

Description:

5 **Method for the design of a regulator for vibration damping at a lift cage**

The invention relates to a method for the design of a regulator for vibration damping at a lift cage, wherein the regulator design is based on a model of the lift cage.

10 Equipment and a method for vibration damping at a lift cage has become known by the Patent Specification EP 0 731 051 B1. Vibrations or accelerations rising transversely to the direction of travel are reduced by a rapid regulation so that they are no longer perceptible in the lift cage. Inertia sensors are arranged at the cage frame for detection of measurement values. Moreover, a slower position regulator automatically guides the lift
15 cage into a centre position in the case of a one-sided skewed position relative to the guide rails, wherein position sensors supply the measurement values to position regulators.

A multivariable regulator for reducing the vibrations or accelerations at the lift cage and a further multivariable regulator for maintenance of the play at the guide rollers or the upright
20 position of the lift cage are provided. The setting signals of the two regulators are summated and control a respective actuator for roller guidance and for horizontal direction.

The regulator design is based on a model of the lift cage, which takes into consideration the significant structural resonances.

25 It is disadvantageous that the overall model has a tendency to a high degree of complexity, notwithstanding refined methods for reduction in the number of poles. As a consequence thereof the model-based regulator is equally complex.

30 Here the invention will create a remedy. The invention, as characterised in claim 1, meets the object of avoiding the disadvantages of the known method and of proposing a simple method for the design of a regulator.

Advantageous developments of the invention are indicated in the dependent patent
35 claims.

Advantageously, in the case of the method according to the invention an overall model of the lift cage with known structure is predetermined. There is concerned in that case a so-termed multi-body system (MBS) model which comprises several rigid bodies. The MBS model describes the essential elastic structure of the lift cage with the guide rollers and the actuators as well as the force coupling with the guide rails. The model parameters are known to greater or lesser extent or estimates are present, wherein the parameters for the lift cage which is used are to be identified or determined. In that case the transfer functions or frequency responses of the model are compared with the measured transfer functions or frequency responses. With the help of an algorithm for optimisation of functions with several variables the estimated model parameters are changed in order to achieve a greatest possible agreement.

Moreover, it is advantageous that the active vibration damping system of the lift cage is itself usable for the transfer functions or frequency responses to be measured. The lift cage is excited by the actuators and the responses are measured by the acceleration sensors or by the position sensors.

This model-based design method of the regulator guarantees the best possible active vibration damping for the individual lift cages with very different parameters.

It is ensured by the above-mentioned identification method that as a result the simplest and most consistent model of the lift cage is present. Advantageously the regulator based on this model has a better grade or a better regulating quality. Moreover, the method can be systematically described and can be largely automated and performed in substantially shorter time.

Based on the MBS model with identified parameters a robust multivariable regulator is designed for reduction in the acceleration and a position regulator for maintenance of play at the guide rollers.

The acceleration regulator has the behaviour of a bandpass filter and the best effect in a middle frequency range of approximately 1 Hz to 4 Hz. Below and above this frequency band the amplification and thus the efficiency of the acceleration regulator are reduced.

In the low frequency range the effect of the acceleration regulator is limited by the available play at the guide rollers and the position regulators to be designed therefor. The

position regulator has the effect that the lift cage follows a mean value of the rail profiles, whilst the acceleration regulator causes a rectilinear movement. This conflict of objectives is solved in that the two regulators are effective in different frequency ranges. The amplification of the position regulator is large in the case of low frequencies and then
 5 decreases. This means that it has the characteristic of a low-pass filter. Conversely, the acceleration regulator has a small amplification at low frequencies.

In the high frequency range the effect of the acceleration regulator is limited by the elasticity of the lift cage. The first structural resonance can occur at, for example, 12 Hz,
 10 wherein this value is strongly dependent on the mode of construction of the lift cage and can lie significantly lower. Above the first structural resonance the regulator can no longer reduce the acceleration at the cage body. The risk even exists that structural resonances are excited or that instability can arise. With knowledge of the dynamic system model of the regulator path the regulator can be so designed that this can be avoided.

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The present invention is explained in more detail on the basis of the accompanying figures, in which:

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Fig. 1 shows a multi-body system (MBS) model of a lift cage,

Fig. 2 shows a guide roller with roller forces,

Fig. 3 shows a setting element with guide roller, actuator and sensors,

25 Fig. 4 shows a schematic illustration of the regulated axes,

Fig. 5 shows the amplification of the measured acceleration and of the identified model,

30 Figs. 6 and 7 show an optimised regulator with the identified parameters for active vibration damping,

Fig. 8 shows a signal flow chart for the design of an H_∞ regulator with regulator and regulator path,

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Fig. 9 shows the course of the singular values of a position regulator in y direction,

Fig. 10 shows the course of the singular values of an acceleration regulator in y direction and

Fig. 11 shows a force signal for excitation of the actuators.

