 1 and 2 respectively.

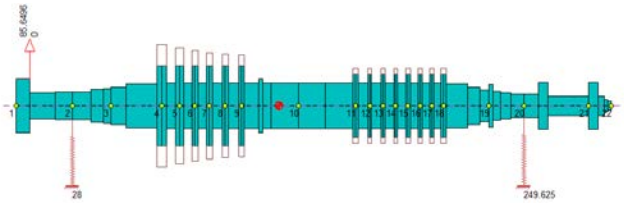

Figure 4 Turbine rotor for Example 1

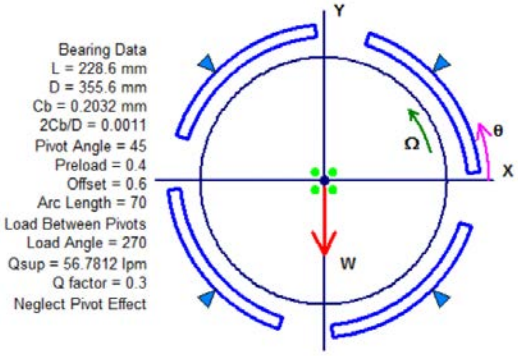

Figure 5 Example 1 Bearing 1 AT STATION 2

The bearings were added to the rotor model by adding a link to the bearing model files in the response program. When the Morton option is selected, the bearing properties as a function of speed are added to the Morton program. In addition the selected overhang rotor cg location and length to the overhang cg is determined from the rotor model. The previous Morton program would have required tables of information coming from two different programs as depicted in the flow chart of Figure 2.

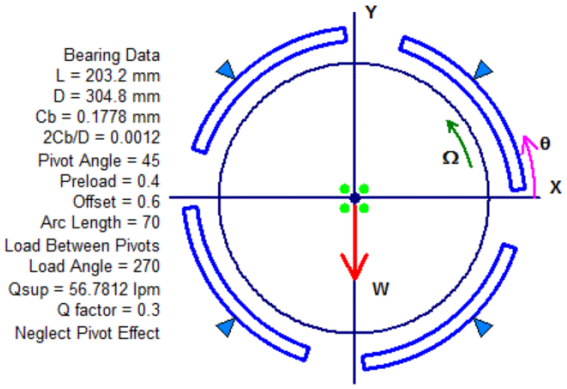

Figure 6 Example 1 Bearing 2 AT STATION 20

Figure 4 shows the imbalance at the left overhang cg. Additional parameters are provided in a window where imbalance level, thermal properties of the rotor shaft, election of the overhang as right or left, and desired speed range for the stability plot. The results of the Morton thermal stability are given in Figure 7 and it is seen that this overhang would most likely have a violent instability if the design speed was at or above 6583 rpm.

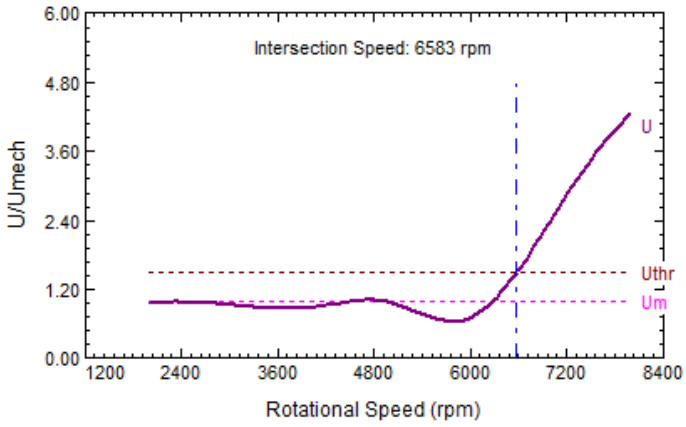

Figure 7 Stability for EX 1 Bearing 1 overhang showing a very rapid increase above 6000 rpm

For the right end overhang the results can be obtained by selecting bearing 2 located at station 20 in the Additional Input Parameters window shown in Figure 8. This is the advantage of the new design program, no tables to generate to run additional bearings.

Figure 9 is the stability plot for the right overhang and the threshold speed is predicted to be 5707 rpm, lower than the left overhang threshold of 6583 rpm. The right end has a smaller amplitude and would likely be less violent.

