IMPROVING RIDE QUALITY IN HIGH SPEED ELEVATORS

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ABSTRACT. This paper presents a set of proposals for improving ride quality in high speed lift systems. The paper starts with a brief description of a mathematical model of the lift system which is used for analytical studies and as a design tool for the study of vibration reduction strategies. An overview of the causes and sources of vibrations is presented, and three ways for improving ride quality in lift systems are proposed. The first is an active suspension system to control resonance in the car/sling isolation, the second is the use of an advanced control strategy to implement car velocity feedback and thereby control the rope modes, and finally non-contacting electromagnetic guide shoes to reduce lateral disturbances in the lift car.

1. INTRODUCTION

The advent of modern high rise building has brought about the need for high speed lift systems to provide quick access within these buildings and one of the main problems in lifts is a loss of ride quality caused by vibration. This study is motivated by the expectation of ever-increasing performance requirements, and the recognition that advanced control techniques may be a cost-effective way of meeting these future requirements.

Ideas on improving ride comfort have been published before, ranging from the use of modal analysis to study the structure of the lift car[1] through to implementation of an advanced control strategy[2]. The problem of reduced ride quality due to vibration is tackled here from the basic level starting with the modelling of the system including its disturbances. This model is used to study the rigid body mechanical resonant modes in the system and the stability of the velocity control loop. Various theoretical studies for improving ride quality were also carried out using this model.

2. SYSTEM DESCRIPTION AND MODEL

The general structure of high speed lift systems is well known, and the particular system studied in this paper is shown in Figure 1. The dynamic arrangement of the complete system can be divided into three sections; these can be thought of separately, but are dynamically coupled. Firstly, there is the electrical drive system which includes closed loop control of the velocity of the main sheave; the second part comprises the ropes and the various masses which they interconnect, the characteristics of which are dependent upon the position of the lift in the shaft. Finally, there is the set of isolation pads which joins the car to the sling.

2.1 Mathematical Model

The ropes (hoist and compensation) play an important role in the dynamics of the lift system because they largely determine the resonances within the system and therefore the amount of vibration in the lift car. The main problem encountered in modelling the ropes is that for a rope to have its mass included in the model, it should ideally be modelled as a distributed parameter

