# Methodology to Identify Noise and Vibrations Problems for Ride Quality Improvements

Raúl Monge, Alfredo Gómez

ITAINNOVA (Instituto Tecnológico de Aragón) C/ María de Luna 7-8, 50018 Zaragoza, Spain
rmonge@itainnova.es - agomez@itainnova.es

**Keywords**: Ride Quality, Vibro-acoustic analysis, Operational Modal Analysis (OMA), Operational Deflection Shapes (ODS), Structural and airborne transmission.

**Abstract.** Comfort is an important issue in the lift industry. This paper shows a methodology used to perform the vibro-acoustic characterisation of lift installations to provide data regarding the functional behaviour and to propose changes in the installations to improve the ride quality in car and to reduce the noise levels in the car. To achieve these objectives, different vibro-acoustic analysis techniques are applied to identify the main frequencies of noise and vibration that could lead to an increase in noise and vibration inside the car. The different techniques applied are FFT analysis, 1/3 octave analysis, vibration transmission by FRF analysis, Operational Modal Analysis and Operational Deflection Shapes. With this analysis, it is possible to determine the components that must be modified in order to improve the design and functionality, with the objective of either reducing the source or minimising transmission paths (airborne or structural), or removing any resonance that may increase vibration and noise inside the car.

## 1 INTRODUCTION

Improving Ride Quality is an important objective of lift manufacturers. The perception of the ride quality by users is based on noise levels, vibration levels and performance inside the car. The perception of lifts by users as "noisy", "vibrating floor" or "sharp starts or stops" can make the brand image worse.

The main causes of the absence of comfort inside lift cars can be produced by noise and vibration sources such as machines (drive), hydraulic pumps, door operators, electrical control cabinets, sliding systems in car and counterweights. However, other causes may also produce an absence of comfort such as resonances of components and insufficient isolation between the sources and the receptor. The passengers inside the car receive noise by airborne and structural transmission, vibration by structural transmission from the sources and a bad kinematic performance of the elevator (jerk, accelerations, vibrations…) [1].

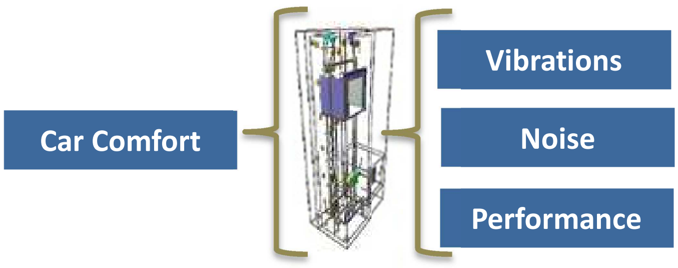



**Figure 1 Causes that influence comfort in the lift car**

The main vibro-acoustic problems that may appear inside a lift car are:

- High Noise Level (Leq) inside the car: this can be due to high noise of the machine, high noise of guide shoes (friction) and high airborne noise transmissibility. Some possible solutions to these situations are to improve the machine sound insulation, the sliding system performance, and the car sound insulation.

- High noise levels at frequency peaks inside the car can be due to resonances of lift components (car panel, car frame, machine frame…) and/or high structural vibration transmissibility from frame to car. Some possible solutions in these cases are “to shift” the resonances and to improve the vibration isolation between the car frame and the car.
- High vibration levels in the car floor can be due to high machine vibration (unbalanced rotating mass, bearing faults, electromagnetic phenomena…), and bad performance (wrong control, car floor resonance, high structural vibration transmissibility). Some possible solutions in these cases are to improve the machine behaviour, to improve the drive control, to “shift” the resonance and to improve the vibration isolation between the car frame and the car.

For these reasons, we present methodologies to identify noise and vibration problems in lifts using measurement equipment, noise and vibration sensors, acquisition hardware and software for dynamic signals analysis.

## 2 IDENTIFICATION OF NOISE AND VIBRATION PROBLEMS

It is possible to apply different techniques to analyse noise and vibration data depending on the targeted result accuracy required according to the problem severity.

The “Vibro-Acoustic characterisation of lift” methodology is used to evaluate the behaviour of the lift and to find possible solutions based on vibro-acoustic and dynamic measurements to improve the ride quality. This methodology does not permit the obtaining of quantitative results of the different sources’ contributions (vibration and noise) to the receptor (lift car user), but it does permit learning about which lift components should be modified to improve the ride quality. However, it is not possible to estimate the improvement without measurements. Different data analysis techniques are applied in this methodology, such as Fast Fourier Transform (FFT), Sound Equivalent Level (Leq), 1/3 Octave band analysis, Frequency Response Functions (FRF), Operational Modal Analysis (OMA) [2], and Operational Deflection Shapes (ODS) [3]. Based on the results of the different techniques and applying correlations between the results, it is possible to gain insights to make improvements in the lift.

