On the Relation Between Operating Conditions and Changes in Vibration Signature: A Case Study in Paper Mill

Basim Al-Najjar
Inst. for Industrial Engineering, Lund University,
Box 118, S-221 00 Lund, Sweden
Inst. for Mathematics, Statistic and Computer Science, Växjö University,
351 95 Växjö, Sweden
e-mail:Basim.Al-Najjar@masda.hv.se

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Abstract: To achieve effective diagnosis and prognosis, relevant and reliable data from the surroundings are required in addition to the vibration measurements. The paper classifies stoppage times and highlights the reasons behind them. Also, it discusses the consequential economic losses incurred by unplanned stoppages. Further, it presents a new approach to envelope alarming, which is called dynamic alarm. It is applicable to identical bearings and those of approximately identical vibration signatures. One of the important conclusions from this study is that it is not only the variation in the machine loading which affects the amplitudes of the bearing defect frequencies, but also the machine speed does. Therefore, changes in the machine speed and load should be considered when interpreting vibration spectra. This will improve the effectiveness of vibration diagnosis and prediction of the time to replacement. During the period covered by this study, which is fifty eight days, it was found that the total stoppage time due to unknown reasons was very large and caused appreciable economic losses of about 2,3 millions SEK. The dynamic alarm is shown theoretically to offer later renewal with fewer failures, and therefore lower cost and higher productivity.

Key Words: Operating conditions, unplanned stoppages, dynamic alarm, vibration monitoring, diagnosis.

Introduction: The vibration spectrum consists of many frequencies of different amplitudes. These frequencies are usually generated by failure causes such as imbalance, misalignment, damage in rolling element bearings or in gear boxes, natural frequencies of the machine parts, etc. [1]. Identification of these frequencies leads to identify the basic causes behind deterioration while damage severity is usually assessed by the amplitude of the considered frequency(s) or frequency band(s). But, the vibration signature is usually changed due to the effect of many factors especially machine operating conditions such as load and speed. Acquisition of better data coverage and quality, which describe these factors in addition to the vibration measurements, eases the task of the maintenance engineer to perform effective diagnosis and prognosis. Frequencies are specific to the bearing but the surrounding structure damps some more than others and occasionally may resonate. The structure responds to the actual frequencies so if machine rotational speed and/or load changes the response spectrum may be quite different. Bearing defect vibration frequencies usually deviate from their theoretical values according to many factors such as faulty installation and operating conditions. These deviations should be considered when interpreting bearing vibration spectra because frequency shift confuses the analyst [2,3]. Therefore, it is important to keep numerate records of the operating conditions such as machine load, speed and operating temperature to distinguish and confirm the reasons behind changes in the vibration signature. In this paper, data and information concern stoppage categories and times, causes behind these stoppages and total stoppage time gathered from a Swedish paper company are presented. The analysis and results of these data are discussed. The concept of the dynamic alarm is presented and explained by an example which is followed by conclusions.

Data Gathering: The paper mill company under study has 4 paper mill machines of different ages. One paper machine, called PM1, was selected for investigation because its database includes more replacements of identical bearings than the other machines' databases. Data from the production and maintenance departments during two months are used in this study. Vibration was the main parameter used for monitoring the machine. The vibration measurements were collected using Microlog and Presim^2 software. The measurements were all in mm/s, RMS. Vibration measurements of ten identical replaced spherical roller bearings of type 23228ck/SKF which are usually used at the driven side of the leading roller of drying cylinders in PM1 formed part of the data. There were no enough replacements from other types to be included. According to the personnel experience, the most troublesome area in the machine was the drying cylinder group. This is why these bearings were selected. The vibration measurements made at these ten bearing were few and done in three directions vertical, horizontal and axial. The measurements in the axial direction are analysed only because they represented bearing condition much better than the measurements in the other directions due to high stiffness and more informative spectra.

The vibration measurements, which cover the period under study (940901-941231), made at these roller bearings are collected from the machine database. The measurements are done only once/day at 940901, 940926, 941013, 941027, 941124, 941208, 941222. Records of machine operating conditions, which were usually registered 2-3 times/day and were reported manually in formal tables, are also gathered for the same period. The data which were formally considered in these reports were: Machine operating speed and load in specific parts of PM1 such as press and drying cylinder groups, number of stoppages and their times, reasons behind stoppages and the machine part where the failure happened, (if they were known). Paper quality and contents were also included. There is small risk for uncertainty in identifying the reasons, i.e. problem area, behind the short stoppages. This uncertainty is due to the deficiencies in the techniques used for detecting and localising any cut in the paper. They were using special lighting system connected to particular sensors which were installed along the paper machine to detect and localise if there is any cut in the paper. Sometimes, it is possible that the operator misses when the light was first activated and consequently the opportunity to identify where the cut is actually happened may be missed.

