RETUNING A GEAR DISC PROFILE TO ELIMINATE HIGH GEAR VIBRATION

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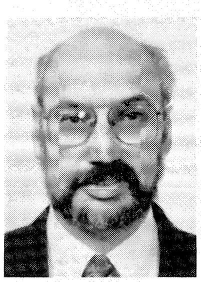
Mr. Hudson graduated with a B.S.M.E. from Newark College of Engineering (1965). He has been a taskforce member on the 4th, 5th, and 6th editions of the API 617 Specification for Centrifugal Compressors, the API task force on Quality Improvement, and the API taskforce on rotordynamics. He has participated in previous symposia as a discussion leader for the topics of compressor maintenance and compressor performance. He has published papers on torsional vibration, lateral vibration, and a paper at the Twenty-First Turbomachinery Symposium. He is a registered Professional Engineer in the State of Wisconsin.



Keith Rouch received a B.S. degree in Engineering from Purdue University (1965), followed by an M.S. degree (1967). He was employed by the Allis-Chalmers Corporation from 1966 to 1985, and received his Ph.D. degree from Marquette University (1978). In 1985, he became Associate Professor of Mechanical Engineering at the University of Kentucky, and Professor in 1993. His professional interests include development and application of finite element methods, analysis of dynamics of rotating machinery, and active vibration control in machining. He teaches courses in finite element methods, systems and controls, dynamics of rotating machinery, and mechanical design at the University of Kentucky and through the National Technological University.

Dr. Rouch has been awarded five patents, and has about 50 publications in journals and proceedings. He has served as Associate Editor of the ASME Journal of Tribology, and is a member of ASME, ASEE, STLE, and SME. While at the university, he has directed twelve students in graduate thesis efforts. He has served as a consultant to a number of firms in finite element applications, rotordynamics, and design reviews.

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ABSTRACT

High gear case acceleration levels, experienced during shop string testing of a compressor train, were traced to gear element structural resonance, which was excited by the gear tooth meshing frequency.

Finite element analysis predicted the original gear natural frequencies and was used to determine gear recontours that successfully eliminated the gear vibration.

INTRODUCTION

Details are presented of a vibration problem with a parallel shaft double helical speed increasing gear that was purchased for a two body horizontally split centrifugal compressor train. The compressors are used in refinery wet gas service. The driver is a fixed speed 6000 hp induction motor. The compressors and gear were purchased to API specifications 617 and 613, respectively.

The gear and motor were individually load tested within the capability of the vendors facilities. The compressor was specified to be full load loop tested with the contract gear, couplings, compressors, instrumentation and the contract separate lube and seal systems.

One of the technical requirements of the contract was to monitor and record all contract vibration and temperature instruments on the gear and compressors. Compliance with normal API limits was expected. Where API limits were not defined, customer technical specifications identified acceptance standards.

PROBLEM

During the compressor string testing, the signal levels from gear case accelerometers, which through most of the testing were approximately 1.0 G, increased to 8.0 to 10.0 Gs in a very narrow speed band. The nature of this problem is presented in

Figures 1 and 2. A waterfall plot is shown in Figure 1 of the gear accelerometer magnitudes recorded during coast down from overspeed. A spectrum plot is shown in Figure 2 of the gear accelerometer magnitudes recorded at 1785 rpm. In addition to the high accelerometer readings, an audible high pitched noise emanating from the gear box could also be heard. A spectrum analysis (Figure 3) of the sound pressure level was recorded with a microphone about 6.0 in from the side wall of the bottom of the gear box. Unfortunately, the 1785 rpm input speed to the gear, at which the problem occurred, was very close to the full load slip speed of the induction motor. The frequency at which these high accelerations occurred, 5720 Hz, corresponded very closely to the tooth mesh frequency. The gear had 196 teeth which created a tooth mesh frequency of 5831 Hz at 1785 rpm. The horizontal and vertical proximity probes at each end of both the gear and pinion did not exhibit any indication of the problem.

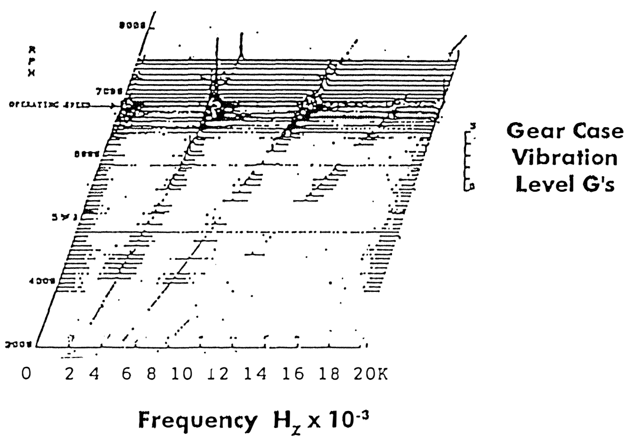

Figure 1. Gear Case Accelerometer Coastdown. Frequency Hz × 10^{-3}.

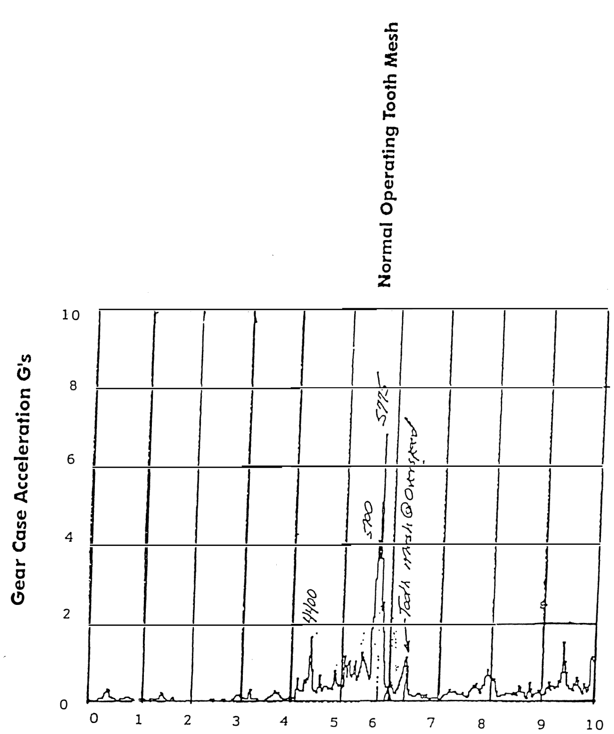

Figure 2. Gear Case Accelerometer Frequency (Hz × 10^{-3}).

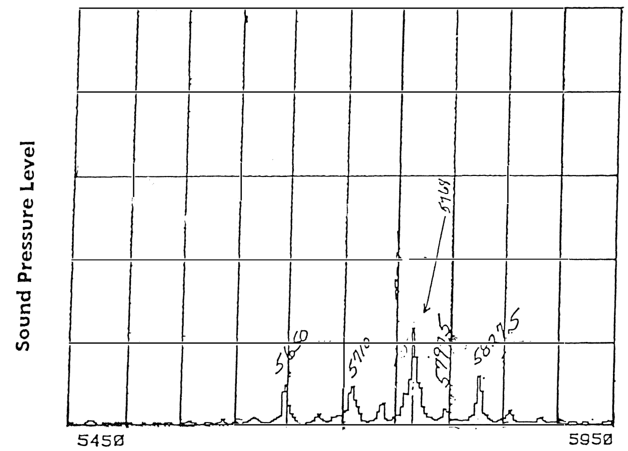

Figure 3. Sound Level Spectrum Hz. "Zoom" Range 5450 to 5950 Hz.