The source contributions to noise inside the lift car methodology consists of applying techniques in order to identify and quantify noise and vibration sources using frequency response and coherence functions [4]. The Multiple Coherence Technique and the Output Power Allocation technique are applied in this methodology.

The Panel Contributions to noise level inside the lift car methodology consists of estimating the contributions of some areas to the acoustic pressure at a chosen point in a lift car based on the vibration panel (velocity). This methodology can be used to evaluate panels with their different vibro-acoustic properties (damping, reverberation….)

Explanations of the different methodologies and applications to real installations are shown in the following points.

### 2.1 “Vibro-Acoustic characterisation on lift” methodology

The vibro-acoustic measurements that are necessary to perform on the lift are basically: Ride Quality (Comfort), Noise at different points, Vibration in machine area, Vibration in car panels and the car frame, and Modal Tests of components. The ultimate objective is to find out correlations between the different results. These measurements are taken with at least two load conditions inside the car during downwards and upwards trips. In some cases, it is necessary to modify the nominal speed of the lift to analyse differences of results in frequency through changes in frequency problems.

The *ride quality* measurements are made in the centre of the car floor. The acquisition will be made by means of specific equipment according to ISO 18738-1 “Measurement of ride quality — Part 1: Lifts (elevators)”. The ride quality of the lift can be obtained through these measurements, and it is also possible to evaluate the behaviour of the lift during trips with different load conditions. The lift behaviour could indicate that the lift cannot achieve the nominal speed due to problems with the inverter parameters, insufficient machine power, error in the load balance, high friction force to sliding between the guide shoes and guide rails. It is also possible to apply the vibration narrow band spectra (FFT) analysis to the vibration time signal to detect the vibration frequency peaks in the range 0-80Hz that contribute more to ride quality.

| | Performance (Q = 0 kg) | | Performance (Q = 100 kg) | |
|---|---|---|---|---|
| | Upward | Downward | Upward | Downward |
| Amax (m/s^2) | 0.56 | 0.55 | 0.57 | 0.56 |
| A95 (m/s^2) | 0.53 | 0.52 | 0.53 | 0.53 |
| Dmax (m/s^2) | 0.57 | 0.58 | 0.56 | 0.57 |
| D95 (m/s^2) | 0.54 | 0.53 | 0.53 | 0.53 |
| Vmax (m/s) | 0.86 | 0.85 | 0.86 | 0.84 |
| V95 (m/s) | 0.85 | 0.84 | 0.85 | 0.84 |
| Distance (m) | 15.40 | 15.40 | 15.40 | 15.40 |
| Jerk max (m/s^3) | 0.95 | 1.05 | 0.97 | 0.95 |
| | Noise (Q= 0 kg) | | Noise (Q= 100 kg) | |
| | Upward | Downward | Upward | Downward |
| Leq (dBA) | 58.0 | 55.0 | 58.6 | 57.6 |
| | Vibration (mg) (Q= 0 kg) | | Vibration (mg) (Q= 100 kg) | |
| | Upward | Downward | Upward | Downward |
| X-A95 (mili-g's) | 2.00 | 3.00 | 2.00 | 3.00 |
| Y-A95 (mili-g's) | 4.00 | 4.00 | 4.50 | 6.50 |
| Z-A95 (mili-g's) | 20.00 | 16.50 | 13.50 | 14.00 |

**Table 1 Ride Quality values**


**Figure 2 Speed curve of lift**

In this case, the ride quality values obtained (acceleration, jerk and vibration values (A-95)) could be considered as acceptable (table 1), but the noise level was considered high. In lift performance (Figure 2) it can be seen that there is a problem in achieving the nominal speed during downward trips without load. This could be due to the machine not having enough electrical power, an error in the load balance or high friction due to sliding between guide shoes and guide rails.

Different measurement points are selected, including points close to machine shaft bearings, machine frame isolator and guide supports. The analysis of these measurement results will give insights to discover vibration problems. The *vibration measurements in the machine* area are carried out by means of accelerometers placed on the machine in radial directions, both vertical and horizontal (see figure 3). These measurements permit the obtaining of rotational frequencies, the power supply frequency and its harmonics, as well as the machine vibration peaks. These measurements allow us to obtain the vibration severity of the machine based on its vibration spectra. The vibration severity is calculated in velocity units (mm/s) at a range from 1Hz to 1000Hz, because gearless machines have low rotational speed. The machine vibration frequencies measured are compared with the technical data label of the machine to verify that the inverter parameters are correct. The technical data of the machine can be seen in table 2.

