Key Dynamic Parameters that Influence Ride Quality of Passenger Transportation Systems

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Abstract Ride quality is a measure of the comfort level experienced by passengers and is intimately associated with their subjective perception and sensitivity to motion and to sound. This measure is affected by noise and vibration of a running system. On the other hand, ride quality is the measure of the product quality of a Passenger Transportation System (PTS) manufacturer. Ride quality of passenger transportation systems is critical for a PTS manufacturer to determine the subjective and objective quality of PTS. This is especially important in high rise (high end) systems. The paper investigates the dynamic interaction of PTS system components and their influence on ride quality.

1 INTRODUCTION

All humans are different and so the perceived performance of a PTS ride, either vertical using a lift or horizontal using a moving walkway or with an escalator in horizontal and vertical direction, is not the same. The acceptance levels of a given PTS assessed by a user group varies from user to user and typically show a wide variance [1].

A PTS contains of a large number of components which interact with each other. These components or sub-systems influence the ride quality of the system [2] and it is the interest of the PTS manufacturer to install and run a systems that makes the ride as smooth as possible. How these components dynamically influence the PTS and how humans respond to it will be investigated. Based on these results, measures to improve the ride performance will be suggested.

2 DEFINITION OF RIDE QUALITY

Ride quality of a PTS is a measure of the comfort level experienced by passengers and is intimately associated with their subjective perception and sensitivity to motion and to sound. For example, in the case of a lift system, to motion and noise levels of the car [3]. Ride quality is then affected by motion and sound quantified using the following parameters:

- Horizontal vibrations [measured in m/s²]:
  Horizontal vibrations are lateral motions (in x- or y-direction). These motions are quantified in terms of the frequency (measured in Hz) and the amplitude (measured in m/s²). However, the amplitude is typically measured in milli-g (1 milli-g = 0.00981 m/s²) [4].

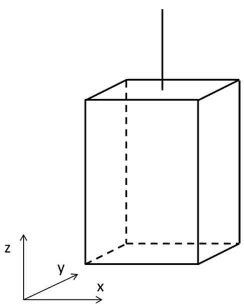


Figure 1 – Arrangement of x-, y- and z-axis

- Vertical vibrations [with the amplitude measured in m/s² and the frequency measured in Hz]: Vertical vibrations represent motions in the direction of the z-axis (for a lift: into or opposite to the direction of travel). A typical cause of this vertical vibration is the elastic behavior of the suspension mean, e.g. a steel-wire rope.
- Acceleration [measured in m/s²]: This is the rate of change in speed in upwards or downwards direction (direction of travel).
  Positive acceleration: increase of speed. Negative acceleration: speed decreases.
- Jerk [measured in m/s³] (time-derivative of acceleration = rate of change of acceleration): The rate of change in acceleration is called jerk.
- Sound pressure [measured in Pa] and Sound pressure level = defined as a logarithmic measure of the effective pressure of a sound relative to a reference value [measured in dB(A)]: Sound corresponds to vibrations which are transmitted through a solid, liquid or gas-form medium. A human being can hear frequencies in the approximate range of 20 to 20,000 Hz.
- Tympanic pressure [measured in Pa] (effect of dynamic change of ear pressure in the middle ear causing discomfort)

The acceptance levels of these aspects not only vary from human to human, but are also different in terms of product range (e.g. low-cost low-rise – middle segment - high-speed/high rise). Therefore, case-specific pass/fail or assessment criteria for PTS are put aside from this paper.

3 IMPACT TO HUMAN DISCOMFORT

Obviously, the response of the complex active structure of a human body to the multiple facets of ride quality is unique from human to human.
The application of (noise and) vibration to a human body can create various subjective effects and at the end cause human discomfort.
Humans are different in terms of body mass, portion of muscles, physical ability or overall condition [5]. And of course, the sensitivity for external stimulation is again different from person to person. If it comes to extrems, very poor ride quality with human over-sensitivity, vibrations can cause stress effects to following physiological areas: Cardio-vascular system, Nerves system, Muscles, Respiratory system.
It is common practice in the Lift Industry to stay within certain limits for critical ride quality aspects to ensure good ride quality:

- Humans are more sensitive to jerk rates than to acceleration, and jerk rates under 1.0 m/s³ are assessed as comfortable. [2]
- High rise/high speed travel (e.g.: travel distances of 400 m and more with maximum speed of 7 m/s usually cause an uncomfortable trip in a PTS.
- Acceleration rates of 0.8-1.6 m/s² improves passenger comfort, however, especially for high rise application these limits are disregarded due to transportation capacity requirements.
- Horizontal vibrations are typically specified below 25 milli-g (peak to peak in x- and y-axis).
- Maintaining the maximum limit sound level of 55 dB (A) is desired in the lift cabin.