INVESTIGATION

After the problem had been documented, A-C Compressor and the customer, who was witnessing the test, contacted Lufkin Industries, the gear vendor. Lufkin did not have an explanation for the problem and questioned the temporary gear test setup at A-C Compressor. The gear had been supported on a large massive cast iron support structure. "Riser" blocks were used at the four corners of the gear to adjust the gear shaft centerline to that of the test driver. The photograph in Figure 4 is used to depict the gear support during the test. Lufkin questioned whether the nonuniform support under the gear may be causing the problem. A-C Compressor had prior experience of test problems caused by support pedestals, however, the problem frequencies were less than 100 Hz.

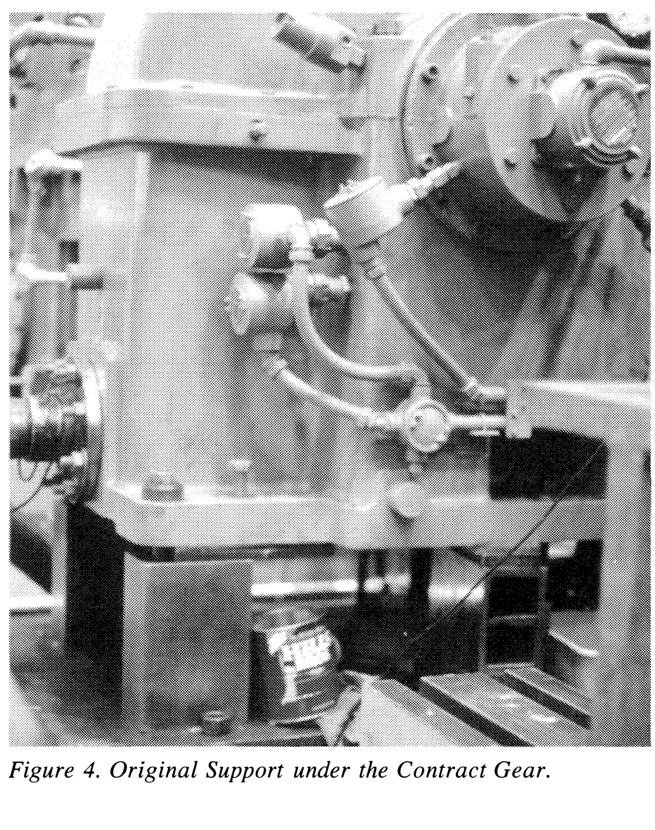

Figure 4. Original Support under the Contract Gear.

The base was modified by replacing the "riser" blocks with a solid 5.0 in thick steel plate machined parallel on both sides (Figure 5). The gear was retested after the base was modified. The base modification did not eliminate or minimize the problem or alter the speed at which it had occurred.

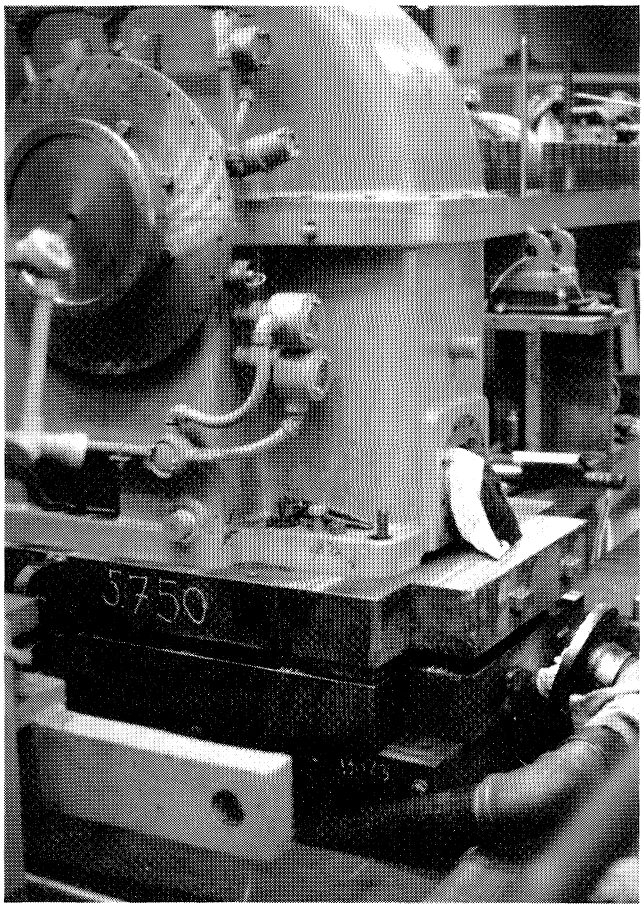

Figure 5. Contract Gear Test Support Modified with a 5.0 In Support Plate.

After the revised support tests were unsuccessful, the gear company, the ultimate user, and the compressor manufacturer met to establish probable causes and courses of corrective action. It was postulated that the most probable cause was a structural resonance of the gear. The assumption was made because of the narrow speed range at which the problem occurred. The vibration appeared at 1782 rpm and was gone by 1788 rpm.

Prior to the purchase of the gear, the end user had compared the proposed gear offering to two similar gears that the end user had in service. One of the gears in operation had the same model number, horsepower rating, identical input speed and a pinion speed 400 rpm less than the proposed gear. The second gear in operation had the same model number, a rating of 5000 hp and identical input and output speeds. The field operation of the two installed gears was very good. There was no record of field problems. The accelerometer magnitudes during operation of both gears were reported as being less than 1.0 G.

When the gearing details were compared, it was found that the two operating gears were constructed with a four diametral pitch, whereas the problem gear was constructed with a six diametral pitch. A comparison of diametral pitch is shown in Figure 6. The design change was intended to increase the number of teeth in mesh to lower the noise level of the gearing. The result of the change had increased the tooth mesh frequency by 50 percent. The two operating gears had tooth mesh frequencies of 3088 Hz and 3956 Hz, whereas the problem gear had a tooth mesh frequency of 5831 Hz.

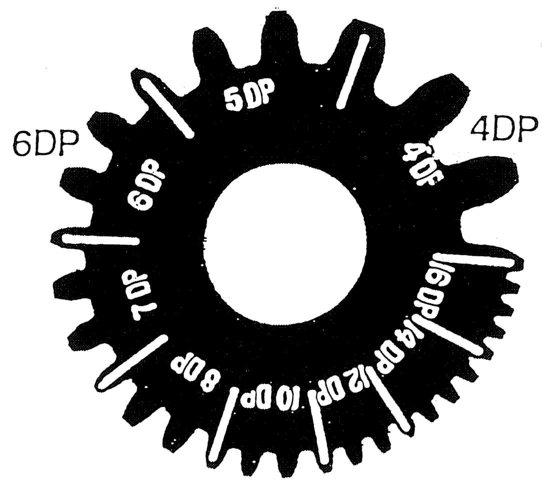

Figure 6. Gear Pitch Template.

TESTING OF THE GEARS TO IDENTIFY STRUCTURAL FREQUENCIES

It was decided to remove the gear cover and "rap" test the gear and measure its resonant signature. Both the gear and pinion rested in their bearings during the test. A plot of the rap test is shown in Figure 7. A multitude of natural frequencies were found in this test but the most notable was a frequency found at 5750 Hz, very close to the accelerometer frequency of 5720 Hz, found during the shop string test.