In order to distinguish the changes in vibration spectra, only the bearings which acquired two or more vibration measurements during the period 940901-941031 are considered significant for the study. The 3rd and 4th drying cylinder groups in the paper machine PM1 were considered for more analysis because they were identified by the company as parts of problem areas. The vibration measurements were picked up at the bearing houses of the driven side, (DS), of the lead rollers, (LR), 041, 056, 059, 064, 065, 072, 075, 080 and, 085 and 095, which lie in the third and fourth drying cylinder groups, respectively. Notice that not all the ten bearings at the driven sides of these LRs could be considered in the analysis because there were no enough vibration measurements. At the bearings where more than one measurement were made during the period under analysis, data about the following are gathered from these measurements, see Tables 1, 2 and 3:

1. Machine speeds recorded by both the vibration technician and operator.
2. Amplitudes of the first multiple (1*X), second multiple, (2*X) and third multiple, (3*X) of the machine speed (in revolution per second), where X and i*. denote the machine rotational frequency and the ith multiple of the frequency, respectively.
3. Amplitudes of the frequencies at the range 15*X - 36*X. Frequencies at this range are, in general, correlated to the changes in the bearing condition [4].
4. The vibration levels which are usually indicated in the commercial vibration-based monitoring programs by one or more of the following:
   - Overall RMS vibration level, to represent the vibration energy in the whole vibration signal.
   - Synchronic RMS vibration level, i.e. the vibration energy in the harmonics to the machine speed frequency.
   - Sub-synchronic RMS vibration level, i.e. the vibration energy in the frequencies less than the machine speed frequency.
   - Non-synchronic RMS vibration level, i.e. the vibration energy in the frequencies which are not harmonics to the machine speed frequency.
5. The load at the third and fourth drying cylinder groups.

Analysis: Data from two types of the above mentioned reports, called report 1 and 2, covered the period 940901 - 941031, are analysed. The analysis is limited to this period due to the lack of more data. The variation in the machine rotational speed at the 3rd and 4th drying cylinder groups during the period under study is explained by the data plotted in Fig. 1. The variation in the machine load at these two cylinder groups, during the same period, is explained in Fig. 2. In reports 1 and 2, the stoppage time is divided into four categories based on the reasons behind stoppages, which are; unknown reasons, (O), miscellaneous, (T), shift in the tambour, (S) and slime, (K). The total stoppage time was about 4593 minutes. The major part of it, about 3980 minutes which is 87% of the total stoppage time, was actually because of unknown reasons, see Fig. 3. The next longest stoppage time, about 475 minutes, i.e. 10% of the total stoppage time, accumulated due to miscellaneous. The latter was not defined in the reports and eventually can be added to the unknown reasons. The machine part behind stoppages are classified into eight areas which are: the 2nd press

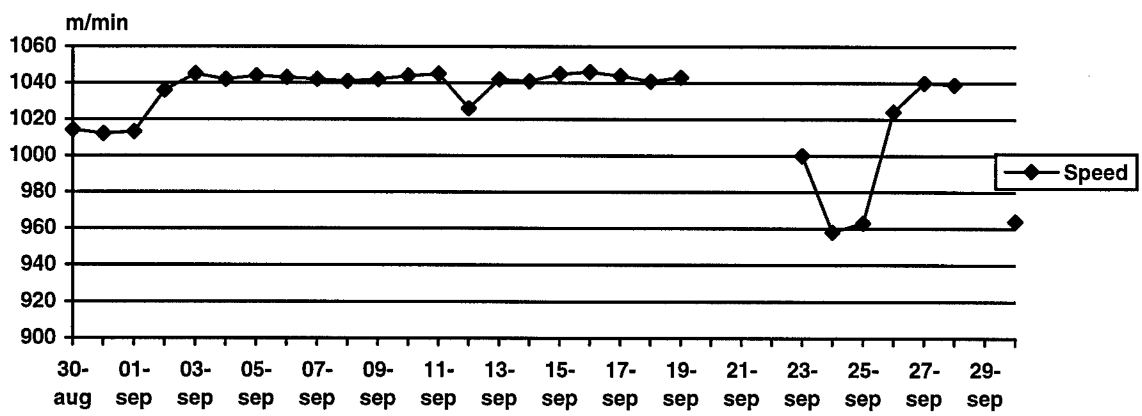

Fig. 1. The variation of the machine speed at the third and fourth drying cylinder groups, PM1. The discontinuity in the curve is due to lack of data at some days.

