Symposium on Lift and Escalator Technologies

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FORWARD

It is with great pleasure that we present the proceedings of a Symposium on Lift and Escalator Technologies, September 2011, organised jointly by The Lift Engineering Section of the School of Science and Technology and The CIBSE Lift Group.

The Lift Engineering programme offered at The University of Northampton includes postgraduate courses at MSc/ MPhil/ PhD levels that involve a 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.

The Symposium brings together experts from the field of vertical transportation and offers an opportunity for graduates and students of the Lift Engineering programme at The University of Northampton to present papers on the subject of their research projects. There will also be keynote addresses by international industry experts invited by the CIBSE Lifts Group.

The papers are listed alphabetically by first author details. 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.

We are grateful to everyone who has submitted papers and in particular our invited speakers: Dr L. Al-Sharif, Mr J. Andrew, Dr G. Barney, Mr A. Scott and Mr R. Smith. We are also grateful to organisations that have supported this venture, as highlighted by their logos below.

Professor Stefan Kaczmarczyk, The University of Northampton and
Dr Richard Peters, The CIBSE Lifts Group

Symposium on Lift and Escalator Technologies

The use of Monte Carlo simulation to evaluate the passenger average travelling time under up-peak traffic conditions

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ABSTRACT

Monte Carlo simulation is a powerful tool used in calculating the value of a variable that is dependent on a number of random input variables. For this reason, it can be successfully used when calculating the round trip time of an elevator, where some of the inputs are random and follow pre-set probability distribution functions. The most obvious random inputs are the number of passengers boarding the car in one round trip, their origins (in the case of multiple entrances) and their destinations.

Monte Carlo simulation has been used to evaluate the elevator round trip time under up-peak traffic conditions. Its main advantage over analytical formula based methods is that it can deal with all special conditions in a building without the need for evaluating new special formulae. A combination of all of the following special conditions can be dealt with: Unequal floor population, unequal floor heights, multiple entrances and top speed not attained in one floor jump. Moreover, this can be done without loss of accuracy, by setting the number of runs to the appropriate value.

This paper extends the previous work on Monte Carlo simulation in relation to two aspects: the passenger arrival process model and the passenger average travelling time.

The software is developed using MATLAB. The results for the average travelling time are compared to analytical formulae (such as that by So. *et al.*, 2002). The results showing the effect of the Poisson arrival process on the value of the elevator round trip time are also analysed.

The advantage of the method over analytical methods is again demonstrated by showing how it can deal with the combination of all the special conditions without the loss of accuracy (five conditions if the passenger arrival model is added as Poisson).

The issues of convergence, accuracy and running time are discussed in relation to the practicality of the method.

Keywords: Monte Carlo simulation, elevator, lift, round trip time, interval, up peak traffic, average waiting time, average travelling time, multiple entrances, highest reversal floor, probable number of stops.

Nomenclature

a is the top acceleration in m/s^2

$AR\%$ is the passenger arrivals expressed as a percentage of the building population in the busiest five minutes

att is the average travelling time in s

awt is the average waiting time in s

CC is the car carrying capacity in persons

d_f is the height of one floor in m

$d_f(i)$ is the floor height for floor i

$d_{f\,eff}$ is the effective floor height used in the case of unequal floor heights in m

$E(d_{total})$ is the expected value of the distance travelled in the up direction in m

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d_G is the height of the ground in m where more than the typical floor height
$E(d_f)$ is the expected value of the floor heights (effective floor height)
H is the highest reversal floor (where floors are numbered 0, 1, 2….N
$HC\%$ is the handling capacity expressed as a percentage of the building population in five minutes
int is the interval at the main terminal in s
j is the top rated speed in m/s^3
L is the number of the elevators in the group
N is the number of floors above the main terminal
P is the number of passengers boarding the car from the main terminal (does not need to be an integer)
S is the probable number of stops
τ is the round trip time in s
t_{ao} is the door advance opening time in s (where the door starts opening before the car comes to a complete standstill)
t_{dc} is the door closing time in s
t_{do} is the door opening time in s
t_f is the time taken to complete a one floor journey in s
t_{pi} is the passenger boarding time in s
t_{po} is the passenger alighting time in s
t_{pB} is the component of the travelling time that the passenger spends boarding and alighting from the elevator car in s
t_{pW} is the component of the travelling time that the passenger spends waiting for other passengers to board and alight from the elevator car in s
t_{pH} *is the component of the travelling time that the passenger spends travelling in the up direction at rated speed in s*
t_{pS} *is the component of the travelling time that the passenger spends stopping when travelling in the up direction in s (accelerating, decelerating times, door opening and closing times)*
t_s is the time delay caused by a stop in s
t_{sd} is the motor start delay in s
t_v is the time required to traverse one floor when travelling at rated speed in s
U is the total building population
U_i is the building population on the i^{th} floor
v is the top rated speed in m/s

1. INTRODUCTION

Monte Carlo simulation is a powerful method that can be used to evaluate the output value for problems that have a number of random inputs, whereby the probability density functions of the input random variables are known. By generating instances of the random input variable in the form of scenarios, and running a large number of scenarios, the expected value of the output of interest can be found by taking the average value of all the scenarios. Scenarios in this paper will be referred to as trials.

Monte Carlo simulation has been effectively used to evaluate the round trip time under up peak traffic conditions [1], in finding an optimum parking policy [2] as well as generating passengers for the purposes of simulating [3]. It offers an advantage over conventional equation based methods where special conditions exists, such as unequal floor heights, unequal floor populations, top speed not attained in one journey and multiple entrances.

This paper extends the application of the method to the calculation of the passenger average travelling time. In order to verify the results of the method, an equation is developed to calculate the average travelling time under up peak traffic conditions assuming top speed is attained in one floor journey, single entrance and equal floor heights. The Monte Carlo simulation results for the average travelling time are then compared to the equation developed in [4]. The equation is then extended in order to cover the case of unequal floor heights.

Analytical methods for elevator traffic analysis have been extensively covered in [5], [6], [7] and [8]. The Poisson passenger arrival model has been extensively covered in [9], [10], [11] and [12]. The case of the top speed not attained in one floor journey is addressed in [13]. The case of multiple entrances has been addressed in [14]. Discrete time-slice Simulation based methods have been developed in [15].

In order to ensure consistency and clarity of the interpretation of the results, the following definitions will be used throughout this paper for the average waiting time (*awt*) and the average travelling time (*att*):

> *awt: The average waiting time will be defined as the period from passenger arrival in the lobby until the passenger starts to board the car. Thus, based on this definition, the average waiting time does <u>not</u> include the passenger boarding time.*
>
> *att: The average travelling time will be defined as the period from the time the passenger starts to board the car until the passenger has left the car at the destination floor. Thus, based on this definition, the average travelling time does include the passenger boarding time. It also includes the passenger alighting time at the destination.*

The equation for the average travelling time is derived in section 2. Verification of this equation using the Monte Carlo simulation method is carried out in section 3. The equation is then further adjusted for the case of unequal floor heights in section 4. The effect of the Poisson passenger arrival model is analysed in section 5. A practical elevator system design example is given in section 6. A number of notes on convergence are presented in section 7. Conclusions and further work is presented in sections 8 and 9 respectively.

2. DERVATION OF THE EQUATION FOR THE AVERAGE TRAVELLING TIME

An equation for the average travelling time has been developed in [4]. The equation is derived in the section using a different approach and in accordance with definition presented earlier.

The approach that will be followed in deriving the average travelling time is to find the expression for each component of the minimum possible time and maximum possible time and the use the average of both.

The average travelling time includes four components:

- The boarding and alighting time for the passenger himself/herself.
- The time the passenger spends waiting for other passengers to board and alight.
- The time the passenger spends during the elevator stoppage time (where stoppage time includes acceleration and deceleration time as well as door opening and closing times).
- The time that the passenger spends in the elevator car travelling at top speed.

The first component, which is the boarding and alighting time of the passenger, is easy to evaluate:

$$t_{pB} = t_{pi} + t_{po} \qquad (1)$$

In order to calculate the second component, it is assumed that on average the passenger will have the remaining $P-1$ passengers ahead of him/her and the other half behind him/her. Thus he/she will have to wait for $\left(\frac{P-1}{2}\right)$ passenger to board the elevator after he/she has boarded; and will have to wait for $\left(\frac{P-1}{2}\right)$ passenger to alight before he/she could alight.

$$t_{pW} = t_{pi} \cdot \left(\frac{P-1}{2}\right) + t_{po} \cdot \left(\frac{P-1}{2}\right) = \left(t_{pi} + t_{po}\right) \cdot \left(\frac{P-1}{2}\right) \tag{2}$$

As for the time spent during elevator stops, it is worth noting that all passengers will at least have to wait for the first stop (rational passenger boarding at the ground cannot alight at the ground and must at least wait for the first stop). Thus all passengers as a minimum must wait for t_s caused by the first stop. As a maximum, a passenger might have to wait for all the S stops above, $S \cdot t_s$. None of the passengers will wait the last stop (door closing at the highest floor, acceleration and deceleration during the express back journey and doors opening at the main entrance) and hence the wait is for S stops rather than S+1 stops. Taking the average of both values above, gives the average time each passenger waits during elevator stops travelling in the up direction:

$$t_{pS} = \frac{S \cdot t_s + t_s}{2} = t_s \cdot \left(\frac{S+1}{2}\right) \tag{3}$$

On average each stop will traverse a distance of $\frac{H}{S}$ floors. All passengers will have to wait for that distance to be traversed at stop speed at least, as any rational passenger cannot board at the main terminal and leave at the main terminal. As a maximum, some passengers will have to wait for the whole H floors to be traversed. The minimum time will be $t_v \cdot \frac{H}{S}$, while the maximum time will be $t_v \frac{H}{S} \cdot S$. Taking the average of both times give an expression for the time spent during travelling at top speed in the up direction.

$$t_{pH} = \frac{t_v \cdot \frac{H}{S} + t_v \frac{H}{S} \cdot S}{2} = t_v \cdot \left(\frac{H}{2}\right) \cdot \left(\frac{S+1}{S}\right) \tag{4}$$

Adding all the four terms provides an expression for the average travelling time:

$$\begin{aligned} att &= t_{pB} + t_{pW} + t_{pS} + t_{pH} = \\ &\left(t_{pi} + t_{po}\right) + \left(t_{pi} + t_{po}\right) \cdot \left(\frac{P-1}{2}\right) + t_s \cdot \left(\frac{S+1}{2}\right) + t_v \cdot \left(\frac{H}{2}\right) \cdot \left(\frac{S+1}{S}\right) \\ &= \left(t_{pi} + t_{po}\right) \cdot \left(\frac{P+1}{2}\right) + t_s \cdot \left(\frac{S+1}{2}\right) + t_v \cdot \left(\frac{H}{2}\right) \cdot \left(\frac{S+1}{S}\right) \end{aligned} \tag{5}$$

Rearranging and assuming that $t_{pi} = t_{po} = t_p$, gives the important final result for the average travelling time:

$$att = t_v \cdot \left(\frac{H}{2}\right) \cdot \left(\frac{S+1}{S}\right) + t_s \cdot \left(\frac{S+1}{2}\right) + t_p \cdot (P+1) \qquad (6)$$

A similar expression for the average travelling time has been derived by So () using a different method and is shown below:

$$att = t_v \cdot \left(\frac{H}{2}\right) \cdot \left(\frac{S+1}{S}\right) + t_s \cdot \left(\frac{S+1}{2}\right) + t_p \cdot (P) \qquad (7)$$

It is worth noting that the expression in equation (6) differs from the one in equation (7) in that it includes an extra t_p where this accounts for the fact that this definition of waiting time includes passenger boarding time, while equation (7) excluded passenger boarding time.

It is also worth noting that equations (6) and (7) implicitly make the following assumptions:

1. Top speed is attained on one floor journey.
2. Incoming up peak traffic only.
3. Equal floor heights.
4. Single entrance.

The equation of the round trip time depends on the values of *S* (probable number of stops), *H* (the highest reversal floor) and *P* (the number of passengers in the car) as shown in equation (16).

$$\tau = 2 \cdot H \cdot t_v + (S+1) \cdot t_s + P \cdot (t_{pi} + t_{po}) \qquad (8)$$

The highest reversal floor is a function of the number of passengers:

$$H = f(P) \qquad (9)$$

The probable number of stops is also a function of the number of passengers:

$$S = f(P) \qquad (10)$$

The number of passengers in the elevator car is equal to the product of the passenger arrival rate and the actual interval:

$$P = \lambda \cdot int_{act} \qquad (11)$$

But the interval is in fact a function of the round trip time as shown in equation (20) below:

$$int_{act} = \frac{\tau}{L} \qquad (12)$$

Combining equations (19) and (20) gives the following result that shows that the number of passengers is a function of the round trip:

$$P = \lambda \cdot \frac{\tau}{L} \qquad (13)$$

As can be concluded from the two equations ((8) and (13)) the round trip time is a function of the number of passengers, but the number of passengers is a function of the round trip time. Thus the equation for the round trip time shown in (8) is an implicit equation of the round trip time that can be only solved by the use of an iterative approach (or other mathematical methods such as conformal mapping [11]). This has been addressed as part of a comprehensive design methodology [17].

When amending the equations for *H* and *S* to address the Poisson passenger arrival model, the term that represents the probability of a passenger not travelling to the i^{th} floor can be amended as shown below.

The probability of a passenger will not travel to floor *i* assuming equal floor populations for constant and Poisson arrival modes is shown below (using equation (11)):

Constant passenger arrival model	$P\left(\overline{pass \rightarrow floor\, i}\right)\Big\vert_{constant} = \left(1 - \frac{1}{N}\right)$	(14)
Poisson passenger arrival model	$P\left(\overline{pass \rightarrow floor\, i}\right)\Big\vert_{Poisson} = \left(\exp\left(-\frac{1}{N}\right)\right)$	(15)

The probability that all the passengers will not go to a floor *i* is (assuming equal floor populations) for both constant and Poisson arrival models is shown below:

Constant passenger arrival model with equal floor populations	$P\left(\overline{all\ pass \rightarrow floor\, i}\right)\Big\vert_{constant} = \left(1 - \frac{1}{N}\right)^{\lambda \cdot int}$	(16)
Poisson passenger arrival model with equal floor populations	$P\left(\overline{all\ pass \rightarrow floor\, i}\right)\Big\vert_{Poisson} = \left(\exp\left(-\frac{1}{N}\right)\right)^{\lambda \cdot int} = \exp\left(-\frac{\lambda \cdot int}{N}\right)$	(17)

And this can be further developed for the case of unequal floor populations as shown below:

Constant passenger arrival model with equal floor populations	$P\left(\overline{all\ pass \rightarrow floor\, i}\right)\Big\vert_{constant} = \left(1 - \frac{U_i}{U}\right)^{\lambda \cdot int}$	(18)
Poisson passenger arrival model with equal floor populations	$P\left(\overline{all\ pass \rightarrow floor\, i}\right)\Big\vert_{Poisson} = \left(\exp\left(-\frac{U_i}{U}\right)\right)^{\lambda \cdot int} = \exp\left(-\frac{\lambda \cdot int \cdot U_i}{U}\right)$	(19)

The probability of all passengers not going to a floor *i* is equivalent to the probability of the elevator not stopping at floor *i*. These expressions are used in deriving the values of *H* and *S* as shown in equations (20) to (27).

The equation for calculating the average travelling time (8) can cope with a number of special conditions such as unequal floor heights and Poisson arrival model by using the calculated for the probable number of stops and the highest reversal floor in accordance with equations (20) to (27).

	Constant passenger arrival model		Poisson passenger arrival model	
Equal floor populations	$S = N\left(1-\left(1-\frac{1}{N}\right)^{\lambda\cdot int}\right)$	(20)	$S = N\cdot\left(1-\exp\left(\frac{-\lambda\cdot\mathrm{int}}{N}\right)\right)$	(21)
Unequal floor populations	$S = N-\sum_{i=1}^{N}\left(1-\frac{U_i}{U}\right)^{\lambda\cdot int}$	(22)	$S = N-\sum_{i=1}^{N}\exp\left(\frac{-\lambda\cdot\mathrm{int}\cdot U_i}{U}\right)$	(23)

	Constant passenger arrival model		Poisson passenger arrival model	
Equal floor populations	$H = N-\sum_{i=1}^{N-1}\left(\frac{i}{N}\right)^{\lambda\cdot int}$	(24)	$H = N-\sum_{i=1}^{N-1}\exp\left(\frac{-\lambda\cdot\mathrm{int}}{N}\right)^{i}$	(25)
Unequal floor populations	$H = N-\sum_{j=1}^{N-1}\left(\sum_{i=1}^{j}\frac{U_i}{U}\right)^{\lambda\cdot int}$	(26)	$H = N-\sum_{i=1}^{N-1}\left(\prod_{j=N-i+1}^{N}\exp\left(\frac{-\lambda\cdot\mathrm{int}\cdot U_i}{U}\right)^{j}\right)$	(27)

3. VERIFICATION

The derivation of the equation for the average travelling time has been necessary in order to verify the use of the Monte Carlo simulation. A repeat of the calculations carried out in [4] has been carried out with the results shown in Table 1. The results show excellent agreement with the calculation results.

Table 1: Verification results for the average travelling time comparing calculation and Monte Carlo simulation.

N	P	Analytical Equation, assuming constant arrival process (7)	Monte Carlo Simulation (assuming constant arrival process)
10	6.4	48.19	48.18
10	16.8	74.46	74.40
13	6.4	53.65	53.67
13	16.8	84.00	84.00
16	10.4	73.27	73.27
16	20.8	101.97	101.97
20	10.4	80.70	80.72
20	20.8	112.85	112.85
23	12.8	94.74	94.65
23	26.4	135.00	135.03

However, the strength of the Monte Carlo simulation method becomes clear when the special conditions exist (such as top speed not attained or multiple entrances), where the calculation method fails to deal with. This will be illustrated later in this paper.

4. CASE OF UNEQUAL FLOOR HEIGHTS

In the case where the floor heights are unequal, this will have an effect on the calculation of the round trip time equation. The equation for the round trip time or average travelling time can be amended as follows in order to account for this case as follows.

The effect of the unequal floor heights can be taken into consideration by assuming an effective floor height $d_{f\,eff}$ that can be inserted into the original round trip time equation.
The effective floor height $d_{f\,eff}$ is the expected value fo the floor height. The effective floor height is the weighted average of all the floor heights multiplied by the probability of the elevator passing through that floor. In order for the elevator to pass through a floor it should travel to any of the floors above that floor. Thus it is necessary to find the probability of the elevator travelling above a certain floor, i.

The probability of the elevator not stopping at a certain floor, assuming equal floor populations is the probability that passenger j will stop at a floor i (assuming equal floor populations and a constant passenger arrival model):

$$P(\,pass\;j\,will\,stop\,at\;floor\,i\,)=\left(\frac{1}{N}\right)\qquad(27)$$

Thus the probability that passenger j will not stop at a floor i is:

$$P(\,pass\;j\,will\,NOT\,stop\,at\;floor\,i\,)=\left(1-\frac{1}{N}\right)\qquad(28)$$

But the car contains P passengers. So the probability that none of them will stop at floor i is the product of all of their respective probabilities:

$$P(\,all\;pass\,will\,NOT\,stop\,at\;floor\,i\,)=\left(1-\frac{1}{N}\right)^{P}\qquad(29)$$

The probability that the lift will not travel any higher than a floor i is the probability that it will not stop on floor $i+1$ or $i+2$ or $i+3$ all the way to floor N. This is expressed as the product of these individual conditional probabilities:

$$P(\text{lift will not travel above floor } i) =$$
$$\left(1-\frac{1}{N}\right)^P \cdot \left(1-\frac{1}{N-1}\right)^P \cdot \left(1-\frac{1}{N-2}\right)^P \ldots\ldots \left(1-\frac{1}{i+2}\right)^P \cdot \left(1-\frac{1}{i+1}\right)^P \quad (30)$$

This can be re-written as:

$$P(\text{lift will not travel above floor } i) =$$
$$\left(\frac{N-1}{N}\right)^P \cdot \left(\frac{N-2}{N-1}\right)^P \cdot \left(\frac{N-3}{N-2}\right)^P \ldots\ldots \left(\frac{i+1}{i+2}\right)^P \cdot \left(\frac{i}{i+1}\right)^P \quad (31)$$

Putting all terms inside the same bracket gives:

$$P(\text{lift will not travel above floor } i) =$$
$$\left(\left(\frac{N-1}{N}\right) \cdot \left(\frac{N-2}{N-1}\right) \cdot \left(\frac{N-3}{N-2}\right) \ldots\ldots \left(\frac{i+1}{i+2}\right) \cdot \left(\frac{i}{i+1}\right)\right)^P \quad (32)$$

This simplifies to:

$$P(\text{lift will not travel above floor } i) = \left(\frac{i}{N}\right)^P \quad (33)$$

Thus the probability that the lift will travel above the floor i is:

$$P(\text{lift will travel above floor } i) = 1-\left(\frac{i}{N}\right)^P \quad (34)$$

Thus the expected value of the travel distance can be calculated as the weighted average of the various floor heights as follows:

$$\begin{aligned} E(d_{total}) &= d_f(1) \cdot \left(1-\left(\frac{1}{N}\right)^P\right) + d_f(2) \cdot \left(1-\left(\frac{2}{N}\right)^P\right) + \ldots \\ &+ d_f(N-1) \cdot \left(1-\left(\frac{N-1}{N}\right)^P\right) + d_f(N) \cdot \left(1-\left(\frac{N}{N}\right)^P\right) \end{aligned} \quad (35)$$

The last term above reduces to zero (as it is impossible for the elevator to pass through floor N). The expected floor height is obtained by dividing the expected total travel distance by the highest reversal floor, H. So the equation for the effective floor height can be expressed as shown below (assuming equal floor populations and a constant passenger arrival model):

$$E\left(d_f\right)=\frac{\sum_{i=1}^{N-1} d_f(i)\cdot\left(1-\left(\frac{i}{N}\right)^P\right)}{H} \tag{36}$$

The same procedure can be used to develop the equation for the case of unequal populations and Poisson passenger arrival model.

Taking an example to illustrate the difference in the effective floor height, a building with 20 floors above ground is analysed. The floor heights are shown below in Table 2. It will be assumed that the floor populations are equal and that the passenger arrival process is constant (rather than Poisson). It will be also assumed that the number of passenger, *P*, is 13.

Table 2: The floor heights for a building with 20 floors above ground.

Floor #	*i*	$d_f(i)$ (m)
L20	21	3.2
L19	20	3.2
L18	19	3.2
L17	18	4.2
L16	17	4.2
L15	16	4.2
L14	15	4.2
L13	14	4.2
L12	13	4.2
L11	12	4.2
L10	11	4.2
L9	10	4.2
L8	9	4.2
L7	8	4.2
L6	7	4.2
L5	6	4.2
L4	5	4.2
L3	4	6
L2	3	6
L1	2	6
G	1	8

Applying equation (24) to evaluate the highest reversal floor gives a value for *H* of: 18.95 (assuming floors numbers run from 1 to 21). Then applying equation (36) to evaluate the effective floor height gives a value of 4.62 m. This can be compared to the average floor height of all floors, which is 4.50 m. A difference of 0.12 m exists per floor.

The average passenger travelling time can be calculated in order to assess the effect of unequal floor heights, using equation (7). Using the parameters shown below, whereby the rated speed is attained in one floor journey, and there is only a single entrance and a constant passenger arrival model is assumed.

t_{do} = 2 s
t_{dc} = 3 s
t_{sd} = 0.5 s

t_{ao} = 0 s
t_{pi} = 1.2 s
t_{po} = 1.2 s
v = 1.6 m/s
a = 1.0 m/s^2
j = 1.0 m/s^3

The calculation and Monte Carlo simulation results for both round trip time and the average travelling time are shown in Table 3 below.

Table 3: Calculation and Monte Carlo simulation results for the round trip time and the average travelling time (all results in seconds).

Floor height used	Round trip time		Average travelling time	
	Calculation	Monte Carlo simulation	Calculation	Monte Carlo simulation
Average of all floor heights (4.5 m)	225.11	225.10	89.76 s	89.83
Effective floor height 4.62 m using equation ()	227.96	227.96	90.55 s	90.55

Using the effective floor height results in a difference of around 3 seconds for the round trip time and a difference of around 1 second for the average travelling time. Moreover, the Monte Carlo simulator is giving identical results to the calculation method of the amended equation.

5. THE EFFECT OF THE POISSON PASSENGER ARRIVAL MODEL

Further investigation is carried out in this section of the effect of the passenger arrival model on the round trip time and the average travelling time. Table 4 shows the average travelling time and the round trip time for a number of buildings using for both the constant passenger arrival model and the Poisson arrival model. It can be seen that the assumption of a Poisson arrival model results in a small reduction of the values of the round trip time and the average travelling time.

Table 4: Round trip time and average travelling time for the two passenger arrival models.

