Springer

KSME

Journal of Mechanical Science and Technology 29 (7) (2015) 2645~2651
www.springerlink.com/content/1738-494x(Print)/1976-3824(Online)
DOI 10.1007/s12206-015-0510-6

Braking performance analysis of an escalator system using multibody dynamics simulation technology †

Chan Jong Park* and Gero Gschwendtner

Escalator System and Innovation Centre, OTIS GesmbH, Modecenterstrasse 17, 1110 Vienna, Austria

(Manuscript Received December 4, 2014; Revised April 9, 2015; Accepted April 9, 2015)

Abstract

An escalator brake system composed of an operational brake and an auxiliary brake is one of the most critical components that directly influence passenger safety. Therefore, understanding the braking performance of an escalator system at the early design stage is imperative. In this article, the application of multibody dynamics simulation for escalator industry is discussed. This study proposes an efficient multibody dynamics simulation modeling approach that considers the dynamic effects of a step band, handrail band, and passenger traffic load, which requires considerable computational resources when the conventional method is employed. The approach also covers a comprehensive simulation modeling of drive machine with gearbox, main drive chain band, operational brake system, and auxiliary brake system to evaluate the escalator's braking performance at the system level. The simulation model is verified with actual measurement data and employed to investigate potential worst case braking scenarios. The dynamic influences of these braking scenarios on the escalator system are discussed as a result.

Keywords: Auxiliary brake; Braking performance; Design of experiment; Escalator system; Multibody dynamics; Operational brake; Relative coordinate formulation

1. Introduction

Activation of an escalator's brake systems brings a moving escalator at operating speed to a standstill within a short period of brake time. The brake systems are the most critical component from the perspective of passenger safety. If a loaded escalator makes a hard stop during a normal operational braking event or an auxiliary braking event, passengers might feel uncomfortable or have a risk of falling especially in down running escalators.

Several engineering efforts have been exerted to determine the influence of braking on passenger safety. The European standard (EN115:1995+A1:1998, EN115-1:2008) [1, 2] suggests various stopping distances depending on different rated speeds and a maximum deceleration to ensure the safe stop of an escalator. The American safety code (ASME A17.1, 2004) [3] also specifies a certain maximum deceleration as a braking performance requirement. Al-Sharif [4] conducted an experimental investigation to understand the relationship between passenger ride comfort and kinematics during an escalator stop. He concluded that the maximum value of deceleration in the traveling direction is the most important indicator in deciding passenger ride comfort in case of braking.

Stopping distance and deceleration are directly measured on the step band under unloaded and loaded escalators at an operating speed to prove compliance to the safety codes on the escalator braking performance. However, the measurement process is not only time consuming, but also requires considerable effort to carry out under passenger load conditions.

When an escalator comes to a standstill during a short period of brake time, both the passengers and the escalator system experience the most severe internal impact forces. If escalator safety components are not designed properly for the critical situations that cause huge internal impact forces, it might lead to a failure mode, resulting in accidents that may injure passengers. Therefore, understanding the system level dynamic behavior of escalator safety components is essential during the early design stage. A typical escalator system is shown in Fig. 1.

The development of modern engineering technology and high performance computing systems has enabled computational analysis to replace rapidly conventional trial-and-error design and testing process in every industry, thereby minimizing development time and cost. In the last two decades, Park et al. [5-7] conducted several pioneering trials to investigate the dynamic characteristics of an escalator system using the

*Corresponding author. Tel.: +43 1 61005 1185, Fax.: +43 1 61005 1146
E-mail address: chan-jong.park@otis.com

† This paper was presented at the Joint Conference of the 3rd IMSD and the 7th ACMD, Busan, Korea, June, 2014. Recommended by Guest Editor Sung-Soo Kim and Jin Hwan Choi

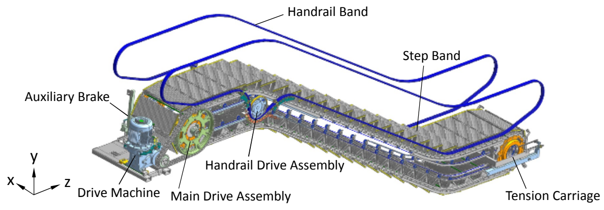


Fig. 1. Escalator system.

computational analysis methodology. However, they did not consider driving mechanisms, such as drive machine, brake systems, and main drive chain band because of the efficiency of numerical analysis and hardware capacity at that time.