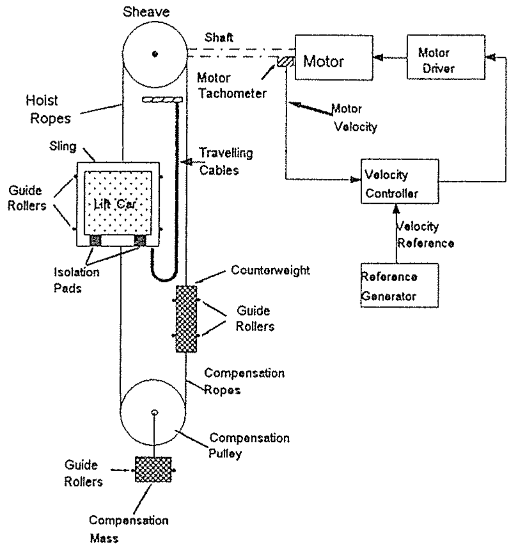


Figure 1 - High Speed Lift System

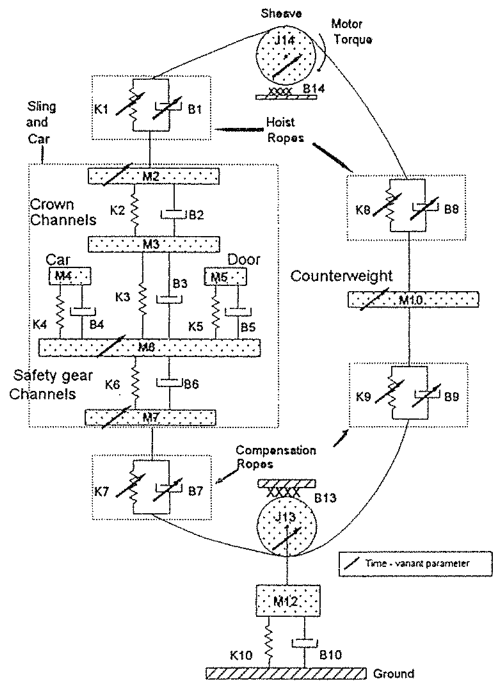


Figure 2 - Mechanical Model of Lift

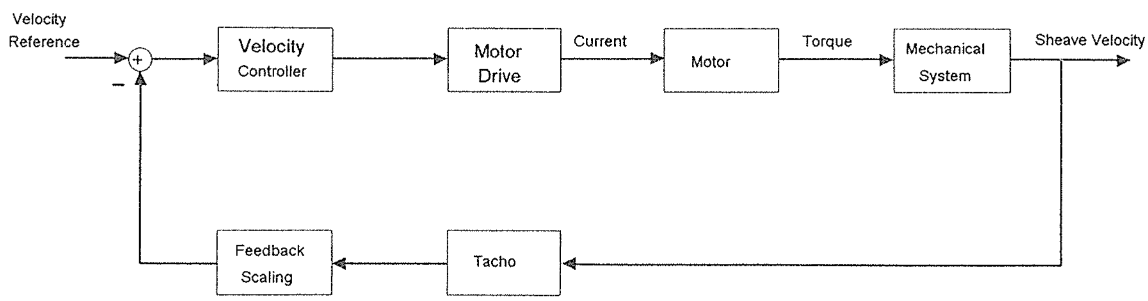


Figure 3 - Block Diagram Representation of the High Speed Lift System

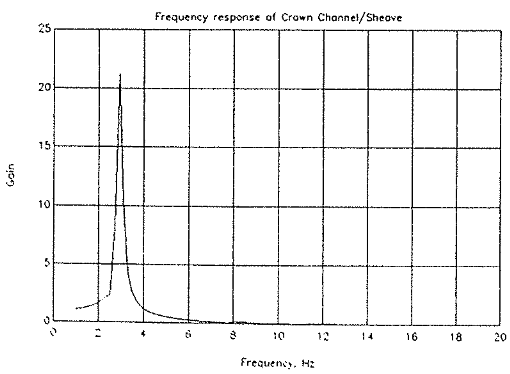


Fig. 4(a) - Rope modes

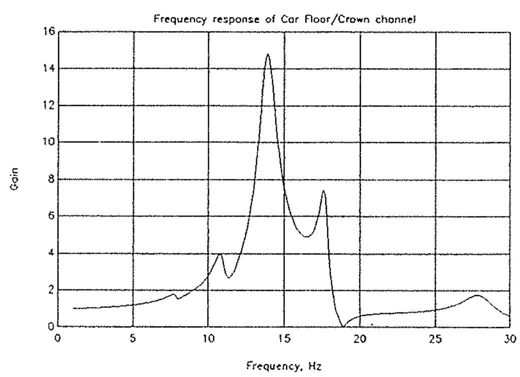


Fig. 4(b) - Isolation pads

Figure 4 - Frequency Response of Lift System (Load: No-load, Position: Mid-shaft)

system using a finite element approach. However this leads to a great deal of complexity, so one aim of the modelling work was to develop a model using lumped parameters which would predict the frequency and time response of the vertical dynamics of the system with acceptable accuracy. A representation of the mathematical model is shown in Figure 2: the model consists of 10 masses with 10 springs and dampers and thus 20 first order differential equations are required to describe its rigid body dynamics. This dynamic model has been validated by experimental testing on a full size lift[3].

The electrical elements of the lift system consist of a velocity controller, a motor drive and a 57 kW d.c. motor. It is important to model the dynamics of the velocity controller as they affect the response of the entire system, although the model of the drive itself is simplified on the assumption of fast-acting current control, which means that the torque constant is the only motor parameter of relevance.

A block diagram showing the complete model of the lift system is seen in Figure 3, where all the mechanical elements are represented by a block in the velocity control loop. Two types of computer simulation were carried out. The first used frequency response analysis to study the variation (magnitude and frequency) in the resonant modes for different lift positions and loads. Secondly, time response simulation was undertaken to investigate the effects of vibration as the lift operates under normal conditions. The frequency response analysis was carried out by linearising the model for different operating conditions (load and position). For the time response simulation a software package the Advanced Continuous Simulation Language (ACSL) was used, and this meant that the time-variant parameters associated with the rope dynamics could be properly incorporated.

2.2 Experimental Work

The result of the modelling and experimental work on the lift system shows that resonances occur in two areas: the ropes and the isolation pads, both of which exhibit large resonant peaks with the result that any random excitation inputs will be amplified at the resonance frequency. Figures 4(a) & (b) show the frequency response between the sling and the sheave, and between the car floor and the sling respectively. The resonant frequency of the rope system is at 3 Hz and the isolation pads at 14 Hz. Note that both these change with passenger load and lift position.

Analysis of the acceleration records on the car floor reveals that most of the random disturbances on the car floor are concentrated at the resonant frequencies of the hoist ropes and the isolation pads. Figure 5(a) shows the acceleration on the car floor during a normal top to bottom journey, and the Power Spectral Density (p.s.d.) of the disturbances during the constant velocity phase of the journey is presented in Figure 5(b). The figure-of-merit used to assess ride quality is the r.m.s. value of the random acceleration during the constant velocity phase of the journey, which for the journey shown in Figure 5 is 0.0048 g_{rms}.