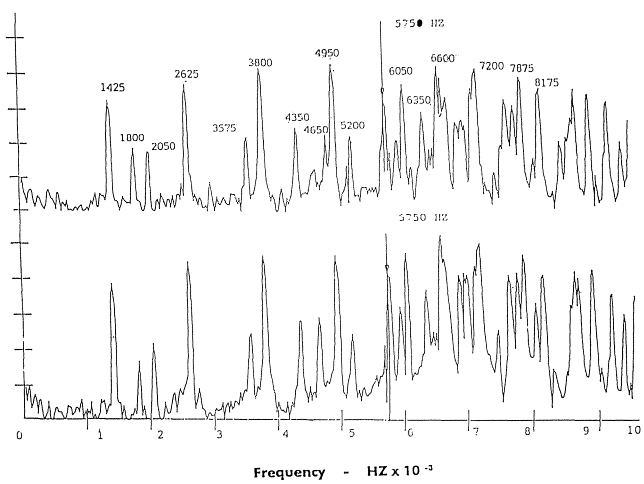

Figure 7. Spectrum from Ring Test—Upper Plot Pinion and Gear in Gear Case. Frequency Hz. × 10^{-3}.

The pinion was removed from the gear in the event there might have been some mesh interaction between the gear and pinion. The gear rap test was repeated with nearly identical results. After the rap tests were performed it was postulated that the gear structural frequency might be a disc mode in which the disc would vibrate in a "pie" shape mode shape. A sketch of the modal patterns of a circular disc is presented in Figure 8. Disc frequencies were defined many years ago in work published by Campbell [1] and Grinsted [2].

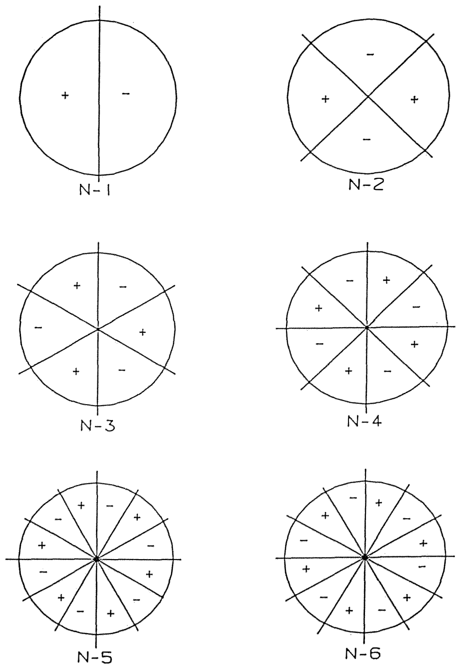

Figure 8. Typical Disc Mode Shapes.

Data Syst Engineering and Testing Services Inc. was contacted to perform an electromagnetic excitation of the gear. The gear shaft was mounted on the floor in shop "horses" with plastic lined "V" shaped saddles. An electromagnet was located at the outer diameter of the side wall of the gear and driven by a variable frequency A-C oscillator. The test setup is shown in Figures 9 and 10. The oscillator frequency was varied until an audible buzz was heard from the gear. The gear side wall was then probed circumferentially with an accelerometer and the apparent mode shape was determined. The tests determined that a "N-3" node existed at 5767 Hz.

PROPOSED REDESIGN OF GEAR TO ALTER STRUCTURAL FREQUENCIES

The results of the testing convinced everyone that the gear geometry should be altered to "retune" the gear. The gear geometry can be seen in Figure 11. The gear blank was a single piece forging approximately 38 1/2 in diameter and 13 1/2 in thick. It was shrunk and keyed on a 8 1/4 in shaft.

It was suggested that if the gear mass were altered, the frequency that coincided with the tooth mesh frequency could be altered by modifying the disc geometry. The gear manufacturer informed the researchers that they had a similar problem previously and attempted to recontour the gear by removing 1.0 in of material along the gear side wall. The gear company described this as "coping" the gear. It was reported that this recontour was not effective in retuning the problem frequency. With this knowledge, it was discussed and decided that a more rigorous approach than "trial and error" remachining should be pursued. The approach to resolve this problem began with a decision to develop a finite element model of the gear and the shaft as

Figure 9. Variable Frequency A-C Oscillator and Electronics Used to Excite and Monitor Gear Resonance.

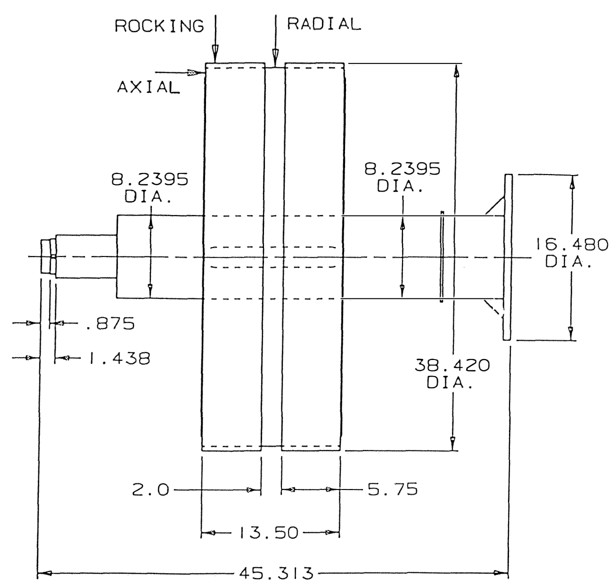

Figure 10. Gear Assembly with Accelerometer Locations to Identify Disc Modes.

designed and compare the test results to the analytical results. If the correlation between the analysis and test data was good, a revised model would be developed that would enable successful retuning of the gear. The compressor company retained Dr. Keith Rouch to model the gear. Dr. Rouch is an experienced consultant in the application of finite element analysis techniques in the field of turbomachinery design.

FINITE ELEMENT ANALYSIS

The first analysis was of the gear blank alone, as depicted in Figure 11, without a shaft. The model was an accurate representation of the machined details of the gear except for the gear teeth. The gear teeth area was modelled as the outer diameter of a simple cylinder with a diameter equal to the turned diameter between the helix. The calculation was evaluated through a frequency equal to twice the tooth mesh frequency. When the results were analyzed, it was determined that six disc modes did exist within this frequency range. In addition, it was revealed that for each of these disc modes there were complex mode shapes that included axial and/or radial displacements within a given circumferential "pie" shaped mode. The frequencies of this first analysis are contained in Table 1.

Table 1. Natural Frequencies (Hz) of the Machined Gear Without Shaft.

Circum. Mode No.	A	B	C	D	E
N-1	575	906	3090	3596	4001
N-2	1483	2524	4450	4710	6434
N-3	2694	3825	5732	5842	7132
N-4	3896	4956	6688	6696	8228
N-5	5062	6000	7311	7636	9505
N-6	6191	6986	8013	8570	10428

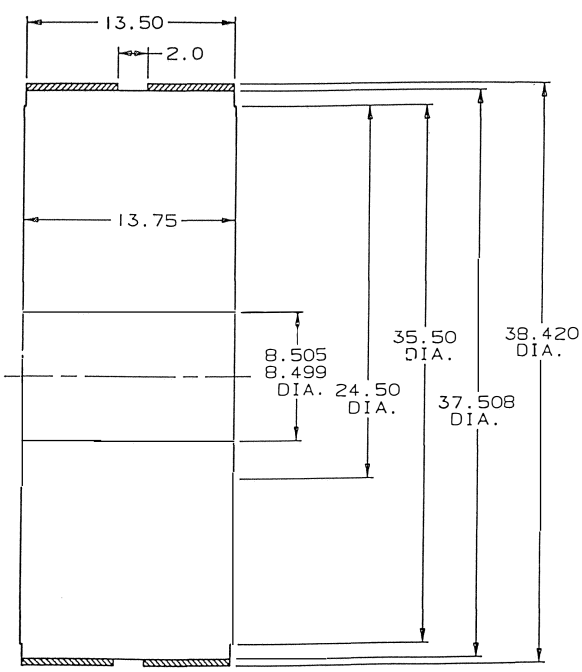

Figure 11. Machined Gear Geometry (Original Design).

