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FOREWORD

It is with great pleasure that we present the proceedings of the 8th Symposium on Lift and Escalator Technologies, 15-16 May 2018, organised by The Lift and Escalator Symposium Educational Trust.

The objective of The Lift and Escalator Symposium Educational Trust is to advance education in lifts, escalators and related technologies. The Trust is a Registered Charity No: 1170947 and is supported by The University of Northampton, The Chartered Institution of Building Services Engineers (CIBSE) and The Lift and Escalator Industry Association (LEIA).

The Lift Engineering programme offered at The University of Northampton includes postgraduate courses at MSc/ MPhil/ PhD levels that involves study of the advanced principles and philosophy underlying lift and escalator technologies. The programme aims to provide a detailed, academic study of engineering and related management issues for persons employed in lift making and allied industries.

The CIBSE Lifts Group is a specialist forum for members who have an interest in vertical transportation. The group meets regularly to promote technical standards, training and education, publications and various aspects of the vertical transportation industry. The CIBSE Lifts Group directs the development of CIBSE Guide D: Transportation systems in buildings, the de facto reference on vertical transportation.

LEIA is the UK trade association and advisory body for the lift and escalator industry with a membership covering some 95% of the lift and escalator industry. LEIA members supply passenger and goods/service lifts, stairlifts, homelifts, lifting platforms, escalators, passenger conveyors and a range of component parts for such products. LEIA members undertake the maintenance and modernisation of more than 250,000 products falling within the scope of the Association. LEIA provides advice on health, safety and standards matters, promotes education and training especially through its distinctive distance learning programme.

The Symposium brings together experts from the field of vertical transportation, offering an opportunity for speakers to present peer reviewed papers on the subject of their research. Speakers include industry experts, academics and post graduate students.

The papers are listed alphabetically by title. The requirement was to prepare an extended abstract, but full papers were accepted from the invited speakers where they preferred to offer them. The submissions are reproduced as they were submitted, with minor changes in formatting, and correction of obvious language errors where there was no risk of changing meaning.

Professor Stefan Kaczmarczyk, and Dr Richard Peters
Co-Chairs and Proceeding Editors

CONTENTS

Coupled Vibration of Rope-guided Hoisting System Under Multiple Constraint Conditions

Lu Yan, Guohua Cao, Ke Wang

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Keywords: rope-guided, hoisting system, coupled vibration, multiple constraint conditions

Abstract. In addition to being used as hoisting ropes, wire ropes are also used as guiding rails to provide guidance for hoisting conveyances in many Chinese hoisting systems of hundred meters or even a kilometer mine shaft and outdoor long-travel lifting systems. The vibratory response in this rope-guided hoisting system differs from flexible rails and gradually attracts researchers' attention. Therefore, modeling of coupled vibration of time-varying hoisting systems with four rope guides is presented by energy method in this paper and the system vibration equation is established with Lagrange multiplier method under multiple constraint conditions. Lateral and torsional coupling vibrations are analyzed under hoisting rope eccentricity, tension and density differences between rope guides. These characteristics provide guidelines for the design of double conveyances hoisting systems to prevent conveyances colliding and lay a foundation for future research.

1 INTRODUCTION

Dynamic Characteristics Investigation of hoisting systems is an important task for system design in mine or elevator hoisting systems. Wire ropes used as guiding rails are common substructures in mine hoisting shafts. Different from high-frequency vibration of the system with rigid rails [1], rope-guided system vibrations are characterized by lower-frequency vibrations and greater lateral amplitudes. Wang [2] simplified the guiding rope as a spring-mass module and established a lateral vibration model with two guiding rope constraints in a vertical mine construction system. Based on Wang's model [2], Cao [3, 4] analyzed the system energetic and later regarded guiding rope as continuum and numerically solved the model with Lagrange multiplier method. Wang [5] built a double drum winding hoisting model with two guiding ropes, but the guiding ropes are regarded as a force acted on conveyance. This shows the dynamics of rope-guided systems attract growing attention. Present dynamic models of rope-guided systems are limited to two guiding ropes, while there are four guiding ropes in many Chinese mine shafts. Therefore, dynamic modeling of hoisting systems with four guiding ropes is built based on Lagrange multiplier method in this paper and lateral and torsional coupling responses are revealed. This work is important for safety and stability of working conveyances.

2 MODELING FOR HOISTING SYSTEM

Geometrical model in a rope-guided hoisting system is shown as Fig. 1a). A conveyance, moving at speed v with mass of m, is suspended by a hoisting rope with length of l(t) and constrained by four guiding ropes each with length of L_i and bottom tension of T_{bi} (i=1, 2, 3, 4), as shown in Fig. 1b). Constraints between conveyance and guiding ropes are demonstrated in Fig. 1c), where w_c and u_c and θ are lateral and torsional responses of conveyance in w-u plane with coordinate origin "O", the hoisting point. (A_i, B_i) is coordinate position of the ith guiding rope at constrained point between the guiding rope and conveyance. w_i and u_i are responses of a guiding rope in w and u directions, respectively. Hoisting rope is made by the twisting of several steel wire ropes, thus the torsion from twisting is assumed to be zero and hoisting ropes are regarded as one rope in this model. Torsion of conveyance is only caused by the movement in the directions of w and u. The model is based on the following assumptions: (1) longitudinal vibrations and their coupled vibration with lateral motions are neglected. (2) Conveyance height and revolute motion around lateral axes are neglected.

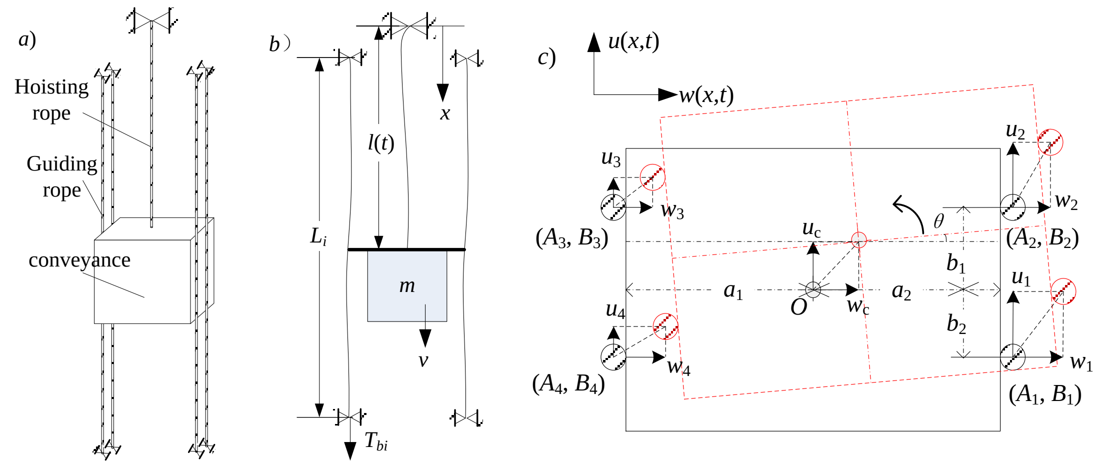

Figure 1 Modeling for hoisting system: *a*) System model, *b*) Mathematical model in vertical direction, *c*) Mathematical model in lateral direction

Kinetic energy T_k and potential energy V_e of the system are:

$$\begin{aligned} T_k &= \frac{1}{2}\int_0^{l(t)} \rho_0 \left[\left(u_{0,t} + v u_{0,x}\right)^2 + \left(w_{0,t} + v w_{0,x}\right)^2 \right] \mathrm{d}x \\ &+ \frac{1}{2}\sum_{i=1}^{4}\int_0^{L_i} \rho_i \left[\left(w_{i,t}\right)^2 + \left(u_{i,t}\right)^2 \right] \mathrm{d}x + \frac{1}{2} m_c \left(\dot{u}_c{}^2 + \dot{w}_c{}^2 \right) + \frac{1}{2} J_c \dot{\theta}^2 \end{aligned} \tag{1}$$

$$V_e = \int_0^{l(t)} \left[\rho_0 \left(l - x\right) + m \right] \left(g - a\right) \varepsilon_0 \mathrm{d}x + \sum_{i=1}^{4} \int_0^{L_i} \left[\rho_i g \left(L_i - x\right) + T_{bi} \right] \varepsilon_i \mathrm{d}x \tag{2}$$

where ρ_0 and ρ_i are length density of hoisting rope and the ith guiding rope, respectively. Subscript x and t respectively denote partial derivatives of variables with respect to x and t. Overdot " • " is derivative with respect to t. Lateral strain measure of hoisting rope and the ith guiding rope are given as $\varepsilon_0 = 0.5(w_{0,x}^2 + u_{0,x}^2)$ and $\varepsilon_i = 0.5(w_{i,x}^2 + u_{i,x}^2)$. J_c is rotational inertia for transverse section of conveyance. Since the hoisting rope is subject to an external force at $x(t) = 0$ due to head sheave vibration, lateral excitations in w and u directions are set as $w_0(0, t) = e_1(t)$, $u_0(0, t) = e_2(t)$. Setting vibration of the hoisting rope as two sections of homogeneous and non-homogeneous solutions yields Eq. (3).

$$\begin{aligned} w_0\left(x,t\right) &= \bar{w}_0\left(x,t\right) + \bar{h}_1\left(x,t\right) \\ u_0\left(x,t\right) &= \bar{u}_0\left(x,t\right) + \bar{h}_2\left(x,t\right) \end{aligned} \tag{3}$$

where $\bar{h}_i\left(x,t\right) = e_i\left(t\right)\left(1 - x/l\right)$, ($i$=1,2), The boundary conditions are:

$$\begin{aligned} & w_i\left(0,t\right) = w_i\left(L_i,t\right) = u_i\left(0,t\right) = u_i\left(L_i,t\right) = 0,\ \ i = 1,2,3,4 \\ & \bar{w}_0\left(0,t\right) = \bar{u}_0\left(0,t\right) = 0 \end{aligned} \tag{4}$$

According to Galerkin's method, time-variant domain $[0, l(t)]$ of the hoisting rope and $[0, L_i]$ of the guiding rope for x can be converted to fixed domain [0, 1] for ζ and η_i, respectively, where $\zeta = x/l(t)$ and $\eta_i = x/L_i$ (i=1, 2, 3, 4). Any variable () in time-variant domain is expressed with $\tilde{()}$ in fixed domain. The solutions of the system are assumed as following forms:

$$
\begin{aligned}
\tilde{w}_0(\zeta,t) &= \sum_{j=1}^{N_1} \varphi_j(\zeta)\, p_{0,j}(t) \\
\tilde{u}_0(\zeta,t) &= \sum_{j=1}^{N_1} \varphi_j(\zeta)\, q_{0,j}(t) \\
\tilde{w}_i(\eta,t) &= \sum_{j=1}^{N_2} U_{i,j}(\eta)\, p_{i,j}(t), i=1,2,3,4 \\
\tilde{u}_i(\eta,t) &= \sum_{j=1}^{N_2} U_{i,j}(\eta)\, q_{i,j}(t), i=1,2,3,4
\end{aligned}
\tag{5}
$$

where $p_{0,j}$ and $q_{0,j}$ as well as $p_{i,,j}$ and $q_{i,,j}$ are generalized coordinates for the hoisting rope and the *i*th guiding rope respectively in *w* and *u* directions, φ_j and $U_{i,\,j}$ are corresponding vibration mode functions:

$$
\begin{aligned}
\varphi_j(\zeta) &= \sin\left(\frac{2j-1}{2}\pi\zeta\right) \\
U_{i,j}(\eta_i) &= \sin(j\pi\eta_i), i=1,2,3,4
\end{aligned}
\tag{6}
$$

According to Fig. 1c), constraint conditions between conveyance hoisting rope and four guiding ropes are:

$$
\begin{aligned}
& g_1 = \tilde{w}_c - \tilde{w}_0(l,t) = 0 \\
& g_2 = \tilde{u}_c - \tilde{u}_0(l,t) = 0 \\
& \begin{cases} g_{2i+1} = w_c - (1-\cos\theta)A_i - B_i\sin\theta - w_i(l,t) \\ g_{2i+2} = u_c + A_i\sin\theta - (1-\cos\theta)B_i - u_i(l,t) \end{cases}, i=1,2,3,4
\end{aligned}
\tag{7}
$$

where (A_1, B_1)=(a_2, -b_2), (A_2, B_2)=(a_2, b_1), (A_3, B_3)=(-a_1, b_1), (A_4, B_4)=(-a_1, -b_2). Since the conveyance rotation angle is small enough, assumptions sin θ=θ and cos θ=1 are valid to obtain the solution. System vibration are given by Equation (8)-(9).

$$
\frac{d}{dt}\frac{\partial T_k}{\partial \dot{Q}_i} - \frac{\partial T_k}{\partial Q_i} + \frac{\partial V_e}{\partial Q_i} = \sum_{j=1}^{10} \lambda_j \frac{\partial g_j}{\partial Q_i}
\tag{8}
$$

$$
\mathbf{M}\ddot{\mathbf{Q}} + \mathbf{C}\dot{\mathbf{Q}} + \mathbf{K}\mathbf{Q} = \mathbf{F} + \mathbf{G}^{\mathrm{T}}\boldsymbol{\lambda}
\tag{9}
$$

where $\mathbf{Q}$=[$p_{0,1},\ldots, p_{0,N1}, q_{0,1},\ldots, q_{0,N1}, p_{1,1},\ldots, p_{1,N2}, q_{1,1},\ldots, q_{1,N2}, p_{2,1},\ldots, p_{2,N2}, q_{2,1},\ldots, q_{2,N2}, p_{3,1},\ldots, p_{3,N2}, q_{3,1},\ldots, q_{3,N2}, p_{4,1},\ldots, p_{4,N2}, q_{4,1},\ldots, q_{4,N2}, w_c, u_c, \theta]^{\mathrm{T}}$ is generalized coordinate vector, $\boldsymbol{\lambda}=[\lambda_1, \lambda_2,\ldots, \lambda_{10}]^{\mathrm{T}}$ is Lagrange multiplier vector, $G_{j,i} = \partial g_j / \partial Q_i$, **M**, **C**, **K** and **F** are matrices of mass, damping, stiffness and force, respectively. In order to simplify solution, transformation square matrix $\mathbf{T_R}$ is introduced. Thus, $\mathbf{Q}_{\mathrm{T}} = \mathbf{T_R}\cdot\mathbf{Q}$, $\mathbf{M}_{\mathrm{T}} = \mathbf{T_R}\cdot\mathbf{M}\cdot\mathbf{T_R}^{\mathrm{T}}$, $\mathbf{K}_{\mathrm{T}} = \mathbf{T_R}\cdot\mathbf{K}\cdot\mathbf{T_R}^{\mathrm{T}}$, $\mathbf{F}_{\mathrm{T}} = \mathbf{T_R}\cdot\mathbf{F}$, $\mathbf{G}_{\mathrm{T}} = \mathbf{G}\cdot\mathbf{T_R}^{\mathrm{T}}$. Setting $\mathbf{Q}_{\mathrm{T}} = [\mathbf{Q}_0^{\mathrm{T}} \ \mathbf{Q}_1^{\mathrm{T}}]^{\mathrm{T}}$, $\mathbf{G}_{\mathrm{T}} = [\mathbf{G_0} \ \mathbf{G_1}]$ yields $\mathbf{Q}_{\mathrm{T}} = \left[-\mathbf{G}_0^{-1}\mathbf{G}_1 \ \ \mathbf{I}_{2N_1+8N_2-7}\right]^{\mathrm{T}}\cdot \mathrm{Q}_1$, where $\mathbf{G_0}$ is an invertible matrix with dimension of 10×10. Letting $\boldsymbol{\Phi} = \left[-\mathbf{G}_0^{-1}\mathbf{G}_1 \ \ \mathbf{I}_{2\mathrm{N}_1+8\mathrm{N}_2-7}\right]^{\mathrm{T}}$, the vibration equation (9) can be written as:

$$\left(\mathbf{\Phi}^{\mathrm{T}}\mathbf{M}\mathbf{\Phi}\right)\ddot{\mathbf{Q}}_1+\left(2\mathbf{\Phi}^{\mathrm{T}}\mathbf{M}\dot{\mathbf{\Phi}}+\mathbf{\Phi}^{\mathrm{T}}\mathbf{C}\mathbf{\Phi}\right)\dot{\mathbf{Q}}_1+\left(\mathbf{\Phi}^{\mathrm{T}}\mathbf{M}\ddot{\mathbf{\Phi}}+\mathbf{\Phi}^{\mathrm{T}}\mathbf{M}\dot{\mathbf{\Phi}}+\mathbf{\Phi}^{\mathrm{T}}\mathbf{K}\mathbf{\Phi}\right)\mathbf{Q}_1=\mathbf{\Phi}^{\mathrm{T}}\mathbf{F}_{\mathrm{T}} \tag{10}$$

3 SOLUTIONS AND RESULTS

Parameters of a rope-guided system are presented in Table 1.

Table 1 Parameters of rope-guided hoisting System

Parameters	Descriptions	Values
m	Conveyance mass and loads	24000 [kg]
T_b	Average bottom tension of guiding rope	56.5 [kN]
ρ_0	Length density of hoisting ropes	6×2.62 [kg/m]
ρ_i	Length density of guiding rope	8.94 [kg/m]
$a_1(a_2)$, $b_1(b_2)$	Cross-sectional dimension of conveyance	0.58 [m], 1.095 [m]
L_i	Length of guiding rope	1012 [m]
H	Hoisting height	1000 [m]
v	Hoisting velocity	9.7 [m/s]

According to Article 388 of Coal Mine Safety Regulations [6], tension differences between guiding ropes are no less than 5% of their average tensions and tensions of two ropes near the mine shaft center are larger than that of the other two. Therefore, setting T_{b1} = 1.025T_b, T_{b2} = 1.075T_b, T_{b3} = 0.975T_b, T_{b4} = 0.925T_b as 5% tension differences, the system excitations from head sheave in w and u directions are set as $e_1(t) = A_1\sin\pi t$, $e_2(t) = A_2\sin\pi t$, respectively, where A_1 = 0.03 m and A_2 = 0.04 m are assumed. Lateral and torsional coupling vibrations are show as Fig. 2 under tension differences of guiding ropes. It can be seen from curves of $w_c(t)$ and $u_c(t)$ that there are only small differences in amplitude of lateral responses under equal tensions of guiding ropes in w or u directions, because there are only excitation amplitude difference of $w_c(t)$ and $u_c(t)$, and the torsional responses are too small to affect the lateral displacement.

Lateral and torsional coupling vibrations under different linear density of guiding ropes and different eccentricity of hoisting rope are shown in Fig. 3 and Fig. 4. The torsional responses are also small within 0.1m eccentricity installation error, while linear density differences of guiding ropes have an obvious effect on torsional vibration of conveyance. Since obvious torsional vibration can affect lateral vibration, linear density of guiding ropes should be well designed to satisfy sufficient space between conveyances and shaft wall.

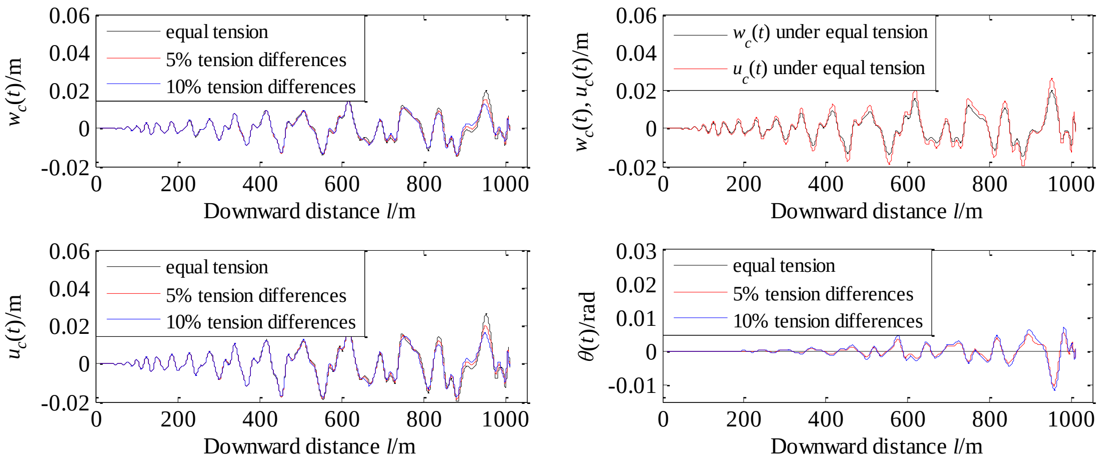

Figure 2 Vibration under tension differences of guiding ropes

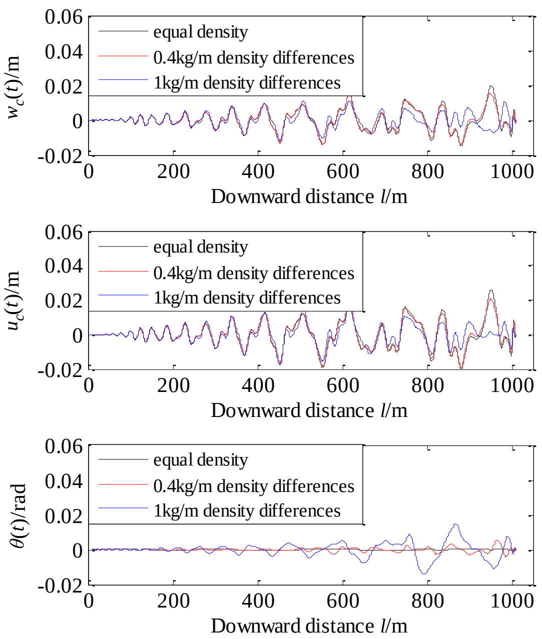

Figure 3 Vibration under liner density differences of guiding ropes

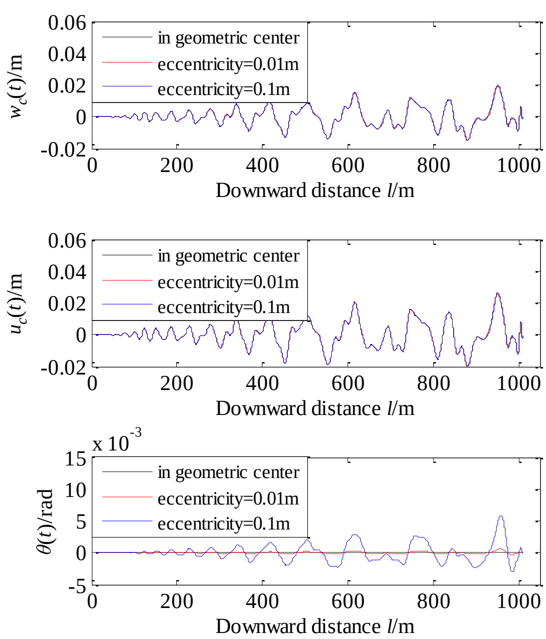

Figure 4 Vibration under eccentricity differences of hoisting ropes

4 CONCLUSIONS

Coupled vibration model of time-varying hoisting systems with four rope guides is built by energy method and the system vibration equation is established with Lagrange multiplier method under multiple constraint conditions. Lateral and torsional coupling responses are obtained under tension and density differences of guiding ropes as well as eccentricity between hoisting ropes and conveyance. These characteristics in some degree may provide guidelines for the design of hoisting system to prevent collision between conveyances and the shaft wall..

ACKNOWLEDGEMENTS

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BIOGRAPHICAL DETAILS

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Dual Rated Speeds Escalator in Rapid Transit System with Extended Ramping Up and Down

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Keywords: Dual rated speeds, automatic switching, rapid transit system, acceleration, ramping up, ramping down, vibration, jerkiness.

Abstract. To cater for different needs of escalator operating speeds in rapid transit systems (i.e. higher rated speed of 0.75m/s during peak hours is for effective discharging of passengers while slower rated speed of 0.50m/s during off-peak hours is for elderly passengers), we have introduced the dual rated speed escalator. Conventionally, the switching between 2 rated speeds can be done either manually through a key switch or automatically when no passengers are detected on the escalators at pre-set timing. However, there is a possibility of not being able to change speed if there are constant passengers coming into the rapid transit station, taking the escalators. Therefore, this shortcoming will be overcome by setting up a schedule timetable to do the safe switching of escalator rated speeds with passengers riding on the escalators with extended ramping up and down without comprising any safety requirements.

This paper presents the case studies conducted on an existing station where a performance–based approach was adopted. The timing for the speed ramping up/down between the 2 rated speeds has been increased to 30 seconds in order to reduce the acceleration which results in minimizing the acceleration (vibration) and the rate of change of this acceleration (jerk). The objective is to ensure that the passenger’s perception are imperceptible and do not experience any abnormal and sudden change of vibration and jerk during the switching of dual rated speeds with extended ramping up and down.

1 INTRODUCTION

Rapid transit system (RTS) is the main mode of public transport in Singapore. The escalators at RTS stations are usually heavily utilized to bring in and discharge large amounts of passengers efficiently and safely. Currently, all escalators are designed and operated at the rated speed of 0.75m/s to meet the high travel demands during the morning and evening peak hours.

In recent years, due to the ageing population in Singapore, statistics have revealed that the increase in escalator incidents in RTS stations are associated with elderly passengers due to their increased reaction time and poor understanding of the proper usage of escalators. Surveys conducted have also shown that elderly passengers are more comfortable with slower speed escalators. In order to accommodate the elderly passengers during off-peak hours while the high throughput (primary rated speed of 0.75m/s) is not required, a secondary rated speed of 0.5m/s is incorporated to operate the escalators during off-peak hours.

Conventionally, switching between two rated speeds can be done either manually through a key switch or automatically when no passengers are detected on the escalator at pre-set timings. By leveraging on new technology, smooth and gradual acceleration and deceleration of escalators are achievable with more precise digital control for the safe automatic switching of speeds while passengers are still riding on the escalators.

The purpose of this paper is to present the concept and philosophy behind the approach and share the implementation, technical challenges and results for the development of the first in the world automatic seamless switching between primary and secondary rated speeds with passenger riding on the escalator based on a scheduled timetable.

2 CONCEPT AND PHILOSOPHY

The objective of the case studies is to quantify the measure of passenger's perception on the effect of the escalator step acceleration (measured as vibration) in the forward horizontal direction (x-axis) and the rate of change of this acceleration (measured as jerk) [1] for the stability of the passengers on the escalator. An accelerometer is utilized to measure the passenger comfort during the escalator ride in accordance to ISO 18738-2 [2]. Following studies in this paper focus mainly on the x-axis as this is the domineering axis for vibration and jerk whereas the escalator step vibration and jerk along the y-axis and z-axis are well restricted by the escalator step chain, step track and up-lifting tracks.

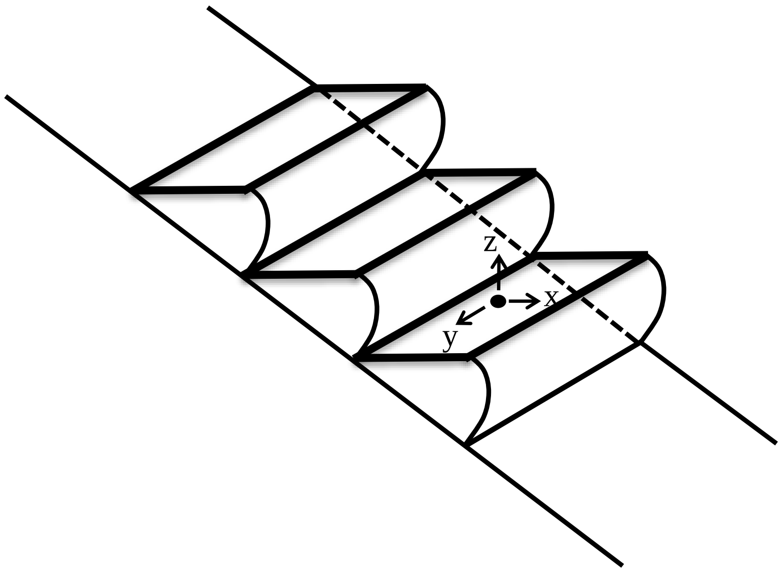

Figure 1 The 3-axis of acceleration for the escalator step

In accordance to SS626 [3] and EN 115-1 [4], escalators or moving walks which start or accelerate automatically by the entering of a passenger shall move with at least 0.2 times the nominal speed and then accelerate less than 0.5 m/s^2.

2.1 Case study on speed ramping up by 3 seconds – Typical case from the standby speed of 0.2m/s (when no one is using escalator) to the rated speed of 0.75m/s

An experiment was conducted on an escalator compliant to SS626 and EN 115-1, where measurements were taken at a rated speed of 0.75m/s as well as the measurements for the escalator speed ramping up from 0.2 m/s to 0.75m/s (within 3 seconds) with an acceleration of 0.18 m/s^2.

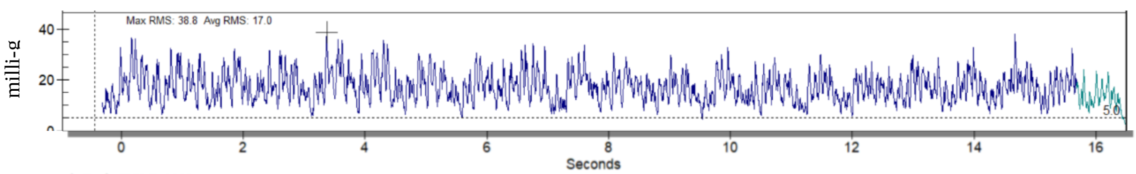

Figure 2 Vibration of escalator step in the x-axis – speed of 0.75m/s

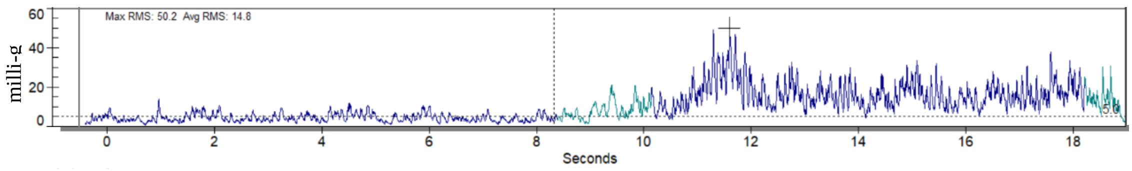

Figure 3 Vibration of escalator step in the x-axis – ramping up from 0.2 m/s to 0.75m/s in 3 seconds (from 9 to 12s)

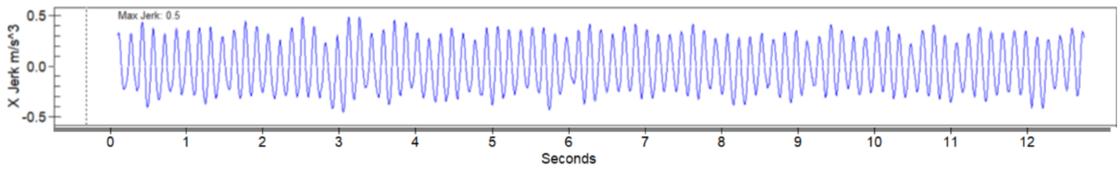

Figure 4: Jerk of escalator step in the x-axis – speed of 0.75 m/s

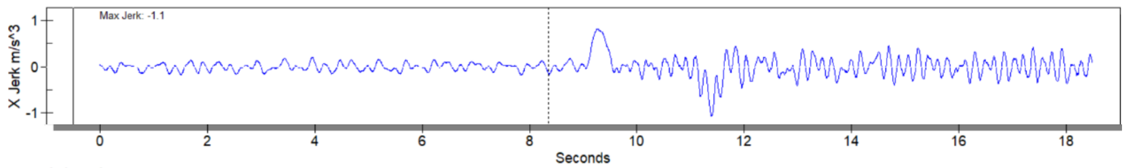

Figure 5 Jerk of escalator step in the x-axis – ramping up from 0.2m/s to 0.75 m/s in 3 seconds (from 9 to 12s)

Table 1 Comparison of Average RMS (Vibration) and Jerk for the speed of 0.75m/s and the ramping up from 0.2 to 0.75m/s in 3s

	Average RMS (milli-g)	Maximum Jerk (m/s^3)
Speed of 0.75 m/s	17.0	0.5
Ramping up from 0.2 m/s to 0.75 m/s	14.8	1.1

From the above readings, the average RMS value, measuring vibration of the escalator step in the x-axis of the ramping up (3s) is slightly lower than the constant speed of 0.75m/s. This is expected as the slower speed at initial stage of ramping tends to have lesser vibration. However, from the jerk graph, it has showed that there is a maximum jerk of 1.1 m/s^3 as compared to 0.5 m/s^3. The maximum jerk (rate of change of vibration) of 1.1 m/s^3 occurs when the speed ramping up starts. This implies that although the acceleration value of 0.18 m/s^2 are within the limits stated in SS626 and EN115-1, there is a higher jerk value due to this acceleration within a 3 second timeframe.

As every individual ability to balance would be different due to their physiology and psychology, therefore the increase in the jerk value may cause the passenger to feel uncomfortable or unbalanced [5].

2.2 Case study on extended ramping up and down in 30 seconds for dual rated speeds of 0.75m/s and 0.50m/s

The escalator is programmed with a scheduled time table to auto-switching over while passengers are riding on the escalators from a primary speed of 0.75m/s during peak hours to a secondary speed of 0.5m/s during the off peak hours and vice versa.

The duration of ramping up and down is set to gradual change between the two rated speeds in 30 seconds. The capability of the variable speed drive is maximized to achieve this function. Theoretically, the acceleration/deceleration of the changeover from 0.5 m/s to 0.75 m/s and vice versa is worked out to be 0.0083m/s^2. The aim is to ensure the effect (vibration and jerk) during auto-switching over of the escalator rated speeds is imperceptible to passengers while they are riding on the escalator.

The measurements for the same escalator operating at different conditions are carried out:

1) Escalator rated speed of 0.5 m/s
2) Escalator rated speed of 0.75 m/s
3) Escalator ramping up speed from 0.5 m/s to 0.75m/s
4) Escalator ramping down speed from 0.75 m/s to 0.5 m/s

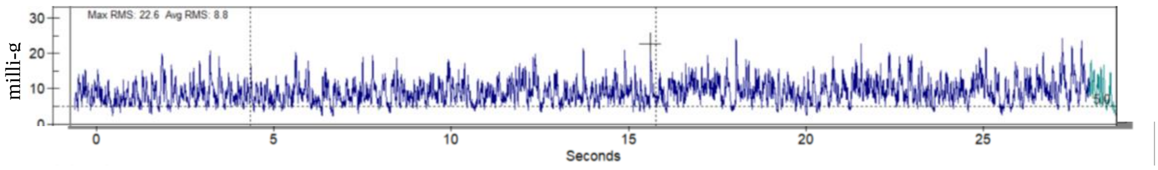

Figure 6 Vibration of escalator step in the x-axis – speed of 0.50m/s

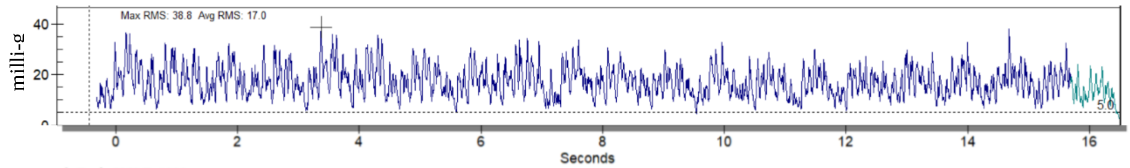

Figure 7 Vibration of escalator step in the x-axis – speed of 0.75m/s

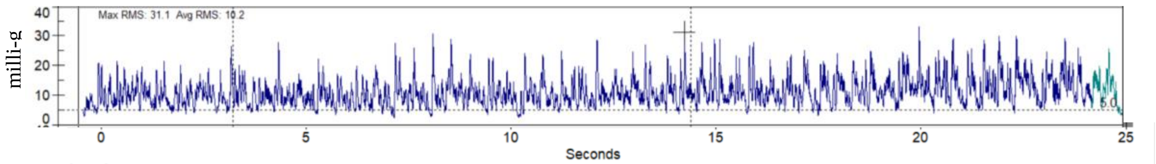

Figure 8 Vibration of escalator step in the x-axis – ramping up from 0.5 m/s to 0.75m/s in 30 seconds

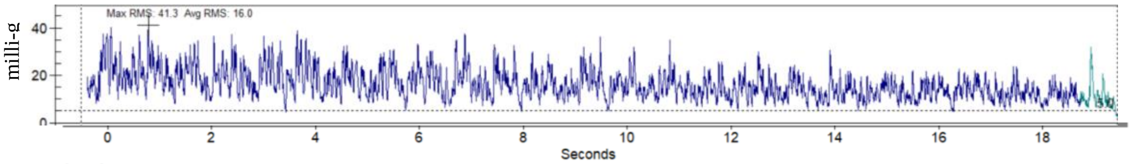

Figure 9 Vibration of escalator step in the x-axis – ramping down from 0.75 m/s to 0.5m/s in 30 seconds

Table 2 Comparison of Average RMS (Vibration) under various operating conditions

	Average RMS (milli-g)
Speed of 0.5 m/s	8.8
Speed of 0.75 m/s	17.0
Ramping up from 0.5 m/s to 0.75 m/s	10.2
Ramping down from 0.75 m/s to 0.5 m/s	16.0

From the vibration of escalator steps in the x-axis under various operating conditions, it was noted that the average RMS values of the step vibration during the escalator ramping up/down are within the limits of the escalator travelling at 0.75 m/s rated speed. There are also no abnormal and large sudden changes of vibration values which could be perceived by the passengers. From the readings, the passenger's perception during the speed ramping up or down could be similar or less perceptible compared with riding an escalator at a rated speed of 0.75 m/s.

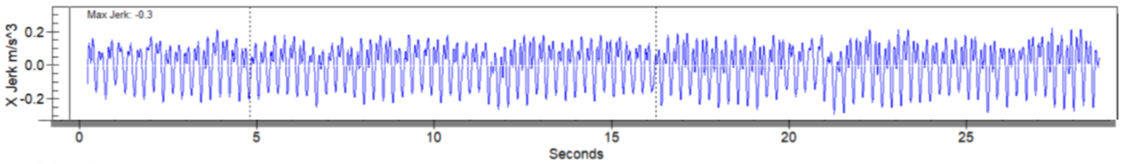

Figure 10 Jerk of escalator step in the x-axis – speed of 0.5 m/s

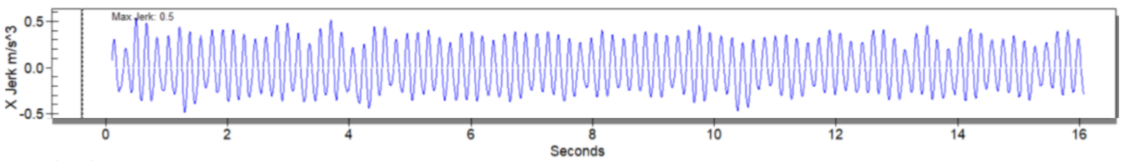

Figure 11 Jerk of escalator step in the x-axis – speed of 0.75 m/s

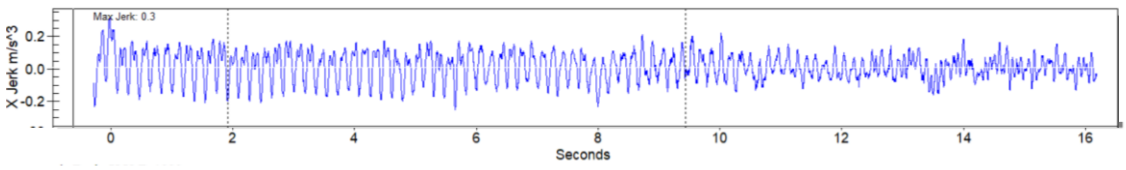

Figure 12 Jerk of escalator step in the x-axis - ramping up from 0.5 m/s to 0.75 m/s in 30 seconds

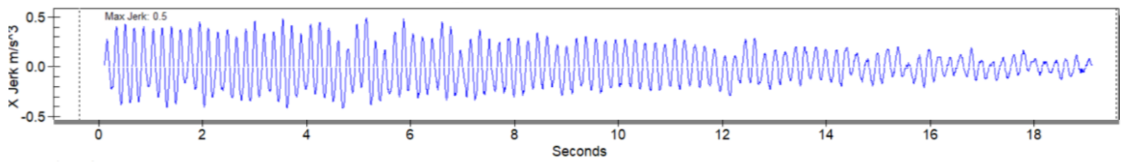

Figure 13 Jerk of escalator step in the x-axis - ramping down from 0.75 m/s to 0.5 m/s in 30 seconds

Table 3 Comparison of Maximum Jerk under various operating conditions

	Maximum Jerk (m/s^3)
Speed of 0.5 m/s	0.3
Speed of 0.75 m/s	0.5
Ramping up from 0.5 m/s to 0.75 m/s	0.3
Ramping down from 0.75 m/s to 0.5 m/s	0.5

The maximum jerk values are compared among various operating conditions. The jerk will have a direct impact on the passenger's ride comfort [6]. For the various escalator operating conditions as indicated in the table, the maximum jerks measured during the extended ramping up and down of the escalator rated speed are also within the limit of the escalator travelling at a rated speed of 0.75 m/s.

From both the graphs of the escalator step vibration and the jerk values, there are also no distinct and large sudden changes in the measured values. Therefore, this shows that the passengers will experience similar or less perceptible vibration and jerk during the ramping up/down in 30s as compared with the ride comfort at the rated speed of 0.75 m/s.

3 SAFETY MEASURES

Necessary safety measures are incorporated accordingly to prevent the escalator from overspeed and underspeed and to prevent the sudden change in the acceleration/deceleration which could result in jerkiness of the escalator.

The safety measures are incorporated as follows:

a) To set the overspeed limit at 0.85m/s, both machine brake and emergency brake will be activated
b) To set the underspeed limit at 0.11 m/s, both machine brake and emergency brake will be activated
c) To limit the acceleration/deceleration to $0.01m/s^2$, machine brake will be activated.
d) Handrail underspeed of -15% will always apply correspondingly to all operating conditions using speed differential between handrail speed and step speed.

4 CONCLUSION

Passenger's ride comfort is subjective. Every passenger's ability to balance under the influence of a forward horizontal acceleration (vibration) and rate of change of the acceleration (jerk) of escalator step differs from one another due to the different physical conditions of each individual.

The purpose of extending the duration of the ramping up/down of escalator speed is to ensure the ramping conditions are gradual & safe while passengers are still riding on an escalator. The case studies of ramping up and down benchmark with the vibration and jerk under the rate speed of 0.75m/s which the current escalator users experienced in RTS of Singapore and was deemed safe. The studies prove that with the extended ramping up and down, the changes are imperceptible to the passengers who do not experience any abnormal and sudden change of vibration and jerk during the switching of dual rated speeds. This proposal of dual rated speed escalator with extended ramping up and down is suitable for a busy RTS environment to accommodate the different needs of escalator operating speeds (i.e. higher rated speed of 0.75m/s during peak hours for effective discharging of passengers and slower rated speed of 0.50m/s during off-peak hours for elderly passengers).

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BIOGRAPHICAL DETAILS

Gan Kong Cheong has been in the Lift & Escalator industry for 20 years. He is presently the Principal Project Manager working for Land Transport Authority of Singapore. He has taken part in the design, installation, testing & commissioning of the a few rapid transit line namely, North East Line, Circle Line, Downtown line and also the present Thomson-East Coast Line which has more than 1200 escalators and 400 lifts. He is a Professional Engineer (Mechanical), Specialist Professional Engineer (Lift & Escalator), Specialist Professional Engineer (Access Platform), Authorised Examiner (Lifting Equipment) and Chartered Railway Engineer (S'pore).

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Melvyn Thong has over 27 years of experience in design, supervision and project management of electrical and mechanical installation for power plants, industrial, commercial and institutional, road and railway transit projects. Over 18 years of experience in the design, supervision and project management of 22kV AC, 400V AC and 750V DC electrical system for Singapore Road and Rapid Transit Systems. Experienced in the design development, implementation and asset renewal of power supply systems for Thomson East Coast Line (TEL), Downtown Line (DTL), Compass Line (NSEW Line), North East Line (NEL), Circle Line (CCL), Sengkang Punggol LRT (SPLRT), Bukit Panjang LRT (BPLRT).

Experienced in high voltage and low voltage reticulation design and is licensed to perform switching in Singapore and has extensive experience in the design and implementation of Mechanical and Electrical Services requirements for Roads and Rapid Transit Systems for RTS and LRT networks. A certified Chartered Electrical Engineer (UK), Chartered Railway Engineer (S'pore) and Professional Engineer in Singapore with a Licensed Electrical Engineering for up to 22kV.

Dynamic Behavior of Traction System with Tension at the Pulley of Compensating Rope

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Keywords: Traction system, dynamics, tension pulley

Abstract: With high capacity and low cost, the traction system is used in various lifting applications. Caused by the effect of hoisting and compensation rope with time-varying length, the longitudinal vibration of the system is varying moderately during the period of operation. In this paper, the dynamic model of the traction system with compensating rope is established based on Lagrange equations of the first kind. The compensating rope is tensioned by a pulley with external force working on it. The dynamics resonance characteristics under different working conditions of bidirectional conveyances and tensioning pulley can be obtained, which can provide reference for the design of long- travel and high-speed hoisting systems.

1 INTRODUCTION

Due to their ability to resist relatively large axial loads, ropes have been widely used in many different applications to support structures, conduct signals, and carry payloads. Many researchers have concentrated on the longitudinal vibrations of the hoisting cable or container for decades. Kaczmarczyk and Ostachowicz [1,2] investigated the longitudinal responses of a hoisting cable in the mine hoisting system and a compensation cable in a high-speed elevator. Ren and Zhu [3] presented the longitudinal and lateral vibrations of a moving two-cable one-rigid-body-car system, in which the rotation of the car is considered. For the spatial discretization of a cable, assumed modes method (AMM) [3] and finite element method [4, 5] are commonly adopted. For the equation of motion, Lagrange's equations [6, 7] or Hamilton's principle can be selected. Terumichi Y. [8] researched the nonstationary vibration of a string with a constant hoisting speed, and the analytical results showed that the axial velocity of the string influenced the peak amplitude of the string vibration at the passage through resonances. But in above references, only single side dynamic model is established and the compensating rope is ignored, so the effect of tensioning pulley and tensioning force on the traction system can't be investigated.

2 MODEL DESCRIPTION

As shown in Fig. 1, the traction system with tension at the pulley is investigated, the length of hoisting cable is donated as $l_i(t)$, and their corresponding velocity and acceleration can be expressed as $v_i(t)$ and $a_i(t)$, respectively.

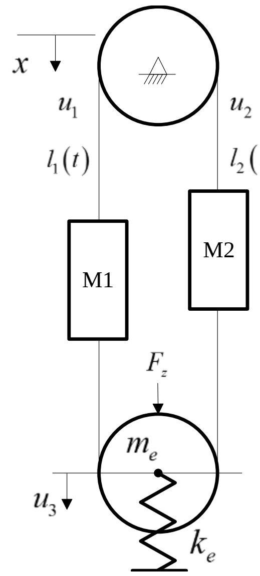


Figure 1 Traction system with tension at the pulley

3 MATHEMATICAL MODEL

The model was established based on the energy method, and the kinetic energy is given as followed:

$$K_e = \frac{1}{2}\int_0^{LL} \rho_1 \left(\frac{Du_1}{Dt} + v_1\right)^2 dx + \frac{1}{2}\int_0^{LL} \rho_2 \left(\frac{Du_2}{Dt} + v_2\right)^2 dx + \frac{1}{2} m_e \dot{u}_3^2 \tag{1}$$

The potential energy of the traction system is a function of the vibration displacement, which is given as followed:

$$E_e = \int_0^{LL} \left[T_1(x,t)\varepsilon_1 + \frac{1}{2} EA\varepsilon_1^2 \right] dx + \int_0^{LL} \left[T_2(x,t)\varepsilon_2 + \frac{1}{2} EA\varepsilon_2^2 \right] dx + \frac{1}{2} k_e u_3^2 \tag{2}$$

$$E_g = -\int_0^{LL} \rho_1 g \cdot u_1(x,t) dx - \int_0^{LL} \rho_2 g \cdot u_2(x,t) dx - m_e g u_3 \tag{3}$$

Where $\rho_i\left(i=1,2\right)$ is the linear density of the lifting cable with concentrated inertia elements m_i attached to it. m_i is the hoisting conveyance mass. $T_i(x,t)$ is the tension at position *x* of the cable. $u_i\left(x,t\right)$ is the longitudinal vibration at position x of the cables; u_3 is the longitudinal vibration of the tensioning pulley, m_e is the quality of the tensioning pulley, k_e is the stiffness of the tensioning spring, LL is the distance between traction pulley and tensioning pulley.

ε is the elastic strain of the cable. Caused by the neglection of transverse vibration, the expression of ε can be obtained as follows:

$$\varepsilon = \frac{1}{2} u_x \tag{4}$$

After the normalization, the kinetic energy, the elastic potential energy and the gravitational potential energy of the system are brought into the Lagrange equation of the first kind.

$$\frac{d}{dt}\frac{\partial K_e}{\partial \dot{q}_i} - \frac{\partial K_e}{\partial q_i} + \frac{\partial\left(E_e + E_g\right)}{\partial q_i} = Q_i + \sum_{k=1}^{n} \lambda_k \frac{\partial g_k}{\partial q_i} \tag{5}$$

In this paper, the constraint conditions at the tensioning pulley can be defined as:

$$g_1 : u_1\left(LL,t\right)+u_2\left(LL,t\right)=2u_3 \tag{6}$$

Using AMM, the solution can be expressed as:

$$u(\xi,t)=\sum_{i=1}^{n}U_i\left(\xi\right)q_i\left(t\right) \tag{7}$$

Where U_i is the assumed mode and set as $\sin(\frac{2i-1}{2}\xi),\left(0\le\xi\le1\right)$. q_i is the generalized displacement and set as $\left[q_1,q_2,u_3\right]^{\mathrm{T}}$. As shown in figure 1, q_1 is the generalized displacement of the left rope, q_2 is the generalized displacement of the right rope, and u_3 is the generalized displacement of the tensioning pulley.