When comparing the p.s.d. of the disturbances for the journey shown in Figure 5 against the frequency responses presented in Figure 4, note that the p.s.d. is taken over a time interval where the dynamics of the lift system have varied, whereas the frequency response presented in Figure 4 is for a particular operating condition. Thus the modes that appear in the p.s.d. do not all occur at the same time but over a time interval, an example being the mode at 8 Hz which is due to the compensation rope when the lift is not at mid-shaft position. The concentration of disturbance energy in the 20 Hz region is due to a structural mode in the sling.

3. CAUSES OF VIBRATION

Vertical vibrations in the lift car can be associated with the resonant modes of the rope and isolation pads of the lift system being excited by random excitation from the drive machinery.

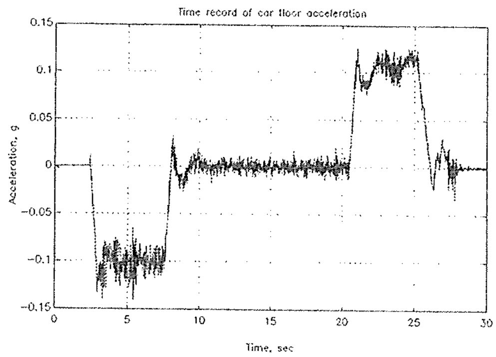


Fig. 5(a) - Car floor acceleration

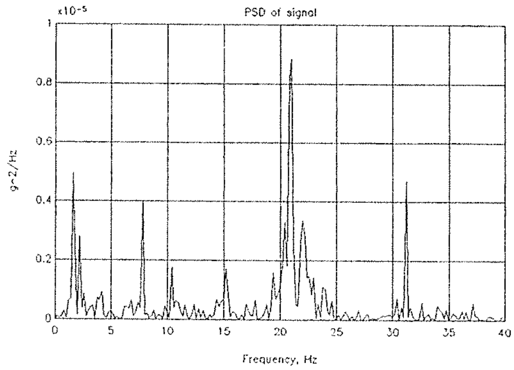


Fig.5(b)-P.s.d. of car floor acceleration

Figure 5 - Analysis of Random Disturbance on Car Floor, No-load (0.0048 g_{rms})

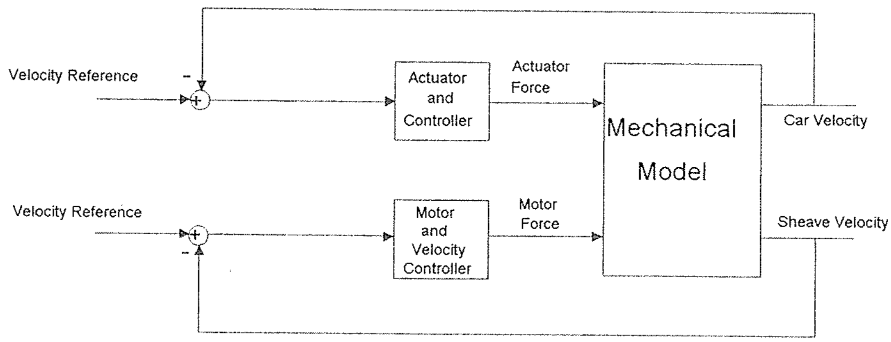


Figure 6 - Block diagram of Lift System with Modified Skyhook Damping

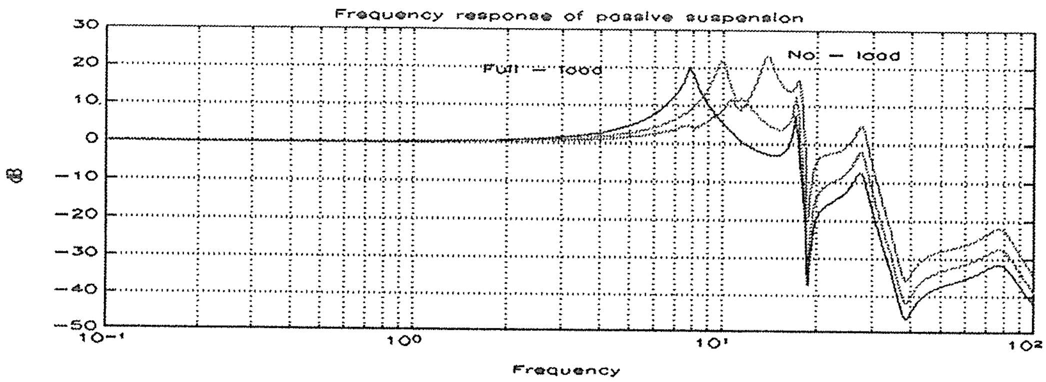


Figure 7 - Frequency Response Between Car Floor and Sling With Passive Suspension (lift at middle of shaft)

Lateral modes are also known to affect ride quality in the vertical direction. The cause of vibration in the horizontal direction is due mainly to disturbance input from the guide rails and guide rollers. Thus one important physical requirement is to have the guide rails as smooth and straight as possible, and the rubber guide rollers maintained at a high level.