It can be seen that each circumferential mode consisted of five complex mode shapes labeled A, B, C, D, and E. Despite many years of design and analysis backgrounds of the numerous people involved, these mode shapes were a revelation. These complex mode shapes are detailed in the following sections.

Mode Shape "A"

This mode contains bending of the disc in the axial direction in addition to the circumferential mode. The deflected mode shape is shown in Figure 12. This "A" mode is the classic disc mode most people would refer to as the disc frequency.

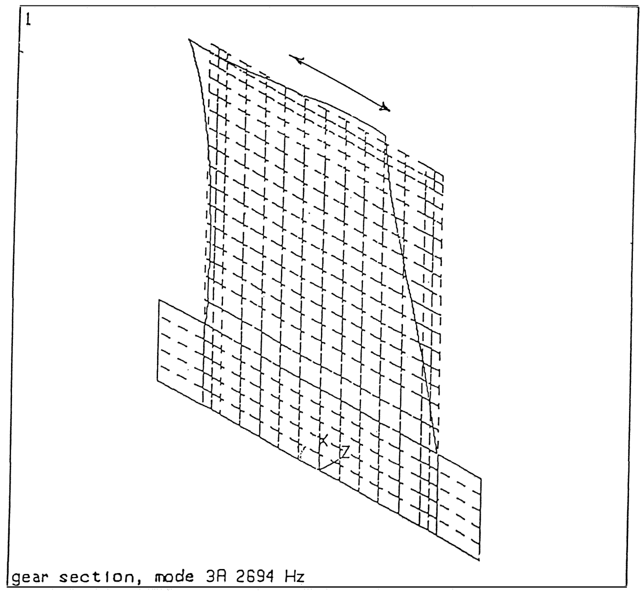

Figure 12. N-3A Gear Mode Shape (Original Design).

Mode Shape "B"

This mode contains radial deflection of the disc in addition to the circumferential mode. In addition, there are concave and convex deflections of the gear side walls. The deflected mode shape is depicted in Figure 13.

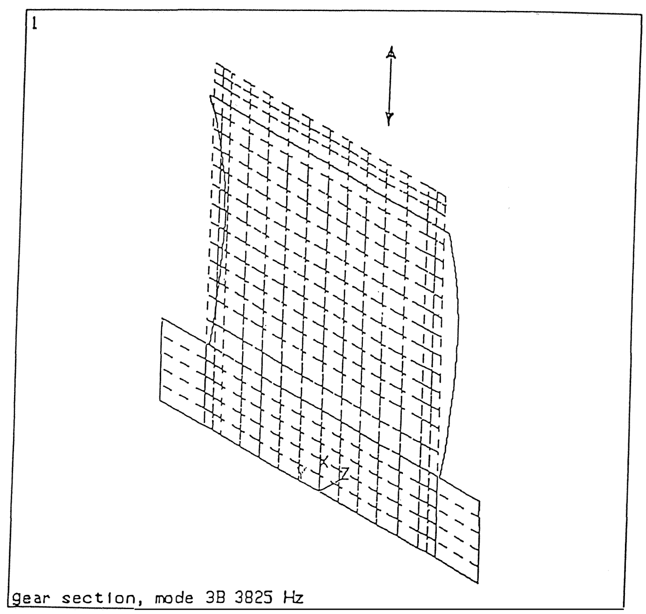

Figure 13. N-3B Gear Mode Shape (Original Design).

Mode Shape "C"

This mode contains complex bending of the side wall of the disc resulting in combined radial and axial deflection or "rocking" at the outer surface of the disc. The deflected mode shape is shown in Figure 14.

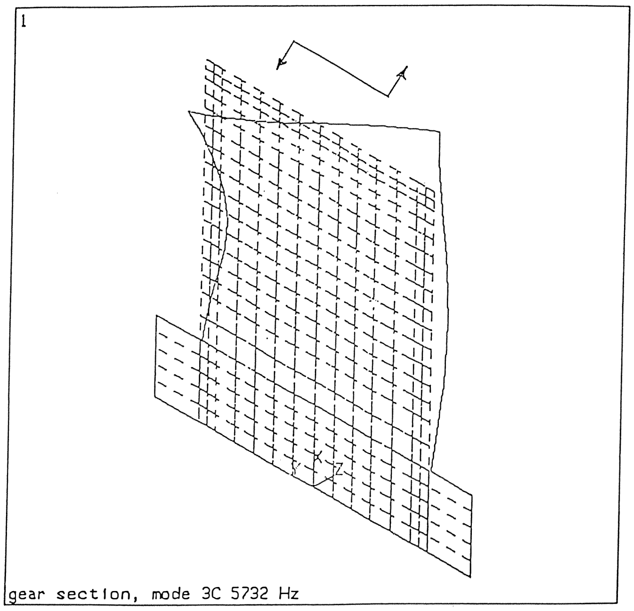

Figure 14. N-3C Gear Mode Shape (Original Design).

Mode Shape "D"

This mode is rather difficult to show in a planer plot. The side walls deflect axially out of phase with each other in conjunction with a substantial tangential deflection along the gear pitch line. This mode shape is depicted in Figure 15.

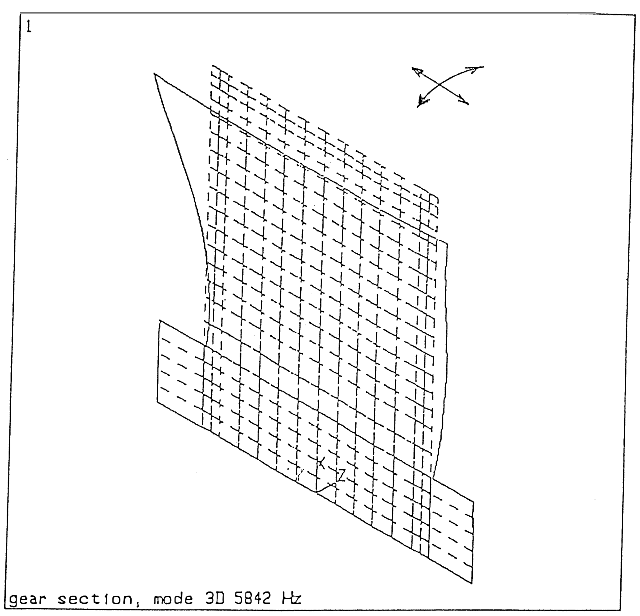

Figure 15. N-3D Gear Mode Shape (Original Design).

Mode Shape "E"

This mode consists of complex concave/convex deflection of one side wall and axial bending of the opposite side wall. A plot of this mode shape is presented in Figure 16.