N	*P*	Analytical Equation, assuming constant arrival process (equation (7))	Monte Carlo Simulation (assuming constant arrival process)		Monte Carlo Simulation (assuming Poisson arrival process)	
		att	*att*	τ	*att*	τ
10	6.4	48.19	48.18	114.26	47.37	111.72
10	16.8	74.46	74.40	170.82	73.90	169.49
13	6.4	53.65	53.67	131.27	53.08	128.83
13	16.8	84.00	84.00	197.40	83.36	195.75
16	10.4	73.27	73.27	180.98	72.60	178.80
16	20.8	101.97	101.97	241.80	101.25	240.21

In general, as the number of passengers changes, the Poisson arrival model results in a smaller value of the round trip time and the average travelling time, as shown in Figure 1 and Figure 2 respectively.

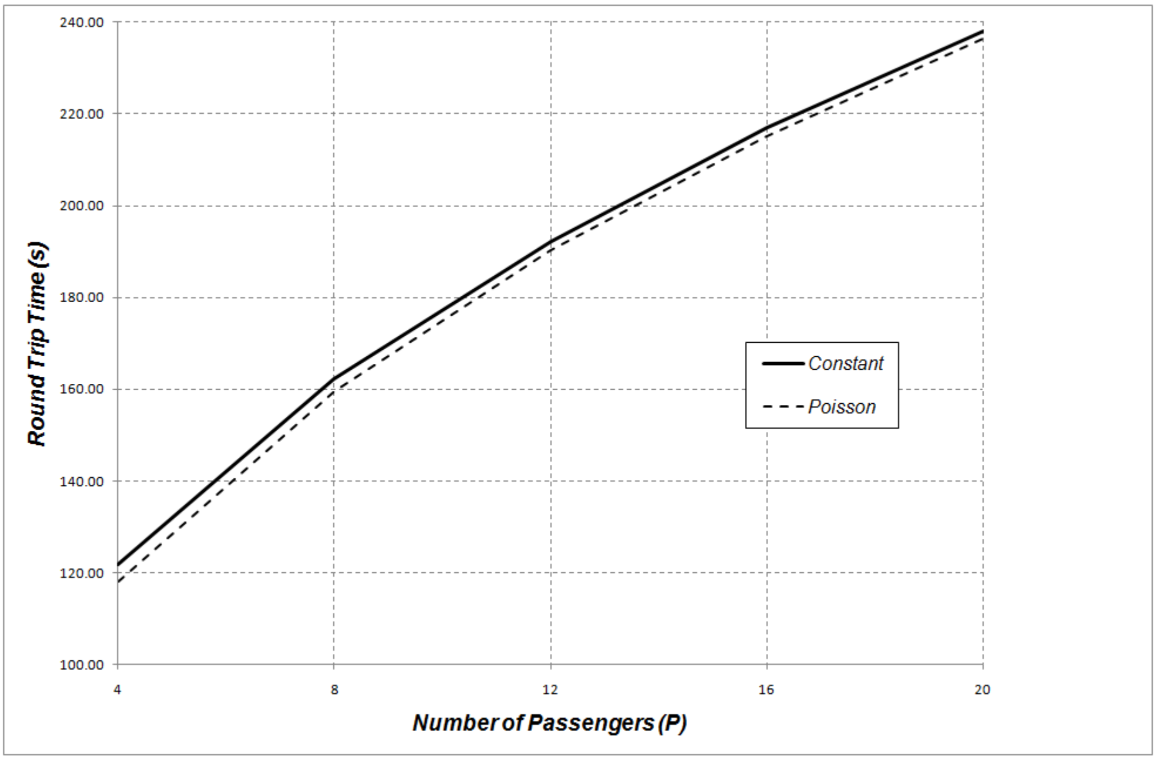

Figure 1: Round Trip Time for a 16 floor building for both constant and Poison arrival passenger models.

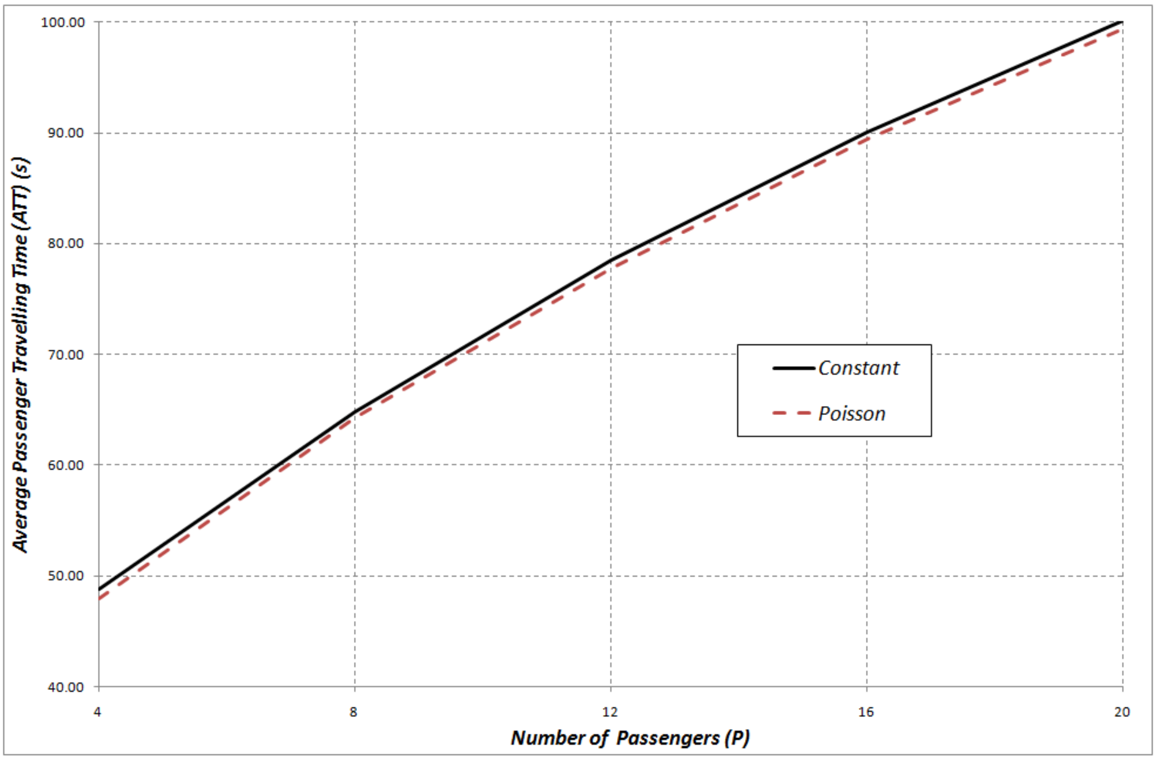

Figure 2: Average travelling time for a16 floor building under constant and Poisson passenger arrival models.

6. PRACTICAL EXAMPLE

In order to illustrate the use of the Monte Carlo Simulation method in the elevator traffic design, the following practical example is presented. The example is shown in order to illustrate the use of the method for the combination of all of the following special cases:

a. Constant passenger arrival model.
b. Unequal floor populations.
c. Unequal floor heights.
d. Top speed not attained in one floor journey.
e. Multiple entrances.

An office building has an arrival rate (*AR%*) of 12%. It is desired to design the elevator system such that a target interval of 30 seconds is achieved. The automated design method developed in [17] is used for the design and the Monte Carlo simulation is used to calculate the round trip time as shown in [1].

The following parameters are used:

t_{do} = 2 s
t_{dc} = 3 s
t_{sd} = 0.5 s
t_{ao} = 0 s
t_{pi} = 1.2 s
t_{po} = 1.2 s
v = 4.0 m/s (top speed will not be attained in one floor journey [16])
a = 1.0 m/s^2
j = 1.0 m/s^3

Table 5: The floor heights, populations and arrival rates for a building with 20 floors above ground.

Floor #	$d_f(i)$ (m)	*Entrance arrival percentage*	*Population*
L20	4	-	30
L19	4	-	38
L18	4	-	38
L17	4	-	38
L16	4	-	38
L15	4	-	38
L14	4	-	38
L13	4	-	38
L12	4	-	38
L11	4	-	38
L10	4	-	38
L9	4	-	38
L8	4	-	38
L7	4	-	38
L6	4	-	38
L5	4	-	38
L4	4	-	100
L3	6	-	100
L2	6	-	100
L1	6	-	100
G	8	70%	-
B1	3.2	10%	-
B2	3.2	10%	-
B3	3.2	10%	-

The resultant design is shown below:

- Constant passenger arrival model
- Round trip time: 177.72 s
- Average travelling time: 71.73 s
- Number of elevators: 7
- Target interval: 30 s
- Actual Interval: 25.39 s
- Actual passenger P: 10.15 passengers
- Car capacity: 13 passengers 1000 kg
- Car loading: 78%

7. NOTES ON CONVERGENCE OF THE MONTE CARLO SIMULATOR

In this section, some analysis is carried out on the convergence of the final result from the Monte Carlo simulator as used to calculate the round trip time and the passenger average travelling time.

In order to achieve better accuracy, the number of trials can be selected. The round trip time results for a sample building are shown in Table 6. For each number of trials, the analysis is carried out 10 times.

Table 6: Effect of the number of trials on the calculation of the round trip time using the Monte Carlo Simulator.

	Number of Trials					
	10	**100**	**1000**	**10000**	**100000**	**1000000**
Readings for the round trip time (s)	150	154.7813	153.8286	154.1935	154.1368	154.1514
	153.87	153.8205	154.3263	154.1499	154.205	154.1547
	152.745	152.9183	153.6842	153.8559	154.1622	154.1546
	155.7375	152.6933	153.7789	153.9662	154.1579	154.1587
	154.6125	153.4088	154.1551	154.0913	154.1548	154.1553
	156.3	155.4473	153.8216	154.2747	154.1166	154.1585
	156.5475	154.0455	154.0831	154.1364	154.1485	154.1510
	162.2175	153.3323	154.5007	154.1614	154.2053	154.1533
	156.5475	153.708	154.4249	154.1944	154.1861	154.1614
	147.75	155.049	154.2289	154.1461	154.1513	154.1557

The results of all the Monte Carlo Simulations are plotted as a scatter diagram in Figure 3 in order to visually convey the relationship between the accuracy of the method against the number of trials. The effect on accuracy of the final answer against the number of trials is plotted in Figure 4. Based on the results in the figure, 100 000 trials are required for accuracies better than ±0.1%.

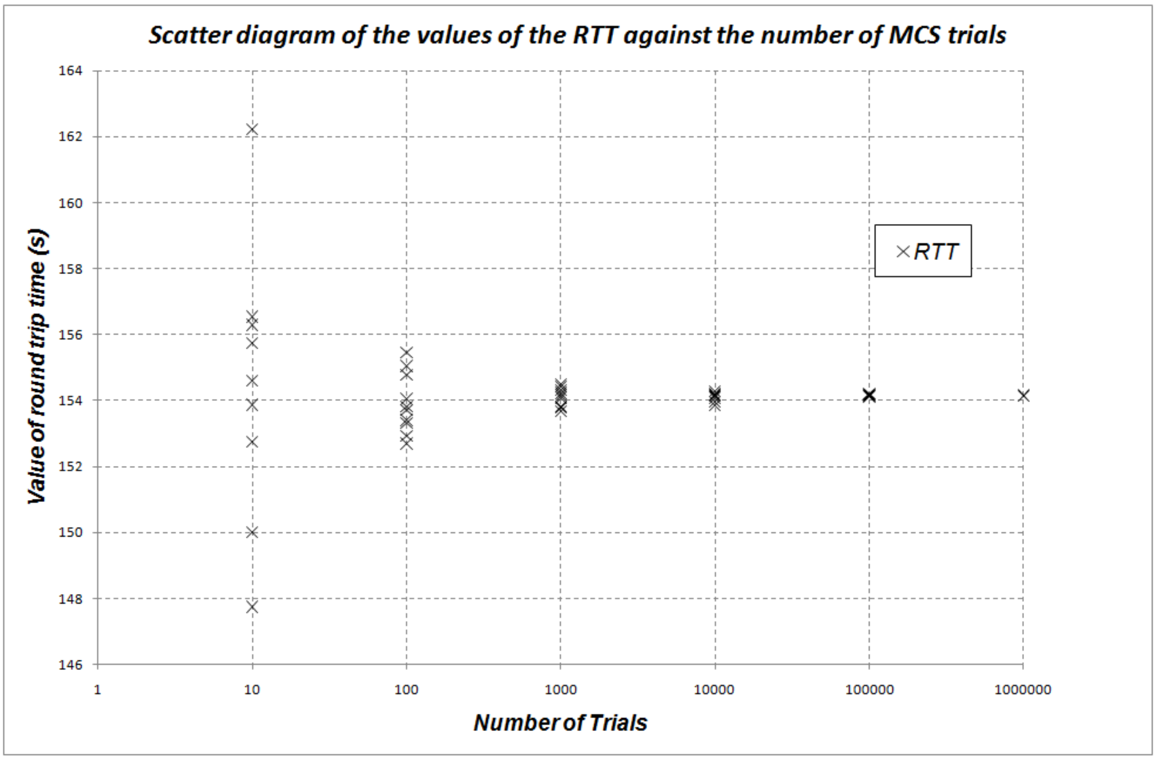

Figure 3: Convergence of the value of the round trip as the number of trials is increased.

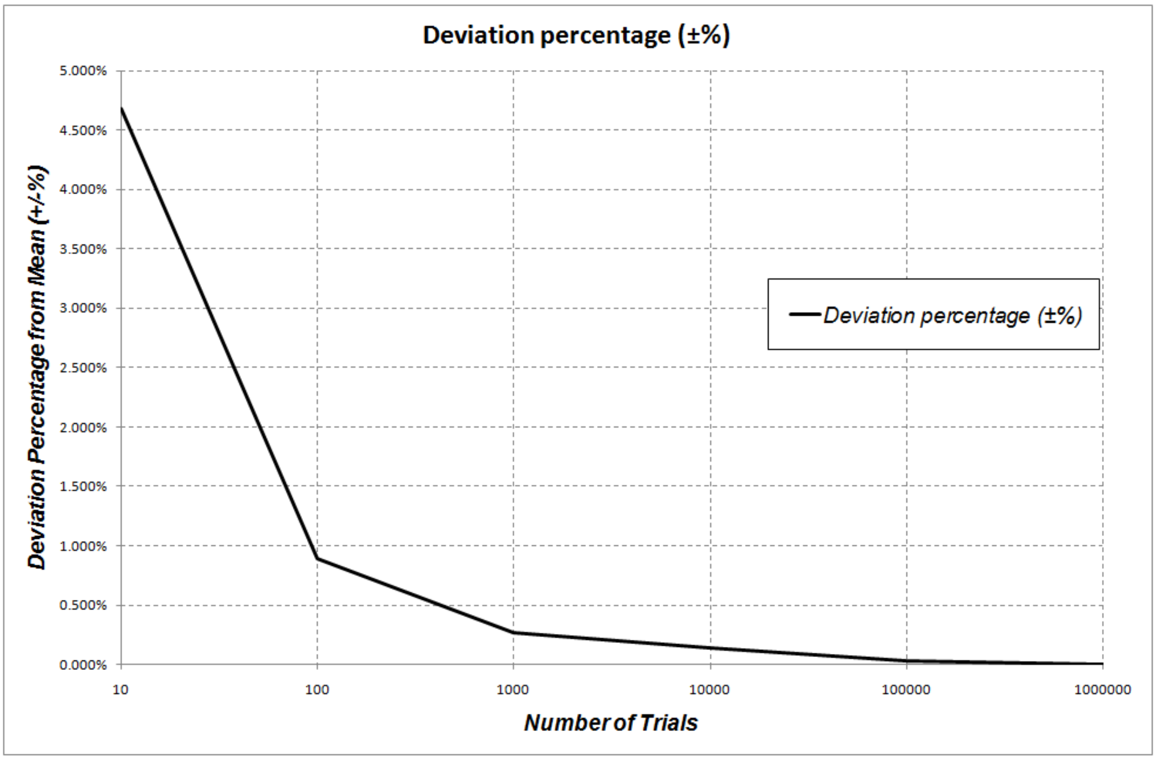

Figure 4: Deviation percentage of the RTT from the mean against the number of trials.

For the example above, an analysis is shown of the running time for the increased number of trials and the resultant accuracy, as shown in below. This provides a guide to the designer in terms of trading off accuracy with running time.

It is worth noting that these running times are based on the running of MATLAB code. Use of other tools, such as C++ for example, would provide much faster software, significantly reducing the running time.

Table 7: Accuracy of the results for different number of trials and the required running time for the Monte Carlo Simulation for the example used.

Number of iterations	Percentage deviation from the mean	Running time (s) (for the example of 10 floors above ground, 13 passengers)
10	±4.678%	<1
100	±0.895%	<1
1000	±0.265%	<1
10000	±0.136%	<1
100000	±0.029%	7
1000000	±0.003%	70

8. CONCLUSIONS

Monte Carlo simulation has been used to calculate the average passenger travelling time in an elevator system under up peak traffic conditions. The results of the Monte Carlo simulation have been verified for the simplest cases using an analytical formula for the average travelling time that has been derived. This verification showed good agreement.

The analytical equation was further developed to deal with the case of unequal floor heights, and further verification was carried out with good agreement. The analytical equations for the average travelling time can be applied to the cases of unequal floor populations and Poisson passenger arrival model.

The strength of the Monte Carlo simulation comes to the fore when the combination of all the five special conditions exists in a building: unequal floor heights; unequal floor populations; multiple entrances; Poisson arrival model and top speed not attained. A practical design example is given to show how the method can be used to calculate the round trip time and the average travelling time.

Commentary is given on the rate of convergence of the method, and the effect of the number of trials on the accuracy of the result. A guide is provided to the designer as to the trade-off between the number of trials, accuracy of the method and the running time.

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Symposium on Lift and Escalator Technologies

Some thoughts on Progressive Safety Gears and Modernisation

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INTRODUCTION

Where a lift has been subject to a modernisation programme, or, more particularly, one or more successive cab refurbishments, resulting in a change of car mass, it is essential that the continued integrity and compliance of the safety gear be confirmed before the lift is returned to service.
In the European context, EN81-1: Annex D specifies a commissioning test with 125% rated load and travelling at rated speed or lower. This test does not check the free fall performance. It is simply a test to ensure that the safety gear has been installed correctly and is functional. Consequently, after a modification it is not sufficient simply to perform the confirmatory test specified in Annex D.

However, there is no currently accepted method to establish free fall performance on the basis of a test with intact suspension. The objective of this paper is to discuss why that may be the case, and to explore possible ways in which, whilst it may not be possible to establish an accurate measure of free fall deceleration, nevertheless, it might be possible, in some circumstances, to establish with a reasonable degree of confidence, whether or not a given installation would have a free fall deceleration within the range required by EN81-1/EN81-2,

THE DYNAMICS OF SAFETY GEAR OPERATION

The dynamic model we shall employ for safety gear operation is shown in Figure 1. In the case of free fall, Figure 1(a), the dynamic equation on the car side will be

$$F_{SG} = (P+Q)(g_n + a_{FF}) \quad \text{.......} (1)$$

whilst in free descent, Figure 1(b), the dynamics will be

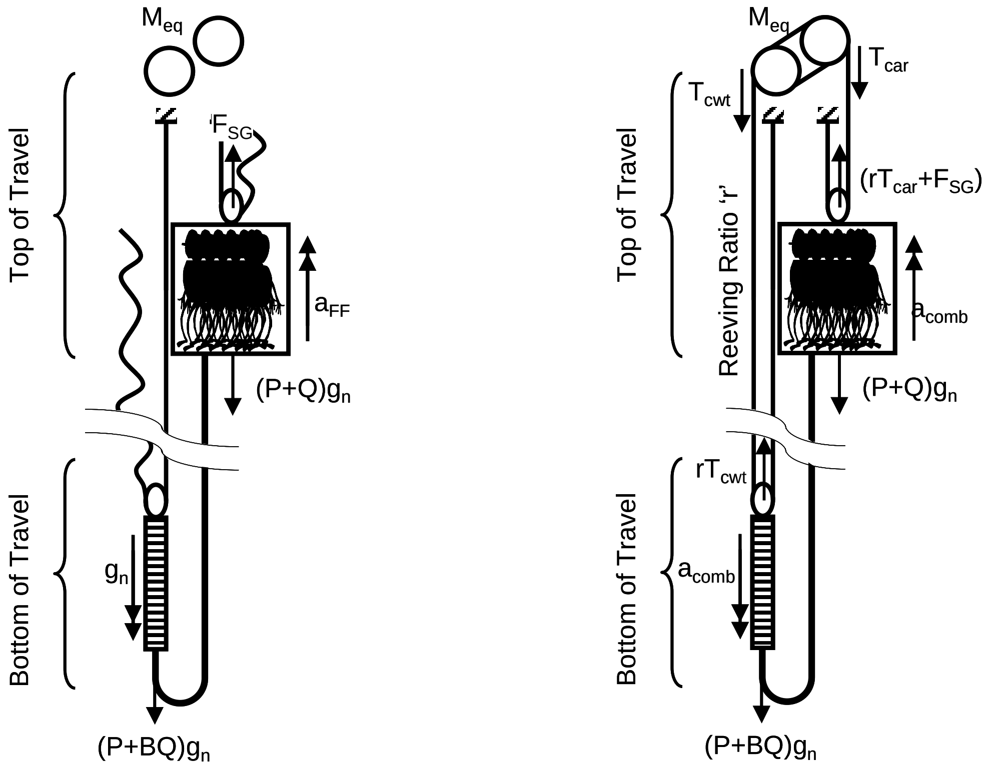
$$rT_{car} + F_{SG} - (P+Q)g_n = (P+Q)a_{comb} \quad \text{.......} (2)$$

$$(P+BQ)g_n - rT_{cwt} = (P+BQ)a_{comb} \quad \text{.......} (3)$$

where the car side 'fixed' mass P is taken to include the mass of ropes on the car side and an appropriate proportion of the travelling cable mass. Combining equations (1), (2) and (3), and eliminating the safety gear force F_{SG} :

$$a_{FF} = \frac{r(T_{cwt} - T_{car}) + (2P + (1+B)Q)a_{comb} - (P+BQ)g_n}{P+Q} \quad \text{.......} (4)$$

Whilst equation (4) defines the relationship between a_{FF} and a_{comb}, the presence of the term $r(T_{cwt}\text{-}T_{car})$ raises a significant difficulty, since, in any particular case, the magnitude of this force is difficult to determine. During an emergency arrest by the car safety gear, rope tension is dependent mainly upon the characteristics of the lift machine. Note that we have assumed for the purposes of this part of the discussion that the descent is arrested entirely by the safety gear, without assistance from the electromagnetic brake or any dynamic braking arrangements.



(a) Free fall **(b) Free descent**

Figure 1 : The dynamic model

The system has a number of rotating inertias which, referred to the traction sheave, we shall designate by an overall equivalent mass M_{eq}, and has an efficiency η less than 100%. In order to accelerate this equivalent mass in the 'car-up' direction the roping system must develop a tension difference (T_{cwt}-T_{car}) such that

$$T_{cwt} - T_{car} = \frac{M_{eq} r a_{comb}}{\eta} \quad \text{...} (5)$$

We can, of course, now eliminate (T_{cwt}-T_{car}) between equations (4) and (5) to express the relationship between free fall deceleration a_{FF} and free descent acceleration a_{comb} (normalised in terms of g_n) entirely in terms of the system parameters

$$\frac{a_{FF}}{g_n} = \frac{\left(1 + B + 2\frac{P}{Q} + \frac{r^2}{\eta}\frac{M_{eq}}{Q}\right)}{1 + \frac{P}{Q}} \frac{a_{comb}}{g_n} - \frac{B + \frac{P}{Q}}{1 + \frac{P}{Q}} \quad \text{....................................} (6)$$

Nevertheless, expression (6) still does not help the problem. On a new installation it could be expected that all the system parameters are known, allowing a reasonably accurate estimate of M_{eq} and η. However, when it comes to a reconstruction or modernisation, if the safety gear and lift machine are to be re-used but the system parameters have changed (e.g. because of a change in system mass), since the machine parameters may be lost in the mists of time, a safety gear test at rated load with intact suspension is unlikely to give an accurate guide to the free fall deceleration, particularly when we consider that in the case of a geared system, the (unknown) value of M_{eq} may be significantly in excess of either P or Q, and will, at the very least, be of comparable magnitude.