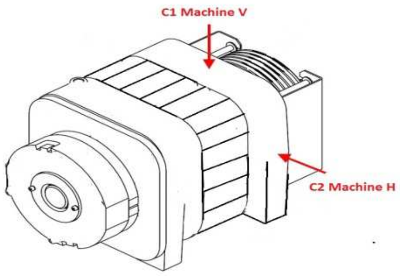



**Figure 3 Accelerometers on Machine**

| Technical data | |
|---|---|
| **Rotation speed ($n_M$)** | **239 $min^{-1}$** |
| **Power supply freq ($F_N$)** | 31,9 Hz |
| **Pulley diameter** | 160mm |
| **Roping** | 2:1 |

**Table 2 Machine Characteristics**

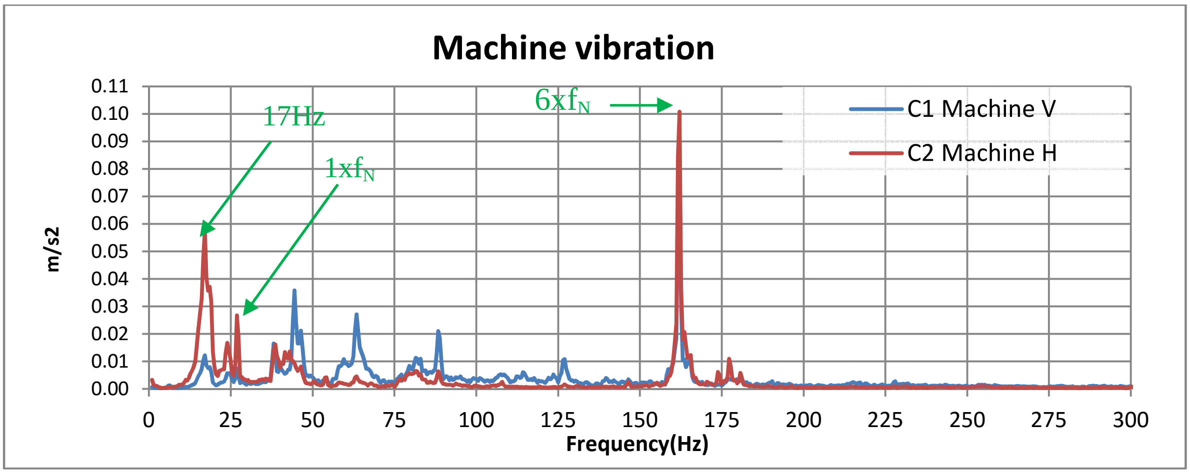



**Figure 4 Machine vibration spectra**

The machine data show that the nominal speed is 1m/s, but previous performance measurements show that the nominal speed is 0.85m/s, therefore for speed V= 0.85m/s the following frequencies are calculated:

$$f_N = 0.85 \text{ x } 31.9\text{Hz} = 27.1\text{Hz} \quad => \quad 6 \text{ x } f_N = 162.5\text{Hz}$$

The vibration spectra show a peak at 27.1Hz, which is $1xf_N$ (power supply frequency), with higher amplitude in horizontal direction and a peak at 162 Hz, which is $6xf_N$ ($6^{th}$ harmonic of power supply), with similar amplitudes in vertical and horizontal directions. A peak at 17Hz also appears in the horizontal direction in the spectra, which could be caused by a pitch of the machine in a horizontal direction or a resonance of the frame structure. The vibration severity is higher in the horizontal direction (1.3mm/s) due to the possible pitch around the machine axial axis above the isolators at low frequency.

The vibration transmissibility through the machine frame isolators can be obtained by means of accelerometers placed on the machine and accelerometers placed on the machine frame. The vibration transmissibility is calculated at the range 0-200Hz (Figure 5), but the transmissibility is analysed only at the main vibration peaks of the machine in a vertical direction because the higher vibration values on the machine frame are related with the main vibration peaks of the machine. The vibration transmissibility is defined as the ratio of the vibration amplitude on the machine frame and the vibration amplitude on the machine plate (FRF – Frequency Response Function). Figure 5 shows that the isolators are not working correctly at around 150Hz.