The MBS model has to reproduce the significant characteristics of the lift cage with respect to travel comfort. Since in the case of identification of the parameters it is possible to operate only with linear models, all non-linear effects have to be disregarded. The first natural frequencies of the elastic lift cage are so low that they can overlap with the so-called solid body natural frequencies of the entire cage.

As shown in Fig. 1, at least two rigid bodies are required for modelling the elastic lift cage 1, namely cage body 2 and cage frame 3. Cage body 2 and cage frame 3 are connected by means of elastomeric springs 4.1 to 4.6, the so-called cage insulation 4. This reduces the transmission of solid-borne sound from the frame to the cage body. For modelling a rigid lift cage 1 it is sufficient to consider cage body and cage frame overall as one body.

The transverse stiffness of cage body 2 and cage frame 3 is substantially less than the stiffness in vertical direction. This can be modelled by division in each instance into at least two rigid bodies, namely cage bodies 2.1 and 2.2 and cage frames 3.1 and 3.2. The at least two part bodies are horizontally coupled by springs 5, 6.1 and 6.2 and can be regarded as rigidly connected in vertical direction.

The guide rollers 7.1 to 7.8 together with the proportional masses of levers and actuators can be modelled by at least eight rigid bodies or also disregarded. This dependent on the associated natural frequencies of the guide rollers and on the upper limit of the frequency range which is considered. Since the natural frequency of the actuator/roller system can lead to instability in the regulated state, modelling by rigid bodies is preferred. These are displaceable relative to the frame only perpendicularly to the support surface at the rail and are coupled with the roller guide springs 8.1 to 8.8. In the other directions they are rigidly connected with the frame.

As is shown in Fig. 2, the guide behaviour or the force coupling between guide rollers and

guide rails is important. Substantially only the two horizontal force components are necessary for formation of the model. The vertical force components, which result from the rolling resistance, can be disregarded. The normal force results from the elastic compression of the roller covering 9.1 to 9.8. The axial or transverse force results from the angle between the straight lines perpendicular to the roller axis and parallel to the rail and the actual direction of movement of the roller centre point.

Mathematically, the following relationships are relevant:

$$F_{RA} = - \tan(\alpha) * F_{RN} * K \quad \{1\}$$

F_{RA} : rolling force in axial direction in [N]
 α : oblique running angle in [rad]
 F_{RN} : rolling force normal to the support surface [N]
 K : constant without dimension, determined by measuring

The force law {1} is at the latest invalid when the limits of the static friction force are reached as well as in the case of a large oblique running angle α . This is rapidly greater at low travel speed and at standstill amounts to approximately 90 degrees. The force law {1} thus applies only to the moving cage.

For the rolling force in axial direction with cage moving, there then approximately applies:

$$F_{RA} = - v_A / v_K * F_{RN} * K$$

$$F_{RA} = - v_A * (F_{RN} * K / v_K)$$

v_K : vertical speed of the cage [m/s]

v_A : speed of the cage in axial direction [m/s]

K is a constant and v_K and F_{RN} can be regarded as constant when the biasing force is significantly greater than the dynamic proportion of the normal force. This means that the rolling force in axial direction is proportional and opposite to the speed in axial direction and conversely proportional to the travel speed of the lift cage.

Transverse vibrations of the cage are thus damped by the rollers like a viscous damper, wherein the effect is smaller with increasing travel speed.

As shown in Fig. 3, the guide rollers 7 are connected with the cage frame 3 by a lever 10 rotatable about an axis 10', wherein the roller guide spring 8 produces a force between lever and cage frame. An actuator 11 produces a force acting parallel to the roller guide spring. A position sensor 12 measures the position of the lever 10 or of the guide roller 7.

5 An acceleration sensor 13 measures the acceleration of the lift cage 3 perpendicularly to the support surface of the roller covering 9 on the guide rail 14. The reference numerals of the respective elements apply as shown in Fig. 1 (for example, at the lift cage 1 at the bottom on the right: 7.1, 8.1, 9.1, 10.1, 11.1, 12.1, 13.1).

