APPLICATION NOTE

Envelope Analysis for Diagnostics of Local Faults in Rolling Element Bearings

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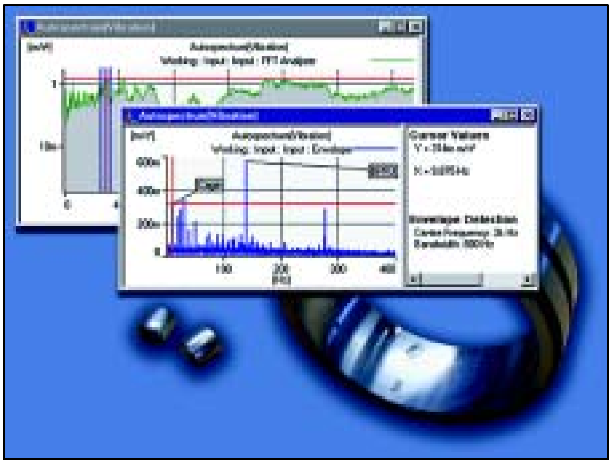
Introduction



Envelope Detection or Amplitude Demodulation is the technique of extracting the modulating signal from an amplitude-modulated signal. The result is the time history of the modulating signal. This signal may be studied/interpreted as it is in the time domain or it may be subjected to a subsequent frequency analysis. Envelope Analysis is the FFT (Fast Fourier Transform) frequency spectrum of the modulating signal.

Envelope Analysis can be used for diagnostics/investigation of machinery where faults have an amplitude modulating effect on the characteristic frequencies of the machinery. Examples include faults in gearboxes, turbines and induction motors. Envelope Analysis is also an excellent tool for diagnostics of local faults like cracks and spallings in Rolling Element Bearings (REB).

The Multi-analyzer System PULSE™ includes Envelope Analysis Type 7773. This application note briefly describes the ideas behind Envelope Analysis of local bearing faults, how Envelope Analysis is implemented in PULSE™, practical considerations, and two case studies.

Rolling Element Bearing Frequencies

Rollers or balls rolling over a local fault in the bearing produce a series of force impacts. If the rotational speed of the races is constant, the repetition rate of the impacts is determined solely by the geometry of the bearing. The repetition rates are denoted Bearing Frequencies and they are as follows:

- BPFO, Ball Passing Frequency Outer Race, local fault on outer race
- BPFI, Ball Passing Frequency Inner Race, local fault on inner race
- BFF, Ball Fault Frequency = 2 * BSF, Ball Spin Frequency, local fault on rolling element
- FTF, Fundamental Train Frequency, fault on the cage or mechanical looseness

The bearing frequencies can be calculated using the formulae given in the Appendix, or by using 'Bearing Calculators' supplied by the bearing manufacturers on either the web or on CD-ROM.

The spectrum of the vibration measured on a machine containing a faulty bearing will contain one or more of the bearing frequencies. Most often, and especially when a fault starts developing, the vibrations caused by a bearing fault will be obscured by the vibrations from other rotating parts like shafts, gears, etc., and the bearing frequencies cannot be seen in either the time history or the spectrum of the vibration. The vibration from the healthy rotating parts will show up in the lower frequency range of the spectrum, i.e., a few harmonics of shaft speeds, tooth mesh frequencies, etc., but fortunately, the bearing fault vibration will manifest itself throughout the spectrum. This is due to the fact that a repetitive impulse, with repetition time = T, has a corresponding line spectrum consisting of all the harmonics of the repetition frequency = 1/T.

Load Variation Modulation

The radial load in the bearing determines the strength of the impact from rolling over a fault. A fault in a stationary bearing race will be subjected to the same force at each roll and consequently all the pulses in the pulse train will be of equal strength/height.

On the other hand, a fault in a rotating race will be subjected to a varying force, the variation repeating itself with the RPM of the race. This means that the pulse train will be amplitude-modulated with the RPM of the race, and in turn all the harmonics in the line spectrum, BPFO or BPFI (whichever correspond to the rotating race), will appear amplitude-modulated by the RPM of the race. Likewise, the BFF caused by a ball/roller fault, will be amplitude-modulated by the FTF, the RPM of the cage.

If there is more than one fault of a kind, the line spectrum will still contain the harmonics of the bearing frequency. Only the ‘shape’ of the spectrum will change depending on the relative positions of the faults.