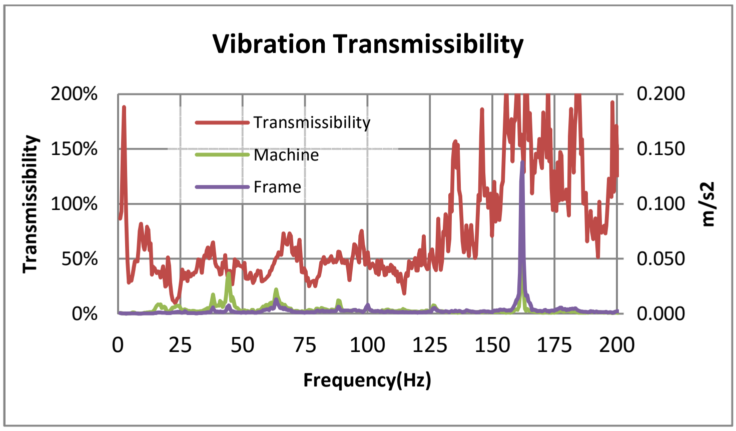



**Figure 5 Vibration transmissibility through isolator**

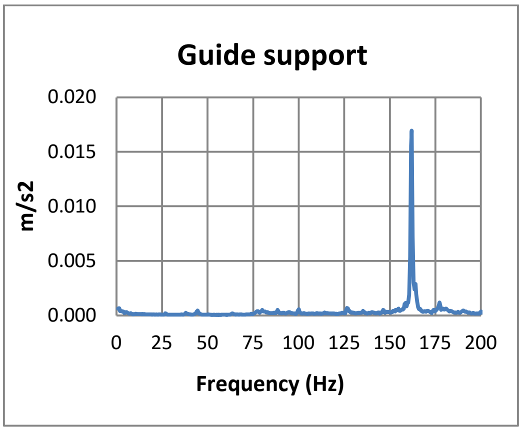



**Figure 6 Guide support vibration**

Furthermore, the vibration on the guide supports (figure 6) near the machine frame is measured by an accelerometer in a horizontal direction, because, depending on the hoistway structure (metallic, cement), the structure vibration could produce a high noise level inside the hoistway and therefore increase the noise level inside the lift car.

The *noise measurements* are taken inside the lift car and in the hoistway, near the machine and above the roof car. These measurements permit the obtaining of the noise equivalent level ($L_{eq}$), the noise evolution during the trips, the noise spectra at 1/3 octave band, and the contribution of the machine frequencies and converter frequencies by means of FFT analysis in spectra 1/3 octave band. The noise levels allow us to estimate whether the noise transmission is mainly airborne or structural through the hoistway to the car.

The noise equivalent levels inside the lift car are lower than the levels in the hoistway (Table 3). The main contribution to $L_{eq}$ inside the car corresponds to the 160Hz band (Figure 7). In the noise spectra in the hoistway, the 160Hz band is not predominant, and there are other bands that have a similar contribution to the $L_{eq}$ (Table 3).

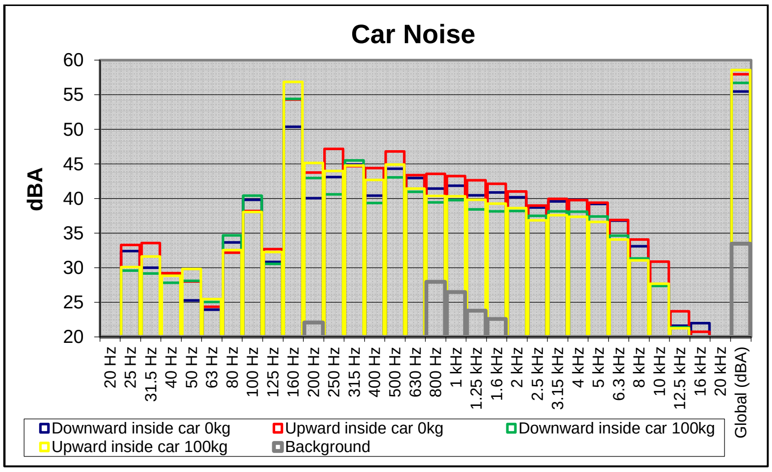



**Figure 7 1/3 Octave band noise in Car**

**Table 3 $L_{eq}$ and $L_{A\_160Hz}$ in Car (dBA)**

| | Noise Equivalent Level (dBA) | | | |
|---|---|---|---|---|
| | Downward 0kg | Upward 0kg | Downward 100kg | Upward 100kg |
| Machine | 61,4 | 63,0 | 61,8 | 62,8 |
| Above Roof | 60,7 | 62,0 | 62,2 | 62,4 |
| Inside Car | 55,5 | 58,0 | 56,7 | 58,6 |
| | Noise Level at 160Hz (dBA) | | | |
| | Downward 0kg | Upward 0kg | Downward 100kg | Upward 100kg |
| Machine | 50,4 | 48,1 | 48,9 | 48,0 |
| Above Roof | 48,6 | 51,6 | 51,0 | 52,4 |
| Inside Car | 50,4 | 54,3 | 54,4 | 56,9 |

The noise FFT spectra inside the car show that the predominant peak appears at 162Hz (6$^{th}$ harmonic of $F_N$) and this peak is the main contributor at the 1/3 octave band of 160Hz (Figure 8). The noise transmission from the hoistway to the car is obtained by deducting the levels above the car roof from the levels inside the car. Positive values (levels inside the car are higher than levels in the hoistway) denote that structure-borne has a higher contribution than airborne (Figure 9). The results show that at 160Hz the noise inside the car is mainly structure-borne. This band contains the 6$^{th}$ harmonic of power supply frequency. The noise level values at 1/3 octave band of 160Hz near the machine, above the roof and inside the car are shown in Table 3.

