Analysis of Elevator Ride Quality and Vibration

by Gregory P. Lorsbach

Introduction

The measurement of elevator ride quality (ride comfort) has become an important subject within the elevator industry. It is now often part of specifications for new and modernized elevator systems. It is also a competitive issue for elevator manufacturing, installation and maintenance companies, because it is a strong indicator of the quality of design (overall structure and components), installation and service of the elevator systems. Additionally, the analysis of vibration and sound that has been collected for ride quality measurements provides the ability to diagnose the function of elevator and escalator system components.

How ride quality is measured strongly affects the results of those measurements. Based on extensive work performed by companies from around the world, an international standard was developed for the measurement of elevator ride quality. The standard is *ISO 18738 Lifts (Elevators) – Measurement of Lift Ride Quality*. ISO 18738 establishes the requirements, methodology and processing techniques required to standardize the measurement and evaluation of elevator ride quality and performance characteristics including acceleration, velocity and jerk. This standard does not try to establish what is or is not acceptable in terms of ride quality. Practically, acceptability has to be considered a moving target. The technology and techniques to provide "good" ride quality will change (hopefully for the better) over time.

Utilizing the standard offers the ability to evaluate and troubleshoot using vibration and sound to identify problem areas and improve ride quality. It is important to remember that we are not simply evaluating vibration and sound, but the vibration and sound that relates to ride quality (i.e., human response to that vibration and sound). This means that we are evaluating vibration that was collected in a specific way and analyzed using specified techniques.

The Vibration Record

First-Order Analysis – Troubleshooting

Data as collected by an instrument may or may not be related to how people feel that vibration, depending upon how the data were processed. For example, Figure 1 displays the vibration and sound level as collected by an EVA-625 prior to processing for ride quality evaluation. Although this is a graphical representation of sound level and accelerometer outputs, the elevator industry generally distinguishes between acceleration and vibration based on the net motion of the car. The evaluation of vibration with respect to human response requires that data as collected by the instrument be weighted (filtered), utilizing the whole-body weighting (Figure 2) as specified in ISO 8041. Paying special note to the weightings in Figure 2, it is clear that people are most responsive to 1-2-Hz vibration frequencies in the horizontal directions (x and y), with sensitivity dropping off at higher and lower frequencies. In the vertical direction (z), we are most sensitive to 5-8-Hz vibration frequencies, with human response

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dropping off at higher and lower frequencies. This has important ramifications from a design standpoint. In the simplest sense, structure design and component choices can be made so as to minimize vibration at frequencies most readily felt and heard.

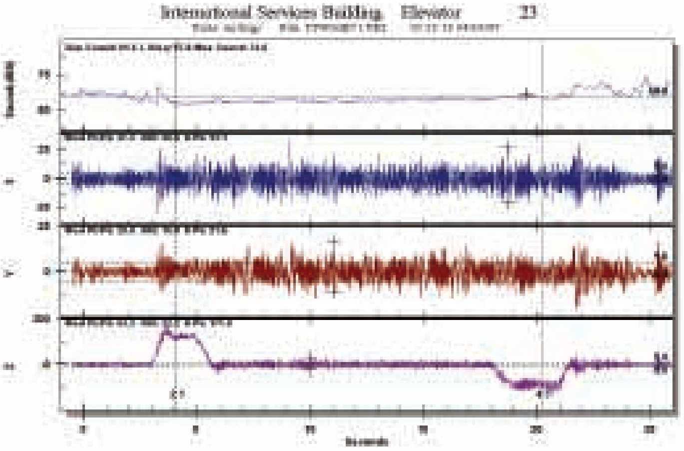

Figure 1: Displayed from top to bottom are sound level, x-axis acceleration (front to back), y-axis acceleration (side to side) and z-axis acceleration (vertical) time histories.

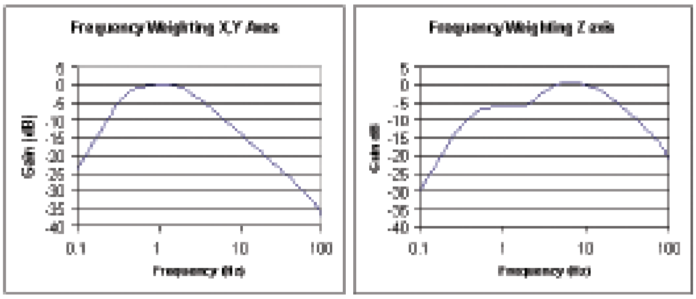

Figure 2: Whole-body frequency weighting

Figure 3 represents the data after it has been processed according to the International Organization of Standardization (ISO) standard and is used to evaluate elevator ride quality. This allows the direct diagnosis of problems that have a negative effect on ride quality. The data that has been processed according to the standard is intended to give meaning, such that an increase in the level of vibration corresponds to an increase in the perception of that vibration.