cylinder group, 3rd press cylinder group, 1st drying cylinder group, 2nd drying cylinder group, 3rd drying cylinder group, 4th drying cylinder group, calender and pope, see Fig. 4. The plot shown in Fig. 4 displays that the stoppage times accumulated due to failures in the 3rd press cylinder group, 1st and 2nd drying cylinder groups and Calender are; 1370, 1120, 786 and

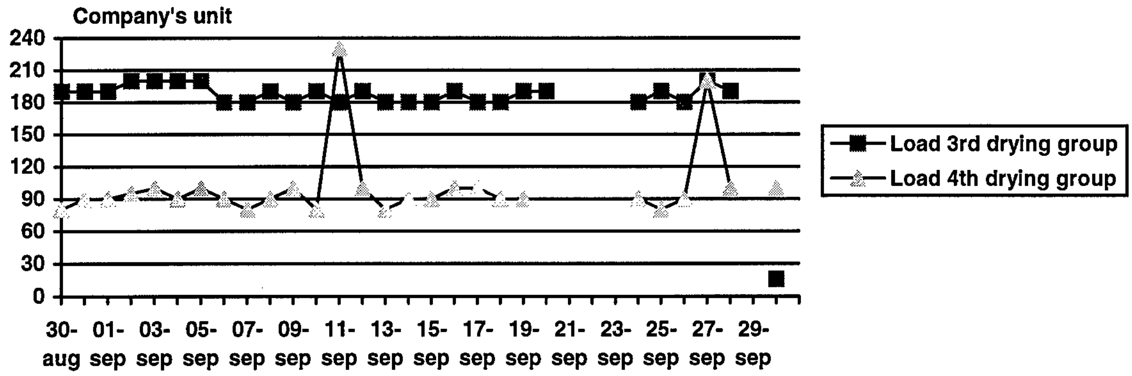

Fig. 2. Variations of the machine load at the 3rd and 4th drying cylinder groups, PM1. The discontinuity in the curve is due to lack of data at some days.

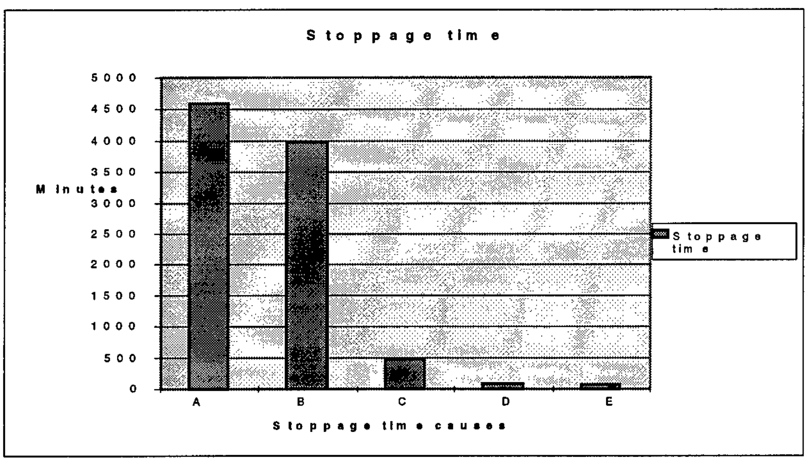

Fig. 3. Total stoppage time, (A), stoppage time due to unknown reasons, (B), stoppage time due to miscellaneous, (C), stoppage time due to shift in the tambour, (D), stoppage time due to slime, (E).