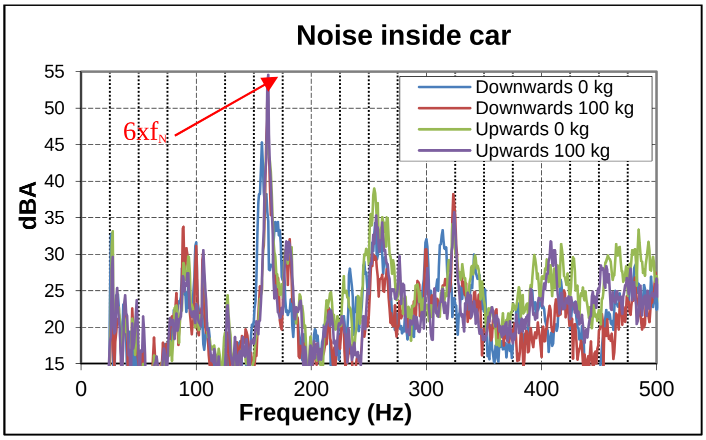



**Figure 8 FFT spectra noise in Car**

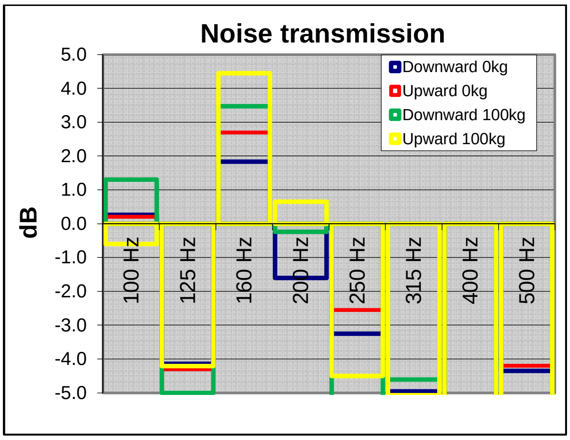



**Figure 9 Noise transmission**

The *vibration transmissibility through the load cells* between car floor and car frame is obtained by means of accelerometers placed on joint points of load cells. The vibration transmissibility (FRF – Frequency Response Function) is calculated at the main vibration peaks on car floor related to the machine vibration frequencies. The vibration transmissibility at 162Hz can be seen in table 4.

| 162 Hz | Vibration transmissibility through load cells | | | |
|---|---|---|---|---|
| | Load Cell 1 | Load Cell 2 | Load Cell 3 | Load Cell 4 |
| Downwards 0 kg | 38% | 42% | 85% | 84% |
| Downwards 100 kg | 32% | 31% | 81% | 81% |
| Upwards 0 kg | 27% | 34% | 59% | 71% |
| Upwards 100 kg | 22% | 22% | 52% | 56% |

**Table 4 Vibration transmission through Load Cells**

The *vibration measurements on the car panels* (lateral, roof and floor) are carried out with the purpose of determining whether there is any car panel that vibrates with a higher amplitude, causing structural noise transmission towards the car passengers. Accelerometers are placed at the centre of the main panels: Lateral panel, keyboard panel, roof panel and floor panel. The FFT spectra (Figure 10) show that the main vibration peaks appear at 27Hz and 162 Hz (1x$F_N$ and 6x$F_N$), especially in the roof panel. Therefore the roof vibration could be increasing the noise inside the car.