Substituting Eqs. (7) into Eq. (5), The equation of motion can be reduced as followed:

$$\begin{aligned}&M\ddot{Q}+C\dot{Q}+KQ=F+G^{\mathrm{T}}\lambda\\&g(q,t)=0\end{aligned} \tag{8}$$

Where G is the Jacobian matrix of the constraint equations, and $\lambda=\left(\lambda_1,\lambda_2,\ldots,\lambda_c\right)$ are Lagrangian multipliers, which denote the constraint forces between two hoisting cables and the tensioning pulley, $Q=\left[q_1,q_2,u_3\right]^{\mathrm{T}}$ is a vector of generalized displacement. By derivation, the forcing term F can be obtained as follows.

$$\boldsymbol{F}_{u,i}=\rho_i v_i^2\int_0^1 U_j{'}\left(\xi\right)\mathrm{d}\xi-\rho_i LLa_i\int_0^1 U_j(\xi)\mathrm{d}\xi-T_i\left(\xi,t\right)\int_0^1 U_j{'}\left(\xi\right)d\xi \quad ,i=1,2 \tag{9}$$

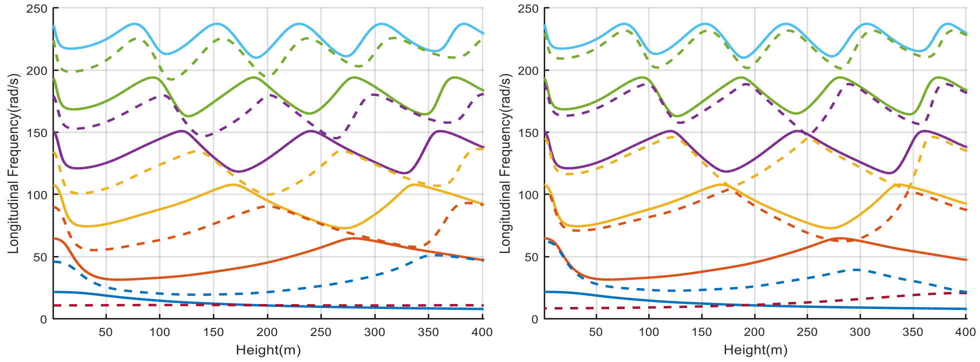
4 NUMERICAL CALCULATION

The parameters used in the numerical simulation are as listed in Table.1.

Table 1 Fundamental parameters of the system

Parameter	Definition	Value
$\boldsymbol{H}$	Travel Height	401 m
$\boldsymbol{V}$	Running Velocity	10 m/s
$\boldsymbol{A}$	Running Acceleration	$2\ \mathrm{m/s^2}$
$\boldsymbol{\rho}$	Rope linear mass density	22.6 kg/m
$\boldsymbol{E_s}$	Rope Modulus of elasticity	1.2×10^{11} Pa
$\boldsymbol{A_s}$	Rope Cross section area	$6.94\times10^{-3}\ \mathrm{m^2}$
$\boldsymbol{M_z}$	Conveyance mass	1.5×10^{4} kg

The frequency characteristics of the two-conveyance system can be obtained by numerical calculation, which is shown in Fig 2. The kinetic equation can be obtained using AMM and the mass matrix M and the stiffness matrix K can be extracted. Finding out eigenvalues of the matrix $\sqrt{K/M}$, and the natural frequencies of each order can be obtained by sorting the eigenvalues at corresponding time.



(a) (b)

Figure 2 The frequency characteristics of the bilateral hoisting system with tensioning pulley and without tensioning pulley

Figure 2-(a) stands for the descend progress of M_1, and Figure 2-(b) is in the opposite direction. In the figures above, the solid lines are the natural frequency of lifting system without tensioning pulley, and the dotted lines are the natural frequency of lifting system with tensioning pulley. From the result above, we can see the natural frequencies of each order of lifting system are decreasing correspondingly caused by the tensioning pulley.

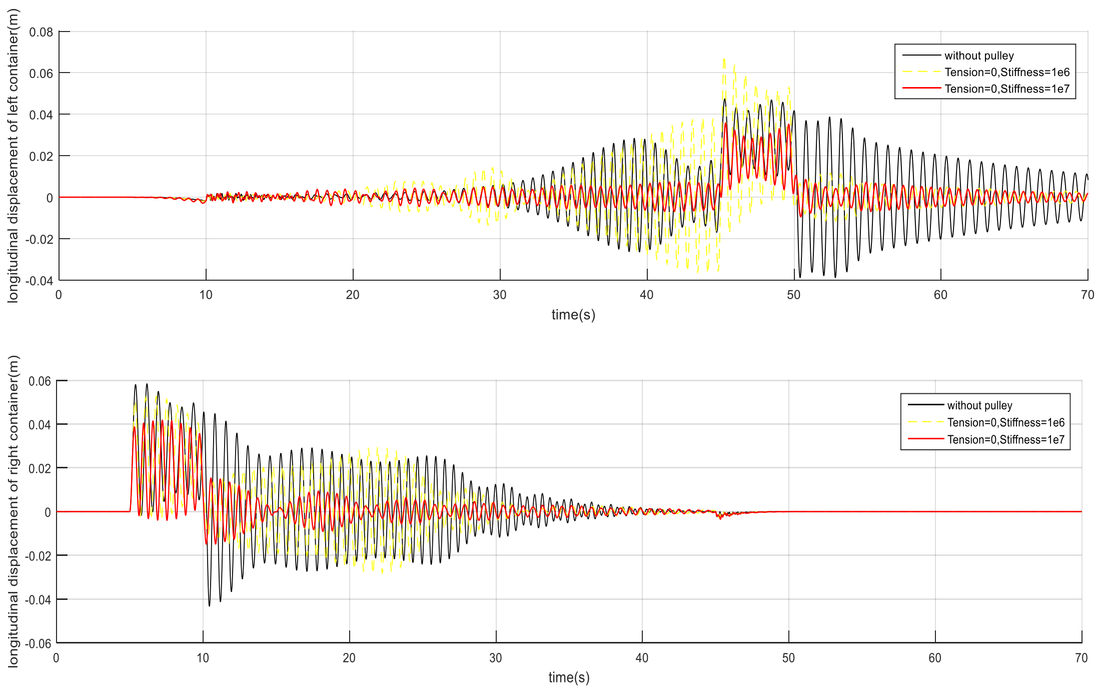


Figure 3 The longitudinal displacement of bilateral conveyance with different tensioning stiffness

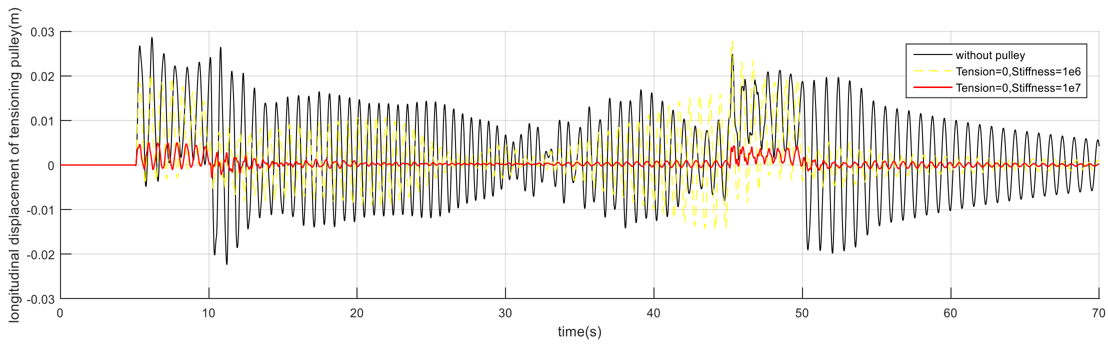


Figure 4 The longitudinal displacement of tensioning pulley with different tensioning stiffness

Firstly, the impact of the tensioning spring is discussed by comparing the numerical solution results of different stiffness working on the pulley. As shown in Fig 3 and 4, the curve trend of hoisting conveyance with and without tensioning pulley are basically similar. By comparing the red and black line in above pictures, the vibration amplitude of hoisting conveyance without tensioning stiffness is much higher than that with a tensioning spring of proper stiffness working on the pulley. However, by comparing the yellow and black line in above pictures, it can be seen that the tension spring can't get its effect if the tensioning stiffness isn’t selected in the appropriate range. The results show that the tensioning stiffness can achieve the vibration amplitude’s suppression of the hoisting conveyance. Similarly, as shown in Fig 4, the longitudinal displacement of the tensioning pulley is also decreased by proper tensioning stiffness working on it.

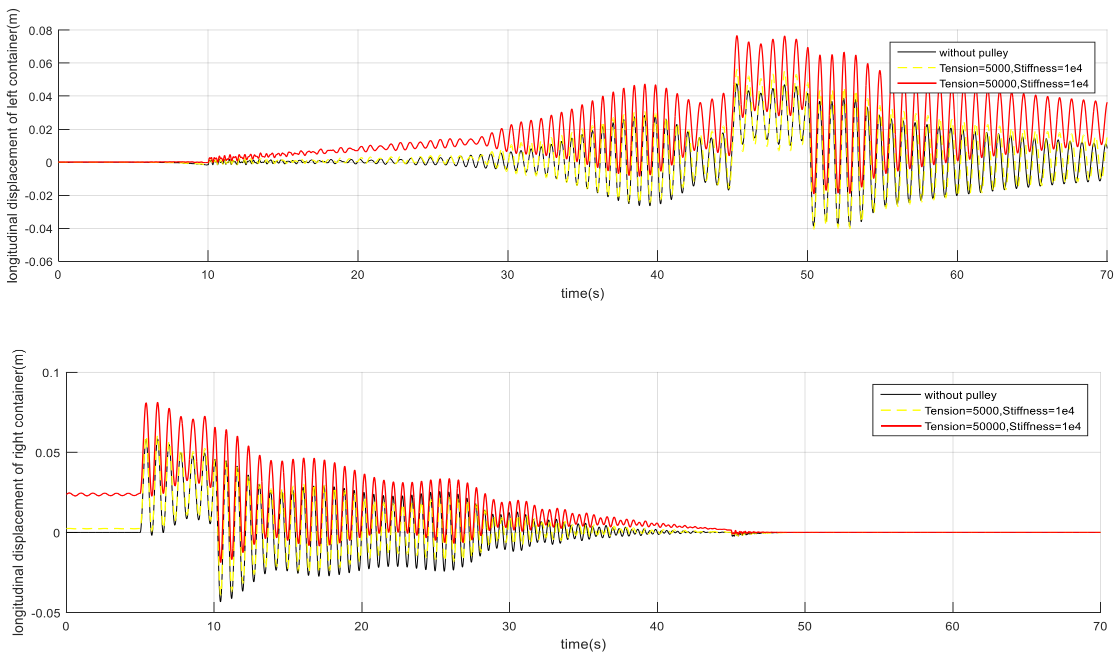


Figure 5 The longitudinal displacement of bilateral conveyance with different tension forces

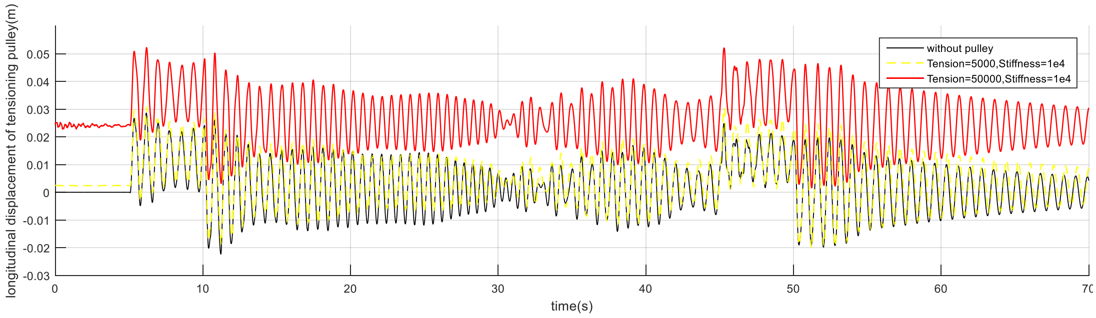


Figure 6 The longitudinal displacement of tensioning pulley with different tension forces

Lastly, the impact of tensioning force is also discussed by comparing different tensioning forces working on the pulley. As shown above, the dynamic amplitude of the conveyances and tensioning pulley aren't substantial changed with the increase of tensioning forces. The difference among different tensioning forces is the change of cables' static displacement at the conveyances and tensioning pulley caused by the tension forces. It can be concluded that the tensioning forces working on the pulley will not affect the longitudinal vibration shape of the traction system.

CONCLUSION

In this paper, the dynamics behavior of the traction system with tension at the pulley is investigated. The cables are spatially discretized using the AMM and the equations of motion are established by Lagrange equations of the first kind. From the numerical results, it can be concluded that the tensioning spring with proper stiffness can suppress the longitudinal vibration at the conveyance and tensioning pulley. However, the tensioning force working on the pulley at the bottom of the traction system can't play an effective role in the vibration suppression. Above all, this theoretical model can predict the response of the traction system, which will lay the foundation for the longitudinal vibration control of elevator hoisting system.

ACKNOWLEDGMENTS

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BIOGRAPHICAL DETAILS

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Experiment Method of Tribology Performance of Braking Material for High Speed Elevator

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Keywords: Testing rig, experiment method, high speed braking, friction coefficient.

Abstract. As a crucial brake component of the elevator, the braking performance of the safety gear is extremely important for the safety and reliability of the elevator. In this paper, an elevator safety gear braking testing rig based on disc brake model was established to simulate the actual high speed braking working condition, and some preliminary results have been obtained.

1 INTRODUCTION

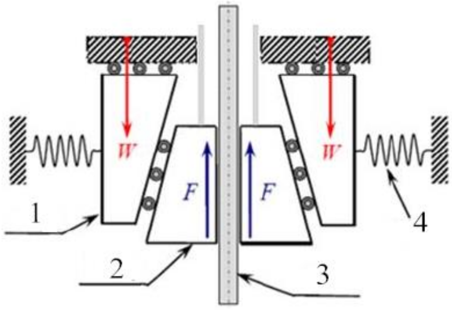
High-speed elevators generally refers to elevators operating at speeds greater than 3m/s so far, which are widely used in vertical transportation for high-rise buildings because of their high transport efficiency[1]. In consideration of the high operating speed, the technical requirements, especially the braking performance, should be more demanding than the conventional elevator. Thus, the safety gear, as a crucial brake component of the elevator [2], whose braking performance is extremely important for the safety and reliability of the elevator, should be emphatically studied. The friction interface problem of the safety gear is mainly to study the influence of the wedge material on the friction performance of the friction interface. Ao et al. study the braking process on progressive safety gear [3]. Chuan Jin reports a numerical simulation model for the safety gear frictional temperature rise [4]. Xiong X et al. study the impact of brake pressure on the friction and wear of carbon/carbon composites [5]. Therefore, the development and testing of wedge materials for the safety gear has become the concern of many elevator companies. Besides, Hyung-Min Ryu built a mechanical structure of high-speed elevator system for a dynamic load simulator [6], and Hussam et al. did a series of experiments on C/C–SiC brake pads for high-performance elevators [7].

In this paper, an elevator safety gear braking testing rig based on disc brake model was established to simulate the actual high speed braking working condition, and thus obtain the effective friction coefficient of the corresponding material.

2 ELEVATOR SAFETY GEAR

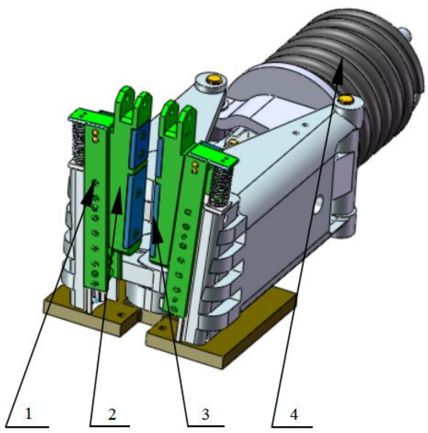
2.1 Elevator Safety Gear

For low-speed elevators, if emergency situations such as wire rope fracture, car over-speed operation, and car drop happen, the speed governor will trigger elevator safety gear by simultaneously lifting the wedges on both sides of safety gear, and the wedges will lock the rail because of the force acting by the springs. Fig.1 illustrates the structural diagram of low-speed elevator safety gear. For high-speed elevators, the spring is placed on the rear side of the safety gear, at the same time a leverage mechanism is introduced to amplify the rear spring force, and thus larger normal loads can be applied to the wedges. Fig.2 illustrates the structural diagram of a high-speed elevator safety gear.



1- gear 2- wedge 3- rail 4- spring

Figure 1 Structural diagram of low-speed elevator safety gear



1- bearing 2- wedge 3- breaking material 4- spring

Figure 2 Structural diagram of high-speed elevator safety gear

2.2 Theoretical model

Since setting the braking test in the vertical hoistway would cause operating risk and safety threat because of the shock on test tower caused by high initial velocity and inertia, the disc braking model is used to establish the testing rig. Fig.3 shows the dynamic analysis of these two models. Thus, the coefficient of fiction of the braking material can be derived according to the force analysis of the brake disc, as shown in Eq. 1.

$$\mu=\frac{F_t}{F_n}=\frac{M}{2\cdot p\cdot A\cdot R}=\frac{J\alpha}{2\cdot p\cdot A\cdot R}=\frac{Ja}{2p\cdot A\cdot R^2}$$

Where μ represents the COF (coefficient of friction); F_t represents the friction force during braking; F_n represents the normal pressure during braking; *A* represents the contact area; *J* represents the total inertia; α represents the angular deceleration of the brake disc; *R* represents the distance from the center of the brake pad to the center of the brake disc; *a* represents the linear deceleration at *R*; *M* represents the total mass of the actual car and passengers; *p* represents the surface pressure to the brake pad.

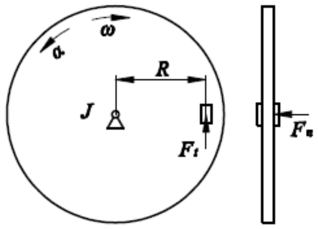
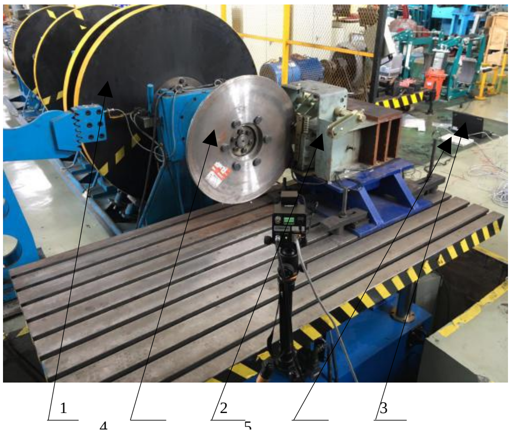


Figure 3 Dynamic analysis of these two models

2.3 Testing rig

The high-speed elevator safety gear testing rig is established as shown in Fig.4, which includes the driving system, the braking system and the data collection system.

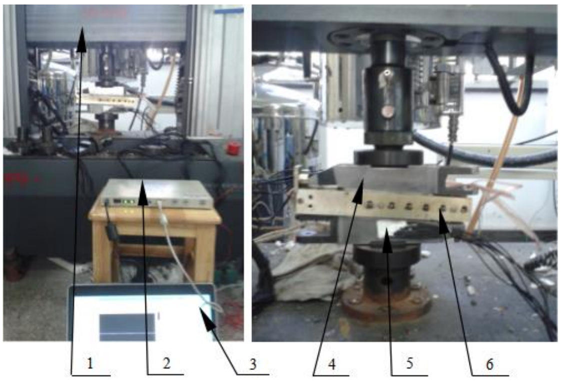


1- inertia disc 2- braking disc 3- safety gear 4- dynamic strain indicator 5- computer

Figure 4 High-speed elevator safety gear testing rig

The driving system mainly includes the motor, the load inertia, and the fixture of devices. The initial speed and load inertia are provided by the driving system. The braking system includes a brake disc and the safety gear. The normal load to the brake pad is exerted by the safety gear, so that a braking torque opposite to the direction of motion is generated in the contact area between the brake disc and the pad, which stops the whole system. The data collection system includes speed and braking pressure data collection systems. The speed data collection system is to collect the motor speed via the encoder during braking process, and thus the angular deceleration can be obtained by doing the discrete derivative of the angular velocity. The braking pressure data collection system is to dynamically monitor the pressure between brake disc and the material pad, the main method is collecting strain data from the wedge surfaces, then obtaining the braking pressure using the pressure-strain calibration curve.

The pressure calibration device is shown in Fig.5. Pressure is applied on the surface of the wedge, and the pressure and strain data simultaneously recorded during the compression process, so that the pressure-strain curve can be obtained.



1- compression testing machine 2- dynamic strain indicator 3- computer 4- wedges

5- bearing 6- strain gages

Figure 5 The pressure calibration device

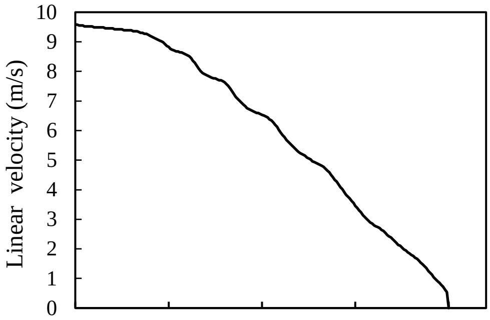
3 RESULTS AND DISCUSSION

3.1 Braking deceleration

Fig.6a) and Fig.6b) show the linear velocity and the linear deceleration at the center of the brake pad respectively. The linear deceleration increases rapidly after a relatively smooth period, indicating that the braking material starts to get in touch with the brake disc when t>0.33s. Besides, due to the instability of braking pressure, the linear deceleration may fluctuate greatly during the braking period. The smooth period between t=0s and t=0.33s can be considered as the consequence of the system damping after the motor has been cut off. Thus, the average deceleration a_{av} of the braking process can be calculated according to these two periods, as shown in Eq.2. It can be told that the average deceleration a_{av} meets the technical requirement for braking average deceleration from the national standard GB7588-2003, that is, the average deceleration of progressive safety gear should be 0.2g~1.0g with a free-falling car equipped with rated load.

$$a_{av} = \frac{V_s}{T_e - T_s} - \frac{V_i - V_s}{T_s} = \frac{9.362}{2-0.33} - \frac{9.582-9.362}{0.33} = 4.94 m/s^2$$

Where V_s represents the linear speed at the beginning of braking; V_{i} represents the linear speed at the beginning of recording; T_s represents the time when braking starts; T_{e} represents the time when braking ends.



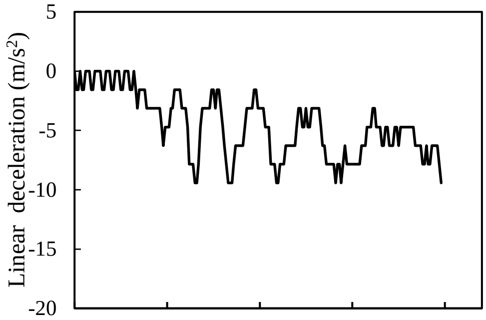


Figure 6 (a) linear velocity **(b) linear deceleration**

3.2 Braking pressure

Fig.7a) shows the pressure-strain calibration curve; it can be approximately considered as proportional relationship. The strain curve during the braking process is shown as Fig.7b). The impact stress wave caused by the contact and the quickly touching down of the wedges and brake disc lead to the instantaneous increase of the surface strain. The strain value fluctuates inevitably after entering the stable braking period mainly because of the differences of the pad- disc contact conditions at different areas, and accordingly the period of strain fluctuation is the rotation period of brake disc. Besides, due to the wear loss of the brake pad, the braking pressure curve will certainly show a damping trend during the braking period, as shown in Fig. 7b).

The average braking pressure $P_{av} = 18.5MPa$ can be obtained by the mean strain of the braking period $\varepsilon_{av} = -339\mu\varepsilon$ and the pressure calibration curve.

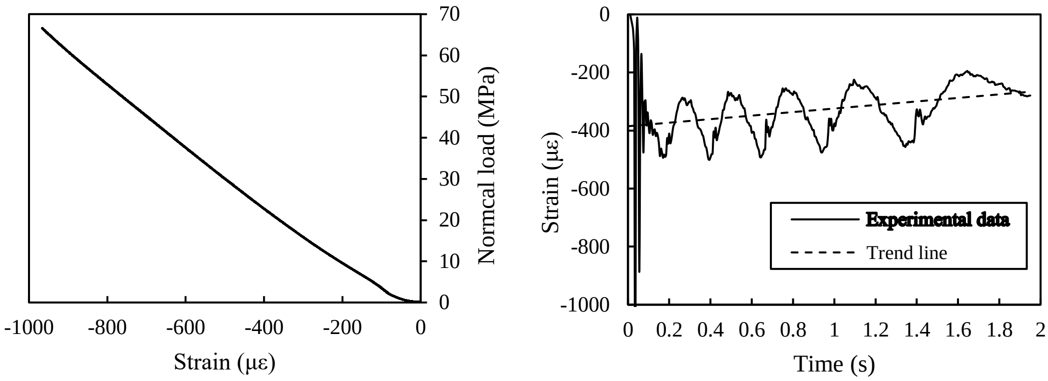


Figure 7 Curves for (a) Pressure-strain calibration and (b) Strain

3.3 Coefficient of friction

According to Eq.1 and data above, the COF of brake material can be obtained, as shown in Eq.3. The COF is 0.198, which is close to the theoretical friction coefficient. Thus, the experimental method proposed in this paper can effectively measure the COF of the brake materials.

$$\mu = \frac{J \cdot a_{av}}{2 p_{av} \cdot A \cdot R^2} = \frac{200 \times 4.94}{2 \times 18.5 \times 10^6 \times 0.0015 \times 0.3^2} = 0.198$$

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Experimental Investigation of Friction and Slip at the Traction Interface of Rope and Sheave

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Keywords: Rope-sheave interaction, Slippage, Digital image measurement

Abstract. In this paper, an exclusive testing rig was built to experimentally investigate the friction and slip at elevator traction interface under different traction conditions. The effect of pre-tension of rope is also discussed which indicates that the full slip occurs earlier and is greater under smaller pre-tension force. The experimental results indicated that slipping occurs at both ends of contact arc first, and then expands to the middle region gradually until the full slip along the sheave occurs. In addition, the full slip occurs earlier under lower rope pre-tension.

1 INTRODUCTION

Early studies on friction between rope and sheave are very limited. The most famous formula was published by Eytelwein [1], who concluded with a fully exponential distribution of normal pressure and rope tension at full slip based on Euler's solution [2]. However, the deduction of the Euler-Eytelwein formula only considered the axial forces in the rope, and Heller [3] extended Euler's work to account for radial shear stresses in the rope by introducing a new parameter of diameter ratio D/d. Usabiaga etc. [4] provided a new model based on a similar assumption as that of Euler-Eytelwein formula, in which the behavior of the wire rope was simplified as a series of elastic springs and body elements. However, Lugrís etc. [5] stated that modeling the rope as a linear spring, although very simple and efficient, is not accurate if the rope deformation along the segment in contact with the sheave is not considered.

In term of experiments, Nabijou and Hobbs [6-8] did a series of research on wire ropes bent over sheaves. The relative movements between wires within a wire rope were first investigated to form a valuable input in predicting wire rope fatigue and then the curvature of wires in single and double helices during bending was calculated to examine the bending strains of wires in a frictionless rope. In addition, Wiek [9] measured the distribution of the local contact pressure in a particular U groove pulley under full stick condition. Inspired by his experiment, Usabiaga etc. [10] developed a similar experimental method to investigate how the imbalance ratio, defined as ratio of rope tensions (T_1/T_2) at ends of the rope, influences the distribution of normal contact pressure by embedding a tri-axial piezo-electronic load cell inside the sheave and measuring the resultant normal and tangential forces.

Feyrer [11] experimentally studied the influence of rope bending stiffness on the contact angle, and identified high local stresses at the ends of contact arc. Such phenomenon was also observed by Ridge etc. [12], in which the cyclic bending strain in the wires of a six-strand right-handed Lang's lay steel wire rope was measured as it was running on and off a pulley. The result showed that all cyclic bending strain signals demonstrate a similar waveform with two peaks: one as the rope moves on the pulley and the other as it moves off. Chen etc. [13] developed a wireless detection system for dynamic rope skid in a friction lifting system. With two child nodes installed on the friction drive wheel and the guide pulley respectively, this detection system was able to send speed signals of friction drive wheel and guide pulley to the host node, which could then calculate the speed difference to monitor the rope skid phenomenon. The disadvantage of their measurement is apparent in that the results present only the overall slippage of the rope, and the information about the slippage distribution is missed. In Oplatka and Roth's work [14], this problem was overcome by fixing a camera and a floodlight on the sheave while rotating. The images of specific points on the rope were captured by the camera, which enabled the calculation of the relative speed between the sheave and the rope.

Although lots of experimental work has been done in investigation of traction drive system, the measurement of slippage evolution and distribution is scarce in the literature. In this work, a new experimental method for detecting dynamic slippage evolution by utilizing image processing technology was developed.

2 THE EULER-EYTELWEIN FORMULA

The traction is usually calculated with the Euler-Eytelwein formula. Figure 1a presents the traction interface between rope–sheave. Consider a differential rope element as shown in Fig.1b and assume the rope does not reach the state of gross slip and ignore the inertia forces, the equivalence of normal forces gives:

$$T(\alpha)\sin\frac{d\alpha}{2}+[T(\alpha)+dT]\sin\frac{d\alpha}{2}=p(\alpha)\frac{D}{2}d\alpha \quad (1)$$

The equivalence of tangential forces together with Coulomb friction law gives：

$$(T+dT)\cos\frac{d\alpha}{2}-T\cos\frac{d\alpha}{2}=fp(\alpha)\frac{D}{2}d\alpha \quad (2)$$

Since the differential angular extent $d\alpha$ is very small, Eq. (1) and Eq. (2) can be simplified as:

$$p(\alpha)=\frac{2}{D}T(\alpha) \quad (3)$$

$$dT=fp(\alpha)\frac{D}{2}d\alpha \quad (4)$$

Combining Eq.(3) and Eq.(4), the Euler-Eytelwein formula is obtained as below:

$$T_1/T_2=e^{f\theta} \quad (5)$$

where T_1 (high tension) and T_2 (low tension) are the rope tensions at both ends as shown in Fig.1a, θ the rope wrap angle, f the equivalent COF, $p(\alpha)$ the distribution of the normal line contact pressure and D is the sheave diameter.

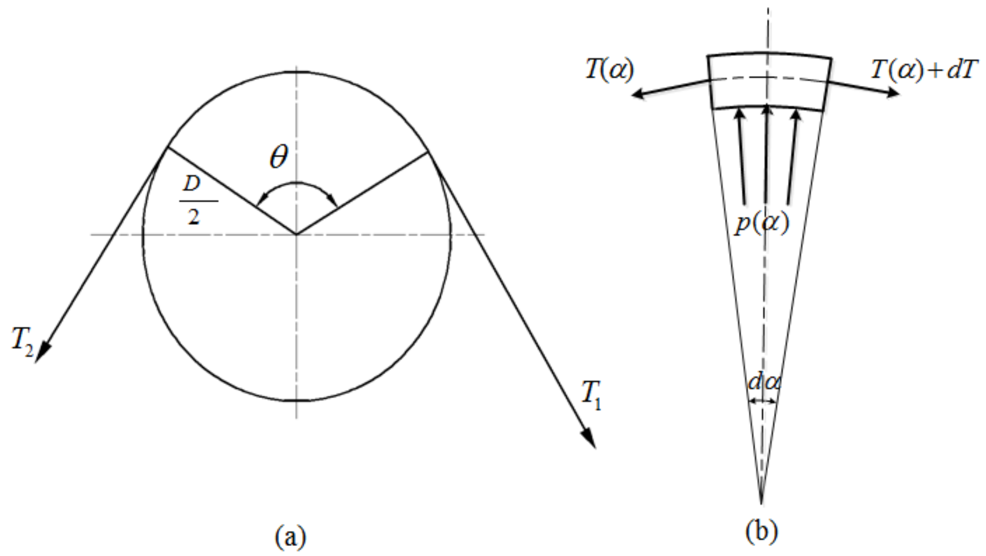

Figure 1 Geometry and notation for contact forces at rope–sheave interaction

The traction friction between the rope and the traction sheave results in a rope tension difference between two ends thus provides the traction capability. For a given traction system, the traction capability depends on the maximum friction at the rope-traction sheave interface which is governed by the Euler-Eytelwein formula. In principle, three traction conditions can be determined by comparing the rope tension ratio to the maximum value predicted by Euler-Eytelwein formula: (1) if $T_1 / T_2 < e^{f\theta}$, the traction of this system is large enough; (2) if $T_1 / T_2 = e^{f\theta}$, then the rope sliding is pending and any disturbance may trigger the slip, thus it is not safe; and (3) if $T_1 / T_2 > e^{f\theta}$, the rope slips on the traction sheave and the traction drive system is not safe as well.

Note that as the simplest theoretical model concerning traction drives, the Euler-Eytelwein formula neglects some factors, including the diameter ratio of sheave and rope, rope weight, the angular velocity of sheave and so on.

3 EXPERIMENTS

3.1 Test rig

The schematic of the test rig is shown in Fig.1. In this test rig, two identical motors (model KONE MX18) are used, one of which is used to drive the traction sheave, and the other is used to provide a constant torque load to simulate the moment due to the elevator system gravity. Two fly wheels are designed to account for the system inertia and a 5:1 gear box is adopted to increase the rotation speed of the fly wheel thus reduce the required fly wheel size.

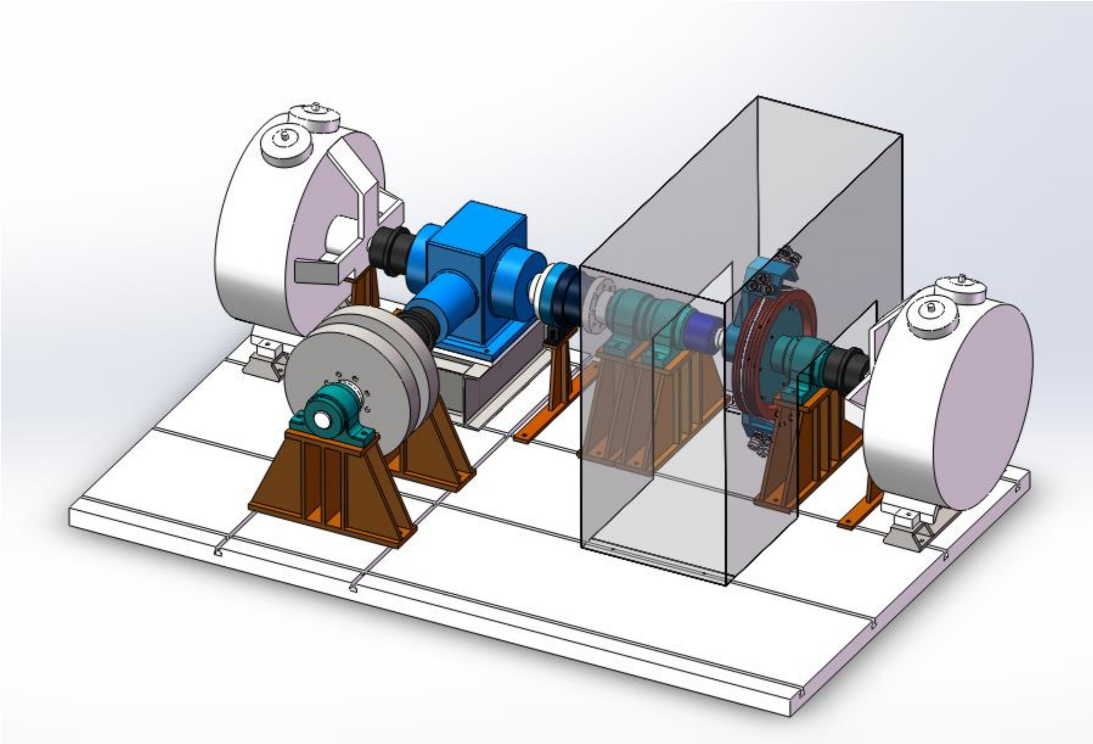

Figure 1 Schematic of the whole test rig

The traction interface consists of the traction sheave and six piece of wire ropes which are wrapped on the traction sheave symmetrically as shown in Fig.2. The ropes ends are fixed on a specially designed fork frame and the rope tension is adjustable with disk springs and screw assemble before the tests. The instantaneous tension force for each rope end during the test is measured with a load cell sandwiched between the disk springs and the nut.

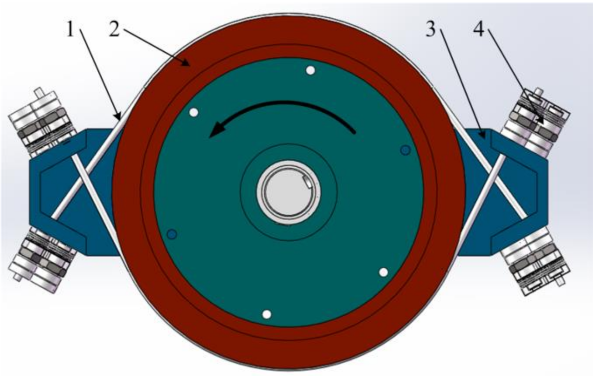

Figure 2 Assembly of traction system: 1-rope, 2-traction sheave, 3-fork frame, 4-rope fixture

3.2 Measurement of rope slippage

Figure 3 depicts the rope slippage measurement system, which consists of a high speed industrial camera (JAI SP-5000C-PMCL), LED light source, the marked object and a computer for image processing.

Figure 3 Structure of slippage measuring equipment

A series of mark sets are purposely made along the traction interface and each set consists of three mark points: one on the wire rope and two on the lateral side of the traction sheave near the edge as shown in Fig.4a. During the traction, the relative positions of those mark sets were recorded with a high speed industrial camera and the raw images are processed first with five detailed processes: image reading, threshold calculation and binarization, dilation, erosion and opening operation. Besides, the experiment for camera calibration was performed by using a circle with diameter 10 mm as calibration pattern. And the results show that the measurement error of displacement is roughly in the range of 0.2mm~0.22mm.

Geometrical relationship of slippage can be calculated after image processing. As shown in Fig.4b, two sequential pictures were taken by the camera and the mark set of former moment is represented as AB_1C while the mark set of latter moment is represented as AB_2C. The relative slippage between the two cases is very small because of very shot exposure time, the length of straight line B_1B_2 is assumed to be equal to the arc length of B_1B_2, thus the approximated slippage of the rope B_1B_2 can be derived as below:

$$\overline{B_1B_2} = \sqrt{\overline{AB_1}^2 + \overline{AB_2}^2 - 2\overline{AB_1} \bullet \overline{AB_2} \bullet \angle B_1AB_2} \tag{6}$$

There are two cases need to be considered in determining the value of $\angle B_1AB_2$.

Case1: if point A locates outside $\triangle CB_1B_2$:

$$\angle B_1AB_2 = \left|\angle CAB_1 - \angle CAB_2\right| \tag{7}$$

Case2： if point A locates inside $\triangle CB_1B_2$:

$$\angle B_1AB_2 = 2\pi - \left(\angle CAB_1 + \angle CAB_2\right) \tag{8}$$

Where

$$\angle CAB_1 = \arccos\left(\frac{\overline{AB_1}^2 + \overline{AC}^2 - \overline{B_1C}^2}{2\overline{AB_1} \bullet \overline{AC}}\right) \tag{9}$$

$$\angle CAB_2 = \arccos\left(\frac{\overline{AB_2}^2 + \overline{AC}^2 - \overline{B_2C}^2}{2\overline{AB_2} \bullet \overline{AC}}\right) \qquad (10)$$

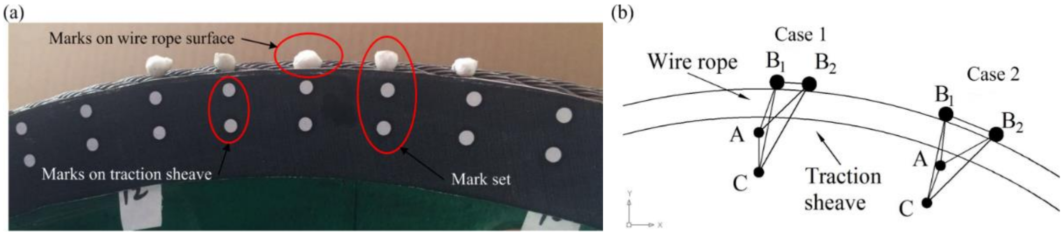

Figure 4 Calculation of slippage (a) mark points (b) Geometrical relationship of mark points

3.3 Experiment procedure

The drive motor together with the traction sheave is started with a preset constant acceleration and the load motor immediately outputs a constant loading torque during its start up. The constant inertial moment generated by the fly wheels due to the constant acceleration, together with the constant torque of load motor, are applied to the traction ropes as traction load. When the traction friction provided by this traction interface is lower than the total traction load, the rope slippage on the traction sheave occurs. The traction interface was monitored and the rope slippage was recorded with the camera during this process.

4 RESULTS AND DISCUSSION

4.1 Slip evolution in wrapping contact zone

Experiments were first conducted to investigate the slip evolution process in the wrapping contact zone, which was partitioned as head region, central region and end region. Note that the head region corresponds to the releasing part of the rope on the sheave. Camera measurement method was adopted to capture the rope slip behavior at the traction interface. In these experiments, about 130 pictures of marked regions in total were taken. However, only 6 pictures with equal time interval in each marked region were selected for slippage demonstration. The selected pictures in different regions are shown in Fig.5 and an obvious relative slip was observed between the marked points on the rope and the corresponding reference points on the sheave in both head and end regions as shown in Fig.5a and Fig.5c, especially in the last picture of each mark set. However, the relative slip is trivial and ignorable in mid region as shown in Fig.5b.

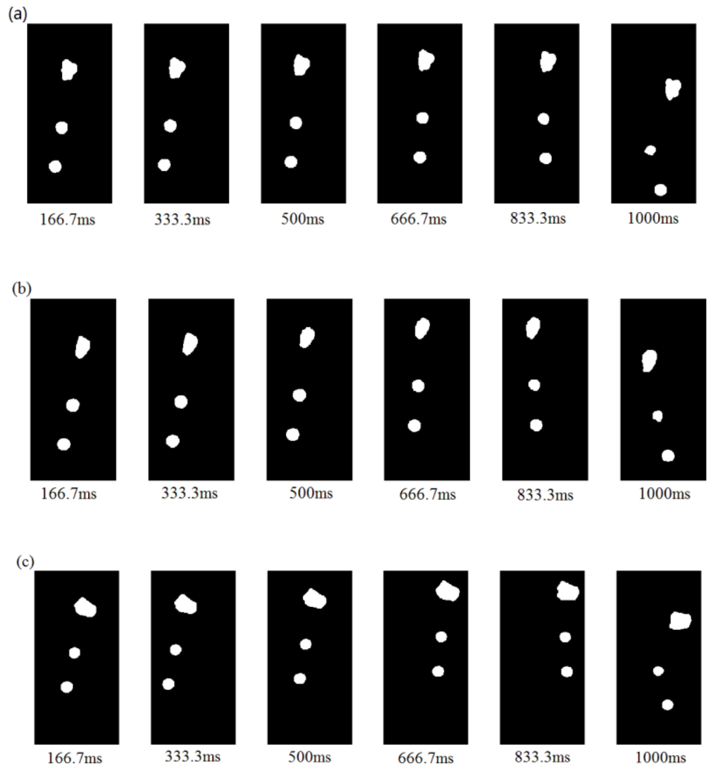

Figure 5 Image sequences of mark points in different regions: a. Head region; b. Mid region c. End region

Following the process in section 2.2, the slippage in different regions can be quantitatively obtained and the slippage developments in different regions are compared in Fig. 6. It can be observed that the slippage in both head and end regions increases sharply after it reaches the approximate critic value 800ms. However, the slippage development of mid region is extremely slow in the whole process. Based on the slippage data, it can be postulated that slipping occurs firstly at both ends of contact arc, and then expands to the middle region gradually until the full slip along the sheave occurs. Note that those marked points will be out of the shooting scope of the camera at about 1000ms, thus the fully slip in the whole contact region is not captured.

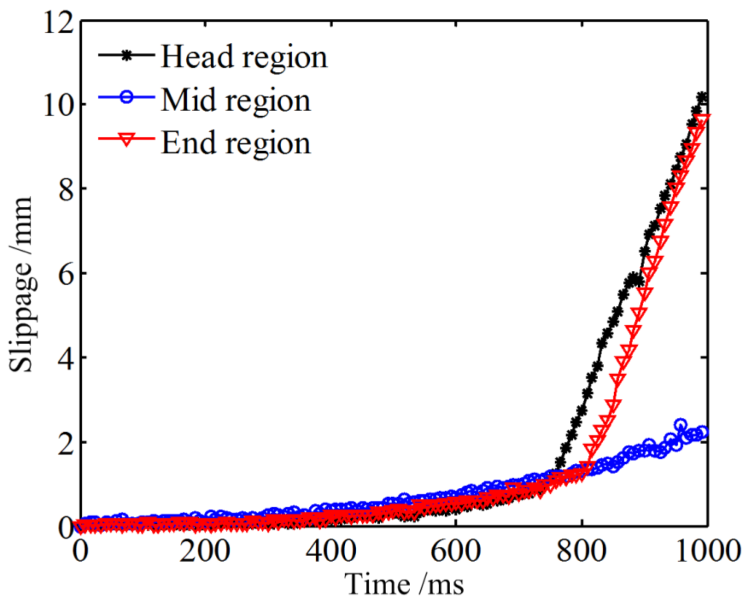

Figure 6 Experimental observation of slippage development in different regions

4.2 Effect of rope preload on slippage development

Two experimental tests, with a rope preload of 3500N and 1800N respectively, were carried out to investigate the effect of rope preload on the slippage development and the results are shown in Fig.7. Note that rope tension at both ends can be obtained by monitoring the corresponding reaction forces and the torque applied to the traction sheave by rope was induced by an imbalance of tension between these two ends of the rope. As shown in Fig.7, this imbalance keeps increasing until the friction force between the rope and sheave reaches the maximum value, after which, a full slip occurs and the rope-end force becomes approximately constant. It can also be observed from Fig.7 that with a lower rope preload, the full slip occurs earlier. Note that effect of pretension on equivalent COF can be also demonstrated by these tests. The rope-end force when slip is occurring can be experimentally measured and the wrap angle is known as 120° in the experiments, then according to Euler's equation, the equivalent COF values can be obtained as 0.405 for a preload of 1800N and 0.491 for a preload of 3500N respectively.

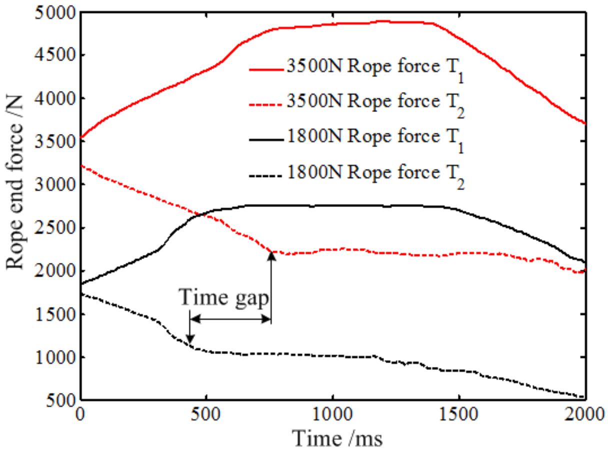

Figure 7 Instantaneous rope-end force behavior under two different pre-loads

The measured instantaneous slippage of head region under two preload levels are plotted in Fig.8. In both cases for head region, a sharp increase of slippage, which means a full slip, occurs at around t=750ms and in low preload case rope slips a little bit earlier. It also can be observed that slippage for 1800N case increases more sharply than that for 3500N case once full slip initiates in these specific experimental tests.

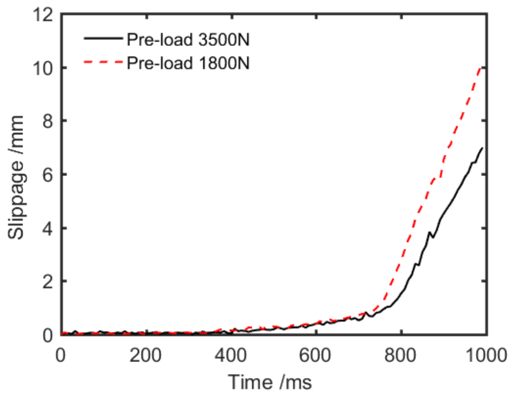

Figure 8 Slippage development of head region under different pre-tension forces

In order to double-check the effect of pre-load on the slippage, the instantaneous rotation speeds of drive motor and load motor were measured with encoders and the speed difference indicates the slippage. The results are plotted in Fig.9a and Fig.9b, corresponding to a preload of 3500N and 1800N respectively. Obviously, a speed difference between drive motor and load motor appears at about t=1050ms for 3500N case and t=800ms for 1800N case, which double confirmed the previous conclusion that full slip occurs earlier under low preload condition. Note that the occurring time of slip measured by camera measurement method and by motor speed difference method are different because the time sequences between both measurement methods do not align.

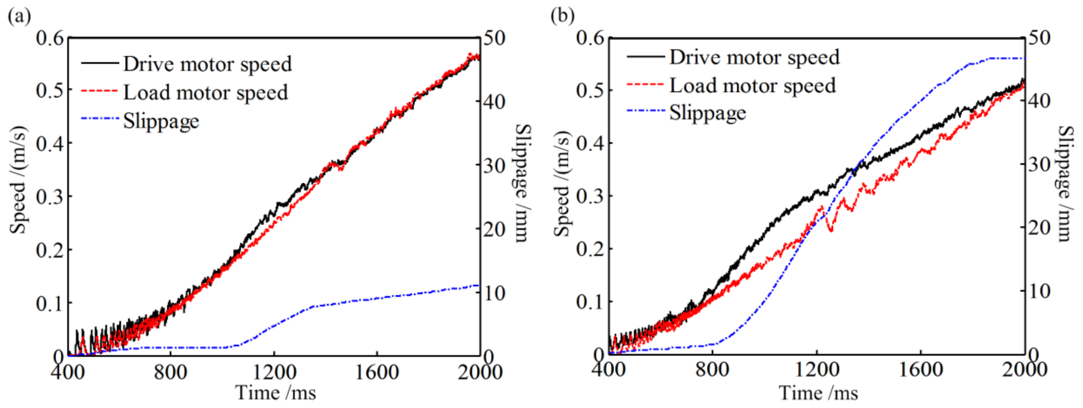

Figure 9 Instantaneous speeds of drive motor and loading motor with a preload of (a) 3500N (b) 1800N

5 CONCLUSIONS

In this paper, some experiments were performed to investigate slippage development at the traction interface between the rope and the sheave. Based upon these results, the following conclusions are obtained:

1. The camera measurement method for slippage monitoring proposed in this work is effective.

2. For the rope-traction sheave interaction model presented in this work, the slip occurs first at both ends, and then expands to the middle region gradually.

3. The full slip occurs earlier under lower pre-load.

6 LITERATURE REFERENCES

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History of the Safety Gear

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Abstract. The safety gear is regarded as the last line of defence in the relatively safe world of lifts. Industry contemporaries recall Elisha Otis declaring "All safe" after cutting the ropes on a platform upon which he was standing and the safety gear preventing his uncontrolled descent. The design of safety gears has moved on significantly from an original proposal to place a bag of feathers in the lift pit to designs that now arrest uncontrolled movement in ascent. This paper is a developing research project which will look at UK patents and standards and tracks the development of the safety gear from the embryonic days of lift installations to the present day. It will contribute to knowledge by bringing together a number of sources of information not previously brought together into a single paper and thus provide a consolidated history of the safety gear.

1 INTRODUCTION

A literature search has revealed limited historical information about safety gears fitted to passenger lifts. Most literature is of a technical or sales nature rather than being in a historical context.

Manufacturers of lifts have been taken over by companies over the years and in many cases historical information about a company's history has been lost. There are a few exceptions, but most information is about the company rather than their products.