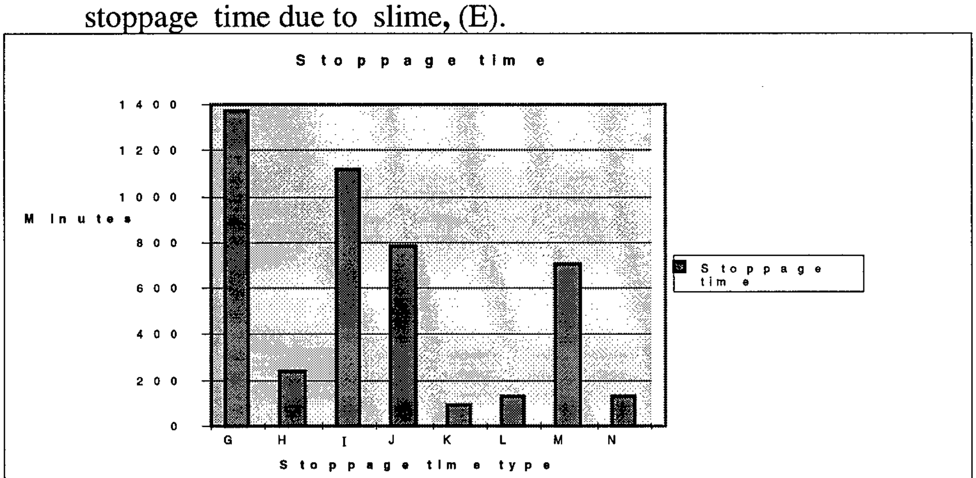

Fig. 4. Stoppage time due to; failures in the 3rd cylinder group, (G), failures in the 3rd press cylinder group, (H), failures in the 1st drying cylinder group, (I), failures in the 2nd drying cylinder group, (J), failures in the 3rd drying cylinder group, (K), failures in the 4th drying cylinder group, (L), failures in calender, (M), failures in pope, (N).

709 minutes, respectively. The number of stoppages experienced by PM1 during this period was about 490 stoppages. It is obvious that there exist distinguishable differences between the machine speeds measured by the operator and that recorded by the vibration technician, see Tables 1, 2 and 3. The minimum differences, about 0 and + 0.02 revolution per second, (rps), were experienced in 941013 and 941222, respectively, and the maximum differences, about 0.85 and 0.75 rps, were in 940901 and 941208, respectively.

The time of the vibration measurement.	941013, 07:13	941027, 06:10
Machine speed measured by vibration technician	547 rpm = 9.12 rps	510 rpm = 8.5 rps
Amplitudes of 1*X, 2*X, 3*X, 4*X, 6*X & 7*X and 8*X	0.31, 1.45, ----, ----, ---- & ---- and 0.17 mm/s	0.24, 1.14, ----, 0.55, ---- & ---- and ----, mm/s
Amplitudes of the frequencies within the range	(194-316) Hz, were 0.17-0.32 mm/s	and (216-312) Hz, were 0.13-0.25 mm/s
Overall RMS	2.04 mm/s	1.71 mm/s
Synchronic	1.88 mm/s	1.61 mm/s
Sub-Synchronic	0.10 mm/s	0.10 mm/s
Non-synchronic	0.79 mm/s	0.57 mm/s
The load at the drying cylinder group	200 company's unit	200 company's unit
Rotational speed according to the operator where the diameter of the cylinder = 605 mm	1040 m/min. = 9.12 rps 941013, 06:00	1006 m/min. = 8.82 rps 941027, 06:00
Differences in rps: (speed measured by vibration technician - speed measured by operator)	0 rps	- 0.32 rps

Table 1. The bearing at the DS of the LR 56 in the axial direction, LR 056 DS A.

The time of the vibration measurement.	941013, 07:52	941222, 09:37
Machine speed according to the vibration technician	547 rpm = 9.12 rps	539 rpm = 8.99 rps
Amplitudes of 2*FTF, FTF: Fundamental train frequency 2*X+2*FTF 1*BPFI, 2*BPFI, BPFI: Ball pass frequency inner. 3*BPFO, BPFO: Ball pass frequency outer. 2*BPFI+2*FTF 3*BPFO+2*FTF 8*BSF, BSF: Ball spin frequency.	0.25, 0.18, 0.36, 0.15, 0.14, ----, ---- , ----- mm/s	---, ---, ---, 1.83, 11.61, 1.13, 1.38, 2.15 mm/s
Amplitudes of 1*X 2*X 16.5*X 20.5*X	0.25, 1.5 , ----, ---- mm/s	1.1, ----, 1.4, 0.97 mm/s
Amplitudes of the frequencies in the range	(100-466 Hz), were about 0.14-0.30 mm/s	(100-440 Hz) were about 0.8- 11.6 mm/s
Overall RMS	1.98 mm/s	13.32 mm/s
Synchronic	1.86 mm/s	8.85 mm/s
Sub-Synchronic	0.12 mm/s	0 mm/s
Non-synchronic	0.66 mm/s	9.95 mm/s
The load at the drying cylinder group	200 company's unit	200 company's unit
Rotational speed according to the operator where the diameter of the cylinder = 605 mm	1040 m/min. = 9.12 rps 941013, 06:00	1023 m/min. = 8.97 rps 941222, 06:00
Differences in the speed: (speed measured by vibration technician - speed measured by operator)	0 rps	+ 0.02 rps