Inter-harmonic Frequencies

In a very loose bearing some inter-harmonics of the relative frequency or speed between the two races will be dominant. Normally, where the outer race is stationary and the RPM of the inner race is the shaft speed, the inter-harmonics will be ½, 1, 1½, 2, 2½,... or $^1/_3$, $^2/_3$, 1, 1 $^1/_3$, 1 $^2/_3$, 2,... harmonics of the shaft speed. By inspecting the bearing frequency formulas it can be seen that the BFFO and BPFI may be very close to the ½ harmonics.

Envelope Detection

The bearing frequencies are present throughout the spectrum (the 1/T line spectrum), but obscured at lower frequencies by other vibrations. However, there is a technique that makes it possible to extract the bearing frequencies from the part of the vibration spectrum where the 1/T line spectrum is dominant, that is, amplitude demodulation:

> A band-pass filter, with centre frequency f_c, filters out the selected part of the spectrum, the output is shifted (heterodyned) to low frequency ($f_c \rightarrow$ DC) and subjected to envelope detection.

If the band-pass filter encompasses a range where the 1/T line spectrum is dominant, the resulting time history will be dominated by the envelope of the original pulse train. This envelope time history can now be subjected to FFT analysis for easy identification of Bearing Frequencies. The figures below illustrate these properties. A synthesised time signal is composed as follows:

A pulse with a repetition time of 25 ms (~ 40 Hz) is subjected to a certain amplitude modulation with a repetition time of 250 ms (~ 4 Hz = 240 RPM) plus random noise, 0 –

1 kHz, with substantially higher power. This resembles a BPFI = 40 Hz fault, load variation modulated by the Shaft-speed = 240 RPM, the bearing situated in a 'noisy' machine.

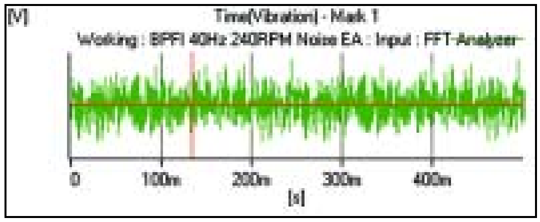


Fig. 1
Time history of the synthesised signal

Fig. 1 shows the time history of the synthesised signal, 12.8 kHz baseband and 0.5 s, representing two cycles of the shaft. The signal is dominated by the noise and the repeated pulse cannot be recognised.

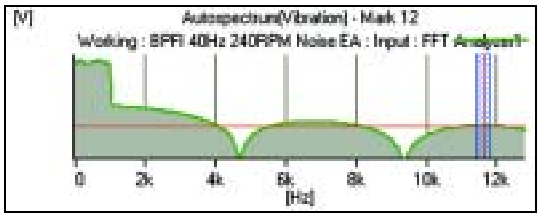


Fig. 2
Baseband spectrum of the synthesised signal

Fig. 2 shows the baseband spectrum, from 0 – 12.8 kHz dominated by the noise and above 1 kHz dominated by the line spectrum of the pulse train (the width of the pulse determines the notches in the spectrum). The notches are at the harmonics of 1/pulse-width, i.e., the wider the pulse, the more frequent the notches.

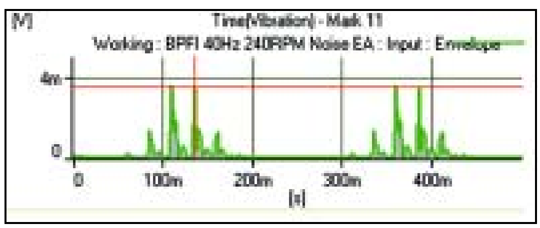


Fig. 3
Time history of the envelope detected in 400 Hz @ 11.6 kHz

Fig. 3 shows the time history of the envelope detected in the 400 Hz span around 11.6 kHz as indicated in Fig. 2. The result is as expected – a pulse train with repetition time = 25 ms and amplitude modulated with a period of 250 ms, i.e., exactly the envelope of the synthesised signal relieved of the contaminating noise.