Although measurement and analysis provide a complete standardized evaluation of performance characteristics of an elevator system, a limited analysis of the vibration at full speed will be made for the purposes of this discussion. Vibration is characterized in terms of the maximum peak-to-peak vibration, and the A95 (typical vibration) between the points that an elevator has traveled 0.5 m from its start position, through to the point at which an elevator has traveled to within 0.5 m of its final position. The units typically used in evaluating vibration are milli(g)s. Bear in mind that vibration is a result of both the moving and control elements that make up an elevator system.

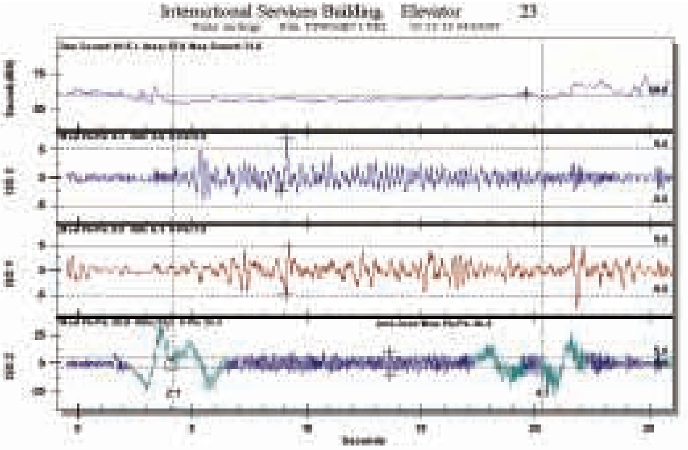

Figure 3: Data used to evaluate elevator ride quality after it has been processed according to the ISO standard

When attempting to evaluate the function of the components of an elevator system, the first approach is to conduct a first-order analysis based on a few simple questions:

1) Is the vibration acceptable?
2) Does the problematic vibration show up in the horizontal axes or vertical axis?
3) Is the vibration impulsive or continuous?

1) Acceptable Vibration

As a worldwide supplier for ride quality instrumentation, I am often asked what is considered "a good vibration level." This question is not easily answered. What is acceptable from a vibration-level standpoint is based on many factors. A primary factor is a competitive issue with respect to the expectations of the local market. Realistically, every elevator company manufactures a system that causes a box to move up and down in response to traffic requirements. Competitive pressures keep the costs for equivalent functionality approximately the same. However, the motion and sound that a rider perceives correlate with the perception of the quality of design, installation and maintenance. It has been my experience that the maximum acceptable (good) vibration level for new or modernized elevators is less than 12 milli(g)s maximum peak to peak, and less than half that for the A95 (typical) peak to peak. It is not uncommon (and therefore achievable) for the maximum peak-to-peak vibration to be less than 10 milli(g)s, and the A95 peak-to-peak vibration to be less than 5 milli(g)s for high-speed elevators. Certainly, there is a relationship to cost. I will often suggest that the user measures an elevator that he or she considers acceptable. Using that data, the user can create internal benchmarks for acceptability.

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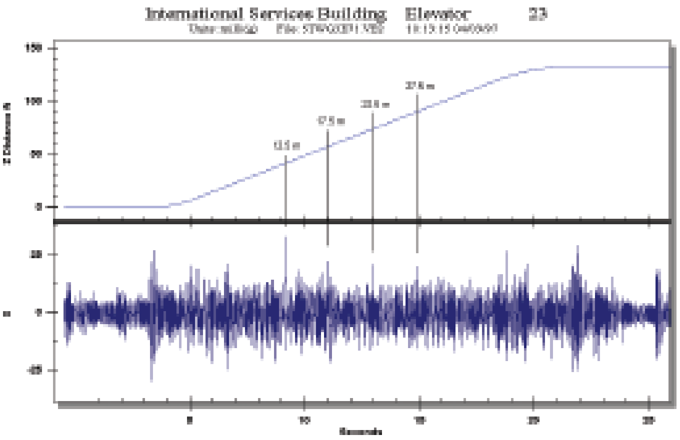

Figure 4: Distance traveled over time

2) Horizontal or Vertical

This is an important question, since vibration sources can be identified based on the axis that is being affected. Knowing the axes affected allows the user to quickly eliminate possible vibration sources. The potential horizontal-axis vibration sources are rail misalignment and/or roller or slide guides. The vertical-axis vibration sources are ropes, sheave, machine, controller/drive or counterweight.