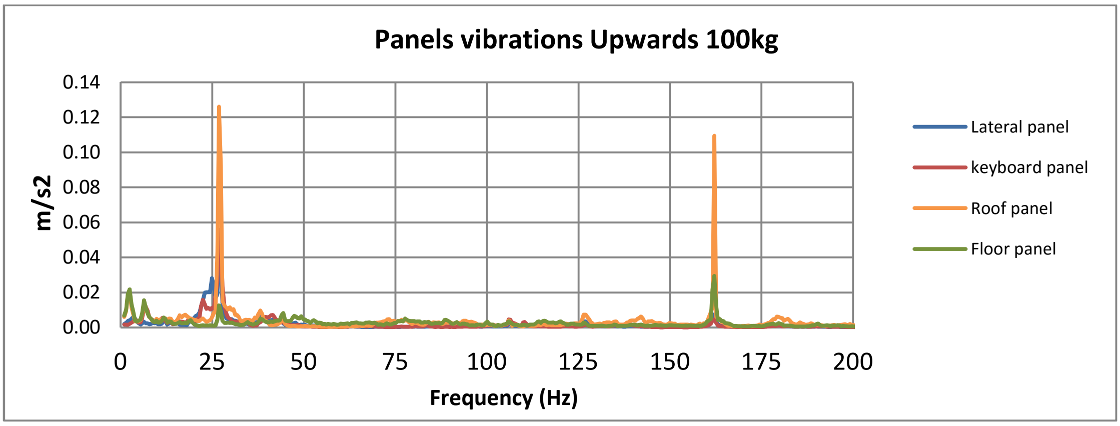



**Figure 10 FFT spectra vibrations of panels**

The *Operational Modal Analysis (OMA)* and the *Operational Deflection Shapes (ODS)* are the techniques that help understand what the vibration shape is like and to get ideas on how to reduce it. In this case it is applied to some components and panels because, based on the different vibration measurements in the lift, it was found that the machine frame and car roof have a higher vibration amplitude at 162Hz. Therefore, the machine frame and roof panel were selected to obtain their mode shapes (2-200Hz) and to check if there is any mode near to harmonics of the power supply frequency that may increase its vibration. The roof has a mode at 166Hz (Figure 11); this mode shape is near 162Hz, therefore this mode may be self-excited by the 6$^{th}$ harmonic of the power supply frequency (162 Hz). The machine frame has the 5$^{th}$ mode shape at 164.2 Hz (Figure 12), therefore this mode may also be self-excited by the 6$^{th}$ harmonic of the power supply frequency, thus increasing the vibration and noise levels in the lift car. The mode shape at 164.2 Hz can be seen in the figure below. The shapes are similar to the *Operational Deflection Shapes (ODS)* obtained during normal operation.

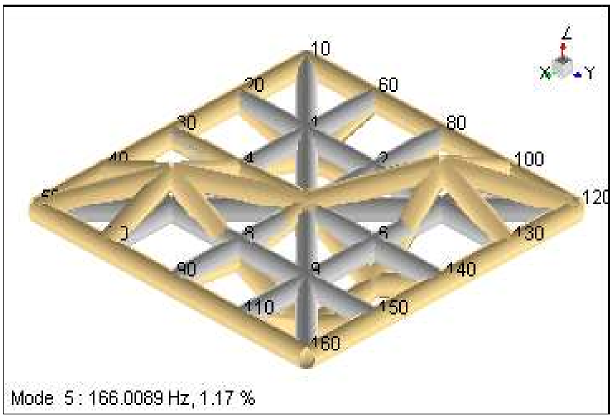



**Figure 11 Mode Shape of roof at 166Hz**

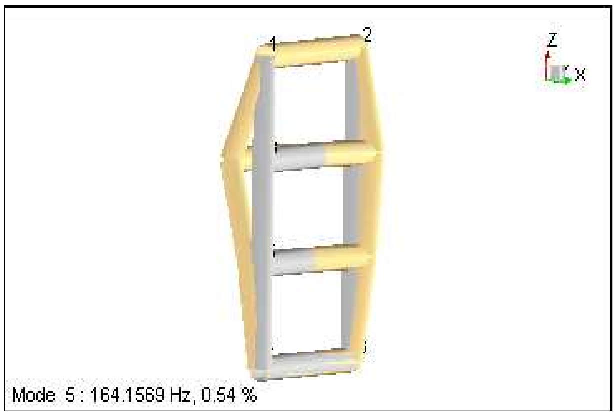



**Figure 12 Mode Shape of Machine- frame at 164 Hz**

Based on the previous analysis, the actions proposed to improve the performance, ride quality, and to decrease the noise and vibrations would be:

- Review the correct load in the counterweight.
- Reduce the friction force between the guide shoes and guide rails.

- Review the machine, trying to minimise the harmonic peaks of the power supply frequency that could affect comfort in the car.
- New design of machine frame (change the modes at 166Hz by design).
- New design of the isolators on the machine frame to work better at the $6^{th}$ harmonic of the power supply frequency. The problem is the high vibration level of the machine frame even though the isolators are well designed.
- Minimise the noise levels of the source of excitation (machine) to decrease the noise level inside the hoistway and minimise the airborne noise inside the car.
- Increase the damping of the roof panel or change its stiffness to decrease its ability to generate noise by vibrations. For instance, adding rubber sheets to the roof.
- Minimise the structure-borne noise inside the car by improving the isolation of the load cell between the car frame and floor frame.
- Improve the isolation between the machine and the guide support to decrease the noise transmission to the adjacent room.

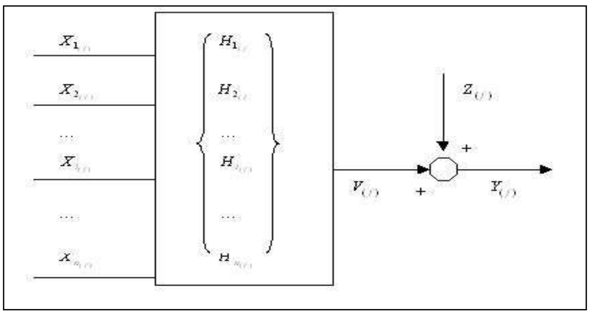
## 2.2 "Sources contributions to noise inside lift car" methodology

This methodology consists of applying techniques to identify and quantify noise and vibration sources, using frequency response and coherence functions in lifts.

The procedure consists of:

- Selecting the receptor and sources of vibration and noise.
- Obtaining the global contribution of the sources selected to the noise inside the car by means of "*Multiple Coherence*" (A function in a frequency domain expressing relationship, causality or dependence.)
- Obtaining the contribution of each source to the noise level inside the car by calculating the Transfer Matrix (H) between each source and the receptor. "*Output Power Allocation*" technique.