It is common knowledge that Elisha Graves Otis designed the first safety gear and his paper presented to the Newcomen Society American Branch in New York 1945 gave good insight into the man himself as well as the product. Another author, John Inglis, based in Australia presented a paper at Elevcon in 1998 entitled "Evolution of Safety Gear" which has provided some interesting drawings for this paper however the paper did not look at changes in standards in a chronological way.

Other historical books were located including "Electric Lift Equipment for Modern Buildings" (1923) by Grierson and "Electric Lifts" by R S Phillips (1973*)* "Giving rise to the modern city" by Jason Goodwin and "A history of the passenger elevator in the 19th century" by Lee Gray (2002).

Two Codes of Practice published by the British Industries National Council in 1935 and 1943 have also been located.

Various books of a technical nature were located however historical and developmental information about safety gears was not covered. The British Standards published since 1970 were reviewed including BS 2655-1 (1970), BS5655-1 (1979) and BS5655-1 (1986) after which the European Norm standards saw the dropping of the BS prefix for EN. These included EN81-1 (1990) including Amendment A3, EN81-20 (2014) and EN81-50 (2014). Manufacturers were also contacted for documentation and information will be reviewed as and when it is received. The literature review has revealed that there is very little information about the development of the safety gear and finding information is difficult but not impossible.

The elimination of the safety gear has been proposed on many occasions which makes this research a piece of legacy work that may become important as a future student can take it forward from the cessation point of the final research such that a chronology from its invention in 1853 can be established. This work can therefore be concluded as being novel and in the educational and public interest.

2 ELISHA OTIS

Otis was born in 1811 in Halifax, Vermont to Stephen Otis and Phoebe Glynn. He moved away from home at the age of 19, eventually settling in Troy, New York, where he lived for five years employed as a wagon driver. In 1834, he married Susan A. Houghton. They would have two children, Charles and Norton. Later that year, Otis suffered a terrible case of pneumonia which nearly killed him, but he earned enough money to move his wife and three-year-old son to the Vermont Hills on the Green River. He designed and built his own gristmill, but did not earn enough money from it, so he converted it into a sawmill, yet still did not attract customers. Now having a second son, he started building wagons and carriages, at which he was fairly skilled. His wife, Susan, later died, leaving Otis with two sons, one at that time being age 8 and the other still a young child [10].

At 34 years old and hoping for a fresh start, he married Betsy, and moved to Albany, New York. He worked as a doll maker for Otis Tingely. Skilled as a craftsman and tired of working all day to make only twelve toys, he invented and patented a robot turner. It could produce bedsteads four times as fast as could be done manually (about fifty a day). His boss gave him a $500 bonus. Otis then moved into his own business. At his leased building, he started designing a safety brake that could stop trains instantly and an automatic bread baking oven. He was put out of business when the stream he was using for a power supply was diverted by the city of Albany to be used for its fresh water supply. In 1851, having no more use for Albany, he first moved to Bergen City, New Jersey to work as a mechanic, then to Yonkers, New York, as a manager of an abandoned sawmill which he was supposed to convert into a bedstead factory.

In his spare time, he designed and experimented with his old designs of bread-baking ovens and train brakes, and patented a steam plough in 1857, a rotary oven in 1858, and, with Charles, the oscillating steam engine in 1860. Otis contracted diphtheria and died on April 8, 1861 at age 49 owning a factory worth not more than $5,000 and employing only 8 or 10 men. [1]

3 THE INVENTION OF THE SAFETY GEAR

"The significant influence of lifts dates from the invention by Elisha Graves Otis of a device capable of keeping a lift from falling even though the hoisting ropes should break"[1]

In 1851 Elisha Graves Otis went to Bergen, New Jersey and then a year later to Yonkers, New York in his employ as master mechanic of the bedstead plants in which his employer, Josiah Maise, was an owner. It was here that he came face to face with his destiny and he designed and installed the first lift equipped with an automatic device to prevent it from falling. [1]

He was destined to move to California however an unsolicited order for two safety lifts had been received from a Mr Newhouse, a furniture manufacturer in whose plant at 275 Hudson Street, New York, a serious accident had just been experienced. The order for these two lifts marked the beginning, in 1853, of the now worldwide association of elevators and the name Otis.[1]

In 1853 an exhibition was held at the Crystal Palace in New York City at which Elisha Graves Otis demonstrated his confidence in his own product by standing on the platform of the lift erected at the exhibition, raising the platform well above the heads of the assembled crowd, and then at the most dramatic point in his oratorical exposition, cutting the rope by which the platform was suspended. Those who had morbidly anticipated a leg breaking crash, however disappointed, were nevertheless impressed with the effectiveness of the Otis Safety when, as a matter of fact, nothing happened. [1] It is said that after the descending platform was arrested that Elisha Otis uttered the words "All Safe Gentlemen". The frame that he used for the demonstration was sold to P T Barnum, the "World's Greatest Showman".

The New York Tribune reported the exhibition and made mention of the Otis invention however it should be noted that they referred to the lift at the exhibition as one for hoisting goods and it was not until three years later that the first passenger lift was manufactured by Otis and installed in a five-story building on the north-east corner of Broome Street and Broadway which belonged to E V Haughwout & Co, dealers in china and glassware. [1]

4 DEVELOPMENT OF SAFETY GEAR DESIGN

According to Inglis[2] the earliest story known about safety of persons whilst travelling in a lift dates back to a Sultan who required a means of lifting people to the upper floor of his castle. It is said that a large bag of feathers was placed in the pit and one of his servants rode in the lift car whilst the rope was cut. The servant apparently survived the fall with only a broken leg and the Sultan therefore concluded that no one would be killed whilst using his lift.

In the early days car guide rails were often timber and a knife action safety gear would embed itself into the timber guides which would have to be replaced after each application. This was obviously an unsatisfactory situation and often rendered a lift out of service for a long period of time whilst replacement guide sections were obtained. This type of safety gear, as with many others, was also plagued by nuisance tripping in the event that the knife blade ran too close to the timber guides or a build-up of detritus would cause it to operate.

Inglis goes on in his paper to a further development where an Italian invented a method of preventing injury in the event of free fall or an overspeed condition in the down direction. This is the first located mention of an overspeed condition. The invention consisted of some rods across the car above the passengers' head with the rods terminating in two rubber diaphragms at their ends. In the event of overspeed in the down direction a passenger would hold onto the bar with the diaphragms taking the force out of the impact when the car hit the buffers. There was the obvious question of how many passengers could be protected by such a device.

The development of high speed lifts necessitated the development of a new type of safety. It is elementary that the purpose of a safety device is not merely to stop the lift platform, since this could be done with absolute certainty by simply letting the platform hit the bottom of the hatchway, but rather to bring the platform to a sufficiently gradual stop to prevent injury. The early safety, which contributed so greatly to the fame and fortune of Elisha Graves Otis, was of the instantaneous type which operated only in the event of slack or broken ropes and was useful only because it applied when the lift had barely started to fall and before it had attained a downward speed greatly in excess of the normal speed, which was slow. Obviously, with a high-speed lift it would be almost as disastrous to stop the lift at high speed with a safety of instantaneous type as it would be too hit the bottom, or at any rate the stop would be more sudden than the human body could stand without injury. [1]

According to Grierson[3] up to about the year 1880 cast iron racks or ratchets were attached to the guides, and a pair of dogs, fixed at the top of the car attached to the single suspension rope, and operated by springs, formed the safety gear. Note the single rope, a situation no longer permitted for lifts although still seen in mine winding and cable car applications. When the rope failed, the springs that operated the dogs engaged with the racks on the guide posts and immediately brought the car to a dead stop. Grierson also states that "safety gear is not ordinarily fitted to counterbalance weights, only the car."

The next important development, according to Grierson, appeared around 1893 and is still extensively in use in Great Britain (bear in mind Grierson was published in 1923) was cam type guide grips. It consists of four serrated steel cams, mounted on two turned steel rods, that, when the necessity arises, rotate and bring the cams into contact with the guide rails or wood backing.

This safety gear was only suitable for slow speed cars (100 ft/min) due to being of practically instantaneous action. The design would also only protect against a too rapid descent of the lift car and was useless for excessive speed in the upward direction. Grierson noted that various manufacturers used different methods of safety gear activation including slack rope activation and a separate safety line connected between the car and counterweight.

In 1878 an overspeed governor of the fly ball type was invented for the purposes of operating a progressive safety gear. This was invented by Charles R Otis.[1]

5 HISTORY OF THE BRITISH STANDARDS INSTITUTION

The history of the British Standards Institution can be found on the BSI website [11]

The chronology shows an interesting start to the standards movement in the UK and the period when other groups were doing the work can be accounted for by looking at this timeline.

According to the website [11] BSI was formed in 1901 by Sir John Wolfe-Barry - the man who designed London's Tower Bridge - BSI was the world's first National Standards Body. The original BSI committee met for the first time on the day Queen Victoria died – 22 January 1901. One of the first standards it went on to publish related to steel sections for tramways.

The BSI Kitemark was first registered by BSI on 12 June 1903 – the same year in which Harley Davidson, Crayola crayons and the Tour de France were born. Originally known as the British Standard Mark, it has grown into one of Britain's most important and most recognized consumer quality marks.

In the years between 1914 and 1945 standardization grew. This is quite interesting as these dates coincide with the start of the first world war and the end of the second world war.

During the 1920s, standardization spread to Canada, Australia, South Africa and New Zealand. Interest was also developing in the USA and Germany.

In 1929 the Engineering Standards Committee was granted a Royal Charter. A supplemental Charter was granted in 1931 changing the name, finally, to The British Standards Institution.

Again, this is significant when you look at the 1935 and 1943 Codes of practice published by the Building Industries National Council (BINC) as in 1931 BSI was a standards organisation in its own right and it would appear the BINC were a competitor.

In 1942 the British Government officially recognized BSI as the sole organization for issuing national standards. Why therefore did BINC publish a Code of Practice in 1943? The answer may lay in the war years as between 1939 and 1945, during World War II, ordinary standards work was stopped, and efforts were concentrated on producing over 400 'war emergency standards'.

At the end of the war in 1945 a BSI Kitemark licence was issued for copper pipe fittings that is still going strong today – it's the longest running BSI Kitemark.

6 PRE-BRITISH STANDARD CODES OF PRACTICES

Prior to publications by the British Standards Institution two standards have been located. These were published by the Building Industries National Council in 1935 and 1943.

6.1 Building industries national council, code of practice for the installation of lifts and escalators, 1935

It is interesting to note who represented various parties on the committee including private companies. There were only 12 people on the committee, but notable lift companies represented were The Express Lift Co (J W Stevens), Waygood Otis (D W R Green and W W Weaver), J & E Hall (E M Medway).

This is the earliest located code of practice or standard so far located about lifts.

The technical requirements with respect to safety gears can be found in appendix 1 of this paper.

6.2 Building industries national council, code of practice for electric passenger and goods lifts and escalators, 1943

The preface to the 1943 edition makes interesting reading and unusually by modern standards makes specific reference to the passing of one of the committee members.

It is also interesting to note that the title of the code of practice changed with the words "for the installation of" being dropped.

The 1943 standard was far more specific and lost some of the quaint features that were in the 1935 standard such as the labelling of overspeed governors with brass labels although attendants in the car remained!

The section applicable to safety gears moved from section 12 to section 16 and the technical requirements stated therein can be found in appendix 2 of this paper.

7 DEVELOPMENT OF BRITISH & EUROPEAN STANDARDS

Further work is required into researching of older standards and codes of Practice as there is clearly a gap between Otis demonstrating his safety gear in 1853 and the 1970 edition of BS2655-1. It is understood that there was a BS2655 published around 1958 but even then, this leaves a gap of over 100 years in documentary evidence of design.

Standards since 1970 have developed as follows:

1970 BS 2655-1:1970: Specification for lifts, escalators, passenger conveyors and paternosters. General requirements for electric, hydraulic and hand-powered lifts

1970 BS 2655-7:1970: Specification for lifts, escalators, passenger conveyors and paternosters. Testing and inspection

1979 BS 5655-1:1979, EN 81-1:1977: Lifts and service lifts. Safety rules for the construction and installation of electric lifts

1986 BS 5655-1:1986, EN 81-1:1985: Lifts and service lifts. Safety rules for the construction and installation of electric lifts

1986 BS 5655-10:1986: Lifts and service lifts. Specification for the testing and inspection of electric and hydraulic lifts

1995 BS 5655-10.1.1:1995: Lifts and service lifts. Specification for the testing and examination of lifts and service lifts. Electric lifts. Commissioning tests for new lifts

1995 BS 5655-10.2.1:1995: Lifts and service lifts. Specification for the testing and examination of lifts and service lifts. Hydraulic lifts. Commissioning tests for new lifts

1998 BS EN 81-1:1998+A3:2009: Safety rules for the construction and installation of lifts. Electric lifts

1999: PAS 32-1:1999: Specification for examination and test of new lifts before putting into service. Electric traction lifts

1999 PAS 32-2:1999: Specification for examination and test of new lifts before putting into service. Hydraulic lifts

2007 BS 8486-1:2007+A1:2011: Examination and test of new lifts before putting into service. Specification for means of determining compliance with BS EN 81. Electric lifts

2007 BS 8486-2:2007+A1:2011: Examination and test of new lifts before putting into service. Specification for means of determining compliance with BS EN 81. Hydraulic lifts

2014 BS EN 81-20:2014: Safety rules for the construction and installation of lifts. Lifts for the transport of persons and goods. Passenger and goods passenger lifts

2014 BS EN 81-50:2014: Safety rules for the construction and installation of lifts. Examinations and tests. Design rules, calculations, examinations and tests of lift components

The requirements of the 1935, 1943 and 1970 standards can be found in the appendices to this paper.

8 FACTORS AFFECTING SAFETY GEAR DESIGN

A number of factors seem to have driven safety gear design over the years.

- The desire to prevent an uncontrolled descent.
- The relationship between speed and safety gear design (principally to protect passengers against injury)
- The desire to protect against uncontrolled ascent as well as descent.

As the research progresses more categories may be identified although it currently appears that from 1853 onward for over a century it was simply a desire to prevent uncontrolled descent to allay passenger fears about this risk.

9 LIFT SPEED AND SAFETY GEAR SELECTION

The 1935 code of Practice allowed instantaneous safety gears to be used on lifts up to 250 f.p.m (1.25 m/s). This was reduced in the 1943 code of practice reduced this speed to 200 f.p.m (1.0 m/s). The reasoning behind this is not known.

BS2655-1 (1970) stated 2.12.3 Safety gears of the instantaneous type may be used for lift cars having a contract speed not exceeding 0.75 m/s or 150 ft/min. A further reduction in the permissible speed.

There was a shift in the 1979 standard BS5655-1 (1979) which introduced the buffered effect for the first time: 9.8.2.1 Car safety gear shall be of the progressive type if the rated speed of the lift exceeds 1.0 m/s. It can be (a) of the instantaneous type with buffered effect if the rated speed does not exceed 1.0 m/s (b) of the instantaneous type if the rated speed does not exceed 0.63 m/s

The buffered effect was rarely used but when it was it involved the sling having an additional and independent travelling section underneath it which had an instantaneous safety gear attached. This safety gear would operate and the forces were reduced by the lift car sling being separated from the safety gear by devices such as hydraulic pistons (rather like buffers) which would take the forces out of the operation.

Different manufacturers came up with different designs for both instantaneous and progressive safety gears.

The BS5655-1 (1986) standard simply mirrored the BS5655-1 (1979) standard as did the EN81-1 (1998) + A3 (2009) standards.

However, the publication of EN81-20 (2014) saw the end of the buffered effect with the wording being amended as follows: 5.6.2.1.2.1 Car safety gear (a) shall be of the progressive type or (b) may be of the instantaneous type if the rated speed of the lift does not exceed 0.63 m/s

The philosophy behind the change in speed between the 1970 and 1979 standards for instantaneous safety gears is not known however it can be seen that the EN81-20 (2014) standard limits instantaneous to a maximum speed of 0.63 m/s whereas the BS2655-1 (1970) standard allowed the higher speed of 0.75 m/s so despite the buffered effect being removed the reduced speed from the 1979 standard is still adopted. The speeds allowed prior to the 1970 standard were up to 250 f.p.m (1.25 m/s), unthinkable by todays standards.

10 FORCES

Initial considerations were that considerations with respect to forces imposed on passengers may have caused the change.

Standards do not specifically state forces applied on a passenger during the operation of a safety gear and the best that can be referred to is the forces permissible in the event of a buffer collision.

BS2655-1 (1970) simply stated in clause 3.4: "Buffers shall be installed under all cars and counterweights. Springs buffers or buffers of rubber or timber may be used."

By today's standards a completely meaningless statement but clearly there is no consideration as to the forces permitted to be placed upon a passenger. In particular it should be noted that timber buffers were permitted.

BS5655-1 (1979) saw the introduction of consideration into forces imposed on a passenger.

Clause 10.4.3.3 stated "With the rated load in the car, in the case of free fall, the average retardation during action of the buffers shall not exceed gn. Retardation of more than 2,5 gn shall not be longer that 1/25 of a second. The speed of impact on the buffers to be considered is equal to that for which the stroke of the buffer is calculated (see 10.4.3.1 and 10.4.3.2)

Whilst the introduction of forces into the standards came with the 1979 standard it is unlikely to account for the difference in speed with reference to safety gear speed between the 1970 and 1979 standards especially with the 1935 and 1943 Codes of Practice reducing permissible speeds as well.

BS5655-1 (1986) mirrored the 1979 standard with the exception the 1/25th of a second was replaced by 0.04 seconds thus replacing the fraction with a decimal.

The introduction of the 2014 standard saw an additional component being introduced when the maximum peak retardation was introduced at 6 gn but the 2.5 gn for a maximum of 0.04 seconds remained.

11 DIRECTION OF LIFT AND SAFETY GEAR SELECTION

As previously stated in BS2655-1 (1970) "A car safety gear shall **not** operate to stop an ascending lift car. If an ascending lift car is to be stopped on account of overspeed then a safety gear shall be fitted to the counterweight for this purpose."

The wording changed in BS5655-1 (1979) to the following: 9.8.1.1 The car shall be provided with a safety gear capable of operating only in the downward direction and capable of stopping a fully laden car, at the tripping speed of the overspeed governor, even if the suspension devices break, by gripping the guides, and holding the car there.

It should be noted that the overspeed governor was introduced into the wording of standards at this point as a mandatory clause for all electric lifts including those with instantaneous safety gears albeit, as previously mention Charles Otis invented the flyball overspeed governor for progressive safety gears in 1878.

Prior to this BS2655-1 (1970) offered an overspeed governor as an option as follows: 2.12.3 The safety gear shall operate to stop and sustain the lift car with contract load in the event of failure of all suspension ropes or chains or their attachments, or in the event of the lift car exceeding a predetermined speed in the downward direction, when the safety gear is operated by an overspeed governor.

BS5655-1 (1986) adopted the same wording as the 1979 standard.

It was not until EN81-1 (1998) Amendment 3 (2009) that the requirement to stop an ascending lift car came into being with clause 9.8.10 which stated: 9.10 A traction lift shall be provided with ascending car overspeed protection means conforming to the following:

9.10.1 The means, comprising speed monitoring and speed reducing elements, shall detect uncontrolled movement of the ascending car at a minimum 115% of the rated speed, and maximum as defined in 9.9.3, and shall cause the car to stop, or at least reduce its speed to that for which the counterweight buffer is designed.

The introduction of EN81-20 (2014) varied the wording to 5.6.1.1 Devices, or combinations of devices and their actuation shall be provided to prevent the car from (a) free fall, (b) excessive speed, either downwards, or up and down in the case of traction lifts, (c) unintended movement, with open doors (d) in the case of hydraulic lifts, creeping from a landing level.

12 CONCLUSION

The history of the safety gear is an interesting subject and it has been identified that there is a long gap of inactivity in design from 1853 to 1935 which requires further investigation.

Two codes of Practice have been located that were published prior to the British Standards Institution (BSI) publishing lift related standards. BSI was formally recognised by the Government as having the sole responsibility for publishing UK standard in 1942 albeit BSI was formed in 1901. The 1935 British Industries National Council document was published prior to BSI being the national body but the 1943 standard was published just after and is therefore likely to have been in the process of being published when BSI were awarded their status.

Sources thus far identified show development from a state of concern about the risk of falling to the quality of the fall should it happen with the introduction of acceptable forces relative to speed.

Furthermore, the need to address other technical issues such as uncontrolled movement in the ascending mode have also caused developments in safety gear design. Research will continue to establish development of the safety gear from its initial design in 1853 to the present day.

In summary the highlights relative to changes in standards have been as follows:

In 1935 the permissible **rated speed** of a lift having an instantaneous safety gear was 250 f.p.m (1.25 m/s). This has gradually reduced over the years to 200 f.p.m (1 m/s) in 1943, 150 f.p.m (0.75 m/s) in 1970 and then to 126 f.p.m (0.63 m/s) for standards in 1979, 1986, 1998 and 2014. A speed reduction of 50%!

The major changes in standards with respect to **force** have been that the 1970 and previous standards made no mention of forces however the 1979 standard brought in a requirement of 2.5 gn for no longer than 1/25th of a second (this being relative to buffers). In 1986 there was no change in quantum but the 1/25th of a second changed to 0.04 seconds. This remained true until the 2014 standard when an additional requirement that the maximum peak retardation didn’t exceed 6 gn was introduced.

With respect to **direction** the 1970 standard stated that the safety gear did not need to stop an ascending car. The 1979 standard introduced that the safety gear needed to be able to stop a fully laden lift car at the tripping speed of the overspeed governor in the down direction. The 1986 and

1998 standards mirrored this but the introduction of the A3 amendment in 2009 made ascending protection necessary and introduced the term uncontrolled upward movement.

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BIOGRAPICAL DETAILS

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David is a Chartered Engineer and Chairs the Educational Trust that manages these Symposiums. He has been in the lift industry since he left school in 1980.

APPENDIX 1:

BUILDING INDUSTRIES NATIONAL COUNCIL, CODE OF PRACTICE FOR THE INSTALLATION OF LIFTS AND ESCALATORS, 1935 – EMERGENCY SAFETY DEVICES

12(a) Every power-driven lift suspended by wire ropes shall be provided with a car safety gear, attached to the car frame and preferably placed beneath the car. The safety gear shall be capable of stopping and sustaining the car with full contract load in the car.

12(b) Every power-driven lift having a travel exceeding 18' shall be equipped with an overspeed governor device which will operate to apply the safety gear in the event of the speed of the car in the descending direction exceeding a predetermined limit.

12(c) The application of the safety gear shall not cause the car platform to become out of level in excess of ¼" per foot, measured in any direction.

12(d) When the safety gear is applied, no decrease in the tension of the governor rope or motion of the car in the descending direction shall release the safety gear.

12(e) It is permissible to release the safety gear by reversing the direction of motion of the machine.

12(f) The safety gear shall operate to stop and sustain the car in the event of failure of the suspension ropes or in the event of the lift exceeding a predetermined maximum speed in the descending direction when a speed governor is fitted.

12(g) Every safety gear shall operate positively and mechanically, independently of any springs used in its construction.

12(h) Any levers or dogs operated by shafts shall be keyed to such shafts by B.S.S No 46 keys.

12(i) The design of safety gear shall provide for its application to both guides and to each side of the guides equally.

12(j) Any additional rope used solely for the purpose of operating the safety gear must be led over independent pulleys, running on independent shafts.

12(k) All bearings for drums and screw shafts in connection with safety gears must be on non-ferrous metals.

12(l) The car speed governor shall be set the cause the application of the safety gear at a speed of not more than 40 per cent., and not less than 15 per cent., above the contract speed provided that no governor shall be required to trip at a speed less than 175 feet per minute and with instantaneous type of safety gear shall trip at a speed not exceeding 250 feet per minute.

12(m) The counterweight safety gear, if provided, may be operated by the same governor and governor rope to operate the car safety gear provided it complies with the requirements for, and for the application of, counterweight safety gears. Provision shall be made to cause the application of the counterweight safety gear at a speed greater than the car safety gear, but at not more than 10 per cent in excess of that at which the car safety gear applies.

12(n) Broken rope safety gears of the instantaneous type may be used on counterweights within the following limits:

Table 1

Contract Speed (ft per min)	Total weight of counterweight in pounds
250	2,000
200	3,000
150	4,000
100	5,000

12(o) Three types of safety gear are recognised at present, viz:

i. Instantaneous type, Limited to speeds not exceeding 200 ft per minute (Type I)

ii. Gradual Wedge Clamp Type, with gradual increasing retarding force (Type G.W.C)

iii. Flexible Guide Clamp Type, with constant retarding force. (Type F.G.C)

12(p) No safety gear shall be permitted to stop an ascending car or counterweight. If an ascending car is to be stopped on account of overspeed, a safety gear shall be fitted to the counterweight for this purpose. The governor may, however, open the motor circuit and apply the brake in the event off overspeed in the ascending direction.

12(q) The governor must be placed where it cannot be struck by the car in case of overtravel and where there is sufficient space for the full movement of the governor parts.

12 (r) The motor control circuit and the brake control circuit shall be opened before or at the same time as the governor trips.

12(s) Governor ropes shall run clear of the governor jaws during the normal operation of the lift.

12(t) The proper tripping speed of the governor shall be stamped on the governor base or on a brass plate attached to the base.

12(u) When replacing governor ropes, they shall be of the same size, material and capacity as the ropes supplied originally and installed by the makers of the lift, except that where a rope of different characteristics is proposed, a test of the car and/or counterweight safety gear shall be made, to determine the fitness of the new ropes.

12(v) The are (area) of contact between the governor jaws shall be such that no serious cutting, tearing or deformation of the rope shall result for the operation of the safety gear.

12(w) It is recommended that governor gears have self-lubricating bearings which do not require frequent attention.

12(x) In the case of a safety gear actuated by means of a rope unwinding from a drum, such drum shall have at least three complete turns on the drum after the safety gear has been applied and the car stopped. The minimum diameter of such drum shall be 5” (Five inches). The device for holding the safety rope or rod in position during normal operation shall be fixed to the steel framework of the car and not to the car bodywork. The ends of the governor rope shall be held by a clevis or other similar means, which shall affect its purpose by friction. The clevis or other holding device shall be supported by or from the steel framework of the car and not fixed to the car bodywork.

12(y) No safety gear shall depend on the completion or maintenance of an electric circuit for its operation. All safety ears shall be applied mechanically.

12(z) The gripping surfaces of car or counterweight safety gears shall not be used to guide the car or counterweight, but shall run free of the guides during normal operation of the lift.

Note: A pawl or ratchet shall not be held a sufficient safety gear for lifts travelling in a vertical or substantially vertical direction.

This standard also went on to describe safety gear testing in section 13 as follows:

13(a) A contract load test shall be made by the lift maker of the safety gear or gears of each new lift, before such lift is put into service for normal and regular operation. This test shall be made to determine whether the safety gear will operate satisfactorily within the specified limits.

13(b) It is permissible to make the runaway test with platform, but without the car bodywork, provided the test weight is equivalent to the weight of the car bodywork plus contract load. This test should be carried out in the presence of the representative of the insurance office concerned or a qualified engineer, who shall issue the certificate of fitness.

13 (c) The maximum stopping distances of cars with safety gears of types G.W.C and F.G.C with contact load in the car, and the minimum stopping distances with the attendant only in the car, shall be as follows:

Table 2

Speed Ft/min	**Max Distance with Contract Load**		**Min Distance with Attendant Only in Car**	
	Type G.W.C	Type F.G.C	Type G.W.C	Type F.G.C
300	7’ 0”	1’ 9”	1’ 6”	0’ 6”
400	7’ 9”	2’ 6”	1’ 8”	0’ 9”
500	8’ 9”	3’ 6”	1’ 10”	1’ 0”
600	10’ 0”	4’ 9”	2’ 1”	1’ 3”
700	11’ 3”	6’ 6”	2’ 4”	1’ 6”
800	12’ 3”	8’ 3”	2’ 8”	1’ 9”

13(d) Stopping distance shall mean the actual slide as measured by the marks on the guides.

13(e) The runaway test shall be made by the lift maker with all electrical apparatus intact, except for the overspeed contact or cut out on the governor. For lifts operating directly from alternating current the governor shall be tripped by hand at the maximum speed obtainable.

13(f) Safety gears shall be examined thoroughly at least annually by the insurance company concerned or a qualified engineer, when the general condition of the working arts, sliding surfaces etc must be approved.

13 (g) At such examination the safety gear shall be applied with the car stationary and the car lowered to ensure the safety gear functions correctly.

APPENDIX 2:

BUILDING INDUSTRIES NATIONAL COUNCIL, CODE OF PRACTICE FOR ELECTRIC PASSENGER AND GOODS LIFTS AND ESCALATORS, 1943 – EMERGENCY SAFETY DEVICES

16(a)(1) Every passenger and goods lift suspended by ropes shall be provided with a safety gear attached to the car frame and preferably placed beneath the car platform.

16 (a)(2) The safety gear shall be capable of stopping and sustaining the car with contract load.

16(a)(3) The safety gear shall operate to stop and sustain the lift car in the event of failure of all suspension ropes, or in the event of the lift car exceeding a predetermined speed in the descending direction when an overspeed governor is fitted.

16(b) No car safety gear shall be permitted to stop an ascending lift car. If an ascending lift car is to be stopped on account of overspeed, a safety gear shall be fitted to the counterweight for this purpose.

Note: An overspeed governor may be used to cause the motor control and brake control circuits to be opened in the event of overspeed in the up direction.

16(c) The car safety gear of every lift having a travel exceeding 20 ft. should be operated by an overspeed governor.

16(d) The application of the safety gear shall not cause the car platform to become out of level in excess of 1 in 24, measured in any direction.

16(e) The motor control and brake control circuits shall be opened at the time the safety gear is applied.

16(f) When the car safety is applied, no decrease in the tension of any rope for applying the safety gear, or motion of the lift car in the down direction shall release the car safety gear. Note: It is permissible to release the safety gear by reversing the direction of the lift machine.

16(g)(1) It shall not be possible for vibration of the car frame to cause the safety gear to be applied.

16(g)(2) No safety gear shall depend upon the completion or maintenance of an electric circuit for its operation.

16(g)(3) Every safety gear shall be applied positively.

16(h)(1) safety gears of the instantaneous type may be used on car frames having a contract speed not exceeding 200 fpm.

16(h)(2) The tripping speed of governors used with instantaneous type safety gears shall not exceed 260 fpm.

16(i)(1) Any rope used for applying the safety gear shall be supported by its own pulley(s) and properly guarded. Such pulleys shall be mounted independently of any shaft carrying the lifting ropes.

16(i)(2) Such ropes shall not be less than ¼ in in diameter and shall be of steel or phosphor bronze.

16(j) The gripping surfaces of the safety gear shall be held clear of the guides during normal operation of the lift

16(k) Any levers or cams operated by shafts shall be fasted to such shafts by means of tapered pins in accordance with BS No 46 part 3, by means of sunk keys in accordance with BS No 46 Part 1, or by equivalent connection.

16(l) Safety gears shall be designed to grip each guide and to operate on both guides simultaneously.

16(m)(1) The rope attached to any safety gear actuating drum shall have not less than three turns of rope remaining in such drum after the safety jaws have gripped the guides and stopped the lift car.

16(m)(2) Bearings for actuating drums and screw shafts shall be of non-ferrous metal.

16(n) Any releasing carrier or other mechanism used for actuating the safety gear shall be carried by the car frame and not by the car enclosure.

16(o) When a release carrier is used to hold a governor rope clevis, such releasing carrier shall effect its purpose by means of a friction device.

The standard also introduced a table with lower rated speeds than the 1935 edition:

Table 3

Governor Tripping Speed FPM	Maximum Stopping Distance	Minimum Stopping Distance	
	Car with contract load (or for counterweight)	Car with attendant only	Car with contract load (or for counterweight)
WEDGE CLAMP SAFETY GEAR			
175	1’ 4”	0’ 10”	0’ 10”
200	1’ 6”	0’ 11”	1’ 0”
300	2’ 0”	1’ 0”	1’ 2”
400	2’ 10”	1’ 2”	1’ 7”
500	3’ 11”	1’ 5”	2’ 0”
600	5’ 2”	1’ 7”	2’ 5”
700	6’ 8”	1’ 11”	3’ 0”
800	8’ 6”	2’ 2”	3’ 7”
FLEXIBLE CLAMP SAFETY GEAR			
175	0’ 10”	0’ 5”	0’ 6”
200	0’ 11”	0’ 6”	0’ 7”
300	1’ 7”	0’ 7”	0’ 8”
400	2’ 5”	0’ 8”	1’ 1”
500	3’ 5”	0’ 11”	1’ 6”
600	4’ 10”	1’ 2”	1’ 11”
700	6’ 5”	1’ 6”	2’ 5”
800	8’ 2”	1’ 10”	3’ 2”

16(p) Any part of a safety gear subject to tension, torsion or bending shall be made of steel.

16(q) The distance travelled by the lift car (or counterweight) between the tripping of the governor jaws and the safety gear jaws beginning to apply pressure to the guides shall not exceed 2 ft 6 in.

16 (r) The maximum and minimum stopping distances of lift cars with wedge clamp or flexible clamp safety gears shall be in accordance with table III above

16(s) The stopping distance is the actual slide as indicated by the markings on the guides made by the safety gear.

16(t)(1) Safety gears of the instantaneous type may be used on counterweights having a contract speed not exceeding 250 fpm.

16(t)(2) Provision shall be made to cause the application of the counterweight safety gear, when operated by an overspeed governor, at a greater speed than, but not more than 10 per cent greater that that at which the car safety gear is applied.

16(u) A pawl and ratchet shall not be held to constitute a sufficient safety gear.

This standard also introduced a separate section on overspeed governors by way of section 17 which stated:

17(a) Overspeed governors shall be placed where there is sufficient room for their proper operation and where they cannot be struck by the lift car or counterweight in the event of overrun

17(b) The overspeed governor shall be adjusted to cause application of the lift car safety gear at a speed within the limits given in table IV

Table 4

Contract Speed	**Governor tripping speed (Percentage of contract speed)**	
Ft. per min.	Minimum	Maximum
Up to 500	115% (1)	140%
Over 500 to 700	115%	133%
Over 700	115%	125%

(1) No governor shall be required to trip at less than 175 ft. per min,

17(c) Each overspeed governor shall be marked with its tripping speed in terms of car speed in feet per minute.

17(d) The motor control and brake control circuits shall be opened before or at the time the governor trips.

17 (e) Governor ropes shall not be less than 5/16 in in diameter and shall be of steel, or phosphor bronze and of suitable construction.

17(f) Governor ropes shall run clear of the governor jaws during normal operation of the lift

17(g) The arc of contact made by the governor rope and the governor sheave shall, in conjunction with a rope tension device, provide sufficient tractive effort to cause proper operation of the governor.

17(h) Governor jaws and their mountings shall be so designed that any cutting, tearing or deformation of the rope resulting from their application shall no6t prevent proper operation of the safety gear.

APPENDIX 3

BS 2655-1:1970: SPECIFICATION FOR LIFTS, ESCALATORS, PASSENGER CONVEYORS AND PATERNOSTERS. GENERAL REQUIREMENTS FOR ELECTRIC, HYDRAULIC AND HAND-POWERED LIFTS

This standard was published in 1970 and ended up with 6 amendments to its original form.

There were four separate sections on safety gear requirements which could be found in sections 2, 3, 5 and 6.

Safety gears shall comply with the following general requirements:

- Every passenger and goods lift shall be provided with a safety gear attached to the car frame and placed beneath the car platform.
- Safety gear shall also be provided on the counterweight where there is an accessible space beneath the travel of the counterweight.
- It shall be possible to release car safety gears by raising the car, and counterweight safety gears by raising the counterweight.
- Each car safety gear shall be operated by means of either a governor or a safety rope. All sheaves or pulleys in contact with any part of this rope, which is normally in motion at the same time as the car, shall have diameters at least 30 times the diameter of the rope.
- A car safety gear shall not operate to stop an ascending lift car. If an ascending lift car is to be stopped on account of overspeed then a safety gear shall be fitted to the counterweight for this purpose. Where an overspeed governor is used, it shall cause the motor control and brake control circuits to be opened in the event of overspeed in the upward direction.
- The application of the safety gear shall not cause the car platform to slope at more than 1 in 25 to the horizontal
- The motor control and brake control circuits shall be opened by a switch on the car safety gear before or at the time the safety gear is applied.
- When the car safety gear is applied, no decrease in the tension of any rope used for applying the safety gear, or motion of the lift car in the downward direction shall release the car safety gear.
- It shall not be possible for vibration of the car frame to cause a safety gear to be applied.
- No safety gear shall depend for its operation upon completing or maintaining an electric circuit.
- The gripping surfaces of a safety gear shall be held clear of the guides during normal operation of the lift.
- Any levers or cams operated by shafts shall be fixed to such shafts by means of welding, sunk keys or by equivalent positive connection.
- Safety gears shall be designed to grip each guide and to operate on the guides simultaneously
- Any shaft, jaw, wedge or support which forms part of a safety gear and which is stressed during its operation shall be made of steel or other ductile material
- The drive to a car governor rope shall be effected from the car frame.
- Any connecting device between a governor rope and car frame (or counterweight) that is intended to be released when the safety gear is applied shall be retained in its normal position by a spring-loaded device
- A pawl and ratchet shall not be used as a safety gear

Intelligent Manufacturing Research Based on Sheet Metal Modularization of Elevator

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Keywords: intelligent manufacturing, order form, modularization, robot.

Abstract. It is an inevitable trend to using general elevator processing equipment, elevator sheet metal modular design and intelligent manufacturing to in order to improve the intelligent manufacturing level of elevator sheet metal production line, improve the work efficiency and reduce the operation workers intensity. The adoption of a general elevator processing equipment plus robot technology production line has a reasonable structure, improves the processing quality, high efficiency, and saves labor, and can meet the automation requirements of elevator sheet metal production.

1 DEMAND FOR AUTOMATIC PRODUCTION LINE OF ELEVATOR SHEET METAL PROCESSING

The elevator is an important means of transportation for vertical transport. The safe, comfortable, efficient and energy-saving operation of the elevator is a strict requirement for the elevator itself. China's elevator production, elevator ownership, and annual growth are the best and world number one in the world. The elevator manufacturing performance is further enhanced in the industrial concentration, the development trend is good, and the industrial structure has undergone significant changes. It shows the situation that the stronger performers getting stronger and the local companies progressing faster and faster. However, small business’s growth slows down and even declines. At the same time, industrial concentration has further increased.

With the continuous economic development, people's demand for elevator products is also rapidly developing. An obvious change is that now elevator-manufacturing companies have become increasingly diverse range of products, such as passenger elevator, medical elevator, freight elevator, escalator and moving walkways elevator. According to the technical points, it is divided into rooms, no rooms, small rooms etc. Multiple varieties, large batches, individualized customizations and errors in the construction of elevator shafts, mass customization have become the mainstream production mode for the development of modern elevator manufacturing. With the increasing demand for personalized products, the market is increasingly competitive. Fiercely, it is decided that elevator companies need to solve the production transformation according to the order. Enterprises urgently need practical product configuration design and manufacturing methods.

At present, elevator industrial strong countries such as Europe, USA, Japan and other countries have generally adopted the automatic production line for elevator sheet metal processing that integrates high-end machine tool equipment and logistics automation as the manufacturing mode for elevator cars and car doors; for example, the flexibility of Italy's Salvagnini Company. Sheet metal processing system is a model of flexible sheet metal processing system technology and is widely used in various industries. However, the production of most elevator cars and car doors in China is still dominated by labor-intensive manual production. The transfer of work pieces and the control of production cycles between processes are mainly realized manually. The gap between key equipment and production lines and the international advanced level is relatively large. The import dependence is strong. With the change of the age structure of the population in our country, there is a shortage of skilled workers. In short, the demographic dividend disappears and the cost advantage of China's manufacturing industry is losing. It can be foreseen that the intelligent manufacturing based on modular elevator sheet metal processing will be the first choice for the transformation and upgrading of China's elevator parts processing enterprises, and it will also be the main research topic of elevator sheet metal processing machine manufacturing enterprises in China.

2 CONSTRUCTION OF SELF-DEVELOPED INTELLIGENT ELEVATOR SHEET METAL PROCESSING PRODUCTION LINE

The production of elevator products has its own particularity, and it is necessary to have a predictive judgment on the market, because the elevator size, power, lifting height and so on required for each different building are different, and need to be based on customized order data parameters. Can quickly form the products needed by users through modularization. The basic method used in the production of elevators is to transform the manufacturing problems of customized products into or into partial batch production problems through product reorganization and process reorganization, that is, new and customized personalized products are provided to users, and actual products are mainly composed of different modules. Then elevator development needs to develop a common platform first, and then carry out a rapid customer-oriented deformation design on this platform. When the elevator load capacity, the structure of the hoistway, the effective area of the elevator room and the use situation and other requirements are different, the mechanical structure of the elevator. The structure and size of the medium-sized room vary greatly. Therefore, the key to the structural design of the elevator is the design of the room.

When the requirements for lift load, well structure, room effective area and use occasion are different, the elevator structure and size must be adjusted accordingly. Therefore, parametric design technology is applied in the elevator room structure system. By modifying the design parameters, the 3D CAD model can be modified, which can greatly improve the repeat design efficiency and can quickly establish the overall product appearance CAD model.

Setting specific module-related parameters at specific locations forms a digital product. The computer can quickly determine the product's specification and model number by identifying a specific number at a specific location then map the code to a specific encoder and decompose it into production-ready parts. The design is divided into various components, and these factors are divided according to the governing relationship to the corresponding manufacturing production lines, collaboration companies, outsourcing and other departments.

2.1 Elevator sheet metal processing intelligent manufacturing line

It is a system integration innovation project involving a wide range, through the upstream parameterized design, through the optimization of the computer to the manufacturing of the modular production process, involving the upper and lower material feeding system, punching, bending, welding and other online equipment and processes Tooling, as well as integrated innovations such as CNC systems, functional components, and automatic controls. Therefore, we must use the thoughts and methods of system engineering to start from the optimization and innovation of the overall solution for elevator sheet metal processing, explore new intelligent manufacturing process flow and methods, and study and summarize the overall solution for the formation of intelligent manufacturing processing units or automated production lines.

According to the requirements of the overall solution, we will focus on solving the current problems in China's general processing equipment design optimization, dynamic monitoring, the establishment of a knowledge expert database, adaptive intelligent control compensation, networked intelligent monitoring, and reliability enhancement; Research and application of processing technology, design and development of green efficient, supporting robots, comprehensively enhance the technical performance and level of enterprise equipment, and meet the development and application requirements of elevator sheet metal processing related fields in China.

2.2 Current Elevator Sheet Metal Processing Flow

The technological development status of intelligent manufacturing processes and equipment for elevator sheet metal processing, and the overall solutions for typical elevator sheet metal processing automatic production lines have the following technical features and development trends.

At present, the workers do most of the elevator sheet metal processing in China independently. Each machine tool such as blanking machine tool, stamping machine tool, bending machine tool and welding equipment consists of 2-3 artificial feeding, blanking, and indexing operations. The completion of the transfer of the upper and lower processes of the vehicle not only has low production efficiency, but also lags behind in production management. At the same time, the labor intensity in the production process has affected the increase in production.

2.3 The overall technical route and characteristics of elevator sheet metal flexible production line

Due to the volume and size of the elevator sheet metal, the developed elevator sheet metal intelligent manufacturing line (shown in Figure 1) consists of general equipment, open CNC robots, manipulators, and follow-up units and automatic feeding units for auxiliary automatic processes. Under the control of the control system, the robot grabs the plate to be processed from the platform truck and sends it to the conveyor belt.

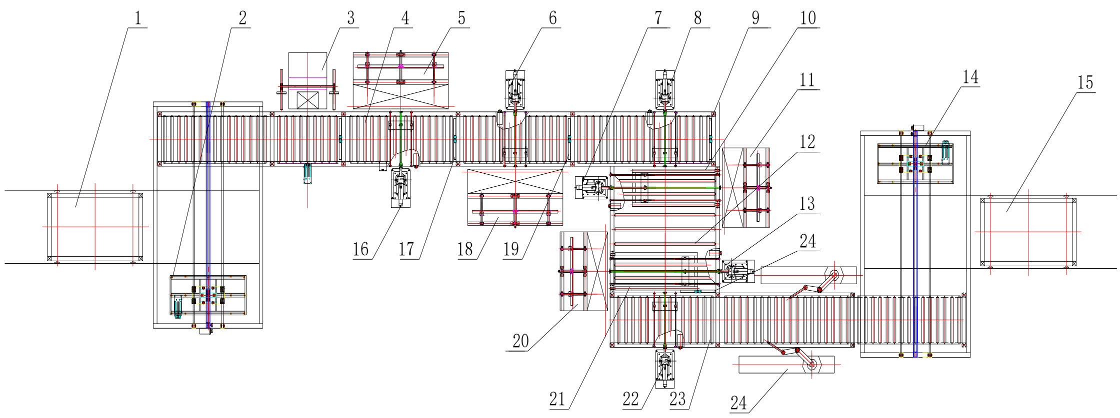

Figure 1 Schematic diagram of elevator sheet metal intelligent manufacturing line

(1) Pick up the robot with sucker 2 to the platform truck 1, lift it, move it to the set position above the conveyor belt 4, and lower it. The conveyor belt 4 transports the work piece to the adjustable stop plate of the punch press 3, and an open CNC robot with a clamp completes work piece positioning, feed, take-out, and plane movement of the work piece. The punching machine completes the punching work, and the robot returns the work piece. On the conveyor belt, the adjustable baffle falls and starts the conveyor belt operation.

(2) After the conveyor belt 4 sends the work piece to the SPS numerical control bending machine 5, the baffle plate 17 works, the robot hand 16 sends the metal plate to the specified position with the push plate, the SPS numerical control bending machine 5 works, and the follow-up unit works to ensure the work piece bending Reliable, complete one-sided bending work. After the baffle plate 17 falls, the conveyor belt 4 sends the work piece to the SPS numerical control bending machine 18, and the baffle 19 works. The robot 6 sends the metal plate to the predetermined position with the push plate, the SPS numerical control bending machine 18 works, and the follow-up unit works to ensure the work piece. The bending is performed reliably and the second bending work is completed. The baffle 19 falls, the conveyor belt 4 sends the work piece to the end, the robot 8 and the baffle 12 work, the robot 7 pushes the metal plate to the specified position with the push plate, the SPS CNC bending machine 11 works, and the follow-up unit works to ensure that the work piece is bent Reliable and complete the third side bending work. The baffle 12 falls. After

the conveyor belt 10 sends the work piece to the SPS numerical control bending machine 20, the baffle 21 works, the robot 13 sends the metal plate to the predetermined position with the push plate, the SPS numerical control bending machine 20 works, and the follow-up unit works to ensure the reliable bending of the work piece. Complete the fourth side bending work. The baffle 21 falls and the robot 22 pulls the work piece onto the conveyor belt 24,

(3) After the conveyor 24 has sent the work piece to the welding position, it is positioned and the welding robot 24 is operated. After the completion, the shutter is dropped and the start conveyor 24 is operated to send the work piece to the end.

(4) The conveyor 24 sends the work piece to the retrieving station, and then the work piece is sent to the platform truck 15 by the sucker-disc open type numerically-controlled robot 14 to complete all the work.

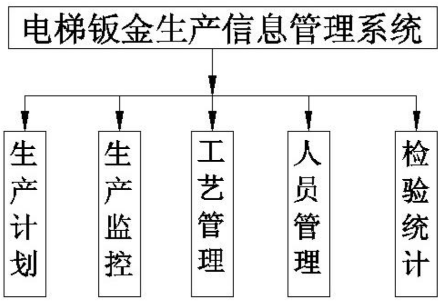

Figure 2 Functional block diagram of elevator sheet metal production information management system

2.4 Management Structure

The intelligent sheet metal production information management system in the elevator sheet metal production line system (function structure shown in Figure 2) completes elevator sheet metal order management, elevator sheet metal production statistics, elevator sheet metal production monitoring, elevator sheet metal processing Management and elevator sheet metal operator management work.

2.5 Design of the equipment layer

The equipment layer includes purchased robots, mechanical arm, and elevator sheet metal processing equipment. The three sets of equipment complete the automatic production line of elevator sheet metal under the control of the decision-making control system, the robot control system and the robot control system, feeding, feeding, stamping, bending and welding.

The horizontal transfer device is a railcar with its own positioning and lifting device and is positioned by the sensor. The material trolley is equipped with three sensors, two for near deceleration and one for stop. The horizontal transfer trolley is driven by two AC servo motors. One drive rolls along the inside of the I-beam and the other drives the up and down movement of the material rack. The trolley has its own battery and can be charged freely while waiting. During the movement, there is no need to consider the problem of entanglement of the accompanying cable. The track of the trolley can be freely selected along any track of the material warehouse, and the material can be taken and fed conveniently. In combination with the PC intelligent control technology, the path can be optimized and the trolley can be moved laterally. The highest efficiency. In the design of the up and down elevator platform, the motor is used to drive the screw rod, and the ball screw driver ball pair moves on a screw rod with two opposite thread directions. When the screw rod rotates, the two ball bearings are close to or far away at the same time and pass through the four-bar mechanism. The pallet lifting the trolley car overall lifts the pallet. Since the screw can drive the ball, and the drive in turn can achieve self-locking, its principle is equivalent to the jack. Therefore, the device is safe and reliable when it is raised and lowered, and it will not be subject to free fall. The railcar adopts four-point support and asynchronous two-point drive, which can guarantee the effective driving force of the rail-to-rail transfer to prevent slipping, and also can guarantee the stability of the railcar and prevent the side-turn or lateral tilting moment. The layout of the entire rail car is shown in Figure 3.

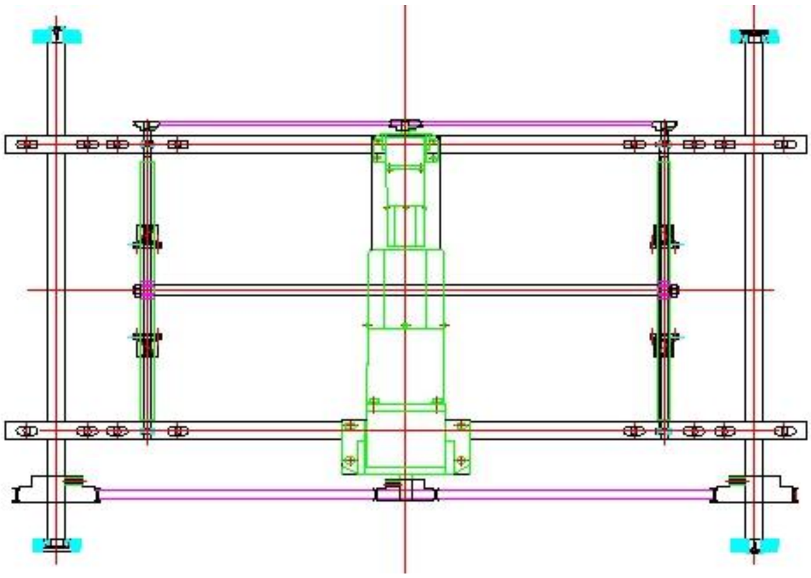

Figure 3 Horizontal transfer trolley car

2.6 Intelligent manufacturing technology elevator sheet metal production line workflow

Intelligent manufacturing technology elevator sheet metal production line workflow, the use of photoelectric sensors, photoelectric switches, in and out of material detection, CCD online detection technology to achieve the entire system of closed control, automatic detection technology can greatly improve the reliability of the equipment and reduce the incidence of accidents probability. After starting the decision system, the robots and robots are started separately to complete the preparation of the robots and robots. By starting the operation of the production line through a touch screen or an open numerical control system, the decision control system waits for the robot, the manipulator to return to zero, and the shaping unit to return to zero. According to the working status, the decision unit notifies the robot to pick up the work piece, send it to the conveyor production line, the production line pneumatically, and the work piece enters the detection part; at the same time, the decision-making control system detects the position of the work piece through the infrared sensor and controls the robot to clamp the work piece. The robot completes the workflow. In the specified position, the robot returns to the waiting position, waits for decision-making control system instructions, punches, bends, and welds the equipment. The robot processes the finished work piece and the mechanical arm places the work piece.

