Vibration frequencies generated by paper machine differential gearbox drives

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ABSTRACT: Vibration data on differential gearboxes found on line shaft driven paper machines are analyzed, and background is given on computing gearmesh frequencies and shaft speeds found inside the gearboxes. Differential cluster mechanics are also covered. An example analysis spreadsheet is provided as a guide for the user; it identifies both good and bad frequencies to look for at each data collection position.

KEYWORDS: Bearings, differential gears, frequencies, gear boxes, mechanical drives, paper machines, rotation, shafts, velocity, vibration analysis.

The purpose of this paper is to aid the user in analyzing vibration data collected on differential gearboxes commonly found on line shaft driven paper machines. Background on computing gearmesh frequencies and shaft speeds found inside the gearboxes are given because it is very common to have a number of gearboxes on the same paper machine with different gear tooth counts, depending on the application, input speed, required draw, required horsepower, or required output speed. Also, differential cluster mechanics are covered because they require a special analysis to determine the resulting shaft speeds and gearmesh frequencies. Furthermore, an example analysis spreadsheet is be provided as a guide for the user. It identifies both "good" and "bad" frequencies to look for at each data collection position. For example, one expects to find 1X GMF (gearmesh frequency). However, a bearing defect frequency or harmonic indicates gearbox bearing problems developing.

Calculating GMF and shaft speeds

To accurately determine the source of gear problems identified by vibration analysis, it is essential to be able to calculate gearmesh frequencies as well as determine intermediate shaft speeds by calculations based on gearmesh frequency and number of teeth on a gear. Worn or misaligned gears show vibration at gearmesh frequency and its harmonics with 1X rpm sidebands, therefore, this section focuses on determining these frequencies. **Figure 1** shows the internal gears and shafts of a differential gearbox.

Gearmesh frequency is calculated by multiplying the number of teeth on a gear by the rotational speed of the shaft to which that gear is attached.

GMF = No. teeth on gear x shaft rpm *(1)*

Example [see Fig. 1 *(2)*]

Line shaft rpm = 962 (this is known)

Number of teeth on line shaft gear = 76

$GMF_{\text{Line shaft gear}}$ = 962 x 76 = 73,112 cycles per minute (cpm)

A meshed gear's shaft speed is calculated on the factual basis that mating gears have the same GMF. Since the first outer differential gear mates with the line shaft gear, the two gears have the same GMF (73,112 cpm). Therefore, the first outer differential gear speed can be calculated if the number of teeth on the first outer differential gear is known (76 teeth in this case).

Shaft rpm = GMF/Number of teeth on gear

Example (see Fig. 1)

GMF = 73,112 cpm

Number of teeth on first outer differential gear = 76

Differential gear rpm = 73,112 cpm/76 teeth = 962 rpm.

Ignoring the differential cluster for now, note from Fig. 1 that the

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PIV (positive infinitely variable) input gear is also in mesh with the first outer differential gear. Therefore, it will also have the same GMF (73,112 cpm) and the PIV input shaft rpm can be calculated since it is known that the PIV input gear has 89 teeth:

PIV input shaft rpm = 73,112 cpm/89 teeth = 821 rpm

Assuming a 1% reduction in the PIV unit (the PIV unit will be discussed in detail in the following section), the PIV output shaft speed would be 821 rpm x 0.99 = 813 rpm. Now, the GMF for the PIV output gear can be calculated since there are 19 teeth on this gear:

$GMF_{\text{PIV output gear}}$ = 813 rpm x 19 teeth = 15,447 cpm

The PIV output gear mates with the 1st primary variable gear (51 teeth). Therefore, the primary variable shaft rpm can be calculated as follows:

Primary variable rpm = 15,447 cpm/51 teeth = 303 rpm

Note, from Fig. 1, that there is a second gear, 2nd primary variable gear (20 teeth), also attached to the primary variable shaft. Even though the 2nd primary variable gear is on the same shaft as the 1st primary variable gear, it will have its own GMF:

$GMF_{\text{2nd primary variable gear}}$ = 303 rpm x 20 teeth = 6060 cpm

The balance of the gear train computations are as follows:

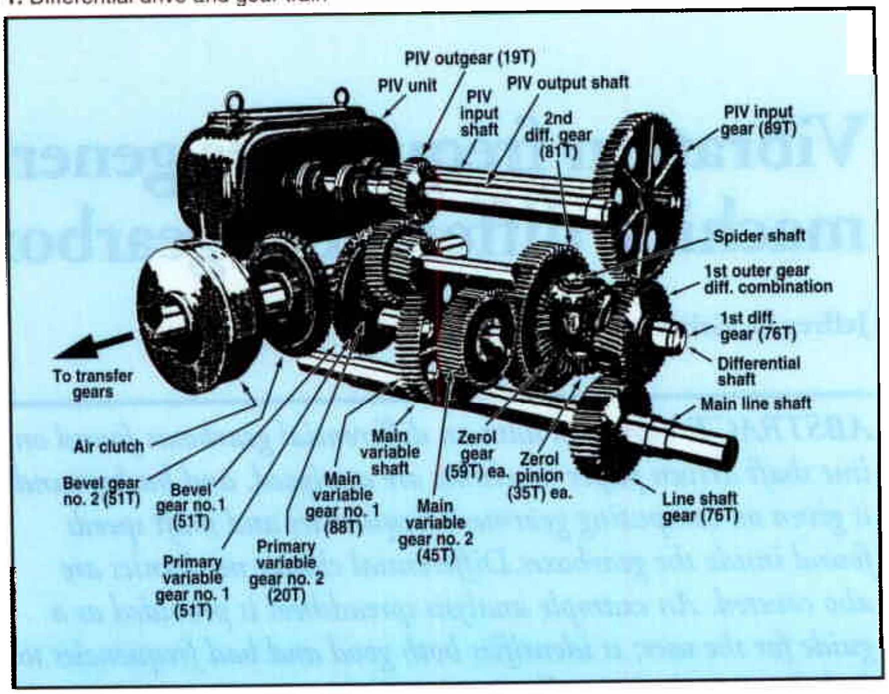
Main variable rpm = 6060 cpm/88 teeth = 68.86 rpm