The multiple coherence provides a measurement of the dependency between a set of *n inputs (X1...Xn)* and *one output (Y)*, independent of the correlations among the inputs.



$$Y_{(f)} = \sum_{i=1}^{n} H_{i(f)} \cdot X_{i(f)} + Z_{(f)} = V_{(f)} + Z_{(f)}$$

$$G_{yy} = \{H\}^t \cdot [G_{xx}] \cdot \{H\}^* + G_{zz} = G_{vv} + G_{zz}$$

$$\gamma_m^2 = \frac{G_{vv}}{G_{yy}}$$

**Figure 13. Multiple Input – Single output model**

In the previous equations, $Y_{(f)}$ is the Fourier transform of the output $y_{(t)}$, $X_{i(f)}$ is the Fourier transform of the input $x_{i(t)}$, $H_{i(f)}$ is the Frequency Response Function between the input $x_{i(t)}$ and the output $y_{(t)}$, $Z_{(f)}$ is uncorrelated noise at the output, and $V_{(f)}$ is the output fraction, related to all inputs. Then the equation is transformed to a spectral matrix, $G_{yy}$ is the output power spectrum, $[G_{xx}]$ is the input spectral matrix, $G_{zz}$ is the output noise power and $G_{vv}$ is the coherent output power and $\gamma^2_m$ is the multiple coherence function.

The "*Output Power Allocation*" identifies the frequency response functions of the transmission (H). Each contribution is defined as the input (X) multiplied by the effect of the transmission path (H). The output can thus be written as a sum of contributions (amplitude and phase) from each input.

$$H_{(f)} = Y_{(f)} / X_{(f)} = \frac{Y_{(f)} \cdot X^{*}_{(f)}}{X_{(f)} \cdot X^{*}_{(f)}} = G_{xy} / G_{xx} \qquad \boxed{\{G_{xy}\} = [G_{xx}] \cdot \{H\}} \qquad \begin{pmatrix} G_{11} & G_{12} & G_{13} & \dots & G_{118} \\ G_{21} & G_{22} & G_{23} & \dots & G_{218} \\ \dots\dots & \dots\dots & \dots\dots & \dots & \dots\dots \\ G_{181} & G_{182} & G_{183} & \dots & G_{1818} \end{pmatrix} \times \begin{pmatrix} H_1 \\ H_2 \\ \dots\dots \\ H_{18} \end{pmatrix} = \begin{pmatrix} G_{1y} \\ G_{2y} \\ \dots\dots \\ G_{18y} \end{pmatrix}$$

The interpretation method is algebraic. The individual contribution and the combining effect are quantified by a triangular scalar matrix. This technique is computationally complicated; the algorithm program and visualisation is implemented in Matlab for each frequency. The visualisation shows the contribution of the different inputs (blue colour) to the output spectrum (red colour). The contribution of the sources at different frequencies can be seen in different diagrams of figure 14. These visualizations permit us to identify the sources which provide the main contributions to the output.

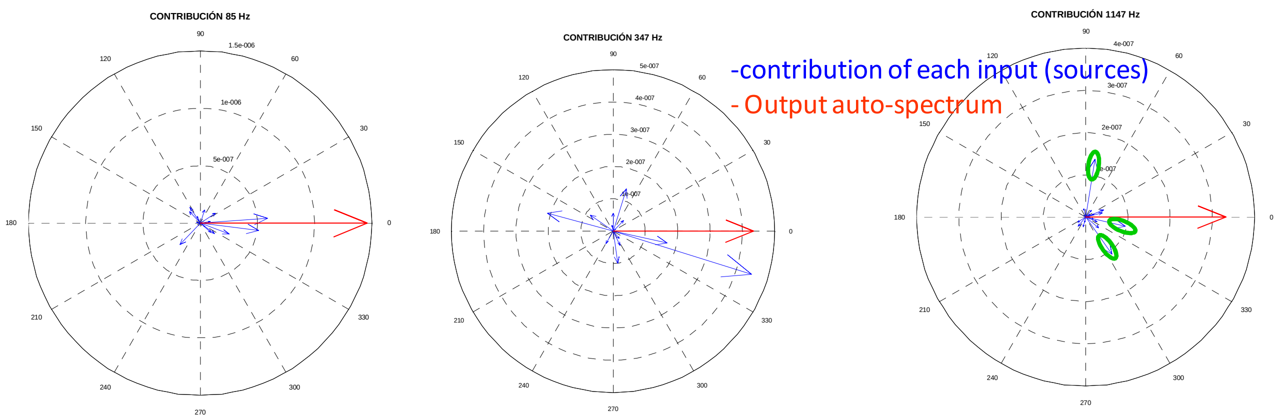



**Figure 14 Visualisation of contributions**

## 2.3 "Panel Contributions to noise level inside lift car" methodology

This methodology is based on considering that the car panels are a series of sources with their sound power. The sound power of sources is obtained by means of acceleration measurements at the different points of car panels. The pressure inside the car will be computed with the sound power sources.