Elevator sheet metal intelligent manufacturing automated production line control is the core of the entire system, complete the control of robots, mechanical arm and equipment. The open CNC system can realize information interaction with touch screens and PCs and improve the openness of the system. The open CNC system control system uses a PC+ control card structure. The open CNC system control system interacts with the decision control system information through the serial port, and controls the mechanical arm actuators of each axis through the control card to realize the robot motion control. The robot control system uses CC-Link and decision control systems for information exchange. According to the instructions of the decision control system, the mechanical arm actuator is controlled to complete the work of feeding the work piece.

3 CONCLUSIONS

To liberate people from heavy, repetitive and cumbersome work and to engage in relatively easy service work is an embodiment of scientific and technological advancement and a development trend of human society.

Automation technology is a solution that partially replaces labor, reduces labor intensity, and improves processing efficiency. Automated machining saves labor costs while improving machining efficiency and machining accuracy, and has good economic benefits. Therefore, for the upgrade needs of China's elevator processing industry, independent research and development of elevator sheet metal modular processing intelligent manufacturing production line, the previous paragraph uses a parametric design, manufacturing uses a modular technology, due to the use of robots to assist the work, not only improved Efficacy, reduced labor, and improved product quality. Without adding bending, stamping machine equipment, etc., it can reduce 2/3 of the staff to complete the same job. By adjusting processing equipment, not only can people be reduced, they can meet the needs of the order market, and they can also significantly increase production. It is the inevitable choice for upgrading and upgrading of elevator manufacturing enterprises in China.

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BIOGRAPHICAL DETAILS

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Lift Planning for High-Rise Buildings

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Keywords: high-rise, lift, elevator, traffic analysis, calculation, simulation.

Abstract. This paper provides an overview of the different ways of providing lift service to high rise buildings. In general, taller buildings need a greater proportion of core space to accommodate the lifts. To reduce core space, often the first option considered is to divide the lifts into two or more zones. Double deck lifts, with two cabs serving adjacent floors at the same time, provide greater handing capacity per shaft. Solutions with two independently roped cars per shaft achieve a similar handling capacity boost, but with added flexibility. For super high-rise buildings, shuttle lifts expressing people to sky lobbies offer further savings in core space. Planned rope-less lifts solutions promise significantly more handling capacity per shaft, freeing mega high-rise buildings of the limits imposed by roped lifts. Often the solution chosen will adopt more than one strategy and technology. The pros and cons of different approaches are discussed, together with a core space analysis of alternative solutions for example buildings.

1 INTRODUCTION

In low rise buildings, there is normally a single lift group which serves every floor. In high rise buildings lifts are often arranged in zones. CIBSE Guide D [1] suggests that it is general practice to serve a maximum of 15–16 floors with a lift or a group of lifts. Al-Sharif et al provide additional rules of thumb for high rise lift planning [2], also including double deck and shuttle lifts drawn from a range of sources. Other options including two cars per shaft [3] are available, and rope-less lifts are planned [4].

This paper discusses the different approaches, together with a core space analysis of alternative solutions for example buildings. There will normally be separate goods lifts and firefighting lifts [1] which are not addressed in this paper.

For a first step in traffic analysis design, let us consider an office building with 60 occupied floors and 50 people per floor requiring an uppeak handling capacity of 12% and maximum interval of 30s. Apply the general analysis round trip time calculation for single [5] and double deck [6] lifts; this assumes conventional control. Design parameters are based on CIBSE Guide D. Core calculations are based on the core area taken by the lift shaft, and do not include lift lobbies.

The results are indicative only, as designs which are more core efficient will allow more people to be accommodated in a building with the same total floor area. After this initial planning stage, most designers would then use simulation to assess the application of destination control and to consider lunch time traffic.

2 SINGLE DECK LIFTS

According to the CIBSE Guide D, a building with sixty occupied floors would typically be served by 4 rises, see Figure 1. If a lift serves too many floors, the number of stops and transit time for passengers at higher floors becomes intolerable. Another consideration is core space. If every lift serves every floor, the taller the building, the higher the proportion of the building is taken by lifts.

	rise 1	rise 2	rise 3	rise 4
60				X
59				X
58				X
57				X
56				X
55				X
54				X
53				X
52				X
51				X
50				X
49				X
48				X
47				X
46				X
45			X	I
44			X	I
43			X	I
42			X	I
41			X	I
40			X	I
39			X	I
38			X	I
37			X	I
36			X	I
35			X	I
34			X	I
33			X	I
32			X	I
31			X	I
30		X	I	I
29		X	I	I
28		X	I	I
27		X	I	I
26		X	I	I
25		X	I	I
24		X	I	I
23		X	I	I
22		X	I	I
21		X	I	I
20		X	I	I
19		X	I	I
18		X	I	I
17		X	I	I
16		X	I	I
15	X	I	I	I
14	X	I	I	I
13	X	I	I	I
12	X	I	I	I
11	X	I	I	I
10	X	I	I	I
9	X	I	I	I
8	X	I	I	I
7	X	I	I	I
6	X	I	I	I
5	X	I	I	I
4	X	I	I	I
3	X	I	I	I
2	X	I	I	I
1	X	I	I	I
ground	X	X	X	X

X represents served floor

I represents express zone

Figure 1 Example four rise arrangement for sixty floor office building

To illustrate the impact of the number of rises on core space, Figure 2 shows the core space by designs meeting the criteria for between 1 and 4 rises.

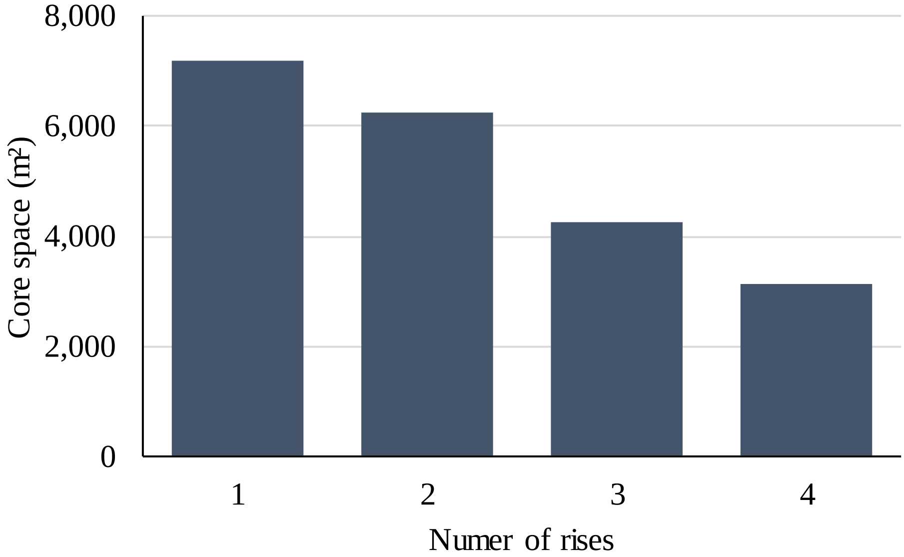

Figure 2 Core space

As is demonstrated in Figure 2, increasing the number of rises reduces core space. The reduction in core space is a consequence of less lifts travelling to higher floors, e.g. where there are four rises there may be are four times as many shafts at the ground floor as there are at the top floor. The reduction in possible stops also reduces the total number of probable stops; with less probable stops, the round-trip time of a lift group is reduced, which increases the handling capacity. This trend in reducing core space with increasing numbers of rises will continue until the lift group is so small that the handling capacity criteria is met with *N* lifts, but >*N* lifts are required to meet the interval criteria.

One of disadvantages in having multiple rises is that interfloor traffic between the zones is restricted. Often there is a transfer floor to enable travel between the zones without returning to the ground floor; for example, to transfer between rise 1 and rise 2 there could be a transfer floor at either level 15 or 16, see Figure 3.

	rise 1	**rise 2**	**rise 3**	**rise 4**
18		X	I	I
17		X	I	I
16	T	X	I	I
15	X	T	I	I
14	X	I	I	I
13	X	I	I	I
ground	X	X	X	X

T represents possible transfer floor location

Figure 3 Extract of Figure 1 showing possible transfer floor locations between rise 1 and rise 2

If the transfer floor is at level 15, people who work in rise 1 on level 15 can use rise 2 when travelling from ground. Indeed, this will be their preferred rise as they will express to their floor without intermediate stops. This is not as intended in the traffic design and would lead to higher passenger demand on rise 2. To avoid this, the car call buttons for level 15 may be disabled during the morning uppeak. This is a practical solution but provides an inconsistency in user interface. Having the transfer floor on level 16 is better for the traffic design, but additional core space is required as rise 1 now extends an extra floor.

Where the number for floors does not divide equally into the number of rises, it is normal to have less floors in the higher rises. Other factors may also influence the choice of floors assigned to rises, e.g. different floor populations, magnet floors such as restaurants, and architectural considerations such as the stepping back of a building at higher floors. Higher rises normally have faster lifts such that the round-trip time is similar for all rises.

3 DOUBLE DECK LIFTS

Double decker lifts have two separate cabs built into a single unit so that the upper and lower cabs serve adjacent floors simultaneously. During peak periods maximum operating efficiency is achieved by restricting the lower cabs to serving odd numbered floors, and the upper cabs to serving even numbered floors. Escalators are provided to link the lower and upper ground floors.

All floors should be of equal height as the two cabins of the double deck lifts must server adjacent floors simultaneously, at all floors. There are solutions for adjusting the distance between cabins, but these introduce additional complexity and operational delays. Double deck lifts suffer from non-coincident stops, see Figure 4.

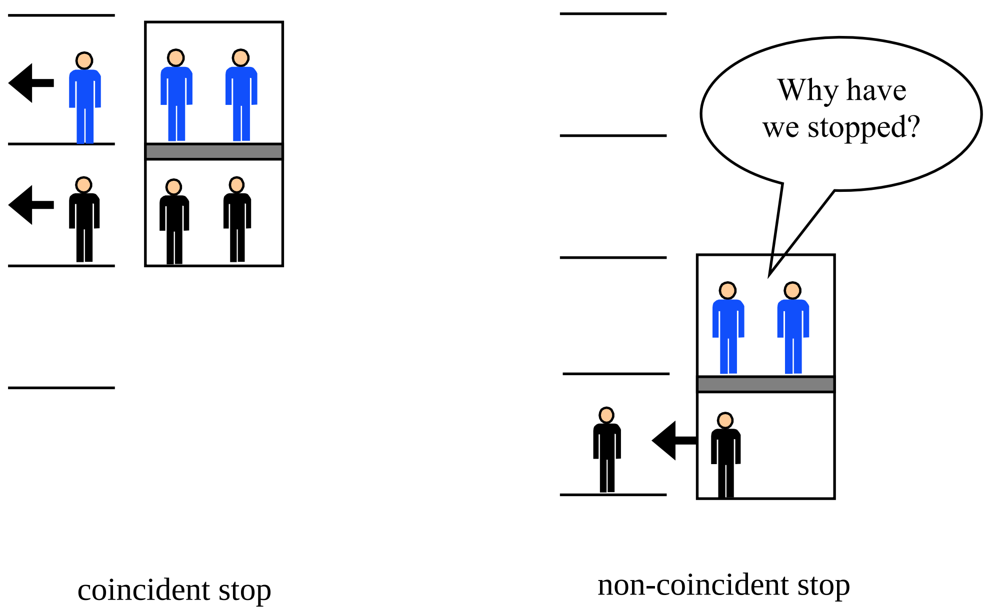

Figure 4 Illustration of coincident and non-coincident stops

The figure of merit for use of double-deck lifts is defined as being the percentage of stops that are coincident to both upper and lower cabs [7]. A high figure of merit is preferable as it can be frustrating for passengers when the lift stops repeatedly, and no-one leaves or enters their lift cab. There will normally be an indicator in the cabins to communicate that the other car is loading or unloading. For the example building consider solutions for one, two and three rises. Figure 5 shows an arrangement with three rises.

	rise 1	rise 2	rise 3
60			X
59			X
58			X
57			X
56			X
55			X
54			X
53			X
52			X
51			X
50			X
49			X
48			X
47			X
46			X
45			X
44			X
43			X
42			X
41			X
40		X	I
39		X	I
38		X	I
37		X	I
36		X	I
35		X	I
34		X	I
33		X	I
32		X	I
31		X	I
30		X	I
29		X	I
28		X	I
27		X	I
26		X	I
25		X	I
24		X	I
23		X	I
22		X	I
21		X	I
20	X	I	I
19	X	I	I
18	X	I	I
17	X	I	I
16	X	I	I
15	X	I	I
14	X	I	I
13	X	I	I
12	X	I	I
11	X	I	I
10	X	I	I
9	X	I	I
8	X	I	I
7	X	I	I
6	X	I	I
5	X	I	I
4	X	I	I
3	X	I	I
2	X	I	I
1	X	I	I
upper ground	X	X	X
lower ground	X	X	X

Figure 5 Example double deck three rise arrangement

With double deck lifts, the possible number of stops is reduced by half; in Figure 5 there are 20 floors in each rise above the lower/upper ground floor. However, with odd to odd and even to even traffic enforced, the lift will not stop more than 10 times during its travel up the building. With two cabins per lift, and the reduction in number of stops, core efficiency is improved. Figure 6 shows the core space for a double deck installation with a single rise, two rises and three rises. In this instance going from two the three rises is offering a small saving in core space.

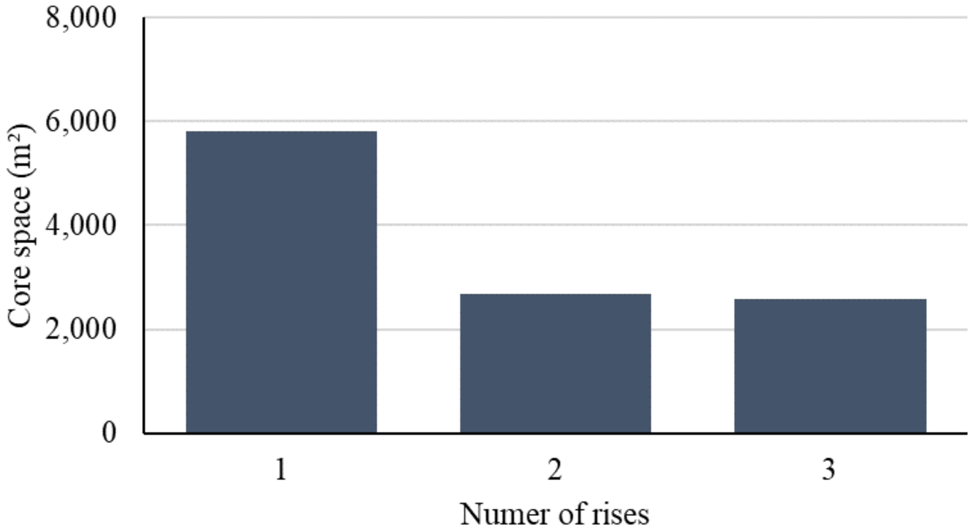

Figure 6 Core space

Transfer between rises can be included, see the discussion in section 2 on transfer floors. Journeys between odd and even floors can be allowed for, especially with destination control. However, this should be avoided at peak times where possible as it interferes with the efficiency of the system.

Although the example analysis presented in Figure 6 is for conventional control, the advantages of destination control with double deck lifts are significant. Aside from the boosting of uppeak handling capacity, an intelligent control system will reduce non-coincident stops.

4 TWO CARS PER SHAFT

Solutions exist for two independent cabins per shaft [3]. For maximum efficiency, two entrances are required with the lower lift serving a lower zone, and an upper lift serving an upper zone. In contrast to double deck lifts, the floor to floor heights do not be to be equal. However, the lower ground floor needs to be sufficiently high for the two cars to serve the lower and upper ground floor at the same time.

In planning the installation, the objective is to have a similar round-trip time for both the lower and upper lifts. This arrangement does not suffer from non-coincident stops, but instead there is a possibility that the lower car must wait for the upper car to move, and vice versa, see Figure 8.

An example with two rises is shown in Figure 7.

	rise 1	rise 2
60		U
59		U
58		U
57		U
56		U
55		U
54		U
53		U
52		U
51		U
50		U
49		U
48		U
47		U
46		U
45		L
44		L
43		L
42		L
41		L
40		L
39		L
38		L
37		L
36		L
35		L
34		L
33		L
32		L
31		L
30	U	I
29	U	I
28	U	I
27	U	I
26	U	I
25	U	I
24	U	I
23	U	I
22	U	I
21	U	I
20	U	I
19	U	I
18	U	I
17	U	I
16	U	I
15	L	I
14	L	I
13	L	I
12	L	I
11	L	I
10	L	I
9	L	I
8	L	I
7	L	I
6	L	I
5	L	I
4	L	I
3	L	I
2	L	I
1	L	I
upper ground	U	U
lower ground	L	L

L represents served by lower car

L represents served by upper car

I represents express zone

Figure 7 Example two lifts per shaft two rise arrangement

At the initial planning stage, this technology can be considered equivalent to double deck lifts for core space. Although operationally different, it can achieve the same efficiencies as there are still two cabins per shaft; the difference is that the cabins are unbound by a connection.

Transfer between rises can be allowed for, as per the discussion in section 2. Transfer between the lower and upper zones of a rise is also possible. However, crossing the zone boundaries reduces the efficiency causing more instances of delayed cars, see Figure 8.

This technology is currently only available with destination control; a conventional control solution would be possible, but without knowing people's destinations, its efficiency would be reduced.

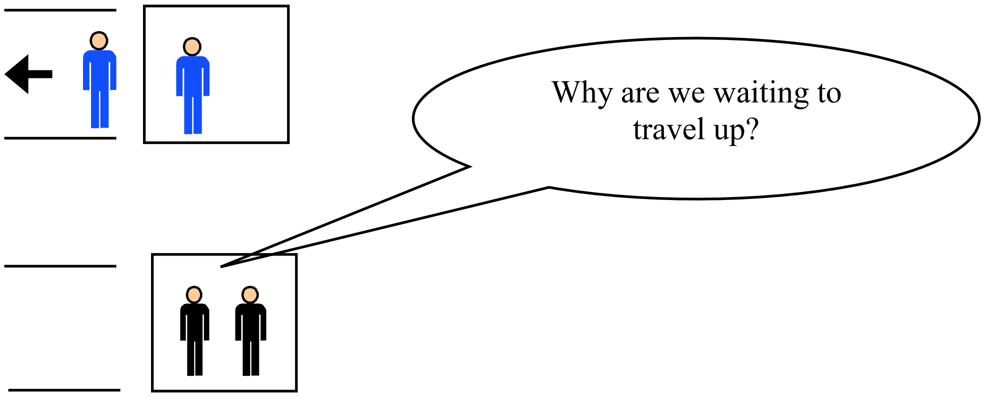

Figure 8 Waiting for second car to move

5 SHUTTLE LIFTS AND SKY LOBBIES

Shuttle lifts express people to one or more sky lobbies part way up the building. Typically, the core saving achieved by omitting express zones for rises 2, 3, etc. is greater than the additional core required for the shuttle lifts. The disadvantage of shuttle lifts is that people need to take two lifts to reach their office floor.

Shuttle lifts may be single, double deck, or two cars per shaft; for a double deck shuttle lift there would be a double deck sky lobby. For a two car per shaft shuttle, the lower deck could express to the lowest populated floor of rise two, and the upper deck to the lowest populated floor of rise 3. Local lifts may also be single, double deck, or two cars per shaft.

The intuitive way of applying shuttle lifts is to move people to the lowest floor served by one of the local lift groups. However, this is not necessarily the most core efficient approach. The top-down solution [8] shuttles people to the bottom of one set of local lifts, and the top of another set of local lifts. Although this is very core efficient, it is not popular as some people are travelling up, and then down.

Figure 9 shows a sky lobby arrangement with double deck shuttle lifts serving local double deck lifts; for upper floors passengers take the shuttle, and then transfer to the double deck rise 2. Figure 10 compares the core space of this arrangement with two rises of double deck lifts.

	rise 1	shuttle	rise 2
60			X
59			X
58			X
57			X
56			X
55			X
54			X
53			X
52			X
51			X
50			X
49			X
48			X
47			X
46			X
45			X
44			X
43			X
42			X
41			X
40			X
39			X
38			X
37			X
36			X
35			X
34			X
33			X
32		X	X
31		X	X
30	X	I	
29	X	I	
28	X	I	
27	X	I	
26	X	I	
25	X	I	
24	X	I	
23	X	I	
22	X	I	
21	X	I	
20	X	I	
19	X	I	
18	X	I	
17	X	I	
16	X	I	
15	X	I	
14	X	I	
13	X	I	
12	X	I	
11	X	I	
10	X	I	
9	X	I	
8	X	I	
7	X	I	
6	X	I	
5	X	I	
4	X	I	
3	X	I	
2	X	I	
1	X	I	
upper ground	X	X	
lower ground	X	X	

Figure 9 Two rises of double deck lifts with a shuttle lift to the upper rise

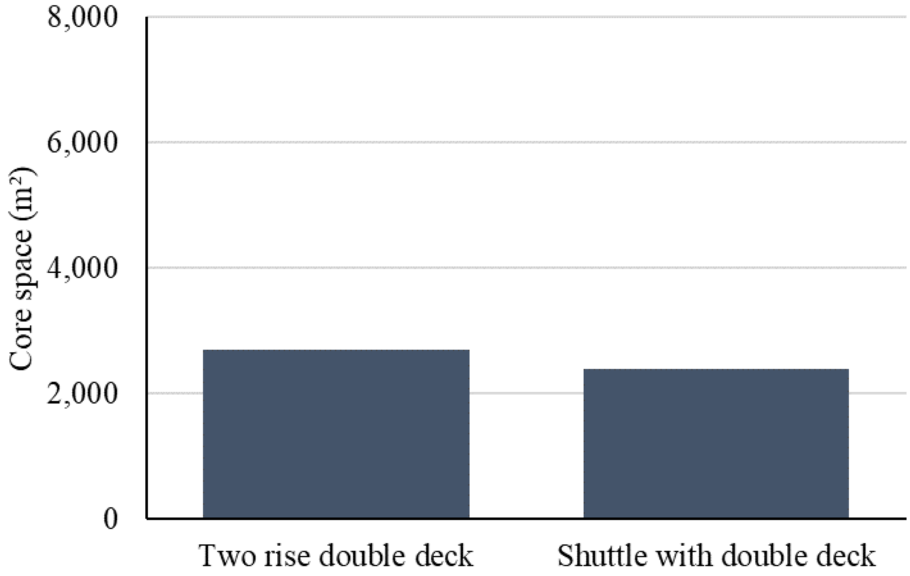

Figure 10 Core space

6 ROPELESS LIFTS

The prospect of ropeless lifts provide a unique opportunity for high rise and mega high-rise buildings. There are many possible variants of this technology and ways to apply it, but for illustrative purposes consider the single deck solution shown in Figure 1 where the express zones are removed, but instead there are ropeless lifts which serve levels G, 16, 31 and 46 in a loop with two up and two down shafts. The uppeak handling capacity of these loops is limited by the rate at which cars can be loaded and dispatched. This yields enormous handling capacities compared to roped lifts.

Figure 11 demonstrates the reduction in core space achieved. Relative core saving will increase with building height. Building height is limited by the increasing percentage of the building taken by lifts as buildings get taller. With ropeless lifts, this limit is raised.

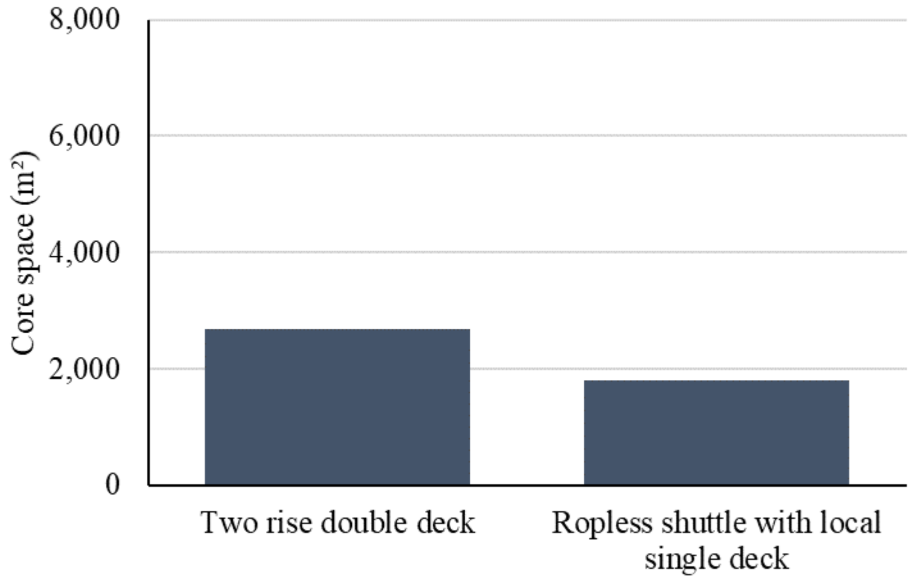

Figure 11 Core space

7 CONCLUSIONS

This paper provides an overview of different ways to provide lift service to high rise buildings. In general, taller buildings need a greater proportion of core space to accommodate the lifts. This core space can be reduced by applying different strategies and technologies which has been illustrated with an example building. Double deck lifts, lifts with two independently roped cars per shaft, shuttle lifts and ropeless lifts have been considered. The different approaches have pros and cons; the final strategy selected will account for cost and of ease of use.

The analysis in the paper has been based on round trip time calculations which provide a good first step in a traffic design. Detailed design would normally consider destination control and involve simulating prospective solutions.

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BIOGRAPHICAL DETAILS

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Longitudinal Coupled Vibration of Parallel Hoisting System with Tension Balance Devices

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Keywords: longitudinal coupled vibration, parallel hoisting system, tension balance device, dynamic behaviors.

Abstract. In long travel lifting systems, multiple parallel ropes are connected to conveyance through a set of termination devices to achieve the lifting process. Due to external excitations coming from different wear levels and manufacturing error of rope grooves on the friction pulley, or rope slipping, each rope length is different and accordingly results in different rope tension. Therefore, a tension balance device is applied as the termination device to reduce the tension differences. In order to describe the dynamic behavior, rope sockets of the tension balance device are simplified into a lumped mass at the end of each rope, and a longitudinal vibration model of a parallel hoisting system with tension balance device is built. Both the normal and unsteady working conditions of the tension balance device are considered. To deal with the complex constraints between ropes and hoisting conveyance, Lagrange multipliers are used, and the equations are numerically solved. The characteristics of longitudinal vibration frequency are depicted and the dynamic behaviors are revealed, which are essential for optimizing the parallel hoisting system with the tension balance device. That will minimize the effects of external excitations and improve the security of the system.

1 INTRODUCTION

Hoisting system is an efficient vertical transportation to lift human and payload to different levels [1, 2]. Hoisting ropes are axially translating media with time-varying length [3, 4]. Hoisting ropes, due to their flexibility, loading conditions, and internal damping characteristics, mainly determine the resonances of longitudinal and lateral vibration of the system [5, 6]. This may lead to large vibration amplitude and mechanical breakdowns of the equipment especially in high-rise systems. Therefore, parallel hoisting system is now used to improve system safety [7]. Kumaniecka and Nizioł[8] analyzed the longitudinal-transverse vibration with a varying-length rope and focused on the parametric resonances of the rope. Huang [9] studied the dynamic stability of a moving string in three dimensions. The partial differential equations of motion are derived by using Hamilton's principle and later simplified as ordinary differential equations by Galerkin’s method. Huang [10] also investigated qualitative aspects of parametric excitation due to a non-constant traveling velocity of a viscoelastic string. Sandilo and Van Horssen studied autoresonance phenomena of one rope in a space–time-varying mechanical system [11].

However, the longitudinal vibration of a hoisting rope with tension balance device is less studied in parallel hoisting system. In this paper, the flexible hoisting system is regarded as a flexible multi-body structure. The governing equations of the flexible hoisting system are developed by using the Lagrange equations technique. The characteristics of longitudinal vibration frequency are depicted and the dynamic behaviors are revealed, which are essential for optimizing parallel hoisting system with the tension balance device. This work will minimize the impact of external excitations and improve the security of the system.

2 DESCRIPTION OF THE MODEL OF A LIFTING SYSTEM

Long travel lifting system mainly includes the following parts by reference to Fig. 1: Sheaves, multiple hoisting ropes, tension balance devices, hoisting conveyance and rigid guide. The end of the rope is fixed on rope sockets (3-a) of the tension balance device. Tension balance devices used to adjust the tension distance between the hydraulic cylinder (3-b) and the base (3-c) according to the tension [12].Multiple parallel ropes are connected to conveyance through a set of termination devices to achieve the lifting process. Eccentric pulleys and sheaves and systematic resonance in the electronic control system are typical causes of longitudinal vibrations of a lifting system. The model is based on the following assumptions: (a) Only longitudinal vibration is considered. (b) Neglecting the influence of the compensation rope.

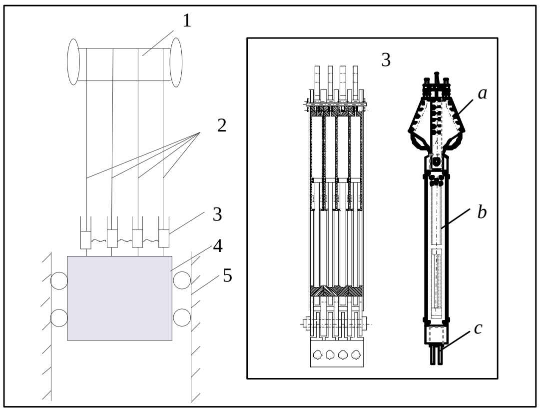

Figure1: Lifting system

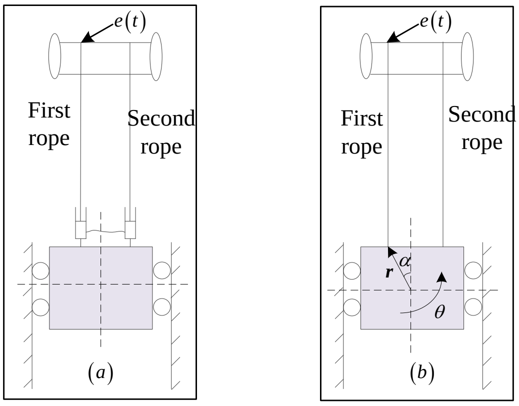

Figure 2: Two hoisting ropes with tension balance (a) and without tension balance (b)

3 VIBRATION MODEL

In order to derive the differential equations of motion, the dynamic model considers the case of n hoisting ropes. The kinetic energy is:

$$T=\frac{1}{2}\sum_{i=1}^{n}\rho\int_0^{l_i(t)}\left(u_{i,t}+vu_{i,x}+v_i\right)^2dx+\frac{1}{2}m_i\left(u_{i,t}\left(l_i,t\right)+vu_{i,x}\left(l_i,t\right)+v_i\right)^2+\frac{1}{2}m\dot{u}_c^{\,2}+\frac{1}{2}J\dot{\theta}^2 \tag{1}$$

where ρ, m_i, m and J are rope mass per unit length, rope mass of sockets on tension balance device, hoisting conveyance mass and inertia moment, respectively. u_i is longitudinal vibration displacement. u_c is the displacement of conveyance. Let $\rho_s = \rho + m_i\,\delta\,(x - l_i)$ for which the kinetic energy can be written as:

$$T=\frac{1}{2}\sum_{i=1}^{n}\rho_s\int_0^{l_i(t)}\left(u_{i,t}+vu_{i,x}+v_i\right)^2dx+\frac{1}{2}m\dot{u}_c^{\,2}+\frac{1}{2}J\dot{\theta}^2 \tag{2}$$

The potential energy and dissipation energy is a function of the vibration displacement:

$$E_e=\sum_{i=1}^{n}\left(\int_0^{l_i(t)}T_{i,s}u_{i,x}dx+\int_0^{l_i(t)}EA\frac{1}{2}u_{i,x}^{\,2}dx\right)+\frac{1}{2}\sum_{i=1}^{4}G_i\theta^2 \tag{3}$$

$$E_g=-\rho_s\sum_{i=1}^{n}\int_0^{l_i(t)}u_i g\mathrm{d}x-mgu_c \tag{4}$$

$$D=\sum_{i=1}^{n}\left(\frac{1}{2}\mu_i\int_0^{l_i(t)}u_{i,t}^{\,2}dx\right) \tag{5}$$

where G_i is guide stiffness, $T_{i,,s}$ is the rope tension. g is the constant acceleration of gravity. And the Lagrangian will be

$$L=T-\left(E_e+E_g\right) \tag{6}$$

The solutions of longitudinal displacement are assumed in the form:

$$u_i = \sum_{j=1}^{R} \psi_j q_{i,j} \tag{7}$$

where $q_{i,j}$ is the generalized coordinate, $\psi_j = \sin\left(i - 1/2\right)\pi\xi$ is the mode shape functions. This Lagrangian leads to the following set of simultaneous differential equations of motion

$$\frac{\mathrm{d}}{\mathrm{d}t}\frac{\partial L}{\partial \dot{q}_i} - \frac{\partial L}{\partial q_i} + \frac{\partial D}{\partial \dot{q}_i} = F_i + \sum_{k=1}^{N} \lambda_k \frac{\partial g_k}{\partial q_i} \tag{8}$$

where, $\boldsymbol{q} = \left[\boldsymbol{q}_1^{\mathrm{T}} \quad \cdots \quad \boldsymbol{q}_n^{\mathrm{T}} \quad u_c \quad \theta\right]^{\mathrm{T}}$, $\boldsymbol{g} \in R^{N\times 1}$ is the boundary conditions vector and its definition are denoted in Section 4, λ_k is the Lagrange multipliers. This is a linear system of homogeneous equations, which can be considered as the $n+1$ components of the block matrix equation

$$[\boldsymbol{M}]\ddot{\boldsymbol{q}} + [\boldsymbol{C}]\dot{\boldsymbol{q}} + [\boldsymbol{K}]\boldsymbol{q} = [\boldsymbol{F}] + \boldsymbol{\lambda}^{\mathrm{T}}\boldsymbol{Q} \tag{9}$$

where the matrices are defined by:

$$\begin{aligned}
\boldsymbol{M}^i &= \rho l_i \boldsymbol{M}_1 + m_i \boldsymbol{M}_0 \\
\boldsymbol{C}^i &= \rho v_i \left(\boldsymbol{C}_1 + \boldsymbol{C}_2 - \boldsymbol{C}_2^{\mathrm{T}}\right) + \mu_i l_i \boldsymbol{C}_1 \\
\boldsymbol{K}^i &= \rho a_i \boldsymbol{K}_1 - \frac{\rho v^2}{l_i}\boldsymbol{K}_2 + \frac{EA}{l_i}\boldsymbol{K}_3 - \mu_i v_i \boldsymbol{K}_4 \\
\boldsymbol{F}^i &= \left(\rho l\left(g - a_i\right) - \rho v_i^2\right)\boldsymbol{F}_1 - m_i g \boldsymbol{F}_2 - \left(\rho g l_i - \rho v_i^2\right)\boldsymbol{F}_3 - m_i a_i \boldsymbol{F}_4
\end{aligned}$$

$$\boldsymbol{M}^{n+1} = \begin{bmatrix} m & 0 \\ 0 & J \end{bmatrix}, \boldsymbol{C}^{n+1} = \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix}, \boldsymbol{K}^{n+1} = \begin{bmatrix} 0 & 0 \\ 0 & \sum_{i=1}^{4} G_i \end{bmatrix}, \boldsymbol{F}^{n+1} = \begin{bmatrix} mg \\ 0 \end{bmatrix}$$

$\boldsymbol{M}_i, \boldsymbol{C}_i, \boldsymbol{K}_i, \boldsymbol{F}_i$ are mode matrixes with boundary conditions and $\boldsymbol{Q}$ is Jacobi matrix related to the constraint equations g_k. After elimination of constraints, we can obtain the complete solution of the problem.

4 CASE STUDY

In this section, simulation results of the two ropes in this paper are discussed.

Case (a):

When guide stiffness G_i is soft as Fig. 2(b). The constraint relationship between the hoisting rope and conveyance without a tension balance device should be fulfilled as followed:

$$g_1 = u_c + r\theta \sin\alpha - u_1\left(1, t\right), \mathrm{g}_2 = u_c - r\theta \sin\alpha - u_2\left(1, t\right) \tag{10}$$

in which, r is the distance to the connecting point as Fig. 2(b), and a rotation around the axis of angle θ (in the positive trigonometric sense).

Case (b):

When tension balance devices are applied in the system as shown in Fig. 2(a), constraint condition is:

$$g_1 = \frac{1}{2}\left(u_1\left(1, t\right) + u_2\left(1, t\right)\right) - u_c \tag{11}$$

5 NUMERICAL CALCULATION

In this simulation, two hoisting ropes one with a tension balance device the other without are selected for comparison. The fundamental parameters of the system are listed in Table 1.

Table 1 Fundamental parameters of the system

Parameter	Value	Unit	Parameter	Value	Unit
m	35000	kg	ρ	8.5	kg/m
m_i	400	kg	G_i	2×107	Nm
EA	1.138×105	N	J	2000	kg·m^2
μ_i	0.4125	kg/s	l_0	2100	m

By numerical calculations, the results demonstrate the nonlinear dynamic interaction between the components of the lifting system, during the hoisting process and when the system is non-stationary. The roots determine the natural frequencies of the system. The natural frequencies are determined from the roots of the frequency equation associated with Eq. (9) by calculating the eigenvalue.

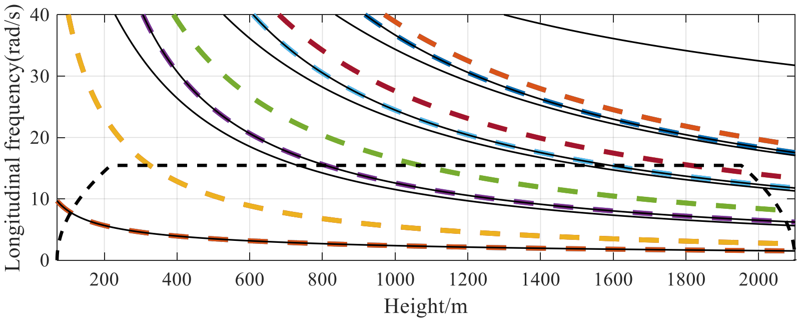

Figure 3: Frequency characteristic diagram (without tension balance device in solid line ω, with tension balance device in dotted line $\bar{\omega}$. Excitation frequency in black dotted line)

In both cases, the first order is longitudinal low frequency and they are exactly the same. As shown in Fig. 3, odd orders of longitudinal frequency are exactly consistent ($\omega_{2i-1} = \bar{\omega}_{2i-1}$), while the even orders are higher than that with tension balance device ($\omega_{2i} > \bar{\omega}_{2i}$). The second order is the rotational frequency of hoisting conveyance.

The numerical output demonstrates the rotational vibration resonates at 700m in Fig. 4 (The frequency diagram also shows the intersection of the excitation frequency and the rotation frequency). Longitudinal vibration of the hoisting conveyance is basically similar between two models. However, the resonance peak in other parts of hoisting rope is much higher than that with tension balance device (Fig. 5). The results show that the tension device fulfills its function and achieves the rope tension balances.

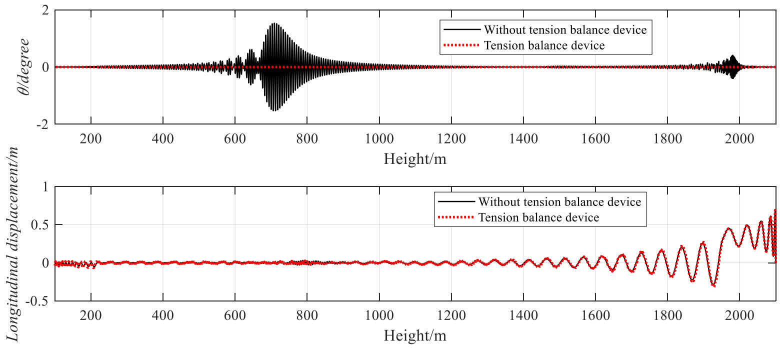

Figure 4: Terminal response when the excitation is applied to the first rope

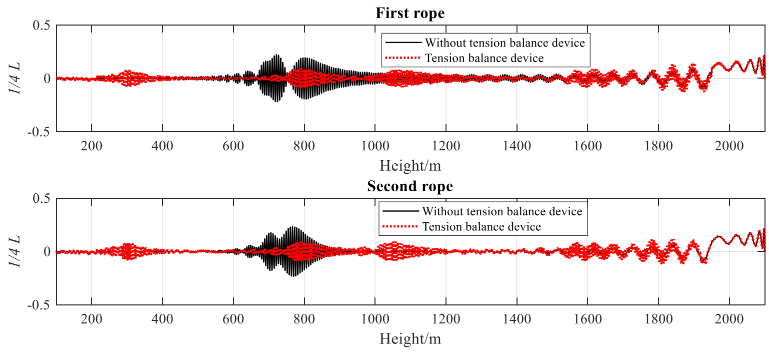

Figure 5: Vibration responses: $u_i(1/4l_i, t)$ with tension balance device or not.

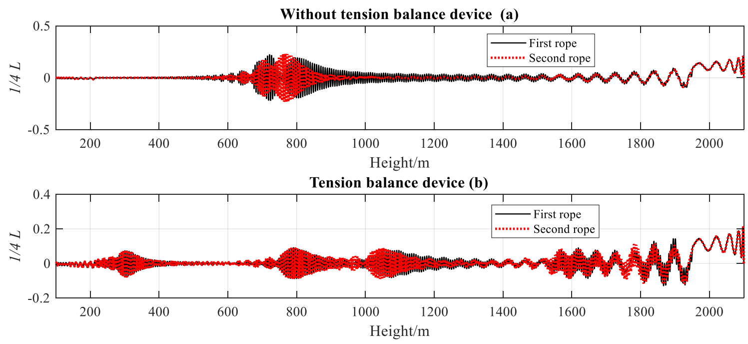

Figure 6: Displacements of two ropes at quarter length with respect to height.

The external excitation is applied at the top end of the first rope and the bottom constraint force acts on the second rope. From Fig. 6(a), the excitation applied to the first rope is indirectly transmitted to another rope through the hoisting conveyance. Since the ends of two ropes are connected to tension balance device directly, their responses are basically the same (Fig. 6(b)).

6 CONCLUSIONS

Vibration phenomena in long-travel lifting systems lead to poor ride quality. Eccentric pulleys and sheaves, systematic resonance in the electronic control system generated vibrations are typical causes of longitudinal vibrations of the lifting system. By modeling longitudinal coupled vibration of parallel hoisting systems with or without tension balance devices, it can be observed that the hoisting conveyance is subjected to rotational vibration while is excited by an eccentric pulley. The tension balance device can be adapted to mitigate the effects of vibrations and rope tension difference. It can avoid the system's rotational resonance and reduce the resonance amplitude reduced to about half. Other passive and active vibration suppression techniques can be utilized to control vibration phenomena.

ACKNOWLEDGMENTS

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People Flow Analysis in Lift Modernization

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Keywords: People flow analysis, lift modernization, site survey, simulation.

Abstract. The modernization and retrofitting of lifts in a large building presents unique challenges. Current trends in offices, such as flexible working times and workplaces, overpopulated office floors and increased communication requirements, can initiate lift modernization, which aims at keeping the building competitive. The requirements of lift modernization vary case by case according to the building type, the condition of equipment and the data available from the existing installation. This article describes three levels of people flow analysis and their benefits to customers. The levels require an increasing involvement to acquire data and perform analysis. This article can also be used as a handbook for choosing the proper level of analysis for a particular modernization project.

1 INTRODUCTION

At the moment, there are more than 14 million lifts and escalators in operation around the world. About one million new devices are installed every year. The lift modernization market increases with the number of aging devices. Modernization is very demanding when key components of a lift system prone to wear, such as the motor, drive, doors or control system, are replaced by new ones, and other good quality components are re-used. Successful modernization depends on careful planning. Therefore, the existing technical solution and the usage of the building need to be very well understood beforehand to be able to select and engineer the optimum new solutions.

The research on people flow analysis concentrates on new buildings [e.g., 1]. Not much research has been published on the principles of lift modernization, or how to estimate the improvement potential in buildings with aging lifts. Modernization is often concerned with many entrance floors, and new control systems such as destination control [2, 3]. In such cases, uppeak calculation is of limited value since it applies only to conventional control and buildings with one entrance floor. Regardless of its limitations, uppeak calculation quickly indicates whether the performance of the existing lifts is within an acceptable range and a further study is worth conducting. Then, simulation of the people flow and new equipment gives a better insight of the effects of the renovation [e.g., 4].

The analysis techniques for modernization are the same as those applied for new buildings. However, the performance of the existing equipment can be measured, and the measured data can be used instead of estimates of important parameters, which often are uncertain for the analysis of a new building. This increases the accuracy of the analysis. The people flow analysis of a modernization is based on the comparison of the existing and the new system, the situation before and after the modernization. Such a comparison shows the potential of the new equipment and provides performance indicators for building owners when deciding on the scope of the modernization.

In this paper, people flow planning service for lift modernization is divided into three levels depending on the complexity of the analysis: (1) basic, (2) advanced, and (3) consultation. A basic analysis follows the well-known principles, which are nowadays applied routinely for new buildings in the design phase. An advanced analysis is based on site surveys that collect data on lift performance, vertical people flow and passenger service quality [5]. A consultation service further widens the scope of the analysis to horizontal people flow, which can be measured by modern lobby sensors. The consultation can then utilize the wealth of data and recommend new arrangements in building guidance as well as in the location of attractions.

2 MODERNIZATION PLANNING SERVICES AND NEEDS

For a long time, lift planning has been based on morning uppeak, which has been considered the worst-case traffic condition for a lift group. This has been true with a conventional control system with up and down call buttons. In uppeak traffic, the destination control system (DCS) can handle more passengers than the conventional control [3]. The worst traffic condition for lifts with the DCS is mixed traffic, which occurs especially in office buildings during lunch-time. Nowadays, lift modernization is often initiated because passenger demand in a building has increased beyond the handling capacity of the existing lifts. Such a situation occurs when either the usage of the whole building or its occupancy levels change. In a typical modernization, the conventional control is replaced with the DCS to eliminate queuing in uppeak.

Modern building architecture prefers multi-purpose buildings and open offices. Personal working places and rooms are changed to meeting rooms or open office areas. A recent study on offices showed that open office spaces create value for users, tenants, and developers [6]. In such an agile workplace, seats can be freely occupied for the time needed, and when vacated can be occupied by another person. The survey revealed that end users spend an average of 54% of their total work time in open office spaces. Another modern trend is office hotels, where anyone can rent a room or a seat for some hours or days. Nowadays, canteens, cafeterias, restaurants and meeting rooms are often placed on any floor around the building. Many companies also encourage employees to expand their professional networks with common get-together hours with free coffee.

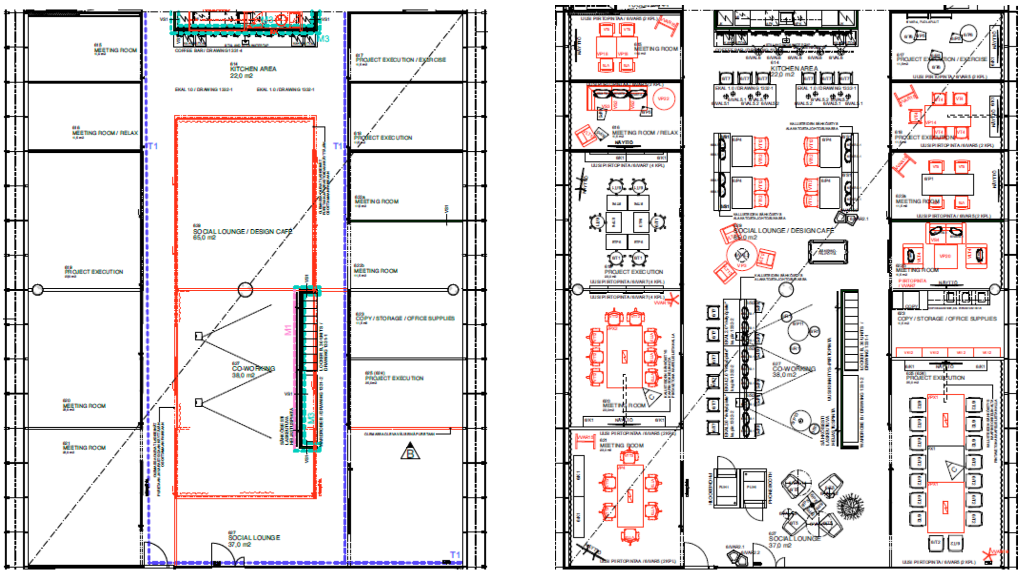

Figure 1 The renovation of a traditional office floor (left) to an open office (right)

The new office trends described above increase interfloor traffic in the building and affect the lift service quality and should thus be taken into account in the renovation process of an office building. As an example, a building with 14 office floors was recently renovated. In the renovation, four office floors were dedicated to open office space (Fig. 1). Also, the restaurant on the lowest floor and one of the renovated floors started to offer free coffee to all employees. After the renovation, the open office area attracted more people to visit the building. They wanted to meet others face-to-face, and, after the renovation, it was easier to find a free seat on the open office floors. The four-lift group which served the building had offered excellent service in the building with 15 seconds waiting times on average before the renovation. After the renovation, the number of passenger journeys had increased by more than 50%, from about 2 250 to 3 500 journeys per day as measured by the lift group [7]. This increased average waiting times to over 20 seconds.

3 BASIC ANALYSIS

Basic analysis follows the footsteps of a typical analysis for a new building and can be done without measurements on site. In addition, standard design criteria for lift groups can be applied [1, 7]. Such benchmarks are important in competing with new buildings. The main input parameters are building population based on floor areas and default lift parameters. Also, typical traffic patterns are used in the analysis. A basic analysis can be done as the first step if working practices, new tenants, restaurants, or other changes to floor utilization increase people flow inside the building.

The effect of the control system on passenger service quality can be studied with a lift traffic simulator such as the Building Traffic Simulator (KONE BTS™) [4]. Typical traffic mixes are simulated with constant passenger demand for at least two hours [9, 10]. For example, in offices the typical traffic mixes are morning uppeak with 100% incoming traffic and lunch traffic (e.g., 45% incoming, 45% outgoing and 10% interfloor traffic).