In the context of '*human-machine/lift*' a unique person interacts with the lift car. The person stands on the cabin floor, typically wearing shoes. These represent the coupling or damping element between the human being and the lift system (cabin). The effects of different damping types (shoes) and the theoretical and experimental transmissibility has been compared in another paper [6]. The dynamic response of the lift cabin can substantially been influenced by the characteristics of the passenger and his behaviour during the lift travel. The dynamic interaction of lift components is defined by and finally counts in that human-machine context: Mass-Damping-Stiffness characteristics.

4 COMPONENTS THAT AFFECT RIDE QUALITY

4.1 Sources of vibrations

The operation of lift systems is affected by vibrations and associated vibro-acoustic noise. This affects ride quality and results in a high level of dynamic stresses in lift components. A good understanding and prediction of vibration phenomena occurring in elevator installations is essential for developing vibration suppression and control strategies in order to design a system which satisfies ever more demanding ride quality criteria.

Vibration sources affecting a lift car involve the car guiding system, suspension and compensating ropes and air flow [7,8]. The underlying causes of vibration are varied, including poorly aligned joints and imperfections of guide, eccentric pulleys and sheaves, systematic resonance in the electronic control system, and gear and motor generated vibrations [9,10].

In high-rise applications elevators are subject to extreme loading conditions. High-rise buildings sway at low frequencies and large amplitudes due to adverse wind conditions and the load resulting from the building sway excites the elevator system. This leads to large vibratory motions of elevator ropes [11,12]. The taller a building, the higher the rated speeds of elevator systems are needed. The dynamic responses become more adverse as the speed increases. Torque ripple generated in the motor causes vertical vibrations of the car [13]. At high speeds guide rail deformations induce large lateral vibrations of the car [14,15]. Furthermore, large aerodynamic loadings due to the airflow around the car result in excessive noise and flow-induced vibrations of the car structure [16,17].

The effects of component dynamics to the ride quality of a PTS are the subject of a number of previous investigations. These show evidence of the complexity of the interaction of multiple components.

A variety of sources affect the lift car and – if a passenger travels in the cabin – the passenger. The primary sources that affect ride quality are:

- guiding system (faulty guide rails and joints)
- suspension and compensating ropes
- air flow

Furthermore, imperfect pulleys and sheaves, a systematic resonance in the electronic lift control system, and motors and gears all generate vibrations which are transmitted into the system. Finally, the very slow excitation of building sway to the lift systems (mainly the ropes) can cause vibration in the cabin [13].

Guide rails: The most essential elements affecting ride quality of a lift are guide rails and their installation condition. Due to the T-profile design and the manufacturing process, guiderails are never perfectly straight and totally rigid [2]. The guide rail irregularities (poorly aligned joints, bends and unevenness) introduce lateral excitation to the car during its travel. The range of excitation frequencies depends on the speed. The fundamental frequency of this excitation is given as

$$\Omega = \frac{2\pi}{\gamma} V \frac{rad}{s} \left(or\ f = \frac{\Omega}{2\pi} = \frac{V}{\gamma}\ Hz\right) \tag{1}$$

where γ represents the wavelength corresponding to the guide rail profile.

For example, in a standard arrangement the two lengths of guide rail are connected together with rectangular cross section fishplate bridging the joint. The joint interface is the critical area as it is extremely difficult to arrange for the second moment of area (*I*) of the fishplate–joint interface to be the same as that of the guide rail. Thus, the bending stiffness (*EI*, where *E* is the Young's modulus) of the guiding system is non-uniform in respect of bending deflections. Therefore, during lift travel with a given speed of V, the excitation frequency *f* transmitted to the car frame can be calculated as

$$f = \frac{V}{L} \ [Hz] \tag{2}$$

where *L* is the rail length. Thus, using the standard rail length of 5 m and the speed range of 10 - 18 m/s this would yield the frequency range of 2 – 3.6 Hz. On the other hand, taking into account the fact that the wavelength due to manufacturing bends are about 0.1 m [8] the frequencies calculated from equation (1) can be over 100 Hz (100 – 180 Hz). In general, the nature of guide rail imperfections should be classified as *nondeterministic*. If the unevenness of guide rails is measured then the record for one rail will be different from that for another one. A nature such as this is referred to as being random or stochastic [15]. Consequently, the response of a car – hoist rope system is also a random phenomenon.

Roller Guides: Roller guides are available in various types and configurations for all kinds of application, such as high speed traction lifts or low rise hydraulic systems.
Under the consideration of stiffness and damping characteristic of these rotating or non-rotating components, it is obvious that they directly affect the vibration and thus the ride performance of a lift car.