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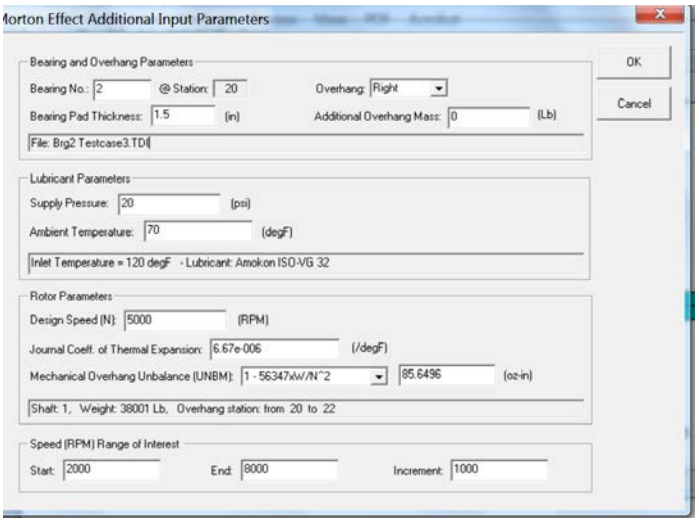

Figure 8 Additional Input Parameters for bearing 2

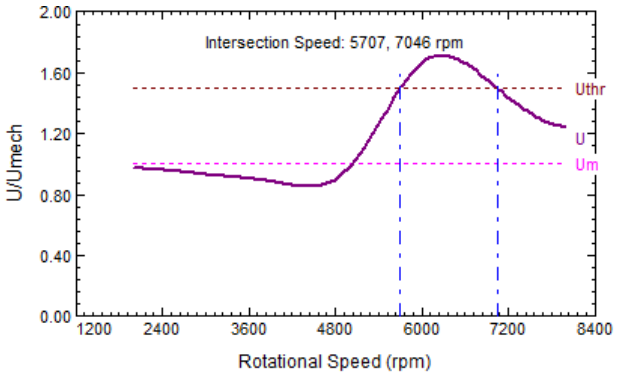

Figure 9 Stability plot for Ex 1 bearing 2 overhang

EXAMPLE 2

The next example is a double overhung turbo-compressor. The model is shown below in Figure 10. The

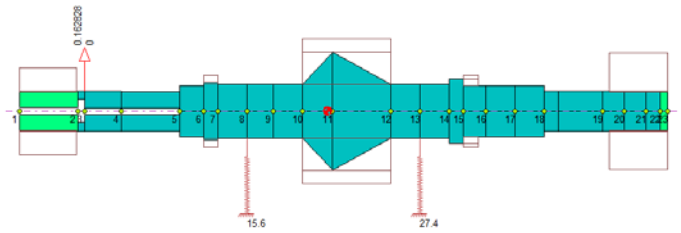

Figure 10 Model of a double overhung compressor

long overhangs and added weight at the ends of the shaft make it a good candidate for a Morton problem. Figure 10 shows the trial imbalance at the cg of the left overhang.

The cg of the right overhang is close to the imbalance shown in Figure 11. This rotor is much smaller than the turbine rotor of Example 1 and the bearings have much smaller diameters hence the operating design speed is much higher. Bearing loads are 663 and 578 N (149 and 131 lb) for bearing 1 and 2 respectively.

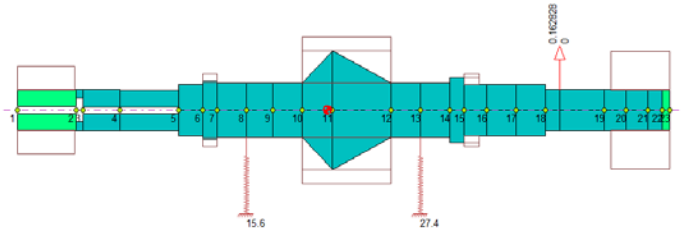

Figure 11 Model of a double overhung compressor showing cg location where the imbalance is located.

The stability calculations for the two ends of this rotor are given below. Figure 12 is the left overhang and is crossing over the threshold at 9,462 rpm and becomes stable at 12,314 rpm. This rotor may be able to run without a violent Morton instability.