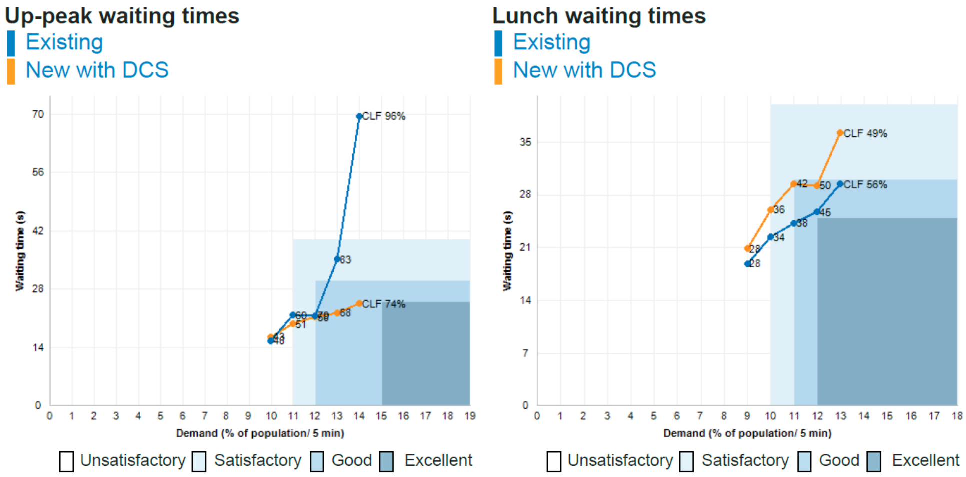

Figure 2 Average passenger waiting times with the conventional and destination control

The results in Fig. 2 show that the DCS can handle more traffic than existing control with the same service quality. CLF refers to car load ratio, i.e. the maximum filling of the car during a roundtrip. Here it is given as per cent of the rated passenger capacity. In uppeak traffic, average waiting times with the conventional control start to steeply increase after passenger demand exceeds 12% of population in five minutes. This result also shows that the handling capacity of the conventional control is 12%. With the DCS, this lift group saturates in uppeak at about 17% demand. Thus, the DCS can handle about 30% larger population or passenger demand compared to the existing group with the conventional control in uppeak traffic. In lunch traffic, average waiting times are a bit longer with the DCS compared to the existing system but remain below 40 seconds.

4 ADVANCED ANALYSIS

If lift components have come to the end of their life cycles, a deeper analysis with a lift traffic simulator is needed. The performance of the current equipment, people flow and passenger service quality should be measured. In the past, a stopwatch and pen and paper have been the only measurement methods in performance studies, but this approach is too laborious for extensive site surveys. Instead, e.g. smartphones with stopwatch and applications with accelerometer offer more efficient ways to measure lift speed, acceleration, jerk and door parameters. An automated device to collect data is preferable, but the use of such may not always be permitted [5]. In the following, the ways of measuring different simulation parameters are described in detail. The measurements aim at a simulation model that closely reproduces the current passenger service quality and lift operation. Such a model can then be applied to provide a realistic prediction of the effect of the modernization.

4.1 Lift cycle

Lift parameters may be considered as simple and straightforward to measure on site, but this is not necessarily the case. The parameters are related to the performance or cycle time, as shown in Fig. 3: door opening and closing time, door closing or photocell delay, motor- and drive-related start delay, lift flight time, door pre-opening or advance door opening time, and passenger transfer time.

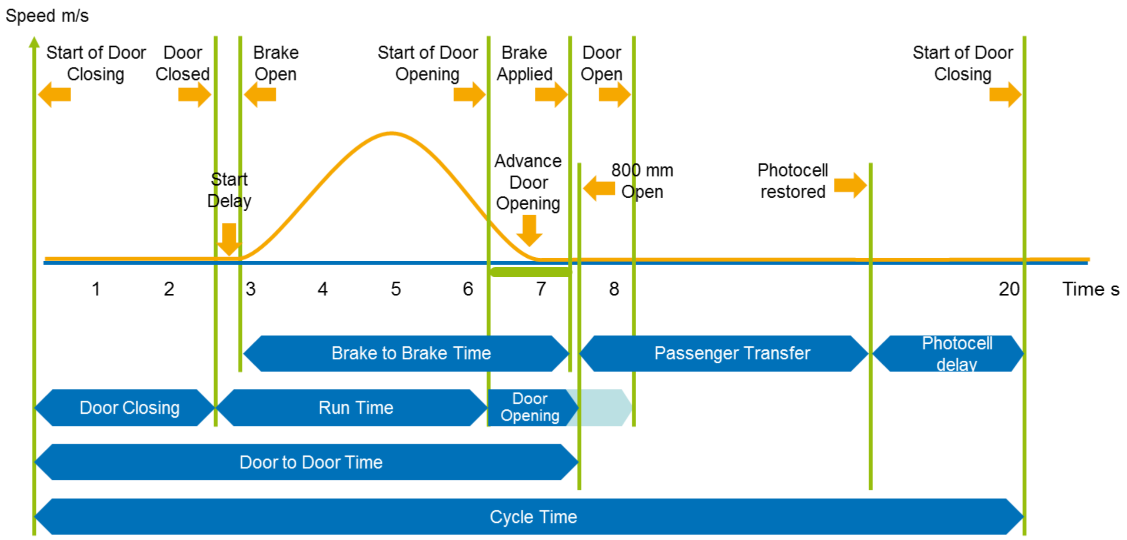

Figure 3 Lift cycle time definitions

Door opening and closing times are probably the easiest parameters to measure even with a stopwatch. Door opening time is often interpreted as until 800 mm open, when passengers can be assumed to start boarding or alighting. Door closing delay is a bit more complicated to measure, but, in the best case, could be seen from lift control parameters. Door pre-opening time is rather difficult to measure accurately. Therefore, if pre-opening is clearly visible, a half-a-second reduction in the performance time can be assumed by default.

People flow analysis requires values for rated speed, and acceleration and jerk so that the lift flight time can be accurately modelled. Usually, acceleration and jerk are not readily available but need to be measured or estimated. An accelerometer provides the most accurate estimations. It is also worth noticing that the flight time model of people flow analysis tools assumes symmetric acceleration and jerk rates during lift acceleration and deceleration. This is not necessarily the case in reality. For example, the approach of a lift to a floor may take longer than assumed by the final jerk. This can be counteracted by higher jerk rates during other parts of the flight. Therefore, an approach where only the time of final approach is measured may result in unrealistically poor lift performance [11].

Passenger transfer time is not exactly a lift parameter. However, passenger transfers constitute a major part of a lift cycle and can be estimated from lift cycles. The key is to understand its role: passenger transfer time is the average time for a single passenger to enter or leave a lift. Thus, if measurement data contains records of how long the doors had been open as well as how many passengers entered and exited during a particular stop, the average transfer time is readily available. If such data does not exist, it is possible to apply commonly used values or to try different values in the simulation to match the simulation results to the actual performance. In practice, two passenger may board a lift side-by-side. Thus, instead of two passengers boarding at, say, 1.0 second per person, the transfer time might be in the range of 1.0 to 1.5 seconds instead of 2.0 seconds.

4.2 Vertical people flow

The easiest way to measure vertical people flow is to observe and record the number of incoming and outgoing passengers in a lift lobby as they board and alight the lifts. However, this method misses interfloor traffic completely. An observer can also travel inside a lift to mark down interfloor passengers [12]. Passengers entering and exiting a lift can also be counted automatically either by a lift or by a 3D-camera-based device that is mounted in the lift ceiling [5].

If passenger counts are associated with the corresponding lift arrival and departure direction, the counted passengers per stop can be aggregated for period t to form statistics of passengers in to or out from a lift on floor i upwards or downwards, e.g., $P_{it,InUp}$. Traffic components, i.e., incoming, outgoing, and interfloor traffic, are then calculated as follows in absolute terms for E entrance floors and N floors in total in the building [7],

$$P_{t,inc} = \sum_{i=1}^{E} P_{it,InUp}, \tag{1}$$

$$P_{t,out} = \sum_{i=1}^{E} P_{it,OutDown}, \tag{2}$$

$$P_{t,intf} = \sum_{i=E+1}^{N} \left[P_{it,InUp} + P_{it,OutDown}\right]. \tag{3}$$

It is straightforward to manipulate these values into a form that can be input to a lift traffic simulator. Usually the traffic components are given as percentages. Passenger demand is given either as persons or as a percentage of the building population per five minutes. Period length, for which the described traffic definition is assumed constant, is typically 5 or 15 minutes. Five-minute periods may describe the changes in traffic more accurately but may be vulnerable to random variations. On the other hand, 15-minute periods are statistically more reliable since the longer periods allow multiple roundtrips for each lift. Fig. 4 shows a typical traffic profile or template for daily traffic in a single-tenant office building [7].

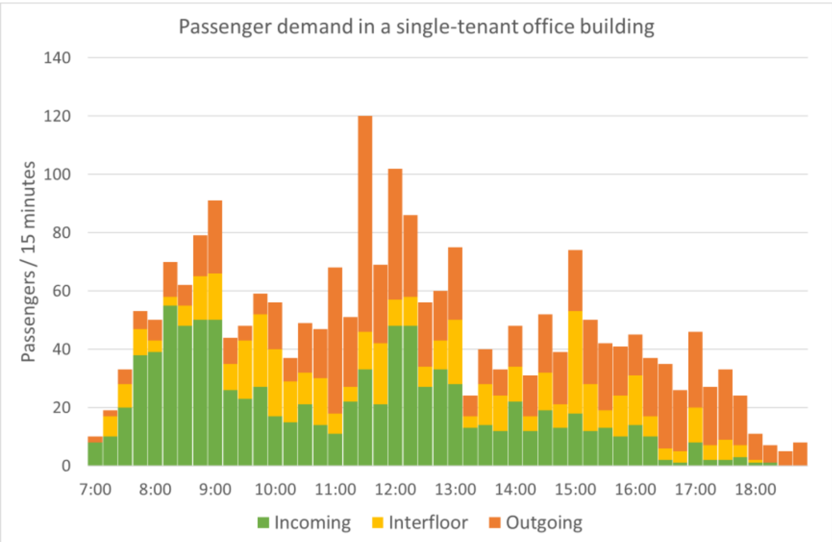

Figure 4 Measured daily vertical people flow

The above model has some limitations. First, it is applicable to a typical high-rise building, where the entrance floors are located at the bottom of the building and all populated floors above them. Second, the model misses traffic between the entrance floors. Third, the sizes of passenger batches travelling together towards the same destination floors cannot be directly observed from the passenger counts, but they can be estimated [13]. Regardless of these limitations, this current *de facto* standard model is sufficiently accurate in describing vertical people flow in tall buildings.

4.3 Building population and floor probabilities

The population on each floor can be defined according to the actual number of occupants, e.g., workplaces in offices, instead of an area-based estimation as in the basic analysis. In addition, a building security system can be used to estimate the absence rate or the number of visitors. If the building or floor usage is planned to change, the population can also be taken into account with such changes.

The automated passenger counts during lift stops can also be used to define building population and, in more complex cases, individual floor usage probabilities. The population, i.e., the number of occupants inside a building, until period T is given in Eq. 4 as the sum of differences between incoming and outgoing passengers. The total population can then be defined as the maximum of these momentary populations over the whole day.

$$POP_T = \sum_{t=1}^{T}\left[P_{t,inc} - P_{t,out}\right], \tag{4}$$

The method is demonstrated in Fig. 5, which shows the cumulative number of occupants in the building for all 15-minute periods of the day. As this example is an office building, the drop in the population after 11:00 signifies the beginning of lunch-time. After lunch, the population does not reach the highest level, about 300 persons. Probably this observation does not mean that many people leave the office after lunch. Instead, many use the stairs to return to their floors, which remains unobservable for the passenger counting in the lifts. Due to this kind of unobservable people flow, the population needs to be reset at some time in the day. In this case, the population can be assumed to reach zero at midnight. In other building types, where such a natural zero-level cannot be assumed, it is possible to find a daily minimum value and shift the population values accordingly.

Figure 5 The cumulative number of occupants (left) and examples of floor probabilities (right)

Individual floor populations can be derived from the detailed floor counts, as shown in Eq. 5. They are important parameters in defining probability distributions from which the traffic simulator generates the origin and destination floors of passengers. These probability distributions vary in time, which is also demonstrated in Fig. 5. The figure depicts two probabilities throughout the day: one that describes the bias between two entrance floors, and another that is the probability of an upper floor being an origin or a destination floor. The proportion of the lower entrance varies from 30% to 90%, where the high percentage during lunch-time is explained by the restaurant on that floor. The population on the upper floor of this example varies between 3% and 12% of the total population if the statistically volatile morning and evening periods are excluded.

$$POP_{iT} = \sum_{t=1}^{T}\left[P_{it,OutDown} + P_{it,OutUp} - P_{it,InDown} - P_{it,InUp}\right] \tag{5}$$

Thus, the probabilities of different floors being passengers' origin or destination is dynamic rather than static as assumed in the standard simulation models. Such situations can be modelled by simulating separately the most important periods of daily traffic, e.g., uppeak and lunch traffic, during which floor probabilities can be assumed constant.

4.4 Passenger service quality

A lift monitoring system such as KONE E-Link can provide data and statistics about lift cycles and passenger calls among other parameters. Especially the observed passenger service quality, such as passenger waiting times, is of great value. The current global trend is that on entrance floors a great percentage of people have their eyes and mind focused on their cell phones. This habit has the effect of making the waiting time seem psychologically shorter than it really is. This is just the opposite when a waiting passenger is alone on an upper floor with eyes focused on the hall lantern. In this latter situation, the wait can seem longer than it really is.

Omitting the psychological effects, the measured service parameters can be compared against the simulated values to validate the simulation models. In such a simulation, all control parameters need to be set as close to the site as possible. In addition, simulations can be repeated several times to achieve statistical accuracy. The site data could be collected for several days and combined into an average template, but such a model loses the relationship between the observed service quality and actual events.

In Fig. 6, an example of matching uppeak and lunch traffic simulation models with the monitoring system data is shown. The data is taken from the renovated office with KONE Hybrid DCS, which poses an additional challenge [14]. Passenger service quality is described by call time on the upper floors since only up and down call buttons are available there. The entrance floor lobbies are equipped with destination keypads, which allows the comparison of passenger waiting times.

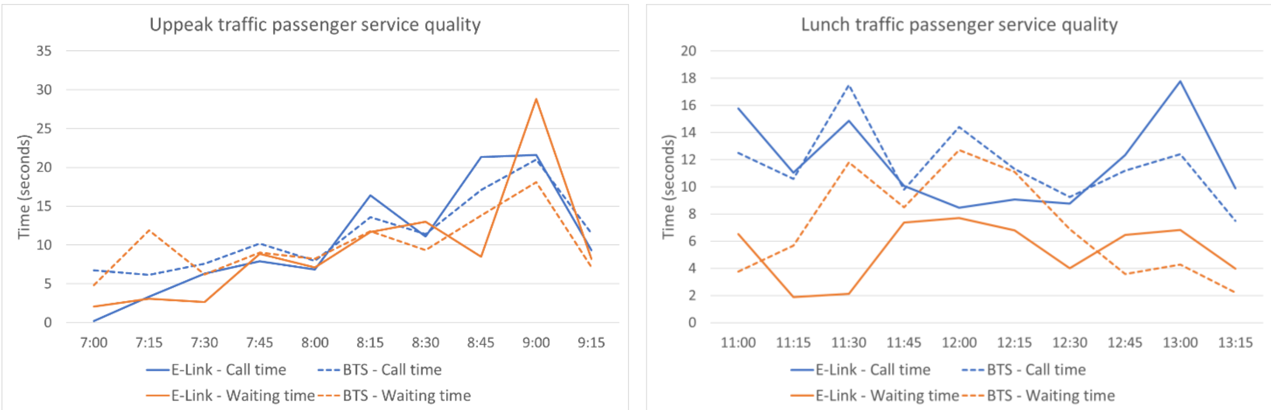

Figure 6 Passenger service quality by a lift monitoring system and a traffic simulation

According to the figure, the simulated values follow the monitoring system data quite well. However, some differences are visible, especially in average waiting times related to destination calls. This may be partly explained by tailgating, i.e., passengers who do not give the call but follow their friends to the lifts. With the assumption of individual arrivals, the simulated waiting and call times become much longer than actually recorded. It turns out that lunch traffic simulation needs to model passenger batch arrivals with the average batch size of 1.5 passengers, which was observed in this building in an earlier study [15]. The batch arrivals has been found a critical step in modelling the real traffic [16].

The average waiting and call times for the whole period can be seen in Table 1. On a statistical level, the simulated averages closely match the actual averages.

Table 1 Overall passenger service quality in uppeak and lunch traffic

	Uppeak		Lunch	
	E-Link	**BTS**	**E-Link**	**BTS**
Average Waiting Time [s]	11.4	12.9	5.7	8.2
Average Call Time [s]	14.4	13.8	12.2	12.6

5 CONSULTATION

The third level of planning service is the most comprehensive one, called building consultation service. The planning concerns not only vertical transportation but planning the entrance floor and transportation device layout to support smooth people flow in the building.

Consultation begins when an expert visits the site to study the current people flow. To determine the vertical transportation and horizontal movement of people, sensors or cameras can be installed in the main entrance floor corridors, doorways, lift lobbies and escalators. The study is based on current user routes and possible plans of future population.

Typical user groups and their routes are sketched in Fig. 7. Information about passenger journeys reveals bottlenecks and even typical passenger behaviour in the building. After becoming aware of the current people flow, both the building owner and the service provider can develop their new plan on a firm basis. The next step in the consultation service is to create a proposal for how the situation can be improved.

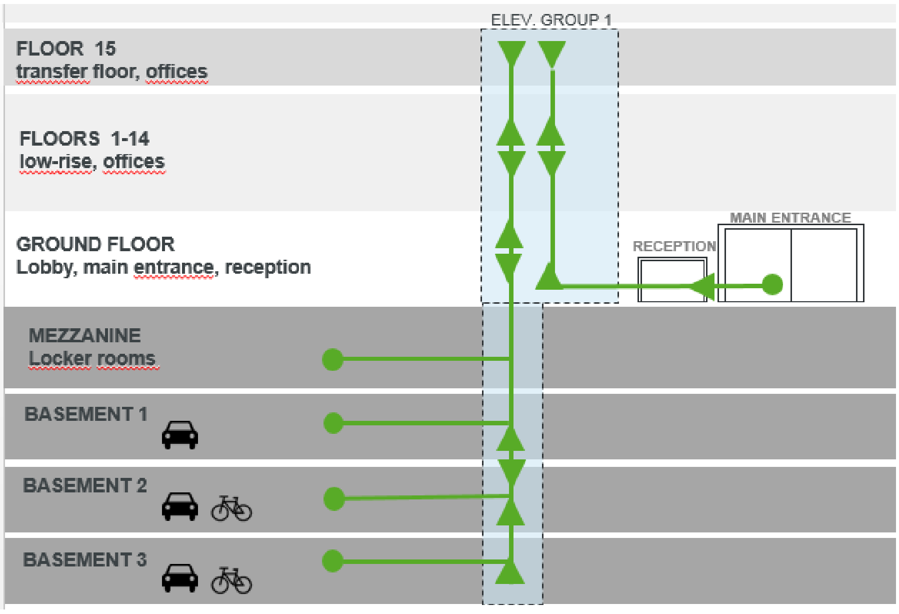

Figure 7 Example user groups and their typical routes in a building

In preparing the proposal for a modernization plan, the location of turnstiles, access control devices and destination keypads can be compared by simulations. Building traffic can be modelled with modern 3D simulation techniques (Fig. 8) [4]. The building layout is defined from architectural drawings showing the rooms, walls and corridors of the building [17]. In horizontal movement, people avoid colliding with each other, walls and other obstacles on the way. They move through doors, turnstiles and lift doors when they are open. In vertical movement, calls given by passengers from the call giving devices are allocated to lifts using real control system algorithms and software. Lift dynamics and lift models describe the physical movement of lifts in the building. Passenger traffic is generated for different user groups utilizing the measured traffic or floor populations. The journey times of different user groups as well as momentary densities in the building reveal the best plan for floor layouts. As a result of modelling the whole building traffic, the most advantageous floor layout, doors, turnstiles and transportation solutions can be proposed for the basis of the modernization.

Figure 8 A 3D model of people flow in a transit station

6 SUMMARY

In this paper, three levels of people flow planning services for lift modernization are discussed. Modernization is a current trend, for instance in Europe, where the lift market is mature. Only some components or the whole lift group can be modernized. According to the need, different levels of studies and modernization plans are proposed. Replacing the control system with a new one requires at least a basic analysis, but if it includes also access control or turnstiles, advanced or consultation level planning may be necessary.

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BIOGRAPHICAL DETAILS

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Regulating Lift and Escalator Safety in Hong Kong

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Keywords: Electrical and Mechanical Services Department, Hong Kong, lift and escalator safety, regulator, facilitator, promotor.

Abstract. Quality and safe lift and escalator services are vital for a densely populated city dominated by high-rise buildings as in Hong Kong. The Electrical and Mechanical Services Department ("EMSD") of the Government of Hong Kong Special Administrative Region ("HKSAR") is responsible for regulating the safety of lifts and escalators in Hong Kong by enforcing the Lifts and Escalators Ordinance (Cap.618) through various means, such as conducting risk-based audit inspections, carrying out prosecution and disciplinary proceedings, implementing codes of practice as well as the registration of contractors, engineers and workers. EMSD has also rolled out various initiatives to facilitate the sustainable development of the lift and escalator trade, including maintenance price survey for lifts, performance ratings of registered contractors, collaboration with the trade and training institutes to recruit new blood etc. In addition, EMSD promotes lift and escalator safety to the public via diversified publicity and education programmes. This paper will examine the ways on how EMSD plays its role as "Facilitator" and "Promoter" in addition to its traditional role as a "Regulator" on regulation of lift and escalator safety in Hong Kong, and the outcomes of its efforts.

1 ENFORCEMENT OF LIFTS AND ESCALATORS ORDINANCE

Hong Kong is the city with the most skyscrapers in the world[1]. There are over 300 skyscrapers which rise taller than 150 meters and over 1,200 high-rise buildings which rise taller than 100 meters in the city. Quality and safe lift and escalator services are vital for such vibrant and vertical city with population density of around 6,830 persons per square kilometer and yearly visitors of more than 58 million[2].

1.1 The old regulatory regime

It takes us back to 1960s when the regulatory control over lift and escalator safety was first implemented in Hong Kong. The Lifts and Escalators (Safety) Ordinance (Cap.327) ("ex-LESO") was enacted in 1960 as the first ordinance to regulate lift and escalator safety in Hong Kong. It provided a legal framework for EMSD to exercise regulatory control over lift and escalator safety in Hong Kong. Since its enactment, the ex-LESO went through various major amendments over the years, but still it was not able to resolve the deep-rooted drawbacks which had hampered enforcement for lift and escalator safety, e.g. ambiguity of responsibilities among owners and property management agents in upkeeping safety of lifts and escalators, lack of a registration system to monitor the quality of frontline workers who carried out lift and escalator works, low penalty level for committing offences, lack of public surveillance on expiry of use permit for lifts and escalators, etc.

1.2 Revamp of the legislative framework

In 2009, the Government of the HKSAR took the initiative to transform the regulatory regime over lift and escalator safety to address the increasing public concerns over lift and escalator safety and to

[1] Top 10 Cities with the Most Skyscrapers in the World, https://wearetop10.com/cities-with-the-most-skyscrapers/

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make the regulatory framework capable of coping with market changes and long term development requirements. Following public consultation and the necessary legislative processes, the Lifts and Escalators Ordinance (Cap. 618) (“LEO”) was enacted and came into full operation in end 2012 to regulate lift and escalator safety and related matters in Hong Kong.

Major modifications of the legislative requirements in respect of lift and escalator safety in the LEO include:-

- adoption of the “Responsible Person” (RP) concept to replace “owner” – both owners and property management agents have the common objective for upkeeping the safe operation of a lift or an escalator;
- expanding the applicability of regulatory control – all lifts and escalators within the territory, including those belonging to the Government of the HKSAR and the Housing Authority, are now under a unified regulatory system;
- strengthening the registration regime – formalized and upgraded the requirements for registration as contractors and engineers, imposed the need to renew registrations at intervals not exceeding five years, introduced the worker registration system to grant legal status to qualified tradesmen, and required practitioners to complete professional development training for renewal of their registrations;
- stiffening the penalties for offences – the maximum fine for offences is increased to HK$200,000, and there are no changes in imprisonment terms; and
- additional measures to improve operational effectiveness and efficiency – the regulator, i.e. EMSD, is empowered to issue “Improvement Orders” to demand rectification, within a specified period, of non-compliances or defective items of a lift or an escalator; and periodic examinations of a lift or an escalator can be advanced by not more than two months to accommodate use permit processing time.

The changes in the legislative requirements of the LEO as compared with the ex-LESO are significant, in particular in (i) the use of RP to replace “owner” to make more stakeholders subject to regulatory control, and (ii) imposition of the registration renewal requirement under which existing registrants would have to give up the perpetuity of their registration status. Notably, the change would be extremely difficult, if not impossible, without the mutual understanding and concerted efforts of stakeholders acting in common pursuit of better lift and escalator safety.

1.3 Regulator for lift and escalator safety

As the regulator, EMSD regulates lift and escalator safety in Hong Kong under the legislative framework of the LEO by various means:-

- Equipment compliance

  The LEO stipulates that a registered lift/escalator contractor who undertakes the installation of a lift/escalator must ensure that the works are not to be carried out unless the lift/escalator and all its safety components[3] are of a type in respect of which the contractor has obtained approval

[3] In accordance with Schedule 2 of the LEO, the safety components for lifts include safety gear, overspeed governor, door locking device, buffer, ascending car overspeed protection means, unintended car movement protection means and any safety circuit for a lift that contains any electronic component, while those for escalators include step and pallet for an escalator.

from Director of Electrical and Mechanical Services (DEMS). In applying the type approval, the following information should be provided for detailed assessment by EMSD:-

- information of the lift/escalator manufacturer, e.g. name, address, history, organization, size of plant, product range, yearly production, brochure, job references, etc.;
- general specifications of the lift/escalator applying type approval, e.g. type examination certificates, model numbers and their applications, type of major components adopted in the lift/escalator model, confirmation from the manufacturer that the brands/models of lift/escalator are designed and manufactured to the requirements of the "Code of Practice on the Design and Construction of Lifts and Escalators" (Design Code) issued by EMSD, etc.;
- type test certificates and reports for safety components issued by approved independent testing institutes;
- technical information of the lift/escalator applying type approval, e.g. installation, operation and maintenance manuals, typical electric schematic diagrams of the power circuits and safety circuits with explanatory notes, supporting drawings, illustrations and calculations, maintenance schedules issued by manufacturers, etc.;
- quality assurance certificate (e.g. ISO 9001) including manufacturers of lift/escalator model and related safety components; and
- arrangement of training and technical support provided by manufacturers.

● Quality control

As stipulated by the LEO, only Qualified Persons[4] (QPs) or persons under the direct supervision of a QP at the place at which the works are carried out are allowed to carry out lift and escalator works. QPs are required to be in possession of the necessary qualifications and experience, and have obtained registrations from EMSD. If not registered, the personnel who want to undertake the lift/escalator works have to work under the direct supervision of a QP. Personnel who have registered under the LEO as contractors, engineers, and workers are required to renew their registrations so as to be able to continue to carry out lift/escalator works. The renewal requirement is to ensure that they are fit and proper to continue to perform the legislative duties conferred or imposed on them.

On the other hand, the LEO imposes restriction on subcontracting lift and escalator works. Except with written approval of EMSD, a registered lift/escalator contractor who undertakes any lift/escalator works (except installation or demolition of a lift/escalator) must not subcontract the works or any part of the works to any other person who is not a registered lift/escalator contractor.

● Setting of standards

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[4] Qualified Person is defined under section 2(1) of the LEO. In essence, a Qualified Person can be taken, in relation to lift/escalator works, as a registered lift/escalator worker or registered lift/escalator engineer employed by the registered lift/escalator contractor who has been contracted to carry out the lift/escalator works. An independent registered lift/escalator engineer can also be regarded as a Qualified Person in respect of any lift/escalator examination being carried out.

The LEO empowers DEMS to issue any code of practice that in DEMS's opinion is suitable for providing practical guidance in respect of any matter concerning the safety of lifts or escalators, including providing practical guidance in respect of the design, use and operation of lifts or escalators and providing practical guidance to persons who carry out any lift works or escalator works. In this regards, EMSD has issued two codes of practice, including (i) the "Code of Practice on the Design and Construction of Lifts and Escalators" (Design Code), which sets out technical details, methods, procedures and safety rules for compliance with the requirements of the LEO on the design and construction of a lift and an escalator, or any associated equipment or machinery of a lift or an escalator, and (ii) the "Code of Practice for Lift Works and Escalator Works" (Works Code), which provides guidance to set out the minimum industry standards for satisfying the requirements under the LEO, taking into account the trade skills and risk perception of the general practitioners. In drafting these two codes of practice, references have been made to relevant safety standards of the European Standards Institution as well as local safety requirements derived from previous lift and escalator incidents happened in Hong Kong, and the trade together with relevant stakeholders have been consulted on the contents with their comments suitably incorporated.

There are also other standards set out for the trade under the LEO to ensure lift and escalator safety. As stipulated in the Lifts and Escalators (General) Regulation (Cap.618A) (a regulation made under the LEO), a registered lift/escalator contractor is required to:-

- post a notice in specified form signifying the suspension of service of a lift/escalator within 4 hours if the normal use and operation of the lift/escalator cannot be resumed within 4 hours from the time at which an incident involving the lift/escalator has come to the knowledge of the contractor; and

- attend to the failure of any emergency device of a lift (i.e. alarm system, emergency lighting, intercommunication system and ventilation fan) within 4 hours from the time when it has knowledge of the failure and notify DEMS in the specified form, within 24 hours after it has knowledge of the failure, if it is unlikely that the failure can be rectified before the end of the 24-hour period.

- Prosecution and disciplinary proceedings

The LEO imposes different levels of fine and imprisonment to reflect the seriousness of different offences. In order to have punitive and deterrent effect against contravention, the penalties for offences have been stiffened in LEO – the maximum fine for offences is increased from HK$10,000 to HK$200,000 while the longest imprisonment remains as 12 months.

Apart from prosecution, the LEO also establishes, as further punitive and deterrent measure, disciplinary proceedings to punish the registered persons (i.e. registered contractors, registered engineers or registered workers) on committing disciplinary offences, which include the followings:-

- committing misconduct or neglect in any professional respect;

- convicting an offence under the LEO;

- obtaining registration or renewal of registration under the LEO by fraud or misrepresentation;

- having failed, without reasonable excuse, to attend before a disciplinary board or an appeal board either as a witness or as a person in respect of whom the board is meeting; and

- convicting in Hong Kong or elsewhere of any other offence that may bring their profession into disrepute.

A complaint alleging a disciplinary offence against a registered person may be made by DEMS or by any other person by submitting the complaint in specified form to DEMS. A disciplinary board, consisting members of professional engineers, registered lift/escalator engineers, registered lift/escalator workers, property managers and laypersons having the role of management committee member or lift/escalator owner, will be established to hear the case and to decide on the disciplinary case. If the disciplinary board determines the registered person in concern has committed the disciplinary offence alleged in the complaint, it may order the person be reprimanded and fined a maximum sum of HK$100,000 for registered lift/escalator contractors or HK$10,000 for registered lift/escalator engineers/workers, and order the Registrar to cancel or suspend the registration of the person.

- Registration system

Under the LEO, the registration regime covers registered lift/escalator contractors, engineers and workers, with different qualifications and experience requirements as shown below.

- Registered Contractor (RC)

  An individual is eligible for application for registration as a lift/escalator contractor, if he/she:-

  - has at least a director, partner or employee who is a corporate member of the Hong Kong Institution of Engineers, or a registered lift engineer and/or a registered escalator engineer;
  - has not less than two other employees with one of them being qualified to carry out lift works and / or escalator works independently;
  - is in possession of necessary facilities, resources and workforce for carrying out lift works or escalator works; and
  - is capable of obtaining technical support from a lift manufacturer or an escalator manufacturer for technological updating, technical training of staff, and sourcing of spare parts.

- Registered Engineer (RE)

  An individual is eligible for application for registration as registered lift/escalator engineer, if he/she has the qualifications and experience specified in any of the following routes:-

ROUTE 1	is a registered professional engineer under the Engineers Registration Ordinance (Cap. 409) in mechanical engineering, marine and naval architecture engineering, electrical engineering, electronics engineering, building services engineering, or control, automation and instrumentation engineering; and
	has at least 2 years' relevant working experience and has the necessary practical experience in lift works or escalator works
ROUTE 2	has a bachelor degree in mechanical engineering, marine and naval architecture engineering, electrical engineering, electronic engineering,

	building services engineering, or such equivalent or higher qualification as recognized by the Registrar; and
	has at least 4 years' relevant working experience and has the necessary practical experience in lift works or escalator works

The applicant has to pass in both the written examination and interview organized by EMSD in order to obtain the registration status.

- Registered Worker (RW)

  An individual is eligible for application for registration as registered lift/escalator worker, if he/she has the qualifications and experience specified in any of the following routes:-

ROUTE 1	has been an apprentice in trade of lift electrician or lift mechanic or equivalent and completed a craft certificate course recognized by the Registrar; and
	has at least 4 years' relevant working experience, of which at least one year was obtained within the 5-year period immediate before the date of submission of the application; and has necessary practical experience and relevant training.
ROUTE 2	has completed a certificate course in building services engineering, electrical engineering, electronic engineering, marine engineering, mechanical engineering, or such equivalent or higher qualifications as recognized by the Registrar; and
	has at least 4 years' relevant working experience, of which at least one year was obtained within the 5-year period immediate before the date of submission of the application; and has necessary practical experience and relevant training.
ROUTE 3	has passed a trade test for lift works or escalator works recognized by the Registrar; and
	has at least 8 years' relevant working experience, of which at least one year was obtained within the 5-year period immediately before the date of submission of the application; and has the necessary practical experience and relevant training.

To regularly assess the competency of these registered persons, the registration status of all these registered persons has to be renewed by application to the Registrar at intervals not exceeding five years.

- Risk-based audit inspections

  The risk-based enforcement approach is adopted for lift and escalator safety regulation. A high level of audit inspections is maintained with a closer focus on lifts and escalators at higher risk and works performed by practitioners with poorer performance track records. In reinforcing the risk-based inspections, reviews are regularly made in respect of different aspects of the regulatory system including:-

- contractors – analyze the level of risk by comparing, for example, the number of complaints against different RCs, tip-off cases, durations and/or frequencies of lifts being put out of services, staff movements, drastic changes in maintenance capacities or workforce level of a RC, etc.;
- lifts/escalators – changeover frequencies of maintenance services providers, age of installations, complexity of installations, number and types of complaints, problematic locations, etc.;
- works – lift works involving major alteration, brand new design or non-standard installation, at unconventional locations, etc.

Currently, the target number of audit inspections for lifts and escalators carried out by EMSD is set at around one in seven lifts/escalators. Despite limited manpower resources, the yearly average number of inspections for lifts and escalators carried out by EMSD has increased from 8,964[5], while the ex-LESO was in force, to 11,207[6], while the LEO was in force, i.e. an increase of about 25%. With such dedication to inspections as well as maintenance and examination works by registered persons, the yearly average of reported incidents[7] due to equipment fault involving per 1,000 nos. of lifts and escalators has decreased from 0.67[8], while the ex-LESO was in force, to 0.15[9], while the LEO was in force, i.e. a decrease of about 78%.

- Shared Responsibility

Under the new regulatory regime of the LEO, the concept of “Shared Responsibility” is adopted. With such concept, different stakeholders, including registered lift and escalator contractors, engineers, workers, RPs[10] for lifts and escalators, the Government of the HKSAR as well as the general public (as users of lifts and escalators), should jointly take part in and share the responsibility for upkeeping lifts and escalators in a proper state of repair and in safe working order. In this connection, EMSD has taken various measures, with the aim of raising stakeholders’ awareness (in particularly that of RPs for lifts and escalators and the general public) of the importance of “Shared Responsibility” in assuring high lift/escalator safety standard and quality lift/escalator services:-

- Clear indication of validity period on use permit

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5 Such figure is the yearly average of inspections for lifts and escalators carried out by EMSD for the years from 2008 to 2012 (i.e. 5 years in total), during which the ex-LESO was in force.

6 Such figure is the yearly average of inspections for lifts and escalators carried out by EMSD for the years from 2013 to 2017 (i.e. 5 years in total), during which the LEO was in force.

7 According to the LEO, when there is a lift/escalator incident belonging to a type as listed in Schedule 7 of the LEO, the Responsible Person for the lift/escalator must inform EMSD within 24 hours after the incident comes to the Responsible Person's knowledge.

8 Such figure is the yearly average of reported incidents due to equipment fault involving per 1,000 lifts and escalators for the years from 2008 to 2012 (i.e. 5 years in total), during which the ex-LESO was in force.

9 Such figure is the yearly average of reported incidents due to equipment fault involving per 1,000 lifts and escalators for the years from 2013 to 2017 (i.e. 5 years in total), during which the LEO was in force.

10 Responsible Person for a lift/escalator is defined under section 2(1) of the LEO as a person who owns the lift/escalator or any other person who has the management or control of the lift/escalator (e.g. representative of building or facility management company).

User surveillance is a very effective means to spot non-compliances. A new use permit arrangement has been introduced in the LEO to replace the posting of safety certificates adopted in ex-LESO to facilitate user surveillance. Succinct and key information, i.e. the expiry date, is now shown on the use permits prominently to enable the public to effectively monitor whether a use permit has expired, and whether or not the lift or escalator has been examined by a registered lift or escalator engineer to confirm its safe working status.


機電工程署 EMSD
准用證(升降機)
Use Permit (Lift)
升降機及自動梯條例(第618章)
THE LIFTS AND ESCALATORS ORDINANCE (CHAPTER 618)
地點編號 Location ID. 1234567 – 001
升降機編號 Lift No. 1
地址 Address 香港安全街安全大廈
SAFETY BUILDING, SAFETY STREET, HONG KONG
屆滿日期 Date of Expiry 15 / 2 / 2014
日Day 月Month 年Year
發出日期 Date of Issue 30 / 1 / 2013
日Day 月Month 年Year
123456789


機電工程署 EMSD
准用證(自動梯)
Use Permit (Escalator)
升降機及自動梯條例(第618章)
THE LIFTS AND ESCALATORS ORDINANCE (CHAPTER 618)
地點編號 Location ID 2345678 – 001
自動梯編號 Escalator No. 1
地址 Address 香港安全街安全商場
SAFETY SHOPPING ARCADE, SAFETY STREET, HONG KONG
屆滿日期 Date of Expiry 15 / 2 / 2014
日Day 月Month 年Year
發出日期 Date of Issue 30 / 1 / 2013
日Day 月Month 年Year
123456789

Figure 1 Use permit for lift (left) and escalator (right) adopted in LEO

- Incident reporting by Responsible Persons

  The LEO stipulates that, if there is an incident[11] relating to a lift or an escalator, the RP for the lift or escalator must within 24 hours after the incident comes to the person's knowledge, notify EMSD and relevant registered lift or escalator contractors in writing. This could advocate proper management of lifts and escalators among RPs who now have legal obligations to ensure the safe operation of the installations under the LEO.

- Announcement of maintenance price figures for lifts

  EMSD has released the average maintenance price figures on the lifts in private residential and commercial premises (which are both based on statistical analysis on the contract prices collected from independent sampling survey on lifts in Hong Kong) on half-yearly basis for public reference since 2014 and 2015 respectively. The sharing of price information serves as a reference for facilitating RPs for lifts to choose among registered lift contractors for provision of lift maintenance services.

- Registered lift and escalator Contractors' Performance Rating System

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[11] Incident is defined under Schedule 7 of the LEO. Lift/escalator incidents to be reported to EMSD include, for instance, death or injury of person involving a lift or any associated equipment or machinery of a lift/an escalator, and, failure of main drive system, safety component and safety equipment of a lift/an escalator.

EMSD has launched the registered lift and escalator Contractors' Performance Rating (CPR) System since 2009. Under the system, EMSD posts on its website the rating indices of RCs in regard of their performance in lift and escalator maintenance services (based on non-compliances identified by EMSD during audit inspections), warning letters issued to RCs, number of reported major incidents, etc. The release of such information for the public's reference injected greater objectivity and transparency into the control process. The information enables RPs for lifts and escalators to make an informed choice in the selection of RCs for the provision of lift and escalator maintenance services and, at the same time, impressed on RCs the need for improving their services so that they can achieve a better performance rating and popular reputation.

2 FACILITATION OF THE SUSTAINABLE DEVELOPMENT OF THE LIFT AND ESCALATOR TRADE

2.1 Understanding the trade

EMSD is not only the regulator, but also the "Facilitator" for the lift and escalator trade in Hong Kong. It keeps its finger on the pulses of the trade through conducting Trade Survey so that necessary assistance can be timely provided to help the sustainable development of the trade. The bi-annual Trade Survey is for:-

- collecting information for analysis of the salary trend for past years and salary structure of workforce (including registered engineers, registered workers and general workers) in the lift and escalator trade;
- canvassing information on the working conditions and working pattern of the workforce (including registered engineers, registered workers and general workers) in the lift and escalator trade; and
- seeking views of various groups of practitioners in the lift and escalator trade regarding the difficulties and problems encountered, ways to attract new employees, issues in the trade which hinder the provision of quality services to the public and how to improve the situation.

It is glad to note from the 2016 Trade Survey results that the satisfaction level of trade practitioners, for both REs and RWs, have been going up and their salaries have increased by around 6.5% a year. This betterment could partially be attributed to EMSD's series of administrative measures to nurture the healthy trade environment that is conducive to enhance lift and escalator safety, including the CPR System, maintenance price survey for lifts in private and commercial premises, the pro forma maintenance contract for use by RPs, etc. With these information announced periodically and publicly, RPs can readily make reference to them when selecting a competent RC to deliver quality services at a fair and reasonable market price.

Apart from the Trade Survey, EMSD also holds regular liaison meetings with trade associations and sets up a "Lift and Escalator Maintenance Working Group" in collaboration with the trade practitioners. On the other hand, a permanent non-statutory body "The Lift and Escalator Safety Advisory Committee (LESAC)" has also been established since 2013 to advise EMSD on matters relating to the administration and enforcement of the LEO and other matters relating to lift and escalator safety referred by EMSD. The establishment of LESAC provides a forum, with a wide participation[12] to facilitate discussions and exchange of views in a broader perspective on matters

[12] The LESAC involves committee members from professional bodies, lift and escalator trade, training institutions, property & facility management sector, general community and the government of the HKSAR.

relating to lift and escalator safety. It has also enhanced community participation which at the same time brought in expertise and experience in related matters for better formulation and implementation of policy by EMSD.

2.2 Attraction of new blood to the industry

Having said of the betterment of the trade, there is a sustained concern of the trade on its aging workforce, with the average age of REs and RWs at 50.2 and 46.7 respectively. To this end, there has been close collaboration between the trade and EMSD in the past years to attract more new blood to join the industry. Starting from 2014, the government and the industries introduced the "Earn & Learn" Scheme to provide cash allowance, guaranteed salary, as well as structured training programme for attracting young people to enter the trade as apprentices. Since then, the number of new apprentice has significantly increased from about 70 a year to over 250 a year in 2016, and such enrolment has been sustained in the current school year. On the other hand, EMSD has also taken the initiative to establish partnership with the trade to launch the "Pilot Cooperative Apprentice Training Scheme (PCATS)" since 2016 to attract more new blood to join the lift and escalator trade. Technician trainees under the Scheme are trained in both RCs and EMSD; and on completion of their 4-year training, will acquire sufficient experience for registration as RW to work independently in RCs.

Apart from recruitment of apprentice, the number of new trainees taking graduate engineer training scheme in RCs has also increased substantially from the usual annual intake of about 5 a year to 15 in 2017. Since it has been planned to ultimately uplift the registration qualification for RE to registered professional engineer, EMSD has been actively encouraging RCs to provide Graduate Scheme "A" Training programme, accredited by the Hong Kong Institution of Engineers, to more graduate engineers.

2.3 Cultivation of innovation and technology

EMSD holds regular seminars to promote new lift and escalator technology. Trade practitioners and professionals will be invited to share their latest products and research results with a view to promoting their use in the trade. These seminars provide a forum, with participation of practitioners from the lift and escalator trade as well as property and facility management, to facilitate discussions and exchange of views on the latest development of lift and escalator technologies.

In view of the tight manpower of the industry, the better and wider use of technology for relieving the manpower is certainly welcome. EMSD has conducted a pilot project on adopting Remote Monitoring System (RMS) for lifts in 2015. As reflected from the pilot project, RMS could effectively reduce 20% of system breakdown and 45% of maintenance suspension time. With the fruitful outcome, EMSD has already extended the implementation of RMS to other government premises and liaised with RCs to adopt RMS for lift and escalator maintenance services in government premises.

Furthermore, in line with the directive of 2017 Policy Address on development of Innovation and Technology (I&T) in Hong Kong, EMSD has launched "E&M InnoPortal"[13]. It is a platform for a public list of the technology needed by various government departments, public bodies and the Electrical & Mechanical (E&M) trade, and it invites the I&T collaboration from innovators in the private sector to solve the problems. On the other hand, universities and start-ups etc. are also welcomed to put on the platform their E&M related innovation and new technologies, including those related to lifts and escalators (e.g. new products developed for enhancing efficiency and safety of lifts

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[13] Website of "E&M InnoPortal": https://www.emsd.gov.hk/minisites/inno/index.html

and escalators), to match the market needs. EMSD will provide venues for trial of suitable projects, conduct prototype testing and pilot projects in a collaborative way, and upload validated performance reports of trial cases to the platform for sharing with the public with a view to jointly promoting and driving the research and development and application of E&M related innovation and technology.

2.4 Promotion of occupational health and safety

EMSD has been devoting continued efforts in promoting the safety of lifts and escalators. Over the years, different types of competitions, e.g. lift and escalator safety quiz, and lift and escalator work safety improvement competition, have been held to enhance the safety of lifts and escalators in the industry and related organizations, thereby reducing accidents and enhancing the trade practitioners' awareness of work safety. EMSD also jointly organizes regular seminars with the Construction Industry Council to promote lift and escalator work safety. On the other hand, issues relating to lift and escalator work safety will be discussed in the regular LESAC meetings, liaison meetings with trade associations and meetings of the Lift and Escalator Maintenance Working Group.

3 PROMOTION OF LIFT AND ESCALATOR SAFETY TO THE PUBLIC

EMSD also acts as a "Promotor" for lift and escalator safety in Hong Kong by implementation of various public education and publicity activities to enhance the public's awareness of the safe use of lifts and escalators and the importance of proper maintenance for these installations.

3.1 Public education on lift and escalator safety

Over the years, EMSD has been carrying out public education on lift and escalator safety by various means. Below are some of major activities of public education:-

- conducting seminars on lift and escalator safety for the general public emphasizing relevant legal requirements under LEO and proper and safe use of lifts and escalators;
- implementing Safety Ambassador Outreach Programme – in 2017, over 400 sessions of outreach talks were conducted for kindergartens, youth centres and elderly centres, reaching over 19,000 participants to promote the proper and safe use of lifts and escalators.
- producing TV Announcements in the Public Interests, leaflets, guidelines, posters, stickers, newsletters, promotional videos, etc. on lift and escalator safety;
- carrying out various publicity activities to promote lift and escalator safety, e.g. carnival, competition, symposium, etc. – in 2017, EMSD has jointly organized the "Building Management Week 2017", which was well received by participating organizations and the public, with Water Supplies Department, Buildings Department, Fire Services Department, Food and Environmental Hygiene Department and Home Affairs Department to promote good practices in quality building management; and
- setting up "Responsible Persons' Corner"[14] at EMSD website to provide one-stop information on lift and escalator safety for reference by RPs.

[14] Website of "Responsible Persons' Corner:
https://www.emsd.gov.hk/en/lifts_and_escalators_safety/responsible_persons_corner/index.html

3.2 Quality Lift Service Recognition Scheme

EMSD has launched the "Quality Lift Service Recognition Scheme" (QLSRS) as a pilot scheme in 2015. It is a voluntary lift service recognition scheme targeting at RPs for lifts of private buildings, with the following objectives:-

- to encourage RPs to enhance the safety level of their lifts and to make the operation of the lifts more effective, reliable and comfortable through implementation of modernisation works; and
- to improve the lift management services of RPs of private buildings to meet the users' increasing demand for quality lift services.

The target participants of the QLSRS are RPs for lifts of private buildings, which include private residential buildings, office buildings, industrial buildings, shopping malls and hotels.

Applicants of the QLSRS are assessed in the following areas:-

- status of implementation of the seven lift modernisation solutions;
- lift management performance of the RP; and
- lift suspension time due to equipment failure.

Qualified RPs who achieved the specified standards will be presented with certificates with relevant ratings (Gold, Silver and Bronze Awards) in recognition of their achievements in implementation of lift modernization works as well as their dedication to continuous provision of quality lift management service. The QLSRS in 2015 has received 94 no. of applications, covering 1,230 no. of lifts, 254 no. of premises and 39 no. of Incorporated Owners / property management companies. 3 Gold Awards, 26 Silver Awards and 16 Bronze Awards were issued. With the support from applicants and the trade, the QLSRS has driven a positive effect in promoting the industry to continuously enhance the quality of lift services. Full implementation of the QLSRS is being planned for launching in late 2018.

3.3 Promotion of lift and escalator modernisation

Lifts and escalators in Hong Kong were installed in different decades. Although they adopted the level of technology appropriate at the time of installation, there is room for improvement to make them safer, more reliable and comfortable with the rapid technological advancement in recent years. In this connection, EMSD has issued the "Guidelines for Modernising Existing Lifts"[15] and "Guidelines for Modernising Existing Escalators"[16] in 2012 and 2016 respectively to encourage RPs to carry out modernization works for their aged lifts and escalators through the recommended solutions.

[15] "Guidelines for Modernising Existing Lifts": https://www.emsd.gov.hk/filemanager/en/content_803/Guidelines%20for%20Modernising%20Existing%20Lifts%20(E).pdf

[16] "Guidelines for Modernising Existing Escalators": https://www.emsd.gov.hk/filemanager/en/content_803/Guidelines%20for%20Modernising%20Existing%20Escalators%20(E).pdf

As a further step-up promotion for lift modernisation, EMSD has set up the "Lift Modernisation Resource Corner"[17] at EMSD website in 2017 to provide one-stop information to RPs regarding lift modernisation. EMSD has also been issuing letters to RPs of aged lifts since 2017 for reminding them of the modernisation solutions applicable to their lifts and encourage them to implement the appropriate recommended solutions as soon as possible.