In this study, an efficient multibody dynamics simulation modeling approach is employed to consider the dynamic effects of step band, handrail band, and passenger load, which consume considerable computational resources if conventional simulation modeling approach is applied. This study also covers a comprehensive simulation modeling of a drive machine with gear train, main drive chain band, operational brake system, and auxiliary brake system to execute braking performance analysis of a complete escalator system. The simulation model is validated with measurement data taken from an actual escalator. Using the verified simulation model, the dynamic stability of a major safety part is investigated with respect to system parameters such as different types of brake system activation, passenger loading conditions, and operating conditions of the main drive chain band. Therefore, the simulation results can provide design engineers with important information for the most critical situations and devise measures to prevent them even in the early design stage.

2. Multibody dynamics simulation for escalator industry application

As modern mechanical systems are becoming increasingly complex and sophisticated, multibody dynamics simulation is quickly gaining popularity in the investigation of system level dynamic behavior in many industries. In the last four decades, several dedicated multibody dynamics simulation software such as ADAMS, DADS, SIMPACK, and RecurDyn have been developed to perform large-scale nonlinear dynamic analysis. Because of the necessity of system level dynamic analysis for a wide range of engineering applications, even commercial FEA codes (e.g., ANSYS Workbench, Altair HyperWorks, SIMULIA Abaqus) and 3D CAD commercial codes (e.g., Autodesk Inventor, Solidworks, Siemens NX, Pro/Engineer) have been integrating small-scale multibody dynamics simulation features into their product-lifecycle-management (PLM) platform.

Two principal approaches in multibody dynamics analysis are present, namely, absolute coordinate and relative coordinate formulation. Each approach has advantages and drawbacks as described below. The absolute coordinate formulation builds up equations of motion using the Cartesian coordinate system where each rigid body has six degrees of freedom in three-dimensional (3D) space. Kinematic movements of each body are determined by defined joint constraint equations. Therefore, the formulation of a complex simulation model is quite straight forward, and accounts mainly for the invention of the earliest commercial multibody dynamics simulation codes such as ADAMS and DADS based on this approach. However, these codes require an efficient and accurate numerical integration algorithm to handle large numbers of equations of motion and joint constraint equations in Differential-algebraic-equation [8] formulation simultaneously. By contrast, the relative coordinate formulation establishes the equations of motion using the minimum number of independent joint coordinates. Therefore, the difficulty of numerical analysis is minimized dramatically compared to the absolute coordinate formulation approach, even though the constraint equations exist because of the cut-joint of the closed-loop system. However, building the formulation of the simulation model is complicated. Commercial codes, such as SIMPACK and RecurDyn, were developed based on the recursive formulation algorithm using the relative coordinate system. These codes generate equations of motion automatically with the minimum number of independent joint coordinates.

Other contributors that cause difficulties in the numerical analysis, such as large difference of inertia properties among the bodies, Coulomb's friction effect and discontinuous contact-impact phenomenon also exist. These difficulties can be overcome by selecting the appropriate physical properties and numerical integration algorithm. The efficiency of numerical analysis is not always considered a top priority especially in the escalator engineering industry. The following engineering aspects also have to be considered when appropriate multibody dynamics code is selected for its applications.

- Handling large numbers of kinematic degrees of freedom for escalator application.
- Handling complicated and huge 3D geometry.
- Productivity and reproducibility in modeling process.
- Modular- and subsystem-based modeling approach.
- Efficient modeling for standard mechanical connection elements (e.g., gear pairs, chain band, belt, bearing).

After considering the aforementioned aspects and characteristics of the escalator engineering industry, RecurDyn [9] is selected to execute the braking performance analysis of an escalator system in this article.

3. Simulation modeling of an escalator system

3.1 Step band

Each step has three step chain links on both sides. Both sides of the step chain band are connected to each other through a chain axle at every three step chain link. The front part of each step is attached to the chain axle to transfer the driving force from the step chain band to the step band. Two step rollers are attached directly to the rear part of each step.