$GMF_{\text{2nd main variable gear}}$ = 68.86 rpm x 45 teeth = 3099 cpm

2nd Differential gear rpm = 3099 cpm/81 teeth = 38 rpm

This approach is used throughout any gear train to determine GMF and shaft speeds. See the following sections on PIV unit and differential cluster to gain a full understanding of the differential drive and gear unit.

1. Differential drive and gear train



PIV unit

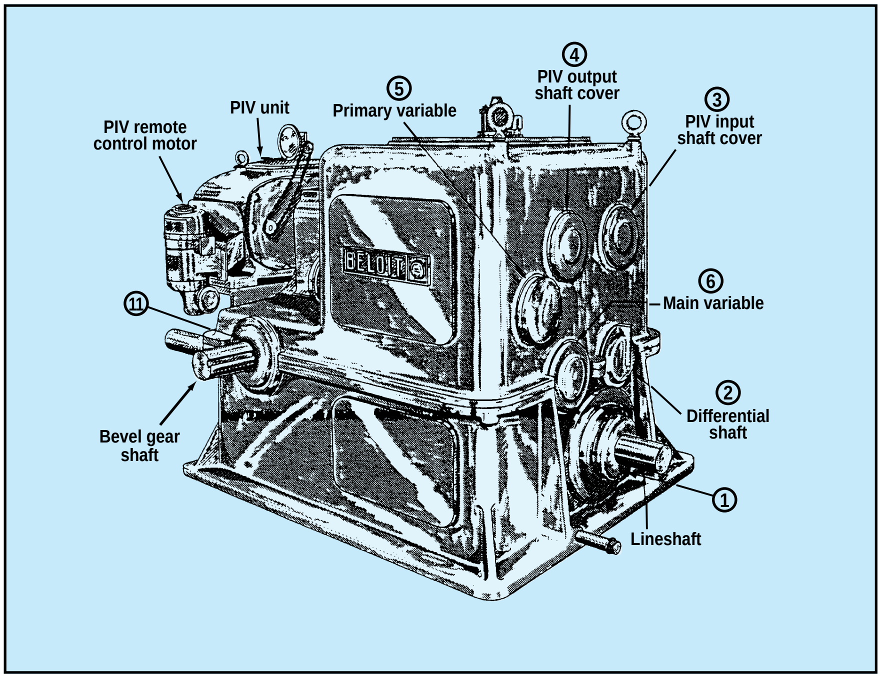
The PIV unit's purpose is to allow small adjustments to the output speed of the gearbox by slightly altering the speed of the 2nd differential gear (see differential cluster section). This allows the process personnel to control the amount of draw on the paper between sections of the paper machine. At this time, no attempt to perform "predictive maintenance" on this unit will be conducted since there are many different mechanical types of PIV units in use. One experienced analyst *(3)* believes it is very difficult to monitor these units with vibration analysis and that preventive maintenance is the best course of action with these units. See the "Owner's Manual" for the PIV unit in your plant to adequately perform the proper techniques of preventive maintenance to keep the PIV unit in the best possible condition and allow the longest possible uninterrupted service.

It is very important that the analyst tach or strobe the input and output shafts of the PIV unit to be able to determine successfully the subsequent gear and shaft rpms in the gearbox as well as the gearbox output speed. In other words, the PIV reduction (or increase) factor must be known for accurate analysis.

Differential cluster

We begin this section with a discussion of how a differential cluster functions and then discuss the calculation of shaft and gear speeds, as well as the expected gearmesh and bearing defect frequencies. The differential cluster in Fig. 1 is made up of the 1st outer differential gear, 1st zerol gear, 2nd zerol pinions, 2nd zerol gear, 2nd outer differential gear, spider shaft, and the differential shaft. The differential cluster follows the theory of beveled differentials without spirals. The first and second outer differential gears are driven by the gears that mate with them. Neither of these gears are attached to the differential shaft, as may be inferred from Fig. 1. They turn freely on two internal bearings that ride on the differential shaft, zerol gears (1 and 2), are permanently attached to and turn at the same speed as the first and second outer differential gears. The zerol gears are not attached to the differ-

2. Non-PIV side of differential gear case



ential shaft either. The combination of the 1st outer differential gear and the 1st zerol gear is called the 1st differential combination, and similarly for the 2nd differential gear and 2nd zerol gear. The zerol gears mate with two zerol pinions mounted in between them. The zerol pinions are mounted on bearings on the spider shaft and are therefore allowed to turn freely around the spider shaft without turning the spider shaft about its longitudinal axis. The spider shaft is mounted to the differential shaft in a perpendicular direction. When there is a net difference in the speed of the 1st and 2nd differential combinations, it will cause the zerol pinions and the spider shaft to rotate about the differential shaft. Since the spider shaft is directly attached to the differential shaft, the differential shaft will rotate at this speed also.