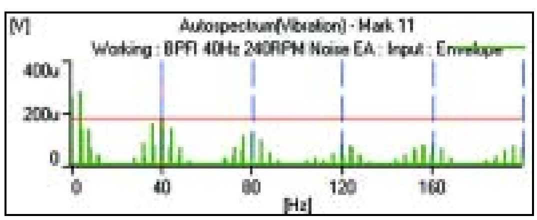


Fig. 4
High-resolution spectrum of the envelope

Fig. 4 is a high-resolution spectrum of the envelope shown in Fig. 3. It shows the first five harmonics of the BPFI = 40 Hz, all amplitude modulated by the Shaft-speed = 240 RPM = 4 Hz accounting for the load variation.

The modulation of the BPFI may be perceived as contaminating the clean 40 Hz, BPFI, spectrum, but it is, on the other hand, very informative. In a real situation where bearing frequencies would not be known in advance, this spectrum is the signature of a rotating race fault, most often the inner race. In the example, the BPFI is the 10th harmonic of the shaft speed. For a real bearing the BPFI and the shaft-speed are not harmonically related (see Bearing Frequencies Formulae in the Appendix). This fact can be used to identify a load modulated race fault.

Envelope Analysis on PULSE™ Type 7773

If the RPM of the spinning race is not known and has to be measured, you need three analyzers to make Envelope Analysis, see Fig. 5.

Fig. 5
Three-analyzer envelope analysis

1. A low frequency FFT Analyzer, to measure the RPM of the spinning race. A tacho signal and a Tachometer is a convenient option, but an FFT analyzer will do fine.
2. An FFT Analyzer set to give the overall baseband spectrum, the spectrum used to determine the setting of the band-pass filter in the Envelope Analyzer.
3. The Envelope Analyzer – this is an FFT Analyzer with an extra 'Analysis mode', i.e., Envelope. The 'Centre Frequency' together with 'Span' control the band-pass filter. The bandwidth of this band-pass filter, 'BP', is twice the 'Span', which is the frequency range of the resulting envelope spectrum. The frequency resolution, *df*, of the envelope spectrum is given by *df* = 'Span'/'Lines'. This means that after having selected the band-pass filter, the frequency resolution is selected by the setting of 'Lines'. All the remaining analyzer properties are unchanged. The outputs from the Envelope Analyzer, Envelope Time History and Envelope Spectrum (and slices hereof) can be post-processed like the outputs from the FFT analyzer.

Practical Considerations

Setting the Band-pass Filter

The band-pass filter should be set where the effect of the bearing fault is predominant in the spectrum. It could be anywhere in the mid-range of the spectrum where there are no other significant contributors except for background vibrations. Peaks and especially new or evolving peaks in the mid-range of the spectrum are normally promising prospects. This is explained as follows: the structure, i.e., the support of the bearing or the complete machine of which the bearing is part, has some natural frequencies or resonances that are very sensitive to excitation. The line spectrum of the bearing fault is present throughout in the spectrum so the fault will excite the resonances, or, in other words, the resonances will amplify the line spectrum, giving at least a better signal/noise ratio, but there are some drawbacks, for example:

1. The bearing frequencies are correct, but the vibration level measured is very sensitive to the placement of the accelerometer and the values may represent the structural properties rather than the bearing condition. Trend analyses can only be made if the position of the accelerometer is fixed and the machine is in no way changed (resonances shift).
2. If the resonances are very narrow, i.e., very lightly damped, they may contaminate the envelope. The resonance will cause the pulse to ring according to the low damping, broadening out the envelope of the pulses. This will first compromise the envelope time history (e.g., possibly put another impact in the shade) and eventually the envelope spectrum as well.

If the machine can be stopped the vibration spectrum of the response to a single blow with a hammer will show the resonances. A resonance is, however, not necessarily excited by the bearing fault, it may coincide with the notches in the line spectrum of the impacts.

Frequency Resolution

The required frequency resolution, *df*, is determined by the frequencies to discriminate. These frequencies are given by the bearing frequencies and the RPM of the races. The lowest expected frequencies are the FTF or possibly the $^1/_3$ harmonics (looseness). If the ½ inter-harmonics are present, these frequencies together with BPFO and BPFI may set the discrimination. The *df* should be 5 – 10 times finer than the wanted discrimination. This may require long measurement time, T > 1/*df* – worth keeping in mind if recorded data is to be used for later off-line analysis.