4 WAY FORWARD TO ENHANCE THE SAFETY OF LIFTS AND ESCALATORS IN HONG KONG

Acting expeditiously on important and emerging issues in the best interests of the public, EMSD will continue to deploy resources effectively to focus attention on where the potential risks are comparatively higher. Looking forward, EMSD will step up regulatory efforts in causing RPs for lifts and escalators to implement modernisation works for aged lifts and escalators to enhance public safety. In particular, EMSD will formulate checklists for assessing the risk level of lifts and require registered lift engineers to use the checklist for risk assessment during annual examination of lifts. If, after assessment, the risk of the lift is identified high, EMSD will consider issuing an Improvement Order requiring the RP to implement modernisation works for the concerned lift within a reasonable time limit. Study will also be conducted to assess whether it is necessary to mandate RPs to modernise their aged lifts. In this regard, EMSD will make reference to the relevant experience of overseas countries, including the implementation status of relevant legislation, consider the feasibility of dedicated supporting measures to RPs for assisting them to comply with the requirements, and conduct regulatory impact assessment and timely consultation with the public and the industry on the proposed initiative.

On the other hand, efforts will be focused on the harmonisation of the Design Code with new EN standards. With EN81:1/2:1998+A3:2009 superseded by EN81-20:2014 and EN81-50:2014 in September 2017, different milestones requiring type-examination certificates complying with the new EN standards have been set for the type approval for new and existing models of lifts as well as safety components. Furthermore, the Design Code is currently under review to incorporate the requirements under the new EN standards. With harmonisation of the Design Code with the new EN standards, the trade could reduce the effort on assessing the compliance of models of lifts on conforming to the requirements for installation in Hong Kong.

Moreover, EMSD will continue to encourage RPs to engage independent REs to resolve occasional disputes between RPs and their appointed RCs, which may arise from the poor performance of the concerned RCs, queries on the needs for repairing/improvement works suggested by the RCs, occurrence of incidents etc., by carrying out independent audit inspections of the concerned lifts and escalators and provision of professional advice accordingly.

Finally and yet importantly, EMSD will continue to act as a "Facilitator" and "Promotor" in encouraging wider use of new technologies in the industry. Promotion of adopting new technologies will not only be confined in lift and escalator maintenance works, but will also be extended to apprentice and safety training in the lift and escalator trade. With the view of achieving quality and safe lift and escalator services, EMSD will continue to collaborate with different stakeholders to explore the possibility of wider use of new technologies in the industry so as to cope with the challenge of increasing demand for manpower resources and make the operation of lifts and escalators more comfortable, safe and reliable.

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[17] "Lift Modernisation Resource Corner": https://www.emsd.gov.hk/en/lifts_and_escalators_safety/responsible_persons_corner/lift_modernisation_resource_corner/index.html

BIOGRAPHICAL DETAILS

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LUI Graham is an Electrical and Mechanical Engineer of the Government of the HKSAR with engineering experience in building services design, operation and maintenance as well as regulatory enforcement on lift and escalator safety in Hong Kong. Currently working as the Engineer of General Legislation Division of EMSD, he is responsible for enforcement of the Lifts and Escalators Ordinance (Cap. 618) in Hong Kong. Mr. LUI is a Chartered Engineer registered with the Engineering Council of the United Kingdom and a Member of the Institution of Mechanical Engineers.

Robots, a New Class of Passenger

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Keywords: Lifts, robots, traffic handling.

Abstract. Lifts are increasingly being called upon to transport robots between floors in multi-story buildings. The robots that are presently available place special demands on lifts and those demands affect traffic handling. The special demands are explained and the impact of those demands on waiting time and transit time are reviewed using simulation.

1 INTRODUCTION

There are two principal types of robots; Industrial Robots and Service Robots. Industrial robots are used for manufacturing [1]. A Service Robot is defined by the International Organization for Standardization (ISO) as a robot "that performs useful tasks for humans or equipment excluding industrial automation applications" [2].

Service Robots are further subdivided into three categories by the International Federation of Robotics (IFR) [3]:

1. Professional service robots
2. Service robots for domestic/household tasks
3. Service robots for entertainment

Service robots for domestic/household tasks are such things as robotic carpet cleaners. Service robots for entertainment are essentially toys. Therefore, this paper addresses only professional service robots.

While there are many types of professional service robots, including some that milk cows and others that have military applications, there is a subset of professional service robots that will ride in lifts with humans.

There are four building types where professional service robots are being applied in increasing numbers. These building types are hotels, office buildings, residential buildings, and hospitals.

In hotels, service robots are being used for room service deliveries and for the delivery and movement of housekeeping carts.

In office, buildings robots are delivering packages and mail from sources outside the building as well as interoffice correspondence.

The growth of e-commerce has caused an increase of package deliveries to multi-story residential buildings. Robots are being used to make the final delivery from the lobby to the residential unit.

Hospitals have used pneumatic tube systems for delivery of medicines and the transport of medical records. Pneumatic tube manufacturers are now offering robots as an alternative to tubes [4].

2 ROBOT CHARECTERISTICS THAT AFFECT LIFTS

2.1 Mass & Size

There are two basic types of professional service robots that are manufactured by several manufacturers that need to use lifts to be effective. One type is a high mass vehicle that is referred to as a tug by one manufacturer [5]. The other type of robot is a low mass unit used for room service [6].

The tug type of robot has a payload of 453 kg. The tug and its lead acid batteries also have significant mass. It is 1164 mm long and 570 mm wide. It has a turning envelope of 1270 mm. The turning envelope is a circle with a diameter of 1270 mm. The area of this envelop is 1.27 m^2.

The room service robot has a total mass, including payload, of 50 kg. This robot is cylindrical and its diameter is 500mm. The turning envelope is also 500 mm which makes the area of the envelope 0.2 m^2.

The room service robot is designed to ride in the lift with human passengers. The mass, size, and turning envelop of the tug type robot needs a much larger lift. Consideration should be given limiting the access of a tug type robot to service lifts.

2.2 Velocity

Both types of robots have velocities of 0.76 m/s. Humans walking to a lift can be assumed to have a walking speed of 1.0 m/s [7]. Walking speed is a component of loading time.

2.3 Kinetic energy

Kinetic Energy is defined by the following equation [8]:

$$KE = \frac{1}{2}mv^2 \tag{1}$$

Where: KE represents Kinetic Energy

m represents mass

v represents velocity

The ASME A17.1 code has a kinetic energy limit and velocity limit for Dumbwaiters with Automatic Transfer Devices [9]. An automatic transfer device is defined as “a power-operated mechanism that automatically moves a load consisting of a cart, tote box, pallet, wheeled vehicle, box, or other similar object from and/or to the car”. The kinetic energy limit is 40 J and the velocity limit is 0.5 m/s during unloading. These limitations do not apply directly to robots riding with human passengers as passengers do not ride in dumbwaiters. However, they give some guidance in the lack of a robot specific standard.

In the case of a tug type robot with a total mass of 600 kg (453 kg payload and 147 kg estimated tug mass), the loading speed would need to be reduced 0.365 m/s during loading and unloading to comply with the 40 J limitation.

A room service robot operating at 0.76 m/s does not exceed the 40J limit. However, it does exceed the 0.5 m/s velocity maximum. Therefore, the maximum velocity during loading and unloading should be limited to 0.5 m/s.

Based on kinetic energy and velocity limits, the loading and unloading times for robots should be greater than the loading and unloading time for humans.

Data needs to be gathered by observing loading operations with robots operating at these speeds. In the absence of this data it would seem logical to add 1.2 seconds to both loading and unloading times when assessing the impact of robots on traffic handling performance. These increased times will increase waiting and transit times.

2.4 Personal Space

Personal Space, as it relates to lifts, defines the number of passengers that will ride in a lift at one time. CIBSE Guide D suggests that an adult male will occupy a space of 0.21 m^2 [7]. This is based on the male being a European or North American and does not include personal space.

Barney proposes that the design capacity of a lift should be based upon approximately 0.263 m^2 per person [10].

Personal space is space between humans. Robots are not humans. There is some research that indicates humans do not view robots as social entities (humans) [11]. The spatial distance that humans will require with robots is probably affected by the same factors that affect spatial distance between humans such as age and culture.

Research is needed in this area. However, in the absence of data, a starting point might be to add a buffer zone of 150 mm around a room service robot. For example, a cylindrical robot with a diameter of 500 mm would occupy, 0.5 m^2 (space diameter of 800 mm).

Personal space for tug type robots should be different because they will need large lifts due to their mass, physical dimensions, and turning envelopes. Tug robots will not be considered in the traffic section of this paper.

3 TRAFFIC EFFECTS

Room service robots are being installed in an existing 4 Star hotel in California. A study of the impact of these robots on passenger traffic was conducted. The following are the building, lift system, and passenger characteristics used for this traffic study.

Building:
- Floors: 25
- Rooms: 510, Located on Levels 4 – 25
- Dispatch Lobby: Level 1
- Occupancy: 440 Guests based on 86% occupancy and 1 person per room.

Lift System:
Number of Cars: 4
- Capacity: 1600 kg
- Speed: 2.5 m/s
- Dispatch Algorithm: Estimated Time of Arrival (ETA).
- Door Type: Center Opening
- Door width: 1066 mm.

Car Loading: 60% by Volume

Passengers:
Humans:
- Loading Time: 1.2 s
- Unloading Time: 1.2 s

Area: .26m²

Robots (room service type):

Loading Time: 2.4s

Unloading time: 2.4 s

Area: 0.5 m²

Traffic Template: Peters (CIBSE) Hotel

Figure 1 below graphically represents the Passenger Demand levels of the Peters (CIBSE) Hotel traffic when applied to the occupancy of the subject hotel.

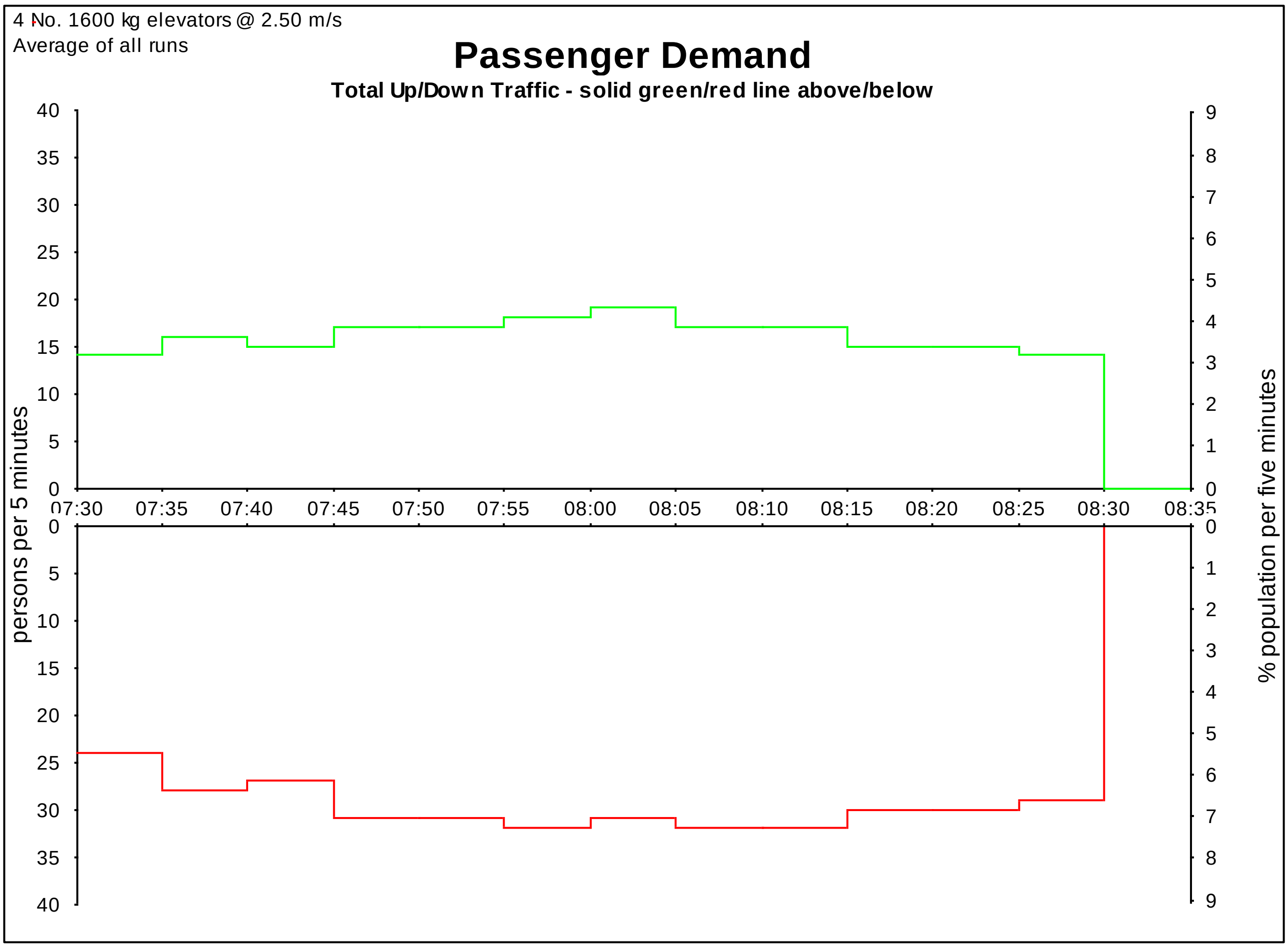

Figure 1 Passenger Demand, Peters (CIBSE) Hotel Template

Figures 2 and 3 show the results of traffic simulations using a Peters (CIBSE) Hotel Template with no robots installed. Figure 2 shows Waiting Times while Figure 3 shows Transit Times.

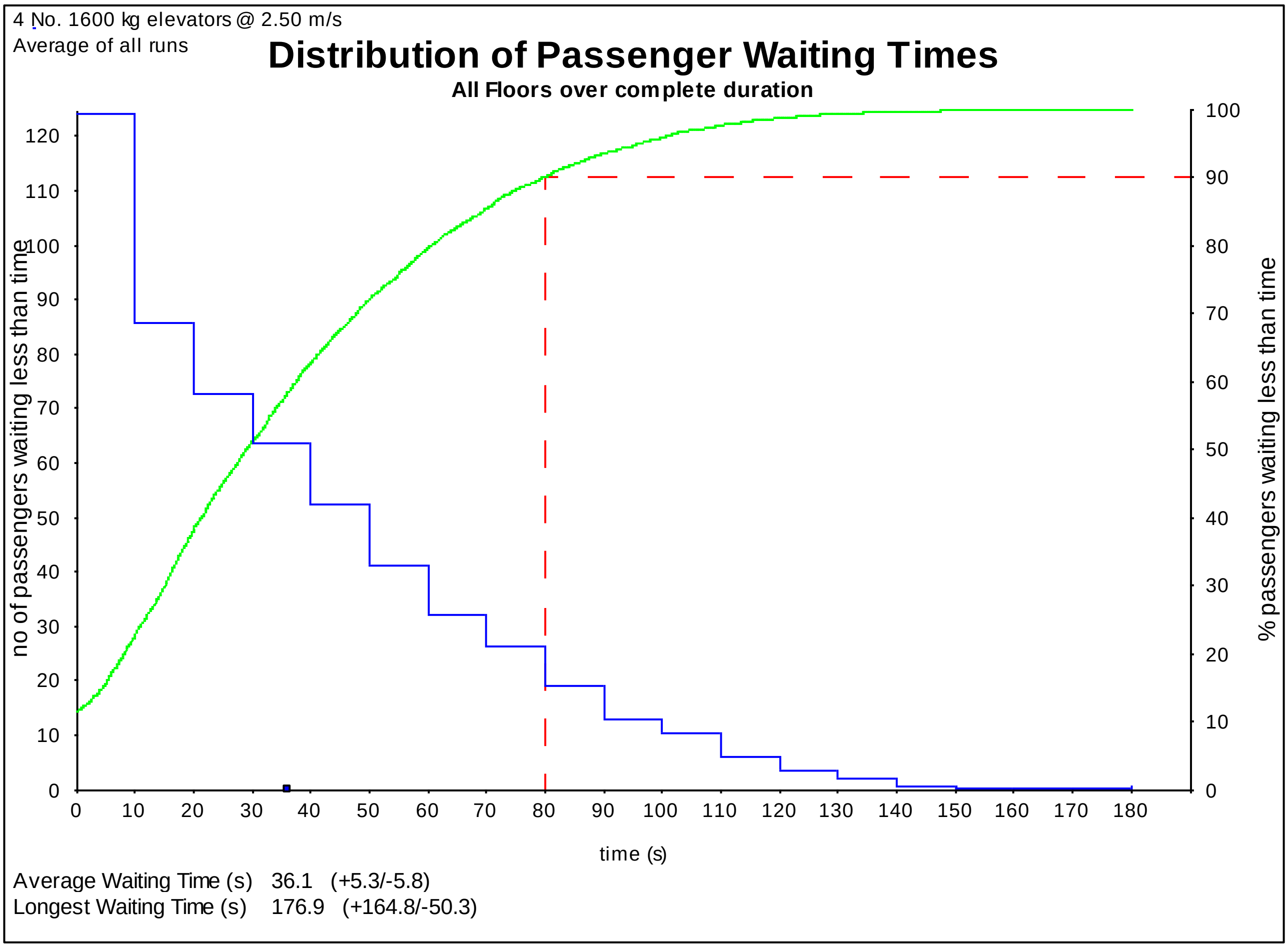

Figure 2 Waiting times without robots

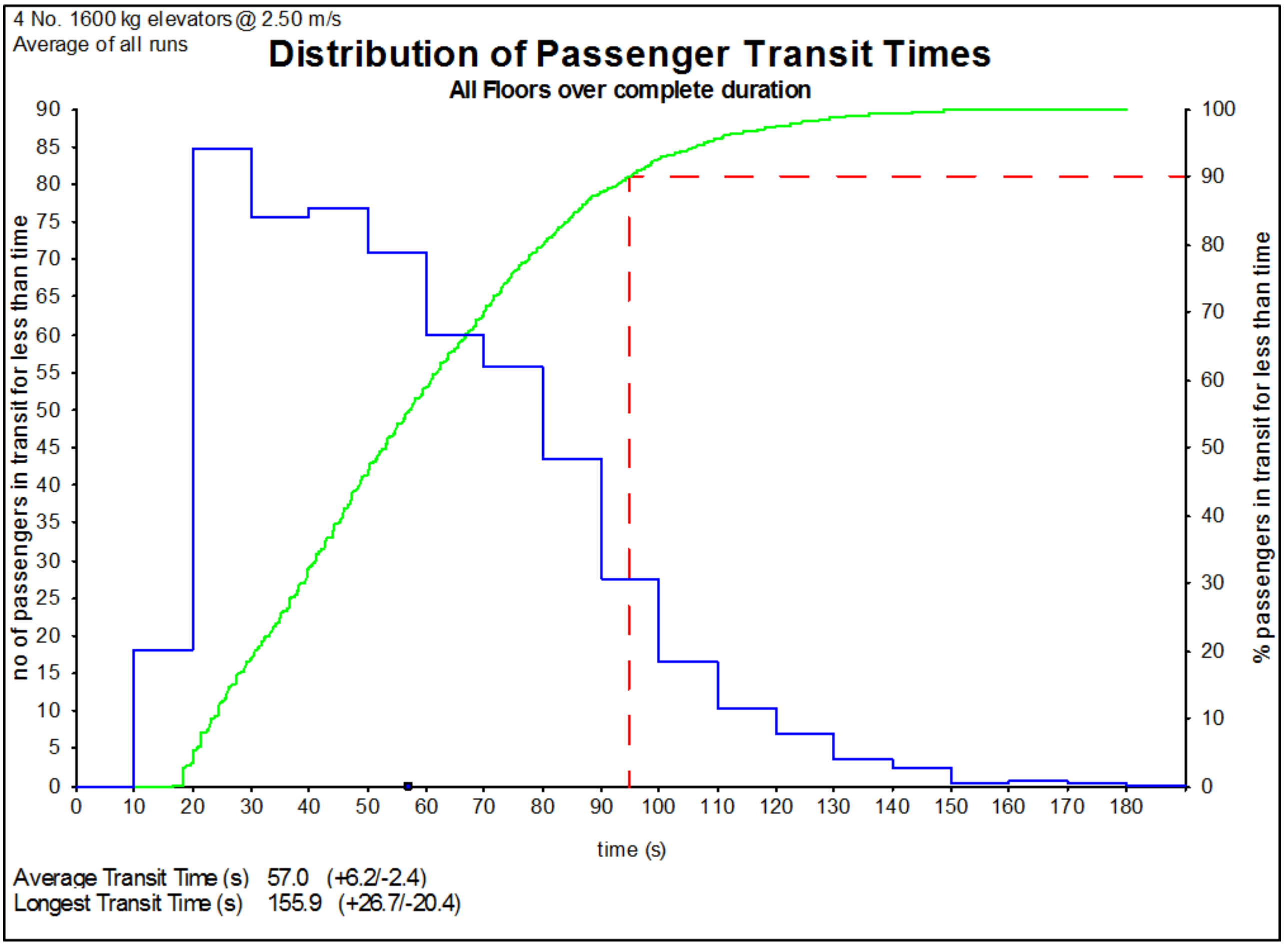

Figure 3 Transit times without robots

Figures 4 and 5 show the results of traffic simulations using a Peters (CIBSE) Hotel Template with one additional person making a room service delivery. The one person makes one delivery round trip every 5minutes. Figure 4 shows Waiting times while Figure 5 shows Transit Times.

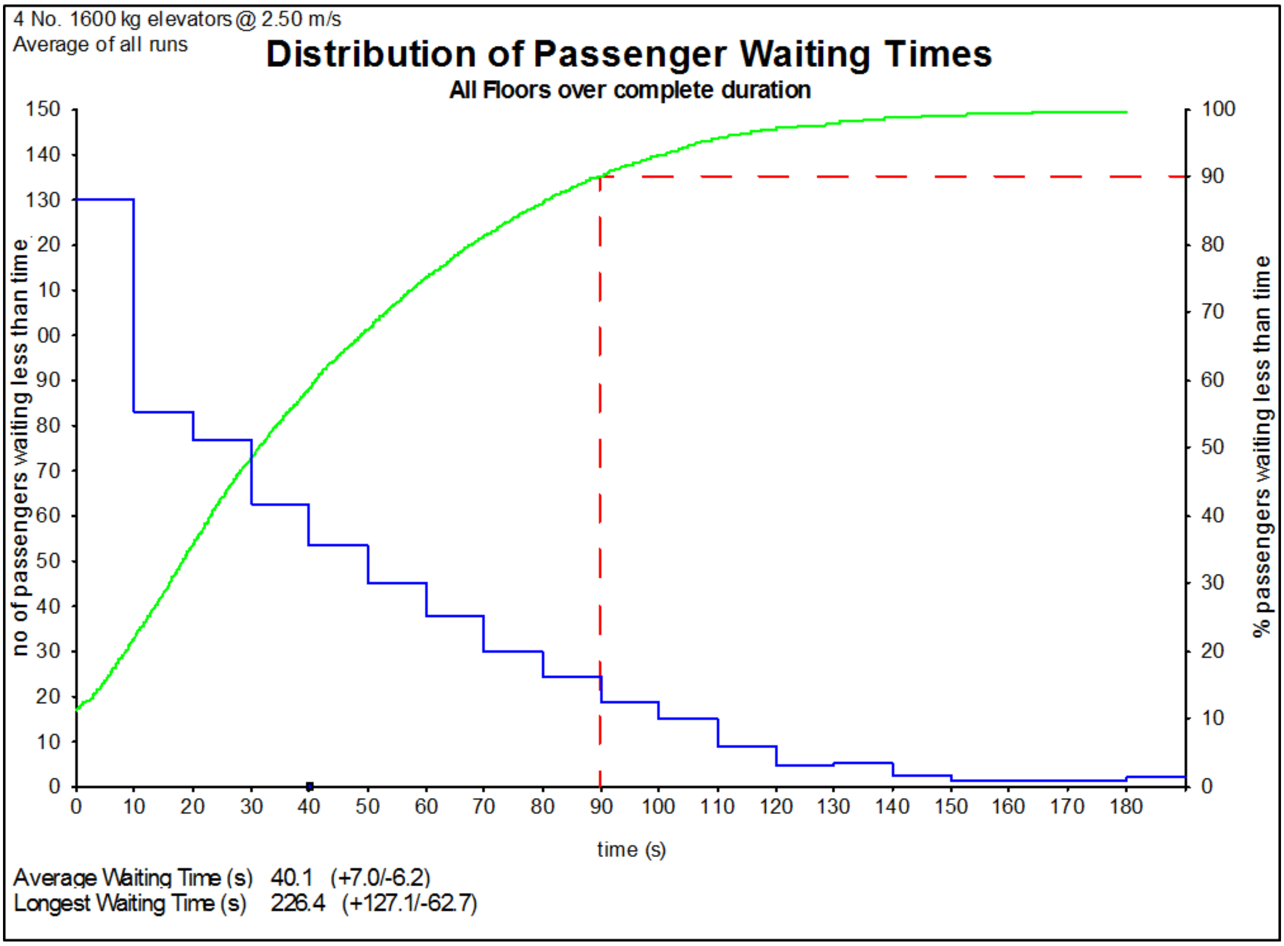

Figure 4 Waiting times with one additional human passenger

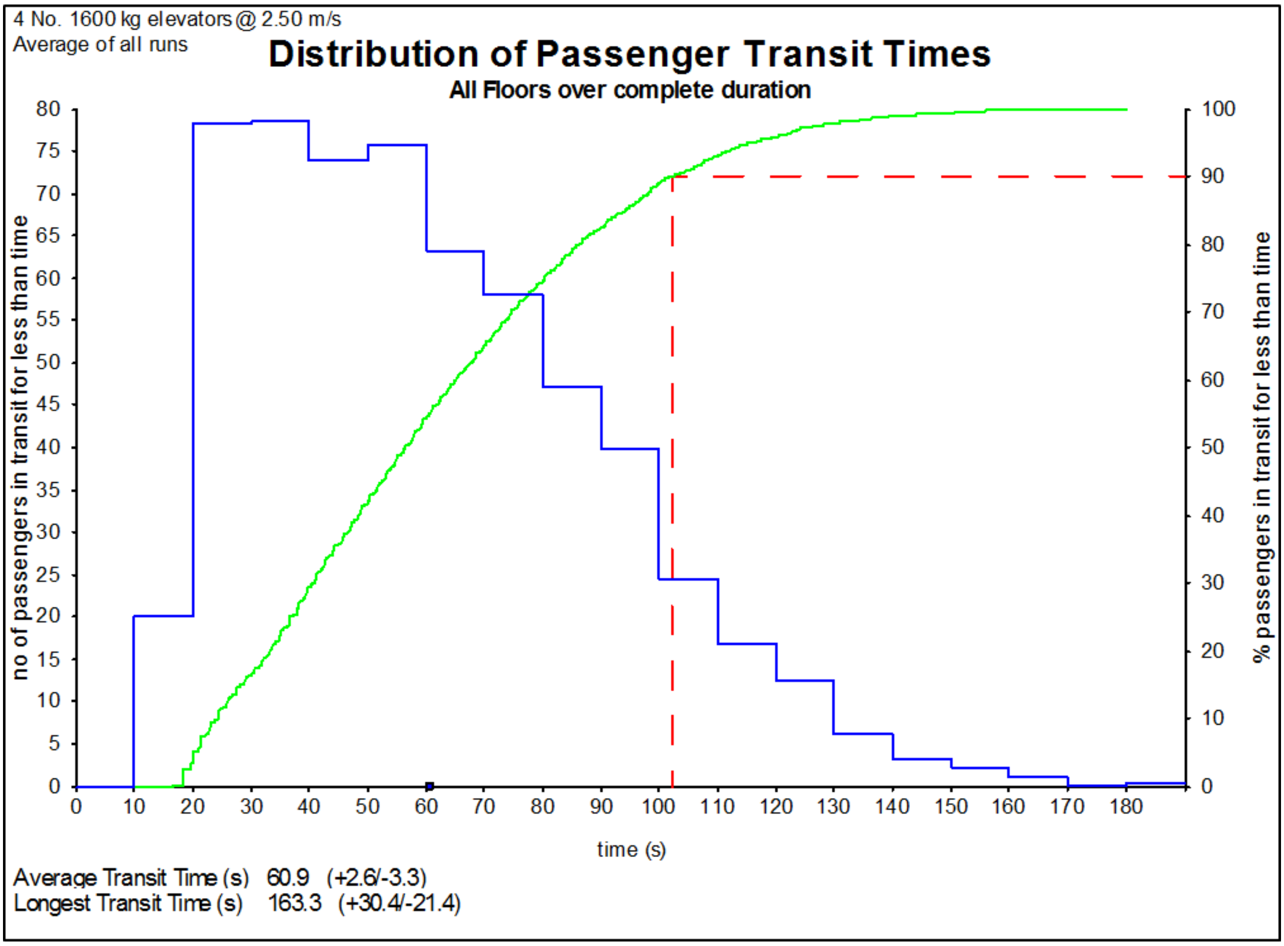

Figure 5 Transit times with one additional human passenger

Figures 6 and 7 show the results of traffic simulations using a Peters (CIBSE) Hotel template with one robot making a room service delivery. The one robot makes one delivery round trip every 5minutes. Figure 6 shows waiting times while Figure 7 shows transit times.

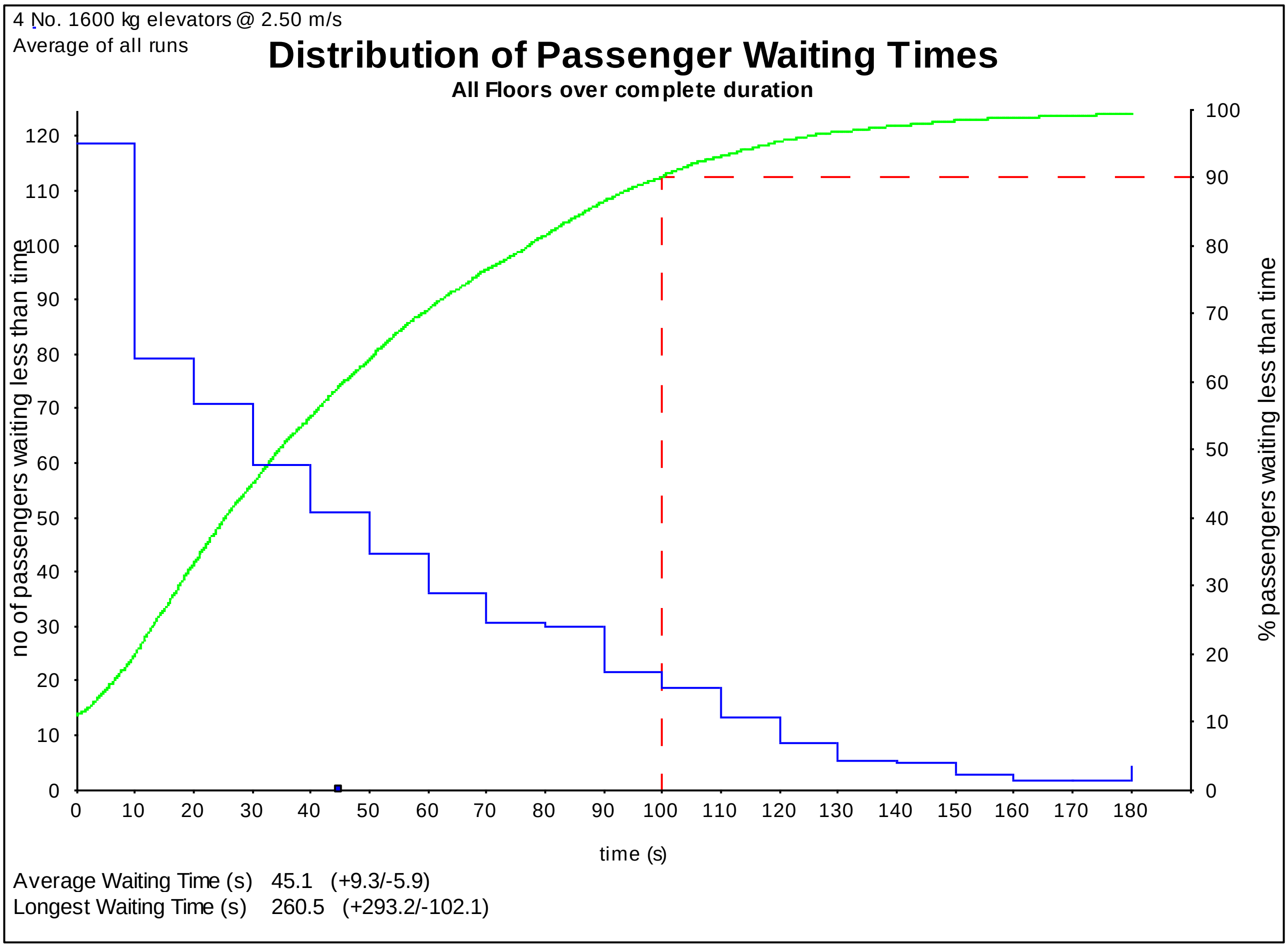

Figure 6 Waiting times with one robot

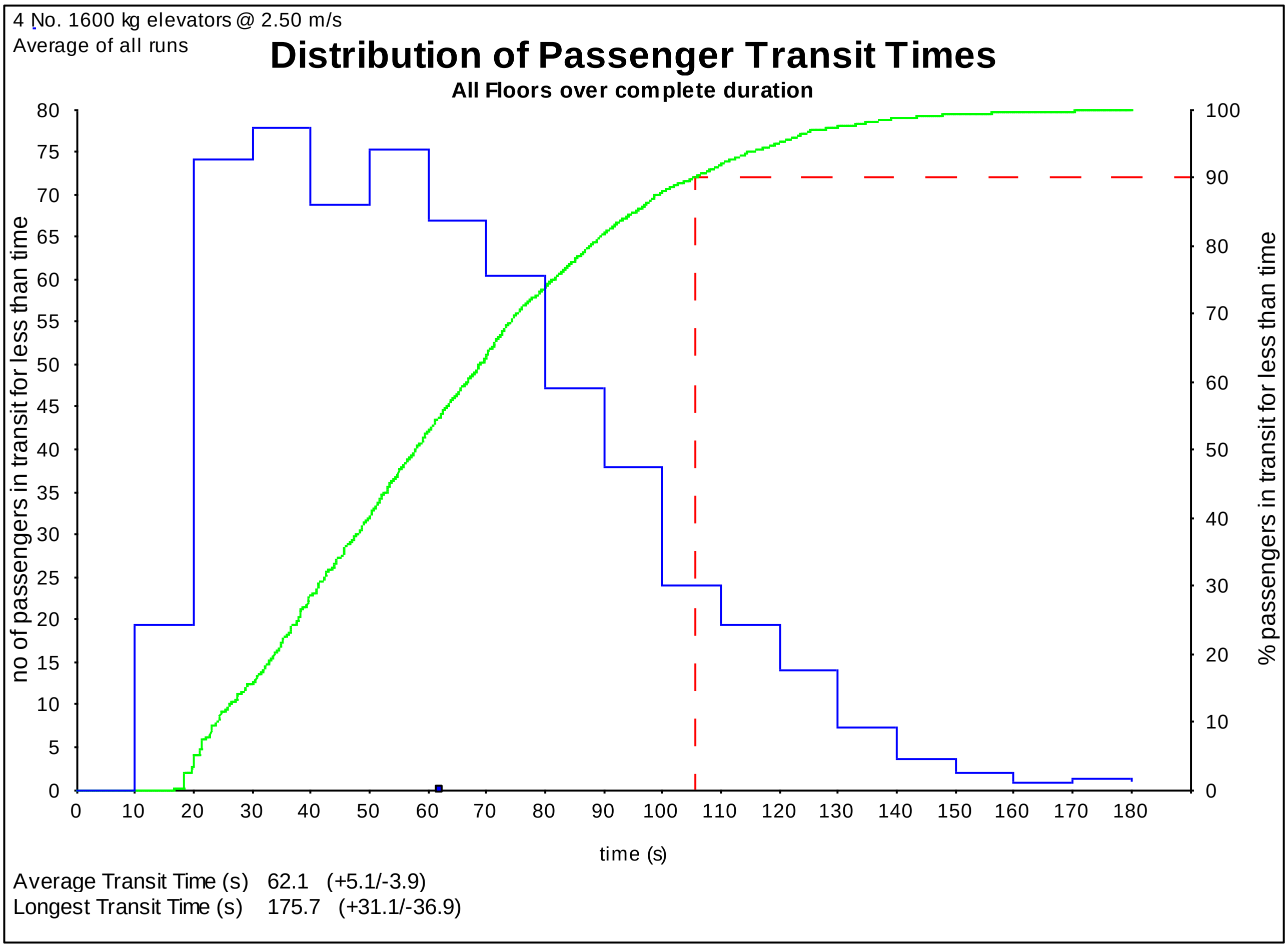

Figure 7 Transit times with one robot

Table 1 summarizes these results.

Table 1 Waiting Times, Transit Times and Times to Destination

	Waiting Time	Transit Time	Time to Destination
No Robots	36.1s	57.0s	93.1s
With 1 additional person	40.1s	60.9s	101.0s
With 1 robot	45.1s	62.1s	107.2s
Additional time for 1 robot compared to 1 person	**5.0s**	**1.2s**	**6.2s**

It should be noted that one robot will increase waiting time by 12.5% more than 1 person due to "personal space" and loading/unloading time.

4 CONCLUSIONS

Robots should be considered as a new class of passenger because they do not yet behave in the same manner as humans.

At present, their speed must be controlled during loading and unloading to either control kinetic energy or velocity. With continuing improvements in machine vision, robots may become better at avoiding collisions with people and property than humans and these restraints could be relaxed.

The physical shape of robots is not the same as humans and so the floor space they occupy is different from humans.

More research is needed to understand the human interaction with robots as passengers. We need to know how much personal separation humans need between themselves and robots.

The speed and personal space characteristics of robots have a negative effect on the traffic handling capacity of lift systems. If robots are planned for a new building, the proposed lift system should be designed acknowledging these effects. If robots are proposed for an existing building, a traffic study should be performed that demonstrates the impact of the robots.

The number of robot installations is growing. Lift traffic consultants need to understand the impact of robots on traffic so they can properly advise their clients.

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BIOGRAPHICAL DETAILS

Rory Smith is Visiting Professor in Lift Technology at the University of Northampton. He has over 49 years of lift industry experience during which he held positions in research and development, manufacturing, installation, service, modernization, and sales. His areas of special interest are Robotics, Machine Learning, Traffic Analysis, dispatching algorithms, and ride quality. Numerous patents have been awarded for his work.

Stress Analysis of Machine Supporting Beam System for Large Tonnage Lift

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Keywords: large tonnage lift, machine supporting beam, stress calculation, Finite Element Analysis

Abstract. With the development of social economy and industry, people's demands for automation, batch production and high efficiency become higher which also stimulate the development of lift industry. Some kinds of lifts have higher requirements for carrying capacity and safety, so the large tonnage lift came into being in large manufacture factories. The distance between the supporting points of the machine supporting beam is wider. During the lift's operation, the deflection and the stress of the machine supporting beam are larger, so a reasonable machine supporting beam structure system is needed. Design of large tonnage lift is beyond the conventional standard of professional design, normally has to meet some special requirements according to the requirements of customers. In this paper, the layout and the stress analysis of the machine supporting beam system for traction machine of large tonnage lift, as well as the corresponding finite element analysis of important load-bearing parts were performed and the checks also carried on, finally the design and material selection of the machine supporting beam system are determined and practically applied.

1 INTRODUCTION

Generally, the common large tonnage lift is the freight lift. It has a rated capacity of 4~15 tons. The «China GB 7588-2003 lift manufacture and installation safety norm» as well as the «GB 25856-2010 freight lift manufacture and installation safety norm» are referred to give dual attention to parameter requirements that the non-specialized operation security and equipment itself needs to satisfy on large tonnage lift [1,2]. The traction machine is an essential component of the lift driving device, and the machine supporting beam of the traction machine is an important supporting component. Due to the structural features of the lift, the machine supporting beam is a load-bearing steel beam supporting the weight of lift car, counterweight and other equipment. The load-bearing beam not only supports the traction machine but also the whole lift. The car, load, counterweight, traction sheave, cable, hoist rope and so on are hung on the load-bearing beam through the traction machine. It is really "a beam of a beam", which bears all static and dynamic loads of the lift. It is generally fixed on the civil load-bearing beam at the top of the lift shaft, and the material used is the channel steel or I-beam. For large tonnage lifts, the role of the machine supporting beam system is self-evident and has irreplaceable effect in ensuring the service life, running safety and comfort of the lift. The bearing beam of overhead type transmission is arranged in the machine room.

2 PRIMARY DESIGN OF MACHINE SUPPORTING BEAM

The roping method of the lift depends mainly on the position of the traction unit, the rated load and the rated speed of the car. In selecting and determining the way of rope winding, it is necessary to consider the high transmission efficiency, reasonable energy consumption and the extension of the service life of the wire rope. Different roping methods have different transmission speed ratios, called traction ratio. It is the ratio of the tangential velocity of the traction sheave to the velocity of the car. According to roping arrangement and traction ratio, the traction lifts can be classified as roped 1:1 or multiple reeving systems. The roping method of the steel wire rope is closely related to the lift machine room type. The rated capacity of the lift is 15 tons, the rated operating speed is 0.5m/s, and the guide rail type is the symmetrical arrangement of two guide rails. After comprehensive consideration, select the sheave roping method with traction ratio 8:1.

The deadweight of the car is 14 tons, the deadweight of counterweight is 21 tons, the traction machine is 1 ton, the rated capacity is 15 tons, and the size of the car is 4000 x 6000 x 3500 according to the standard. The machine supporting beam system is the focus of this design, including the counterweight return sheave beam, diverting pulley beam, the auxiliary machine supporting beam, the middle supporting beam, and the main machine supporting beam. The whole machine supporting beam system is shown in Fig. 1. A schematic showing the roping arrangement of traction wire is shown in Fig. 2.

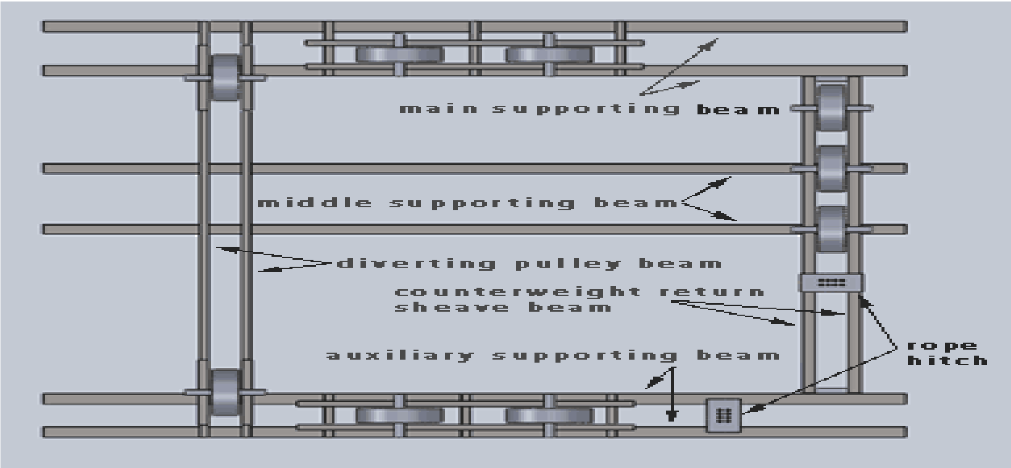

Figure 1 Layout of whole machine supporting beam system

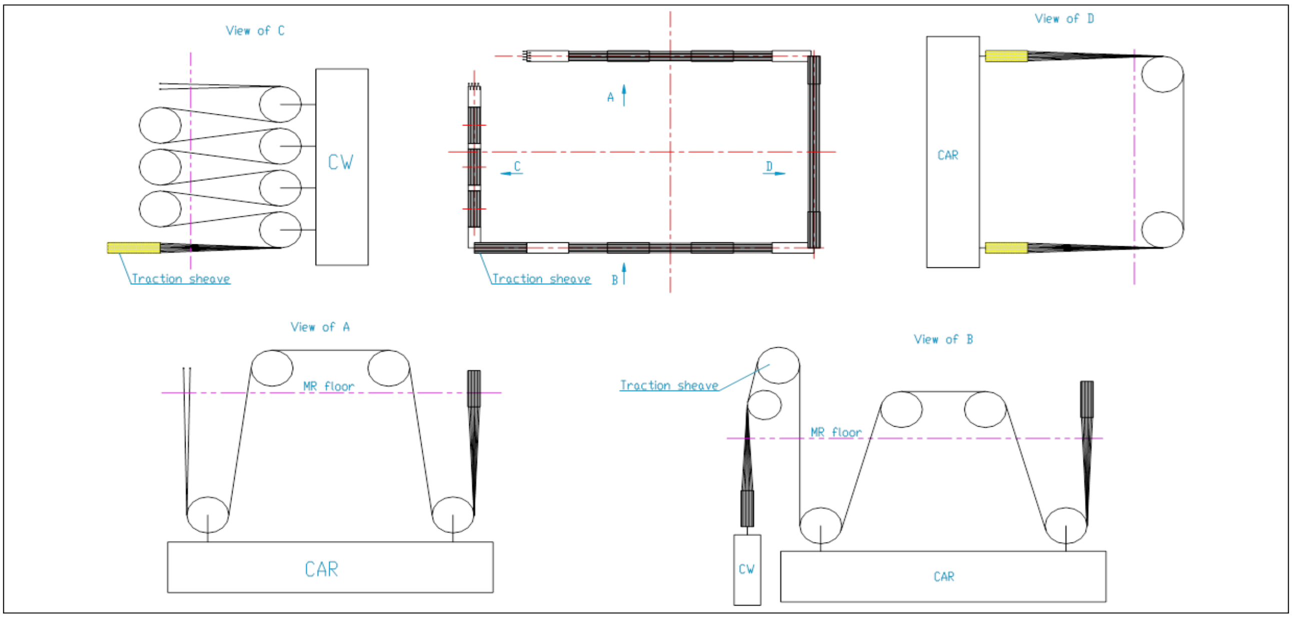

Figure 2 Schematic showing the roping arrangement

3 DETERMINATION OF BEAM SYSTEM PLANE POSITION AND INSTALLATION

Depending on the position for steel wire rope hole in the machine room, the location of the machine supporting beam system can be determined. The position of the traction machine's girder is determined according to the established standard line of the well plane layout, connecting the line of the car center to the counterweight center and the position of the bolt hole of the machine chassis.

Because the traction machine beam is the main bearing part of the lift, the length of the beam should be long enough to avoid the on-site welding.

The installation of the traction machine beam will depend on the different running speeds, the traction mode, the top layer height of the shaft, the sound insulation layer, machine room height, and the plane layout of all parts inside the machine room. Then, consider the specification and interaction distance of the machine supporting beam. Different installation methods can be determined according to these factors. After installation, the traction rope and machine supporting beam are not allowed to have friction when the lift is running. It is necessary to ensure that the traction rope operates smoothly to ensure the safe operation of the lift.

When installing the traction machine beam system, China GB/T10060-2011 "lift installation and acceptance specification"[3] must be followed: the traction machine supporting beam imbedded in the load-bearing wall, its supporting length should exceed the center of wall thickness 20mm, and should not be less than 75mm. For a brick wall, a reinforced concrete or metal beam, which can bear its load, should be placed under the beam to prevent the insufficiency, such as concrete and brick cracking and fragmentation under the support of the traction machine girder (I-beam), as this would not effectively guarantee the long-term safe operation of the lift.

The unevenness of each beam should not be more than 0.5/1000mm. The allowable height difference of two adjacent machine supporting beams should not be greater than 0.5mm, and the parallel deviation between them should not be greater than 6mm. The total equality is based on the connection line between the car and the counterweight center, to avoid causing the verticality of the traction sheave to not meet design requirements, as if the traction rope tension is improperly adjusted, it can result in traction rope groove jumping[4,5].

The common I-steel machine supporting beam is not directly placed on the floors or brick walls. This is to prevent mechanical and electromagnetic vibrations caused by the traction machine, which would lead to floor vibrations, affecting the residents nearby [6].

The loader beam embedded into the load-bearing wall is a hidden project. Before plugging, the quality of the beam installation is checked according to the requirements of GB50310-2002 "acceptance specification for construction quality of Lift Engineering"[7].

4 STRESS ANALYSIS AND CHECK

The whole machine supporting beam system includes the counterweight beam stress calculation, the diverting pulley beam stress calculation, the auxiliary machine supporting beam stress calculation, the middle supporting beam stress calculation, the main machine girder stress calculation.

4.1 Stress calculation of counterweight beam

First, take the counterweight return sheave beam as the simply supported beam; consider the most unfavorable conditions, namely all stress points being in the central span of girder [8].As shown in Fig.3, F_M: Support reaction force of the CW beam on the supporting point of the main supporting beam; F_V: Support reaction force of the CW beam on the supporting point of the auxiliary supporting beam; F_{CC}: Support reaction force of the CW beam on the 1# middle supporting beam; F_{CD}: Support reaction force of the CW beam on the 2# middle supporting beam; F_{C1}: Load force on the 1# pulley block on the CW beam5250kg=the weight of CW/4; F_{C2}: Load force on the 2# pulley block on the CW beam5250kg=the weight of CW/4; F_{C3}: Load force on the 3# pulley block on the CW beam5250kg=the weight of CW/4; F_{C4}: Load force on CW rope head 2625kg=the weight of CW/8； Lc: The CW beam spacing between the supporting points on the main supporting beam and the auxiliary supporting beam4550mm; Lc1: The CW beam spacing between the supporting points on the main supporting beam and the 1# pulley block on the CW beam 520mm； Lc2: The CW beam spacing between the supporting points on the main supporting beam and the 2# pulley block on the CW beam 1360mm; Lc3: The CW beam spacing between the supporting points on the main supporting beam and the 3# pulley block on the CW beam 2200mm ; Lc4: The CW beam spacing between the supporting points on the main supporting beam and the rope head plate on the CW beam 2940mm.

The supporting point of the CW beam on the 1# middle supporting beam coincides with the 2# pulley block on the CW beam. The supporting point of the CW beam on the 2# middle support beam coincides with the 3# pulley block on the CW beam.

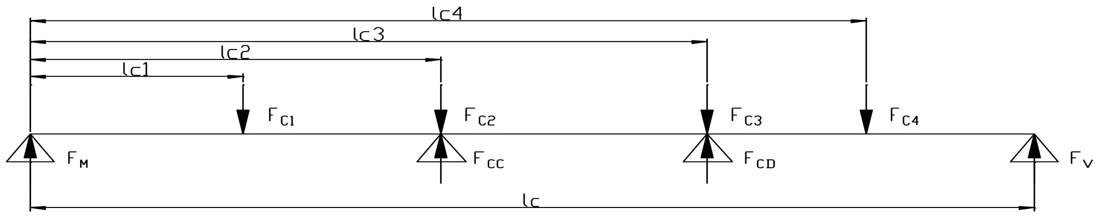

Figure 3 Force diagram of counterweight beam

From the force equilibrium equation,

$$F_{C1} + F_{C2} + F_{C3} + F_{C4} = F_M + F_V + F_{CC} + F_{CD}$$

From the moment equilibrium equation,

$$F_{C1} \times lc1 + F_{C2} \times lc2 + F_{C3} \times lc3 + F_{C4} \times lc4 = F_V \times lc + F_{CC} \times lc2 + F_{CD} \times lc3$$

$$v_{AB}(F_{C1}, lc2) + v_{AB}(F_{C2}, lc2) + v_{AB}(F_{C3}, lc2) + v_{AB}(F_{C4}, lc2) + v_{AB}(F_{CC}, lc2) + v_{AB}(F_{CD}, lc2) = 0$$

$$v_{AB}(F_{C1}, lc3) + v_{AB}(F_{C2}, lc3) + v_{AB}(F_{C3}, lc3) + v_{AB}(F_{C4}, lc3) + v_{AB}(F_{CC}, lc3) + v_{AB}(F_{CD}, lc3) = 0$$

$v_{AB}(F_{Cx}, lcx)$ denotes: simply supported beam AB, under the action of force F_{Cx}, the deflection of beam at L_{CX} point. In order to simplify the calculation, it is assumed that the deflection displacement of the CW beam is 0 on the supporting points of the 1# and the 2# middle supporting beam.