Imperfections of rotating components such as traction sheave, diverter pulleys: Rotating components can only be made to its ideal round condition with huge manufacturing efforts. As a consequence, the rotating components deviate from the geometrical dimensional data and an imbalance excites the system with a frequency of its rotational velocity.

Rope dynamics: Due to their flexibility, hoist (suspension) and compensating ropes are susceptible to vibrations. These vibrations are transmitted to a lift car which often results in a ride quality which is unacceptable [18,19]. Elevator ropes can vibrate in the vertical (longitudinal) direction and lateral (horizontal) in-plane and out-of-plane directions (see Figure 4). An important feature of an elevator system is that the ropes are of time varying length. Furthermore, the number of passengers on board (load) changes. Consequently, the dynamic characteristics of the system vary during travel. In particular, the natural frequencies of the ropes vary slowly during the elevator car motion rendering the system non-stationary. An adverse situation arises when one of the slowly varying rope frequencies approaches near the frequency of a periodic excitation existing in the system. This results in a passage through external resonances.

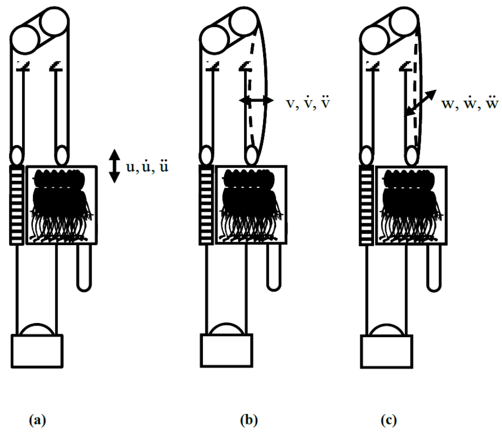


Figure 2 - Longitudinal (a), lateral in-plane (b) and lateral out-of-plane (c) vibrations of elevator ropes [19]

Overall, during the system operation the range of resonance frequencies of the ropes may vary from below 1 Hz to over 100 Hz. As far as the interactions between the ropes and the car are concerned, an elevator will not vibrate throughout its travel at high amplitudes but will ‘pass through’ a resonant vibration at some particular stage in the travel. An important excitation source relevant the rope – car assembly interactions is the low frequency building sway. This excitation will affect the modes below 1 Hz [21].

4.2 Human frequency band

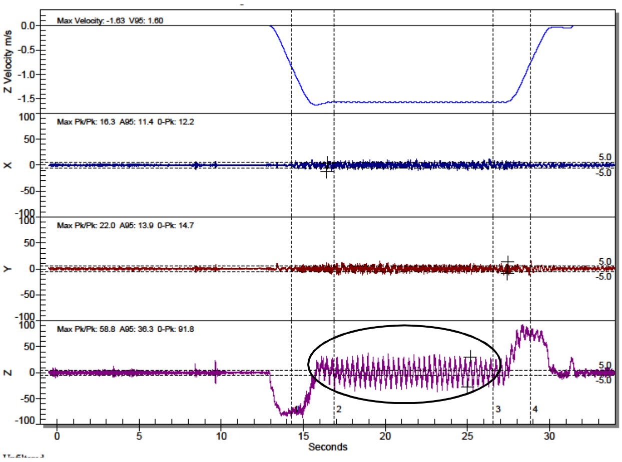
The human hearing range defines the band of frequencies that humans can hear and is usually mentioned as 20 to 20,000 Hz, whereas there are substantial variation between individuals, especially at high frequencies. The sensitivity for noise also varies with frequency. However, the noticeable range of vibration for human beings is even below the minimum of hearable noise.

5 DYNAMIC INTERACTIONS

It is the physical effect of resonance that causes issues with noise and vibration in PTS. The primary external resonance arises when the frequency of external excitation becomes close to one of the natural frequencies of the system. The natural frequencies of a given component is associated with one of the many standing wave patterns by which that component could vibrate. Any component can be forced into vibration at one of its natural frequencies (harmonics) if another interconnected object acts upon it with one of those frequencies.

With focus on the issue of resonance and its influence to the dynamics of lift cars the following practical example is considered.

- The system comprises a lift system with a car of rated speed 1.6 m/s and travel height 23.6 m with multi-reeving roping arrangement.
- The car sufferes from excessive vertical (z-axis) vibrations (peak-to-peak acceleration amplitude of over 58 mg; please see the time measurement record in Figure 3 and the corresponding FFT frequency spectrum in Figure 4)
- The fundamental frequency of the response is about 3.25 Hz which is close to the rotational speed in the diverter pulley system determined as 3.18 Hz.