Therefore, two different tracks for the step rollers and step chain rollers along escalator truss structure are formed.

Main drive sprocket engages with each step chain roller to drive the step band. The mass of step and step chain assembly functions as an additional rotational moment of inertia on the main drive assembly. The effects of inertia on the steps, step chains, chain axles, and rollers can be described as Eq. (1).

$$I_1 = N_1 m_1 r^2 , \tag{1}$$

where N_1 denotes total number of steps, m_1 is the mass of a step and step chain assembly per unit length, and r is the pitch radius of the step chain sprocket. The step band generates both an additional inertia effect on the main drive assembly as well as friction forces against the movement caused by the dead-weight of moving step parts. A friction force is generated by the roller bearing on the step rollers and step chain rollers and can be considered as the following formula applied directly on the main drive assembly as a torque.

$$\tau_1 = 2m_1 g\left[2N_2 + N_3\cos(\alpha) + N_3\sin(\alpha)\sin\left(\frac{\alpha}{2}\right)\right]\mu_1 , \tag{2}$$

where g denotes acceleration of gravity, N_2 is the number of flat steps in the landing area, N_3 is the number of steps in incline area, α is an incline angle, and μ_1 is the bearing friction coefficient of step rollers and step chain rollers.

3.2 Handrail band

An escalator has two continuous handrail bands on both sides. A sliding friction is generated between the inner layer of the handrail band and handrail guidance profile. Friction forces along the linear guidance profile are calculated based on the effects of gravity with a friction coefficient. Friction forces on different curvatures along the handrail guidance profile are derived by Eytelwein's formula [10] assuming that the centrifugal effect from handrail mass is sufficiently small because of the slow handrail speed.

$$f_i = \begin{cases} f_{i-1} + m_i g\cos(\alpha) & \text{if, } i \text{ at linear profile} \\ f_{i-1} e^{\mu_2 \theta_i} & \text{if, } i \text{ at curvature profile}. \end{cases} \tag{3}$$

In Eq. (3), f_{i-1} is a cumulative handrail friction force until previous handrail guidance profile, m_i is a mass of the handrail segment, μ_2 is a friction coefficient between the inner layer of the handrail band and handrail guidance profile, and θ_i is the contact angle around the curvature profile.

The stick-slip sliding friction characteristics of the handrail band are realized in the simulation model by employing the modified Coulomb's friction model proposed by Threlfall [11] as described in Eq. (4). Therefore, the stick-slip sliding friction force of the handrail band can be applied efficiently toward the opposite direction of the handrail movement.

$$\mathbf{f_f} = f_t \frac{\mathbf{v_t}}{v_t} \cdot \left[1 - e^{-\frac{3v_t}{v_\varepsilon}}\right] \qquad if, v_t < v_\varepsilon \tag{4}$$

where f_t is the magnitude of total handrail friction force, $\mathbf{v}_t$ is the handrail velocity vector, v_t is the handrail speed, and v_ε is the characteristic speed where steady-state sliding movement starts. Total handrail friction force is applied directly on the main drive assembly based on the geometrical relationship between the handrail drive pulley and other handrail drive sprockets.

3.3 Drive machine

Drive machine consists of an electric motor, operational brake system, gear train, and gearbox output shaft sprocket. The nominal rotational speed of the gearbox output shaft sprocket is maintained by reducing the fast rotating speed of the motor shaft by the gear ratio from the gear train. As a result, the torque generated from the motor is increased by the gear ratio and transmitted into the gearbox output shaft sprocket.

The gear ratio of the gear train is implemented as a kinematic constraint between the motor shaft and gearbox output shaft. Efficiency of the gear train is considered separately for driving and driven conditions. Inertia effect of the gear train is considered as an equivalent mass and moment of inertia. Therefore, multiple stages of the gear train inside the gearbox can be implemented efficiently in the simulation model instead of considering actual gear-tooth meshing contacts that usually consume huge computational time.