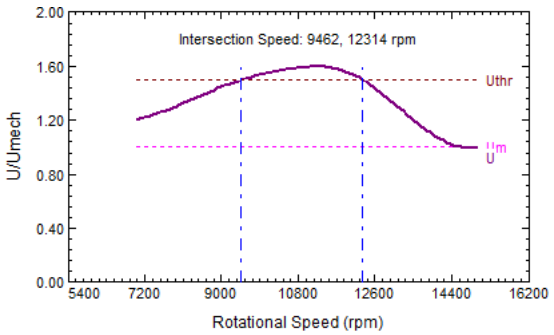

Figure 12 Stability of EX 1 bearing 1 left overhang

The result for the right overhang shows an onset at 10,747 rpm and regains stability at 12,020 rpm. It is not likely to be a problem, even more so than the left overhang.

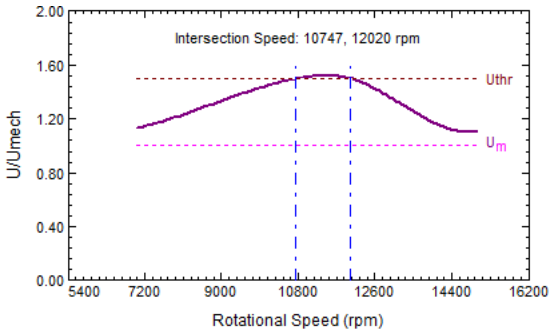

Figure 13 Stability of Ex 2 bearing 2 right overhang

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EXAMPLE 3

The third example is a more complicated train analysis. The simple model was made to test the train capability of the new program since the system now has a potential problem at four overhang locations as shown in Figure 14.

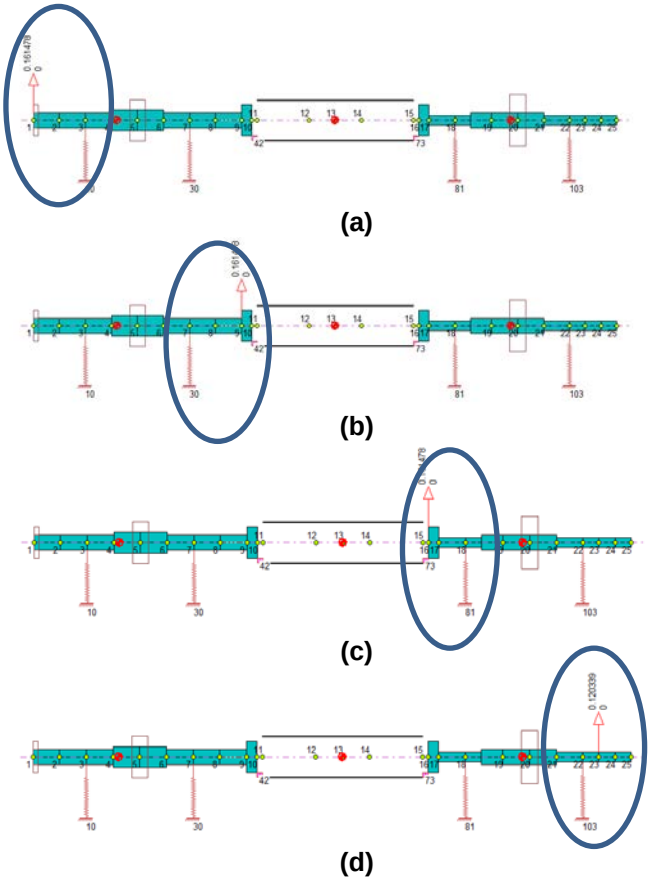

Figure 14 Rotor models showing cg of overhang for the four bearings in the train of Example 3

The bearings for the first shaft, bearings 1 and 2 have a diameter of 76 mm (3 in.) and length of 38 mm (1.5 in.) and bearing clearance of 100 micron (4 mils). The second rotor bearings 3 and 4 have a diameter of 50 mm (2 in.) and length of 25 mm (1 in.) and bearing clearance of 75 micron (3 mils). All 4 bearings are 5 pad with load between pivot. The load on the 4 bearings are 823, 560, 489, and 423 N (185 126, 145, and 95 lb) for bearing 1, 2, 3, and 4 respectively.