Calibration, Scaling and Units

Envelope analysis reveals the bearing frequencies correctly. But, due to the unknown structural transmission path from the fault to the accelerometer, the strength of the vibration (= acceleration) probably reveals more about the placement of the accelerometer and the properties of the structure than the severity of the fault. This means that determination of bearing frequencies can be done without any considerations regarding calibration, scaling, or units. However, it is always recommended to calibrate the equipment to make sure that it performs correctly.

If trend analysis is the subject, the accelerometer should not be displaced and the envelope band-pass filter should not encompass a structural resonance but rather be set where the overall spectrum is flat.

Case 1 – Envelope Analysis on a Scraped Bearing

This case study demonstrates the importance of frequency resolution. Too coarse a resolution may lead to wrong conclusions.

A standard deep groove bearing Type 6302 was worn out and scraped. The bearing was very loose and in operation very noisy. To identify the fault(s) of the bearing it was subjected to envelope analysis. For that purpose the bearing was set up as shown in Fig. 6.

Fig. 6
Bearing mounted in lathe

The bearing was fitted to a shaft and mounted in the spindle of a lathe. The speed of the Inner Race becomes the speed of the spindle. The Outer Race is stationary and subjected to a certain load controlled by the position of the dummy tool mounted in the cross slide of the lathe. An accelerometer, Type 4398, was mounted on the cross slide to pick up the lateral vibrations caused by loaded bearing.

The setup is very close to a real measurement situation. The bearing is under load and the vibrations picked up by the accelerometer are those caused by the bearing and by all the rotating parts in the lathe itself. The bearing was subjected to various loads and speeds. The following analysis was done with ‘medium’ load and a speed of 1916 RPM.

Analysis

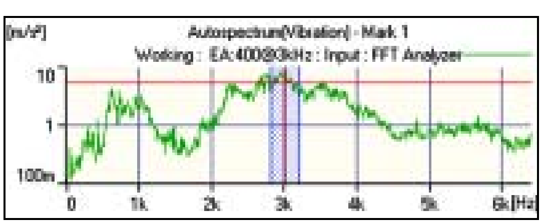


Fig. 7
Baseband FFT spectrum of bearing in lathe

The baseband FFT analysis is shown in Fig. 7. No attempt was made to identify (hammer test) structural resonances. A couple of trial analyses were made and the filter settings, 400 Hz @ 3 kHz, proved to be a good choice. The No. of Lines was set to 6400, giving *df* = 32 mHz.

Using the SKF Online Bearing Calculation, the bearing frequencies of Type 6203 at 1916 RPM were found to be:

– Shaft Frequency = 31.91 Hz

– FTF = 11.54 Hz
– BPFI = 142.59 Hz, 4½ inter-harmonic shaft = 143.61 Hz, Δ = 1.02 Hz
– BPFO = 80.81 Hz, 2½ inter-harmonic shaft = 79.78 Hz, Δ = 1.03 Hz
– BFF = 106.56 Hz

These are the potential fault frequencies. The frequency discrimination, Δ = 1 Hz = 30 * *df*, means that the selected *df* = 32 mHz is better than good.

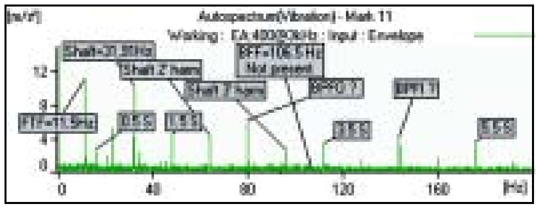


Fig. 8
Envelope spectrum of bearing in lathe

Fig. 8 shows the resulting envelope spectrum. The harmonics and ½ inter-harmonics of the shaft frequency are all there and this, together with a significant FTF, indicates a very loose bearing. This complies with the subjective impression of the bearing.

The BFF is not present, i.e., no indication of fault on balls. Ball faults are the most difficult bearing faults to find – the ball may roll in such a way that the fault is sometimes excited and sometimes not. A fault in a roller, on the other hand, stays in 'place' and will steadily be excited).