Now we will focus on calculating the frequencies (shaft speeds, gear speeds, gearmesh frequencies, etc.) that are generated by the differential cluster. Assuming that we know the gear teeth counts and the PIV input to output ratio, the 1st and 2nd differential combination speeds can then be calculated. For purposes of example calculations, the 1st differential combination will be turning at 962 rpm and the 2nd differential combination will be turning at 38 rpm. The equation for calculating rotational speeds in a differential cluster is as follows *(4)*:

$$\omega_a \pm \omega_b = 2\omega_{arm}$$

where

ω_a = 1st differential combination

ω_b = 2nd differential combination

ω_{arm} = speed at which the spider shaft is spinning.

Note that the direction in which each gear is turning is important to the calculation of the final speed of the arm (spinning of the spider shaft). If the two gear combinations (1 and 2) are turning in the same direction, their speeds will be added together and then divided by 2 to determine the final speed at which the spider shaft is spinning. This is equal to the rotational speed of the differential shaft since it is attached to the spider shaft. However, if one of the differential combinations is turning in the other direction from the other, then the speeds would be subtracted from each other and then divided by 2 to determine the final differential shaft speed. Note that the direction in which the differential shaft will be rotating will be the direction of the fastest turning differential combination.

In the example, from Fig. 1, the two differential combinations are turning in the same direction, therefore:

$$\omega_{arm} = (968 + 38)/2 = 500 \text{ rpm}$$

Since there are bearings in the zerol pinions, it is necessary to be able to calculate their rotational speeds to determine the location of bearing defect frequencies in a spectrum. The speed at which the zerol pinions turn is equal to the algebraic difference between the zerol gears multiplied by the number of teeth on the 1st zerol gear to obtain the zerol gear and pinion mesh frequencies. The 1st zerol gearmesh is then divided by the number of teeth on the zerol pinion to obtain the speed at which the zerol pinions rotate about their own spider shaft.

$$\omega_{zerol\ pinions} = \omega_a \pm \omega_b \text{ x (Number of teeth on 1st zerol gear)/(Number of teeth on the zerol pinion)}$$

where

ω_a = 1st differential combination speed

ω_b = 2nd differential combination speed.

Note: The algebraic difference is used in this equation. Therefore, if the differential combinations are rotating in the same direction their speeds are subtracted.

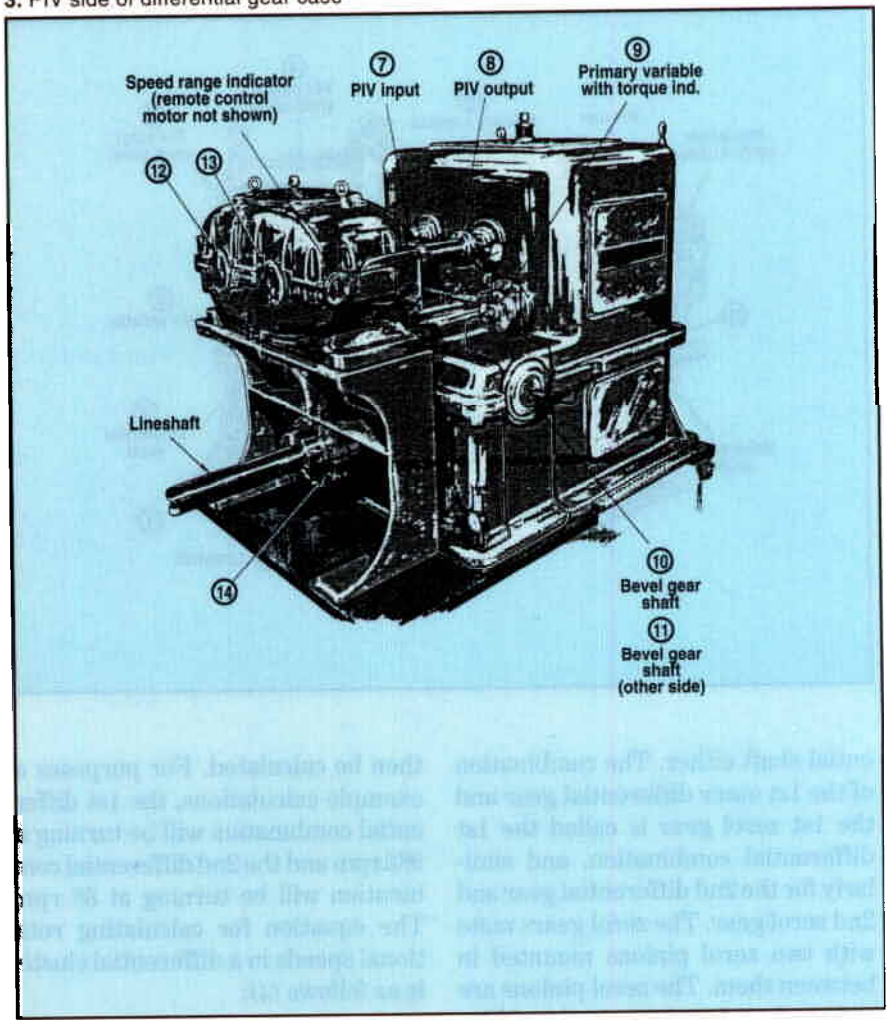
$$\omega_{zerol\ pinions} = [(962 \text{ rpm} - 38 \text{ rpm}) \text{ x } 58 \text{ teeth}]/35 \text{ teeth} = 1531 \text{ rpm}$$