3.4 Operational brake system

An operational brake system is implemented in the drive machine and intended to stop running escalator on both directions within a certain stopping distance and deceleration for safety. This system also maintains an escalator stationary. Dual circuit operational brake system consists of a flywheel as the brake drum, two independent brake levers with brake pads, two brake springs, and a solenoid. The system is generally installed in the fast rotating motor shaft to minimize the required brake torque to stop an escalator. A common practice to comply with code requirements for the stopping distance and deceleration is to adjust the moment of inertia of the flywheel.

The solenoid forces are generated to overcome the brake spring forces and remove the braking torque during normal operating conditions. After a signal from the controller deactivates the solenoid forces, the braking torque is applied on the brake drum by the friction forces on the brake pads because of the spring forces. Mechanical and electrical time delays from the brake springs, controller, and solenoid place the escalator system in a free-wheeling condition, in which the escalator system is governed only by the inertia, gravity, and internal

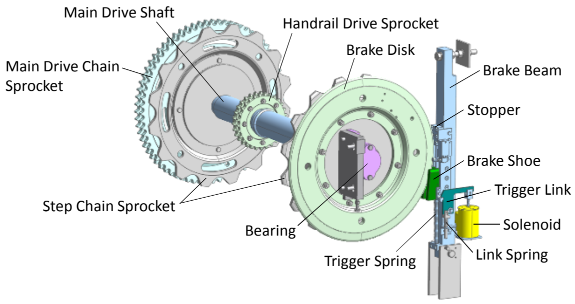


Fig. 2. Auxiliary brake system.

system friction for a short period.

3.5 Auxiliary brake system

The auxiliary brake system is an additional brake intended to increase the safety of an escalator in emergency. An escalator with a rise of more than 6 m should be equipped with an auxiliary brake system according to the safety code of EN115 [1, 2]. The installation of an auxiliary brake system below 6 m rise is also recommended for public escalators. ASME A17.1 [3] requires a device that causes the application of a brake system on the main drive shaft to stop an escalator if the drive chain loses connection with the drive machine. In this study, a frictional wedge-type auxiliary brake system installed on the main drive shaft is considered for analysis as shown in Fig. 2.

Solenoid force can overcome the link spring force to deactivate the auxiliary brake system under normal operating conditions. After a signal from the controller causes the solenoid to de-energize, the trigger link rotates around the hinge point, which is attached to the brake beam, and the brake shoe is disengaged from it through the link spring force. As a result, the brake shoe is pushed upwards by the trigger spring and generates braking force on the main drive shaft by its engagement with the brake disk. The braking force on the brake does not exceed a certain level because of position of the stopper. Therefore, the braking force can be adjusted based on the combination of longitudinal position of the brake beam and vertical position of the stopper.

Mechanical and electrical time delays from the springs, controller, and solenoid in the auxiliary brake system also cause the escalator system to free-wheel for a short activation time.

3.6 Interfaces between drive machine and truss

Four machine feet are supported on two beams across the truss in the upper landing area. The feet are fixed on the two beams with four clamps to maintain the position and orientation of the machine in its installation position during operation. The interface of the four machine feet with two beams and four clamps is implemented by compression only surface-to-surface contacts with steel-to-steel contact properties. Therefore, the combination of bolt reaction forces on the four clamps and the reaction forces on the four machine feet determines the dynamic behavior of the drive machine.

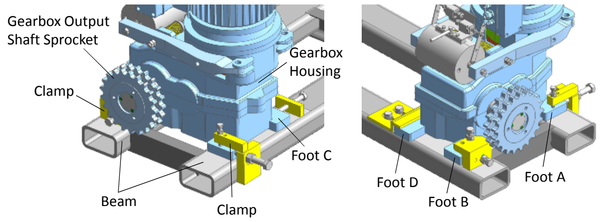


Fig. 3. Interface between drive machine and truss.

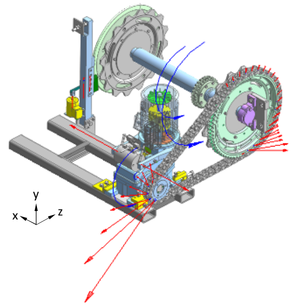


Fig. 4. Escalator simulation model for braking performance analysis.