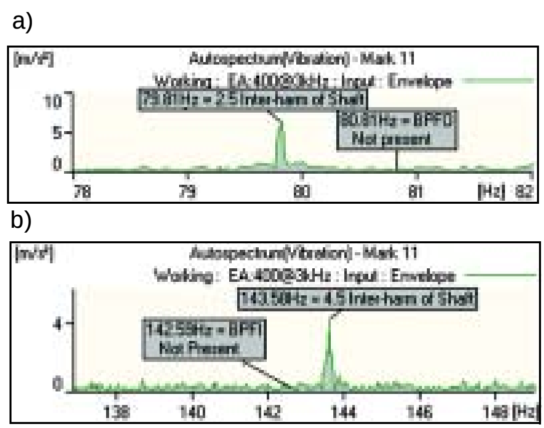


Fig. 9
(a) Expansion of the envelope spectrum around BPFO
(b) Expansion of the envelope spectrum around BPFI

The two components BPFO and BPFI seem to be there, but the question is whether these components are the bearing frequencies or the ½ inter-harmonics? As seen in Fig. 9 these components are not the bearing frequencies but the 2½ and 4½ inter-harmonics of the shaft frequency.

Conclusion

The bearing suffers from severe looseness and may have a damaged cage (FTF). There is no indication of local faults in either races or on the balls.

Case 2 – Envelope Analysis on a Propeller Shaft Bearing

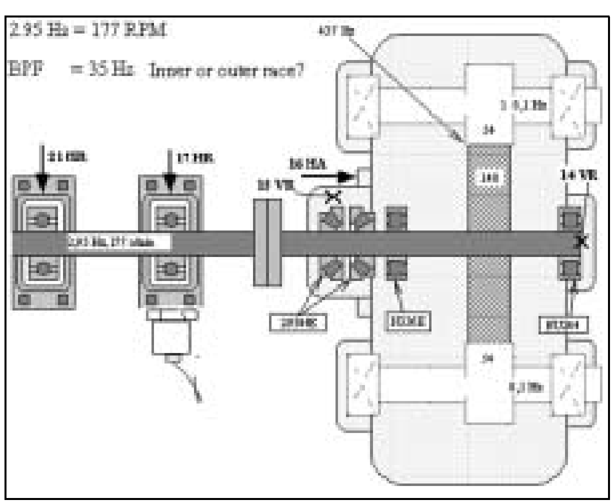


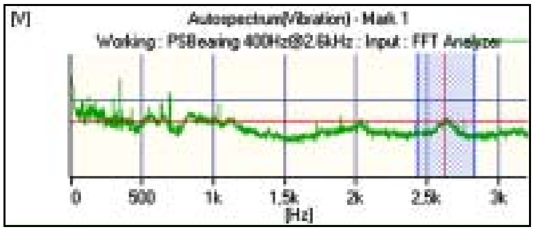
Fig. 10
Blueprint of propeller shaft and gearbox

Excessive vibration and noise in and around a ship gearbox and propeller shaft required vibration analysis to identify the problem.

As you can see in Fig. 10, two gearings and a variety of ball and roller bearings are possibly involved in the problem. Apart from the normal high noise level there was also a new impact-like noise indicating a bearing fault. Using a DAT recorder, several vibration measurements were made, and later analysed. The measurement taken at the bearing housing (17 HR), was promising and, therefore, used for this case study.

The bearings in the gearbox were known by type, but the shaft bearings were unknown. What was known was the rotational speed of the propeller shaft = 177 RPM = 2.95 Hz and, from a former fault, an associated Ball Passing Frequency at 34 – 35 Hz at that speed.

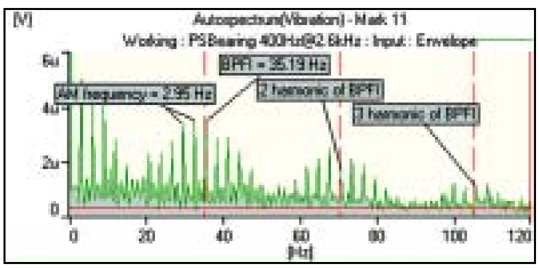
Fig. 11
Baseband spectrum of bearing vibration



Analysis

The baseband FFT analysis, 0 – 3.2 kHz, as shown in Fig. 11, exhibits discrete frequency components, some of which are caused by gear meshing, etc. The high vibration level at very low frequencies may be caused by shaft unbalance and misalignment but also by low frequency impacts, i.e., a bearing fault. The spectrum also shows some peaks that may represent structural resonances. The one at 2.6 kHz turned out well. The band-pass filter was selected to 400 Hz and the No. of Lines to 800, giving a *df* = 250 mHz. This resolution is one tenth of the shaft frequency, which, in this case, is considered to be the required minimum discrimination – the potential load-variation-modulation frequency.