Data collection locations

As with most rotating equipment, the best place to acquire data from the differential gearbox is as close to the bearings as possible since vibration travels from its source through the shaft and out through the bearings to the foundation. **Figure 2** *(2)* is the differential gearbox showing it from a view of the non-PIV end. Positions 1 through 6 have been designated on this end of the gearbox at each accessible bearing housing. Note that position 11, the drive to the transfer case, is also shown in this view. **Figure 3** *(2)* shows the gearbox from the PIV end and positions 7 through 14 are visible on this view. These position numbers will be referenced in **Table I**.

In addition to vibration measurements, certain shaft speeds should be documented in the field during data collection using either a strobe light or tachometer. The line shaft speed must be measured to have a starting point for the calculations. The PIV input shaft can be calculated using the earlier mentioned methods. However, it is wise to also record the PIV input shaft speed to verify the gear tooth counts in the gearbox as well as the shaft speed calculations. The PIV output must be measured since it is difficult to know the speed increase or reduction setting across the PIV unit. The PIV output speed will be used to determine the shaft speeds and gearmesh frequencies associated with the primary and main variable as well as the 2nd differential combination. At a minimum, the lineshaft and the PIV output must be measured to calculate all gearmesh, shaft speed, and bearing frequencies.

3. PIV side of differential gear case



Differential gearbox frequency analysis

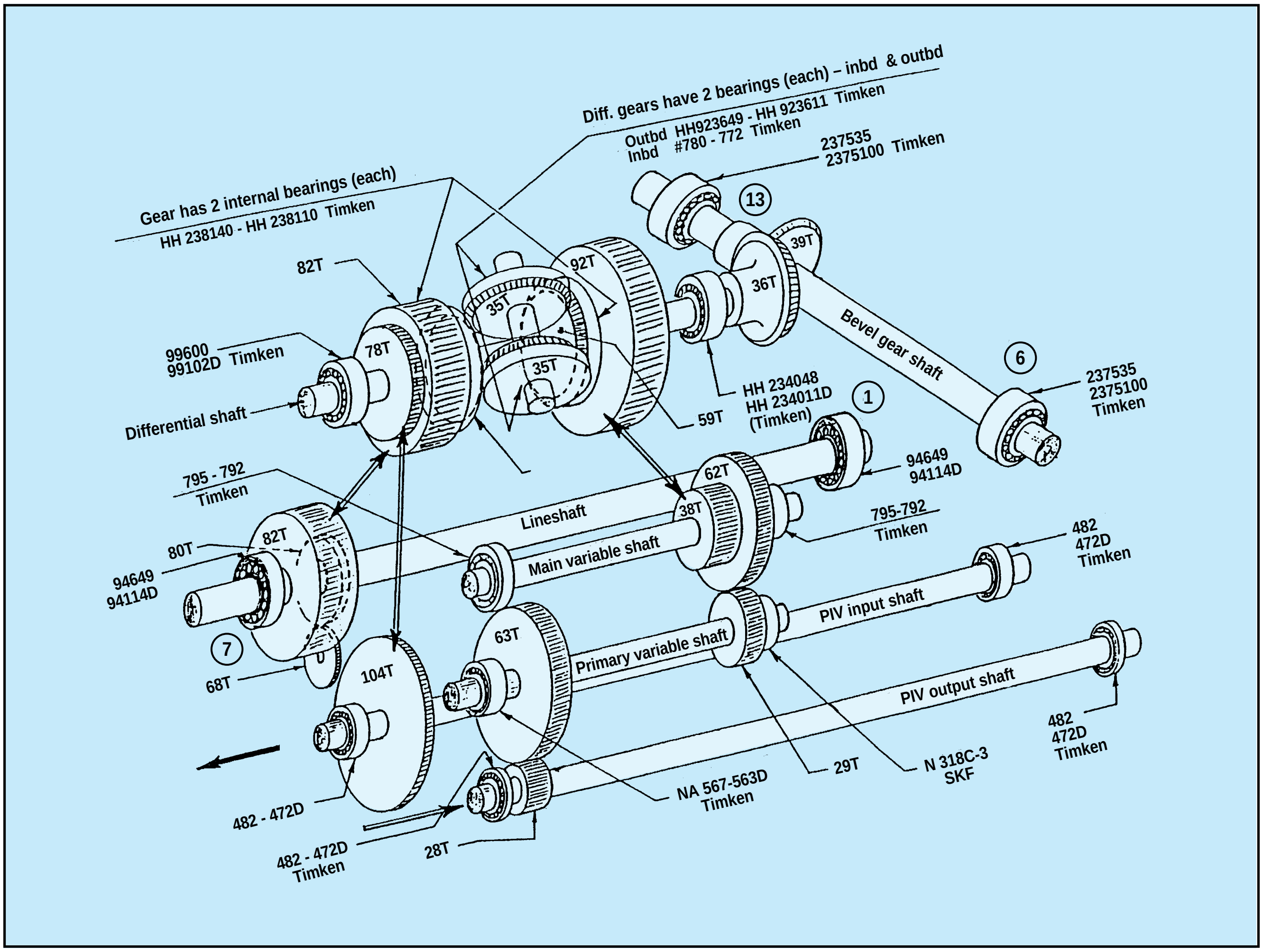
Many vibration frequencies may be generated by the paper machine differential gearbox. In order to properly analyze the "peaks" in the vibration spectra a "spreadsheet" should be developed for each differential gearbox being analyzed. Table I is a guide to be used by the analyst to create a spreadsheet specific to each gearbox under analysis. The rpms, gear tooth counts, and bearings listed in Table I are meant only as an example and do not represent any particular differential gearbox in use on a paper machine. The analyst must determine the exact drive system, gear tooth counts, shaft rpms, and bearing numbers to create a new spreadsheet particular to each different differential gearbox for proper analysis. The "shaded" areas of Table I indicate the information required to be obtained by an analyst for each differential gearbox under analysis.