3.7 Passenger traffic load

Passenger traffic load on an escalator system not only increases the additional moment of inertia in the system, but also applies additional torque on the main drive shaft because of the effects of gravity and friction. Therefore, the influence of passenger traffic load is considered as the following two formulae in the simulation model.

$$I_2 = N_3 m_2 r^2 \tag{5}$$

$$\tau_2 = m_2 g\left[N_3 \sin(\alpha) \pm \left(2N_2 + N_3 \cos(\alpha)\right)\mu_1\right] r. \tag{6}$$

In the equation, m_2 is the passenger load per step.

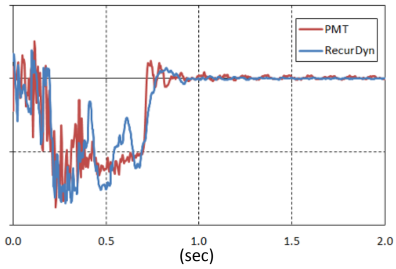
3.8 Main drive chain band

The gearbox output shaft and main drive chain sprockets are connected each other by a duplex roller chain band to transmit driving force from the drive machine or braking force of the operational brake system into the escalator system. The rigid chain links are connected to each other with force elements that rotate freely around their x-axis as shown in Fig. 4. The force elements have the same stiffness as the chain links. The chain roller in each inner chain link has steel-to-steel and surface-to-surface contacts with both sprockets.

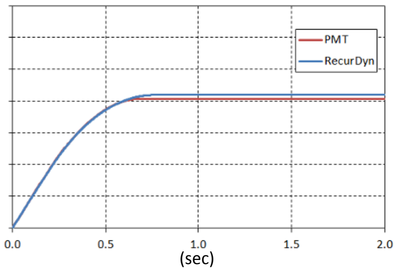
4. Verification of simulation model

The simulation model is verified with PMT [12] measure-

Fig. 5. PMT measurement of an actual escalator.



(a) Deceleration

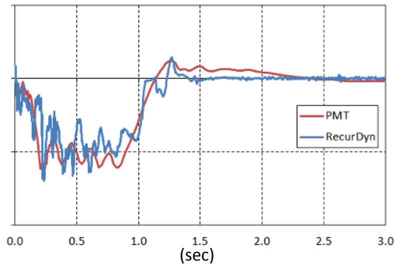


(b) Stopping distance

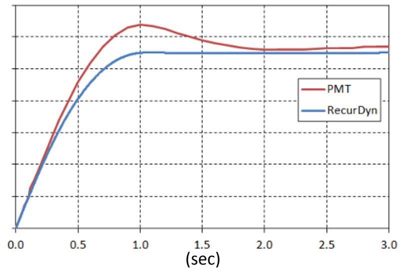
Fig. 6. Comparison for operational brake activation.

ment data taken from an actual escalator with a 6 m rise and 30° incline angle. Fig. 5 shows the deceleration and stopping distance, which are measured on a step in the travel direction when the brake systems are activated. The model is a down running escalator at 0.5 m/sec speed under empty load condition. Measurements are conducted when the operational brake system and the auxiliary brake system are activated separately.

Comparison between the simulation results and measurement data is shown in Fig. 6 for operational brake activation and in Fig. 7 for auxiliary brake activation. Both figures highlight a certain level of variable deceleration during the period of brake time until the escalator comes to a standstill. The deceleration measured from the actual escalator has less fluctuation than the simulation results because of the damping effect of the step chain rollers, which was not considered in the simulation model. The dynamic behaviors characterized by the maximum deceleration peak value can be estimated close to the actual escalator system within a 5.5% error range. Therefore, the multibody dynamics simulation model developed in this article can efficiently represent the actual escalator system in further investigations on the braking performance.



(a) Deceleration



(b) Stopping distance

Fig. 7. Comparison for auxiliary brake activation.

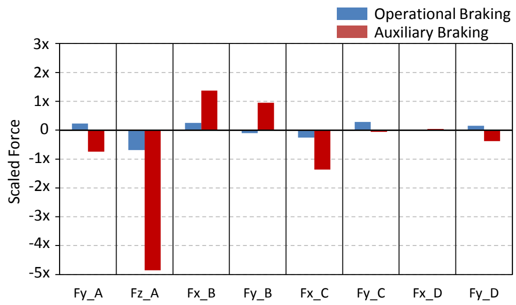


Fig. 8. Reaction forces on the machine feet under empty load.