Table 2. The bearing at the DS of the LR 75 in the axial direction, LR 075 DS A.

The differences in the speed measurements were positive, i.e. the speed recorded by the vibration technician was higher than that registered by the operator. But, the measurement made in 941027 at LR 056 was -0.32 rps. All the frequencies specified in Table 1 acquired higher amplitudes at the higher machine speed independent of the machine load. This is also true in Table 3 when it concerns bearing defect frequencies, e.g. Fundamental Train Frequency, (FTF). The literature was found mainly to confirm this analysis see for example [1]. This is because the vibration levels due to, e.g. imbalance and misalignment are usually proportional to the machine speed. From Table 2, it obvious that high frequencies such as 16.5*X and 20.5*X acquired much higher amplitudes at 941222. The increment in the overall RMS vibration level is obviously due to the appreciable increment in the amplitudes of the synchronic and non-synchronic frequencies. All together may interpreted as indications of bearing defects which led finally to replace the bearing at 941226. The machine load was the same at the two opportunities. The differences in the recorded speed registered by the operator and vibration technicians are negligible.

The time of the vibration measurement.	940901, 07:01	941208, 15:28
Machine speed according to the vibration technician	584 rpm = 9.73 rps	548 rpm = 9.13 rps
Amplitudes of	2* FTF = 1.96 mm/s	2*FTF = 0.55 mm/s
Amplitudes of the frequencies	100-300 Hz are about 0.16-0.6 mm/s	100-560 Hz are about 0.17-0.6 mm/s
Overall RMS	1.92 mm/s	2.53 mm/s
Synchronic	1.62 mm/s	2.45 mm/s
Sub-Synchronic	0 mm/s	0 mm/s
Non-synchronic	1.03 mm/s	0.64 mm/s
The load at the drying cylinder groups	190 company's unit	180 company's unit
Rotational speed according to the operator where the diameter of the cylinder = 605 mm	1013 m/min. = 8.88 rps 940901, 06:00	957 m/min. = 8.38 rps 941208, 06:00
Differences in the speed: (speed measured by vibration technician - speed measured by operator)	+ 0.85 rps	+ 0.75 rps

Table 3. The bearing at the DS of the LR 67 in the axial direction, LR 067 DS A.

Results and Discussions: The plot of the stoppage time accumulated due to failures in the machine parts specified, see Fig. 4, reveals that the problem areas ranked according to their economic importance are the 2nd press cylinder group, 1st and 2nd drying cylinder groups and Calender. The problem areas distinguished by this study contradict those which were identified based on the personnel experience. Machine rotational speed and loading were not constant during operation, and they varied partly stochastically and partly according to production plans. The result of data analysis presented in Tables 1, 2 and 3 reveal appreciable changes in the vibration amplitudes of the frequencies specified when the machine speed and load were changed at the different operating conditions. The machine speed was always recorded by the vibration technician at the times when vibration measurements were done in addition to the record kept by the operator which includes one or two measurements of the speed daily. In most cases, the machine speeds recorded by the vibration technician and operator were made at different times of the day. This may cause some uncertainty in the differences of the speeds recorded by these two persons.

Almost all the frequencies stated in Table 1 reveal higher amplitudes at lower machine speeds. The reason is probably due to the initiation of bearing defects during the interval followed the first measurement, i.e. during the period 941013-941222. The rapid development of these defects resulted finally in the replacement of the bearing at 941226. In

Table 3, the bearing defect frequencies, represented by FTF and the non-synchronic frequencies, acquired higher amplitudes while all the other frequencies acquired lower amplitudes. This is probably due to the higher load, i.e. higher load may result in high amplitudes in the bearing defect frequencies. This is true because changes in the machine speed may cause bearing defect frequencies only to shift.