ALTERNATIVE APPROACH

If, during safety gear operation, the car side deceleration exceeds g_n (i.e. by engaging the safety gear with a partial, or no load in the car), then the car is decoupled from the remainder of the system. Consequently, the average deceleration during such a test will indicate the free fall performance of the safety gear at that particular state of load. Assuming that the average safety gear force, F_{SG} is constant, independent of the total car side mass, this will allow us to estimate the free fall deceleration $a_{FF.}$

Based on a mid-range ratio of car side mass to rated load P = 1.6Q (allowing for rope mass in the value of P), Figure 2 indicates the range of free fall safety gear setting and partial load ($0 \le q \le 1$)for which the car deceleration a_{FF} would be equal to g_n based on the expression

$$\frac{F_{SG}}{Qg_n} = \left(\frac{P}{Q}+1\right)\left(1+\frac{a_{FF}}{g_n}\right) = \left(\frac{P}{Q}+q\right)\left(1+\frac{a}{g_n}\right) \quad \text{.....................................} (7)$$

In the case of a modification to the car, we will assume that the modification to the design has established the revised car mass, and consequently the car side fixed mass P including, as before, the relevant proportion of the mass of compensation ropes/chains, suspension ropes, travelling cables etc.

With a partial load, if the system is subjected to an overspeed test, then 'a' ($\ge g_n$), the average deceleration value during stopping may be measured. After the car has stopped, it must be established that the safety gear has engaged fully, otherwise the test must be repeated with the car loaded to a point where this will occur. The difficulty is, of course, that it is not guaranteed to achieve a deceleration $\ge g_n$ with a partial load sufficient fully to engage the safety gear.

However, assuming that this hurdle is overcome, expression (7) can be applied to provide an estimate of the free fall acceleration value a_{FF} :

$$\frac{a_{FF}}{g_n} \approx \frac{\left(\frac{P}{Q}+q\right)\frac{a}{g_n} - (1-q)}{\left(\frac{P}{Q}+1\right)} \quad \text{...} (8)$$

If a no load test fully engages the safety gear, this expression simplifies to

$$\frac{a_{FF}}{g_n} \approx \frac{\frac{P}{Q}\frac{a}{g_n} - 1}{\left(\frac{P}{Q}+1\right)} \quad \text{..} (9)$$

Provided that the calculated value of a_{FF} falls well within the permitted range (for EN81-1; $0.2g_n \le a_{FF} \le 1.0g_n$), then it may be assumed that the free fall performance of the safety gear with the revised car mass will comply. Clearly, if the calculated value of a_{FF} is close to the lower limit ($0.2g_n$), then the result must be treated with caution, since the methodology will give no more than an approximation to the actual value for a_{FF}. Furthermore, such a test must be treated with extreme care, and the test does not obviate the necessity for an "Annex D" test. It may require several tests to establish the maximum load at which the safety gear will impose a deceleration $\ge g_n$. Multiple testing of safety gears is absolutely not recommended. Finally, if the lift is subjected to a car side deceleration in excess of g_n, then counterweight jump will occur, with consequent severe shock both to the suspension and to the elevator machine, particularly a geared machine, and may lead to internal damage in a gearbox.

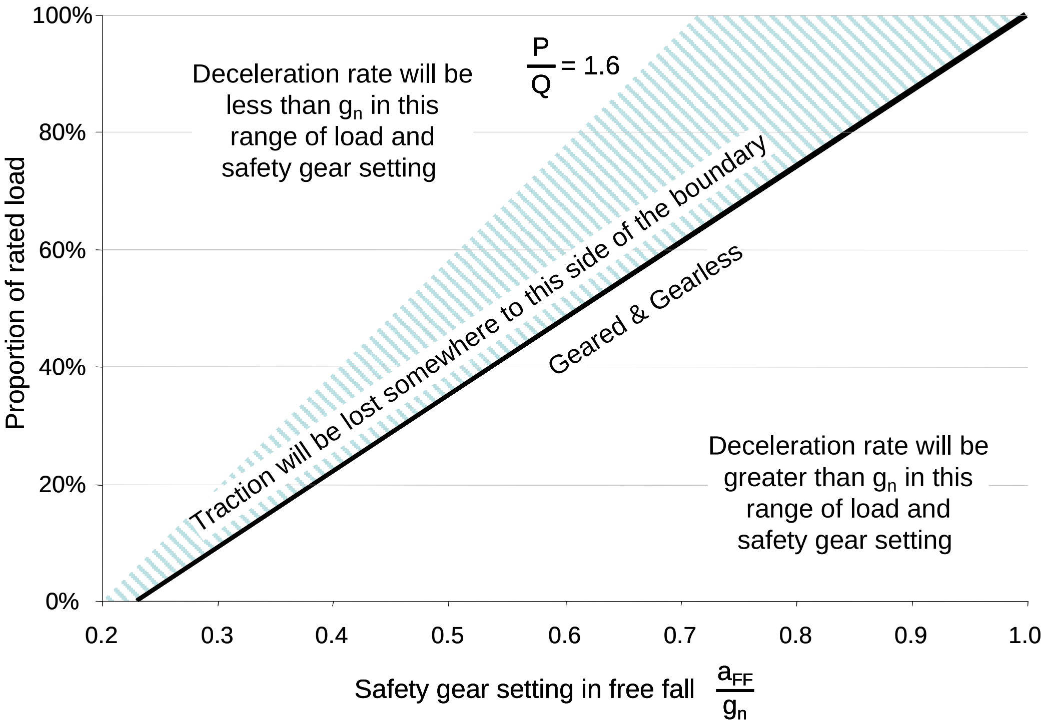


Figure 2 : Safety gear performance in free descent

POINT FOR DISCUSSION

It is becoming clear [1,2] that North American practice is quite relaxed about the prospect that a progressive (Type B) safety gear may have a setting which, whilst it will arrest a car in free descent, may or may not arrest a free fall. The North American view seems to be that if a performance specification similar to that required by EN81-1 were to be adopted, 'typical' safety gear settings would be increased, with the consequence that during 'real time' emergency stops, with a partial passenger load, an intact suspension and with both electromechanical brake and any dynamic braking circuit assisting with the arrest, the more severe deceleration rate is likely to result in a greater incidence of passenger injury. Given the strict inspection regimes extant in North America, the probability of a suspension failure can, to all intents and purposes, be discounted, not withstanding the catastrophic nature of the hazard. Given that higher speed lifts are, in the main, controlled by systems not linked to mains frequency, the probability of 'uncontrolled overspeed' of the system is much higher than that of suspension failure. It is thus considered more important to protect passengers, as far as possible, from injury consequent upon severe deceleration during an arrest with intact suspension, than to guarantee arrest in free fall.

Furthermore, given our discussion of safety gear testing, after a modification, the test specified in A17/B44 (and BS2655 : Part 1), allows a site test to confirm compliance, whilst acknowledging that if the stopping distance under such a test is at or near the maximum permitted, then the safety gear may not be capable of arresting the car in free fall.

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Symposium on Lift and Escalator Technologies

Energy Models for Lifts[1]

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ABSTRACT

"*Energy modelling is a complex subject*" – Peters *et al*, 2004 [3]

The intention of this paper is (1) to explain some work which is being carried out at the International Standards Organisation (ISO) level (2) to suggest a simple energy reference model to support this work; and (3) to develop a simple energy model that could be employed in a public domain traffic simulation program to predict energy consumption.

1 ISO DRAFT STANDARD DIS/25745-1

A Working Group of an International Standards Organisation's Technical Committee (TC178/WG10) has developed a draft standard DIS/25745-1 "*Energy performance of lifts, escalators and moving walks – Part 1 Energy and verification*". This standard sets out the procedures to be used when making energy measurements and verifying that energy usage during the life cycle of a lift installation. It does not grade, or provide energy certification for lifts, escalator and moving walks as happens now for boilers, refrigerators, washing machines, etc.

The Working Group has proposed a simple pragmatic procedure that should be easy to carry out, uses readily available measuring equipment, is repeatable, and allows periodic verification checks to be carried out.

2 ENERGY MEASUREMENT FOR A ISO REFERENCE CYCLE

The proposal is to measure the running energy consumed by a lift during a ISO Reference Cycle. The ISO Reference Cycle comprises running an empty lift car from one extreme landing (highest/lowest) to the other extreme landing (lowest/highest) and back again. The lift carries out one cycle of its normal door operations at each terminal landing. These include opening, closing and dwell times. The energy consumed for at least ten cycles should be measured and an average energy consumption value (in Wh) for a single ISO Reference Cycle determined.

Care needs to be taken to ensure all the energy used to operate the lift is included. For example sometimes the main power and the ancillary power (lights, fans, alarms, trickle chargers, displays, etc.) are often supplied by separate feeders. Non lift function energy consumers such as car and machine room heating, cooling and lighting are not to be included.

After the terminal landings cycling test the lift should be maintained stationary, for five minutes, at one terminal landing. A power measurement (in W) can then be made. This gives the standby power consumed. "Green" lift equipment manufacturers will thus be sure to reduce the idle power consumption by turning off all energy hungry lighting and controllers within this five minute period of grace.

The procedure just described requires a fairly sophisticated energy/power measuring instrument together with a skilled operator. So the second part of the standard indicates how to verify continuing energy consumption. This can be achieved by measuring the line currents at the same time as the energy measurements are made. Later an inexpensive, simple current meter (amp

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probe), applied by the less skilled service mechanic, can be used to detect any changes in the energy consumption. For example, the car might become heavier if it was re-fitted with mirrors (more energy consumed); or the less energy demanding if the incandescent car lighting were to be replaced by low energy units (less energy consumed). Or the door timings might have changed.

The currents that are measured for the verification check do not necessarily need to be exactly in proportion to the energy graph as the power factor (cosφ) values at the different car loadings will vary. However, if as time passes these current values do not change, then it can be assumed that the energy consumption remains the same as it was when first measured and the verification current readings taken.

This energy/power/current measurement procedure can be part of the final commissioning tests for a new lift and could be carried out for an existing lift on request.

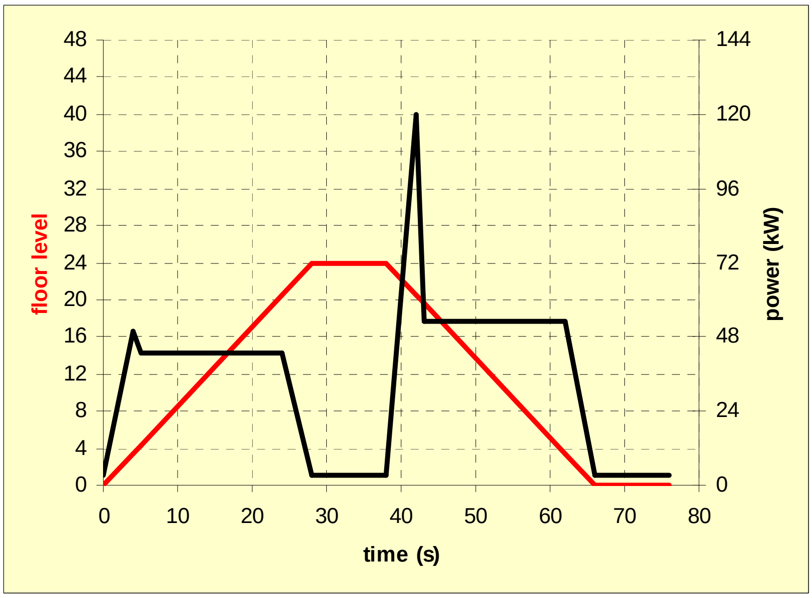

Figure 1 Idealised ISO Reference Cycle

Figure 1 illustrates an idealised ISO Reference Cycle where the empty lift moves from the lowest terminal floor (red line) to the highest terminal floor, carries out its door operations, returns to the lowest terminal floor and carries out its door operations. The power consumed (black line) shows a lower, power consumption as the empty car moves up, under the influence of the (heavier) counterweight, than when the empty car moves down, pulling up the counterweight.

3 ENERGY REFERENCE VALUES FOR A LIFT

The lift now has two measured values: one for the running energy consumed (Wh) during a ISO Reference Cycle and another for the power consumed (W) when in standby mode. These figures apply only to the lift that has been measured and no other. No two lifts are the same even if they share the same rated load and rated speed and are in the same building. Obvious differences include: the travel distance between terminal landings, different door operating times, no of entrances, the counterbalancing ratio, the weight of the car, car balance, the type of guide shoes, roping factor, number of car entrances, drive system, effect of the maintenance regime, etc, etc.

If a purchaser of a lift wishes to be seen to be "green", or is required to be by the terms of any building energy certification process, then the two reference figures should be obtained before an order is placed.

So where do these figures come from?

It is expected that suppliers will know their product sufficiently well (after all they have sized the drive machine and the indicated the supply cable specification, etc.). It is also to be hoped that they will have energy models available for their products and thus be able to easily supply these two figures. Of course the purchaser will confirm them at the time of final test. Energy consumption could thus become a selection criterion between manufacturers.

4 THE ISO REFERENCE CYCLE

How can a simple ISO Reference Cycle model be developed?

Figure 1 shows an idealised ISO Reference Cycle, which comprises four main parts:

(1) power consumption for an empty car travelling up (28 s)
(2) door operation at the highest landing (10 s)
(3) power consumption for an empty car travelling down (28 s)
(4) door operations at the lowest landing (10 s)

The parts (1) and (3) are further subdivided. There is a peak power on start up, which reduces to the running power when rated speed is reached. At the end of the running time the power falls to the idle power (1.0 kW). Remember idle power is not standby power. It is the power consumed between the lift running and it entering the standby mode of operation.

The energy consumed during the ISO Reference Cycle is the area under the graph in Figure 1, in watt-hours (Wh). This can be simply calculated as a set of triangles and rectangles.

5 OBTAINING DATA FOR THE MODEL

The data required are:

Peak power up empty
Running power up empty (at rated speed)
Peak power down empty
Running power down empty (at rated speed)

Time to reach peak power up
Time to reach rated speed up
Time to reach peak power down
Time to reach rated speed down

Door timings

The idealised graph in Figure 1 assumes:

> the time to reach the peak power from starting up (or down) is equal to the theoretical time to reach the rated speed (*tvm*)

and

> the time from reaching peak power to falling to the running power value (at rated speed) is equal to 1.25 *tvm* (125%).

These are reasonable approximations. The time (*tvm*) to reach rated speed (*vm*) is given by:

$$tvm = \frac{vm}{a} + \frac{a}{j}$$ (source CIBSE Guide D: 2010, A2-2 [2])

where: a is the value for acceleration (m/s^2); j is the value for jerk (m/s^3)

6 EXAMPLES OF THE USE OF A SIMPLE ISO REFERENCE CYCLE MODEL

Al-Sharif, Peters and Smith [3] in 2004 obtained data for a lift with a rated load of 1800 kg and a rated speed of 2.0 m/s with 42% counterbalancing. The lift had a regenerative drive. Power data for up and down movements with 0%, 25%, 50% 75% and 100% car loads were obtained. Figure 2 shows a graph of this installation using the data for an empty car (0%) given in Table 1 for a car starting at the highest terminal floor.

Table 1 Spot data for a regenerative drive system (1800 kg)

Car load (kg)	Car load (%)	Power running down (kW)	Power starting down (kW)	Power running up (kW)	Power starting up (kW)
0	0	23	30	– 9.0	3.0

The numbers are rounded for simplicity. The idle power is 2.0 kW.

So what does a simple energy model using this data look like? A plot of the power used by an empty car for a downwards trip would look something like the Figure 3, which is a close facsimile of Figure 2. The calculation of energy used (the area under the curve) gives:

Running energy down	110.7 Wh
Running energy up	-34.9 Wh
Total running energy	75.8 Wh
Door operations	8.9 Wh
Total energy	84.7 Wh

As the ISO Reference Cycle occupied 56 seconds, if cycling had continued for one hour (about 64 Reference Cycles, 128 stops) the energy consumed would be 5.4 kWh (cost about £0.54 at 10p per kWh).

Figure 4 is example based on measurements for another lift. It shows an ISO Reference Cycle for an empty car trip down and then up between terminal floors. This lift has a rated load of 1500 kg, a rated speed of 4.0 m/s, is in a 24 floor building with a 62 HP (46.3 kW) hoist motor, 50% counterbalanced. The black line shows the power consumed. The plot is idealised, an actual plot will have irregularities similar to those shown in Figure 2. The values are from the empty (0%) car load row in Table 2.

Table 2: Data from actual (1500 kg) installation used to obtain Figure 4

Car load (kg)	Car load (%)	Power (kW)			
		running down	starting down	running up	starting up
0	0	53	120	43	50
750	50	13*	100	13*	100
1500	100	43	70	53	130

* 3.0 kW is controller plus ancillaries =10 kW for inefficiency.

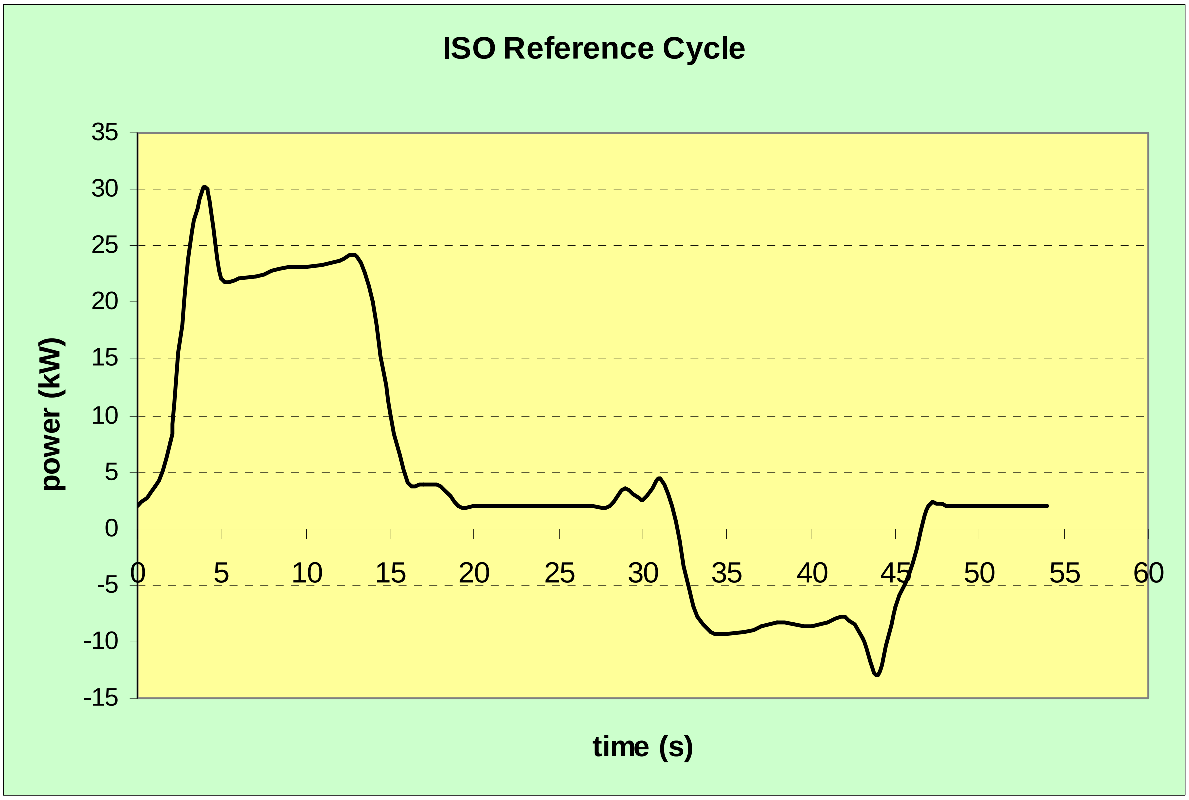

Figure 2: An ISO Reference Cycle for an empty car (1800 kg lift)

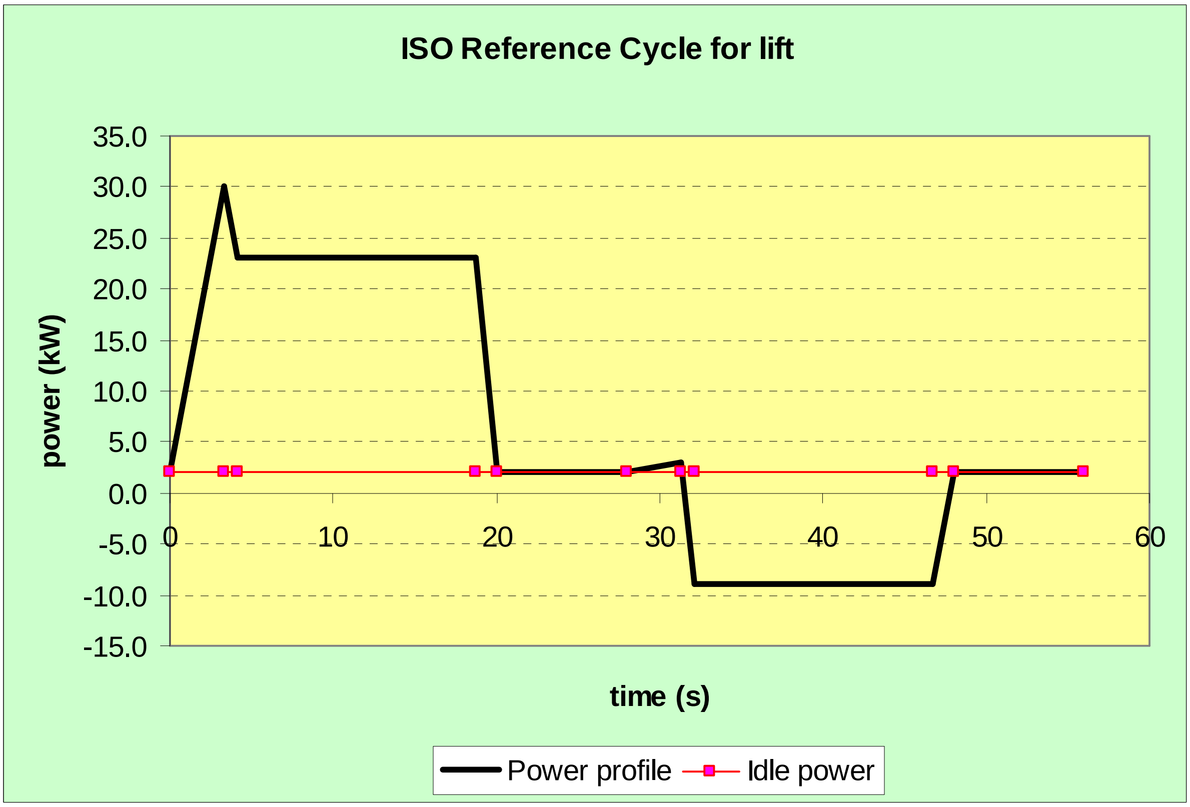

Figure 3 Model plot for installation of Figure 2 (1800 kg lift)

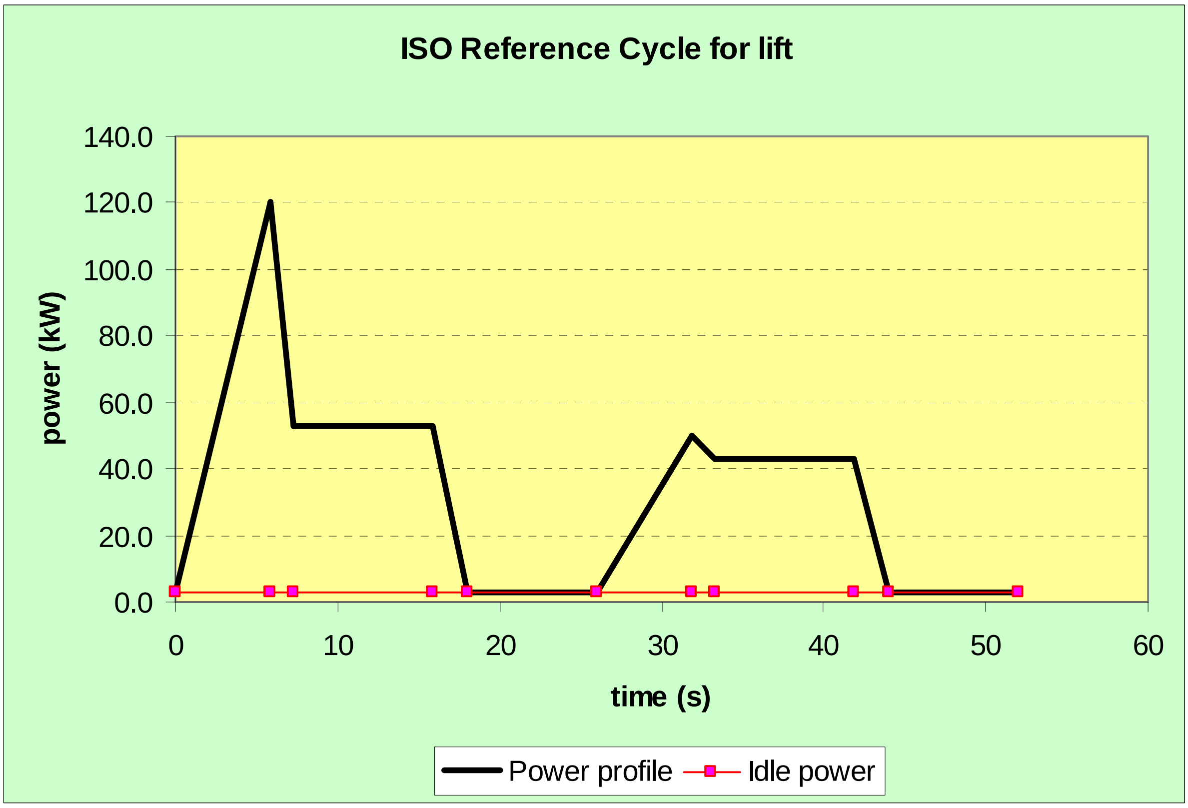

Figure 4 Reference cycle for a 24 floor office building (1500 kg lift)

Table 2 also shows the power required for starting and running for the car loads of 50% (balance) and 100% car loading in both directions of travel. These entries were obtained from the record made[2] when the lift was tested in 1993.