M_{C1}=F_M*lc1

M_{C2}=F_M*lc2-F_{C1}*(lc2-lc1)

M_{C3}=F_M*lc3-F_{C1}*(lc3-lc1)- F_{C2}*(lc3-lc2)+ F_{CC}*(lc3-lc2)

M_{C4}=F_M*lc4-F_{C1}*(lc4-lc1)- F_{C2}*(lc4-lc2)- F_{C3}*(lc4-lc3)+ F_{CC}*(lc4-lc2)+ F_{CD}*(lc4-lc3)

These support reaction forces can be obtained: F_M=-2819kg, F_V=-507.5kg, F_{CC}=-7474.7kg, F_{CD}=-7573.9kg, similar method can be used to solve the bending moment at other supporting points. M_{C1}= 1465804.81 kg*mm, M_{C2}= -576357 kg*mm , M_{C3}= -749770.12kg*mm , M_{C4}= 0.00008kg*mm. Generally, I-beam or channel steel is chosen for this kind of beam, and the cross section can be selected by comparing the parameter table of channel steel or I-beam. Select the material Q235 first, its yield strength is 235Mpa, choose allowable safety factor, general 3 ~ 4, according to the maximum bending moment and safety factor, the minimum section bending coefficient required by the beam can be obtained. Due to the installation space of pulley, 32a# channel steel is used for CW beam. The cross section coefficient of the beam is 950cm^3; the maximum stress of the beam is 15.43Mpa. We can obtain the bending moment diagram on the CW beam as shown in Fig.4, deflection of CW beam as shown in Fig.5, the minimum deflection of CW beam is -0.1mm.

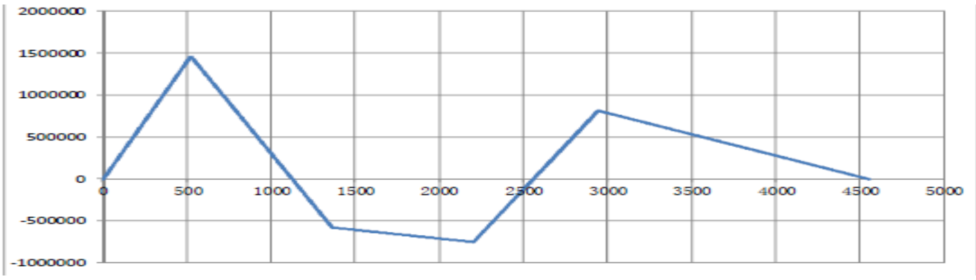

Figure 4Bending moment diagram of counterweight beam

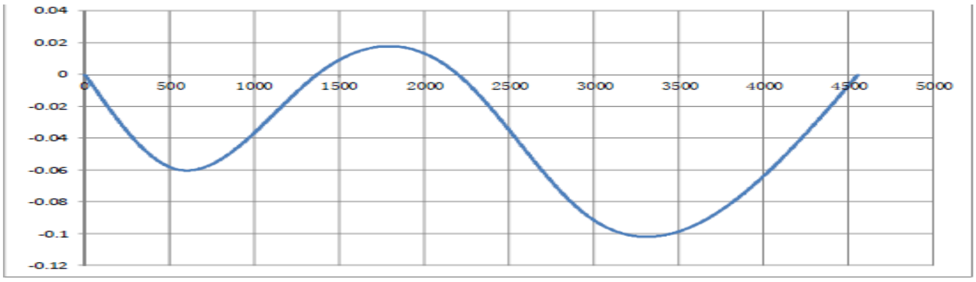

Figure 5 Deflection of counterweight beam

4.2 Stress calculation of middle support beam

Figure 6 Force diagram of middle support beam

The middle supporting beam is simplified as a simply supported beam.
F_{ZA}: support reaction force of the left support of the middle support beam ;
F_{ZB}: support reaction force of the right support of the middle support beam ;
F_{Z1}: Load force of CW beam applied to the middle support beam 7575kg（calculate the maximum of two loads） ;
lA: The length between the left and right support points of the middle support beam, 6300mm ;
lz1: The length between the support point of the middle support beam on the CW beam and the middle support beam left support, 5850mm ;
lz2: The length between the support point of the middle support beam on the CW beam and the middle support beam right support, 450mm。
$F_{ZA}=F_{Z1}*lz2/lA$
$F_{ZB}=F_{Z1}*lz1/lA$
$M_{Z1}=F_{ZB}*lz2$
These support reaction forces can be obtained: F_{ZA}=-541.1kg, F_{ZB} =-7034 kg, the bending moment at the supporting point be solved: M_{Z1}= 3165267.69 kg*mm. Select the material Q235 first, its yield strength is 235Mpa, choose allowable safety factor, general 3 ~ 4, according to the maximum bending moment and safety factor, the minimum section bending coefficient required by the beam can be obtained. 32a# I-steel is used for the middle support beam. The cross section coefficient of the beam is 692 cm^3; the maximum stress of the beam is 45.74Mpa. We can obtain the bending moment diagram on the middle support beam as shown in Fig.7, the deflection of middle support beam as shown in Fig.8, the minimum deflection of middle support beam is -3.7mm.

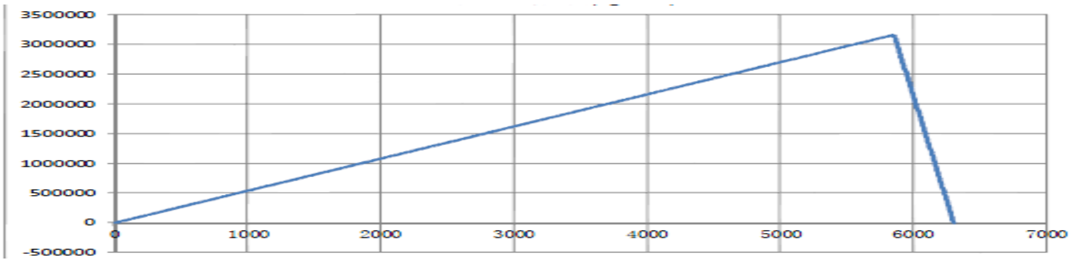

Figure 7Bending moment diagram of middle support beam

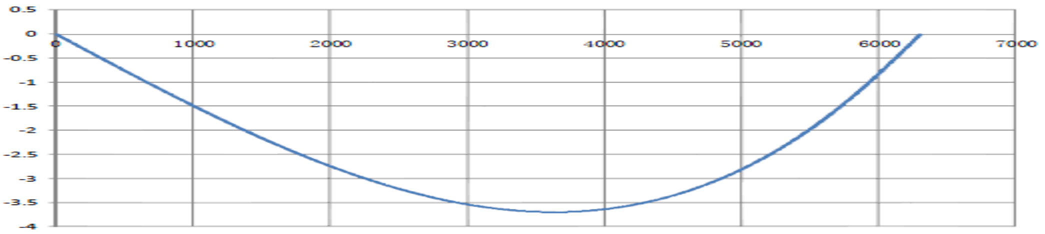

Figure 8 Deflection of middle support beam

4.3 Stress calculation of main supporting beam

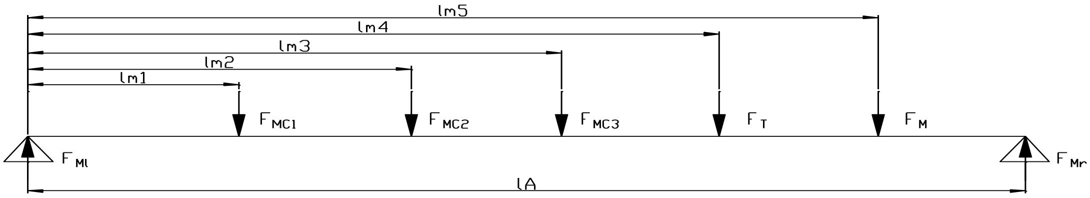

Figure 9 Force diagram of main support beam

F_{Ml}: support reaction force of the left support of the main support beam ;
F_{Mr}: support reaction force of the right support of the main support beam ;
F_{MC1}: Loading of diverting pulley beam on the main support beam, 3625kg ;
F_{MC2}: loading of diverting pulley frame left support point on the main support beam, 2360kg ;
F_{MC3}: loading of diverting pulley frame right support point on the main support beam, 2360kg ;
F_T: Loading of traction machine on the main support beam, 5000kg ;
F_M: Load of CW beam applied to the main support beam, 2820kg ;
lA: The length between the left and right support points of the main support beam, 6300mm ;
lm1: The length between the support point of the diverting pulley beam on the main support beam and the main support beam left support, 1350mm ;
lm2: The length between the left support point of the diverting pulley frame on the main support beam and the main support beam left support, 2410mm ;
lm3: The length between the right support point of the diverting pulley frame on the main support beam and the main support beam left support, 3990mm ;
lm4: The length between traction machine center and main girder left support, 5450mm ;
lm5: The length between the support point of the CW beam on the main support beam and the main support beam left support, 5850mm ;
In the two main support beams, the 1# main girder is close to the car center, the 2# main girder is far from the car center. Among them, the 1# main girder is subjected to greater force, and the above parameters and the following checking calculations are for the 1# main girder.

$F_{Ml}=(F_{MC1}*lm1+ F_{MC2}*lm2+ F_{MC3}*lm3+ F_T*lm4+F_M*lm5)/lA$

$F_{Mr}=F_{MC1} + F_{MC2} + F_{MC3} + F_T +F_M- F_{Ml}$

$M_{MC1}=F_{Ml}*lm1$
$M_{MC2}=F_{Ml}*lm2-F_{MC1}*(lm2-lm1)$
$M_{MC3}=F_{Ml}*lm3-F_{MC1}*(lm3-lm1)- F_{MC2}*(lm3-lm2)$
$M_T=F_{Ml}*lm4-F_{MC1}*(lm4-lm1)- F_{MC2}*(lm4-lm2)- F_{MC3}*(lm4-lm3)$
$M_M=F_{Ml}*lm5-F_{MC1}*(lm5-lm1)- F_{MC2}*(lm5-lm2)- F_{MC3}*(lm5-lm3)- F_T*(lm5-lm4)$

These support reaction forces can be obtained: F_{Ml} = -6047kg, F_{Mr} = -10118kg, the bending moment at the supporting points be solved: M_{MC1}=8163160.7kg*mm, M_{MC2}=10730254kg*mm, M_{MC3}= 10827874.94kg*mm, M_T=7000000kg*mm, M_M= 4553196.345kg*mm. Select the material Q345 first, its yield strength is 345Mpa, choose allowable safety factor, general 3~4, according to the maximum bending moment and safety factor, the minimum section bending coefficient required by the beam can be obtained. 40a# I-steel is used for the main support beam. The cross section coefficient of the beam is 1090cm^3; the maximum stress of the beam is 99.34Mpa. We can obtain the bending moment diagram on the main support beam as shown in Fig.10, the deflection of main support beam as shown in Fig.11; the minimum deflection of main support beam is -10.37 mm.

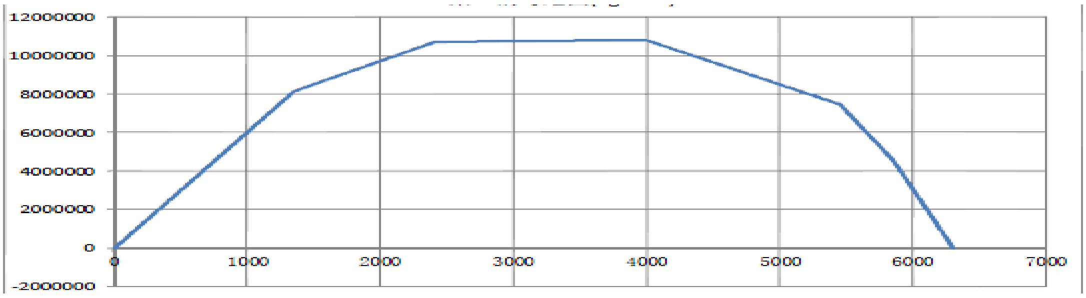

Figure 10 Bending moment diagram of main support beam

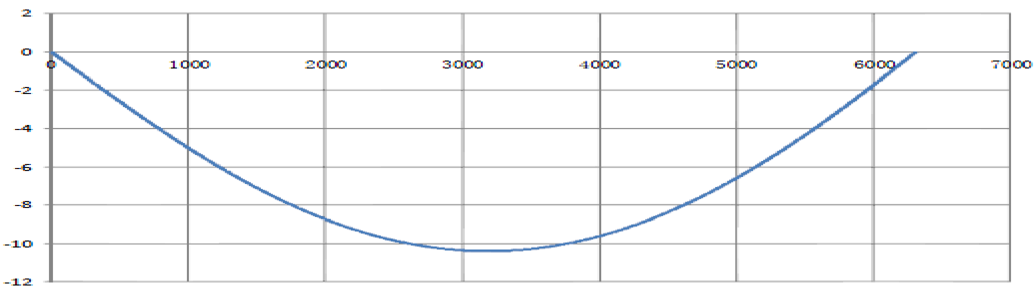

Figure 11 Deflection of main support beam

4.4 Stress calculation of auxiliary support beam

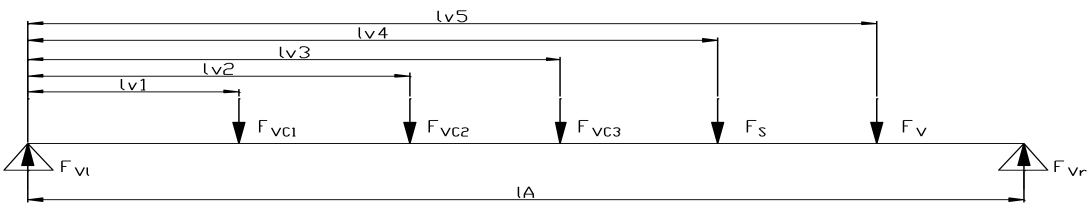

Figure 12 Force diagram of auxiliary support beam

F_{Vl}: Support reaction force of the left support of the auxiliary support beam;
F_{Vr}: Support reaction force of the right support of the auxiliary support beam；
F_{VC1}: Loading of diverting pulley beam on the auxiliary support beam, 3625kg；
F_{VC2}: Loading of left support point of diverting pulley frame on the auxiliary support beam, 1815kg；
F_{VC3}: Loading of right support point of diverting pulley frame on the auxiliary support beam,1815kg；
F_S: Load of the car rope head plate applied to the auxiliary beam, 1815kg；
F_V: Load of the CW beam applied to the auxiliary beam, 510kg；

lA: The length between the left and right support points of the auxiliary support beam,6300mm；
lv1: The length between the support point of the diverting pulley beam on the auxiliary support beam and the auxiliary support beam left support, 1350mm；
lv2: The length between the left support point of the diverting pulley frame on the auxiliary support beam and the auxiliary support beam left support, 2410mm；
lv3: The length between the right support point of the diverting pulley frame on the auxiliary support beam and the auxiliary support beam left support, 3990mm；
lv4: The length between traction machine center and auxiliary girder left support, 5050mm；
lv5: The length between the support point of the CW beam on the auxiliary support beam and the auxiliary support beam left support, 5850mm；
In the two auxiliary support beams, the 1# auxiliary girder is close to the car center, the 2# auxiliary girder is far from the car center. Among them, the 1# auxiliary girder is subjected to greater force, and the above parameters and the following checking calculations are for the 1# auxiliary girder.

$F_{Vl}=(F_{VC1}*lv1+ F_{VC2}*lv2+ F_{VC3}*lv3+ F_T*lv4+F_V*lv5)/lA$

$F_{Vr}=F_{VC1} + F_{VC2} + F_{VC3} + F_S +F_V- F_{Vl}$

$M_{VC1}=F_{Vl}*lv1$

$M_{VC2}=F_{Vl}*lv2-F_{VC1}*(lv2-lv1)$

$M_{VC3}=F_{Vl}*lv3-F_{VC1}*(lv3-lv1)- F_{VC2}*(lv3-lv2)$

$M_S=F_{Ml}*lv4-F_{VC1}*(lv4-lv1)- F_{VC2}*(lv4-lv2)- F_{VC3}*(lv4-lv3)$

$M_V=F_{Ml}*lv5-F_{VC1}*(lv5-lv1)- F_{VC2}*(lv5-lv2)- F_{VC3}*(lv5-lv3)- F_S*(lv5-lv4)$

These support reaction forces can be obtained: F_{Vl} = -5031kg, F_{Vr} = -10118kg, the bending moment at the supporting points be solved: M_{VC1}=6791785.74kg*mm, M_{VC2}=8282095kg*mm, M_{VC3}= 7635800.076kg*mm, M_S=5000000kg*mm, M_V = 2047071.54kg*mm. The material of auxiliary support beam is the same as main support beam (40a# I-steel), the allowable deflection, according to 1/500 of the beam span, the maximum stress of the beam is 75.98Mpa. We can obtain the bending moment diagram on the auxiliary support beam as shown in Fig.13, the deflection of auxiliary support beam as shown in Fig.14; the minimum deflection of auxiliary support beam is -7.59mm.

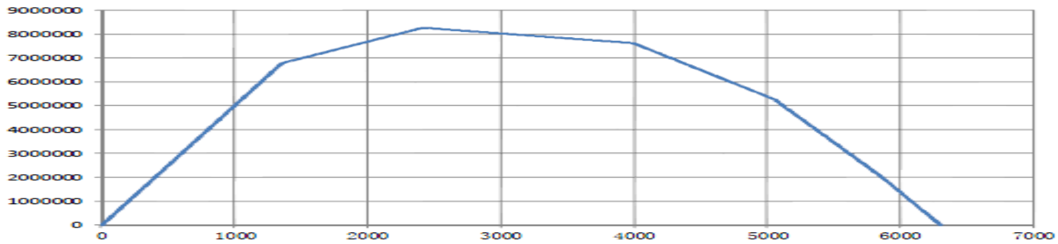

Figure 13 Bending moment diagram of auxiliary support beam

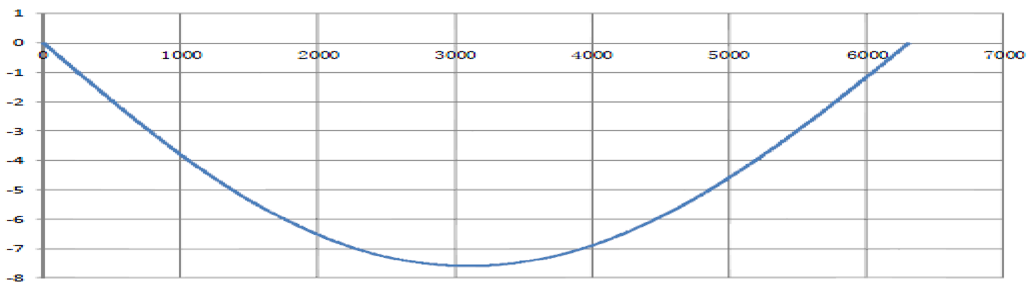

Figure 14 Deflection of auxiliary support beam

4.5 Finite element analysis results

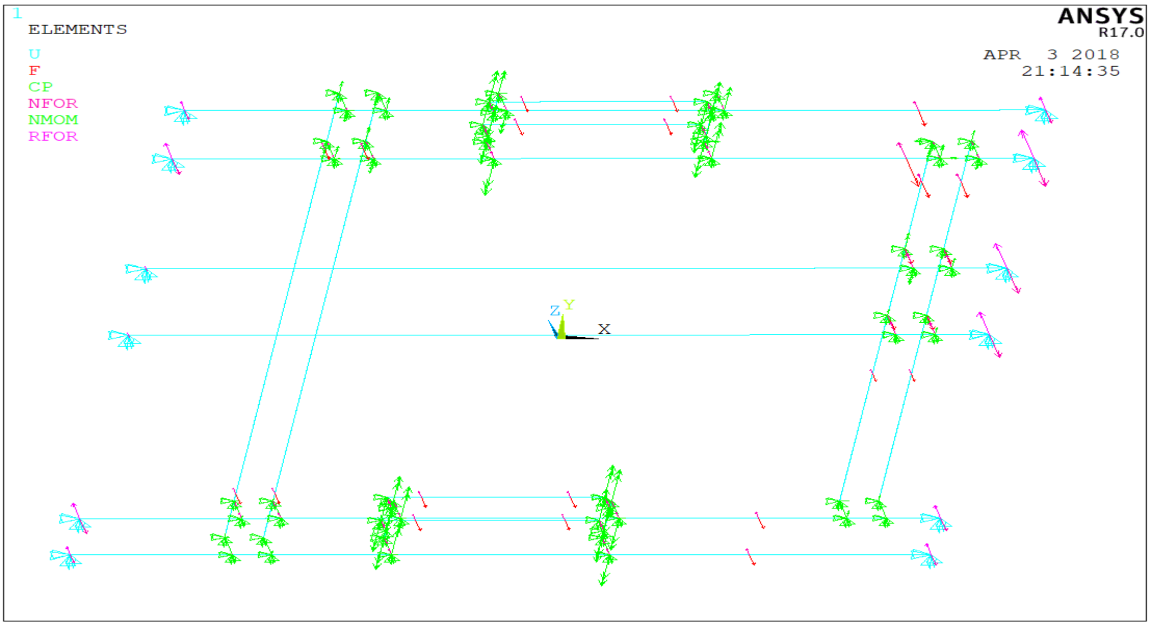

Figure 15 Simplified model of the machine supporting beam system

Figure 15 is a simplified model of the machine supporting beam. The beam is calculated by the Beam 189 element (a three-dimensional beam element in ANSYS).

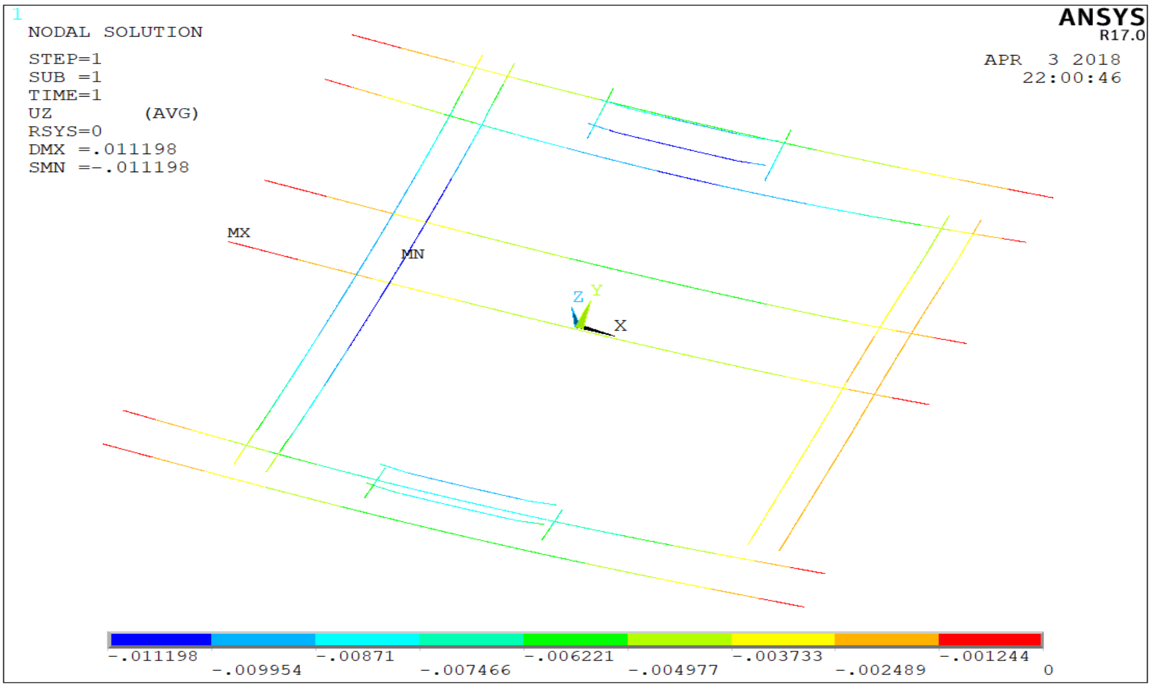

Figure 16 Deflection of Z direction

Figure 16 is the deflection of the machine supporting beam in the vertical direction when it is loaded. The maximum deflection of Z direction is 11.2mm, which occurs on the left diverting pulley beam. The maximum Z deflection on the main and auxiliary beam is 10mm.

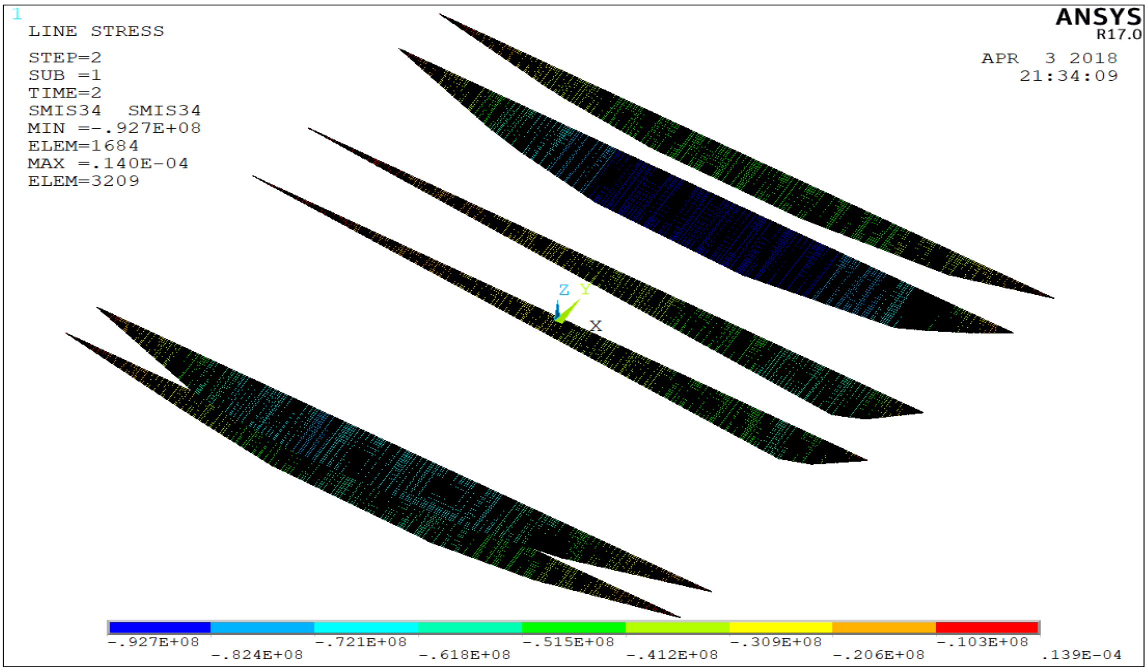

Figure 17 Bending stress on the beam system

Figure 17 is the bending stress for 3 groups of beams (main support beam, auxiliary support beam, middle support beam).

A table has been listed for comparing the key results from finite element analysis with the results obtained by theoretical calculation, including the worst stress and deflection at different locations of machine supporting beam system.

Table 1 comparison of the results of the two methods

Machine support beam system	Theoretical calculation		finite element analysis	
	Maximum bending stress	Maximum deflection	Maximum bending stress	Maximum deflection
Counterweight beam	15.43Mpa	0.1mm	20.3Mpa	2.7mm
Main support beam	99.34Mpa	10.37mm	92.7Mpa	10.04mm
Auxiliary support beam	75.98Mpa	7.59mm	73.1Mpa	7.89mm
Middle support beam	45.74Mpa	3.7mm	61Mpa	5.45mm

5 CONCLUSIONS

According to the machine supporting beam system design, the machine room layout,and the plan of hole position for wire roping, determine the position of diverting pulleys, and the position of rope fastening. The theoretical calculation is based on the actual layout size of an elevator. The main and auxiliary support beam select Q345, the other beams choose Q235, and their yield strength are 345 MPa and 235MPa respectively. The safety factor generally selects 3~4. The main and auxiliary support beam use 40a# I-steel, the counterweight beam and middle support beam use 32a# channel steel and I-steel. The stress and restriction are applied in ANSYS interface, and proven within the allowable stress range. Therefore, this can not only verify whether the material used is reasonable or not, but also can check the correctness of the theoretical calculation.

It can be seen from the table that the results obtained by the two methods are deviant. The two methods are simplified calculation methods, and there were deviations from the actual values; the deviation between the actual value and the theoretical calculation should be greater. In the calculation of the counterweight beam, the displacement of the four supporting points of the counterweight beam (the main support beam, the two middle support beams and the auxiliary support beam) is assumed to be 0, but in fact, the four supporting points of the counterweight beam will follow the displacement of the supporting points of the main support beam, the two middle support beams and the auxiliary beam and then shift. This is the biggest difference, which is the main reason for the deviation. In addition, the finite element is also calculated by a simplified model. In practice, there are total 12 constraints on both sides of the beam in addition to the constraints of the displacement (ux,uy,uz), also partial torsional deflection (rotx,roty,rotz) constraints. In addition, the traction machine and the machine support beam are in practice a surface contact, and only two force points are simplified in the theoretical calculation.

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BIOGRAPHICAL DETAILS

Dr.-Ing. Xiaomei Jiang now works for School of Mechanical Engineering, Changshu Institute of Technology & Jiangsu Key Laboratory of Elevator Intelligent Safety. She is an engineer with more than a decade of experience in engineering design. She graduated from Soochow University in 2009, received her doctorate and published more than 30 papers. She also hosts and participates in the completion of a number of scientific research projects.

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Systems Engineering Approach: Postgraduate Programme Bridging the Gap Between the Theory and Industrial Practice in Lift (Elevator) Engineering

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Keywords: Lift (Elevator) Engineering, Systems Engineering, Postgraduate Programme, Safety Standards, Design.

Abstract. The paper provides a comprehensive introduction to, and an appraisal of, a postgraduate Lift Engineering programme. The programme is based on systems engineering approach and has been designed to transfer the underpinning knowledge required for effective advanced engineering design, research and management in the lift (elevator) making and allied industries. The provision evolved from the distance learning programme which was originally developed following the introduction of the first edition of the European standard EN 81-1:1977. Some parts of the programme have been modified appropriately to reflect other national codes such as ASME/ANSI A17.1. The programme comprises the Masters - level course. The research degree programme offers then an opportunity for successful candidates to study towards PhD / MPhil. An example is discussed to illustrate how research-informed learning aids the solution of complex Lift Engineering design problems. The analysis demonstrates how practice, learning and research are integrated into the programme.

1 INTRODUCTION

In view of the present world-wide interest in the development of safe and cost-effective means of vertical transportation the importance of engineering education for technical staff employed within the (Lift) Elevator Industry cannot be overestimated. The principles underlying Lift Engineering involve a broad range of subjects including Electrical and Electronic, Mechanical, Civil and Production/ Manufacturing Engineering. A successful academic programme in Lift Engineering should therefore integrate those areas [1,2].

This paper presents an academic postgraduate programme that combine practice, learning and research in elevator engineering. The provision evolved from the distance learning programme which was originally developed following the introduction of the first edition of the European standard EN 81-1:1977. Some parts of the programme have been modified appropriately to reflect other national codes such as ASME/ANSI A17.1. The programme comprises the Masters - level postgraduate course which is outlined in the paper.

2 THE LIFT ENGINEERING PROGRAMME

2.1 MSc course structure and delivery

The Masters (MSc) course is composed of compulsory and elective/ designated taught modules, plus an independent, industry-based research study presented in the form of a dissertation [3]. The compulsory taught modules are concerned with Lift Applications Engineering, Codes and Standards and Management of Contracts, all of which are essential. Elective modules provide students with the opportunity to pursue their own specialization within the industry and currently include Lift Component Applications, Hydraulic Systems, Control Systems, Utilization of Materials, Dynamics and Vibrations and Vertical Transportation Systems.

The MSc is delivered in a distance learning regime. In this regime the emphasis is on learning rather than teaching. The tutorial team is staffed and supported by a combination of experienced educational practitioners, together with experienced practitioners drawn from the UK lift industry. The tutors fulfil the role of facilitators of learning. Furthermore, the acquisition of the skills of self-learning is a primary and specific aim of the provision. The tutorial team in collaboration with the lift industry has been involved with the design, development and operation of a Distance Learning course in Lift Technology since 1983. Thus, learning materials for the MSc course have been designed and are continuously revised for use by distance learning students building on and developing from the 35 years of operation of the distance learning provision.

2.2 MSc dissertation and research projects

Research projects form an integral part of the course and each candidate at his/her final stage of study is required to propose and justify a research topic as a subject of the dissertation. This involves the identification of research objectives, the selection of appropriate methods with regard to the research problem, the presentation of the research work plan and an initial review of relevant literature. Subsequently, after the proposal is accepted, the candidate manages his / her own time and activities to bring the project to a successful conclusion. The candidates have access to specialized literature and research resources at the University. The student maintains a chronological record of the work undertaken in pursuit of the project which is periodically submitted to his / her tutor. This forms an important element in compensating for the reduced face to face contact between student and tutor as compared with a similar, but full time student.

Over the last few years over seventy successful projects were completed and MSc dissertations submitted. They cover a broad range of topics and reflect both the students' interests and the industry needs demonstrating strong relationship between practice and theory across a number of elevator technology areas. Two book volumes with reviews of the MSc dissertations have been published by Elevator World [4,5]. The topics cover a broad range of problems such as the effect of building sway on elevator ropes, ventilation and passenger comfort, power comsumption, firefighting and evacuation, usage and utilizing lifts for the differently abled people, safety gear performance, code requirements for interfaces between building systems and elevator systems, accidents involving luggage trolleys and/or shopping carts on escalators .

2.3 Research degree programme

The research programme provides an opportunity for the MSc graduates to continue their studies towards higher research degrees (PhD/MPhil). The programme environment offers an opportunity for students to network with a variety of contacts through research seminars and conference events.

Each academic year commences with the annual Symposium on Lift and Escalator Technologies organized in conjunction with the Chartered Institution of Building Services Engineers (CIBSE) Lifts Group and the Lift and Escalator Industry Association (LEIA), This event provides opportunities for students, practitioners and engineers from industry and academia worldwide to network and discuss the latest training, education, research and innovation developments. The symposium event is now in its 9th edition and the next conference is taking place later this year, from 19th to 20th September 2018.

3 SELECTED TOPICS: LIFT BRAKING AND STOPPING

3.1 Electromechanical braking and emergency arrest

The issue of slowing and bringing the lift car safely to rest is one of the most important problems in the design of a lift installation. This problem is addressed in the MSc syllabus [3] in the context of the traction drive system and the relationships between braking, drive control and traction are comprehensively treated throughout the course learning materials.

This involves the electromechanical brake and the entire range of situations with which it might have to deal, including normal and emergency conditions, considering the interfaces and linkages between the brake and the control systems, and between the brake and the lift car. In accordance with EN 81-20:2014 [8], the electromechanical brake alone must be capable of stopping and holding 125% of the rated load. But even the most modern lift system will be required to stop under the action of electromechanical braking if there is an unconventional event such as the opening of the landing door whilst the lift is in motion, or an interruption of the power supply, for example [1].

However, the discussion of the issues above is predicated upon the assumption that the traction system remains intact and that deceleration of the system is achieved by a braking torque applied to the traction sheave. Thus, the dynamics of the stopping / arrest of the lift car are limited by the available traction. Therefore, it is necessary to investigate the ultimate systems for arresting uncontrolled motion by acting directly on the car.

The ultimate safety system to stop a lift car in the event of overspeed consists of the following elements:

- an overspeed governor set to trip at a pre-determined speed at least 115% of rated speed. An electrical trip should de-energise the drive and engage the electromechanical brake before the car speed, either up or down, reaches this tripping speed.
- a safety gear located on the car (and, in some circumstances, on the counterweight), which will arrest an uncontrolled overspeed in the down direction and
- a suitable device, such as a rope brake or sheave brake, which will arrest an overspeed in the up direction (or any unintended movement in either direction).

Fig. 1 shows the main components of a system for emergency arrest in the down direction [1]. An overspeed governor located in the machine room or in the upper part of the hoist way is connected to the safety gear system on the car by the governor rope. The governor rope is a complete loop, with both ends terminated on the safety gear system on the car, after passing around a loaded tensioning pulley in the pit. The two basic types of mechanical overspeed governor (rocking arm and pivoted bob-weight types) are shown in the diagram together with the three basic types of safety gear - instantaneous (type A) either cam type or captive roller, and progressive (type B). On the car, the governor rope is terminated at the top of the car and connected to the safety gear via a safety gear operating rod.

3.2 Lift car - safety gear performance analysis

Consider a simplified diagram of the car – safety gear interaction shown in Fig. 2(a). In the scenario considered here the car suspension failure is assumed. The car is represented by a rigid body of mass m acted upon by the safety gear braking force F_{sg}. If at the time instant t_1 the car has a speed of v_1 and at the time instant t_2 the speed is v_2 the application of the principle of work and energy [6] yields

$$\frac{1}{2}mv_1^2 + mgy_1 - F_{sg}\Delta y = \frac{1}{2}mv_2^2 + mgy_2 \tag{1}$$

where $\Delta y = y_1 - y_2$ is the distance travelled by the car when being slowed down by the safety gear actions and g is the acceleration of gravity (9.81 m/s^2).

Fig. 2(b) shows the results (velocity, position plots) of a drop test to examine the performance of a safety gear device to be installed in a lift car of mass m = 10207 kg [7]. In Fig. 2(c) the mean acceleration of the mass is shown. It is evident from the test results that during the test the free fall of the mass is arrested at the time instant t_1 ($\approx 2.1\,\text{s}$) and then over the time interval $\Delta t = t_2 - t_1 \approx 3.45 - 2.1 = 1.35\ \text{s}$ the car continues to descend at a near constant speed (of about 12.5

m/s). Thus, the braking force developed by the safety gear is of inadequate magnitude, and it is just large enough to balance the car weight ($F_{sg} \approx mg$). Thus, the safety gear needs to be re-designed.

The required braking force to decelerate the car from v = 12.5 m/s to rest can be determined from (1) by setting $v_2 = 0$ so that the following equation is obtained

$$\frac{1}{2}mv^2 + mg\Delta y - F_{sg}\Delta y = 0 \tag{2}$$

The braking force is then expressed as

$$F_{sg} = \frac{\frac{1}{2}mv^2 + mg\Delta y}{\Delta y} \tag{3}$$

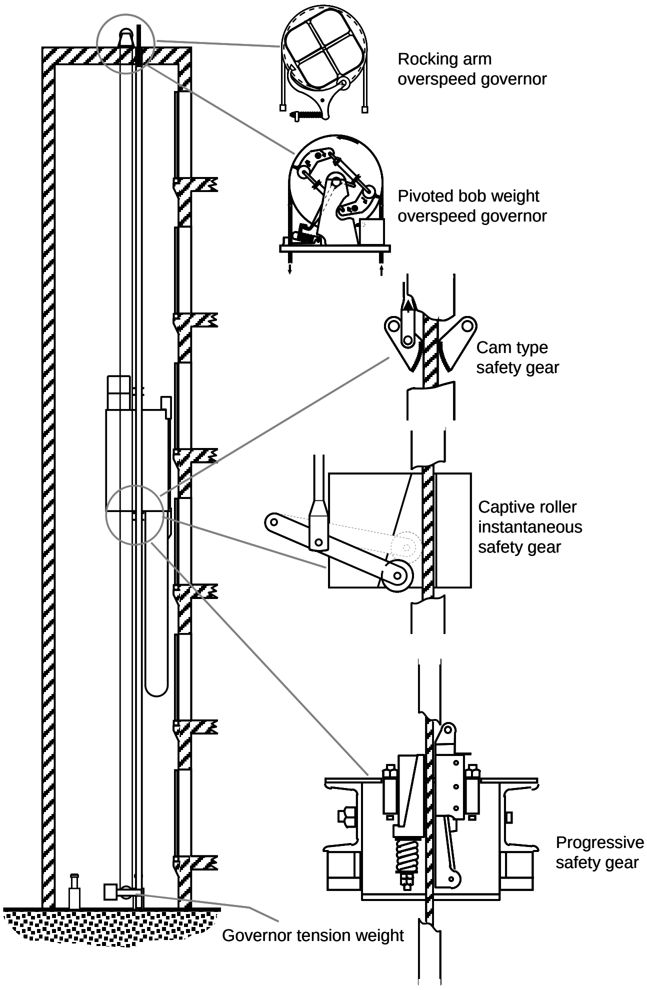
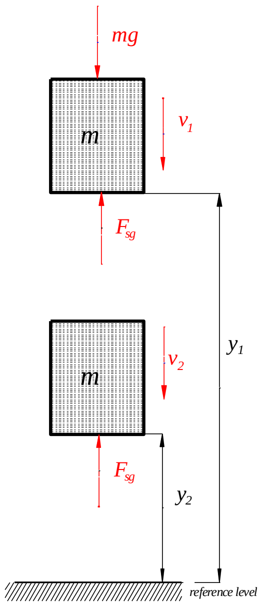
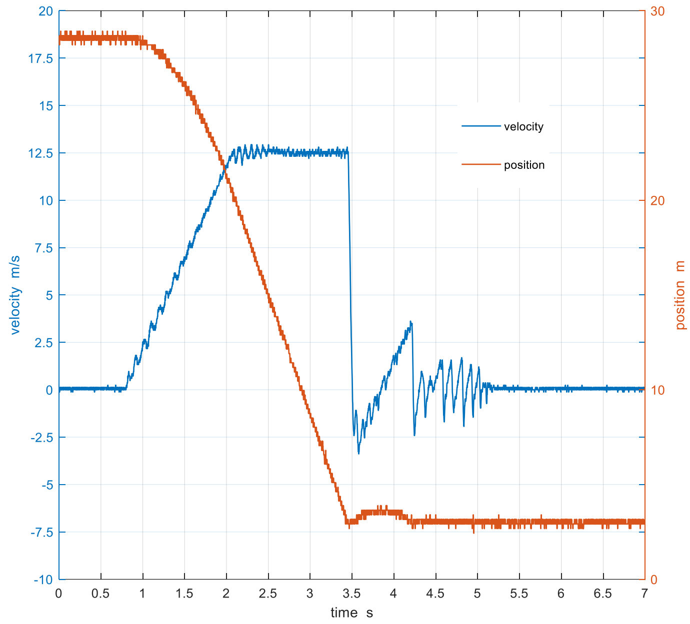


Figure 1 Safety gear – car system [1]

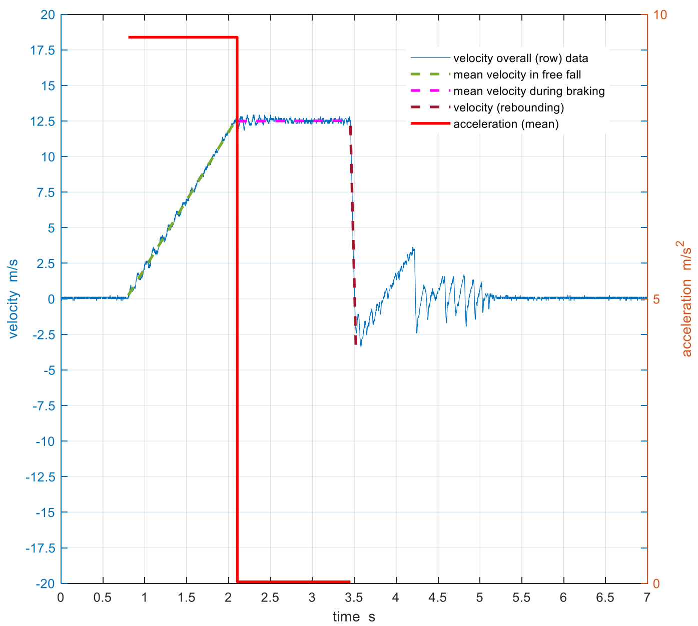
By using $\Delta y = v^2/2a$ in Eq. 3, where a denotes the deceleration rate, and setting the deceleration rate as a = 0.6g (as per the nominal deceleration requirement in EN 81-20 [8]) the required safety gear braking (friction) force is determined as 161.2 kN. During the test the braking force applied was about $F_{sg} \approx mg = 100.75$ kN. Thus, the required increase is significant and can be achieved by various means explored below.



(a)



(b)



(c)

Figure 2 Safety gear action (a) simplified model; (b) test results (c) velocity – acceleration plots

For a progressive (Type B) safety gear, the braking forces were investigated in one MSc dissertation [4,9] extracting data from drop test results of a family of safety gears and comparing these with the literature and other safety gears. The braking force is generated by the interaction of the braking surfaces (gibs) of the safety gear and the guide rail. To a first approximation, this braking force for a single safety gear can be modelled as:

$$F_{sg} = 2\mu R \tag{4}$$

where R is the reaction force between the braking surfaces and the guide rail and μ is the coefficient of friction between the sliding surfaces. The factor of 2 comes from the two pairs of surfaces in contact on a single guide rail. Thus, in looking to increase the braking force of the safety gear, there are two avenues to investigate:

- Increasing the coefficient of friction, μ. However, μ is determined by the selection of materials for the safety gear gibs and machined steel guide rails. A significant increase could be made by changing to materials used e.g. as used in automotive brake pads and by changing the design.
- Increasing the reaction force (generated by springs) consistent with the design of the safety gear e.g. limitations from the strength of the safety gear housing, heating of the braking surfaces, and avoiding excessive damage to the sliding surfaces which would tend to limit the reaction force used.

This is not to imply that the value of μ is constant. It has been documented at least as far back as 1865 that the coefficient of friction for railway brakes was lower at higher running speeds and was also dependant on the reaction force *R*. These dependencies were recognised in the lift literature and are implied by numerous progressive safety gear drop test results. The MSc dissertation studied this speed dependence where the variation of the coefficient of friction with rubbing speed for a single gib/ guide rail interaction was modelled as:

$$\mu = \mu_0 e^{-cRv} \tag{5}$$

where v is the sliding speed, μ_0 is the coefficient of friction when $v = 0$, *R* is as before, *c* is a constant. After analysing results, a more refined model was suggested.

This research project was typical of many MSc dissertation projects as it was a piece of research based on a study of application design so involved close collaboration with the industry and yielded not only a useful academic result but also a practical one; one of the outcomes of the study was that the safety gear design studied had its nominal load increased for use at lower tripping speeds.

4 CONCLUSIONS

The Lift Engineering provision has been developed to integrate three key elements: practice, learning and research. It forms a complete provision for lifelong learning bridging the gap between the theory and industrial practice. The provision involves a modular MSc level course taught through distance learning. The taught modules cover a broad range of areas relevant to the theory and practice in the field of lift technology. In order to progress to the dissertation stage a student must achieve a pass in each of the compulsory modules and the two elective modules (at the first or second attempt). Subsequently, the student is required to undertake advanced independent study leading to the MSc dissertation which is considered essential to achievement of the award. The course is tailored to the needs of those who are employed in the lift manufacturing and allied industries and is supported by the Lift and Escalator Industry Association. Flexible structure of the course and distance learning regime of study minimizes time away from work and benefits both the employer and the employee. Research project forms an integral part of the course and gives students an opportunity to conduct an independent study making use of the skills and knowledge acquired elsewhere in the course. The programme offers an opportunity for successful candidates to study towards PhD/MPhil research degrees.

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BIOGRAPHICAL DETAILS

Stefan Kaczmarczyk is Professor of Applied Mechanics and Postgraduate Programme Leader for Lift Engineering at the University of Northampton. His expertise is in the area of applied dynamics and vibration with particular applications to vertical transportation and material handling systems. He has been involved in collaborative research with a number of national and international partners and has an extensive track record in consulting and research in vertical transportation and lift engineering.

Rory Smith has over 49 years of experience in all aspects of the lift industry including sales, installation, maintenance, manufacturing, engineering, research & development. He has worked for ThyssenKrupp Elevator for the last 23 years. Prior to becoming involved in ThyssenKrupp's Internet of Things, he was Operations Director, ThyssenKrupp Elevator Middle East. His scientific interests include, operations management, high rise - high speed technology, ride quality, traffic analysis, dispatching. To date he has been awarded numerous patents in these areas and has many pending patents.

Nick Mellor has worked for the UK's Lift and Escalator Industry Association (LEIA) as Technical Director and Managing Director since 2012 and has been in the industry for 26 years. Nick was in the inaugural cohort of the MSc in Lift Engineering at Northampton. More recently, as an Associate Lecturer, he has done some tutoring on the MSc.

The Dynamic Interactions in High-Rise Vertical Transportation Systems

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Keywords: High-rise Lift (Elevator) system, Ride quality, Resonance, Suspension/ compensating ropes.

Abstract. High speed and high capacity vertical transportation (VT) installations in the modern built environment service buildings of nearly 1000 m tall. Tall buildings are susceptible to large sway motions when subjected to wind loading or earthquake excitations. The low frequency sway motions cause resonance interactions in lift car/ counterweight suspension system, compensating ropes and overspeed governor ropes. This leads to poor ride quality and a high level of dynamic stresses which may result in damage to the installation. This paper presents the systems engineering approach to predict and quantify transient and steady-state resonant vibrations taking place in high-rise lift applications. The results and conclusions presented in the paper demonstrate that a good understanding of the dynamic behaviour of VT systems is essential for developing design strategies that minimize the effects of adverse dynamic responses so that the installation will operate without compromising the structural integrity and safety standards.

1 INTRODUCTION

The design and operation of high-performance systems for passenger transportation in the modern high-rise built environment present many technical challenges due to adverse dynamic responses and interactions that often arise due to various sources of excitation present in these systems [1]. In the modern high-rise built environment traction drive lift (elevator) systems are used.

The performance of lift installations in tall towers and buildings can be substantially affected by the behaviour of the host structure adverse environmental phenomena [2,3]. Strong wind conditions and earthquakes cause tall buildings to vibrate (sway) at low frequencies and large amplitudes creating a base motion excitation mechanism acting upon all building non-structural components. When the host (building) structure sways a broad range of resonance phenomena occur in the lift system with large whirling motions of ropes and cables being developed that are coupled with vertical motions of the car, counterweight and compensating sheave and often result in damage caused by the impact against the lift equipment located in the shaft and/or against the shaft walls. Aerodynamic phenomena affect the performance of high-speed lift systems. At high speeds the air flow around the car – frame assembly induces excessive vibrations and noise [4,5]. During the lift travel large air pressure differences between the front and rear of the car are being generated. Furthermore, the aerodynamic effects due to multiple cars running in the same shaft should also be considered [6,7].