3) Impulsive or Continuous

As we inspect the unfiltered x-axis time history closer, with respect to distance traveled (Figure 4), it is apparent that there is a series of bumps. Using the EVA Vibration Analysis Tools software, it is determined that the bumps are separated by one rail length (located at 12.5, 17.5, 22.5 and 27.5 m from the point at which the elevator started). This would lead one to the conclusion that there are misalignments at those points (one rail length apart) that are causing excess vibration. However, this is the vibration sensed by the EVA-625. When addressing ride quality, it is desirable to address the vibration that people feel.

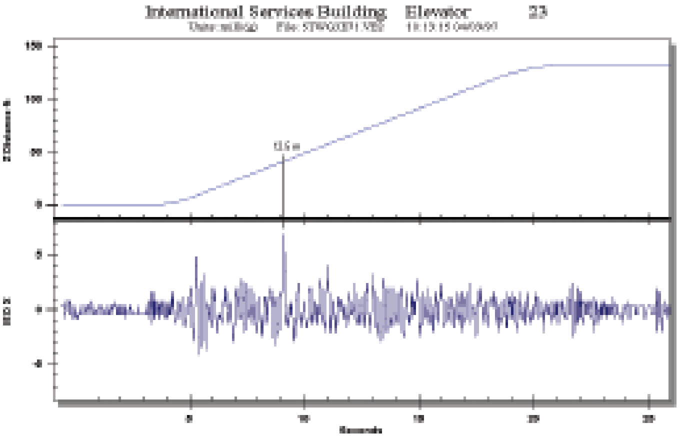

Figure 5: Distance traveled over time, filtered using the specified ISO standard on lift ride quality filter

Figure 5 shows the same record after filtering using the filter specified in the ISO standard on lift ride quality. Clearly, the signal is very different, since the apparent bumps located at 17.5, 22.5 and 27.5 m in the unfiltered data are no longer readily apparent, while the bump located at 12.5 m is clearly visible. This approach allows the maintenance company to address the vibration that a rider would feel and not waste time on vibration that people do not feel. This is an example of impulsive motion. When dealing with impulsive vibration in the horizontal axis, it is usually safe to conclude that it is related to a specific location in the hoistway. Fortunately, the use of the EVA Elevator/Escalator Vibration Analysis Tools software allows the user to precisely locate the bumps in the hoistway.

When addressing continuous vibration, the question of horizontal or vertical remains important. Continuous horizontal vibration is either the result of something that affects the entire hoistway (e.g., rail misalignment), or the source of vibration is traveling with the elevator (e.g., rollers). In the most general sense, horizontal vibration sources are located within the hoistway or even on the car.

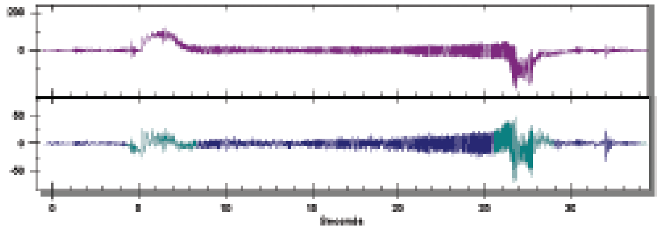

Figure 6: The upper portion of this table shows acceleration for reference, while the lower portion is the ISO-filtered data used for ride quality measurements.

Some of the sources of continuous vertical vibration are related to the ropes, sheave, machine, controller or counterweight. Referring to Figure 6, the vertical-axis continuous vibration is readily apparent. It is clear that there is a strong vertical vibration throughout the record. However, the vibration level greatly increases as the elevator travels from the bottom floor to the top floor. We often use the Fast Fourier Transform (FFT) vibration-level-versus-frequency tool to perform a second-order analysis when finding the source of vibration.