The main source of vertical random disturbance is the drive machinery, with secondary inputs from the guide rollers and from the aerodynamic effect of the lift car. Vibration originating from the drive machinery is transmitted through the ropes and appears as vertical vibration on the lift car. Present vibration reduction methods concentrate on the reduction of random disturbance inputs to the lift system, but there is a limit to the amount of reduction in random disturbances these methods can yield.

Controlling the dynamics of the lift system is an alternative scheme for improving ride quality. With the knowledge that random disturbances felt by passengers are concentrated at the resonant frequencies of the ropes and isolation pads, the obvious solution would be to reduce the resonant peaks, thus reducing the amount of amplification of random disturbances at these frequencies. The low damped rope modes can be controlled with the use of *car velocity* feedback instead of sheave velocity feedback. The resonance due to the isolation pads can be reduced with the aid of active suspension strategies. Non-contacting electromagnetic guide shoes can be used to reduce the random disturbance inputs from the guide rails, and this also reduces the accuracy requirements for the guide rails and rollers.

A patent taken out by Kone Elevators [4] addresses the problem of vertical vibrations with the use of active damping but with a different control strategy to the one presented in this paper. A strategy for controlling non-contacting electromagnetic guide shoes has been proposed in a patent by the Otis Elevator Company[5].

4. ACTIVE SUSPENSION

The term " active suspension " simply means that the suspension (the isolation pads in the case of a lift system) includes some kind of controllable actuator which is able to supply energy to the system, instead of just storing or dissipating energy as in the case of a passive suspension. The way the energy in the active suspension is regulated depends on the type of control strategy and actuator configuration used.

4.1 Active Damping Strategy

Various strategies have been investigated and comparisons made in terms of ride quality improvement and force requirement by the actuator. The most promising strategy was a modified version of the "sky-hook" damper[6], which can provide extra damping to the system without increasing high frequency transmissibility. The purpose of active damping is to reduce the amplification at the resonant frequency of the isolation pads. The actuator is connected in parallel with the rubber isolation pads, and is controlled such that the pads take the static load in the car and the actuator force provides the extra damping. Consequently, there is no change in the natural frequency of the suspension nor in the static deflection characteristics.

The extra damping force is realised by generating an actuator force proportional to the difference between the absolute velocity of the car and the reference, thus creating a type of skyhook damping. A bandpass filter is also included in the active suspension controller to remove low frequency and d.c. components in the force output as these do not contribute to reducing vibrations. A block diagram of the system with modified skyhook damping is shown in Figure 6. The lift system has two control loops, the motor velocity control loop and the active suspension control loop.

4.2 Computer Simulation Results

Computer simulation was used to predict the improvement in ride quality with an active damping strategy. The assessment was based on three criteria, namely, the reduction in the resonant peak in the frequency response of the suspension, the reduction in random vibration on the car floor due to random disturbances in the system, and the actuator force requirement. Figures 7 & 8 show the predicted frequency response between the car floor and the sling with and without active damping; the magnitude of the resonant peak is significantly reduced with active damping.

The random disturbance input to the actual system was measured on the sling with an accelerometer (100 Hz bandwidth) and found to have a value of 0.012 g_{rms}. This value was used in the simulation to predict the improvement in ride quality with an active suspension. Table 1 presents the value of the random acceleration on the car floor and the actuator force requirement due to a random input on the sling. The random input used in the simulation is a white Gaussian noise source band-limited to 100 Hz with a single pole low pass filter and with a known r.m.s value. The p.s.d. of the disturbance on the car floor is shown in Figure 9 which presents the distribution of the disturbance energy. Note that there is a large reduction in vibration in the 14 Hz region, but not around the rope mode at 3 Hz because it is unaffected by the actuator. Computer simulation results suggest that better ride quality with fifty percent or more reduction in random vibration can be achieved with the use of an active suspension system.

4.3 Implementation Outline

Implementation of the active damping strategy involved the installation of a force actuator between the car floor and the bottom channels of the sling, together with the construction of an electronic controller to implement the active damping strategy. The force actuator is required to deliver a peak force of ± 1700 Newtons and an operating bandwidth of 50 Hz. An electromagnetic actuator was chosen, mainly for its robustness and high bandwidth.