This paper reviews and presents a systematic analytical approach to predict and quantify the transient and steady-state resonant vibration phenomena taking place in high-rise lift systems due to of the host structure motions. It is demonstrated that a good understanding of the dynamic behaviour of the main components of the lift systems is essential for developing design strategies that minimize the effects of adverse dynamic responses so that the installation will operate without compromising the structural integrity and safety standards.

2 DYNAMICS OF A HIGH-RISE LIFT INSTALLATION

The traditional traction-driven high-rise lift installation comprises the lift car/ counterweight system driven by tractive forces developed between the traction shave and the suspension means - long slender continua (LSC) such as steel wire ropes (SWR) or light-weight composite ropes [7]. In this arrangement an additional set of ropes, tensioned by the weight of the compensating sheave, are used for the compensation of tensile forces over the traction sheave.

2.1 Dynamic model with base excitation due to the structure motion

A schematic diagram of the dynamic model of the lift system is shown in Fig. 1. The modulus elasticity, cross-sectional effective area and mass per unit length of the ropes are denoted as E_1, A_1, m_1 and E_2, A_2, m_2 for the compensating ropes and the suspension ropes, respectively. The compensating ropes are of length L_1 at the car side and the suspension ropes are of length L_2 at the counterweight side, respectively. The length of the suspension rope at the car side and the compensating rope at the counterweight side are denoted as L_3 and L_4, respectively. The lengths of suspension ropes and compensating cables are time-varying $L_i = L_i(t)$, $i = 1,\ldots,4$. The masses and dynamic displacements of the car, counterweight and the compensating sheave assembly are represented by M_{car}, M_{cwt} and M_{comp}, q_{M1}, q_{M2} and q_{M3}, respectively. The speed and acceleration/ deceleration of the car are denoted by V and a, respectively.

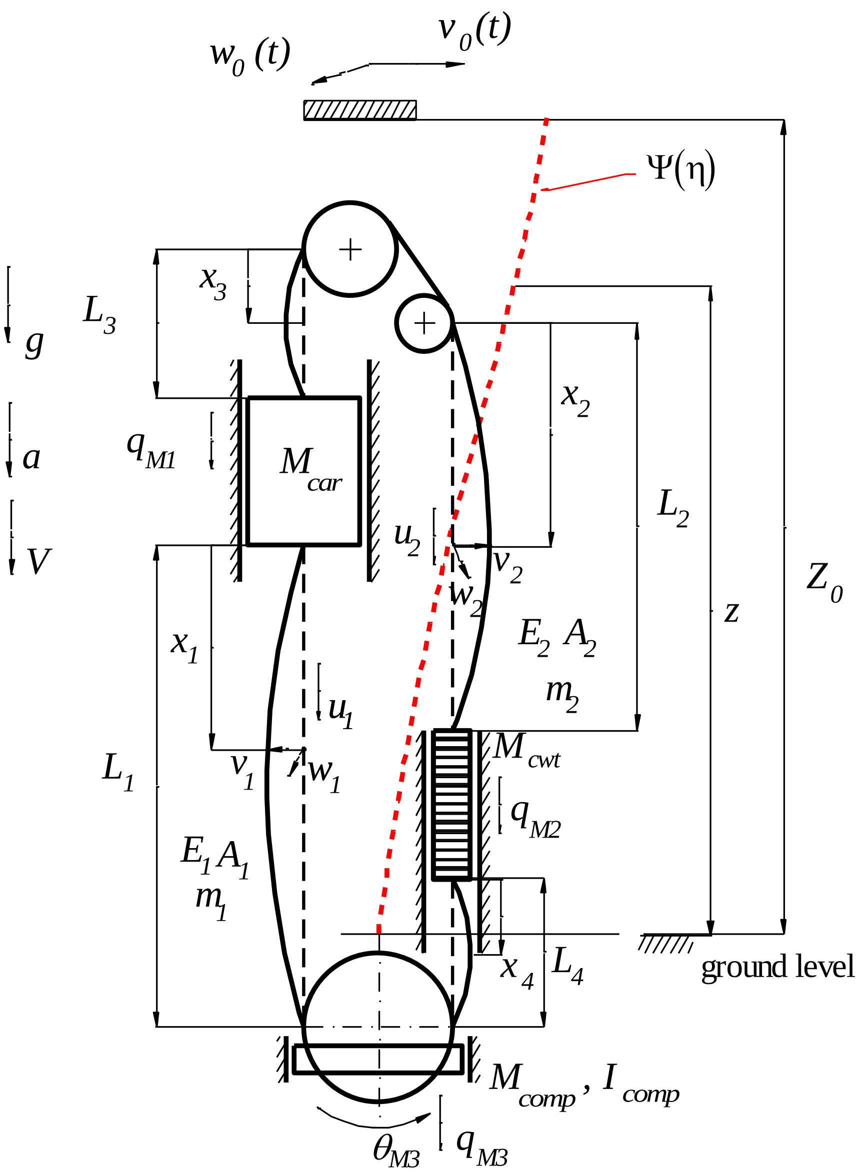

Figure 1 Simplified model of a high-rise lift system

The base motion excitation due to the building structure sway, which results in the in-plane motion $v_0(t)$ and out-of-plane motion $w_0(t)$ at the building top level, acts upon the suspension ropes and compensating cables that suffer from large dynamic displacements. Due to the variation of their lenghths the natural frequencies change during travel, rendering the system nonstationary. An adverse situation arises when the building is excited its natural frequency and vibrates periodically. This in turn may result in external, parametric and internal resonances in the lift system.

2.2 Mathematical model

Eqs (1) represent the mathematical model, based on the diagram in Fig. 1, with the excitation mechanism expressed by functions defined in terms of the deformations of the building represented by the shape function $\Psi\left(z/Z_0\right)$.

$$
\begin{aligned}
& m_i\bar{v}_{itt} - \left\{T_i - m_i\left[V^2 + \left(g - a_i\right)x_i\right] + E_iA_ie_i\right\}\bar{v}_{ixx} + m_i g\bar{v}_{ix} + 2m_iV\bar{v}_{ixt} = F_i^{v}\left[t, L_i\left(t\right)\right],\ i = 1,\ldots 4,\\
& m_i\bar{w}_{itt} - \left\{T_i - m_i\left[V^2 + \left(g - a_i\right)x_i\right] + E_iA_ie_i\right\}\bar{w}_{ixx} + m_i g\bar{w}_{ix} + 2m_iV\bar{w}_{ixt} = F_i^{w}\left[t, L_i\left(t\right)\right],\ i = 1,\ldots 4,\\
& M_{car}\ddot{q}_{M1} - E_1A_1e_1 + E_2A_2e_3 = 0;\\
& M_{cwt}\ddot{q}_{M2} - E_1A_1e_4 + E_2A_2e_2 = 0;\\
& M_{comp}\ddot{q}_{M3} + E_1A_1e_1 + E_1A_1e_4 = 0,\\
& I_{comp}\ddot{\theta}_{M3} - RE_1A_1e_1 + RE_1A_1e_4 = 0,\\
& e_1 = \frac{1}{L_1\left(t\right)}\left[u_1\left(L_1,t\right) - q_{M1}\left(t\right) + \frac{1}{2}\int_0^{L_1}\left(\bar{v}_{1x}^2 + \bar{w}_{1x}^2\right)dx_1 + \frac{\Psi_1^2}{2L_1\left(t\right)}\left(v_0^2 + w_0^2\right)\right],\\
& e_2 = \frac{1}{L_2\left(t\right)}\left[q_{M2}\left(t\right) + \frac{1}{2}\int_0^{L_2}\left(\bar{v}_{2x}^2 + \bar{w}_{2x}^2\right)dx_2 + \frac{\left(\Psi_h - \Psi_2\right)^2}{2L_2\left(t\right)}\left(v_0^2 + w_0^2\right)\right],\\
& e_3 = \frac{1}{L_3\left(t\right)}\left[q_{M1}\left(t\right) + \frac{1}{2}\int_0^{L_3}\left(\bar{v}_{3x}^2 + \bar{w}_{3x}^2\right)dx_3 + \frac{\left(\Psi_{car} - \Psi_{mach}\right)^2}{2L_3\left(t\right)}\left(v_0^2 + w_0^2\right)\right],\\
& e_4 = \frac{1}{L_4\left(t\right)}\left[u_4\left(L_4,t\right) - q_{M2}\left(t\right) + \frac{1}{2}\int_0^{L4}\left(\bar{v}_{4x}^2 + \bar{w}_{4x}^2\right)dx_4 + \frac{\Psi_{cwt}^2}{2L_4\left(t\right)}\left(v_0^2 + w_0^2\right)\right],\\
& 2q_{M3} - u_1\left(L_1,t\right) - u_4\left(L_4,t\right) = 0
\end{aligned}
\tag{1}
$$

In this model e_i denote the quasi-static axial strains in the ropes, $\bar{v}_i\left(x_i,t\right), \bar{w}_i\left(x_i,t\right)$, represent the dynamic displacements of the ropes, T_i, denote the rope quasi-static tension terms, and a_i are the acceleration / deceleration rates of the car /counterweight, where $i = 1,2,\ldots, 4$. The continous slowly varying nonlinear system (1) is discretized by using the Galerkin method [2] and the resulting ordinary differential equation (ODE) set of nonlinear equations is solved numerically.

3 CASE STUDY AND NUMERICAL RESULTS

Fig. 2 and Fig. 3 show the variation of the first two lateral natural frequencies (ω_1, ω_2) of the compensating ropes and the suspension ropes, respectively, in a lift installation servicing a 250 m tall building. The ropes are of standard SWR type and their natural frequencies are plotted against the corresponding length of each rope section, with the in-plane and out-of-plane excitation frequencies (Ω_v, Ω_w; $\Omega_v < \Omega_w$) represented by red horizontal lines, respectively.

On the other hand, the variations of the first four vertical mode natural frequencies ($\hat{\omega}_i$, $i = 1,\ldots,4$) are illustrated in Fig. 4. It is evident that those frequencies are much higher than the resonance frequencies of the building structure ($\hat{\omega}_i / \Omega_{v,w} > 3.5$, $i = 1,2,3$). It can be observed that there are length regions where the lateral fundamental and secondary resonance conditions arise in the system. For example, when the length of the car suspension rope is about 153 m the fundamental resonance of the car suspension system is taking place. The in-plane and out-of-plane displacements of the ropes are shown in Fig. 5. It can be observed that the displacements grow in time due to the resonance condition. The lateral responses of the ropes are coupled with the vertical motions of the car, counterweight and the compensating sheave assembly. These motions are shown in Fig. 6 vs time, with the frequency spectrum (obtained by the application of fast Fourier transform) illustrated in Fig. 7. It is evident that substantial motions of the vertical masses occur, with the dominant frequency being twice the frequency of the building sway. The dynamic interactions in the system result in tubular (whirling) motions of the ropes which is illustrated in Fig. 8 where the trajectory of the displacements of mid-span section (in-plane response vs. the out-of- plane response) are plotted.

4 CONCLUDING REMARKS

The dynamic behaviour of a high-rise lift system subjected to excitations arising from the building motions can be analysed by the application of suitable models. The models and simulation techniques can then be used to predict a range of dynamic interaction and resonance phenomena. The resonance frequencies of the ropes can be shifted / changed by the use of different masses of the compensating sheave assembly. The frequencies of the suspension ropes depend on the mass/ weight of the car (and the corresponding mass of the counterweight) as well as on the car loading conditions. Thus, more advance strategies, such as the active stiffness method [8], can be developed to minimize the effects of adverse dynamic responses of the system. It should be noted that the nature of the dynamic conditions present in high-rise building systems is such that small changes of the natural frequencies of the structure might result in large changes of the resonance conditions that arise in the lift installation.

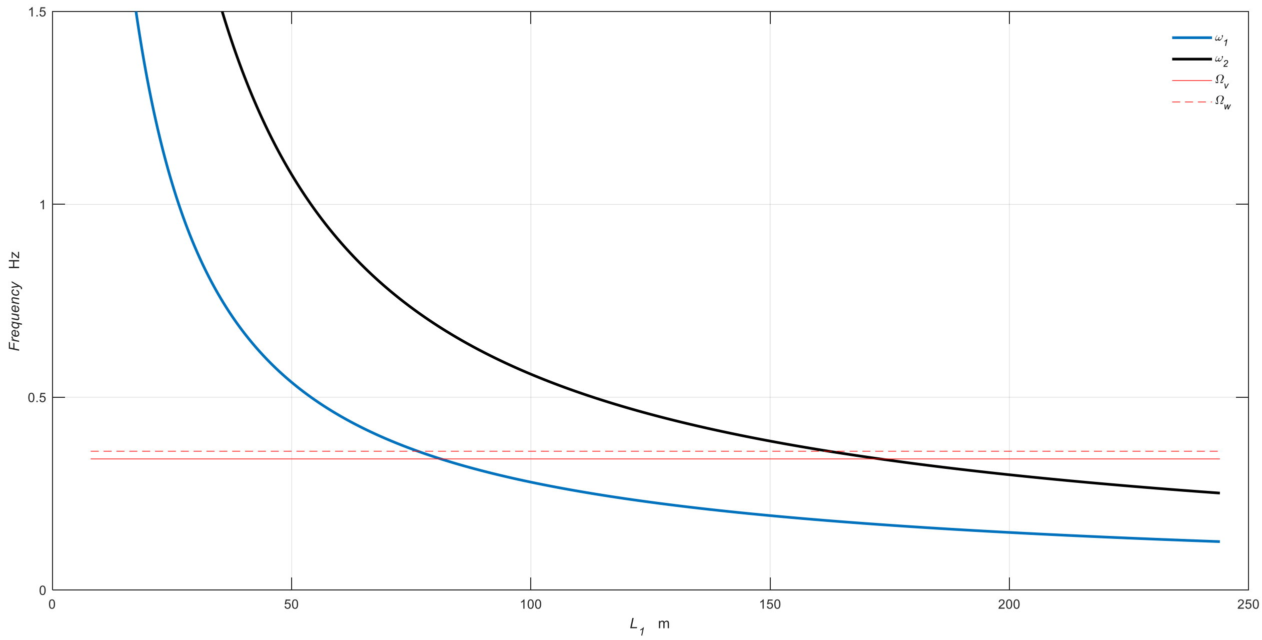

Figure 2 The natural frequencies: compensating ropes at the car side

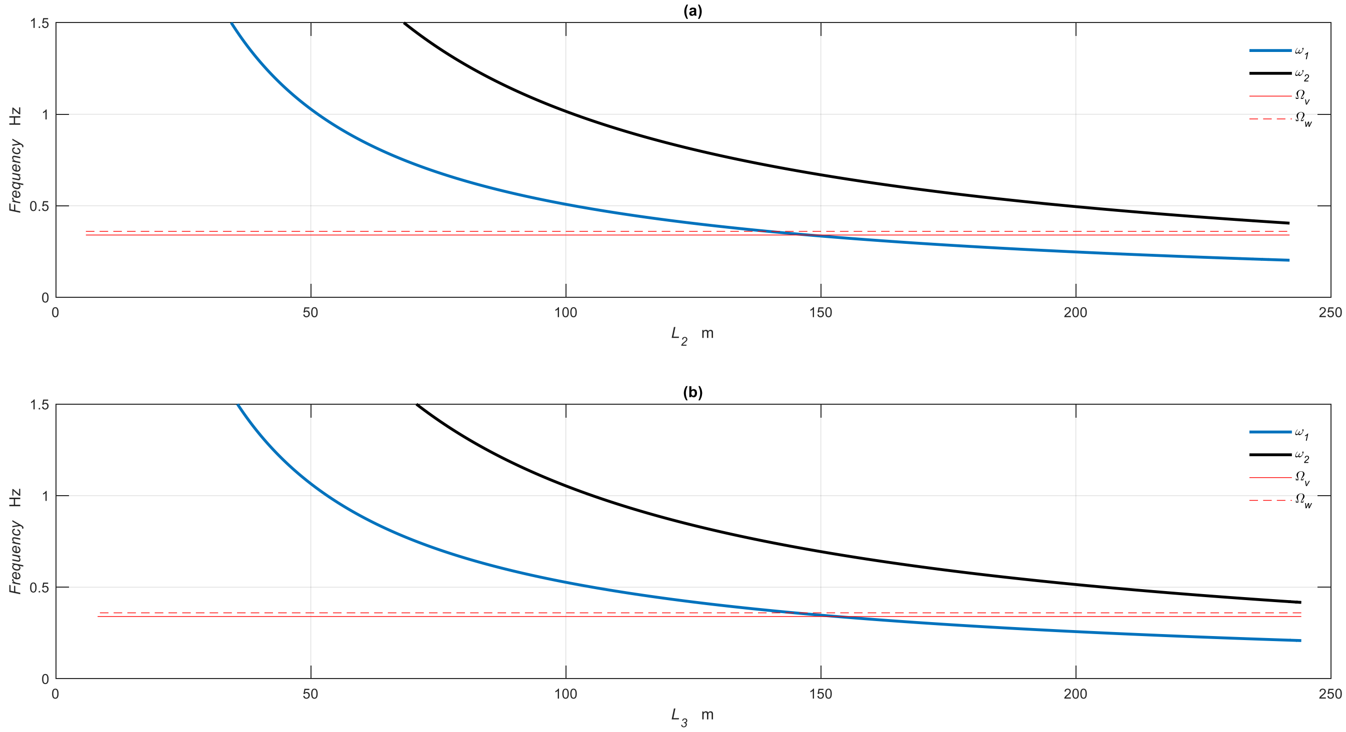

Figure 3 The natural frequencies: suspension ropes: (a) at the counterweight side and (b) at the car side

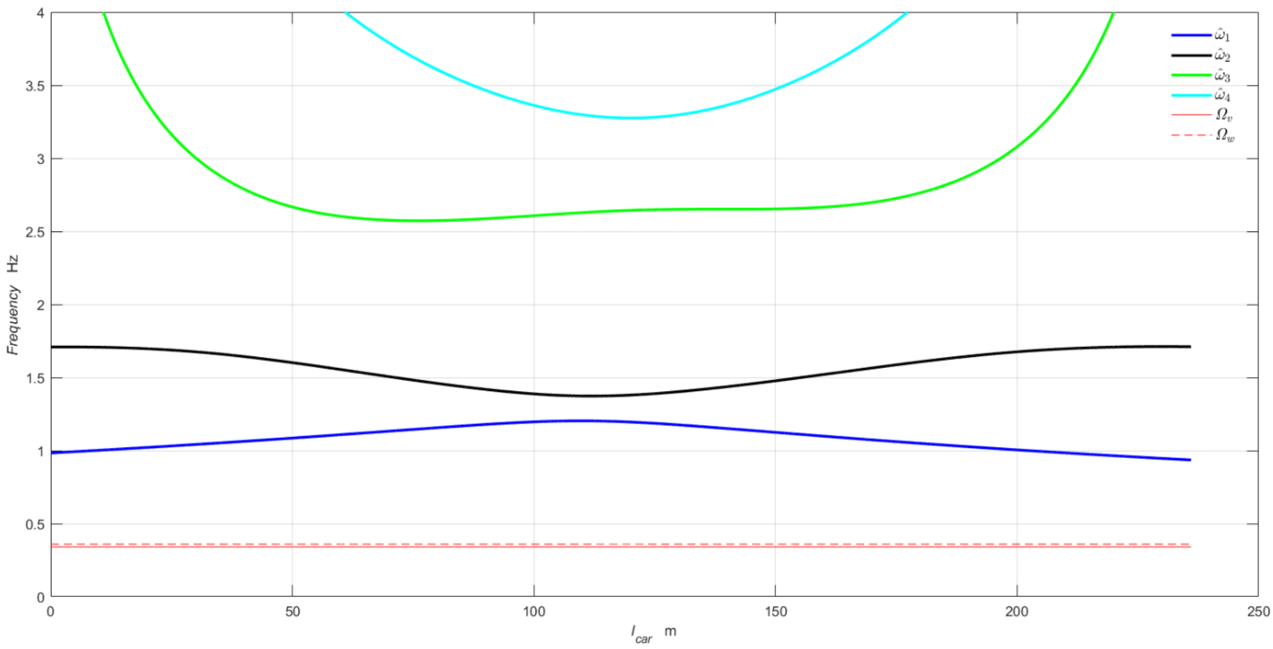

Figure 4 The natural frequencies: vertical modes

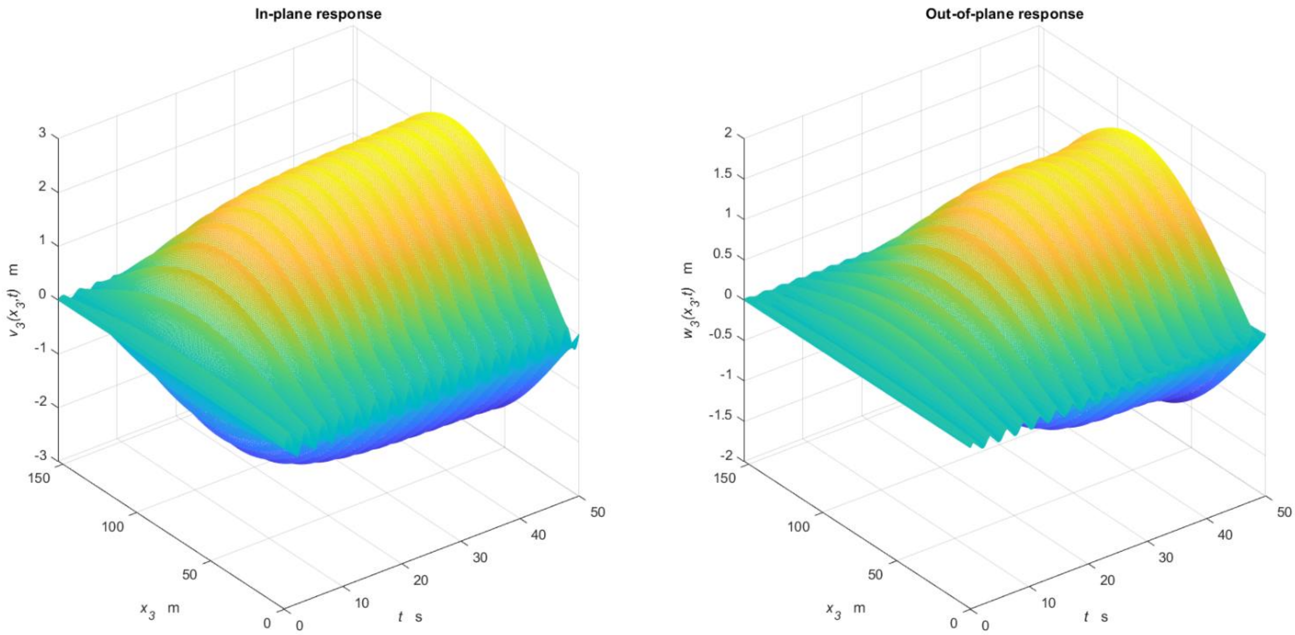

Figure 5 Displacements of the car suspension ropes

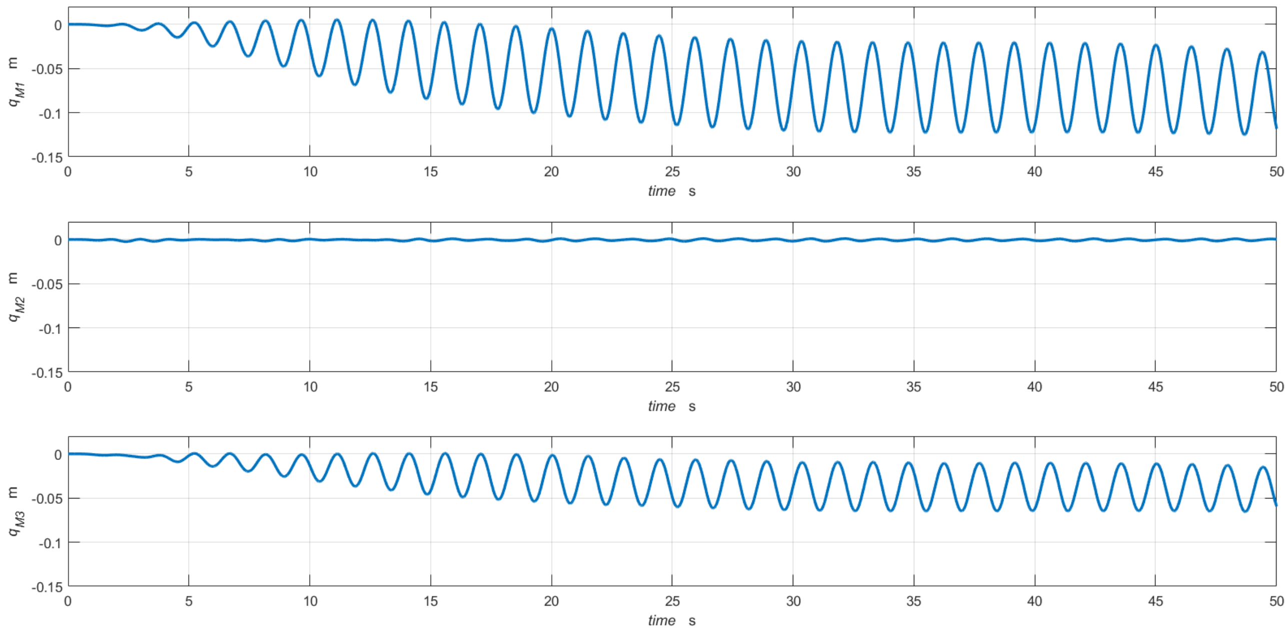

Figure 6 Vertical displacements of the car (q_{M1}), counterweight (q_{M2}) and compensating sheave (q_{M3})

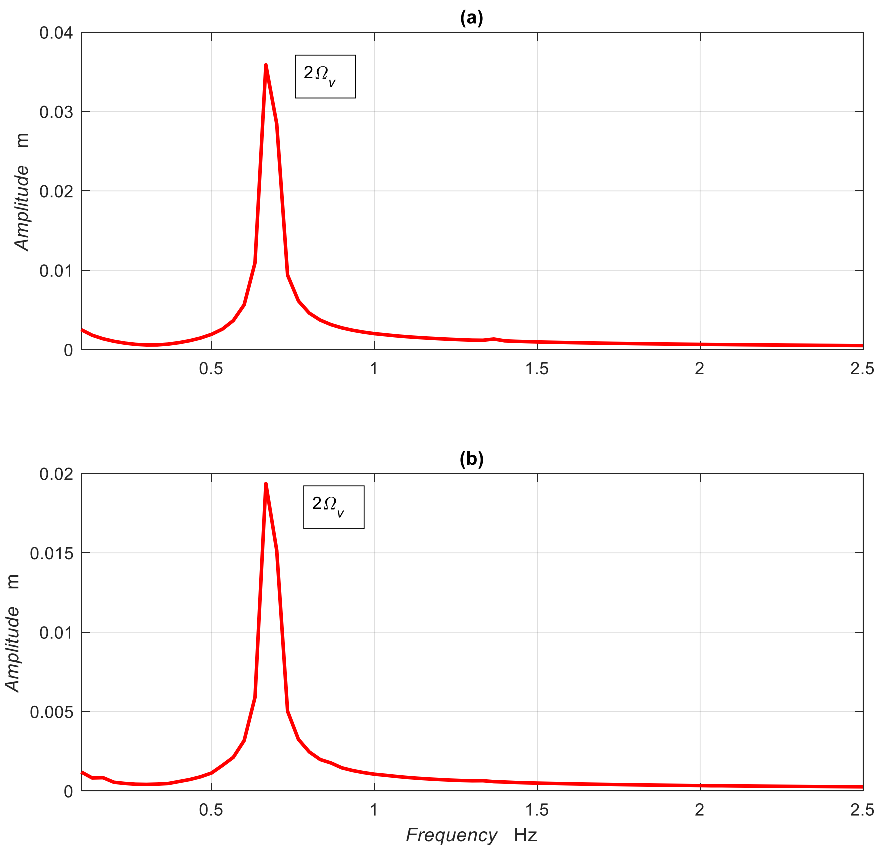

Figure 7 Frequency spectrum of vertical responses of the car (a) and compensating sheave (b)

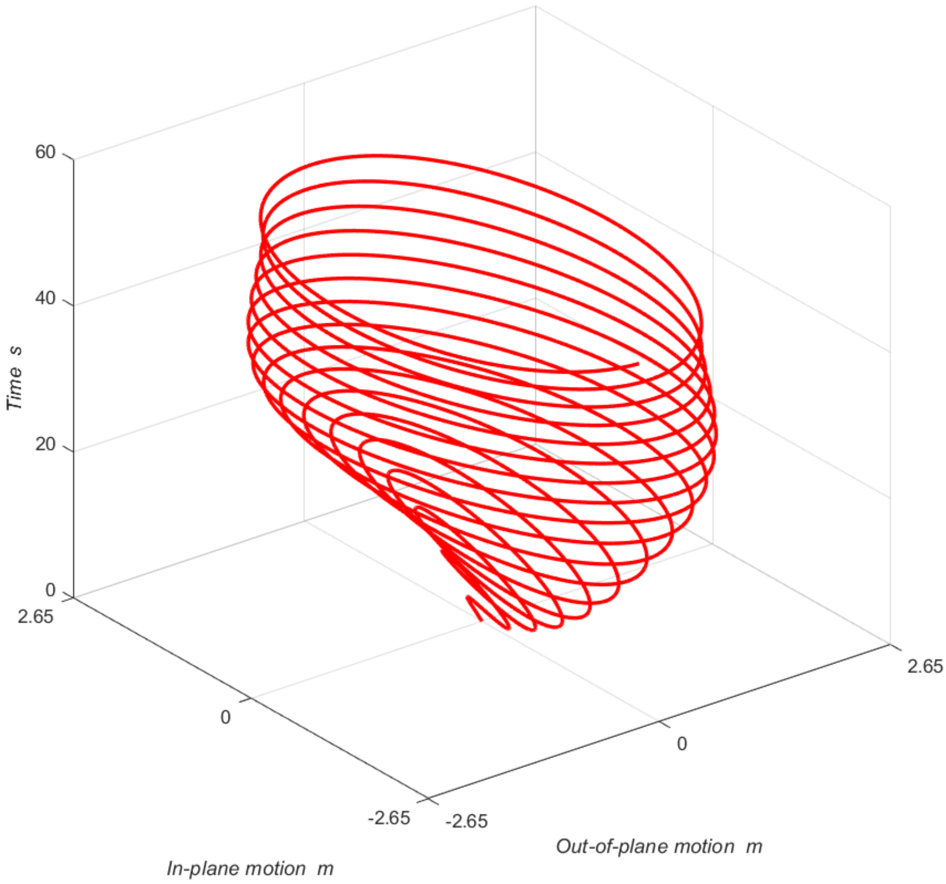

Figure 8 Whirling motions of the car suspension ropes (at x_3 = 78 m)

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BIOGRAPHICAL DETAILS

Stefan Kaczmarczyk has a master’s degree in Mechanical Engineering and he obtained his doctorate in Engineering Dynamics. He is Professor of Applied Mechanics and Postgraduate Programme Leader for Lift Engineering at the University of Northampton. His expertise is in the area of applied dynamics and vibration with particular applications to vertical transportation and material handling systems. He has been involved in collaborative research with a number of national and international partners and has an extensive track record in consulting and research in vertical transportation and lift engineering. Professor Kaczmarczyk has published over 100 journal and international conference papers in this field. He is a Chartered Engineer, elected Fellow of the Institution of Mechanical Engineers, and he has been serving on the Applied Mechanics Group Committee of the Institute of Physics.

The Study of Hoisting System for Vertical Shaft Construction Without the Protection of Guided-cable

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Keywords: Hoisting System, double pendulum, lateral vibration.

Abstract. The sinking bucket, known as the hoisting conveyance of the system, is used to transport the waste pile, water, miners and sinking equipment. In the case of a construction shaft when the sinking bucket is lowered downward from the sinking platform to the bottom of a well and it needs to pass through the sinking platform, the bucket is segregated from a guided carriage and it descends or rises without the protection of the guided-cable. This adverse condition is a common phenomenon in the mines vertical shaft. The hoisting rope is a steel wire rope with a low damping, so it is easy to cause oscillating resonance. This study is concerned with the theoretical modeling and simulation verification of the oscillating resonance of the sinking bucket and hoisting cable. A large transient amplitude and steady-state oscillation of the payload may often occur which may result in emergency scenarios. It will threaten the safety of the miners' lives and delay the process of construction of vertical shaft. Thus, the research on the dynamic characteristics of the sinking bucket without the protection of guided-cable is necessary.

1 INTRODUCTION

The flexible steel wire rope is one of the most popular flexible mechanical structures used in engineering due to its advantages in lightweight, lower cost and higher sensitivity, that has been extensively studied both theoretically and practically in recent decades[1, 2]. The translating media with variable length is widely applied in modern industrial fields, such as high elevator cables[3, 4], cable-driven parallel sinking platform[5-7], tethered satellite[8, 9], gantry crane[10, 11] and mining hoists[12, 13]. Bao[14] studied longitudinal vibration of flexible hoisting systems with time-varying length, the governing equations are developed employing the extended Hamilton's principle considering mutual influence of the rigid motion and deformation of flexible hoisting systems. Kumaniecka[15] investigated analysis of parametric resonance of the longitudinal-transversal vibrations of a rope in a non-linear set of partial differential equations with varying length. Hoisting system for vertical shaft construction includes hoisting cable, guided-cable, sinking platform, and bucket. Due to the fact that the lateral stiffness of the guided rope is smaller, the lateral response of the moving hoisting bucket in cable-guided hoisting system for construction shafts is investigated by Wang [16].

However, the bucket segregated from a guided carriage when it is through the sinking platform and descends or rises without the protection of guided-cable (Fig.1). The model in Fig. 1(a) shows the 'bucket' as a concentrated mass m1. Thus, this model represents a simple pendulum, with the mass m_1 suspended on the cable of length $l(t)$. On the other hand, Fig. 1(b) shows the 'bucket' as a double pendulum with the concentrated mass m_2 representing the payload. The bucket double pendulum together with the payload form a double pendulum with time-varying length and their oscillations induced by external disturbances that is a great security risk. The flexible nature of the physical structure causes oscillating resonance and degrades their effectiveness and safety. This study is concerned with the theoretical modeling the oscillating resonance of the bucket and hoisting cable in the plane. Hoisting systems without the protection of a guided-cable are highly complex dynamic systems that require detailed simulation and analysis in order to achieve acceptable levels of swing angle.

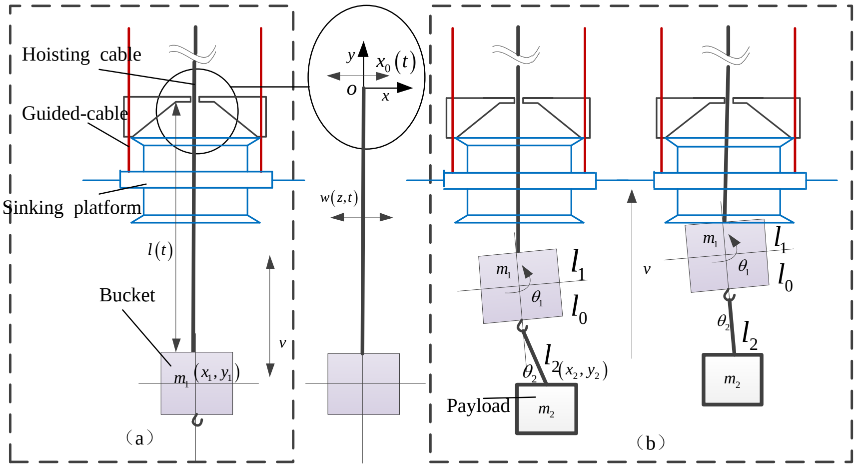

Figure 1 The schematic diagram of hoisting system without guided-cable: (a) Single pendulum; (b) Double pendulum;

2 MODELING

Frame Oxy is the fixed inertia frame (Fig.1), $w(y,t)$ is the position of the flexible cable with respect to the frame Oxy at the position y for time t. The lateral displacement of the cable end is denoted as x. l_1 and l_0 denote the length of the center of the bucket to the upper and lower contact point. l_2 denotes the length of the lower contact point of the bucket to the upper contact point of the payload. θ_1 and θ_2 are the swing angle of the bucket and the payload with respect to the vertical axis, pendulum length, respectively(Fig.1(b)) , respectively. I_1 and I_2 are the inertia of the bucket and the payload, respectively. m_1 and m_2 are the mass of the bucket and the payload, respectively; ρ is the cable density; g is the gravitational acceleration, T_s is cable tension; d_i is constant damping coefficient. Their displacements are expressed as

$$\begin{aligned} &x_1 = x + l_1 \sin\theta_1, y_1 = -l - l_1 \cos\theta_1 \\ &x_2 = x + (l_1 + l_0)\sin\theta_1 + l_2 \sin\theta_2, y_2 = -l - (l_1 + l_0)\cos\theta_1 - l_2 \cos\theta_2 \end{aligned} \tag{1}$$

Its corresponding velocity can be obtained by taking the derivative of position with respect to time as following:

$$\begin{aligned} &\dot{x}_1 = \dot{x} + l_1\dot{\theta}_1 \cos\theta_1, \dot{y}_1 = -\dot{l} + l_1\dot{\theta}_1 \sin\theta_1 \\ &\dot{x}_2 = \dot{x} + (l_0 + l_1)\dot{\theta}_1 \cos\theta_1 + l_2\dot{\theta}_2 \cos\theta_2, \dot{y}_2 = -\dot{l} + (l_0 + l_1)\dot{\theta}_1 \sin\theta_1 + l_2\dot{\theta}_2 \sin\theta_2 \end{aligned} \tag{2}$$

The kinetic energy, the potential energy and dissipative energy of the system can be expressed as

$$T = \frac{1}{2}m_1\left(\dot{x}_1^2 + \dot{y}_1^2\right) + \frac{1}{2}m_2\left(\dot{x}_2^2 + \dot{y}_2^2\right) + \frac{1}{2}\rho\int_0^l \left(\frac{\partial w}{\partial t} + v\frac{\partial w}{\partial y}\right)^2 dy + \frac{1}{2}\sum_{i=1}^{2} I_i\dot{\theta}_i^2 \tag{3}$$

$$V = m_1 g\left(l + l_1(1-\cos\theta_1)\right) + m_2 g\left(l + (l_1 + l_0)(1-\cos\theta_1) + l_2(1-\cos\theta_2)\right) + \rho g\int_0^l y dy + \int_0^l T_s \frac{1}{2}\left(\frac{\partial w}{\partial y}\right)^2 dy \tag{4}$$

$$D = \frac{1}{2}d_0\dot{x}^2 + \frac{1}{2}d_1\dot{\theta}_1^2 + \frac{1}{2}d_2\dot{\theta}_2^2 + \frac{1}{2}d_l\int_0^{l(t)}\left(\frac{\partial w}{\partial t}\right)^2 dy \tag{5}$$

The solutions of lateral displacement are assumed in the form:

$$w(y,t) = \sum_{j=1}^{n}\psi_j q_j^c + \left(1 - \frac{y}{l}\right)x_0(t) \tag{6}$$

where q_j^c is the generalized coordinate, $\psi_j(\xi)=\sqrt{2}\sin\left((j-1/2)\pi\xi\right)$ is the mode functions, $x_0(t)$ is the boundary excitation at the guided carriage (Fig.1) that can be assumed to be a simple harmonic function.

The government equation can be obtained from the action quantity into the Lagrange equation [17, 18]

$$\frac{\mathrm{d}}{dt}\left(\frac{\partial T}{\partial \dot{q}_j}\right)-\frac{\partial T}{\partial q_j}+\frac{\partial D}{\partial \dot{q}_j}+\frac{\partial V}{\partial q_j}=\sum_{i=1}^{N}\lambda_i\frac{\partial g_i}{\partial q_j},\left(j=1,2\cdots n\right) \tag{7}$$

In which, $\boldsymbol{q}=\left[\boldsymbol{q}^c,x,\theta_1,\theta_2\right]^{\mathrm{T}}$; $\boldsymbol{g}\in R^{N\times 1}$ is the vector of geometric boundary conditions hoisting cable and the bucket, and λ_i is the Lagrange multipliers. From Fig. 1, only one constraint condition is given by $g_1=x-w(l,t)=0, N=1$.

After ignoring high order terms, the dynamic equations (7) can be rewritten as

$$\begin{bmatrix}\boldsymbol{M}_c & \\ & \boldsymbol{M}_p\end{bmatrix}\ddot{\boldsymbol{q}}+\begin{bmatrix}\boldsymbol{C}_c & \\ & \boldsymbol{C}_p\end{bmatrix}\dot{\boldsymbol{q}}+\begin{bmatrix}\boldsymbol{K}_c & \\ & \boldsymbol{K}_p\end{bmatrix}\boldsymbol{q}=\begin{bmatrix}\boldsymbol{F}_c \\ \boldsymbol{0}_{3\times 1}\end{bmatrix}+\boldsymbol{\lambda}^{\mathrm{T}}\frac{\partial \boldsymbol{g}}{\partial \boldsymbol{q}} \tag{8}$$

where $\boldsymbol{M}_{\Box}$ denotes the inertia matrix, $\boldsymbol{C}_{\Box}$ is the damping matrix, $\boldsymbol{K}_{\Box}$ denotes the stiffness matrix, and $\boldsymbol{F}_{\Box}$ represents the force input vector. These matrixes are expressed as follows:

$$\boldsymbol{M}_c=\rho l\boldsymbol{M}_1,\quad \boldsymbol{C}_c=\rho v\left(\boldsymbol{C}_1+\boldsymbol{C}_2-\boldsymbol{C}_2^{\mathrm{T}}\right)+d_l l\boldsymbol{C}_1,\quad \boldsymbol{K}_c=\rho a\boldsymbol{K}_1-\rho v^2/l\boldsymbol{K}_2+\left((g-a)\sum_{k=1}^{2}m_k/l\boldsymbol{K}_3+(g-a)\rho\boldsymbol{K}_4\right)-d_l v\boldsymbol{K}_5$$

$$\boldsymbol{F}_c=-\rho\left(v^2x_0(t)/l-v\dot{x}_0(t)\right)\boldsymbol{F}_1-\rho\left(l\ddot{x}_0(t)-ax_0(t)\right)\boldsymbol{F}_2-(g-a)\left(\sum_{k=1}^{2}m_k/l\boldsymbol{F}_3+\rho\boldsymbol{F}_4\right)x_0(t)$$

$$\boldsymbol{M}_p=\begin{pmatrix} m_1+m_2 & \cos\theta_1\left(l_1m_1+(l_0+l_1)m_2\right) & l_2m_2\cos\theta_2 \\ \cos\theta_1\left(l_1m_1+(l_0+l_1)m_2\right) & I_1+l_1^2m_1+(l_1+l_0)^2m_2 & l_2m_2\cos(\theta_1-\theta_2)(l_0+l_1) \\ l_2m_2\cos\theta_2 & l_2m_2\cos(\theta_1-\theta_2)(l_0+l_1) & m_2l_2^2+I_2 \end{pmatrix}$$

$$\boldsymbol{C}_p=\begin{pmatrix} d_0 & 0 & 0 \\ 0 & d_1 & 0 \\ 0 & 0 & d_2 \end{pmatrix},\boldsymbol{K}_p=\begin{pmatrix} 0 & 0 & 0 \\ 0 & gl_1m_1+gm_2(l_0+l_1) & 0 \\ 0 & 0 & gl_2m_2 \end{pmatrix},$$

where, $\boldsymbol{M}_i,\boldsymbol{C}_i,\boldsymbol{K}_i$ and $\boldsymbol{F}_i$ are corresponding the mass, damping , stiffness coefficient matrices and the coefficient force vectors, respectively.

3 SIMULATION ANALYSIS

Suppose the hoisting system is an ideal power source and the bucket moving with the expected motion trajectory. In most situations, the bucket is not only loaded inside and usually needs, in practice, to hoist heavy equipment on the bottom of the bucket. When the bucket is raised to a certain height, the payload is hung at the bottom of the bucket with a certain inclination angle. In addition, for safety reasons, the initial displacement of the hoisting system should be limited in a certain range. Table 1 shows the main parameters of the system and the initial conditions are the angles $\theta_1(0),\theta_2(0)$.

Table 1 Main parameters and initial conditions

Parameters	Physical significance	Value
m_1	The bucket mass	500kg
m_2	The payload mass	500kg
ρ	Cable density	2kg/m
$l(0)$	Initial hoisting cable length	30m
l_0, l_1, l_2	Geometry length	0.5m,0.5m,1.5m
I_1, I_2	Inertia moment	200kgm^2, 200kgm^2
$\theta_1(0), \theta_2(0)$	Initial swing angle	0° ,10°
d_i, d_l	Damping coefficient	1.12,0.005

The difference between a double pendulum and a single pendulum as showed in Fig.2. Compared with the double pendulum, lateral amplitude of the bucket is quite small. When the bucket is separated, it is usually just below the sinking platform and subjected to the excitation of a guided carriage and external disturbances. The swing motion of the hoisting cable and the bucket is a substantial low-frequency oscillation. They swing more severely and the residual swing is sometimes difficult to dissipate, which may knock and damage the other parts of sinking platform. Neglecting the high order term model can still describe the real behavior of a double pendulum hoisting of the small angle assumption. However, it can't fully reflect the overall influence of the lateral vibration of the swing of hoisting cable.

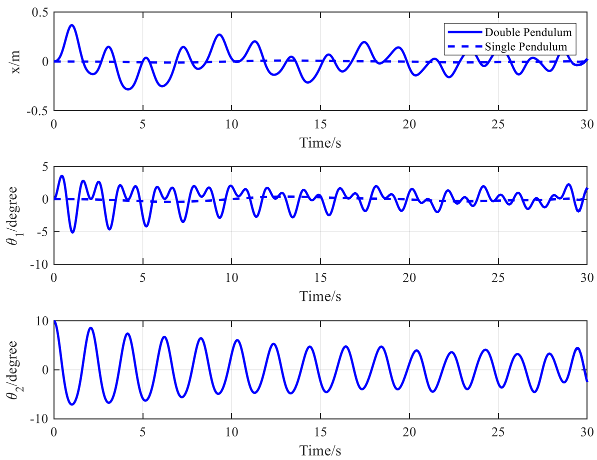

Figure 2 The difference between a double pendulum and a single pendulum: Dotted line: a single pendulum; Soild line: a double pendulum.

The two natural frequencies of the double pendulum system can be obtained by the empirical formula [19] [20]. The natural frequencies $\omega_{1,2}$ are described as follows

$$\omega_{1,2} = \sqrt{\frac{g}{2}\left((1+R)\left(\frac{1}{l}+\frac{1}{l_2}\right) \mp \sqrt{(1+R)^2\left(\frac{1}{l}+\frac{1}{l_2}\right)^2 - 4\frac{1+R}{l \cdot l_2}}\right)}, \tag{9}$$

where, $R = m_1 / m_2$ is the mass ratio of payload and bucket. It can be seen that the two nature frequencies not only depend on the length of the cables but also depend on the mass ratio.

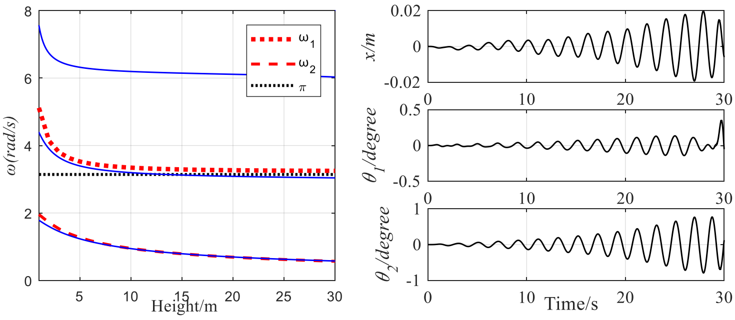

Figure 3 The frequency of swing angle and hoisting cable

The frequency of swing angle and lateral frequency hoisting cable as showed in Fig.3. One can obtain a natural frequency ω from

$$\left| \omega^2 \begin{bmatrix} \boldsymbol{M}_c & \\ & \boldsymbol{M}_p \end{bmatrix} - \begin{bmatrix} \boldsymbol{K}_c & \\ & \boldsymbol{K}_p \end{bmatrix} \right| = 0 \tag{10}$$

The first two order frequencies of the nonzero eigenvalues of the matrix are in a agreement with the frequencies of the swing angle. The second order frequency of swing angle is slightly higher than the result of matrix nonzero eigenvalues that's because the Eq.(9) does not take into consideration the mass of the hoisting cable. When the excitation frequency is πrad/s, so it is easy to cause oscillating resonance in Fig.3. The third order frequency is the lateral frequency of the hoisting cable higher than the swing angle frequency.

A set of initial values of the swing angle is set, and four-order Runge-Kutta numerical computing method is used to solve the equations governing the response of the system. Its plane trajectory can be obtained. The numerical results demonstrate that the plane trajectory of a double pendulum with different initial angles, the displacement boundary shown in the black dotted line. For example, in Fig. 4, the largest amplitude of payload reaches over 0.32m ($\theta_2(0)=15^\circ$) beyond the boundary line. Its displacement is not greater than the boundary line within the dis-placement boundary never exceeds with 10°initial angles. The position of the payload must be kept in the circle of the central point during the hoisting.

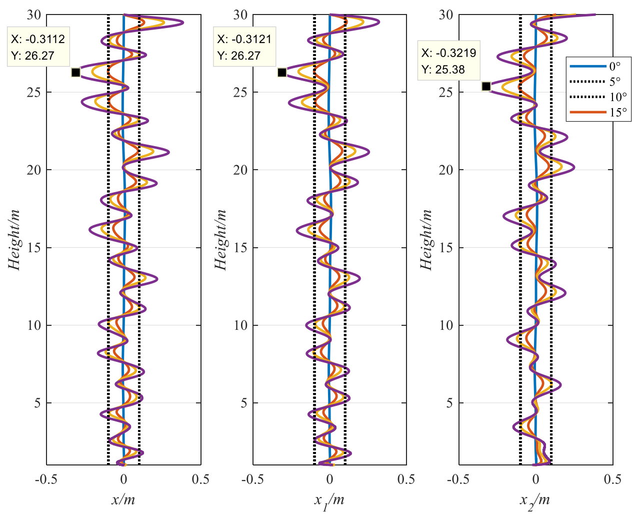

Figure 4 Comparison of the response of the system with different initial angles

4 CONCLUSION

This paper presented theoretical analyses of the hoisting system dynamics for vertical shaft construction without the protection of a guided-cable. The equations of motion of the hoisting system for vertical shaft construction comprising a bucket, a pay-load and hoisting cable excited by external disturbance are derived in this paper. The natural frequencies of the systems depend on the length of the cable, and the bucket-payload mass ratio and is used to predict the response of the system. Frequency characteristics analyses show that both the hoisting cable mass has effects on the second-mode frequency. The external disturbance of the payload will cause oscillation in a substantial low-frequency. Numerical simulation results show that the energy is passed from the payload to the bucket and hoisting cable. Once the initial inclination angle continues to grow, collision phenomenon between the bucket and the other parts of sinking platform in the hoisting process might occur which may lead to the swing of the payload and damage to the components of sinking platform.

5 ACKNOWLEDGMENTS

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BIOGRAPHICAL DETAILS

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