Second-Order Analysis

When discussing continuous vibration, we often refer to vibration resulting from rotating elements. (A series of impulsive events can also be called continuous.) The driving elements of an elevator have a number of rotating components, including the sheaves, motor and gears within the gearbox. Problems in these areas can have a significant effect on vertical vibration level and ride quality. Each of those can be characterized by a rotational frequency. The rotational frequency can be calculated by finding the diameter of the element (via direct measurement) and the speed of the elevator (via EVA-625 data analysis). As an example, assuming a 400-mm sheave diameter (*d*) and 6-mps elevator (*v*), the rotational frequency (*f*) is calculated such that:

d = 400 mm = 0.4 m
v = 6 mps
Circumference (C) = πd = 3.14159 X (0.4 m) = 1.2566 m
Sheave Rotational Frequency = v/C = 6/1.2566 = 4.77 rotations per second = 4.77 Hz (≅4.75)

If the FFT (spectrum) of the vertical-axis vibration signal indicates significant energy at about 4.75 Hz, then we can correlate that with the sheave. It is also important to realize that this would be the fundamental frequency and that some higher-order harmonics may also be present (i.e., 9.5, 14.25, 19 Hz, etc.), as well. The same approach can be applied to such components as guide rollers. Additionally, the motor/worm/ring gear rotational frequencies may be identified in the vertical-axis vibration signal.

A good example of using the FFT is demonstrated (again referring to Figure 6 and with a speed of 2.5 mps, geared). It is obvious that the vibration level increases as the car travels from bottom to top. The lower portion of Figure 6 indicates that the perception of that vibration increases, as well. The first thought upon arriving at this site was that the vertical vibration was related to the ropes. Had this been the case, however, there would likely have been a significant change in frequency, as analyzed through the FFT.

If one imagines a guitar string with a constant tension, as the string gets shorter, the frequency would increase. To test this, the FFT was used to evaluate the frequency content of the signal at different points during the trip (Figure 7). The spectrum of the vertical axis (Figure 8), just after the elevator reaches full speed, indicates that the dominant frequency is about 26.5 Hz. About halfway through the trip, the dominant frequency is still about 26.5 Hertz (Figure 9), although the amplitude has increased by about 150%. Just prior to deceleration, nearly at the top, the dominant frequency remains at about 26.5

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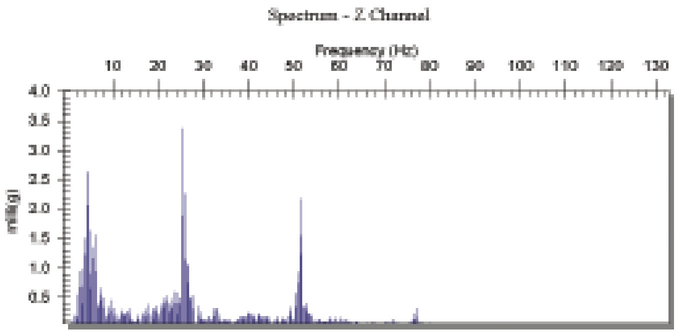