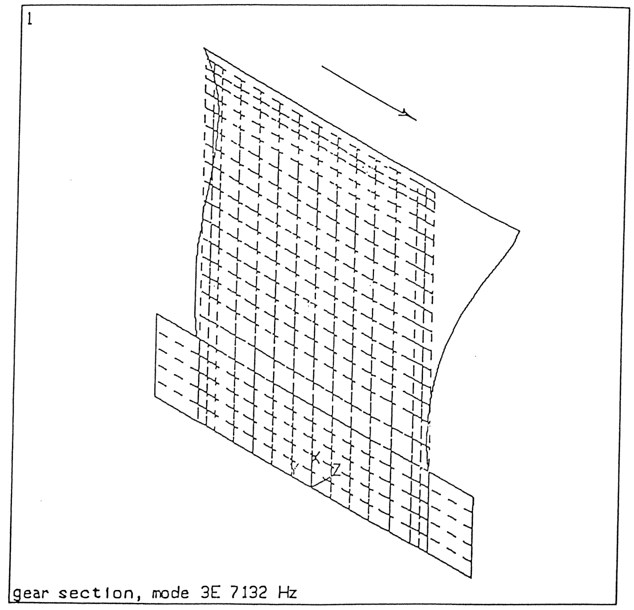

Figure 16. Gear Mode Shape (Original Design).

The previous electromagnetic testing had revealed a N-3 mode at 5767 Hz. The calculations revealed that there are two N-3 frequencies. A "C" mode existed at 5732 Hz and a "D" mode at 5842 Hz. The exciter test frequency, 5767 Hz, was 0.61 percent above the "C" mode and 1.28 percent below the "D" mode. The relative proximity of the test data and analytical results was very encouraging and indicated the finite element approach to redesign might indeed be successful.

In order to see if any of the modes might be modified by other factors such as stresses created by shrink fits and centrifugal stresses, another run was performed. The results of this calculation reveals virtually no effect from shrink fits and rotation (Table 2).

Table 2. Natural Frequencies (Hz) of the Machined Gear with a Shrink Fit on the Shaft at 1785 RPM.

Circum. Mode No.	A	B	C	D	E
N-1	575	906	3089	3601	4004
N-2	1484	2524	4450	4710	6434
N-3	2694	3826	5732	5842	7132
N-4	3897	4957	6688	6696	8228
N-5	5063	6000	7311	7637	9498
N-6	6191	6987	8013	8570	10429

Because the accelerometer running frequency of 5720 Hz, the static rap test frequency of 5750 Hz, the electromatic excitation test frequency of 5767 Hz and the calculations were very close, but not exact, it was decided to repeat the electromagnetic excitation tests. Armed with the new knowledge of the complex mode shapes, the gear vibration would be closely examined to determine if these mode shapes could be excited and documented.

The tests were performed with the exciter located at the outer diameter side wall for axial excitation, at the center groove between the helices for radial excitation, and at one edge of the outer diameter of the gear teeth for "rocking" excitation. All the modes are revealed in Table 3 that could be excited, the manner in which they were excited, and the frequency where actual resonance was determined. The correlation error was found to vary from as little as 0.5 percent to as great as 4.7 percent. From the test data it was established that only the "E" mode could not be excited.

Table 3. Calculated and Tested Natural Frequencies (Hz) with % Error, of the Machined Gear Mounted on the Shaft.

Circum. Mode No.	A	B	C	D	E
N-1	575	906	3090	3596	4001
N-2	1483 (1443)A	2524 (2481)RC	4450 (4661)RE	4710	6434
N-3	2694 (3632)RE	3825 (3768)RC	5732 (5701)RE	5842 (5728)A	7132
N-4	3896 (3804)A	4956 (4922)RC	6688 (6650)RE	6696 (6600)A (6601)RC	8228
N-5	5062 (4947)A	6000 (5937)A (5939)RC (5941)RE	7311	7636	9505
N-6	6191 (6063)A (6064)RE	6986	8013	8570	10428

A = Axial excitation
RC = Radial excitation center of gear
RE = Radial excitation edge of gear

Circum. Mode No.	A	B	C	D	E
N-1	NF	NF	NF	NF	NF
N-2	-2.7%	-1.7%	+4.7%	NF	NF
N-3	-2.2%	-1.0%	-0.5%	-2.0%	NF
N-4	-2.4%	-0.7%	-0.6%	-1.4%	NF
N-5	-2.3%	-1.0%	NF	NF	NF
N-6	-2.1%	NF	NF	NF	NF

NF = Resonance not found

REDESIGN ANALYSIS

Two styles of redesigns were considered. Type 1 included coped side walls. A sketch of the modified design is presented in Figure 17. Two cope depths 1.0 in and 2.0 in were analyzed. The type 2 design included four 4.0 in diameter holes entirely through the gear, as shown in Figure 18. The analysis determined that the holes made very little change. The largest change of any frequency was 2.2 percent (Table 4). It was decided not to pursue this design.

Table 4. Natural Frequencies (Hz) of the Machined Gear with Four Through Holes.

Circum. Mode No.	A	B	C	D	E
N-1	584	923	3057	3608	4002
N-2	1468	2526	4431	4684	6408
N-3	2648	3761	5637	5823	7138
N-4	3834	4828	6550	6652	8237
N-5	4994	5837	7258	7466	9403
N-6	6123	6822	7945	8396	

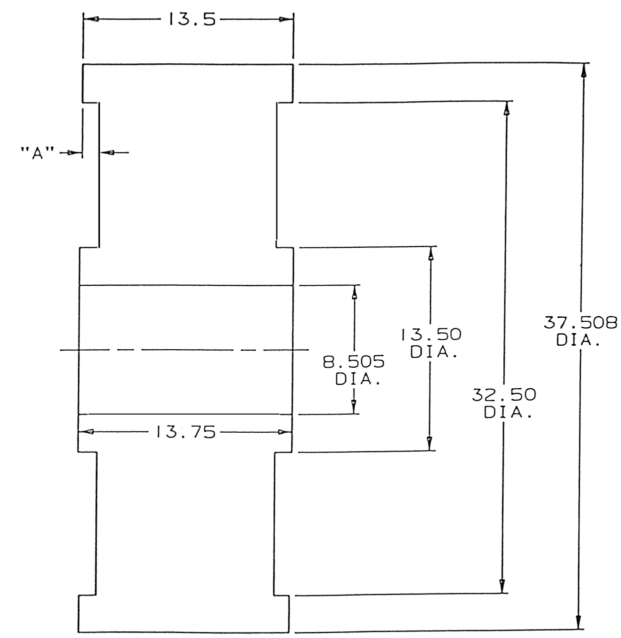

Figure 17. Proposed Redesign with "Coped" Sidewalls of Depth "A."

The results are contained in Table 5 for the 1.0 in side wall coping. Previous testing had shown that the responsive frequencies, except for the N-2/C mode, were lower than calculated. The N-3/D mode was calculated to occur at 5917 Hz. If the modified design had the same test variance (- 2.0 percent) as the original gear, its modified N-3/D mode would occur at 5798.7 Hz. It was felt that this would be too close to the tooth mesh frequency.

Table 5. Natural Frequencies (Hz) of the Machined Gear Modified with 1" Coping of the Sidewall.