Dynamic Alarm: The alarm levels are usually set by a person not well trained for such tasks. It is rare that this person has the required knowledge of machine design, failure causes, defect development mechanisms, failure modes, identifying the significant frequencies and their relation to defects, etc. The setting of envelope alarm is, in general, done by "experts", but the possible changes in the machine speed, which cause frequency shift, are usually not handled reliably. When monitoring a vibration spectrum, the useful data to be observed are: The significant frequencies, amplitudes of the significant frequencies, new and disappeared frequencies and the noise level. The significant frequencies are those which can be utilised to assess which machine elements are deteriorating and the damage severity, and to track defect development. In the commercial VBM programs, which are usually used in paper mills, there are three types of alarms; Overall alarm to monitor the overall vibration energy, envelope alarming to monitor all the spectral patterns, and selective frequency band alarming. Overall vibration energy is the main factor for establishing tables for alarm systems [5]. The statistical alarm limits are usually based on some unrealistic assumptions such as normality of the vibration measurement values even during the defect development phase [6,7]. At alarm, extra vibration measurements at more frequent intervals are usually made before taking a decision on when to renew the bearing.

A dynamic envelope alarm consists of several warning levels and is prepared in advance, i.e. bearing significant defect frequencies are identified in the envelope alarming as multiples of the machine speed, and normalised to 1 Hz, (1 rps). Thus, it would be applicable for identical bearings and those of approximately identical vibration signatures and independent of the machine speed because frequencies are identified relative to the actual rps. Particular warning messages arise indicating deterioration severity and which frequency(ies) is (are) involved. Three levels would, however, be necessary to indicate defect initiation, development and the replacement level [8], but this is rarely done. Significant frequencies and amplitudes can be determined from the machine vibration history. Different operating conditions may cause variations in the vibration signature, i.e. new frequencies and different amplitudes, which were not considered in the original envelope alarming. The user should have the opportunity to change the envelope alarming to fit the operating conditions.

Identification of defect causes is not easy for the technician especially, when thousands of measuring points are required to be assessed say monthly. By means of a software program, which is prepared specially for identification of defect causes, it is possible to identify characteristic defect vibration frequencies and evaluate the state of the interesting bearings easily and effectively. The available tools are not effective enough due to their inability of considering changes in the machine speed, bearing defect frequencies and operational conditions. The assessment of the bearing state should become easier when the dynamic envelop alarm is established and has run for a while to accommodate data. Alarm setting for each measuring point should be handled by software.

Example: The data used in this example are not all real but they are reasonable and based on the author's practical experience within paper mill industry. Assume that we have n nominally identical rolling element bearings installed at the driven sides of n drying cylinders, in a paper mill machine. In such a case the bearings can be considered to be exposed to approximately the same operating conditions, such as load, speed and temperature. Let the vibration frequencies which can be used to detect and follow damage developments in these bearings at an early stage be; 1*BPFO, 2*BPFO, 1*BPFI, 2*BPFI, 1*SBF, 2*SBF, 1*FTF, 2*FTF, (1*FTF+1*BPFO), (1*FTF+BPFI) and (1*SBF+1*BPFO) [2]. Also, let the three alarm levels, i.e. defect initiation, development and replacement, of each of these frequencies be; (0.3, 0.5, 0.95), (0.2, 0.35, 0.85), (0.25, 0.4, 0.9), (0.2, 0.4, 0.8), (0.3, 0.4, 0.9), (0.25, 0.35, 0.8), (0.2, 0.4, 0.8), (0.15, 0.3, 0.6), (0.15, 0.3, 0.6), (0.2, 0.4, 0.7) and (0.2, 0.3, 0.6), respectively, measured in mm/s. These levels can be estimated from the VBM database for the machine when enough identical bearing replacements have been recorded [2,7,8]. Make the alarm level at each frequency cover a range equal to 1.5% at either side of the designated frequency to compensate for the variations in the bearing defect frequencies. It is possible then to set the three-level envelope alarm for the above frequencies, manually or through the software. For easiness, let the location of each of the chosen vibration frequencies in the envelope alarm shift to the right or left precisely the same amount of increment or reduction in the planned running speed, respectively. Also, let the alarm levels increase or decrease by double the amount of increment or reduction in the planned load, respectively. Now, assume that at time $t_2 > t_1$, the vibration and operating conditions are measured once more simultaneously, and it is found that the vibration spectrum is not appreciably changed, but the running speed and load are increased by 10% each. This means that all bearing defect frequencies are shifted by about 10% to the right of their original positions and their amplitudes are increased by at least 20%. In this case, using a three-level dynamic envelope alarm, which should be adjusted manually (or automatically), no warnings would be expected even if operating conditions are changed appreciably, see Fig. 5.