4.4 Electromagnetic Actuator

The arrangement of the electromagnetic actuator is shown in Figure 10, consisting of 2 back-to-back U-shaped electromagnets sharing the same back iron to form a H-shaped core. The 4 limbs each carry a coil and there are two armatures which are connected together to give an actuator capable of generating bidirectional forces. The core of the electromagnet is connected to the sling and the armatures to the car floor. The coils on the top U-core (upper limbs, coils 1 and 2) are connected in series and will excite flux path A when current is passed through the coils, thus generating a downwards force on the car. The same principle applies when the bottom U-core (coils 3 and 4) is energised but this will generate an upwards force. The core and armatures of the electromagnet are constructed of laminated transformer steel sheets and the coils are wound with copper wire. The size and weight of the electromagnetic actuator is dependent on the peak force requirement, and for this particular design the entire actuator weighs 32 kg.

The electromagnetic actuator by itself is a non-linear device because the force output is a function of the square of the flux density in the air-gap. Also, with constant excitation the flux density in the air-gap is inversely proportional to the length of the air-gap. Due to the non-linear nature of the electromagnetics, it is difficult to control the force output of the actuator directly with the input voltage. A block diagram of the strategy used for controlling the force output of the actuator is shown in Figure 11. A flux feedback loop is used to maintain the flux density in the air-gap at the demand value over the operating range of the air-gap size. This flux control loop uses a PI controller to obtain a high closed loop bandwidth of about 60 Hz. Since there is a square relationship between flux density and output force, it is possible to establish linear control over flux density in the air-gap by having a "square root circuit" to convert force demand to flux density demand as the input to a flux control loop.

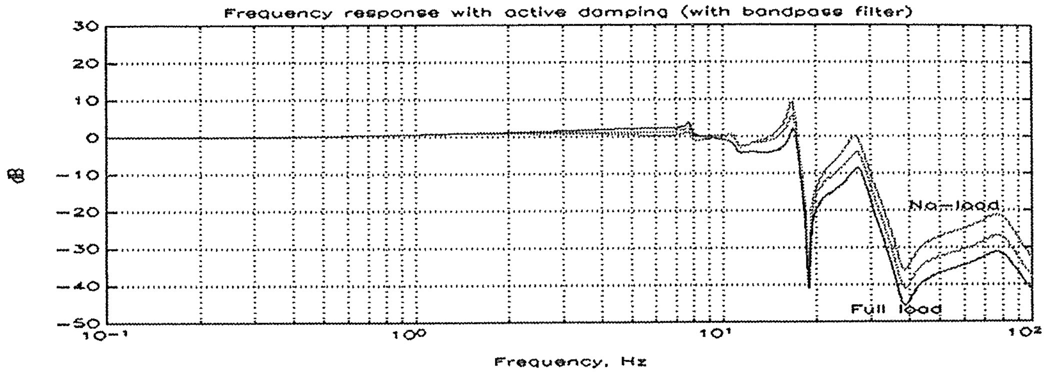


Figure 8 - Frequency Response Between Car Floor and Sling With Active Damping (lift at middle of shaft)

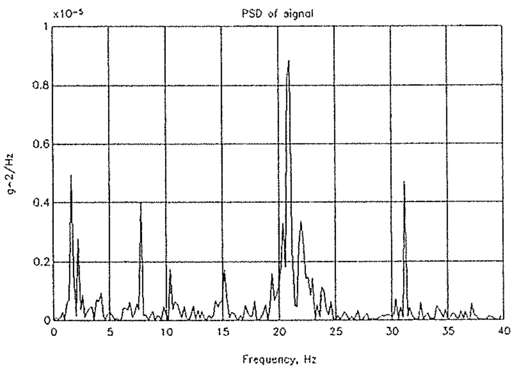


Figure 9a - With isolation pads (Measured)

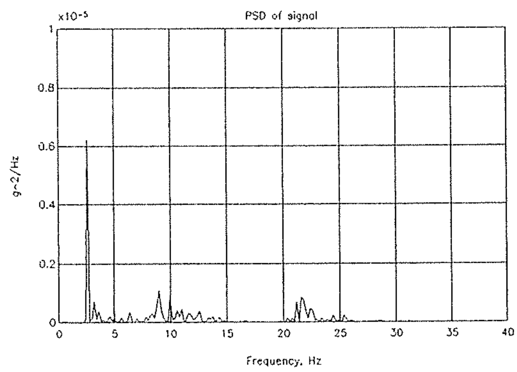
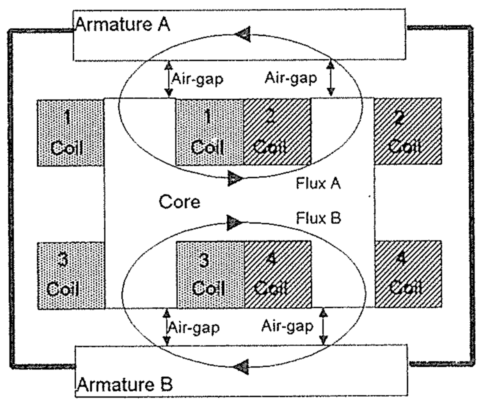