Circum. Mode No.	A	B	C	D	E
N-1	594	933	3057	3439	4001
N-2	1360	2513	4447	4547	6553
N-3	2449	3729	5549	5917	6903
N-4	3554	4755	6477	6769	7858
N-5	4630	5657	7327	7376	9040
N-6	5662	6470	8021	8275	9956

The results are contained in Table 6 for the 2.0 in side wall coping. It is of interest to see that where the 1.0 in coping had raised the N-3/D mode 65 Hz, the 2.0 in coping resulted in a frequency only 12 Hz higher, an increase of only 0.2 percent above that of the original gear. In addition, the N-4/C mode had reduced to 5749 Hz a location for concern. This modification was obviously not the answer.

Table 6. Natural Frequencies (Hz) of the Machined Gear Modified with 2" Coping of the Sidewall.

Circum. Mode No.	A	B	C	D	E
N-1	592	953	3011	3125	4001
N-2	1193	2448	4111	4421	5630
N-3	2152	3528	4948	5854	6011
N-4	3151	4350	5749	6360	7381
N-5	4103	5002	6577	7066	8388
N-6	4994	5600	7460	7915	9139

Three more analyses were made to see if the coping dimension could be optimized. Runs were made for 0.5 in, 1.5 in, and a tapered cope ranging from 0.5 in deep at the start of the inner diameter recess to 2.0 deep at the outer diameter end of the relief. In all cases, the N-3/D mode frequency varied by no more than 1.4 percent. Also the 0.5 in cope design and the tapered cope design positioned the N-3/D modes near the tooth mesh frequency.

In retrospect, this analysis enabled the authors to understand statements made by the gear manufacturer that coping the gear side wall had not worked on a previous problem.

After some reflection of the N-3/D mode shape, it was apparent that there was significant tangential deflection of the gear outer diameter.

It was assumed that modification to the center groove between the gear mesh might alter the frequencies of the gear. A model was generated which kept the identical geometry with one modification. A center groove 2.35 in wide by 2.5 in deep radially was added to the model. The results are shown in Table 7. A comparison to the original gear results revealed that considerable changes were made in "C" modes. Changes occurred in other modes to lesser degrees. A summary is presented in Table 7 of the changes from the original model. It was felt that a combination of coping and center groove modification could produce a gear with adequate separation from the tooth mesh frequency. A number of these calculations were made, but for the sake of brevity are not tabulated. The final design that gave the widest separation from tooth mesh frequency was a gear with a groove 2.0 in wide by 2.0 in deep between the helices with 0.5 in coped from the side walls. The final redesign frequencies are listed in Table 8. The "D" mode frequency proved the most difficult to retune. The analysis revealed that the N-3/D mode would be 3.2 percent below the gear mesh frequency. In addition, the excitation tests done previously had located the actual N-3/D mode 2 percent below the calculation. It was expected the test results for the modified gear would be similar. The closest frequency above the gear mesh frequency was the N-6/B mode with a frequency of 6132 Hz. This mode was 4.8 percent above the gear mesh frequency. Previous excitation tests had a maximum error of -2.3 percent. The separation margins were narrow; however, they represented the best design that evolved from a multitude of finite element analysis modes. It was decided to remachine the gear to this configuration.

Table 7. Natural Frequencies (Hz) of the Machined Gear Modified with a 2.35" Wide by 2.5" Deep Center Groove.

Circum. Mode No.	A	B	C	D	E
N-1	595	927	3127	3594	4000
N-2	1451	2582	4036	4659	4711
N-3	2530	3868	4304	5719	6245
N-4	3582	4569	5186	6623	7826
N-5	4611	5208	6267	7467	8888
N-6	5632	5988	7318	8295	9954

Table 8. Natural Frequencies (Hz) of the Machined Gear Modified with a 2.0" Wide by 2.0" Deep Center Groove and 0.5" Coping of the Sidewall to 30.75" O.D.

Circum. Mode No.	A	B	C	D	E
N-1	597	932	3103	3534	4001
N-2	1403	2556	4361	4642	4807
N-3	2458	3804	4751	5663	6221
N-4	3507	4716	5327	6613	7796
N-5	4538	5395	6266	7513	9209
N-6	5559	6132	7269	8386	

Some mention should be made of the modal configuration of the coped side wall and center groove redesign. The "A" mode remained primarily axial (Figure 19), and the "B" mode remained primarily radial (Figure 20). The "C" mode took a unique mode shape onto itself. The "C" mode consisted of significant out of phase axial displacement of the segment on either side of the center groove as shown in Figure 21. The "D"

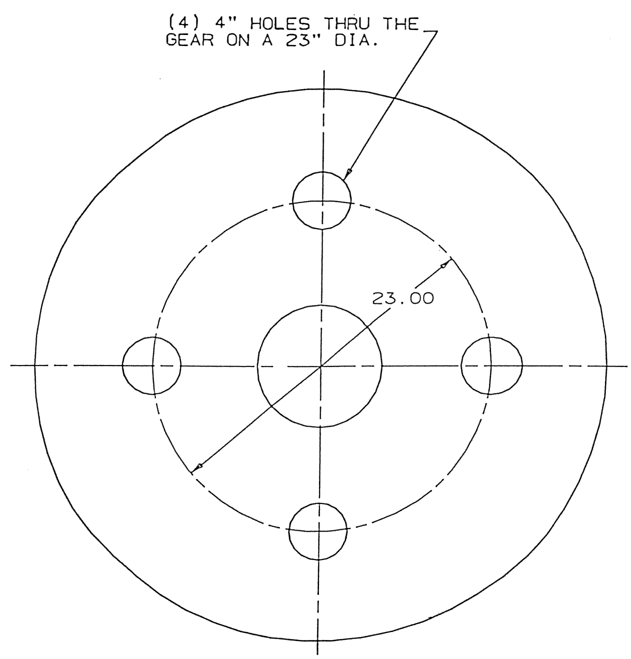

Figure 18. Proposed Modification—Four Through Holes.

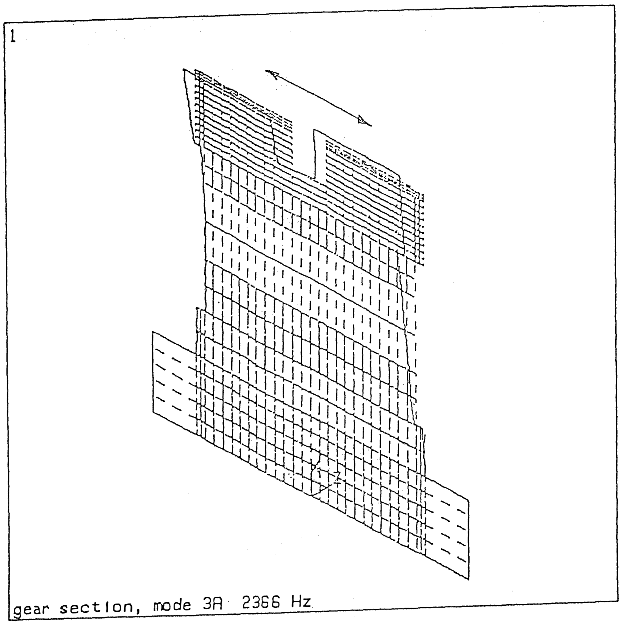

Figure 19. N-3A Gear Mode Shape (Modified Design).