"Good" and "bad" frequencies

Table I lists frequencies that may be generated by the differential gearbox, as well as the harmonics of the fundamental frequency. The analyst can expect to see frequencies such as shaft speed, gearmesh (GMF), 1 or 2 harmonics of each of these, as well as shaft speed sidebands around GMF and GMF harmonics under acceptable operating conditions. However, the appearance of other frequencies such as bearing defect frequencies or multiple families of shaft speed sidebands around GMF and GMF harmonics indicates problems in the gearbox. Sidebands of the listed frequencies are not included in this table, however, analysis of the number of sidebands as well as a trend of sideband amplitudes are essential to determine properly the machine condition.

Due to the number of frequencies generated by the differential gearbox, and the fact that low frequency vibration can travel through many mechanical components, the table has been broken into position numbers for analysis purposes. This allows the analyst to focus on the

4. Locations of bearings on shafts and gears



mechanical components that are closest to each bearing measurement location without "muddying" the analysis by trying to identify every frequency that occurs at every data collection position. For a description of the data collection points, see the section on data collection locations above.

Bearings

It is important for the analyst to identify all of the bearings for each differential gearbox under analysis. **Figure 4** illustrates common locations where bearings are mounted on the shafts and gears. Note that two bearings are sometimes used right next to each other in the design of these gearboxes. If necessary, additional bearings should be added to the Table I example spreadsheet to account for additional bearings not included. It is also the analyst's responsibility to acquire the bearing defect coefficients—ball pass frequency inner and outer race (BPFI and BPFO), ball spin frequency (BSF) and fundamental train frequency (FTP)—for each bearing in the gearbox.

A special note about the bearings for the differential and zerol gears must be made since both the inner and outer race of these bearings rotates. The inner races of these bearings are directly attached to the differential shaft and rotate at the speed of the differential shaft. The outer races of these bearings are directly attached to their respective gears. The system characteristics are such that the races always rotate in the same direction; therefore, the net speed that the bearing is rotating is computed by subtracting the gear speed from the differential shaft speed. This net speed is used as 1X rpm in the bearing analysis. Be aware that the typical bearing defect coefficients are based on a stationary outer race and a rotating inner race (a minus sign is used in the BSF and FTF equations). However, the 1st differential and 1st zerol gears have outer races that rotate faster than the inner race resulting in a net outer race rotation with a stationary inner race. In these cases, the sign in the BSF and FTF equations must be changed to a plus for these bearings. The BPFO and BPFI coefficients remain the same. To perform bearing defect calculations, the following equations can be used:

I. Example frequencies generated by a differential gearbox

Pos. No.	*(Source) Component*	*Symbol*	*rpm Mult,*	*1X*	*2X*	*3X*	*4X*	*5X*	*6X*	*7X*	*8X*
1	Line shaft input	LS	1	962	1,924	2,886	3,848	4,810	5,772	6,734	7,696
	Line shaft bearing No,1	BPFI	13.176	12,675	25,351	38,026	50,701	63,377	76,052	88,727	101,402
	Timken 94649	BPFO	10.824	10,413	20,825	31,238	41,651	52,063	62,476	72,889	83,302
	(94000 Series)	BSF	4.863	4,678	9,356	14,035	18,713	23,391	28,069	32,747	37,426
		FTF	0.451	434	868	1,302	1,735	2,169	2,603	3,037	3,471
	Line shaft gear teeth	LSG teeth	76								
	Line shaft gearmesh	GMF (LS)	1	73,112	146,224	219,336	292,448	365,560	438,672	511,784	584,896
2	1st Diff, gear teeth	DG1 teeth	76								
	1st Diff, gear rpm	DG1	1	962	1,924	2,886	3,848	4,810	5,772	6,734	7,696
	1st Diff, gearmesh	GMF (DG1)	1	73,112	146,224	219,336	292,448	365,560	438,672	511,784	584,896
	2nd Diff, gear teeth	DG2 teeth	81								
	2nd Diff, gear rpm	DG2	1	38	77	115	153	191	230	268	306
	2nd Diff, gearmesh	GMF (DG2)	1	3,099	6,197	9,296	12,395	15,493	18,592	21,691	24,789
	Differential shaft rpm	DS	1	500	1,000	1,500	2,001	2,501	3,001	3,501	4,001
	Differential shaft bearing No,1	BPFI	12.81	6,407	12,813	19,220	25,627	32,033	38,440	44,846	51,253
	Timken 99600	BPFO	10.19	5,096	10,193	15,289	20,385	25,482	30,578	35,674	40,770
	(99000 Series)	BSF	8.4	4,201	8,402	12,603	16,804	21,005	25,206	29,408	33,609
		FTF	0.44	220	440	660	880	1,100	1,320	1,540	1,760
	Differential shaft bearing No, 2	BPFI	9.458	4,730	9,460	14,191	18,921	23,651	28,381	33,111	37,842
	Timken HH234048	BPFO	6.542	3,272	6,544	9,816	13,087	16,359	19,631	22,903	26,175
	(HH234000 Series)	BSF	2.609	1,305	2,610	3,914	5,219	6,524	7,829	9,134	10,439
		FTF	0.409	205	409	614	818	1,023	1,227	1,432	1,636
	(1st Diff, gear - Diff, shaft) rpm (for bearing defect calcs)			462							
	1st Diff, gear bearing	BPFI	10.22	4,720	9,441	14,161	18,881	23,602	28,322	33,042	37,763
	Timken HH238140	BPFO	7.78	3,593	7,187	10,780	14,373	17,967	21,560	25,154	28,747
	(H238100 Series)	BSF		0	0	0	0	0	0	0	0
		FTF		0	0	0	0	0	0	0	0
	(2nd Diff, gear - Diff, shaft) rpm (for bearing defect calcs)			462							
	2nd Diff, gear bearing	BPFI	10.22	4,720	9,441	14,161	18,881	23,602	28,322	33,042	37,763
	Timken HH238140	BPFO	7.78	3,593	7,187	10,780	14,373	17,967	21,560	25,154	28,747
	(H238100 Series)	BSF	3.555	1,642	3,284	4,926	6,568	8,210	9,852	11,494	13,136
		FTF	0.432	200	399	599	798	998	1,197	1,397	1,596

Table 1 continued.

Pos. No.	(Source) Component	Symbol	rpm Mult,	1X	2X	3X	4X	5X	6X	7X	8X
	1st Zerol gear rpm	ZG1	1	962	1,924	2,886	3,848	4,810	5,772	6,734	7,696
	(1st Zerol gear - Diff, shaft) rpm (for bearing defect calcs)			462							
	1st Zerol gear bearing	BPFI	10.22	4,720	9,441	14,161	18,881	23,602	28,322	33,042	37,763
	Timken HH238110	BPFO	7.78	3,593	7,187	10,780	14,373	17,967	21,560	25,154	28, 747
	(H238100 Series)	BSF		0	0	0	0	0	0	0	0
		FTF		0	0	0	0	0	0	0	0
	2nd Zerol gear rpm	ZG2	1	38	77	115	153	191	230	268	306
	(2nd Zerol gear - Diff, shaft) rpm (for bearing defect calcs)			462							
	2nd Zerol gear bearing	BPFI	10.22	4,720	9,441	14,161	18,881	23,602	28,322	33,042	37,763
	Timken HH238110	BPFO	7.78	3,593	7,187	10,780	14,373	17,967	21,560	25,154	28, 747
	(H238100 Series)	BSF	3.555	1,642	3,284	4,926	6,568	8,210	9,852	11,494	13,136
		FTF	0.432	200	399	599	798	998	1,197	1,397	1,596
	Zerol gear No, 1 teeth	ZG1 teeth	59								
	Zerol pinion teeth	ZP teeth	35								
	Zerol gear and pinion mesh [(ZG1-ZG2)* (ZG1 teeth)]										
		GMF (ZP & ZG)	1	54,501	109,002	163,503	218,004	272,505	327,006	381,507	436,007
	Zerol pinions rpm	ZP	1	1,557	3,114	4,672	6,229	7,786	9,343	10,900	12,457
	Zerol pinions bearing No, 1	BPFI	8.307	12,935	25,871	38,806	51,742	64,677	77,612	90,548	103,483
	Timken HH923649	BPFO	5.693	8,865	17,730	26,595	35,460	44,325	53,190	62,055	70,920
	(HH923600 Series)	BSF	2.413	3,757	7,515	11,272	15,030	18,787	22,545	26,302	30,060
		FTF	0.407	634	1,268	1,901	2,535	3,169	3,803	4,436	5,070
	Zerol pinions bearing No, 2	BPFI	11.77	18,328	36,656	54,984	73,312	91,639	109,967	128,295	146,623
	Timken 780-772	BPFO	9.23	14,373	28,745	43,118	57,491	71,863	86,236	100,609	114,981
	(778 Series)	BSF	7.92	12,333	24,666	36,998	49,331	61,664	73,997	86,329	98,662
		FTF	0.44	685	1,370	2,055	2,741	3,426	4,111	4,796	5,481
	Bevel gear No, 1 teeth	BG No, 1 teeth	50								
	Bevel gear No, 1 mesh	GMF (BG1)	1	25,006	50,013	75,019	100,026	125,032	150,038	175,045	200,051
3 & 7	PIV input gear teeth	PIVI teeth	89								
	PIV input shaft rpm	PIVI	1	821	1,643	2,464	3,286	4,107	4,929	5,750	6,572
	PIV input gear mesh	GMF (PIVI)	1	73,112	146,224	219,336	292,448	365,560	438,672	511,784	584,896
	PIVI bearing No, 1	BPFI	11.217	9,215	18,429	27,644	36,858	46,073	55,287	64,502	73,717
	Timken 482-472D	BPDO	8.783	7,215	14,430	21,645	28,860	36,075	43,291	50,506	57,721
	(475 Series)	BSF	3.948	3,243	6,486	9,730	12,973	16,216	19,459	22,703	25,946
		FTF	0.439	361	721	1,082	1,443	1,803	2,164	2,524	2,885
	PIVI bearing No, 2	BPFI	11.217	9,215	18,429	27,644	36,858	46,073	55,287	64,502	73,717
	Timken 482-472D	BPDO	8.783	7,215	14,430	21,645	28,860	36,075	43,291	50,506	57,721
	(475 Series)	BSF	3.948	3,243	6,486	9,730	12,973	16,216	19,459	22,703	25,946
		FTF	0.439	361	721	1,082	1,443	1,803	2,164	2,524	2,885