10 Four lower guide rollers 7.1 to 7.4 together with actuators and position sensors are provided at the lift cage 1. In addition, four upper guide rollers 7.5 to 7.8 together with actuators and position sensors can also be provided. The number of acceleration sensors 13 required corresponds with the number of regulated axes, wherein at least three and at most six acceleration sensors are provided.

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As shown in Fig. 4, for the active vibration damping of the lift cage 1 the number of axes is reduced from eight to six, or four to three axes when active regulation is only at the bottom. A triplet of signals F_{n_i} , P_{n_i} , a_{n_i} for actuator force, position and acceleration belongs to each axis A_{n_i} . The index i is the continuing numbering in the respective axial system and n stands for the number of axes of the system.

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The signals of the lower and the upper roller pair between the guide rails 14.1 and 14.2 are combined as follows: The force signal F_{6_i} for the actuators 11.1 and 11.3 or the force signal F_{6_4} for the actuators 11.5 and 11.7 is divided into a positive and a negative half.

25 Each actuator is controlled in drive only by one half and can produce only compressive force in the roller covering. A mean value is formed from the signals of the position sensors 12.1 and 12.3 and the same applies to the position sensors 12.5 and 12.7. A mean value is similarly formed from the signals of the acceleration sensors 13.1 and 13.3 or 13.5 and 13.7. Since the acceleration sensors 13.1 and 13.3 or 13.5 and 13.7 lie on one axis and are rigidly connected by the lower or upper cage frame, they in principle measure the same and in each instance one sensor of the respective pair can be omitted.

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In the case of measuring travels, one or more actuators is or are controlled in drive by a force signal as shown in Fig. 11 and the lift cage 1 is so excited to vibrations transversely the travel direction that clearly measurable signals arise in the position sensors 12 and in the acceleration sensors 13. So that the correlation of the measurements with the force

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signals can be reliably determined, usually only one actuator or actuator pair is controlled in drive. As shown in Table 1 at least as many measuring travels are necessary as active axes are provided.

5 Table 1

Excitation: one or more simultaneously	Measurements: all simultaneously	
F6 ₁	P6 ₁	a6 ₁
F6 ₂	P6 ₂	a6 ₂
F6 ₃	P6 ₃	a6 ₃
F6 ₄	P6 ₄	a6 ₄
F6 ₅	P6 ₅	a6 ₅
F6 ₆	P6 ₆	a6 ₆

The frequency spectrum of the force signals as well as the measured position signals and
 10 acceleration signals are determined by Fourier transformation. The transfer functions in
 the frequency range or frequency responses $G_{i,j}(\omega)$ at the angular frequency ω as
 argument are determined in that the spectra of the measurements are divided by the
 associated spectrum of the force signal. In that case i is the index of the measurement
 and j is the index of the force.

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$$G^P_{i,j}(\omega) = \frac{P_i(\omega)}{F_j(\omega)}$$

$$G^a_{i,j}(\omega) = \frac{a_i(\omega)}{F_j(\omega)}$$

$$G(\omega) = \begin{bmatrix} G^P(\omega) \\ G^a(\omega) \end{bmatrix}$$

20 $G^P_{i,j}(\omega)$ are the individual frequency responses of force to position and $G^a_{i,j}(\omega)$ are the
 individual frequency responses of force to acceleration. The matrix $G^P(\omega)$ contains all
 frequency responses of force to position and matrix $G^a(\omega)$ all frequency responses of

force to acceleration. The matrix $G(\omega)$ arises from the vertical combination of $G^p(\omega)$ and $G^a(\omega)$.

For a 6-axis system there thus results $2 \times 6 \times 6 = 72$ transfer functions and for a 3-axis system $2 \times 3 \times 3 = 18$ transfer functions. In the case of cages having a centre of gravity lying on the axis between the guide rails 14.1 and 14.2 the couplings and the correlation between the two horizontal directions x and y are weak. For that reason only approximately half the transfer functions is further used, the remaining being excluded due to inadequate correlation.

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The MBS model of the cage is in general a linear system. If this contains non-linear components, a fully linearised model is produced in an appropriate operational state by numerical differentiation. In the linear state space the MBS model is described by the following equations:

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$$\begin{aligned}\dot{x} &= Ax + Bu \\ y &= Cx + Du\end{aligned}$$

x is the vector of the states of the system, which in general are not externally visible. The states of the system in the present case are:

positions and speeds of the centre of gravity in the solid body model, as well as rotational angles and rotational speeds. Derivations of the states are speeds and accelerations. Speed is thus both state and derivation.