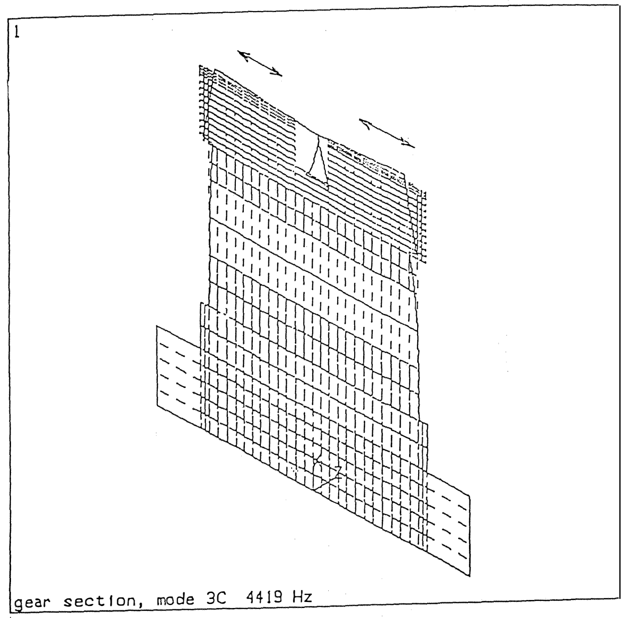

Figure 21. N-3C Gear Mode Shape (Modified Design).

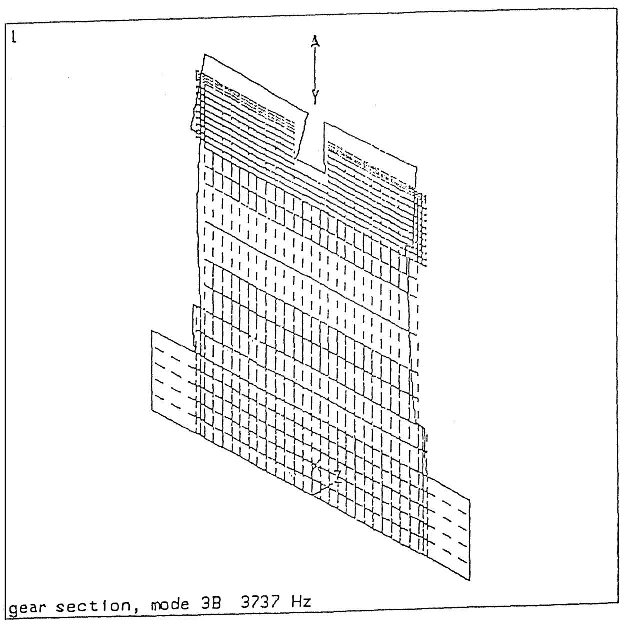

Figure 20. N-3B Gear Mode Shape (Modified Design).

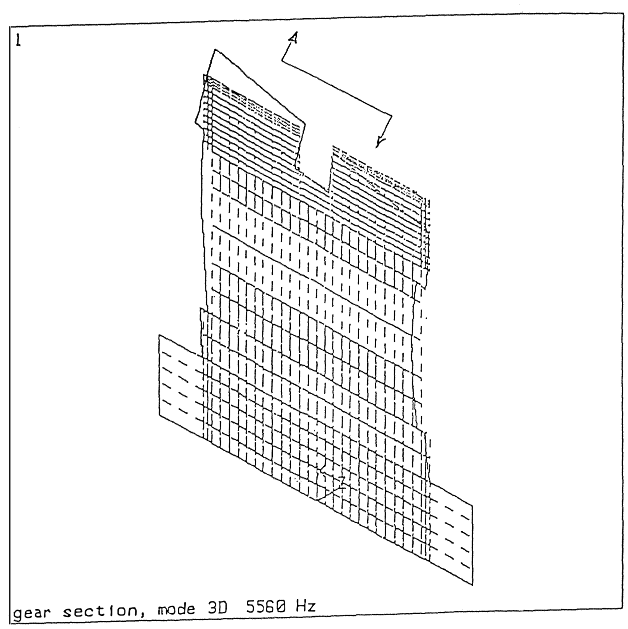

Figure 22. N-3D Gear Mode Shape (Modified Design).

mode became the "rocking mode" (Figure 22) and the "E" mode became the "circumferential/axial" mode (Figure 23).

The remachining of the gear proved to be somewhat of a challenge. The gear was AISI 4340H through hardened to 335-377 Brinell hardness (335-377 BHN). It took nearly 24 hours of round-the-clock machining to recontour the gear. The gear was left mounted on the shaft for the remachining. A photograph of the remachined gear is presented in Figure 24.

After the gear was remachined, it was planned to "rap" test the gear for a signature and then re-excite the gear electromagnetically to verify the gear frequencies.

When the gear was rap tested, everyone was aghast at the results. The gear as before had numerous spikes that identified resonant frequencies. Unfortunately one of the peaks occurred at 5775 Hz, only 25 Hz different from the original gear design. This frequency appeared whether the hammer blow was given in the radial or axial direction. The signature of the machined gear is represented in Figure 25.

The gear was positioned for a test with the electromagnetic exciter. The gear was excited in the axial direction and radially at the center of the groove and again at the radial edge of the gear. The original test had located 14 frequencies where resonance was detected. The modified gear when tested revealed eight resonant frequencies. Unfortunately, one of the resonant frequencies turned out to be 5780 rpm. The summary of the resonant frequencies found during the exciter test are shown in Table 9.

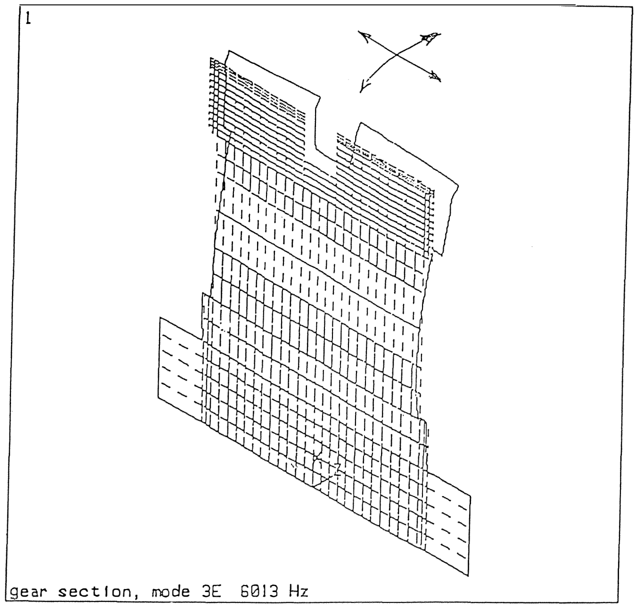

Figure 23. N-3E Gear Mode Shape (Modified Design).

Table 9. Natural Frequencies (Hz), Calculated and Tested, of the Actual Gear Redesign with Gear Tooth Mass Included. Lower Table Lists Error Between Tested and Calculated Frequencies.

Circum. Mode No.	A	B	C	D	E
N-1	584	922	3035	3476	4001
N-2	1361 [1327]	2527	4117	4554	4566
N-3	2366 [2349]	3737	4419 [4383]	5560	6013
N-4	3355 [3352]	4477	5121	6468	7538
N-5	4324	5055 [5065]	6095 [6135]	7310	8694
N-6	5280 [5326]	5739 [5780]	7068	8113	9664

[Test Frequency]

Error in Frequency (Percent)

Circum. Mode No.	A	B	C	D	E
N-1	--	--	--	--	--
N-2	-2.5	--	--	--	--
N-3	- .7	--	- .8	--	--
N-4	- .1	--	--	--	--
N-5	--	+ .2	+ .96	--	--
N-6	+ .9	+ .7	--	--	--

Despite the testing and long hours put into redesign and remachining the gear all the work appeared to be for naught. The machining that had been done could not be reversed. Fortunately, they had decided not to remachine the spare gear. It was decided to install the gear and retest it.