Real machines have had vibrations with symptoms similar to the Morton Effect but all cases with high synchronous vibration are not actually due to the Morton Effect. Some cases are due to loose turbine disks on hot machines, some are due to rubs, and some may be due to a long coupling thermal instability. The discussion of some of these other problems can be found in recent technical papers [18, 19, 20]. Light rubs can cause sub-synchronous vibration at 50% of rotor speed, while heavier rubs can cause thermal bowing, usually at mid-span.

The stability of the current example has been studied by the new Morton Effect program. This requires four runs, one for each overhang as shown in Figure 14. The results are given in Figure 15.

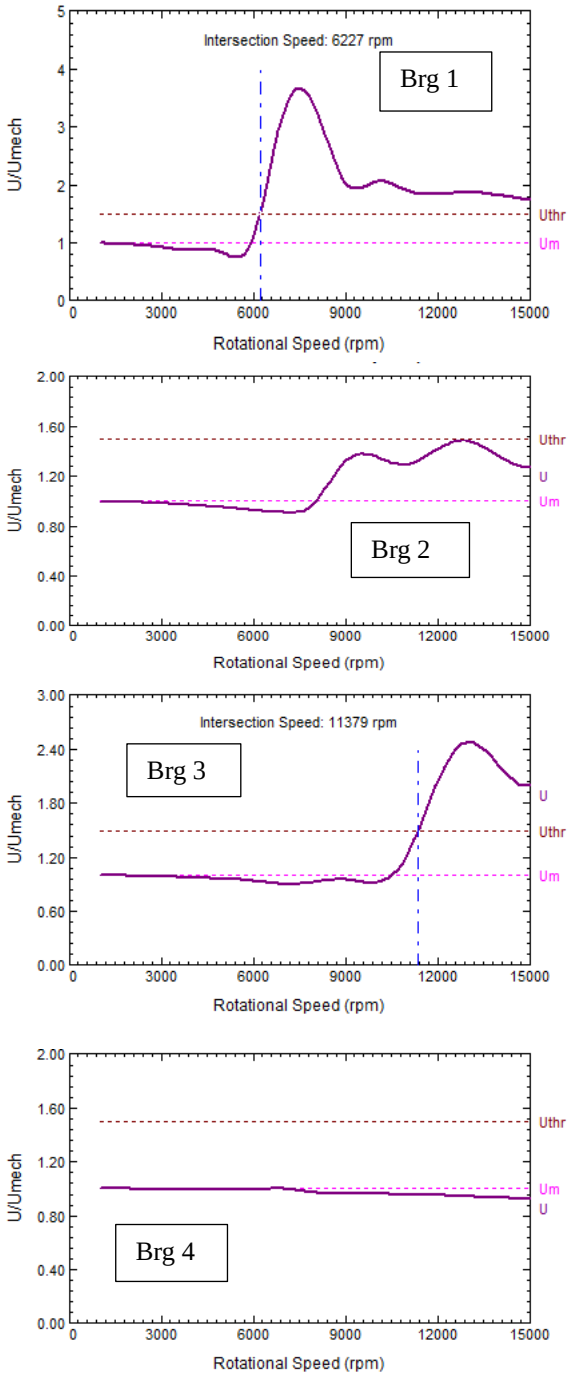

Figure 15 Rotor stability for Example 3

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It is clear that bearing 1 overhang has the best chance of developing a Morton Effect instability just beyond 6000 rpm and the overhang of bearing 3 has a potential instability at just beyond 11,000 rpm. The bearing 2 overhang has a very small chance of seeing a problem and bearing 4 has no chance of instability. .

To eliminate the instabilities is more challenging but it will require much less time to reach a solution with the new program capability.

CONCLUSIONS

This paper gives a summary of the most recent work on the Morton Effect instability prediction. The new version of the overhang analysis was linked to the Dyrobes rotor analysis in an attempt to produce a capability that could be easily run in a short time if the rotor model and bearing files are available. The previous day or more of time required to run the analysis for a 2 bearing machine has been reduced to less than 15 minutes to get the current design prediction. If the design is predicted to be unstable, the resolution of the problem will also require much less time.

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