Figure 9b - With active damping (Simulated)

Figure 9 - P.s.d. of Disturbance on Car Floor due to 0.012 g_{rms} Input on Sling (no-load)



Armatures connected to car floor
Core connected to sling

Figure 10 - Cross Sectional View of the Electromagnetic Actuator (Side Elevation)

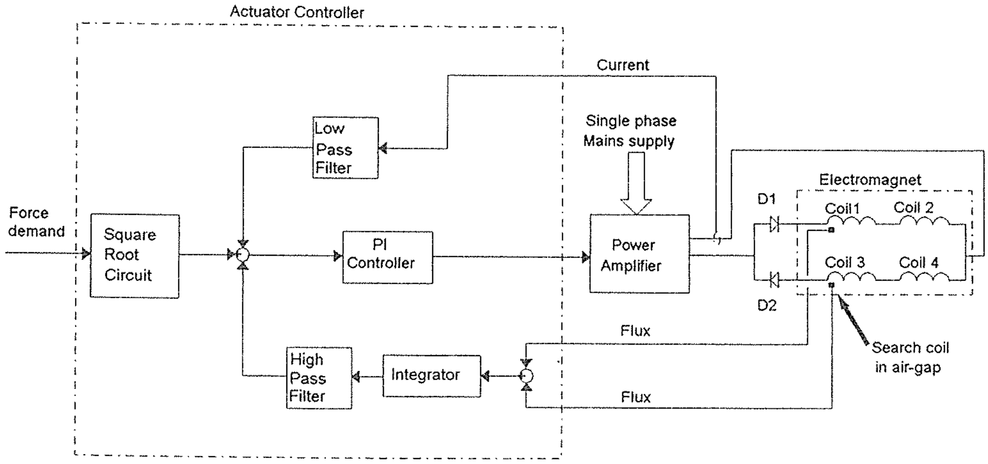


Figure 11 - Electromagnetic Actuator's Control Elements

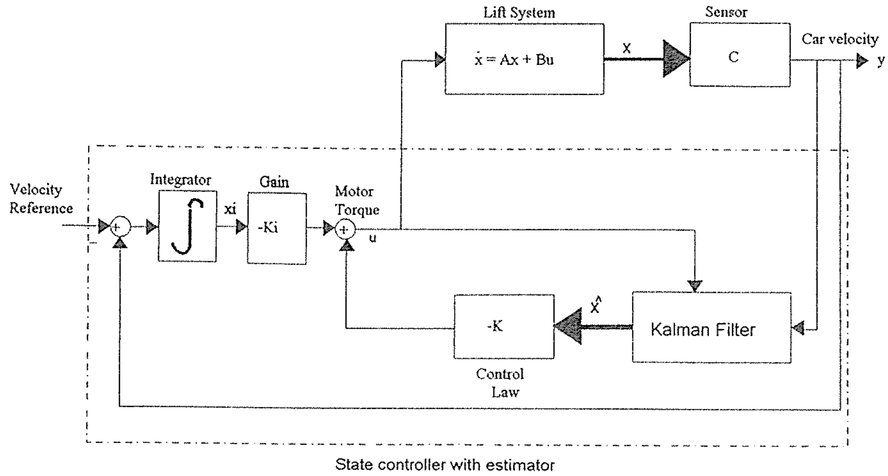


Figure 12 - Car Velocity Control Using State Feedback

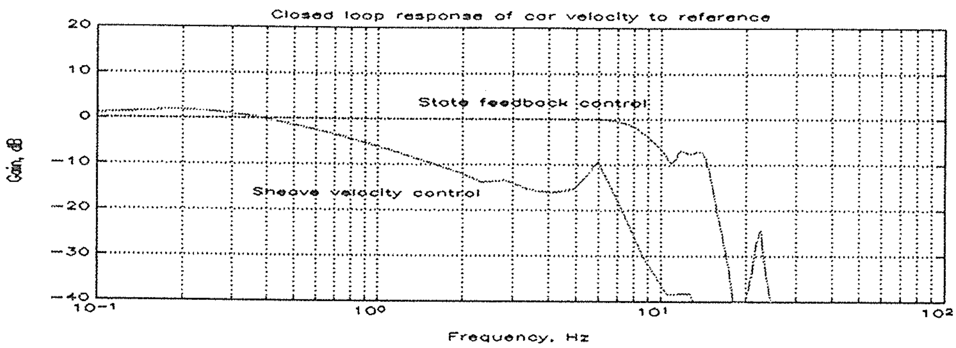


Figure 13 - Closed Loop Frequency Response of Car Velocity to Reference (mid-shaft, half load)