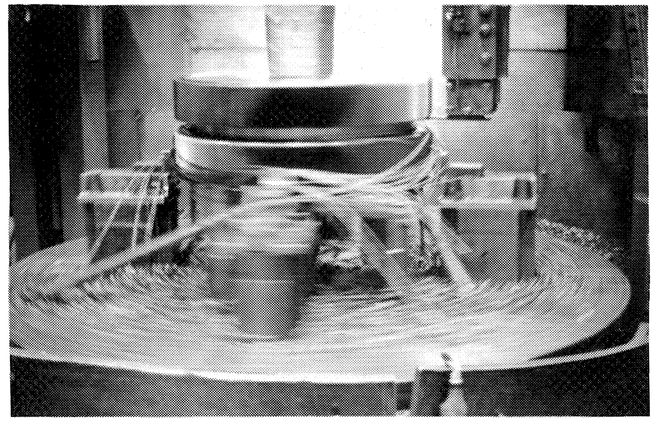

Figure 24. Gear Being Machined for the Modified Design Center Groove Addition.

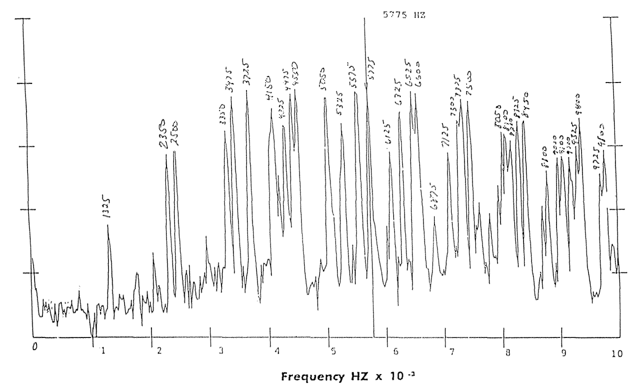

Figure 25. Spectrum from Ring (Modified Design). Frequency Hz × 10^{-3}.

While the gear was being reassembled, Dr. Rouch was given the test results to review. The finite element model was pursued to ensure there were no modelling errors. The final remachining instructions and the dimension of the remachined gear were all double checked and verified as being correct. The mesh detail was refined in the areas of the center groove and side coping, all to no avail. Finally, the model was modified by adding the actual mass of the gear teeth at the root diameter of the teeth where the model had previously stopped. No change was made to the model to alter the stiffness in the region of the gear teeth. This final modification resulted in the calculated N-6/B mode of 5739 Hz, only 0.7 percent below the actual exciter test frequency of 5780 Hz. Refer to Table 9 for a comparison of the test and calculated frequencies.

After this model was developed and its accuracy verified, further redesigns were considered. It was found that a 1.0 in deep cope would remove all the modes from the tooth mesh frequency except for the N-3/E mode. Since the "E" mode could never be excited, this was assumed to be a safe design. It was assumed the gear would have to be recontoured a second time.

Shortly after the calculations were completed, the gear was retested. To the amazement of all involved, the gear accelerometer never registered above 2.0 Gs throughout the range of operation up to and including overspeed. A peak hold plot is shown in Figure 26 of the accelerometer on the gear.

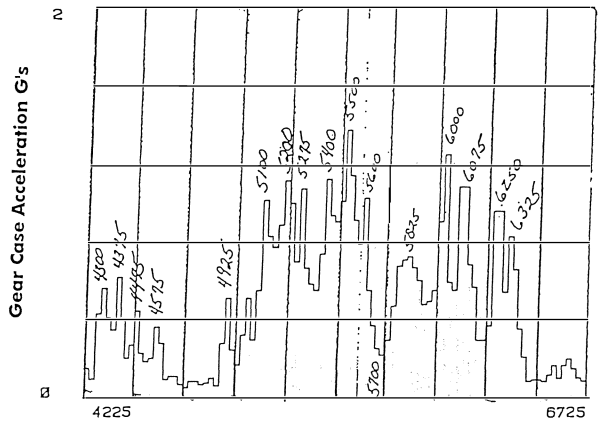

Figure 26. Peak Hold Gear Case Acceleration (Modified Gear Design) "Zoom" Range 4225 to 6725 Hz.

After the test, the option to revise the gear coping on the second gear was discussed. It was decided to machine both gears identically to each other. The signature of the second gear, when it was part of the second string test, was very similar to the first gear.

The elapsed time from when the investigator was first contacted to when the recontoured gear was retested was 15 days.

The compressor train is installed and has been operating over a year with low values of gear case acceleration.

SUMMARY

A gear case vibration with a magnitude of 10.0 Gs was found to exist in an extremely narrow speed range very close to the operating speed of the 6000 hp induction motor. The gear manufacturer had experienced one prior occurrence of this type of problem without any record of damage, however they could not assure the end user of reliable long term operation. Testing of the gear established a disc mode structural resonance of the gear. Finite element analysis predicted, with good accuracy, many of the frequencies found when the gear was tested. The finite element analysis also identified modal contours that were previously unknown to all parties involved.

Gear redesigns established by finite element analysis did not have the same correlation accuracy to tested frequencies until the model was refined to include the mass of the gear teeth.

Even though the redesigned gear had a structural frequency in the vicinity of tooth meshing frequency, within 0.4 percent of the original gear, it was not excited during the shop string retest.

While the gear natural frequency in the vicinity of the tooth mesh frequency was similar to the original design, the mode shape was not. The modal configuration of the redesigned gear was not excited by the gear tooth passing frequency.

CONCLUSIONS

- A gear structural natural frequency was excited during a near full load string test even though the gears were manufactured to a AGMA Quality Level 13.
- Even though prepurchase investigation identified the proposed gear as a near duplicate to operating equipment, contract design refinements resulted in a change that caused the gear resonance.
- Finite element analysis results were very close to frequencies found when the gear was electromagnetically excited. The analysis also identified many complex modal configurations associated with a given "pie" shaped disc mode.
- Due to the complex mode shapes and the multiplicity of structural natural frequencies, the gear required a very deliberate combination of geometry revision to properly position the various structural natural frequencies to avoid excitation by the tooth mesh frequency.
- Finite element redesign predictions did not correlate as well as the original calculations. The model had to be refined to include gear teeth mass to obtain good agreement with test data.
- Even though the redesigned gear had a structural natural frequency very close to that of the original gear, in the proximity of the tooth mesh frequency, it was not excited during the retest of the gear.
- The presence of a structural natural frequency in the vicinity of tooth mesh frequency is not inherently bad design. The modal configuration of the structural frequency and the resultant forcing functions are influential in whether or not the gear will respond.
- The loaded shop string test proved very beneficial in detecting a gear vibration that may have led to serious gear tooth damage and expensive down time of the refinery. The 15 day time span to resolve and rectify the root cause of the vibration most likely would not have been matched under field conditions.

REFERENCES

1. Campbell, W., "The Protection of Steam Turbine Disk Wheels from Axial Vibration," Presented at the ASME Spring Meeting (May 1924).
2. Grinsted, B., "Nodal Pattern Analysis," AMI Mech. E., May 1951, Proceedings of Institution of Mechanical Engineers (1952).