However, this is not possible if the envelope alarm is not adjusted, which is the case in general when using commercial VBM programs including one or, sometimes, two-level envelope alarms. Thus, the replacement of the bearing is indicated even though it is still functioning. For example, frequency b=112.1 Hz, which has 0.5 mm/s amplitude, will shift to the right by about 11 Hz and its amplitude may increase to 0.6. This means that frequency b will shift to the range which is very close to that occupied previously by the frequency (f+b)=117 $\pm$ 2 Hz. In other words, frequency b would exceed the third envelope alarm level and lead to unnecessary bearing replacement, when the real amplitude of frequency b remained at only 52% of the replacement level 0.95 mm/s, see Fig. 5. In general, using a three-level envelope alarm, whether dynamic or not, for bearing defect frequencies increases the bearing's useful life, because the bearing is usually replaced as soon as its vibration level exceeds the normal, i.e. when it is still below the second alarm level [8]. A dynamic three-level envelope alarm would increase the bearing useful life appreciably and reduce the number of planned and unplanned replacements because the ambiguities, which arise due to changes in speed and load, and cause inaccuracy in the assessment of the bearing condition, will be eliminated. In many cases, setting and adjusting envelope alarms manually is very difficult and expensive especially when it concerns thousands of measurement points. Now, if these three envelope alarms are normalised to one RPS and produced in a software program, which is automatically adjustable depending on operating conditions, then envelope alarm setting and adjusting for all n bearings will be easier and cheaper.