Implementation of this control strategy for the actuator requires accurate sensing of the flux density in the air-gap. Search coils embedded into the pole faces of the actuator are used to measure the flux density in the air-gap. Hall sensors can also be used but they are expensive and not as robust. A search coil works on the principle of electromagnetic induction and produces an output voltage that is proportional to the rate of change of the flux density in the air-gap; air-gap flux density is obtained by integrating this induced voltage. The disadvantage of using a search coil is that it is not able to measure flux density at very low frequencies, causing the flux density feedback loop to be effectively open at these frequencies. This is overcome by using a low frequency current feedback to give closed loop control at low frequencies where the search coil is ineffective. The implementation of the search coils and the current feedback loop enables the force output to be controlled from d.c. upwards. The active suspension strategy does not demand any d.c. forces, thus the current feedback is used mainly to remove drift within the flux loop.

The control elements for the electromagnet shown in Figure 11 consist of search coils in all the pole faces to measure the flux density in the air-gap, measurement of the coil's current, two filters and a PI controller, a square root circuit to convert force demand to flux demand, and a power amplifier to drive the coils.

5. CAR VELOCITY CONTROL

The current method used for controlling the velocity of the lift car is to control the velocity of the sheave. This is not the ideal way as there is a rope with elastic properties between the car and the sheave. The bandwidth of the sheave velocity controller must be kept low (under 1 Hz) to prevent exciting rope resonance, thus there is no high bandwidth control on the lift car. However, if velocity feedback can be obtained from the lift car it would be possible to control the car velocity to a higher bandwidth thus maintaining some control over the dynamics of the ropes. This would reduce the amplification of random disturbances by the rope modes.

Car velocity control is extremely difficult to achieve using classical control methods due to a large phase shift in the open loop frequency response between the car velocity and velocity reference, caused by the elastic hoist ropes. Also, the fact that the system is time variant poses further constraint on such a controller. A state feedback car velocity controller would overcome the limitations of a classical controller in terms of stability and robustness.

A method of car velocity control using state feedback with integral control[7] is shown in Figure 12. A state feedback controller requires feedback of all the system states (x), which is achieved with the use of a state estimator, a Kalman filter[8] in this application. A new state, x_i is created, which is the integral of the error between the measured car velocity and the velocity reference. The feedback gain matrix is obtained using optimal methods, in this case minimising a cost function by solving the Riccati equation. The mathematical description of this controller has been left out here but it has been published in [9].

5.1 Computer Simulation Results

Computer simulation of the above control strategy shows that it is possible to achieve direct control of the car velocity up to 7 Hz. This bandwidth is chosen because it is above the rope modes. Figure 13 presents the closed loop frequency response of car velocity to velocity reference, and the control bandwidth has been greatly improved. Simulation results also predict that the state controller is robust enough to cope with the variation in system parameters.

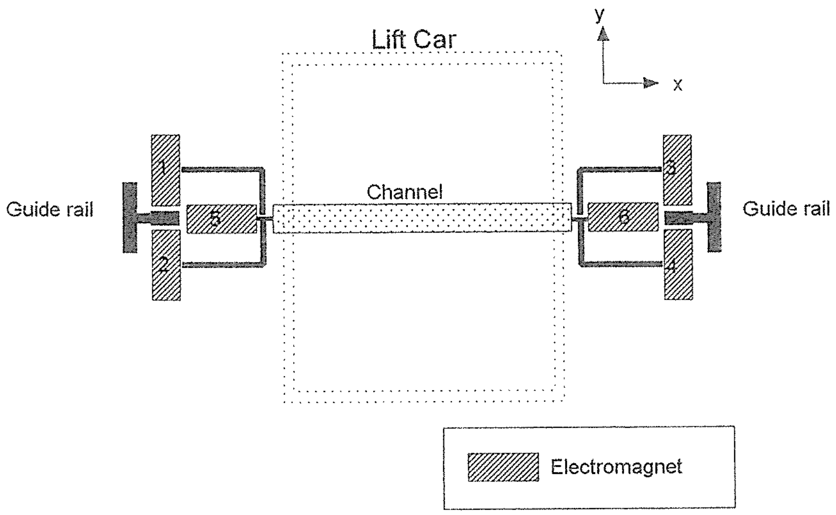


Figure 14 - Configuration for Non-Contacting Electromagnetic Guide Shoes (Top Elevation)