It is interesting to note that at balanced load (50%) the power taken is 13 kW. This is made up of 3 kW idle power supplying the controllers and ancillaries and 10 kW to overcome inefficiencies.

7 TRAFFIC PATTERNS

No one can predict the usage pattern of a lift (Barney: 2003 [1]). It is a bit like predicting how the stock market will perform. Many assumptions are made by experienced traffic designers when sizing a lift installation. This is why some naïve developers get it wrong as they lack that experience.

Traffic simulators are used to study the behaviour of a particular design. Thus it would be useful to be able to study the energy behaviour at the same time.

This is possible as a lift traffic simulator "knows" the passenger load in the car, the direction of travel, the number of passengers entering/leaving, the travel distance, door timings, etc. If the power used for each individual car load and each individual direction and distance of travel were known (they could be in a matrix) then the simulator could estimate energy consumption.

To insert an energy model into a traffic simulation program requires more data than that shown in Table 2. However, Table 2 provides enough entries to establish Table 3, by assuming a linear

[2] The document used was BS5655-10: 1986 "Certificate of test and examination for lifts" and the data was recorded in Section A5(c) "Measurement of the electrical system" for empty, balanced and fully loaded cars. The latest test documents (PAS32/BS8486) do not record such data).

relationship between the grey cell entries. In practice the relationship will be nonlinear. Thus a simple table can be developed for use in a traffic simulator.

Table 3: Extended entries (1500 kg lift)

Car load (kg)	Car load (%)	Power (kW)			
		running down	starting down	running up	starting up
0	0	53	120	43	50
75	5	49	118	40	55
150	10	45	116	37	60
225	15	41	114	34	65
300	20	37	112	31	70
375	25	33	110	28	75
450	30	29	108	25	80
525	35	25	106	22	85
600	40	21	104	19	90
675	45	17	102	16	95
750	50	13*	100	13*	100
825	55	16	97	17	103
900	60	19	94	21	106
975	65	22	91	25	109
1050	70	25	88	29	112
1125	75	28	85	33	115
1200	80	31	82	37	118
1275	85	34	79	41	121
1350	90	37	76	45	124
1425	95	40	73	49	127
1500	100	43	70	53	130

8 EXAMPLES OF AN ENERGY MODEL IN A SIMULATION PROGRAM

8.1 Uppeak traffic

Consider Figure 5. This shows the spatial movements (red line) of the example 1,500 kg lift during the morning uppeak traffic demand. Table 4 gives the data used.

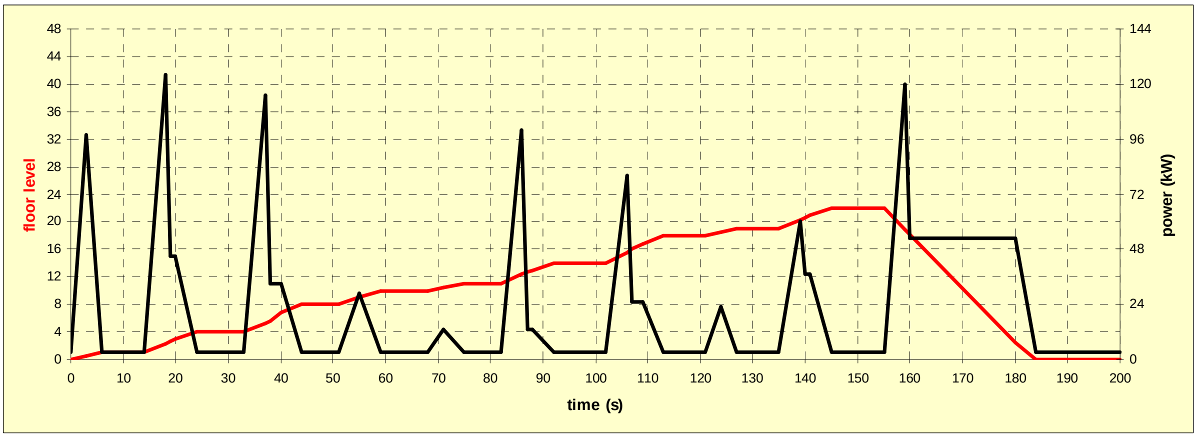

Figure 5 Power profile for a typical uppeak traffic pattern (1500 kg lift)

The lift leaves Floor 0 with 20 passengers and calls at nine floors with various numbers of passengers alighting. Thus the load reduces until the last passengers exit at Floor 22. The lift then returns empty to Floor 0. Note the balance load is achieved as the lift leaves Floor 11.

Where the lift only moves one floor, eg: 0>1, 10>11, 18>19 the graph shows a low peak power as rated speed is not reached (shown * in Table 4). Where the lift moves two floor, eg: 8>10 rated speed is just reached before the slow down sequence is initiated. In all other cases the lift reaches rated speed as indicated by the step in the profile, although it may only be for a short time, eg: 1>4, 19>22.

The energy profile has been idealised for the purpose of illustration. This would not be necessary in a simulation program as the actual profiles can be calculated. Once again the energy consumed is the area under the profile. This can be easily calculated by a simulation program.

Table 4: Data used to construct Figure 5 (1500 kg lift) (figures rounded)

Floor	Number of passengers			Car load (%)	Peak power starting (kW)	Running power (kW)	Total door operating time (s)
	In car on arrival at floor	Leaving car at floor	In car on departure from floor				
0	0	0	20	100	130/98*	n/a	0
1	20	2	18	90	124	45	8
4	18	3	15	75	115	33	9
8	15	1	14	70	29	n/a	7
10	14	3	11	55	17/13*	n/a	9
11	11	1	10	50	100	13	7
14	10	4	6	30	80	25	10
18	6	2	4	20	31/23*	n/a	8
19	4	2	2	10	60	37	8
22	2	2	0	0	120	53	8
Other data are: Time to reach rated speed: 4.0 s. Passenger transfer time 1.0 s per passenger. Flight times: one floor – 6.0 s, two floors – 8.0 s, three floors – 9.0 s, four floors – 10.0 s. Door open and door closing times: 3.0 s each. * Single floor jumps – peak not reached.							

8.2 Down peak traffic

Figure 6 shows a down peak traffic situation for the example 1,800 kg lift. It loads six passengers at Floor 20, which takes 10 seconds including door times. The lift then successively calls at Floors 19, 18 and 17 loading six passengers each. Because the flight time between two adjacent floors is only six seconds the peak starting currents are not reached and are estimated at 2/3rds of the measured peak. Once the lift leaves Floor 17 it regenerates power back into the mains supply. It should be noticed that of the 80 seconds from loading at Floor 20 until the lift arrives at Floor 0, the lift is only moving for 40 seconds.

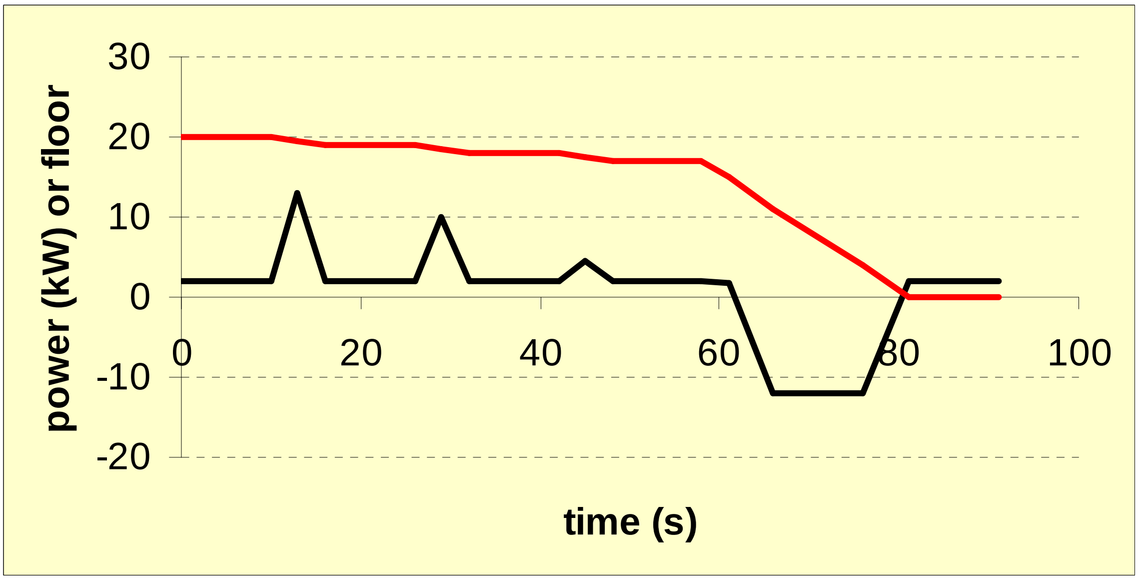

Figure 6 Power profile for a typical down peak traffic profile (1800 kg lift)

9 DISCUSSION AND CONCLUSIONS

The method for taking energy measurements of an actual system using the ISO Reference Cycle will be as accurate as the instruments used and the skill of the user. The same conditions apply to the electrical current measurements made for verification. The two measurements obtained should give a good view of how well a lift is performing at the time of measurement and over time.

Prediction of the two ISO numbers is not difficult. The simple energy model proposed, based on the ISO Reference Cycle, relies on a number of simplifications, as discussed in Section 5. Errors in the values used will affect the shape of the power/energy profile as shown in Figures 2 and 3. However the energy used in the peaks is small compared to that used when the lift is running. As the running power is likely to be known with good accuracy, little error should occur. In any case lift suppliers usually know their product very well and will have accurate values for all these parameters.

Energy usage prediction is much more difficult. The simple model proposed can be employed to calculate energy usage. More data is required, which used to be collected when a lift was commissioned (tested/adjusted). This data, as shown in Table 2, enables an interpolation on a linear basis. This is not strictly correct as electric motors are magnetic devices and exhibit significant nonlinearities. Using data such as that shown in Section 8 allows a reasonable attempt to be made to predict the energy used for SPECIFIC traffic patterns. A striking feature is how little energy is used.

It is important to note that a real energy profile varies with the direction of travel and car position in the well, and is not symmetrical, ie: exhibits nonlinearity.

Figure 5 shows an energy profile for uppeak traffic and Figure 6 shows the energy profile for a down peak traffic. These emulations are not precise, but, if the proposed model is embedded in a simulation program, then a more precise calculation can be made, which will be as accurate as the data provided.

The energy measurement of building services is being required more and more by various regulations, for example, in order to comply with the energy certification of buildings. Modern lifts (and some older ones too) are already very efficient, especially those based on counterbalanced systems. However, it is wise to prove this to energy inspectors and standard methods of energy measurements, conformance checking and modelling are necessary to do this.

It can be expected that third party[3] and manufacturer modelling and simulation programs will include energy modelling as the need for it arises. It will then be possible to more accurately predict energy usage. This can be particularly useful when considering energy reduction measures.

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[3] al-Sharif, L., Peters R. and Smith, R, "Elevator Energy Simulation Model", IAEE, Elevator Technology 14, April 2004.

WARNING

The data used to plot the graphs are based on real systems, but they have been idealised and the numbers rounded to illustrate the discussion.

[3] Visit www.peters-research.com

[4] Visit www.cibse.org

Symposium on Lift and Escalator Technologies

A Reliable Forecast of Lift System Wear

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INTRODUCTION

Lift System Condition Monitoring to support servicing activities has hitherto been restricted to calling up fault storages of the actual lift control systems and occasionally of the lift drive units, each manufacturer using its own concepts.
Sensors detecting the amount of wear in a lift system are currently used – if at all – in form of mobile systems only. They allow random tests to be made of the cab's acceleration behaviour, to determine how the noise is developing or to detect the rope tension and put respective measurements at the technician's disposal uncommented.
Such systems are used to conduct initial tests aiming at converting the interval-based maintenance activities common to lift systems into a condition-oriented or even proactive maintenance.

POINT OF DEPARTURE

Current maintenance strategies for lift systems. The lift system maintenance concept prevailing worldwide is a combination of reactive, preventive and in initial stages also condition-oriented servicing activities. Preventive maintenance of lift systems is carried out on the basis of intervals: within fixed intervals or after reaching a certain number of rides service technicians initiate measures to retard any further reduction of the wear potential e.g. by topping up gearbox oil, greasing the guide rails, etc. At the same time they usually check the degree of wear of certain lift components such as guide shoes or brake linings. The latter is already a first and simple attempt to carry out a condition-oriented service: based on the information available (e.g. of the wear), deadlines are determined on which components need to be replaced in order to prevent any unplanned system failure or even a safety-critical condition to develop.

Condition Monitoring in an industrial environment. Today and in nearly all industrial areas Condition Monitoring is one of the mainstays needed to efficiently operate and service technical plants. This concept is based on a regular and/or permanent recording of the condition of the machine by measuring and analysing meaningful physical parameters. The technological developments achieved in sensor technology, tribology and microprocessor technology allow an unparalleled quantity and quality of information to be used for the maintenance of production machinery. An industrial environment cannot be pictured without Condition Monitoring any more. It must more or less be regarded as a compelling requirement for a condition-oriented and/or proactive maintenance.

The benefit of Condition Monitoring. The more comprehensive the maintenance strategy and the requirements it has to meet, the more distinctive will be the significance of Condition Monitoring.
In trying to achieve maximum plant efficiency, Condition Monitoring can be of assistance in a number of ways:

- by improving the safety against failure on the basis of efficient forecasts relating to defects (and the resulting prevention),
- by minimising downtimes on the basis of an integrated planning of repair measures specified by the Condition Monitoring,

- by maximising the service life of components by preventing any conditions that shorten the life, and
- by a cost-reducing and nearly full use of the component's wear potential.

Condition Monitoring is composed of three steps:

1. determining the condition, i.e. measuring and documenting relevant machine parameters reflecting the current condition of the machine,
2. comparing the condition; reflecting the comparison of the actual condition with a specified reference value (with a growing plant complexity usually determined empirically) and
3. the diagnosis which has to use a comparison of the condition to pinpoint any possible fault as early as possible and to determine its cause.

CONDITION MONITORING IN LIFT SYSTEMS

Hardly any technical measuring systems are offered on the lift market for the first of the Condition Monitoring steps, the determination of the condition. It is only for the intermittent monitoring of vibration and noise data that ride quality measuring systems conform to ISO 18738 such as the EVA system[1] or the LiftPC system[2] can be used. These for example allow information on the condition of the system to be recorded at the time inspections are carried out and long-term developments to be established. But short-term or transient events cannot be detected and a link with other data such as the load condition, temperature, etc. is quite difficult.

A continuous monitoring of the physical lift system parameters in real time would allow long-term trends as well as erratic or transient changes in condition to be recorded. Any subsequent comparisons of the condition and diagnosis algorithms could then fall back on a comprehensive data stock and generate maintenance suggestions.

Condition Monitoring pilot project in lift systems. As early as 2004 Henning installed prototypes of a lift system condition monitoring system in eleven lift systems of the chemicals group BASF. Apart from acceleration and vibration sensors, also sensors monitoring the traction sheave speed, the current hoisting height, the overall cab load and the individual rope tensions were used. The measurements were analysed by an industrial personal computer located directly at the lift system and the results of this analysis were transferred by remote data transmission to a data centre. The main component of the Condition Monitoring system, a vibration and acceleration sensor, was directly fitted on top of the cab. In this position it could record the actual ride movements of the cab as well as the cab guides, door movements and - indirectly via the ropes - also the behaviour of the drive unit.

For each ride the recorded data of all sensors were converted to specific characteristic values and checked to see if they exceeded any limits. Then the characteristic values of each ride of one day were combined to one statistic mean value. These mean values resulting from several hundred rides per day were used for actual trend monitoring purposes. The following two examples e.g. show a trend over several days based again on thousands of rides.

[1] www.pmtvib.com
[2] www.henning-gmbh.de

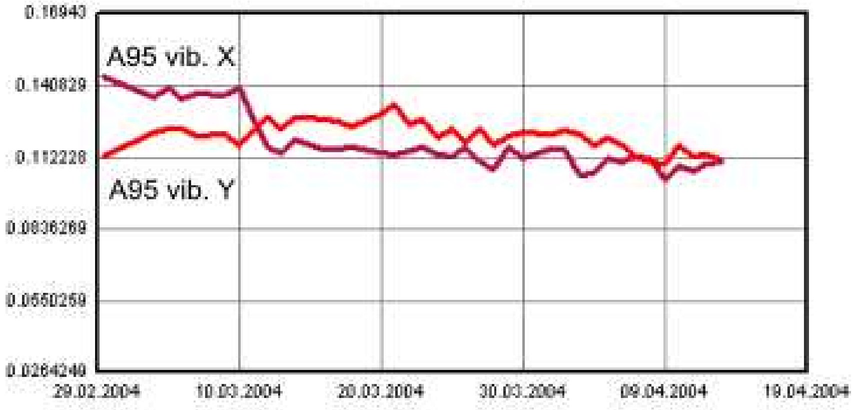

Figure 1: Vibration behaviour of the lift cab in the two horizontal directions in space. One clearly recognises the replacement of the cab guides on March 11.

At the start of the recording period shown in Fig. 1 the slide guides of the cab are already worn out. On March 11, 2004, the guides were replaced by new guide shoes. One clearly recognises that the vibrations in direction X (vertical to the actual distance between guide rails) are reduced immediately. On the other hand the vibration behaviour parallel to the actual distance between guide rails increases before again dropping to the original value after a period of some 25 days. The vibration course in direction Y can be explained by a non homogenous actual distance between guide rails over the entire hoisting height of the lift system: the new slide guides must be allowed to first "grind in" in this direction is space. The diagram shown now simply allows a limit to be determined for the vibration behaviour in direction X which the system is not allowed to exceed and – should it be exceeded – a guide shoe maintenance suggestion to be tripped.

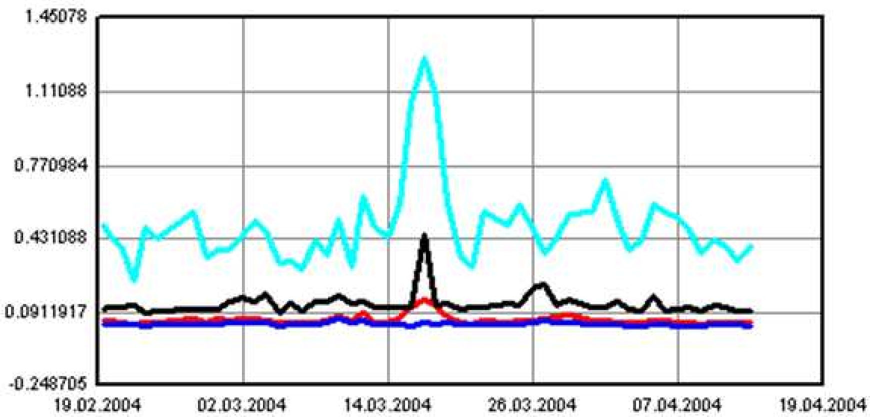

Figure 2: Characteristic vibration values of the movements of the cab door. One clearly recognises that the movement is impaired between March 15 and 17.

The second example (Fig. 2) shows four characteristic vibration values for the door movements. The period between March 15 and 17 is out of the ordinary, the event being of a sporadic nature this time: the guides of the cab door were contaminated by winter grit probably originating from the tyres of a fork lift truck. In this particular case the automatic door monitoring system tripped an alarm and the fault was eliminated within a relatively short time so that door rollers and guides could not suffer consequential damages.

Apart from the vibration data a measurement of the individual rope tensions and of the loading condition has proven to be extremely relevant. As a matter of course the loading condition affects the vibration levels so that these can only be evaluated in combination with the actual load. The individual rope tensions in the rope set should also be taken into account. A replacement of the motor torque by the motor speed generates a trend in the lift industry to use increasingly thinner ropes and higher suspension ratios. Rope research shows that the rope bending capacity is continuously reducing with the diameter [1]. A smaller traction sheave diameter to rope diameter ratio (D/d) additionally reduces the bending capacity; this also applies to multiple rope deviations. This immensely boosts the influence of only one badly adjusted rope of one rope set: the wear of the rope can for example reduce the life of the entire rope set by 60 % if one rope merely deviates by 15 % from the mean value of the single rope loads (see calculation of the rope life by K. Feyrer [2]).

Based on the pilot project conclusions and the exhaustive examination of measuring methods suitable for lift systems, Messrs. Henning have devised in the past few years a Condition Monitoring system for lifts the development of which will be completed at the end of 2012 with a field test in Germany. This system uses an intelligent vibration sensor permanently monitoring the wear of important lift system components. The sensor is mounted on top of the cab and autonomously detects (without being connected to the lift control system) the current ride condition so that door movements, ride starts, constant rides, etc. can be examined separately. In each of these ride conditions significant characteristic values are generated which in their entirety allow long-term trends as well as erratic or transient changes in condition to be detected and fully documented. Even gearboxes and motors can be indirectly recorded since vibrations are transmitted to the sensor via the suspension gear. The sensor is able to make a distinction between numerous wear aspects of critical components such as doors, drive units and guides. At the same time sensors detect the load on each suspension rope and therefore also the load in the cab.

The system has adequate interfaces allowing it to be connected to higher-ranking building management systems. Under favourable conditions, significant changes in the transmitted characteristic values will then generate a warning well before the failure limit of a component is reached so that the required servicing activity can be planned in advance and is no longer subject to fixed maintenance intervals.

SUMMARY

Condition Monitoring already widely used in other branches of industry is still largely ignored in the lift industry. Even though only a small number of lift systems need servicing strategies ending up in a condition-oriented and proactive maintenance, a cost-intensive preventive servicing strategy is the only alternative for lift systems which are part of a production process, which are used in public sectors to secure the mobility of people with physical impairments or which are indispensable for representation purposes. The partially massive cost reductions affecting lift components in the past few years can only be compensated by adequate countermeasures in form of a monitoring of safety-relevant and function-critical components. Automatic Condition Monitoring systems provide an efficient solution and warrant an optimum resource efficiency combined with a high plant availability.

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Symposium on The Lift and Escalator Technologies

Interdependencies Between the development of a Belt type Suspension and Transmission mean and lift components/system design

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INTRODUCTION

In today’s lift systems the steel wire rope is the most commonly used technology for suspension and transmission means. The steel wire rope technology used in Lift and hoisting applications has worked very well for more than 100 Years. Constant improvement in wire rope design, selection and combination of material, as well as advances in manufacturing technology has helped to gain the reputation that lifts are one of the safest transportation systems for Humans.
The component and system design for traction type lift application using steel wire ropes as well as the construction of steel wire ropes in today’s technology state, is the best found compromise at this point in time. Codes and standards have been implemented and tailored to create the framework for steel wire ropes in elevator applications to insure safety and consistency in lift applications.
Compromise in case of improvement means, addressing an isolated item does have an impact on other system areas. Furthermore, there are interdependencies which have to be addressed and can influence the design of a complete system significantly.
Implementation of improvements in one area of the lift system will lead to strive for the best compromise on the remaining system, with the goal the overall solution has improved compared to the previous best compromise.
Although the steel wire ropes have matured over the last decades they still have some disadvantages which are part of the compromise for the overall lift application. Disadvantages such as sheave diameter are to big (D/d 40), the weight, elongation and traction issues do not allow the development of advanced elevator system design without addressing those problems/restrictions.

Recent research and current development of the belt technology demonstrates the efforts made by a number of companies to circumvent the disadvantages of steel wire ropes. Although the currently introduced belt technology still uses steel wire cords within the belts, some of the disadvantages of the traditional steel wire ropes are addressed, for example the reduction of traction sheave diameters and traction issues. Future development of belt system technologies focuses on belt systems without steel wire ropes inside. This addresses an even broader range of today’s compromises made in Lift systems.

This presentation provides an outline of a Master Thesis in progress and will highlight the interdependencies between the development of a new belt type suspension and transmission means and the impact this has on the Lift system as well as on system component design. The final Thesis will act as an input and help the system and component designer to identify, calculate and address issues throughout the design process with focus on belts systems *without containing steel cords.*

GENERAL IDENTIFICATION

The initial focus in relation to Suspension and Transmission means clearly is on some of the main properties / terms used by Lift designer, Engineers or component developers. The properties / terms listed below address the most critical criteria of the new to be developed Suspension and Transmission means and will be used as a base line throughout this text.