Table 1 continued

Pos. No.	*(Source) Component*	*Symbol*	*rpm Mult,*	*1X*	*2X*	*3X*	*4X*	*5X*	*6X*	*7X*	*8X*
4 & 8	PIV output shaft ratio	Ratio (PIV)	0.99								
	PIV output shaft RPM	PIVO	1	813	1,627	2,440	3,253	4,066	4,880	5,693	6,506
	PIV output gear teeth	PIVO teeth	19								
	PIV output gearmesh	GMF (PIVO)	1	15,452	30,904	46,356	61,808	77,260	92,713	108,165	123,617
	PIVO bearing No, 1	BPFI	11.217	9,122	18,245	27,367	36,490	45,612	54,735	63,857	72,979
	Timken 482-472D	BPFO	8.783	7,143	14,286	21,429	28,572	35,715	42,858	50,001	57,143
	(475 Series)	BSF	3.948	3,211	6,422	9,632	12,843	16,054	19,265	22,475	25,686
		FTF	0.439	357	714	1,071	1,428	1,785	2,142	2,499	2,856
	PIVO bearing No, 2	BPFI	11.217	9,122	18,245	27,367	36,490	45,612	54,735	63,857	72,979
	Timken 482-472D	BPFO	8.783	7,143	14,286	21,429	28,572	35,715	42,858	50,001	57,143
	(475 Series)	BSF	3.948	3,211	6,422	9,632	12,843	16,054	19,265	22,475	25,686
		FTF	0.439	357	714	1,071	1,428	1,785	2,142	2,499	2,856
5 & 9	Primary variable gear No,1 teeth	PV G1 teeth	51								
	Primary variable shaft rpm	PV	1	303	606	909	1,212	1,515	1,818	2,121	2,424
	Primary variable gearmesh No, 1	GMF1 (PV)	1	15,452	30,904	46,356	61,808	77,260	92,713	108,165	123,617
	Primary variable gear No, 2 teeth	PV G2 teeth	20								
	Primary variable gearmesh No, 2	GMF2 (PV)	1	6,060	12,119	18,179	24,239	30,298	36,358	42,418	48,477
	Primary variable bearing No, 1	BPF1	11.295	3,422	6,844	10,267	13,689	17,111	20,533	23,955	27,377
	Timken Na567-5630	BPFO	8.705	2,637	5,275	7,912	10,550	13,187	15,825	18,462	21,100
	(565 Series)	BSF	3.713	1,125	2,250	3,375	4,500	5,626	6,750	7,875	9,000
		FTF	0.435	132	264	395	527	659	791	923	1,054
	Primary variable bearing No, 2	BPFI	7.66	2,321	4,642	6,963	9,283	11,604	13,925	16,246	18,567
	SKF N3 18C-9	BPFO	5.34	1,618	3,236	4,854	6,472	8,090	9,708	11,325	12,943
	(N318 Series)	BSF	2.712	822	1,643	2,465	3,287	4,108	4,930	5,752	6,574
		FTF	0.411	125	249	374	498	623	747	872	996
6	Main variable gear No,1 teeth	MV G1 teeth	88								
	Main variable shaft rpm	MV	1	69	138	207	275	344	413	482	551
	Main variable gear mesh No, 1	GMF1 (MV)	1	6,060	12,119	18,179	24,239	30,298	36,358	42,418	48,477
	Main variable gear No, 2 teeth	MV G2 teeth	45								
	Main variable gear mesh No, 2	GMF2 (MV)	1	3,099	6,197	9,296	12,395	15,493	18,592	21,691	24,789
	Main variable bearing No, 1	BPFI	12.65	871	1,742	2,613	3,484	4,355	5,226	6,098	6,969
	Timken 796-792	BPFO	10.35	713	1,425	2,138	2,851	3,563	4,276	4,989	5,702
	(795 Series)	BSF	9.52	656	1,311	1,967	2,622	3,278	3,933	4,589	5,244
		FTF	0.45	31	62	93	124	155	186	217	247
	Main variable bearing No, 2	BPFI	12.65	871	1,742	2,613	3,484	4,355	5,226	6,098	6,969
	Timken 795-792	BPFO	10.35	713	1,425	2,138	2,851	3,563	4,276	4,989	5,702
	(795 Series)	BSF	9.52	656	1,311	1,967	2,622	3,278	3,933	4,589	5,244
		FTF	0.45	31	62	93	124	155	186	217	248
7	PIV input (see posn 3)										
8	PIV ouput (see posn 4)										
9	Primary variable (see posn 5)										