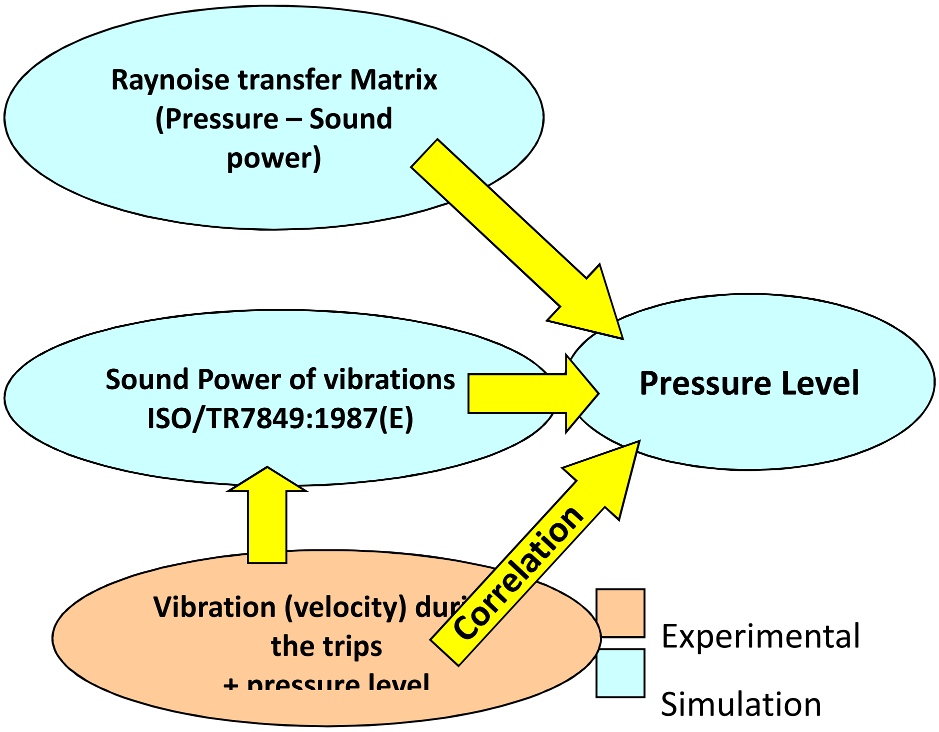



**Figure 15 Methodology summary**

The Transfer Matrix is obtained with Raynoise software. This software is based on the Mirror Image Source Method and the Ray Tracing Method.

The acoustic sources of the panels are determined from the acceleration measurements in different points of panels according to standard ISO/TR 7849 [5]. The sound power radiated by a vibrating surface may be determined by:

$$P_s = \rho c \bar{v}^2 S_s \sigma$$

$P_s$=Sound power
$\rho c$ = specific acoustic impedance of the fluid
$v^2$ = Mean square surface-averaged velocity
$S_s$ = surface area
$\sigma$ = radiation ratio

Finally, the sound pressure evaluation is calculated, and the model is correlated with the experimental results.

## 3 CONCLUSIONS

The “Vibro-Acoustic characterisation on lift” methodology enables us discover how much the different sources (vibration and noise) qualitatively affect the receptor, therefore it enables us to know which lift components need to be changed in order to achieve our goal, namely, to improve the ride quality. However, the improvements are estimated roughly.

The “Sources contributions to noise inside lift car” and “Panel Contributions to noise level inside lift car” methodologies permit obtaining quantitative results of the contribution of the sources, but their application and analysis are complex. Thus, the time spent is much greater in comparison with the first methodology.

Therefore, the first methodology is perfect for troubleshooting activities whilst the other two are more advisable for R&D activities.

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## BIOGRAPHICAL DETAILS

### Raúl Monge

Mechanical Engineer by the University of Zaragoza. His professional career in the Institute has been linked to different R&D areas, above all, solutions to noise and vibration problems in different systems and components (lifts, automotive, construction machinery, railway) with knowledge in applying different techniques, such as system dynamic response (frequency analysis), experimental modal analysis, operational modal analysis, operational deflection shapes. Noise contribution of sources, airborne and structure-borne transmission of sound. Isolation of vibration by means of passive elements and semi-active elements, including the system simulation model

### Alfredo Gómez

Alfredo Gómez graduated from the University of Zaragoza with a MEng degree in Mechanical Engineering complemented with a MS degree in Vehicle Systems Dynamics. He also holds a Master's degree in Innovation Management by the Polytechnic University of Madrid. Currently he is working at ITAINNOVA as technological innovation consultant. In recent years his career has been linked with innovation management and technology transfer. Regarding technical areas he is experienced in mechanical design, energy efficiency, noise and vibration within the field of machinery and mechanical equipment, particularly in the lift industry.