Breaking strength
Weight,
D/d 40,
Elongation,
Traction,
Discard criteria,
Life cycle,
Handling / Maintenance.

Although it is acknowledged the list can be extended, but for the purpose of this text the list will be restricted to the above mentioned terms.
If each term is used as a headline and the direct relationship this headline has to the lift system will be described in general and listed, the list will serve as an input for the development of a new suspension and transmission means with the ultimate goal to improve all of the named areas.

Breaking strength. A minimum Safety Factor of 12 or higher as a general rule, will results in a certain minimum number of suspension members, or in suspension members with increased strength. There is a direct dependency to safety and system capacity.

Weight. The weight of the suspension and transmission means has a direct influence of the overall static system mass as well as dynamic masses (inertia etc.). [3]

D/d 40. Traction sheave geometry e.g. diameter, width, groove size, in conjunction with diameter / thickness of suspension and transmission mean. [1]

Elongation. Permanent elongation (stretch over time) and elastic elongation (dependent on dynamics such as load changes and acceleration changes) directly impact the system. [4]

Traction. Sheave surface design in regards to geometry and Material in conjunction with material and dimensions of the suspension and transmission means. [1]

Discard and replacement criteria. Currently visual inspection and broken wire counts, diameter reduction as well as magnetic field or resistance measurements methods are used to detect remaining Breaking strength, loss in traction, or overall deterioration over time.

Life cycle. Number of bending cycles, bending conditions e.g. number of reverse bends in systems, distance between pulleys, environmental influences, etc. [3]

Handling / Maintenance. Delivery to construction site, installation procedures e.g. end terminations. Maintenance requirements such as lubrication and cleaning, etc.

Relationship to lift system. *Table.1* below takes the above terms and lists them in the left column from top to bottom (note this is no classification). The top row represents some of the major components of a lift system. The bolded Capital "**X**" demonstrates direct dependency whereby the smaller "x" shows indirect or less influence.

Terms/Components	Motor	Sheave	Brake	Safety gear	Compensation system	Termination	Controller
Breaking strength	x	x	x	x	**X**	**X**	x
Weight	**X**	x	**X**	**X**	**X**	x	**X**

D/d 40	**X**	**X**	**X**	x	x	**X**	**X**
Elongation	**X**	**X**	x	x	x	**X**	**X**
Traction	**X**	**X**	**X**	**X**	**X**	x	**X**
Discard Criteria	x	x	x	x	x	**X**	**X**
Life cycle	x	**X**	**X**	**X**	**X**	x	**X**
Handling/ Maintenance	x	**X**	x	x	**X**	**X**	**X**

Table 1 Suspension/ transmission criteria in relation to major components

It has to be acknowledged that the components Motor, Sheave and Brake are often named as one assembly simply referred to as the *Lift Machine*, but for the development of belt type suspension and transmission means it is important to view these components individually.
The term *D/d 40* actually refers to a ratio between the sheave diameter and suspension rope based on code requirements for steel wire ropes, which may not apply for a belt system without steel wires inside.

Relationship to component. Up to this point the dependencies can be seen generic and apply to all types of Suspension and Transmission means for traction type Lift systems. With the finished design and known properties of the new suspension and transmission mean the system and component developer can follow the matrix above and evaluate the dependency for each component on a defined Lift system based on the properties.
This can be achieved by listing the main parts and parameters of the component [2] as indicated in the example for the Motor in *Table 2* below.

Criteria/ parts prop.	torque	speed	power	Shaft load/size	Bearings	dimensions
Weight	**X**	x	**X**	**X**	**X**	**X**
D/d 40	**X**	**X**	**X**	**X**	**X**	**X**
Elongation	**X**	**X**	x	x	x	**X**
Traction	**X**	**X**	**X**	**X**	x	**X**

Table 2 Suspension/ transmission criteria in relation to Motor parts and properties

Calculating values e.g. torque, speed, power, shaft load, bearings depend on the belt properties and criteria listed in the left column, this in return influences the dimensions of the Motor and creates input for the system designer to design the best compromise in relation to the new suspension and transmission mean.
System impact. A new suspension and transmission mean allows the designer to create new system approaches. This can be based on the changed components, based on the belt properties, or a combination of both. For example an increase in traction could allow new lightweight systems. Fig.1 below indicates some of the interdependencies this could have to a new system.

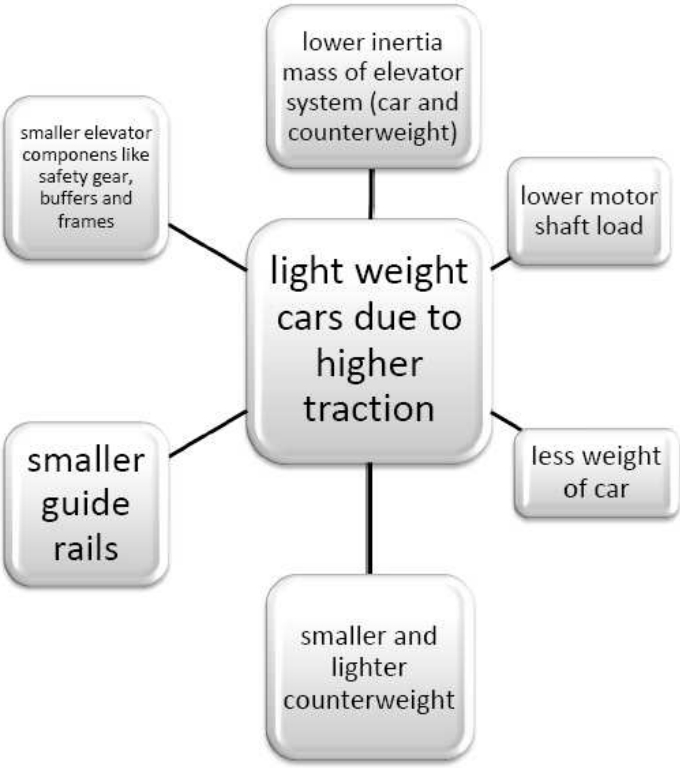

Fig.1

SUMMARY

This extended abstract from a Master Thesis in progress identifies some of the interdependencies between a belt type suspension/transmission mean with the Lift system and major components of the system described on a few examples. The text and tables demonstrate relationships and dependencies which require detailed investigation and calculations not only for the development of the belt but also on the system and component level. The required level of investigation on all aspects of a lift system will enable the system and component designer to think outside the box and apply solutions for new system approaches.

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Symposium on The Lift and Escalator Technologies

The Analysis of Excitation Sources and the Dynamic Responses in Lift Systems

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ABSTRACT

Traditionally, lifts were equipped with machine rooms that contained the drive unit and hoisting motor. Machine room-less lifts (MRL) now have these components located in the shaft and are required to achieve acceptable values of vibrations, airborne noise and structure borne noise. The transmission paths of noise and vibration indicate that they originate from various sources. The possibility to predict the response of systems and sub-systems can reduce development time and allows for specific design changes at an early stage. In the design phase the calculation of system natural frequencies and sub-system natural frequencies enables identification of resonance conditions. The identification of fundamental and harmonic frequencies of all components within the lift system enables quick allocation of excitation sources. The following discussion will briefly examine simulation techniques and identify the basic formulas involved in identifying excitation frequencies. The paper continues with methods of data analysis techniques.

INTRODUCTION

Lifts are highly complex dynamic systems that require detailed simulation and analysis in order to achieve acceptable levels of ride quality in the lift system. In the design phase of a lift system, it is important to have a prediction of the noise and vibration, firstly at a system level and secondly at a subsystem level. The system level addresses the complete lift system and the sub-systems can be further categorised as: machine, suspension media, guide rails, car and counterweight.

Therefore, it is necessary to integrate simulation and analysis into the design process in order to accelerate component and system development.

The results of the simulation and analysis drive the choice of the design solutions and can be considered as predictive engineering. The first step is to analyse the structural behaviour based on the calculation of natural frequencies and mode shapes.

In order to understand the dynamic analysis of a lift system, a mathematical model of the system must be developed to fully understand the response of the system and the systems natural frequencies.

Understanding the main sound sources and excitation frequencies enables targeted definition of design changes, in order to avoid critical resonance phenomena. A resonance phenomenon occurs when an excitation frequency is near the natural frequency.

Design and simulation are therefore imperative at an early stage of a project. Combining this with advanced measurement and analysis, it is possible to understand the noise and vibration propagation path and validation of the theoretical models.

SIMULATION

With simulation tools, such as Acoustic prediction tool, Liftsim and Matlab it is possible to predict the system and subsystem components behaviour. Simulation of systems allows for analysis over a wider range of load and size configurations compared to actual testing and reduces the costly task of physical testing. Moretti [1] suggests that in order understand the systems response, it important to simulate the sub-systems from excitation to response.

New technologies, software programs and the ever increasing availability of computational capabilities have driven the simulation opportunities in the lift industry today. Originally, simulation tools were first introduced in order to guarantee the integrity of components at a sub-system and system level, ensuring that they comply with code requirements. Predictive engineering is applied early in design phases, allowing structural simulation of load and stress analysis for verification purpose.

Today, simulation tools have been adapted to give an engineer the freedom to evaluate also aspects of lift ride quality and critical design decisions. The simulation of system and components at a development stage will help to define system and component specifications.

Roberts [2] indicates that simulation and virtual prototyping is a key factor to achieve cost effective means of designing lifts, in order to meet the expectations of the ever increasing demands on ride quality.

EXCITATION FREQUENCIES

With the knowledge of the excitation, at a system level to a subsystem level, together with the simulation and analysis of the structural behaviour, it is then possible to predict the response.
The calculation of the excitation frequencies will enable identification to see if the frequency is a velocity dependent frequency or not. Excitation frequencies for lift systems are generally dependent on the rated speed of the lift, the corresponding roping factor and the geometry of lift components, e.g. the radius of rotating parts. Detailed information on bearing design and elements will help to identify if a faulty bearing is the cause of a disturbance. Once all the relevant information about the system and the components is available, the excitation frequencies can be calculated. The basic formulas required are as follows.

The rotational frequency, rpm of the motor traction sheave, is calculated as:

$$RPM = \frac{i * v * 60}{\pi \frac{D}{1000}} \tag{1}$$

Where i is the roping factor, v is the rated speed and D is the diameter of the traction sheave.
In equations 1 to 4, the diameters are in millimetres instead of meters, they are divided by 1000 for the conversion to meters.
The rotational frequency of the motor traction sheave in Hz, is calculated as:

$$f_{sheave} = \frac{i * v}{\pi \frac{D}{1000}} \tag{2}$$

The rotational frequency of the magnetic poles in Hz is calculated as:

$$f_{MagneticPoles} = \frac{i * v * p}{\pi \frac{D}{1000}} \tag{3}$$

p is the number of magnetic pole pairs.
Excitation frequencies in Hz for roller guide shoes are calculated as:

$$f_{RollerGuide} = \frac{v}{\pi \frac{D_{Rg}}{1000}} \tag{4}$$

Where v is the rated speed and D_{Rg} is the diameter of the roller guide.

Rope Lay excitation frequencies in Hz are calculated as:

$$f_{RopeLay} = \frac{i * v}{L_{RopleLay}} \tag{5}$$

Where i is the roping factor, v is the rated speed and $L_{RopleLay}$ is explained by Janovsky [3].

For evaluation of all excitation frequencies, It is recommended to calculate the data in an excel table. With the data consolidated in a table, it is possible to identify the fundamental frequencies and the corresponding harmonics.

DATA ANALYSIS

Data recording and data analysis are very important aspects of excitation identification. Today in the lift industry, there are numerous hardware and software packages available and utilised by field personnel, in order to record and analyse data sets. For quick measurements in the field, on problem installations, or to validate consultation specifications they are quite a handy tool and can be utilised by most field technicians. Unfortunately, most of them are very limited in the sampling rates and do not offer adequate analysis of the sound recorded, due to the fact that they only record the noise level and not the sound pressure.

With advanced measurement and analysis tools, it is possible to understand the noise and vibration propagation path in order to validate the theoretical models.

To examine the spectrum of a signal, the time domain must be converted to the frequency domain. This technique is known as *Fast Fourier Transformation (FFT).* Spectrograms are a very efficient way to represent data, and to compare and understand excitation and resonance frequencies throughout the entire trip.

An example of this is demonstrated in Figure 1, where a resonance conditions have been clearly identified by their intensity. The darker the colour indicates higher frequency amplitudes that can be related to time and the position in the shaft.

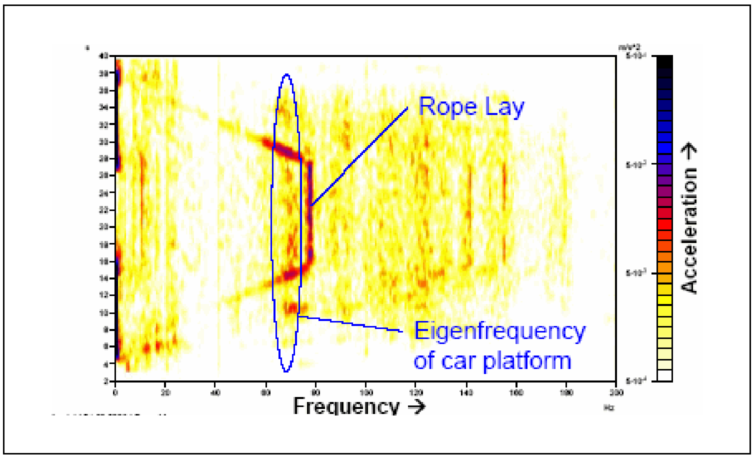

Figure 1. Spectrogram

CONCLUSION

The principles of dynamics form the foundation for the analysis and design of engineering systems. Lifts have to be designed in order to avoid the excitation frequencies that result in a resonance condition. The identification of natural frequencies and mode shapes are essential, in order to develop lift systems to operate optimally, within the buildings that they are designed for. The design of a lift system must not only consider the ride quality felt by passengers in the car. The objective is to achieve adequate ride quality with a combination of minimum transmissions of structure-borne noise and vibrations into the building structure and adjacent rooms.

Identification of all possible excitation sources and vibration transmission paths will allow for targeted design concepts to ensure adequate isolation is present, in order to mitigate disturbances from the system.

Today's lift market is changing from the typical layouts where a machine room was supplied, to cost driven versions of MRL lifts. MRL lifts therefore have to be designed differently in order to compensate the higher shaft noise levels, vibrations and structure borne noise values.

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Development of a Control Method for Speed Pulsation in Escalator's Chain

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INTRODUCTION

Chain drives are used in escalator mechanisms to transfer movement from the motor to the steps and handrails with high-efficiency and synchronization. The chain consists of rollers and links that connect the rollers. The movement of the motor is transferred to the chain by a sprocket that engages the links. However, the rigidity of the links prevent a smooth contact between the chain and sprocket while it is possible with a belt drive. Because of that, the chain winds around the sprocket in a polygonal shape that produces variation in the horizontal speed of the chain even though the sprocket rotates with a steady speed. Such changes in horizontal chain speed are referred to as pulsations. The pulsations are transferred to the steps of the escalator and decrease the comfort of passengers.

The proposed approaches to suppressing pulsation include shaping the chain rail with protrusions or depressions just before the sprocket teeth to vary the horizontal speed of the escalator steps so as to maintain a constant speed within the range where passengers ride (1) and to use an inverter to control the motor rotation speed to suppress the pulsation in the horizontal section of the chain.(2)

The former approach requires machining the rail into a geometrically-determined irregular shape, and the latter basically requires a means of using the sprocket phase and step speed data as feedback to satisfy the condition of constant drive speed, as well as a control circuit that uses that data to control the motor speed. Both approaches will increase system cost.

This paper proposes a new method to control the pulsation of chain speed keeping the constant rotational speed in the motor. It's a method which makes the roller speed change moving roller track adding a new type of rail next to the sprocket.

CHAIN DRIVING PRINCIPLE

We explain here the operating principle of the chain drive mechanism using the schematic diagram presented in Fig. 1. The chain consists of rollers that are connected at regular intervals by links. The chain winds around a sprocket so that the chain moves when the sprocket turns. The roller speed V_n is expressed by Eq.(1) and Eq.(2).

$$V_n = \frac{\Delta X_n}{\Delta t} = \frac{X_{n-1} - X_n}{\Delta t} \qquad (1)$$

$$X_n = \sqrt{P^2 - R^2(1-\cos(n\Delta\theta))^2} - R\sin(n\Delta\theta) \qquad (2)$$

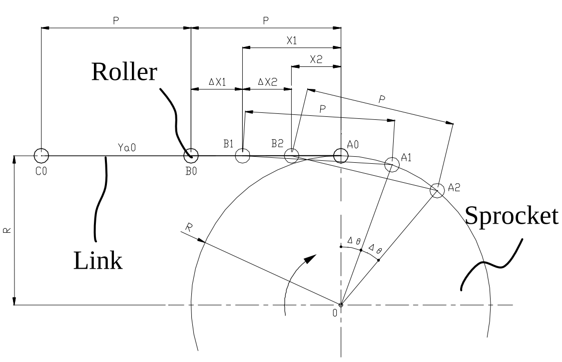


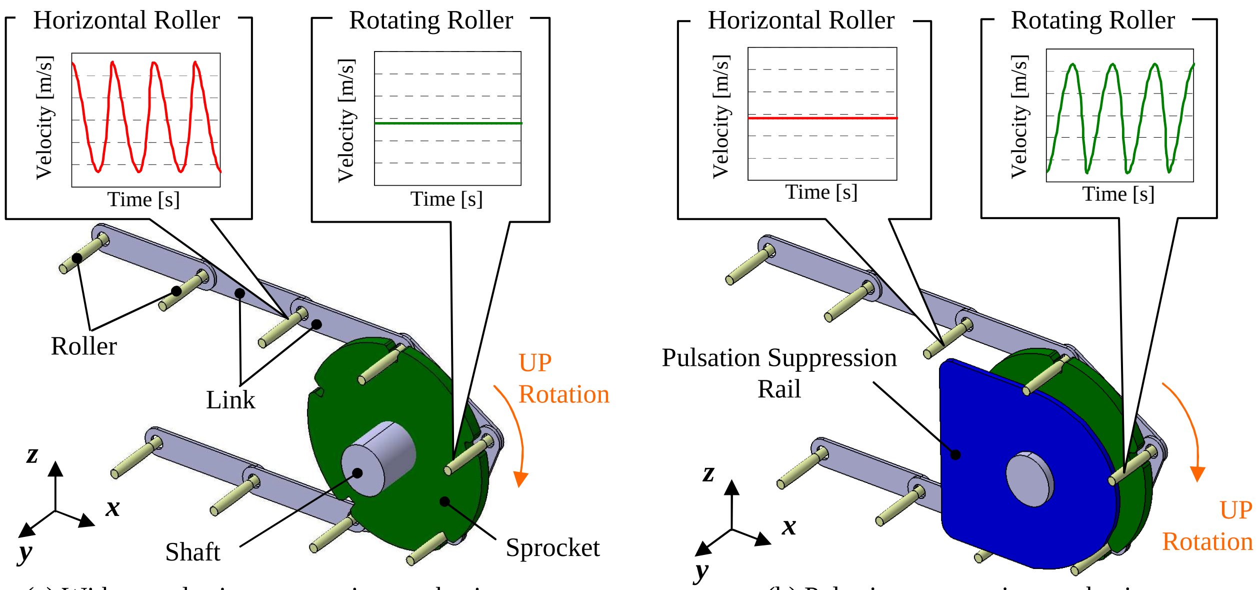
Fig.1 Pattern diagram of an escalator drive part

METHOD OF SUPPRESSING PULSATION

We propose here a mechanical method of suppressing pulsation in which the trajectory of the chain rollers is altered by placing a fixed chain rail that is easily machined and easily installed at the position where the chain turns. The conventional chain mechanism and the proposed mechanism are illustrated schematically in Fig. 2.

In the conventional mechanism, shown in part (a) in Fig. 2, the chain rollers in the rotating part engage the sprocket teeth and move in an arc along the pitch circle and leave the sprocket at the bottom. At that time, because the sprocket moves at a constant speed, the circumferential speed of the rollers is also constant. Nevertheless, a pulsation that corresponds to the length of the chain links occurs in the horizontal sections of the chain.

In the proposed mechanism, on the other hand, the rollers are pulled along by the teeth of the sprocket, but they follow the contour of the fixed rail in the rotating part as shown in Fig. 2 (b). That change in the roller trajectory in the rotating part alters the speed so as to cancel out the pulsation and produce a constant roller speed in the horizontal section of the chain.



(a) Without pulsation suppression mechanism (b) Pulsation suppression mechanism

Fig.2 Construction drawing of pulsation suppression mechanism

DESIGN AND ANALYSIS

Pulsation Suppression Method 1. The roller speed in the rotating part is expressed by $V_r = R_r\omega$. In this first design, the radius of the roller from the center of the sprocket in the rotating section, R_r, is reduced to lower the circumferential speed at the position where the horizontal roller speed is higher and R_r is increased to increase the circumferential speed where the horizontal roller speed is lower so as to make the horizontal roller speed constant. (The rotation speed, ω, is constant.)

The path defined by the designed pulsation suppression rail and the roller and sprocket tooth engagement section are illustrated in Fig. 3 (a); the sprocket tooth is illustrated in Fig. 3 (b). We performed kinematics and dynamics analysis simulations in which the sprocket illustrated in Fig. 4 (a) rotated clockwise for upward drive. The waveforms for horizontal chain speed for a movement of one pitch for suppression mechanism 1 and without suppression are presented in Fig. 4 (c) for comparison.

The effect of pulsation suppression mechanism 1 is a reduction of pulse amplitude by 18% for upward drive relative to the case without suppression. However, some change in speed occurs, and we understand from the analysis results that the design of path before and after roller and sprocket engagement is important.

Pulsation Suppression Method 2. Based on the results obtained with proposed method 1, we designed a new chain rail and tooth shape that takes into account point; it is to make the path just before and after the onset of roller and sprocket tooth engagement as smooth as possible. An additional constraint on the second design is that there be no change in height so that existing escalator components can be used without modification.

The rail shape (path) and the roller engagement section designed for method 2 are illustrated in Fig. 4 (a) and Fig. 4 (b) show the sprocket tooth shape. Fig. 4 (c) respectively presents the simulation results for the speed of horizontal roller movement of one pitch length under suppression method 2 and without suppression.

We see from Fig. 4 (c) that the speed pulsation is controlled to produce a constant horizontal speed. These results confirm that suppression method 2 can reduce the pulsation amplitude by 2% for upward driving relative to the case without suppression.

After the simulation confirmed the speed pulsation suppression effect of the proposed method, we next fabricated a prototype pulsation suppression rail and sprocket, and installed them on an actual escalator to test the suppression effect.

The results of the prototype testing revealed almost no difference in the comparison of waveforms with current escalators, but they did confirm, in part, changes in speed that were not observed in the simulations.

We discuss those speed variations with reference to Fig. 6. In current chain mechanisms, the roller engages with teeth that have a circular bottom as we see in Fig. 6 (a), so the roller position is uniquely determined by the sprocket and does not move within the tooth shape regardless of the tension on the links to the left or right.