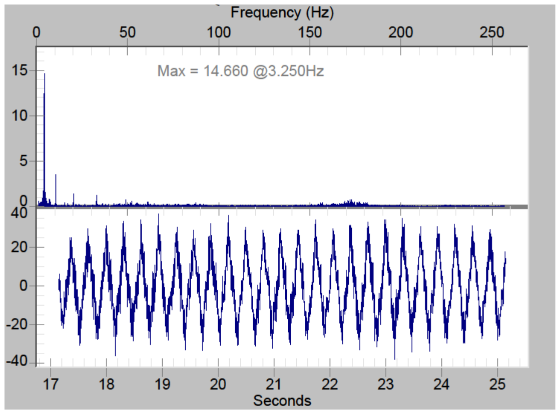


Figure 3 (left) - Time measurement record

Figure 4 (right) – Corresponding FFT frequency spectrum

- The vertical (bounce) vibrations are transmitted to the car through the suspension rope system which can be explained using the spring-mass model shown in Figure 5 where K_e represents the effective stiffness coefficient of the suspension ropes, M is the mass of the lift car assembly, *s(t)* represents the motion excitation at the traction sheave end due to excentricity or out-of-roundness error and *x(t)* is the response / displacement of the car assembly.

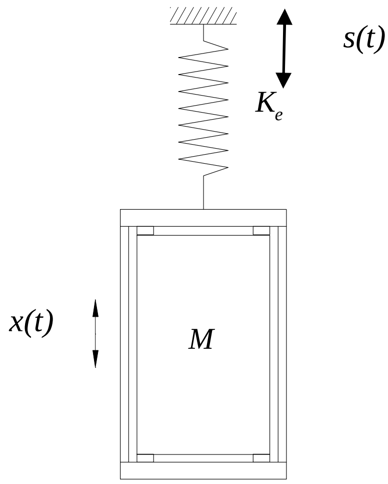


Figure 5 - Spring-mass model

The natural (resonance) frequency of the car-suspension system can be calculated using the following equation

$$\omega = \sqrt{\frac{K_e}{M}} \ \frac{\text{rad}}{s} \quad \left(f = \frac{\omega}{2\pi} \text{Hz} \right) \tag{3}$$

The response can then be determined by solving the following equation [5]

$$\ddot{x} + 2\zeta\omega\dot{x} + \omega^2 x = s_{\max}\sqrt{\omega^4 + 4\left(\zeta\omega\Omega\right)^2} \sin\left(\Omega t + \theta_0\right) \tag{4}$$

where the overdots denote the time derivavtive, $s_{\max}$ is the maximum displacement due to the excitation at the sheave, Ω represents the frequency of the pulley, ζ is the damping ratio and r represents the frequency ratio . The maximum displacement of the car is then determined as [9]

$$x_{\max} = s_{\max}\sqrt{\frac{1+4\zeta^2 r^2}{\left(1-r^2\right)^2+4\zeta^2 r^2}} \tag{5}$$

so that the corresponding acceleration amplitude is given as

$$a_{\max} = s_{\max}\Omega^2\sqrt{\frac{1+4\zeta^2 r^2}{\left(1-r^2\right)^2+4\zeta^2 r^2}} \tag{6}$$

The natural frequency changes during the lift travel and was determined from the braking tests as 3.125 Hz at the bottom landing and 4 Hz at the top landing, respectively (please see Figure 6 and 7).

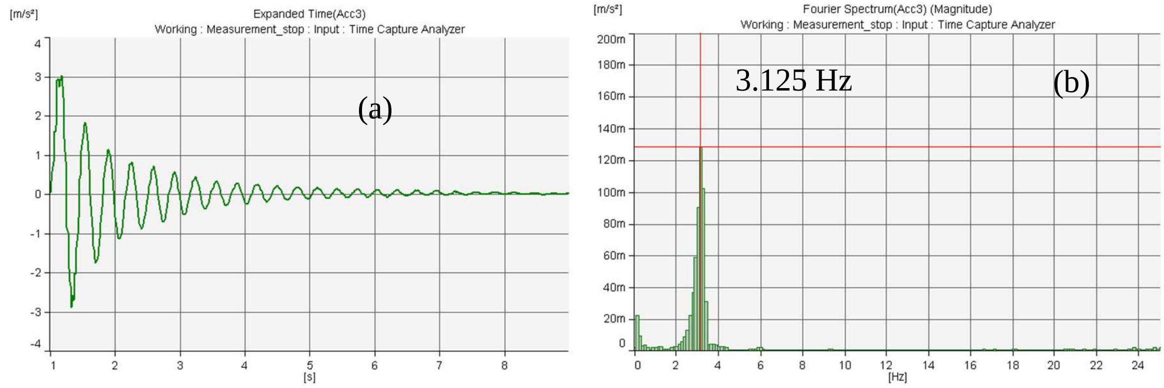


Figure 6 - Vibrations measured at the car sling after the brake was applied when approaching the bottom landing (a) time response; (b) Fourier frequency spectrum.