Table 1 continued

Pos No,	(Source) Component	Symbol	rpm Mult,	1X	2X	3X	4X	5X	6X	7X	8X
10	Output shaft (tending side)										
	Bevel gear No, 2 teeth	BG No, 2 teeth	50								
	Output shaft rpm	OS rpm	1	500	1,000	1,500	2,001	2,501	3,001	3,501	4,001
	Bevel gear No, 2 mesh	GMF (BG No, 2)	1	25,006	50,013	75,019	100,026	125,032	150,038	175,045	200,051
	Output shaft bearing No, 1	BPFI	11.798	5,901	11,801	17,702	23,602	29,503	34,403	41,304	47,204
	Timken 237536	BPFO	9.202	4,602	9,204	13,807	18,409	23,011	27,613	32,215	36,817
	(HM23750D Series)	BSF	3.915	1,958	3,916	5,874	7,832	9,790	11,748	13,706	15,664
		FTF	0.438	219	438	657	876	1,095	1,314	1,533	1,752
	Output shaft bearing No, 2	BPFI	11.798	5,901	11,801	17,702	23,602	29,503	35,403	41,304	47,204
	Timken 237535	BPFO	9.202	4,602	9,204	13,807	18,409	23,011	27,613	32,215	36,817
	(HM237500 Series)	BSF	3.915	1,958	3,916	5,874	7,832	9,790	11,748	13,706	15,664
		FTF	0.438	219	438	657	876	1,095	1,314	1,533	1,752
11	Output shaft (drive side)										
	(see posn 10)										
12	PIV No, 1			(bearings unknown at this time)							
13	PIV No, 2			(bearings unknown at this time)							
14	Line shaft output										
	(see posn 1)										

$$BPFI = N_b/2[1 = (B_d/P_d)\cos\theta] \text{ x rpm}$$

$$BPFO = N_b/2[1 - (B_d/P_d)\cos\theta] \text{ x rpm}$$

$$BSF = P_d/2B_d\{1 - [(B_d/P_d)\cos\theta]^2\} \text{ x rpm}$$

$$FTF = 1/2[1 - (B_d/P_d)\cos\theta] \text{ x rpm}$$

where

N_b = Number of balls or rollers

B_d = Ball or roller diameter, in. or mm

P_d = Bearing pitch diameter, in. or mm

θ = Contact angle, degrees.

Gearbox condition

The evaluation of the gear, shaft, and bearing condition for differential gearbox drives follows the standard vibration “rules” for these components. These “rules” are outside the scope of this paper. However, they may be found in a variety of sources or by contacting Technical Associates of Charlotte for literature pertaining to vibration frequency analysis *(1)*.

What else is there to monitoring differential gearboxes?

There are other factors, beyond the scope of this paper, with which the analyst must become familiar to effectively monitor differential gearbox drives. They include: low-speed analysis, as the primary and main variable shafts can operate below 100 cpm, the proper low-speed instrumentation, setting up alarm levels and narrowband alarms, time waveform analysis to detect cracked or broken gear teeth, the proper data collection parameters, such as maximum frequency (FMAX), lines of resolution, and vibration units, and the general set of vibration frequency analysis “rules” mentioned earlier.

Conclusions

There are pitfalls that may be encountered by a vibration analyst when diagnosing data collected from paper machine differential gearbox drives if the analyst is unfamiliar with the calculation of expected frequencies from these gearboxes. The PIV unit is set such that a reduction or increase between the incoming shaft speed and the outgoing shaft speed occurs. This speed change must be tached or strobed to accurately calculate the resulting shaft, gear, and bearing frequencies. The differential gear cluster and differential shaft work in a unique fashion that does not follow the standard rules for calculating shaft speed reductions or increases based on the common gearmesh calculations. Also, the bearings that support the differential cluster gears have both inner and outer race rotation, thereby resulting in bearing defect frequencies based on the net difference between the speed of the these two races, as well as factoring in a change in the bearing defect equations depending on whether the inner race rotates faster than the outer race. This paper has provided the analyst with a method to identify the spectral frequencies that can appear in vibration analysis of differential gearbox drives. There are other important factors such as data collection instrumentation and vibration analysis procedures that must be used by the analyst to effectively evaluate the gearbox condition.

The author would appreciate any additional information on particular problems or observations that the reader may have encountered in the course of monitoring differential gearbox drives.

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