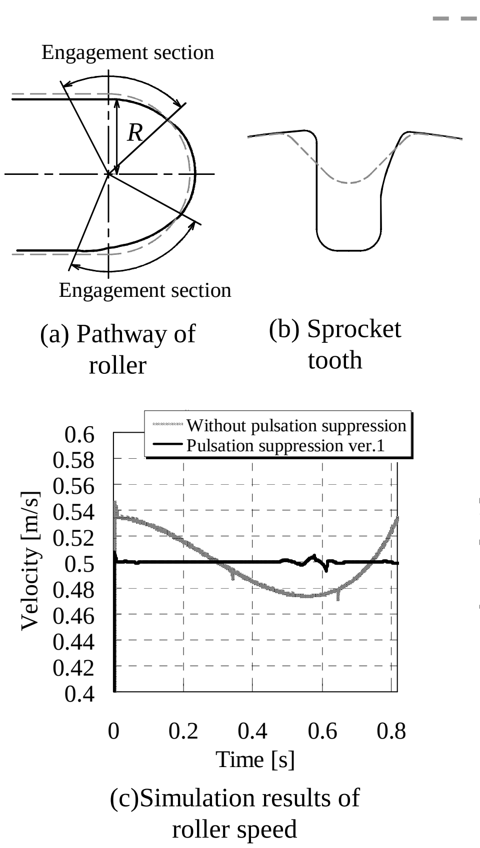


Fig.3 Pulsation suppression mechanism ver.1

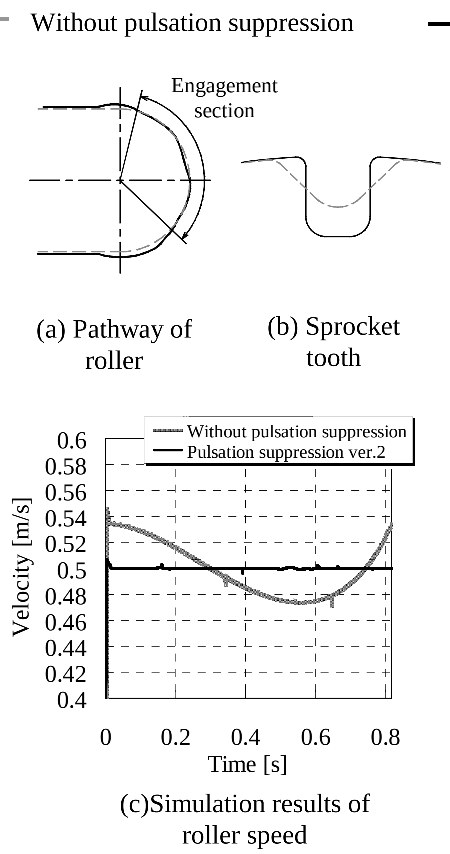


Fig.4 Pulsation suppression mechanism ver.2

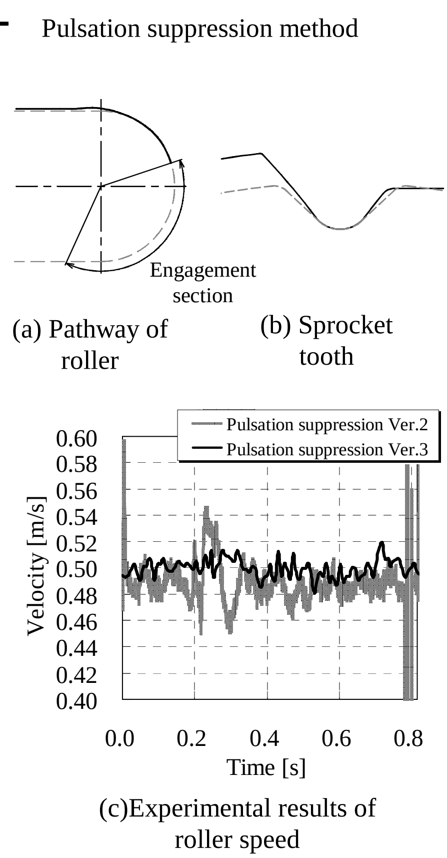


Fig.5 Pulsation suppression mechanism ver.3

In method 2, on the other hand, as shown in Fig. 6 (b), the roller position is determined by both the sprocket and the pulsation suppression rail. As a result, the roller can move by the amount of backlash allowed by the sprocket tooth shape, and moves within the tooth shape due to the tension of the links. Experiments have shown that such movement results in variation in chain speed.

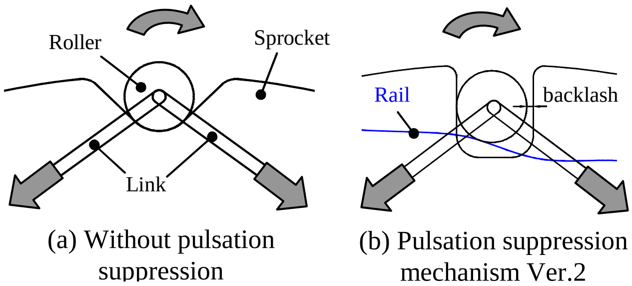


Fig.6 The cause of generating of speed pulsation

Pulsation Suppression Method 3. Building on the results obtained for method 2, we proposed pulsation suppression method 3 to solve the problem of the movement of the roller within the tooth shape.

Method 3 has two features; one is pulsation suppression rail that is placed only in the section of the rotating part defined by the angle through which a single roller, and the other is round-bottomed sprocket tooth. Because the only thing that affects the horizontal speed of a roller is the roller that is in front of it and is in the rotating section, the pulsation suppression chain rail is placed only in the section defined by the angle through which one tooth of the sprocket advances. In the section where there is no chain rail, the roller engages the sprocket firmly at the bottom of the tooth and does not move within the tooth.

The roller path of the pulsation suppression chain rail designed for method 3 and the section in which the roller and sprocket engage are illustrated in Fig. 5 (a); the sprocket tooth shape is illustrated in Fig. 5 (b).

We fabricated a prototype chain rail and sprocket that implement method three, installed them in an actual escalator, and measured the roller speed in the horizontal section of chain. The results for a movement distance of one pitch for method 2 and method 3 are presented in Fig. 5 (c). In proposed method 3, there is no movement of the roller within the sprocket tooth, and the variation in chain speed is greatly suppressed.

CONCLUSION

We have proposed here a mechanism for producing a constant horizontal chain speed to suppress the phenomenon of speed pulsation that is caused by polygonal motion in a chain drive. The proposed method places a fixed chain rail at the drive sprocket to alter the trajectory of the chain rollers so as to geometrically achieve constant horizontal chain speed. The shapes of the fixed chain rail and sprocket teeth were designed with progressive improvement by performing analysis with a computer-aided kinematics and dynamics tool and testing the result in an actual machine to produce three methods successively.The final result is confirmation of the pulsation suppression effect by installation of a prototype in an actual escalator.

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Mathematical Modelling of Comparative Energy Consumption between a Single-loop Curvilinear Escalator (The Levytator) and an Equivalent Pair of Linear Escalators

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INTRODUCTION

This paper describes a technique for mathematically modelling the comparative energy consumption between two types of escalators deriving energy differential functions. Using numerical analysis this paper shows how the energy consumption may vary under different load patterns. The paper concludes that the use of a Levytator is almost always more energy efficient than a pair of conventional straight escalators.

We have focused on energy consumption in operation as we believe the Levytator's carbon footprint in manufacture and disposal would be significantly less than a pair of conventional escalators. If you need details on this, please contact the authors of this paper.

The Levytator: Conventional escalators follow a straight line. The return path of the step travels underneath the useable steps beneath the housing. In order to provide both up and down paths of travel, two conventional linear escalators are needed.

The Levytator is designed to follow any reasonable curve. Its unique patented step design using vertical bearings, allows one Levytator to provide both up and down directions of travel as both set of steps are part of one loop. The Levytator only needs one power source to drive the steps whereas a conventional escalator needs two motors.

Structure: This paper sets out a method for mathematically modelling the differential power consumptions between a single Levytator configuration and a pair of conventional escalators for the same rise. The construction of the mathematical model is set out in Part 1-Overview. The calculations using some simple assumptions are set out in Part 2-How Green is the Levytator. The numerical analysis and the shape of the energy functions are detailed in part 3.

PART 1 OVERVIEW

The performance analysis of the Levytator consists of the comparison of power demand of two escalators that have travelling passengers in two opposite directions and the Levytator with the same geometry. The steps of the process are:-

- To produce the equation of total power demand P^* for two escalators ("up" and "down") with the same geometry (length l and canting angle α) traveling in opposite directions;
- To produce the equation of total power demand P^{**} for the Levytator (the length of incline bands - $2l$, total length – S);
- To calculate the relation between power demands depending on the dynamic parameters.

The calculations based on the Newtonian dynamics and the energy conservation law.

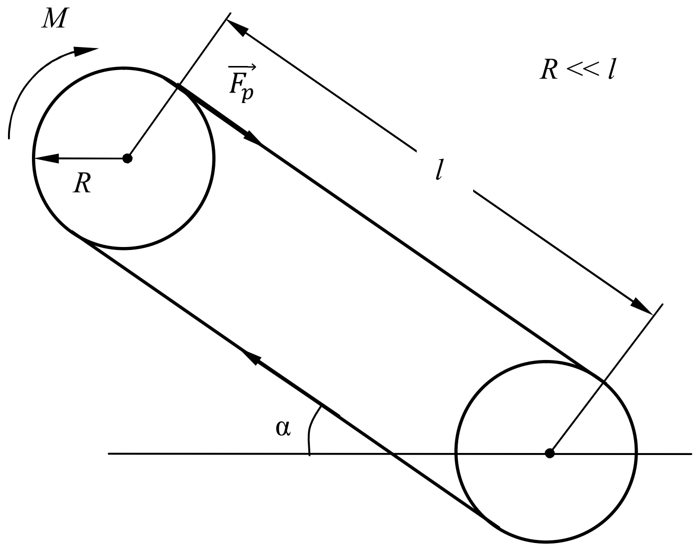

Fig 1 A single conventional escalator

Figure 1 shows a stylized representation of a single conventional escalator rising at an angle of α. The engine has an effective propulsive force of F_p working on an effective radius of R on an escalator of effective length l. Letting m_e be the mass of the escalator, m_1 the mass of passengers going up, m_2 the mass of the passenger going down, V the velocity of the escalator band, η the efficiency coefficient of the engine and μ is the coefficient of friction due to the band of the escalator, we can derive the power demand for a pair of conventional escalators to give the following equation.

$$P^{*} = \frac{V}{\eta}[\mu g \cos\alpha\,(2m_e + m_1 + m_2) + g \sin\alpha\,(m_1 - m_2)]$$

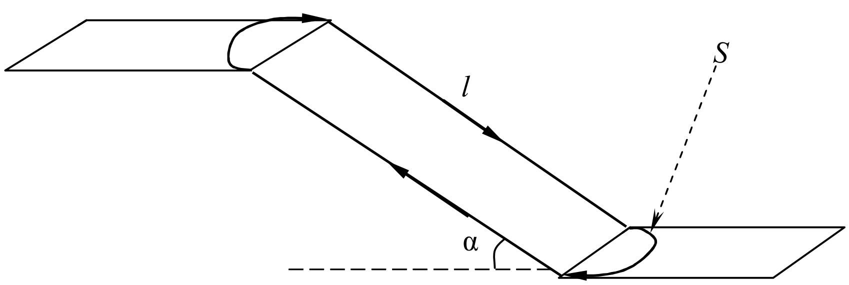

Fig 2 The Levytator

Figure 2 shows the configuration of the Levytator showing both the upward and downward paths of the loop. It is configured to be equivalent to two conventional escalators of effective length l. Since the Levytator's return loop is the downpath, we introduce another variable S, the total length of the Levytator loop. Using the same variables as above, we can derive the power demand for the Levytator as

$$P^{**} = \frac{V}{\eta}\left[\mu m_e g \cos\alpha + \mu m_e \frac{S-2l}{2l} g + (m_1 - m_2) g \sin\alpha + \mu(m_1 + m_2) g \cos\alpha\right]$$
$$= \frac{V}{\eta}\left[\mu g (m_e(\cos\alpha + \frac{S}{2l} - 1) + (m_1 + m_2)\cos\alpha) + (m_1 - m_2) g \sin\alpha\right]$$

Therefore, we can derive the Green Coefficient as follows:-

$$\frac{P^{*}}{P^{**}}=\frac{\mu\cos\alpha\,(2m_e+m_1+m_2)+(m_1-m_2)\sin\alpha}{\mu(m_e(\cos\alpha+\frac{S}{2l}-1)+(m_1+m_2)\cos\alpha)+(m_1-m_2)\sin\alpha)}$$

PART 2 – HOW GREEN IS THE LEVYTATOR

By applying some reasonable assumptions, we can illustrate the 'Greenness' of the Levytator against a pair of conventional escalators by applying these assumptions to the power demand equations derived above in the mathematical models.

We have made the following assumptions in producing the indicative calculations.

The rise for both systems is 7.5m with both systems travelling at a speed of 0.5 m/s. The effective length of the incline l of both systems is 15 m and the number of visible steps is 39. We assume that the step sizes and width are equivalent (1 m wide and 0.38 m deep) and have similar masses. We also assume the average mass of a single passenger is 75 kg. We have assumed the energy conversion efficiency η is the same for both at 90% and the effective coefficient of friction μ is 0.25.

According to the reference paper published by the Royal Academy of Engineering [1], we derive that 1kW per hour in a coal-fired station typically produces 0.9 g of CO_2.

We have modelled the following three cases as an illustration of the relative 'greenness' between the Levytator and a pair of conventional escalators.

Results	Power, P^{*} and P^{**} (kW)	Green Coefficient, $\frac{P^{*}}{P^{**}}$	Power per person per trip, $P^{*}_{per\ person\ per\ trip\ up\&down}$ and $P^{**}_{per\ person\ per\ trip\ up\&down}$ $(kW-hr)$	CO_2 emissions per person per trip, (g)
Escalator	14.91	1.25	0.003148	2.83
Levytator	11.96		0.002525	2.25

Fig 3 Full loaded both up and down

Results	Power, P^{*}_{UP} and P^{**}_{UP} (kW)	Green Coefficient, $\frac{P^{*}_{UP}}{P^{**}_{UP}}$	Power per person per trip UP, $P^{*}_{per\ person\ per\ trip\ up}$ and $P^{**}_{per\ person\ per\ trip\ up}$ $(kW-hr)$	CO_2 emissions per person per trip UP, (g)
Escalator	13.72	1.27	0.0058	5.22
Levytator	10.77		0.0046	4.05

Fig 4 Half loaded with empty downward path

Results	Power, P^{*}_{UP} and P^{**}_{UP} (kW)	Green Coefficient, $\frac{P^{*}_{UP}}{P^{**}_{UP}}$	Power per person per trip UP, $P^{*}_{per\ person\ per\ trip\ up}$ and $P^{**}_{per\ person\ per\ trip\ up}$ $(kW-hr)$	CO_2 emissions per person per trip UP, (g)
Escalator	5.752	2.05	0.00243	2.18
Levytator	2.8		0.0012	1.08

Figure 5 Half loaded with upward path empty

PART 3 – CONCLUSION

From the tables in Part 2, we can show theoretically that the Levytator is more ‘Green’ than a pair of conventional escalators in a similar configuration. We have also modelled several variations of the assumptions (e.g. different values of μ etc). In the main paper we show diagrams from MathCad using different numerical analysis.

Obviously, the accuracy any mathematical model is dependent on the selection of the main parameters to be modelled. Having completed this model, we could refine it further by breaking down μ to include friction between the step bearings and its guide tracks etc. However, we believe we have modelled the key parameters.

The technique shows that we can develop mathematical models to predict likely power demands even before the system is built. By using simple mathematical tools, we can express our intuition that the Levytator is likely to be more energy efficient in some more reasoned and logically argued form. It is also a powerful method to show the energy efficiency of a system before it is built and Elena is researching the application of such techniques to marine systems.

In our attempts to commercialise the Levytator, we have focused on its unique feature of being able to follow any reasonable curvilinear path. This particular modelling exercise has highlighted to us the opportunity to ‘sell’ the Levytator as a ‘Greener’ and more energy efficient solution than conventional escalators.

One final note, in our numerical calculations, there are certain combinations of factors that suggests the Levytator, rather than consume energy, may generate energy!

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Vibrations in a Lift System

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INTRODUCTION

In this paper various models of the dynamic behaviour of a lift car are discussed. The dynamic responses due forces and motion excitations have been analysed. As an example, the results of a computer simulation to demonstrate the effects of the excitations have been presented.

VIBRATION SOURCES IN LIFT SYSTEM

Guide rail excitation:

There are several cases under which the guide rail installation will cause excitation in lift system:

1- Missalignment of the joints: this case is the most common case of excitations in lift system. The excitation is of shock type. The magnitude of this shock depends on the physical placement of two guide rail edges at joints. The quality of guide rail surface treatment and fishplates are the affecting parameters. The applied shock will be transmitted to the car through the guide devices. Generally the guide devices are equipped with spring damper mechanism. The spring mechanism with lower the applied shock magnitude, the damping mechanism will dissipate the energy in the system during a period of time.
2- DBG variations: DBG (Distance Between Guide Rails) variation will cause change in reaction forces of the guide device (guide shoe or guide roller). The frequency of this variation is very low.

Traction machine excitations

1- Torque ripple: naturally the torque of all electrical machines is not quite smooth. This depends on the internal structure of the machines. In case of asynchronous machines which are coupled with reduction units the torque ripples are reduced to a very low level; however in this case the reduction unit itself could be a source of ripples in the machine output. In case of permanent magnet synchronous machines torque ripples are of much more importance, since there is no reduction unit. During recent years machine manufacturers has improved so many solutions to overcome this issue, some of them improved special feedback systems to overcome the ripples (with the use of digital signal processors) and some has improved the rotor slip angle and magnet placements in order to reach the maximum smoothness in output torque.
   Torque ripple will transmitted to the car - through the suspension ropes which are elastic mediums - in form of longitudinal vibrations. As reported by Schindler [1] in model 3300 and 5300 the recorded values are of less than 25mg inside the car.
2- Traction sheaves and rope impact: one of the major sources of excitation created in traction system is the noise and vibration created because of rope and sheave profile impact. They are of random order in magnitude and direction.

Air turbulence effect

Motion of the car and counterweight in lift well will create turbulence when they pass each other. As reported by SHI Li-qun et al. [2] during experiment when the distance of the car and counterweight is changed by $0.1, 0.2, 0.3\ m$ in the lift well the positive lateral forces to the car changed $3, 2, 1.5$ times and negative lateral forces changed $7, 5, 3$ times from the time counterweight was far away from the car.

Unbalanced Rotational movements

Exhausting fan and the door operator machine are the major sources of the noise and vibration excitation in lift system. This kind of vibrations has harmonic nature. Both sources require special isolation mechanism in order to minimize the effect. Basically these kind vibrations create noise rather than movement.

Building structure:

The building structure is always subject to excitation and vibration during operation of the lift. Sometimes effects such as the building sway which may is caused by wind and turbulence may indice vibration to the lift system. On the other hand the vibrational sources of the lift may excite the building structure and create noise or in cases like safety gear engagement may cause larger magnitude vibrations.

Vibration and energy transfer

Vibration is normally defined as a periodic or repeating motion of the body/ object under consideration. Vibrational motions are directly related to the kinetic energy of the object. In a system of particles or solid bodies linked together by constraints this energy can be transmitted between objects through contact / collision or excitation.

Lift car dynamic model

As a part of this study a computer based modeling of a lift car has been developed in Matlab / Simulink software to study the effect of a guide rail joint misalignment. The lateral force is from guide roller reaction caused by a $1\ mm$ step at guide rail joints (see Figure 1). The displacement is applied to upper and lower guide roller within $1\ sec$. The general schematic diagram of the model is illustrated in Figure 2. The roller guide is modeled as illustrated in Figure 3. The physical properties of the lift car are calculated with the use of a real scale computer based modeling in Inventor Software by Autodesk. Unfortunately real time modeling process of the suspension ropes in Matlab requires substantial programming efforts, which is not practical while using personal computers. Thus, for this modeling the suspension ropes are considered as a solid rod of small mass.

The car ($W1800\ x\ D1400\ x\ H2300\ mm$) is considered fully balanced in XY plane and weight vector ($732Kg$) of the car is considered at the center of gravity, the placement of the CG in vertical alignment is considered at $1220\ mm$ from the guide roller ends. All guide devices are equipped with 4 $^{kN}/_{m}$ springs with natural length $20cm$ and active length of $10cm$.

Analysis results show a $2\,mm$ displacement in $Y\ axis - lateral\ moevemnt$ and $1\,mm$ Movement in $Z\ axis - vertical\ movement$ Figure 4.

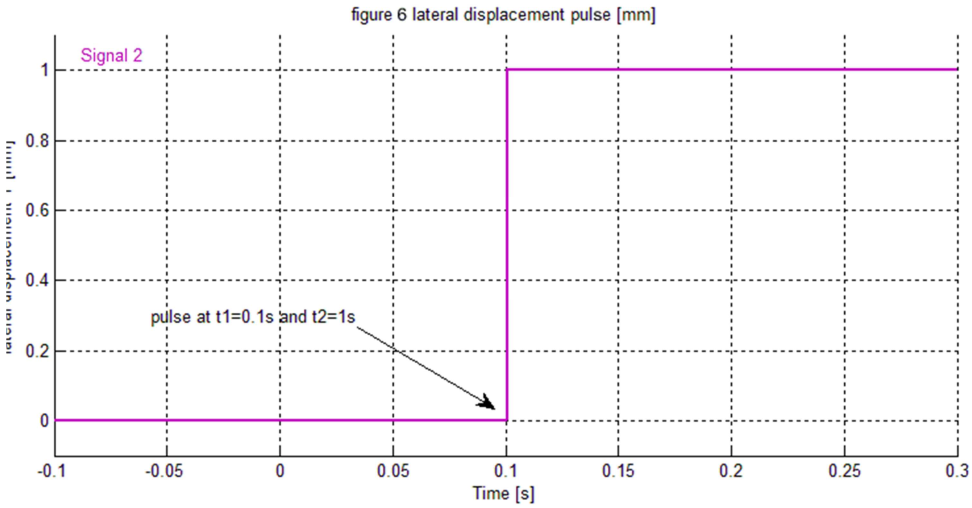


Figure 1

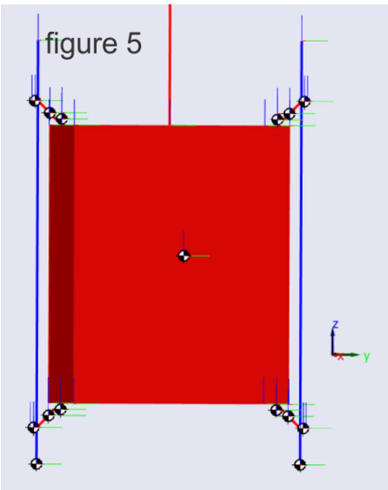


Figure 2

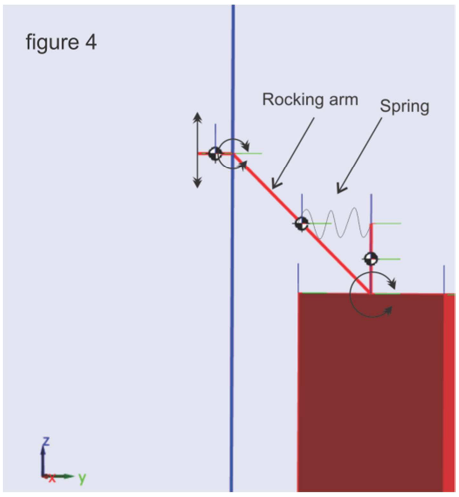
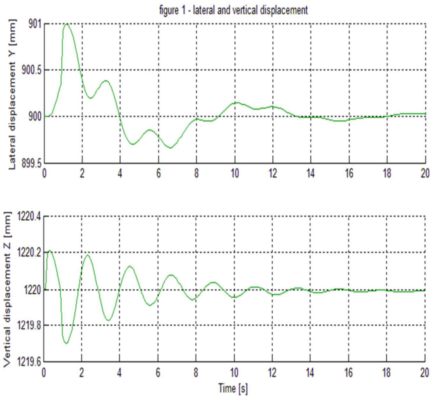


Figure 3



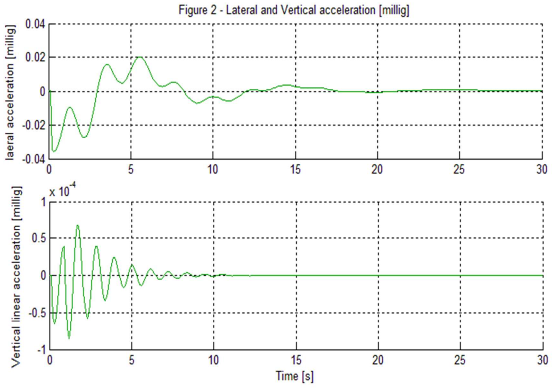


Figure 4 **Figure 5**

Also the relative acceleration values are shown in Figure 5.

SUMMARY

The ride quality and vibration responses in lift systems depend on many effects. By modeling the lift system it can be seen that the lift car is subjected to rotational vibration while is excited by a lateral forces.

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Is the Gearbox Dead?

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INTRODUCTION

By 2009 the relatively cheap synchronous permanent magnet gearless machines that had been originally developed for MRL applications were being applied more widely, with many manufacturers offering packages consisting of machines, bedplates and divertor pulleys aimed at the modernisation market and many consultants specifying these systems for low to medium rise buildings because of their perceived benefits with regard to reduced running costs and general eco-friendliness as exemplified by environmental assessment methods such as BREEAM made it a good time to carry out an objective study comparing the capital and running costs of schemes using a traditional geared machine with schemes using a gearless machine for a range of real life modernisation applications.

The Perceived Benefits of PM Gearless Machines. The main points are as follows:

- They are more efficient and can be used with regenerative drives, thus saving energy;
- One machine model can be applied to a larger range of applications than a geared machine thus making it more economical to hold stocks “on the shelf” reducing lead times;
- Cleaner than a geared machine because no oil reservoir is required;
- Machines are designed to be low maintenance and should offer savings on long term maintenance costs.