The vector $\dot{x}$ contains the derivations of x according to time. y is a vector which contains the measured magnitudes, thus positions and accelerations. The vector u contains the inputs (actuator forces) of the system. A, B, C and D are matrices which together form the so-termed Jacobi matrix by which a linear system is completely described. The frequency response of the system is given by

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$$\hat{G}(\omega) = D + C(j\omega I - A)^{-1} B.$$

$\hat{G}(\omega)$ is a matrix with the same number of lines as measurements in the vector y and the same number of columns as inputs in the vector u and contains all frequency responses of the MBS model of the cage.

- 5 A Jacobi matrix contains all partial derivations of a system of equations. In the case of a linear system of coupled differential equations of 1st order, these are the constant coefficients of the A, B, C and D matrices.

10 The model contains a number of well-known parameters such as, for example, measurements and masses and a number of poorly known parameters such as, for example, spring rates and damping constants. It is necessary to identify these poorly known parameters. The identification is carried out in that the frequency responses of the model are compared with the measured frequency responses. The poorly known model parameters are changed by an optimisation algorithm until the minimum of the sum e of
15 all deviations of the frequency responses of the model is found by the measured frequency responses.

$$e_{i,j}(\omega) = \frac{|G_{i,j}^{\wedge}(\omega)| - |G_{i,j}(\omega)|}{\sqrt{|G_{i,j}(\omega)|}} \cdot w(\omega)$$

$$e = \sum_i \sum_j \sum_{\omega} [e_{i,j}(\omega)]^2$$

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$w(\omega)$ is a weighting dependent on frequency. It ensures that only important components of the measured frequency responses are simulated in the model.

25 An optimisation algorithm can be briefly circumscribed as follows: A function with several variables is given. A minimum or maximum of this function is sought. An optimisation algorithm seeks these extremes. There are many various algorithms, for example the method of fastest degression seeks the greatest gradients with the help of the partial derivations and rapidly finds local minima, but for that purpose can pass over others. Optimisation is a mathematical procedure used in many fields of expertise and an
30 important area of scientific investigation.

Fig. 5 shows the frequency-dependent amplifications of the acceleration measured and of the identified model. $|G_{1,1}^a|$ means amount or amplitude of the transfer function or of the frequency response of force to acceleration with the output acceleration from axis 1 and with the input force from axis 1. Dimension: $1 \text{ mg/N} = 1 \text{ milli-g/N} = 0.0981 \text{ m/s}^2/\text{N} \sim 1$
 5 $\text{cm/s}^2/\text{N}$.

Fig. 11 shows the force signal for excitation of the actuators 11. The excitation is carried out by a so-termed random binary signal, which is produced by means of a random generator, wherein the amplitude of the signal can be fixedly set, for example to $\pm 300 \text{ N}$,
 10 and the spectrum is widely and uniformly distributed.

The model with the identified parameters forms the basis for the design of an optimum regulator for active vibration damping. Regulator structure and regulator parameters are dependent on the characteristics of the path to be regulated, in this case on the lift cage.
 15 The lift cage has a static and dynamic behaviour which is described in the model. Important parameters are: masses and mass inertia moments, geometries such as, for example, height(s), width(s), depth(s), track size, etc., spring rates and damping values. If the parameters change, then that has influence on the behaviour of the lift cage and thus on the settings of the regulator for vibration damping. In the case of a classic PID
 20 regulator (Proportional, Integral and Differential regulator) three amplifications have to be set, which can be readily managed manually. The regulator for the present case has far above a hundred parameters, whereby a manual setting in practice is no longer possible. The parameters accordingly have to be automatically ascertained. This is possible only with the help of a model which describes the essential characteristics of the lift cage.

25 The regulation shown in Fig. 6 is divided into two regulators connected in parallel: A position regulator 15 and an acceleration regulator 16. Other structures of the regulation are also possible, particularly a cascade connection of position regulator and acceleration regulator as shown in Fig. 7. The regulators are linear, time-invariant, time-discrete and they regulate several axes simultaneously, hence the designation MIMO for Multi-Input,
 30 Multi-Output. n is the continuing index of the time step in a time-discrete or 'digital' regulator.

The updated states $x(n+1)$ for the next time step are calculated so that they are available
 35 there.

A dynamic system is time-invariant when the described parameters remain constant. A linear regulator is time-invariant when the system matrices A, B, C and D do not change. Regulators realised on a digital computer are always also time-discrete. This means they make the inputs, calculations and outputs at fixed intervals in time.