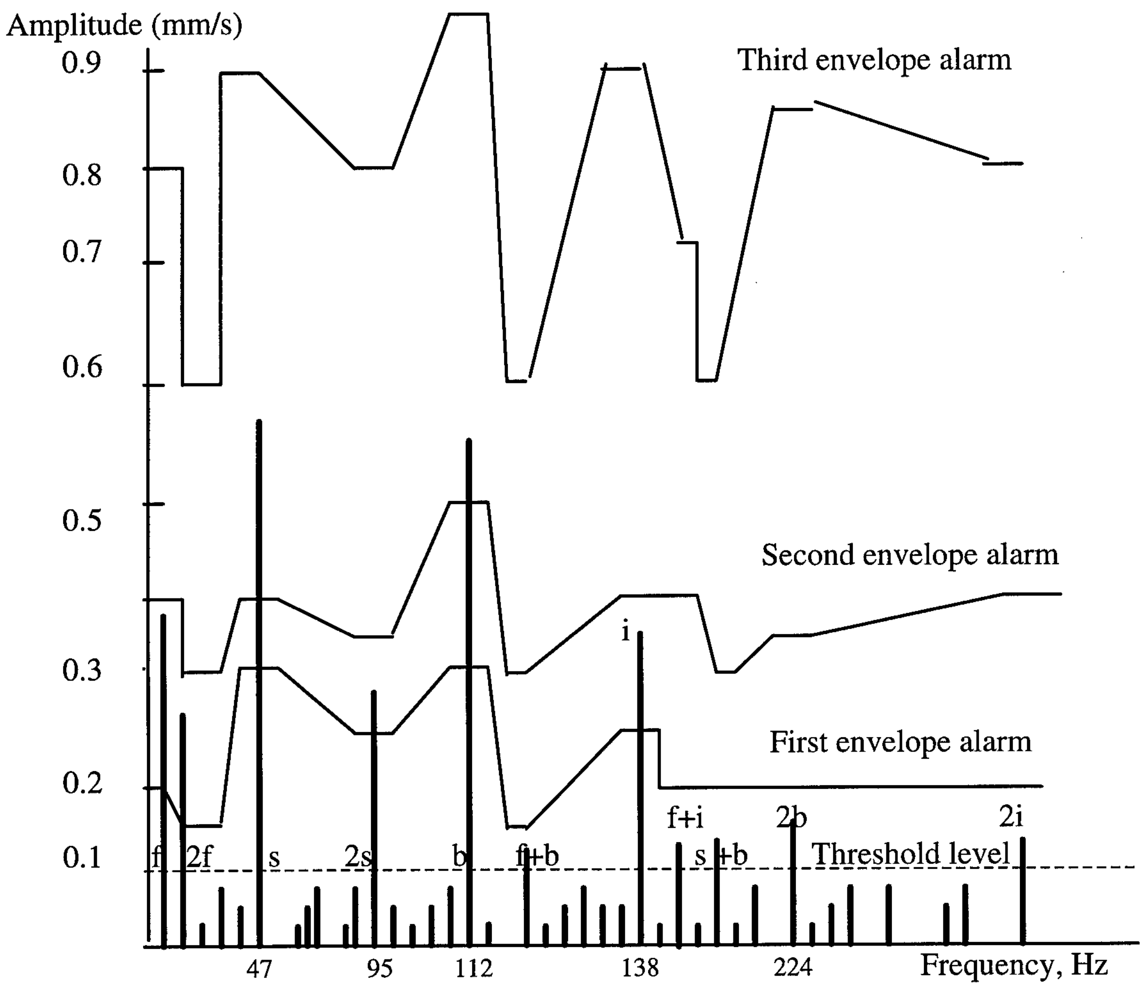

Fig. 5. A typical vibration spectrum reveals only the defect vibration frequencies of the bearing 23052 cck/SKF and the three assumed envelope alarm levels, where b, 2b, i, 2i, s, 2s, f, 2f, (f+b), (f+i) and (s+b) are 1*BPF0, 2*BPFO, 1*BPFI, 2*BPFI, 1*SBF, 2*SBF, 1*FTF, 2*FTF, (1*FTF+1*BPFO), (1*FTF+BPFI) and (1*SBF+1*BPFO), respectively. The speed is assumed to be 10 rps. For the bearing 23052 cck/SKF, FTF=0.45 Hz, SBF=4.73 Hz, BPFO=11.21 Hz and BPFI=13.79 Hz, all normalised to 1 rps. All the frequencies below the threshold level are considered insignificant.

Conclusions: It is beneficial to keep real-time measurements of the operating conditions particularly the machine speed and load to be used in conjunction with the vibration measurements, to eliminate ambiguities in deciding whether the increase in the vibration level is due to more deterioration or to the increment in the operating conditions. The second press cylinder group, first and second drying cylinder groups and Calender are identified in this study as the essential problem areas in the machine under consideration. The total stoppage time due to unknown reasons was very large and causes appreciable economic losses, about 2,3 millions SEK during fifty eight days (this is calculated in the basis that each hour causes losses equal to about 30 000 SEK). Thus, it would be beneficial to invest in the work aims to analysing and eliminating the basic reasons behind these economic losses.

Due to the lack of enough and reliable data, the analysis were limited to few rolling element bearings which made the results in the form of indications instead of statistical evidence. Thus, better data coverage and quality from the surroundings and more frequent vibration measurements help to achieve many goals such as detecting defect initiation and following its development, effective diagnosis and prognosis, effective control of the condition of machinery and production losses. The dynamic envelope alarm is shown theoretically to offer later renewal with fewer failures, and therefore lower cost and higher productivity. Using the dynamic envelope alarm, it is possible to identify defect vibration frequencies and to evaluate the state of the bearings easily and effectively.

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AUTHOR

Basim Al-Najjar: BSc in mechanical engineering, Licentiate Technology in Quality Technology from Department of Mechanical Engineering/Division of Quality Technology, Linköping University and Ph.D. in production management/Terotechnology from Institution for Industrial Engineering, Lund University, Sweden. Many years experience in maintenance, design and construction. Experience, as a consult, in using vibration analysis and problem shooting, teaching, improvements and research in maintenance and especially in Condition-based Maintenance within Swedish paper mill companies. Publications in Condition-based Maintenance in international journals and conferences. Senior lecturer at Lund University and Växjö University, Sweden.

Address: Växjö University, 351 95 Växjö, Sweden
Tel. +46-470-708422
Fax. +46-470-84004
E-mail: Basim.Al-Najjar@masda.hv.se