Possible Disadvantages of PM Gearless Machines. The main points are as follows:

- Most machines are designed for use with new MRL package lifts, i.e. lightweight lift cars and multi-reeved pulleys (2:1 systems being common with 4:1 and even 6:1 systems used for larger capacity lifts) whereas a traditional lift will have heavier cars and 1:1 roping;
- Many packages designed for modernisation use rope diameters and pulley diameters smaller than permitted by EN81-1 to convert existing 1:1 roped systems to multi-reeved systems;
- The machines may need “exotic” arrangements of divertor pulleys to increase the angle of wrap of the ropes on the sheave to achieve traction;

METHODOLOGY

Machine Selection. As each machine manufacturer has developed their own individual methods of machine selection, system calculations were developed from the coursework and the relevant sections of EN81-1 to select the machines. Compliance with the requirements of EN81 with regard to rope diameter and minimum rope to sheave ratios was considered of prime importance. Unfortunately this disqualified some gearless machine ranges from consideration, as they used ropes smaller than 8 mm in diameter. The manufacturers with the widest ranges of machines capable of covering the full range of applications considered (1:1 or 2:1 roped up to and including 2000 kg rated load and 2.0 m/s rated speed) were chosen, namely Alberto Sassi S.p.A. for the geared machines and Leroy Somer for the gearless machines.

Estimation of Energy Consumption. BREEAM is the most commonly used environmental assessment method used in the UK and their methodology used in 2008 made reference to ISO Draft standard ISO/DIS 25745-1:2008[1,2] [1]. The method outlined in draft standard for calculating the theoretical energy usage wasn't good enough as it didn't give any guidance on the estimation of the number of trips per annum and placed undue emphasis on the reduction of the counterbalance ratio. A more comprehensive methodology was found on the Energy-Efficient Elevators & Escalators (E4) website and this was used instead [3]. Figures for the number of trips per annum were taken from the UK section of the E4 interim report [4]. For the gearless machines the energy usage was calculated separately with and without regeneration.

Unfortunately neither publication gave any guidance on estimating the power required when the lift was on standby. It was assumed that the overall standby power for the worst case (i.e. installations without automatic shutdown on idle) would include elements required by the drive, controller & indicators (40 W) [5]; the door gear (15 W per car entrance for powered doors only) [6] and the car lighting (5 kW per 100 kg rated load, double this for hospital lifts).

Estimation of Costs. The capital cost items that needed to be considered for each scheme were:

- The machine and associated rope guards, bedplates & divertors from the machine manufacturer's price lists (Sassi or Leroy Somer).
- Ropes (Gustaf Wolf from Re-ropes Ltd).
- Drives (Control Techniques "Unidrive SP" from Leroy Somer). For the gearless schemes the drive cost was assumed to double if the drive was regenerative.
- Compensation (Dätwyler flat belt type from A&A).

The running costs comprised electricity and maintenance. Electricity was difficult to estimate because of the plethora of available tariffs.EDF Energy's standard domestic tariff for the London area [7] was finally chosen for use as a benchmark. After some discussion with a colleague selling maintenance it was decided to exclude this element from the running costs as the cost of a contract is primarily determined by the age and availability of spares for the equipment, so in this case the cost of a maintenance contract would likely be the same for both geared and gearless schemes.

Applications. The applications were chosen from actual modernisations that had been undertaken by Kone in 2008/2009. These ranged from 2000 kg goods lifts with manual doors in a retail unit to small lifts in residential units and included lifts in offices & hospitals.

RESULTS

With regard to the capital costs in every case considered the geared machine was the cheapest option and the gearless machine with regenerative drive was the most expensive; and the gearless machine with regenerative drive consumed the least energy, the geared machine the most. Further analysis is required to ascertain whether the energy savings made by the use of a gearless machine with regenerative drive can ever be sufficient to offset the increased initial outlay.

Fig. 1 shows the relative costs of each scheme for each of the case studies after 15 years assuming energy costs rise by 10% each year. The case studies were arranged in order of usage, with 1 having the lowest use and 15 the highest. It can clearly be seen that gearless solutions are generally more economical for applications with usage in excess of 300,000 trips per annum. Most of the case studies follow the same pattern, the exceptions being case 2 which had a relatively high rated speed, and case 14 which had an extremely high usage.

The power required of a lift motor depends on the rated load and the rated speed, so comparing the costs against the product of the rated load and rated speed as shown in fig.2 gives a further

[1] Both BREEAM and ISO/DIS 25745-1 have since been updated.

insight into the point at which it would be economically feasible to use a gearless machine in preference to a geared machine.

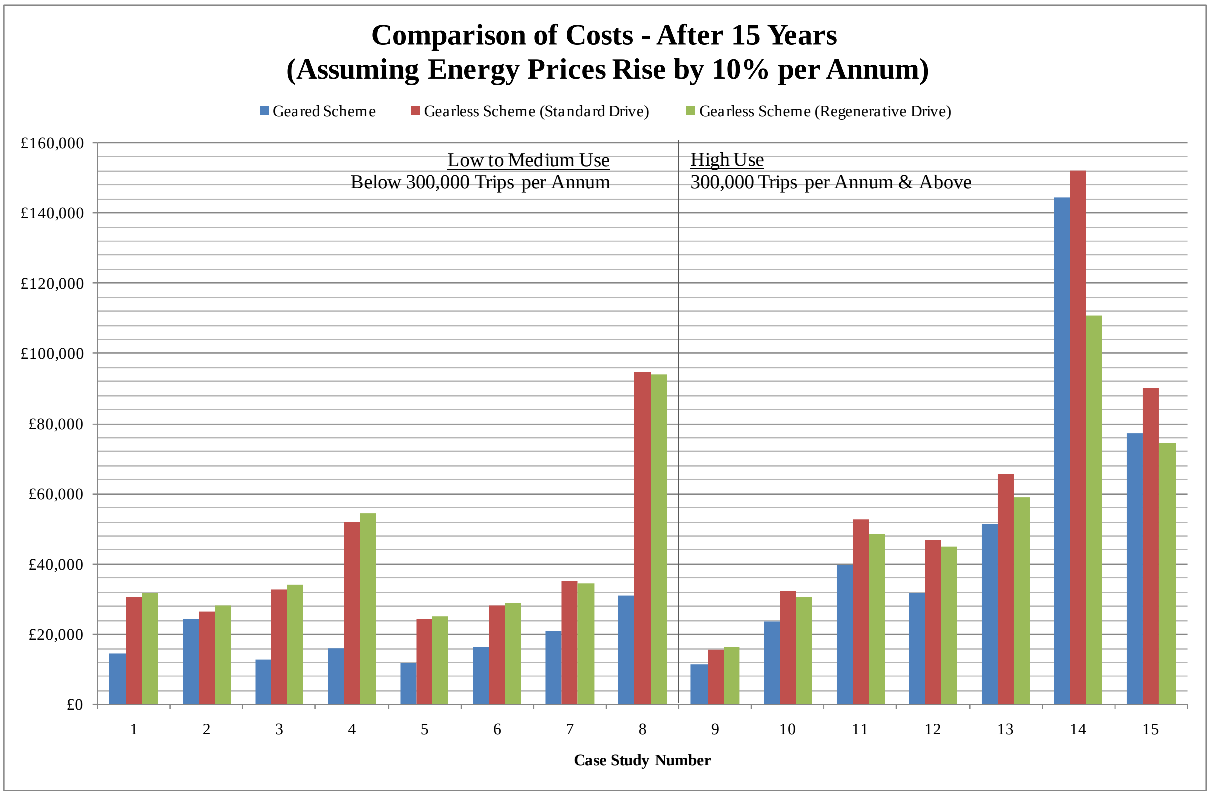

Figure 1

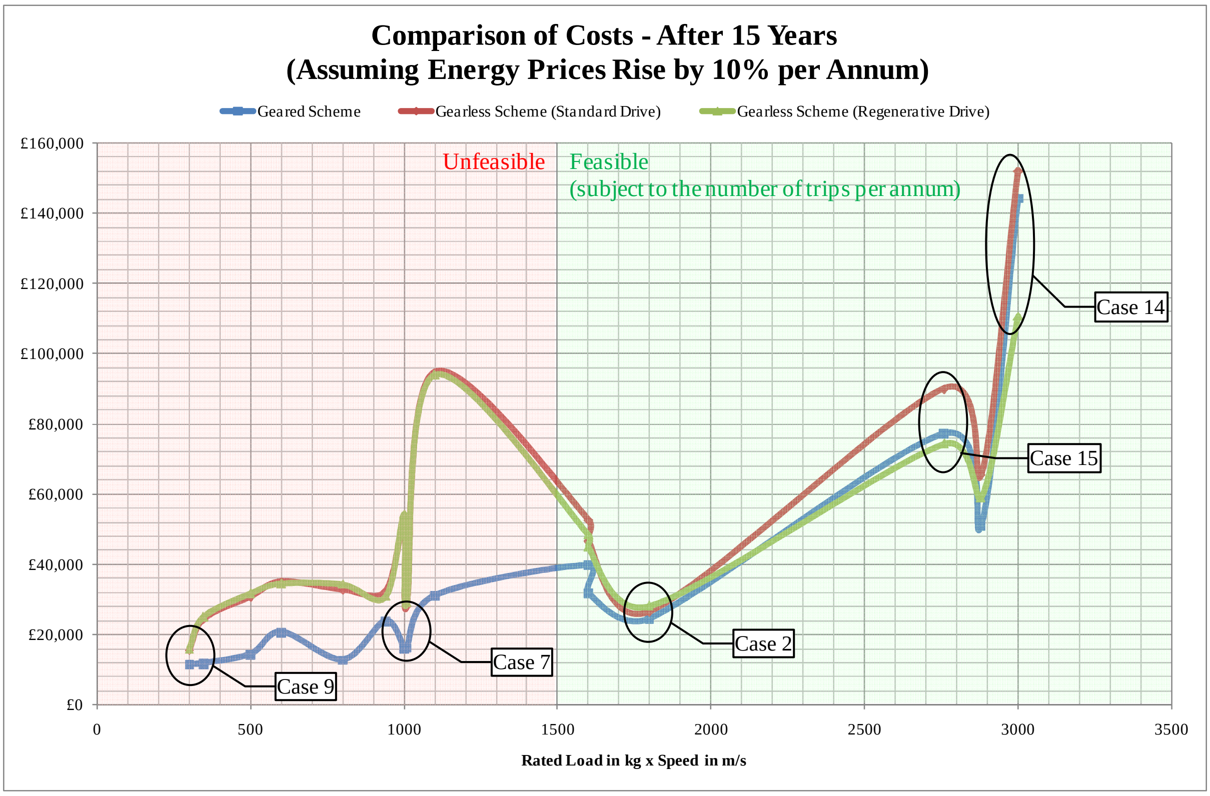

Figure 2

It is informative to look at tab.1 the data table used to prepare fig. 2 for further insights.

Rated Load x Rated Speed (kgm/s)	300	346.5	500	600	800	938.7	1000	1008	1100	1600	1600	1800	2760	2880	3000
Trips per Annum	300,000	50,000	30,000	200,000	30,000	300,000	30,000	50,000	200,000	300,000	300,000	30,000	800,000	300,000	500,000
Geared Scheme	£11,507	£11,827	£14,457	£20,741	£12,809	£23,866	£16,065	£16,239	£31,161	£39,786	£31,878	£24,571	£77,289	£51,184	£144,400
Gearless Scheme (Standard Drive)	£15,764	£24,441	£30,858	£35,099	£32,775	£32,326	£52,215	£28,234	£94,758	£52,712	£46,650	£26,613	£90,108	£65,632	£152,160
Gearless Scheme (Regenerative Drive)	£16,282	£25,206	£31,831	£34,625	£34,335	£30,828	£54,375	£29,106	£94,085	£48,512	£44,921	£28,209	£74,351	£59,131	£110,720
Case Study	**9**	**5**	**1**	**7**	**3**	**10**	**4**	**6**	**8**	**11**	**12**	**2**	**15**	**13**	**14**

Table 1

CONCLUSIONS

As an rough rule of thumb: if the rated speed multiplied by the rated load exceeds 1500 kgm/s a gearless machine is worth considering, but a regenerative drive only where the lift is likely to exceed 200,000-300,000 trips per annum.

Since the completion of the work energy prices have risen substantially and seem set to rise at a greater rate than anticipated, however it is inevitable the price of permanent magnet gearless machines will rise significantly in the near future as the price of the neodymium used to make the magnets has risen tenfold over the past year [8], whilst the prices of geared machines have not risen significantly over this time. Perhaps the worm gear is due for a renaissance!

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A View to The Application of Linear Motors in Vertical Transportation

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Key Words: Linear motors, roped, rope-less, elevator, elevators

ABSTRACT

Linear motors[1] have been considered to be used as propulsion means in different vertical transportation systems[2], as alternatives to conventional systems, in both roped and rope-less configurations. This paper is a summary and collection of points, ideas and comments from different related, albeit not all, texts and sources dealing with the advantages and challenges in this regard, with a conclusion in the end.

GENERAL CONCEPT

Figure1 shows a schematic illustration showing examples of different arrangements for linear motors, replacing propulsion means in conventional passenger elevators.

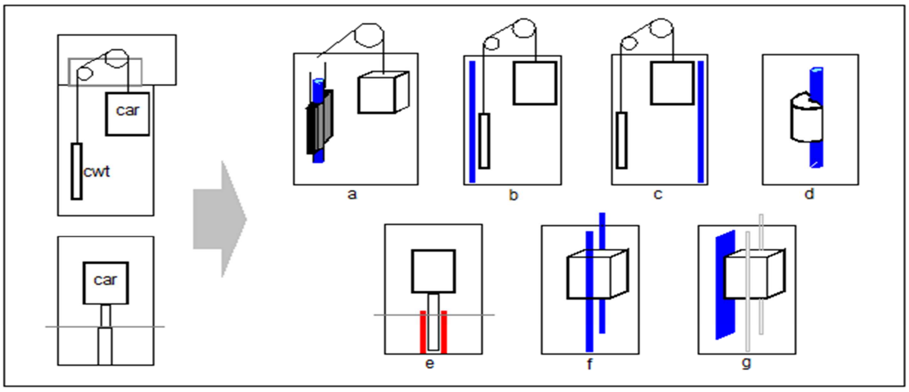

Fig. 1

Roped and rope-less alternatives are shown in blue and red. (Readers may refer to [1-5] for more detailed demonstrations and explanation).

POINTS, ADVANTAGES AND CHALLENGES

Wire rope itself. Extending the length of travel has limitation due to the extra load on the ropes, and also vibration. A limit of 700-800 meters according to ropes weight and safety factor criteria is

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[1] Electromagnetic motors capable of providing direct linear motion, similar in principles to their rotary counterparts. Particularly: Linear Induction Motors (LIMs), Linear Synchronous Motors (LSMs), and Linear Switched Reluctance Motors (LSRMs).

[2] Including passenger elevators and some related mechanisms, as well as other elevating or lifting devices used in different areas.

given in [6], and a 600-750 meters practical limit in [7]. A rise limitation of about 1200 meters is mentioned according to the weight and strength of ropes, car weight and safety factor, as well as vertical oscillations [8]. In gold mining applications, however, achievable depth by a single roped system is around a maximum of 3000 meter, yet not sufficient [9-11].

Ropes elimination consequences. With no ropes, there would be no counterweight leading to savings and improvements in hoistway and car spaces which consequently result in interesting new ideas for vertical
transportation in buildings: Having more than one car in a single shaft with more convenience, and having the movement of elevator cars not limited to a vertical path. These seem to be the most interesting points in the application of linear motor driven elevators, with consequent dramatic changes in building circulation and traffic patterns, according to [12-26] and also [1,2,5,8][1].

On the other hand, in a rope-less elevator, energy and power demand is higher due to the lack of counterweight which offsets a percent of payload in conventional roped systems. Without a counterweight, energy consumption could increase by a factor of three to eight [27]. Such systems would consume four to eight times the power a similar roped elevator's needs [7]. Removing the ropes and counterweight would probably increase power demand and energy losses by approximately 6-7 times [19].Another say regarding linear induction motors is that they are only 60-70% efficient in power use, compared to 90% for conventional rotary motors [13].

Elimination of Driving Unit and Mechanical Parts. In a roped configuration, with no traction machine and machine room, the exerted load on the building is reduced, as well as starting torque and current to overcome inertial forces. Overall mechanical efficiency is improved, and the inherent noise with gearings and pumps is eliminated [1]. In a rope-less system, lack of cables, gears and wheels provides a smoother and quieter ride [25]. Also maintenance requirements are minimal due to the lack of moving parts, cables and hydraulic equipment [26].

On the other hand, these eliminations will result in challenging issues: These would be safety and braking system for a rope-less elevator, and emergency operations and passengers rescue.

More complicated safety system [1], operation in the event of power failure [25,28], emergency stop problems in either direction, and feasibility of manual release for passengers [19], are issues considered and mentioned in this regard.

Other Challenges. Some other points also exist in the application of linear motors.

Technical issues. Maintaining the distance (air gap) between stationary and moving parts, i.e. between the rotor and the stator, is of importance in the systems with linear motors.
Attractive forces exist in there and such issue leads to changes in performance and speed, as well as loosening of the fixing bolts. Utilizing rolling guides, or a tubular motor instead of a flat one can help [2]. Attraction force working across the air gap tends to pull the secondary and primary parts against each other, which can be eliminated in perfectly symmetrical motors [29].

Utilization of linear motors causes the presence of a magnetic field in the car and also in the hoistway. CIBSE Guide D mentions a linear motor mounted directly on the lift car would expose the occupants to intense magnetic fields and, possibly, high noise levels [30]. Ishii states that the high leakage flux in superconducting magnets having no magnetic core, may disturb the surroundings (people) [8].

[1] Since some similar sources are referenced in different sections, references numbers may not be in numerical order hereafter.

Different environmental effects on such systems would be another issue. Questions and doubts regarding fire hazard, environmental impact upon drive mechanism, and oscillation in event of an earthquake, have been mentioned in [28]. Sensitivity of the system to dust is pointed out in [31].

Finally, thermal efficiency is another issue in linear motors. The properties of the motor depend on the temperature of its parts and temperature conditions would change motor conditions during the move [29]. In case of high duty cycles or in locked rotor conditions (moving element stopped), overheating can occur in linear induction motors [26].

Cost and Financial issues. A part of cost related issues is due to the power and efficiency problems previously mentioned. Another part relates to the production costs as well as maintenance and inspection. Miravete states that production costs problem due to the high motor length [31]. Another say is that the capital and operating costs of rope-less elevators would take decades to pay back through rents gained on the space they would save, equipment costs will also be high due, in part, to the need for full height linear motors [32]. Similar concerns are mentioned in [29,33].

OTHER AREAS OF APPLICATION

Areas for the use of linear motors in vertical transportation cover more fields and ideas, examples of those are as follows.

Otis offer a system which provide a combined movement path for elevator cars, in which vertical ride is by means of ropes (like a conventional traction elevator), and horizontal transport by linear motors [32].

Linear induction motor driven elevators, as well as linear electric actuators for doors and hatches, are of interest to be used on US Navy. Low maintenance and reducing shipboard manning play a role here [34]. Another example with 150% overload capacity and speed of 0.75 meter per second is mentioned in [26].

Elevators door mechanism is another example for the application of linear motors. Again low maintenance is a factor here, as door related problems are the cause of up to 40% of service calls [35]. More examples and applications can be found in [31,36,37].

Application in different environments would be another use for linear motors. Linear synchronous motor based systems can be used as freight elevators and vertical platform lifts in industrial areas [26]. Automatic baggage handling systems and leisure riders are other examples [31].

A low-rise linear switched reluctance motor elevator to serve a few levels, as a low cost solution for improving seniors mobility in their own homes, has been considered [38].

CONCLUSION

Application of linear motors in vertical transportation has been a subject over many years. They offer interesting advantages, and at the same time impose number of problems and challenges. Important point here is the conflict which exists and makes it not easy to have a quick comparison and decision regarding the replacement of conventional elevators and lifting systems: Extended travel height is an advantage, but having a linear motor along the whole shaft is costly – Maintenance is supposed to be lower, but in another term could be higher – mechanical efficiency could be higher, but power consumption is also higher in rope-less applications, and overall efficiency is a challenge – Savings would be in high rise buildings space, but costly maintenance and operation would exist, and so on. It can be concluded that application of linear motors in vertical transportation in the near future would be confined to specific areas and especial buildings, e.g. in especial industrial or army applications, or in a futuristic architectural design, where specific characteristics or necessities play a dominant role, and the overall positive points compensate for negative issues. However, such systems seem to have a strong potential to be alternatives to conventional elevators on a commercial scale, no matter if not in the next few years.

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The Reliance on Testing for Modernised Lifts

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INTRODUCTION

When a lift is nearing the end of its working life then it is usually time for a modernisation, this will encompass upgrading several major components of the lift system with newer and more efficient components than before. These are usually lighter in weight and smaller than previous equipment due to the technological advancements that have been made over the years, they are also easier to obtain. This is mainly due to the introduction of the Lifts Directive [1] and subsequent Lift regulations [2] that came into force July 1999; this opened the flood gates for all member states to be able to trade effectively and safely due to the conformance procedures and subsequent CE marking that can be enacted by law. This has further been reinforced by the latest Machinery Regulations [3] with its 'intended use' certificate of incorporation.

Unfortunately this system of compliance is not considered when using the modernisation model for lifts and subsequently a great deal of reliance is left down to the test procedures adopted, but do the test procedures cover all eventualities and leave the lift totally safe to use? Furthermore which test procedure do we adopt?

All of the components chosen for the modernisation have the appropriate CE mark but collectively when placed as a complete system do not afford the same seal of approval, this is due to there being no legislative requirements for the system calculations to be performed, furthermore the simple action of weighing the car and counterweight does not always take place.

This research intends to highlight the issues by following three modernisations from start to finish with a view towards testing to see if there are any obvious frailties that come to light. If there is no car weight or it has been guessed then the knock on effects filter through to most of the major components, traction calculations, emergency braking decelerations and safety gear decelerations and sliding distances all of which are fundamental calculations carried out for a 'new' installation are affected, but none of these are recorded or asked for on a test sheet. Without the back up of conformity procedures for a modernisation the test procedure and recording should be such that these calculations *must* have been carried out before the test or it *cannot* be completed and placed back into service.

METHODOLOGY

The methodology to achieve the main aim and objectives of this dissertation has been to research the 'new lift' conformity assessment procedures within the Lifts Directive [1] and subsequent Lift Regulations [2] and current codes and standards, which in turn back up the final test results against the modernisation guidelines and test procedures.

This can be shown by following through to completion the modernisation of a lift at;

The Brunel Shopping Centre Swindon Bay 14 lift3A Goods lift.

The following on site tests were carried out with the test engineer and the findings recorded.

- Traction tests as set out within BS 8486-1:2007 section 5.4 [4]
- Balancing of the car at 50% achieved through half of the rated load placed within the car and the current readings taken from the VVVF drive at the halfway point in both the up and down directions as the car is running. Weight is then added or taken away from the car accordingly until the readings are the same in both directions, the amount of weight that has

been taken out or added will be the required amount removed or added to the counterweight accordingly.

- Brake tests as in BS8486-1:2007[4]

The actual angle of wrap was measured on site using a tape measure and calculated.
The calculations for traction using Euler's formula as from Janovsky [5] and BS EN 81-1:1998 + A3:2009[6]. The calculations do not take into account the inertia of the diverters or machine and are calculated with 125% rated load and with the car near the lowest floor.
Acknowledgement that all of the relevant data has been supplied by the third party specialists who specify the tolerances for the 'intended use' and certificate of incorporation and of the maintenance company who supplied test data and information of the modernisation.
Only one scenario is included in this review and the findings require further research.

RESULTS

By looking at The Brunel Shopping Centre Swindon Bay 14 lift3A Goods lift the following data can be ascertained:

Brunel Centre Specification of Motor Details from Sassi

INSTALLATION		**GEAR**	
Type of roping	1:1	Quantity	1
Car Speed[m/s]	0.64	Type	MB95
Duty Load[Kg]	3000	Traction sheave position	To be defined
Car Weight[Kg]	3300	Ratio	1/58
Car Travel[m]	9	Traction Sheave Ø[mm]	650
Ropes weight[Kg]	50	Ropes no. x Ø[mm]	6 16
Counterweight [Kg] (%)	4800	Angle groove[◦] U	30
Comp Chains[Kg] (%)		Angle undercut groove[◦]	87
Shaft Efficiency[%]	80	Pitch of grooves[mm]	21
Machine	TOP	Brake voltage DC	185
Type of diverters	On	Flywheel	
Number of diverters	1	MAX static load[Kg]	12000
Out of balance load [Kg]	1,582	MAX out of balance[Kg]	1,648
Static load[Kg]	11,150	Sync traction speed	0.59
Minimum alfa angle[◦]	156		
Diverter Ø[mm]	650	**MOTOR**	Ziehl Abegg
Acceleration[m/s^2]		Frame size	VFD200L-4
Electric cable mass[Kg]		Power[kW asyncr]	23
Ropes breaking load[Kg]		Poles	4
		RPM	1100
		Voltage [V]	400
		Frequency [Hz]	38
		Regulation	VVVFClosedlo
		Sts./h.	240
		Running current	49
		Starting current	96

Table 1.