Figure 8

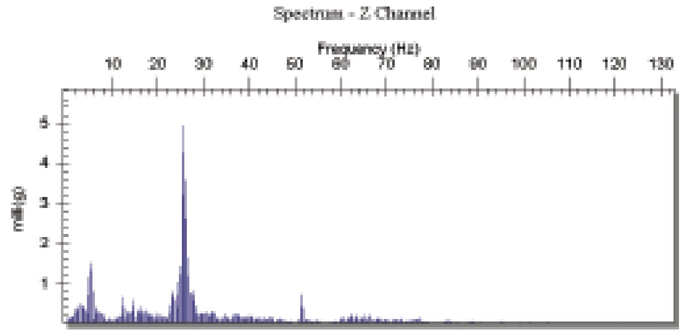

Figure 9

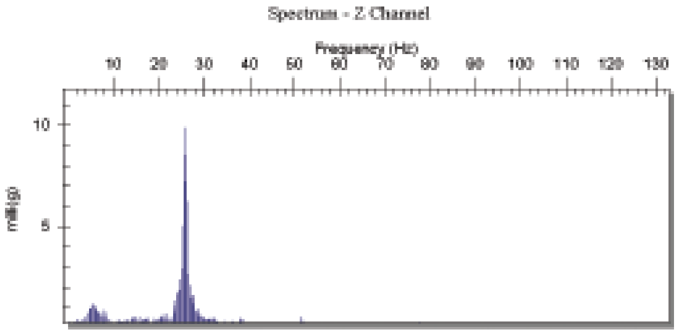

Figure 10

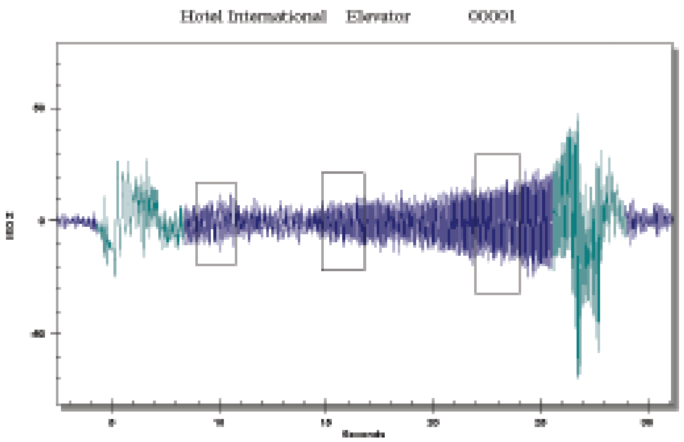

Figure 7: Frequency of the signal at different points in the trip

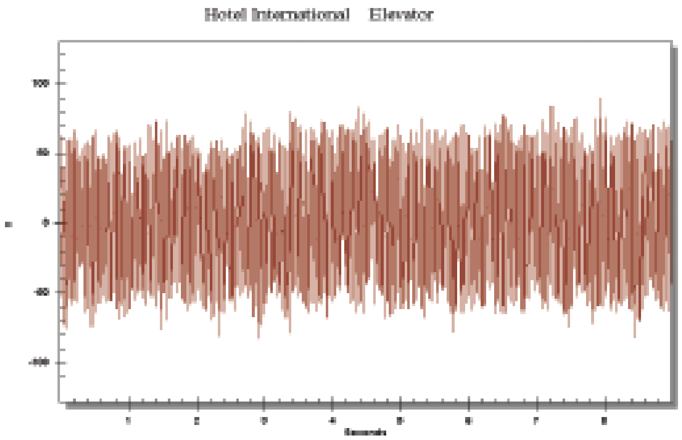

Figure 11

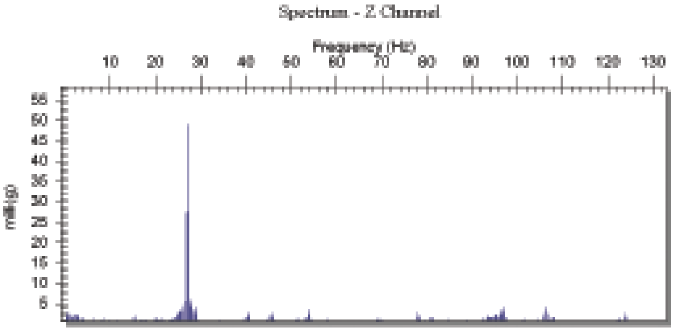

Figure 12

Hz (Figure 10), but the amplitude has tripled. This indicates that there has been no significant change in the frequency of vibration as the elevator traveled from the bottom to the top of the hoistway. Using this approach, the ropes and sheave can be eliminated. The next step was to attach the EVA-625 accelerometer directly to the gearbox and make a measurement while the elevator was moving.

Analyzing the vibration signal (Figure 11), the spectrum (Figure 12) indicates that the dominant frequency was about 27 Hz, or nearly the same as that measured on the floor of the car. This allows us to conclude that the source of the vibration (and poor ride quality) was the machine (gear mesh frequency).

Conclusion

It is important to remember that successful field personnel within the elevator industry are necessarily clever and analytical (problem solvers). Although they may have not been exposed to vibration analysis as part of their education or experience, they can apply basic and powerful techniques to analyze vibration and quickly evaluate the condition of most elements of an elevator system. Furthermore, it is a simple matter to determine if repairs or changes that had been made to an elevator had the desired effect of improving ride quality.

Additionally, there is significant value to evaluating ride quality prior to the turnover on new installations to establish a "fingerprint" for that installation and to measure before and after modernization to determine what may be required from a bidding standpoint and establishing the improvement for the customer after completion. 🌐