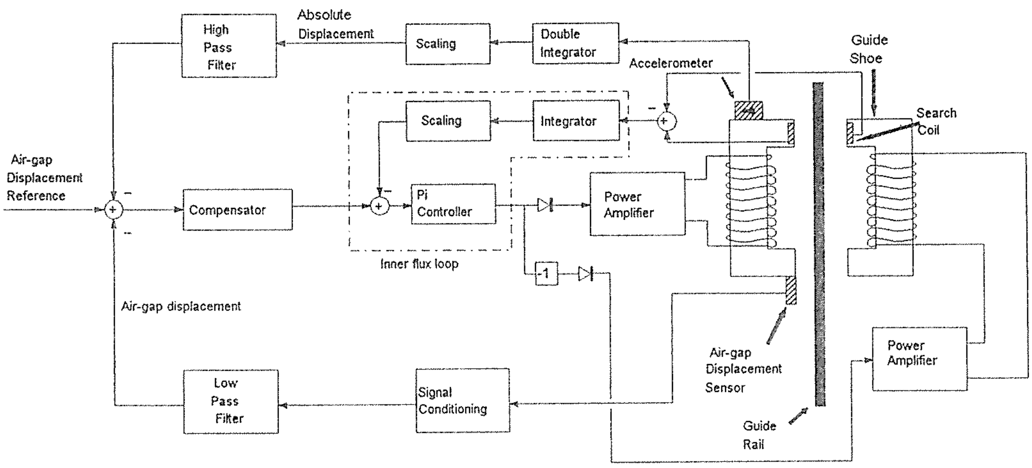


Figure 15 - Control Strategy for a Pair of Electromagnetic Guide Shoes

	No load	Balanced load	Full load
Measured (g_{rms})	0.0048	0.0038	0.0045
Simulated (g_{rms})	0.0054	0.0042	0.0025
Predicted with active damping (g_{rms})	0.0024	0.0019	0.0014
Actuator Force Requirement for Active Damping			
Peak (Newtons)	850	1074	1709
r.m.s. (Newtons)	185	270	434

Table 1 - Random Acceleration on the Car Floor and Actuator Force Output Due to 0.012 g_{rms} Disturbance Input on the Sling

6. NON-CONTACTING GUIDE SHOES

A strategy for implementing non-contacting electromagnetic guide shoes onto standard T-section guide rails is presented in Figure 14. This strategy uses three pairs of electromagnets (numbers 1&2, 3 & 4, and 5 & 6) to guide one end of the lift car (i.e. top or bottom); a similar set would be used to guide the other end of the lift car. The lift car is held in position by controlling the attractive force generated by each electromagnet. Forces acting on the lift car in the positive y-direction are controlled by electromagnet numbers 1 & 3, while electromagnet numbers 2 & 4 react to forces in the negative y-direction. Forces in the x-direction are controlled likewise using electromagnet numbers 5 & 6.

Each pair of electromagnets is capable of producing bidirectional forces, and is controlled individually using its own sensors and controllers. The control strategy for a pair of electromagnets is shown in Figure 15. This control strategy requires the use of 3 sensors: an accelerometer to measure the lateral acceleration of the guide shoes, a non-contacting displacement sensor to measure air-gap displacement, and search coils in the pole face of each electromagnet to measure the flux in the air-gap. The inherent non-linear characteristic of the electromagnet is linearised by a high bandwidth inner flux loop using flux density feedback. A positive input to this inner control loop would energise one electromagnet thus producing a force in one direction, a force in the opposite direction is achieved by a negative input. The two outer control loops consist of a pair of complementary filters: a low bandwidth air-gap displacement loop and a high pass absolute displacement loop. The controller in the outer loop is used to improve system stability, and thus should have phase advance characteristics.

The air-gap displacement loop keeps the air-gap displacement at the preset value for low frequencies, which allows good tracking of the guide rail. Random disturbance inputs to the system are removed by the absolute position feedback loop. This strategy enables the electromagnetic guide shoe to track the guide rail and at the same time remove high frequency disturbances from the system.

Random disturbances due to unevenness on the guide rails (e.g. joins, bumps) are acted upon by the high bandwidth absolute displacement loop. The disturbance force on the guide shoe due to a small bump on the guide rail would be detected by the accelerometer and appears as a change in absolute position. The outer control loop would demand a force to be generated in the electromagnet to oppose this disturbance force.

External forces acting on the lift car (e.g. sudden load change, air flow around the car) and therefore onto the guide shoes are acted upon by both the outer control loops. An impulsive force input will be detected by the accelerometer and the outer controller would demand a change in air-gap flux density to counteract this force whereas a steady-state force input (e.g. unbalance loading) is acted upon by the air-gap displacement loop. Thus the guide shoe would appear stiff to an external force acting on it.

7. CONCLUSION

This paper examined ways of improving ride quality in high speed lifts mainly by controlling its mechanical dynamics and reducing its susceptibility to random disturbances on the lift car. Although some of this work has been researched and published, implementation of these ideas onto new lift systems is not imminent. Initial studies using computer simulation have shown that significant improvement in ride quality can be achieved. The next stage would be to implement such ideas onto a prototype lift system to assess its performance and potential.

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