Fig. 12
Envelope spectrum of bearing vibration



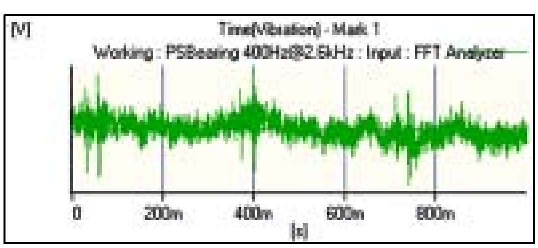
The envelope spectrum in Fig. 12 shows the characteristic signature of an inner race fault:

The harmonics of the BPFI = 35.19 Hz load-variation-amplitude modulated by the shaft frequency = 2.95 Hz; there are no signs of other bearing frequencies. After replacement of the bearing, the diagnosis was confirmed upon inspection.

Envelope Time History

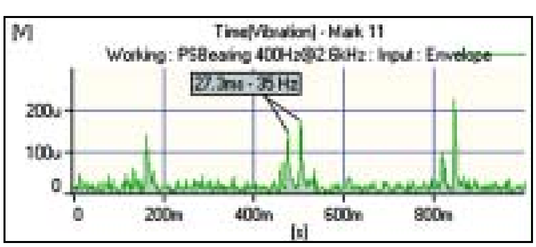
Bearing frequencies are best identified in the envelope spectrum. Studies of the envelope time history may be valuable in determining when and how many impactive events take place. As mentioned, the time study can reveal more than one fault of a kind.

Fig. 13
Baseband time history



The time study is not really addressed in this application note. Fig. 13 and Fig. 14 display the baseband and the envelope time histories of the propeller shaft bearing, giving an impression of the potential of the envelope time history.

Fig. 14
Envelope time history

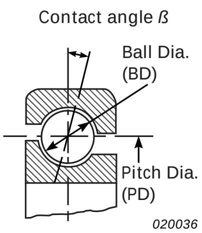


The baseband time history is quite noisy, but by exerting some good will, it is possible to identify three impulsive events within 1 s corresponding to one event per shaft cycle.

One event per cycle cannot be related to a bearing fault, but the envelope time history shows that there are three groups of impulsive events per second. Within the group it is possible to identify peaks spaced by 27.3 ms corresponding to the 35 Hz = BPFI. Only 5 – 6 of the 35 ball passing events per second (where the fault is in the load zone) can be seen. The distance between the groups of impulses is exactly equal to the time period of one shaft cycle confirming the amplitude modulation of the impacts by the shaft speed. The only things seen in the baseband time history are the most energetic impacts. The others, which are easy to see in the envelope, are difficult or impossible to determine in the baseband time history.

Appendix

Bearing Frequencies Formulae



n = Number of balls or rollers
f_r = relative rev./s between inner and outer races

Impact Rates f(Hz) assuming purerolling motion:

BPFO, Outer Race Defect: $f(Hz) = \frac{n}{2} f_r \left(1 - \frac{BD}{PD} \cos\beta\right)$

BPFI, Inner Race Defect: $f(Hz) = \frac{n}{2} f_r \left(1 + \frac{BD}{PD} \cos\beta\right)$

BFF, a Ball Defect: $f(Hz) = \frac{PD}{BD} f_r \left[1 - \left(\frac{BD}{PD} \cos\beta\right)^2\right]$

The formulae apply to the shown deep groove bearing and only in the case of pure rolling operation. It is recommended to use Bearing Frequency Calculators supplied by bearing manufacturers, e.g., www.skf.com

References

Search for Envelope Analysis, Type 7773 in:
PULSE Knowledge Library (VP 7800), part of PULSE software

Technical Review, No. 1 1987 Vibration Monitoring of Machines on www.bksv.com (BU 0029)

Product Literature

- System Data: IDA^e Hardware Configurations for PULSE, Types 3560C, 3560D and 3560E (BU 0228)
- System Data: Software for PULSE, Types 3560C, 3560D, 3560E with Types 7700, 7701, 7702, 7705, 7707, 7770, 7771, 7764, 7772 and 7773 (BU 0229)

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