The so-termed H_∞ method is used for the regulator design. Fig. 8 shows the signal flow chart of the H_∞ design method with closed regulating loop. The main advantage of the H_∞ design method is that it can be automated. In that case the H_∞ standard of the system to be regulated is minimised by closed regulating loop. The H_∞ of a matrix A with $m \times n$ elements is given by:

$$\|A\|_\infty = \max_i \sum_{j=1}^n |a_{i,j}| \quad (\text{maximum 'lines sum'})$$

The system to be regulated is the identified model of the lift cage 1 with the designation P for plant as shown in Fig. 8. The desired behaviour of the regulator K with the reference numeral 17 is produced with the help of additional weighting functions at the input and output of the system.

- w_v models the interferences in the frequency range at the input of the system
- w_r is a small constant value
- w_u limits the regulator output
- w_y has the value one

Fig. 8 is a diagram for the design of the regulator by the H_∞ method. w is the vector signal at the input and is composed of v and r . z is the vector signal at the output, wherein $z = T \cdot w$. T is composed of regulator, regulating path and weighting functions. P_6 or a_6 forms the feedback in the closed regulating loop, in the case of separate design of position regulator or of acceleration regulator. F_6 is the output or the setting signal of the regulator. The H_∞ standard is minimised by $\|z\|_\infty / \|w\|_\infty = \|T\|_\infty$. For that purpose there is again necessary an optimisation algorithm which changes the parameters of the regulator until a minimum has been found.

Fig. 9 shows the course of the singular values of a position regulator in y direction. This has predominantly an integrating behaviour.

Fig. 10 shows the course of the singular values of an acceleration regulator in y direction.

5 This has a bandpass characteristic.

Singular values are a measure for the overall amplification of a matrix. An $n \times n$ matrix has n singular values. Dimension: $1 \text{ N/mg} = 1 \text{ N/milli-g} = \text{N}/(0.0981 \text{ m/s}^2) \sim 1 \text{ N}/(\text{cm/s}^2)$.

Patent Claims:

1. Method for the design of a regulator for vibration damping at a lift cage (1), wherein
5 the regulator design is based on a model of the lift cage, characterised in that an overall model of the lift cage (1) is used with model parameters which are known to greater or lesser extent or estimated, wherein the parameters for the lift cage which is used are identified by comparison of the transfer functions or the frequency responses of the model with the measured transfer functions or the measured frequency responses and the model
10 parameters are changed in order to achieve the greatest possible correspondence with the measured frequency responses, wherein the model together with the identified parameters serves as a basis for the design of an optimum regulator for active vibration damping.
2. Method according to claim 1, characterised in that the active vibration damping
15 system of the lift cage (1) is itself provided as measuring equipment for the transfer functions or frequency responses to be measured, wherein the lift cage (1) is excited by means of actuators (11) and the responses are measured by means of acceleration sensors (13) or by means of position sensors (12).
- 20 3. Method according to claim 1 or 2, characterised in that the model parameters are changed by means of an optimisation algorithm until the minimum of the sum (e) of all deviations of the frequency responses of the model from the measured frequency responses is found.
- 25 4. Method according to claim 3, characterised in that the deviations between the frequency responses of the model and the measured frequency responses are weighted by a frequency dependent value $w(\omega)$ in the calculation of the sum (e).
5. Method according to one of the preceding claims, characterised in that the
30 regulator (17) is designed with the help of the H_∞ method.
6. Method according to claim 5, characterised in that the regulator (17) comprises a position regulator (15) which controls the actuators (11) in drive in dependence on the position of the lift cage (1), wherein the guide elements (7) adopt a predetermined position,
35 and that the regulator (17) comprises an acceleration regulator (16), which controls the

actuators (11) in drive in dependence on the acceleration of the lift cage (1), whereby vibrations occurring at the lift cage (1) are suppressed.

7. Method according to claim 6, characterised in that the position regulator (15) and
5 the acceleration regulator (16) are connected in parallel, wherein the setting signals of the position regulator (15) and the acceleration regulator (16) are added and supplied to the actuators (11) as a summation signal.

8. Method according to claim 6, characterised in that the position regulator (15) and
10 the acceleration regulator (16) are connected in series, wherein the setting signal of the position regulator (15) is fed to the acceleration regulator (16) as an input signal.

9. Method according to one of claims 6 to 8, characterised in that the position
15 regulator (15) and the acceleration regulator (16) are effective substantially in different frequency ranges.

10. Method according to one of the preceding claims, characterised in that the multi-
body system (MBS) model for an elastic lift cage comprises at least two bodies describing
the cage body (2) as well as the cage frame (3) or for a rigid lift cage (1) comprises cage
20 body (2) and cage frame (3) overall as one body.