The car weight had been estimated and not correctly weighed *before* and *after* the modernisation, the actual weight of the new car was found to be 2700 kg some 600 kg lighter than estimated.

The original diverter had been kept and is positioned within the shaft on the steelwork, the new raft with the new machine positioned above in the motor room. The actual on site angle of wrap was 133^0 instead of the suggested minimum of 156^0.

By looking at the above information and what is actually fitted on site we can carry out some basic calculations to back up the test procedure findings. Firstly the Critical Traction Ratio can be calculated at the minimum angle of wrap as suggested, also with what is actually onsite and with the varying weights given against actual. The calculations do not take into account the inertia of the diverters or machine. Eulers Critical Traction ratio as shown in EN 81-1+A3:2009[6] and used in Janovsky [5].

$$\frac{T1}{T2} < e^{f \times \alpha}. \qquad (1)$$

Eq 1.

f = friction factor;
α = angle of wrap of the ropes on the traction sheave in radians;
$T1$, $T2$ = forces in the portion of the ropes situated at either side of the traction sheave.

Ratio of tensions for suggested and actual

T1 Suggested	69663.75	T2 Suggested	47088
T1 Actual	63777.75	T2 Actual	41202
T1/T2 Actual	**1.547929**	T1/T2 suggested	**1.479438**

Table 2.

Showing Critical Traction ratio for suggested and actual

Traction condition	Coefficient of friction μ	Friction Factor f	Angle of suggested wrap (156^0) α	Critical traction ratio(suggested) $e^{f\alpha}$	Angle of actual wrap (133^0) α	Actual Critical traction ratio $e^{f\alpha}$
Normal Loading	0.10	0.19	2.72	**1.66**	2.33	**1.54**
Emergency Braking	0.09	0.17	2.72	**1.61**	2.33	**1.50**

Table 3.

DISCUSSION

What can be seen from Table 2 is the discrepancy for the weight of the car has led to an increase in the ratio of tensions between actual and suggested. This would not have had an impact solely on its own in this case. By looking at Table 3 Critical Traction ratio (suggested) at the angle of wrap of 156^0 Euler's formula would still hold true for both instances, however coupled with the decrease in the actual angle of wrap to 133^0 has led to the figure for normal loading to be equal to the actual T1/T2 ratio so in essence Euler's formula holds true although with any increase in acceleration traction will be lost as the emergency braking figure shows. In reality the lift passed the test procedures prescribed and some two years later traction was being lost under the emergency scenario. Many factors could have an influence on this, the groove angle would only need to increase by a couple of degrees to decrease the friction factor to make the formulae untrue, therefore wear on the sheave due to set up or poor manufacture could have contributed over the two year period tipping the balance. Whilst testing the lift the angle of wrap would not have been asked for to check or stipulated neither for the weight of car, the test sheet adopted was BS5655:10[7], these calculations, angles and weights should have been carried out and set up before installation. This immediately highlights how critical these factors are even with the 25% redundancy that is built in to the calculation. It also highlights that even after the unit has been tested for traction it has in reality 'papered over the cracks' that the modernisation relies on the testing process to prove the system is safe, rather than backing up the calculations of the desired new system, however this still does not specify how close the ratio becomes before we need to increase the angle of wrap or change the type of groove and angle or undercut to account for wear and tear of the system. The installation company embarked to rectify this issue by installing an additional diverter under the raft to increase the angle of wrap to 180^0 this being the easiest and most practical solution; however this would decrease the life of the ropes due to the reverse bends. The unit was again tested to ensure traction is not lost using the Dynamic Braking test as laid out in BS 8486-1:2007 section 5.4[4]. However the new calculation was never carried out with the new figures and the braking force of the brake never checked.

CONCLUSION

Although the majority of companies embarking on a modernisation of a lift would indeed carry out the fundamental task of weighing the car before modernising and calculating the weight of the sum of the components to be added to ensure the correct figures for calculating traction, there are a lot that do not. The above project is a prime example where estimates and assumptions are made and not checked and although on this occasion the test procedure adopted did result in traction not being lost at time of test some two years later that was not the case.

There are already enough safety codes and standards that give guidance on how to successfully achieve a safe and reliable lift system from a major modernisation; however these are just guidance and not legally binding which results in the reliance on testing procedures again and again. But the test procedures themselves, although comprehensive, do not fully cater for a major modernisation as yet and could be improved to force those that 'do not' to '*do*'.

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Symposium on Lift and Escalator Technologies

The Use of Multi Car / Single Well Lift Systems To Add Value

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INTRODUCTION

The goal of any passenger lift system design for modern buildings is to meet the required performance criteria whilst occupying the minimum amount of valuable floor space. Nowhere is this goal more relevant than in the development of major speculative office buildings where every additional square metre of space that can be made available as lettable area is a constant focus of any competent design team.

This paper focuses on the added value a multi car / single well lift system brought to the development of The St Botolph Building, a large speculative office development in the City of London that was completed in 2010.

BACKGROUND

Office development in London during the start and middle of the last decade was an exciting time where speculative buildings were easily funded and driven through planning and construction fast to meet the burgeoning demand from a booming global economy.

Property developers, particularly speculative ones, are nevertheless focused on maximizing the value of their schemes and one of the key drivers to value is lettable area. Put simply the more space there is in an office building that can be let to tenants and generate rental income, the more value there is to the developer.

The St Botolph Building is a prime example of the type of high-end, speculative office development that characterized the London market last decade. The site is located at the Eastern boundary of the City of London in Aldgate, and was originally to be the home of a 52 storey tower which would have been the first building in the City to offer over 1,000,000 ft^2 of lettable office space.

The developer’s appetite to undertake such a project speculatively, without a pre-let tenant, was however somewhat abated and the scheme reverted to a more modest building that had received planning permission back in 1999.

The vertical transportation consultant for the project was Grontmij, formerly Roger Preston & Partners, whose design team was led by the author.

THE CHALLENGE

The St Botolph Building is not a tall building standing as it does some 60 m above street level; it is however still a large building offering over 500,000 ft^2 of lettable area with some of the thirteen above ground floorplates in excess of 40,000 ft^2.

With a requirement to occupy the building at a theoretical density of one person per 10 m^2 NIA less 15% absenteeism (equivalent to 1 person per 11.7 m^2 NIA), the lifts need to serve around 5000 people and deal with a theoretical morning peak flow rate approaching 750 people per five minutes.

As is always the case with large office buildings the main passenger lifting strategy is very important to define the building’s cores and determine the net lettable areas.

Initial lift traffic analyses (completed to the general recommendations of the British Council for Offices 2005[1] and the Chartered Institution of Building Services Engineers CIBSE Guide D 2005[2]) showed the building required two groups of conventional single deck passenger lifts; a group of eight and a group of six, operating in a high / low peak time zoned configuration.

Although this approach is conventional and with many precedents, we wished to innovate and explore options to reduce the space taken by the lift cores and thus increase the net lettable area for our client.

Working in close conjunction with both the developer and the architect, the vertical transportation consultant team initially reviewed a double-deck option before discounting it as too energy hungry in off peak traffic, requiring too much space for the headroom and machine room and being ill-suited to the variable floor to floor heights that existed in the base of the building to accommodate future trading floors.

The team then turned to consider other ways of getting more lift performance from less space.

THE OPPORTUNITY

Multi car lift systems take their design lead from double-deck lifts in as much as they feature two lift cars operating in a single well, but then evolve the concept by running both cars independently. This configuration has many potential advantages when compared with double-deck. Independent cars are lighter and therefore require smaller machine sizes with commensurately lower energy consumption; headroom and machine room space requirements become more appropriate to smaller building forms and variable floor heights can be accommodated with ease.

Various lift manufacturers are known to be researching and developing multi car / single well systems though ThyssenKrupp’s TWIN is the only system that has currently reached the global marketplace. The first such system was installed on a pilot project at Stuttgart University in 2002 since when more than 100 installation have been commissioned around the world.

The key opportunity however was all about space, and when initial analysis of The St Botolph Building suggested a multi car solution could potentially return more than 30,000 ft^2 back to net lettable (at an estimated, amortized value to the scheme of more than £4M), it quickly became apparent that a further analysis of the potential value of the system to the project was more than justified.

The developer was rightly focused on understanding all the issues around adopting such an innovative system in their building. The benefits were clear; more space to let. Potential risks were predominantly commercial in selecting a solution that could only be provided by one supplier but there was also a need to understand the safety credentials of the system, to test the market’s acceptance of multi car systems and their prerequisite full destination control, and to consider the architectural requirements of the system particularly with respect to two lobby levels.

Detailed lift traffic analysis showed that the conventional, single deck, two groups of eight and six solution could be replaced with a single multi car group of 16 cars running in a single core of eight wells.

Detailed due diligence over the next few months concluded in January 2007 with the developer’s commitment to use a multi car system in The St Botolph Building and create what has become the largest single group of multi car lifts currently operational anywhere in the world.

DELIVERING TWIN

By this stage the architect had started to explore the opportunities that the multi car core solution could provide and were looking to place the lifts in a central atrium extending the full height of the building and, just to add to the challenge, all the cars were to be scenic wallclimbers.

An expansive reception area greets people entering the building where escalators transport passengers up and down to two lift lobbies at upper and lower ground respectively. 16 wallclimbers operate in eight lift wells; the upper cars are 1600 kg / 21 person running at 2.5 m/s, the lower cars also 1600 kg / 21 person running at 2.0 m/s. In peak traffic periods the upper cars serve from the upper ground lobby to a "high" zone at levels 8 to 13 inclusive, the lower cars from the lower ground lobby to a "low" zone at levels 1 to 7 inclusive. The system is full destination control via bespoke lobby terminals.

Less abled people and those not wishing to use the escalators travel to the lower ground lobby via a shuttle lift which also descends further to link with the basement car park and cycle parking.

The need to provide unhindered access for all created another challenge for the design team. To comply with the system's rigorous safety requirements, the cars must remain a minimum distance from each other at all times. This not only defines the floor-to-floor height between the main lobby levels but also means the lower cars cannot access the top two floors of the building. With less abled passengers arriving at the lower ground lobby it was therefore necessary to find some way of getting an upper car to serve the lower ground floor.

This was achieved by creating a "virtual pit", a deeper than normal pit that allows four of the lower car to travel down beneath the lower ground floor level sufficiently for the upper cars to serve the lower ground lobby and thereby provide lift service to the upper two floors of the building. The use of the "virtual pit" is determined automatically by the destination control system. Special service buttons are located on select call panels at all floors allowing the control system to recognise a call from a less abled person and provide lighter loaded cars, extended walking time allocations and slower door times.

Safety was a key area for the due diligence and design development process to focus on. It was clear from the outset that any independent multi-car system could not comply with the fundamentals of the EN81-1 code as it currently exists, so the manufacturer had secured Notified Body approval from the German TÜV to demonstrate compliance with the Essential Safety Requirements of The Lift Directive. At the request of the project team this approval was also subsequently assessed and ratified by LRQA in the UK.

The system calls for many unique design characteristics including:

i) A special door lock safety circuit as most landing doors are served by both cars.

ii) An electronic collision prevention device system that uses as its basis the intelligence of the destination control to prohibit car movement that would result in the cars needing to move within a pre-defined distance from each other.

iii) Independent position controllers that communicate with each other and constantly check the position of the cars in the well. Car position is determined via a laser barcode system for enhanced, consistent accuracy. Should the position of the cars breach the pre-defined safety space, an emergency stop will be initiated on both cars simultaneously. In the unlikely event that this fails to bring the cars to a stop, safety gear will be applied, again to both cars simultaneously.

iv) The collision protection system is calibrated on a site-specific basis such that in any loading and speed condition the safety space on the roof of the lower car is always in accordance with the requirement of EN81-1.

v) Colour coding of major components to provide differentiation between the upper and lower cars. This colouring is most prevalent in the machine room and pit but also extends to the car roofs and bases as part of the architectural aspirations.

As with all good design in today's world, energy efficiency was embedded and allowed the building's BREEAM assessment to take both available credits for energy efficient vertical transportation design. The escalators feature variable speed running, automatically slowing to a reduced speed during periods of no traffic to conserve energy and extend lifetime. The destination control system constantly monitors actual demand and minimises the number of cars required to be in service to maintain predefined average waiting times. The intelligent control system cycles the cars taken out of service so that hours in service are balanced across the group. The bespoke destination control screens feature automatic power down during periods of inactivity, again to conserve energy consumption and extend screen lifetime.

SUMMARY

The St Botolph Building is a global landmark that clearly demonstrates the real value that the adoption of an innovative multi car passenger lift system can deliver. Such a successful outcome is founded on many variables such as economic climate, developer's appetite for risk, architectural design, etc., but for many substantial commercial developments multi car systems should be considered as a viable option to deliver

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Adam Scott is a Technical Director at Grontmij, the fourth largest multi-disciplinary engineering consultancy in Europe. He heads up the specialist Vertical Transportation group and has worked in the lift industry for the last twenty years. He is a past Chairman of the Chartered Institution of Building Services Engineers (CIBSE) Lifts Group and continues to represent CIBSE on the British Standards Institute (BSi) MHE/4 committee. He recently chaired the CIBSE Guide D Steering Committee in the production of the latest 2010 edition and was part of the Vertical Transportation Technical Committee for the 2009 edition of the British Council for Offices (BCO) guide to specification.

Symposium on Lift and Escalator Technologies

Designing Elevator Installations Using Modern Estimates of Passenger Demand and Currently Available Elevator Technologies

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INTRODUCTION

The quantity of passengers to be transported by a lift system is a primary consideration in lift system design.

Research indicates that passenger demand in modern office buildings is significantly different to the assumptions formed many decades ago, but still applied to most modern designs.

The number and type of lifts required to provide a proper and efficient lift service may need to be revised based on these findings. These changes in lift system design have economic and environmental consequences that are favorable.

HISTORICAL REPRESENTATIONS OF PASSENGER DEMAND

A plot of passenger demand depicts the level of passenger traffic in a group of lifts over a period of time. Figure 1 shows estimated passenger demand for an office building over the working day with a population of 1000 people. This has been generated by applying the example of office passenger demand presented by Strakosch [1] over 40 years ago. In this representation of passenger demand, passengers travelling up are shown in the top section of the graph, with passengers travelling down in the lower section.

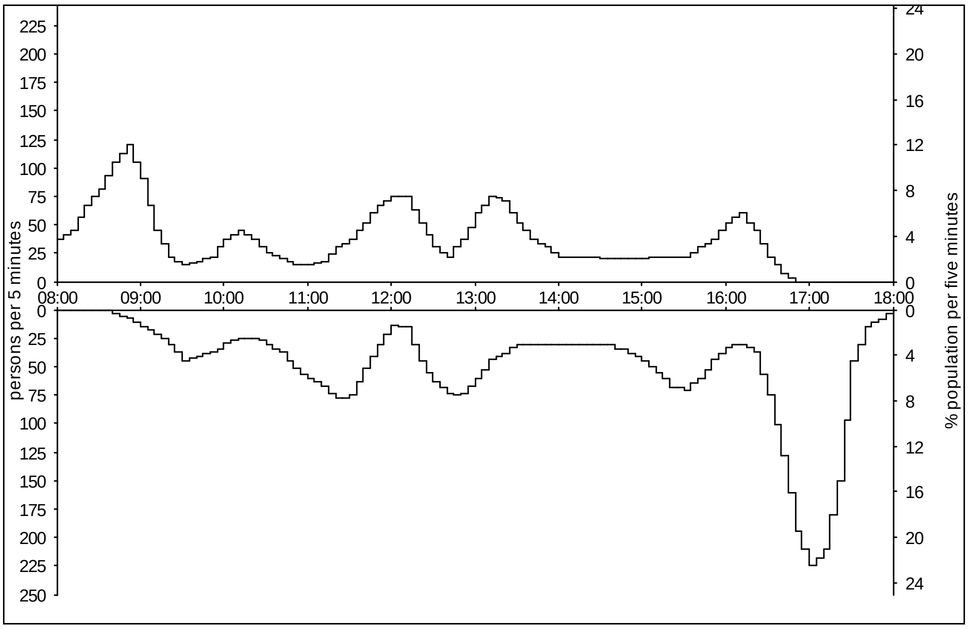

Fig. 1 Strakosch Passenger Demand

It was generally believed that the most demanding traffic type was the morning up peak. This belief was reinforced by research conducted by Barney that showed that lifts have between 20% and 60% more capacity during non up-peak conditions [2].
It has been assumed by many in the lift industry that most office buildings had a pattern of passenger demand similar to those in Figure 1.

MODERN BUILDINGS

How people use lifts and the traffic patterns that their use generates has changed since 1923, when Basset Jones published formulae for the expected number of stops a car will make during a round trip [3]. Summarising the results of a series of peak time traffic surveys carried out between 1993 and 1997 Peters concluded, "Morning traffic peaks are less marked in buildings than they were when traditional up peak design criteria were formulated. In work-related buildings occupied during the day, the busiest period appears to be over the lunch period" [4]

In 2000 Siikonen presented a traffic pattern that represents traffic measured in a modern installation [5].

This pattern is quite different from the traffic pattern presented by Strakosch. Siikonen shows a lunch up peak that is the same size as the morning up peak. Additionally, the down peak at lunch is more intense than the evening down peak. These observations raise a question, are the differences in the patterns due to the unique nature of the building studied by Siikonen or have traffic patterns changed over the years?

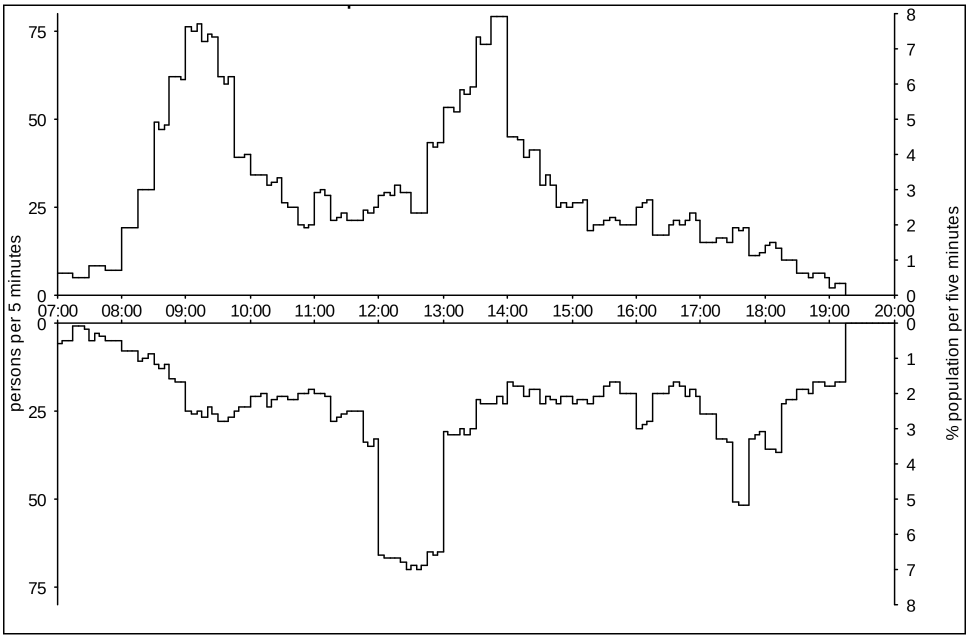

Fig. 2 Siikonen Passenger Demand

In order to better understand modern lift traffic, data was collected at a number of office buildings in different parts of the world including Europe, North America and the United Arab Emirates.

In most cases data was collected by manual count. However, in one building, data from three groups of lifts in a corporate headquarters building was gathered electronically.

Figure 3 shows the results of lift traffic surveys for seven separate groups of lifts [6]. The surveys were undertaken applying a methodology defined by Peters and Evans [7]. The passenger demand is normalised against observed population to allow results to be compared between buildings

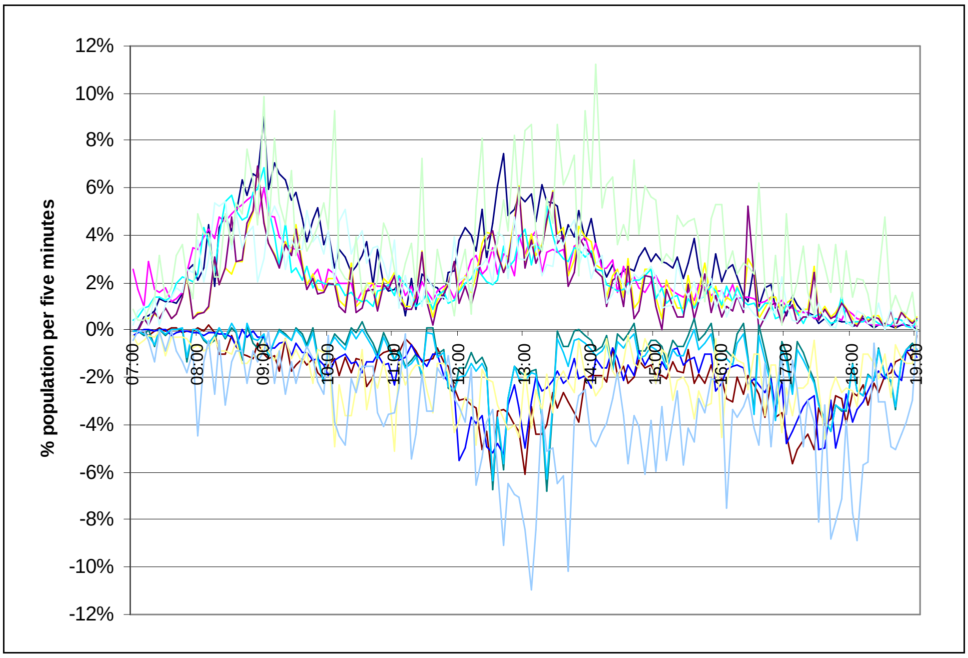

Fig. 3 Observed Passenger Demand

On average, the mix of traffic in modern buildings during the morning up peak was found to be approximately 85% incoming, 10% outgoing and 5% inter-floor.

On average, the mix of traffic in modern buildings during the busiest part of lunch was found to be 45% incoming, 45% outgoing and 10% inter-floor.

None of the groups surveyed have the sharp down peak that is seen in the Strakosch pattern. In modern office buildings with professional workers, a significant portion of office workers are working later than in previous years.

SIMULATION

Simulation can be used to evaluate lift system performance based on modern traffic levels and traffic patterns. Simulation can also apply modern dispatching algorithms such as those based on destination selection. The performance of advanced drive and door systems can also be modelled.

Figure 4 shows the relative performance of two lift systems in a hypothetical building as determined by simulation and using modern traffic mixes.

The Up Peak Round Trip Time (UPRTT) method indicated this building should be fitted with 6 1350kg lifts operating at 2.5m/s.

The traditional system used a generic group control system and had the performance criteria used in the UPRTT method. The Optimized system used a proprietary destination based system and high performance door and drive systems

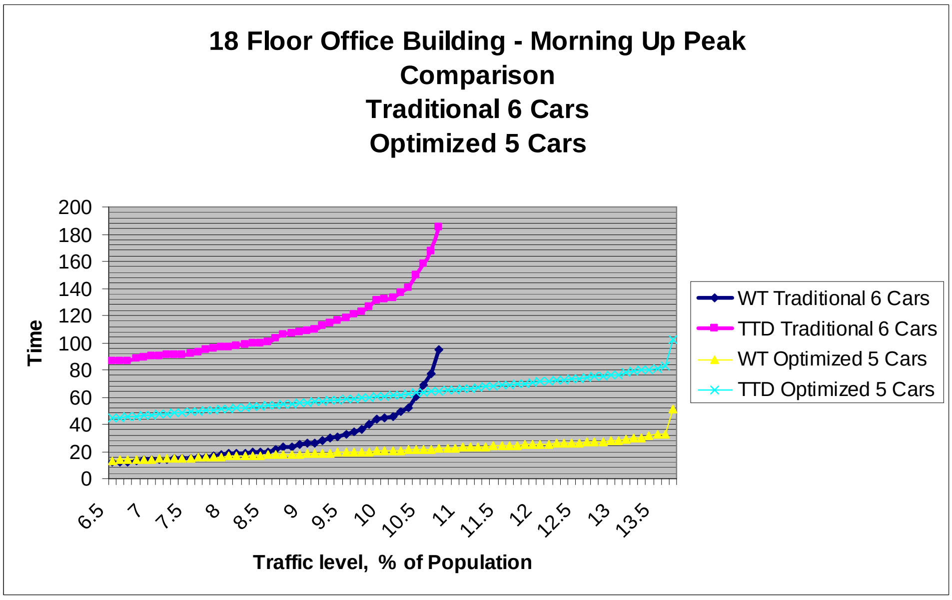

Fig. 4

The Optimized system with 5 cars outperforms the Traditional system with 6 cars.

CONCLUSIONS

This research has indicated that in many cases it may be possible to install fewer lifts than would be indicated using the UPRTT method and still achieve excellent traffic handling.

Simulation was found to be a better method of predicting lift system traffic handling performance than the UPRTT method

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