- It is evident that the range of resonance frequencies is close to the rotational frequencies of the diverter pulleys.

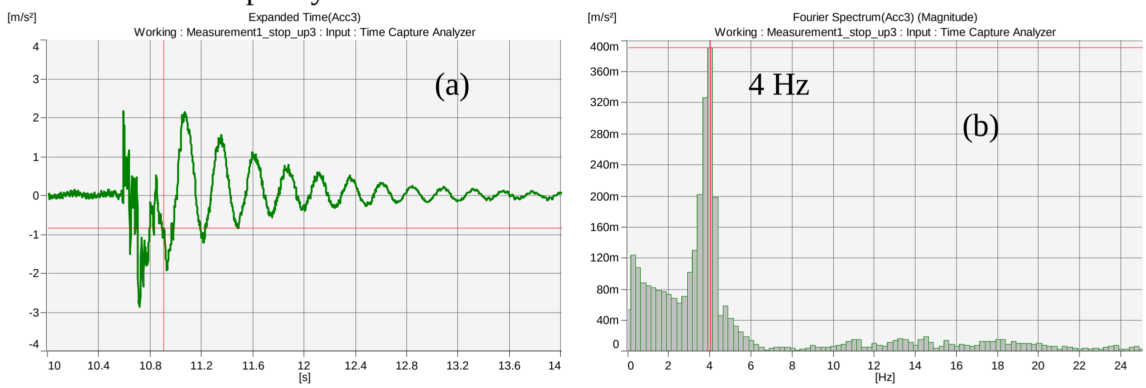


Figure 7 - Vibrations measured at the car sling after the brake was applied when approaching the top landing (a) time response; (b) Fourier frequency spectrum.

In these conditions, if the empty car mass is 15,800 kg and the rated load is 9,100 kg, the suspension stiffness coefficient, with the car approaching the bottom landing is about K_e = 10.545 MN/m, the natural frequency calculated according to equation (3) is 3.275 Hz. Assuming that the out-roundness error results in the maximum displacement $s_{\max}$ = 0.1 mm, with a small damping ratio of 1%, the maximum displacements of the fully loaded car calculated from equation (5) may reach 2.5 mm and the corresponding acceleration determined from equation (6) will be over 100 milli-g.

- The dynamic interactions between the pulleys, suspension and lift car are responsible for compromising ride quality of the system.

6 IMPROVE NOISE AND VIBRATION PERFORMANCE

The following three basic principles can be applied to mitigate the effects of / reduce noise emission and vibration in mechanical systems:

1. Prevention. To reduce the strength of the source.
2. De-coupling. To interrupt the noise/vibration path.
3. Damping. To absorp the energy of noise/vibration.

However in lift systems, due to their unique design and principles of operation, not all available measures can be applied when it comes to an issue of noise and vibration mitigation.

CONCLUSION

The assessment of the influence of noise and vibration in a PTS is based on the subjective passenger perception. Therefore there is limited evidence of how this influence can be quantified.
However, the causes of noise and vibration and their effects (reponses) can be quantified. If the excitation forces are identified the responses are determined through the application of experimental techniques and/or calculated using analytical techniques and/ or computer simulation.Relevant mitigation measures to reduce their effect can then be applied.
In regards to the key dynamic parameters that influence a PTS it can be concluded:

- PTS ride quality can be be assessed by the introduction of certain thresholds, such as the maximum acceleration, jerk and/ or noise levels. However, common limits of these values have not been agreed, as they strongly relate to market segments or country-specific acceptance levels.
- Noise and vibration can cause multiple, possibly severe damage in human physiological sub-systems. However, human response to noise and vibration is as diverse as humankind is and in most cases uncritical in PTS. However, in some cases excessive vibration might lead to failure of the design components and compromise passenger safety.
- PTS ride quality is affected by various sources of excitation that are related to elements such as rails, rail joints, roller guides, imperfect rotating components or ropes.
- The most critical phenomenom is resonance (forced vibration caused by an excitation originating from an external source), as this often leads to in instability of the vibrating system.
- Active and passive measures can be applied to mitigate the effects of resonance, noise and vibration.

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