5. Results of braking performance analysis

From a design standpoint, drive machine feet are critical parts for activated escalator brake systems. These parts should provide enough strength to support the huge impact forces during the short stopping events. Therefore, the reaction forces on the four machine feet are considered as major braking performance indicators for an escalator system. All braking performance studies are executed on the verified simulation model instead of an actual escalator system.

Reaction forces on each machine foot location are presented in Fig. 8 for operational braking and auxiliary braking events under empty passenger load condition. The highest reaction force is observed under the auxiliary brake activation and is concentrated mainly on the longitudinal direction of the machine foot A compared to the other machine foot locations. Fig. 9 shows the machine foot reaction forces for both braking events under full load passenger condition. The brake test load specified in the EN115-1:2008 [2] is considered as full passenger load. The highest reaction forces are also observed on the longitudinal direction of the machine foot A and vertical

Table 1. System parameters for full factorial design.

Parameters	Level 1	Level 2	Level 3
Type of braking	Operational	Auxiliary	Both
M.D.C. tension	Low	High	-
Passenger loading	Empty	Full	-

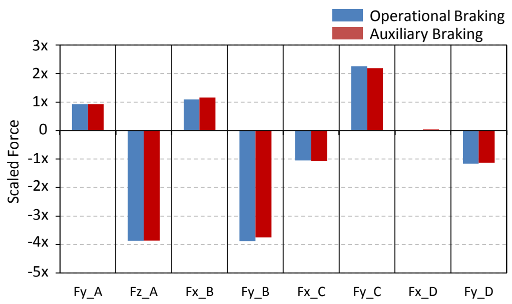


Fig. 9. Reaction forces on the machine feet under full load.

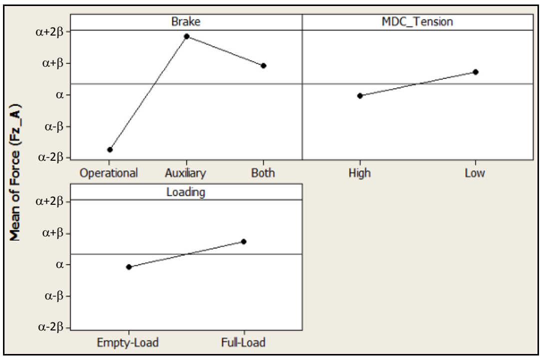
direction of the machine foot B during both braking events. Interestingly however, these values are smaller than the longitudinal reaction force on machine foot A under an empty passenger load condition. In general, the die-casting members of four machine feet are sufficiently strong to support the impact forces in the vertical direction. Therefore, carrying out further braking performance studies on the longitudinal reaction force component of the machine foot A is reasonable.

The design principles indicate that the operational and auxiliary brake systems have different dynamic influences on the escalator system when activated. The main drive chain tension and passenger loading condition might also have different levels of impact on the braking performance of an escalator system. These three system parameters usually change from unit to unit depending on the operating conditions in a specific escalator service area. In other words, a certain combination of the three system parameters will occur during an ordinary operation of the unit. Therefore, the type of braking event, main drive chain tension, and passenger loading conditions are considered as main parameters for the study of braking performance in this article.

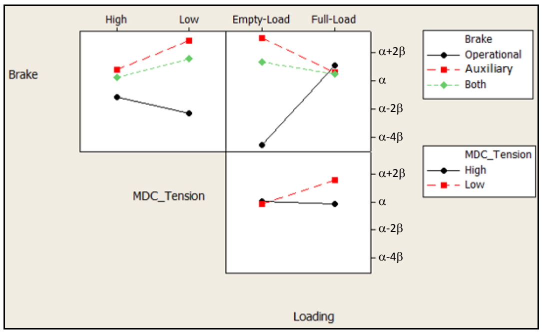
The influence of the three defined system parameters on the braking performance of an escalator system is investigated and the simulation results are interpreted in a systematic manner by using the statistical data analysis approach through the MINITAB software [13]. By considering the full factorial design, design of experiment is performed on the longitudinal reaction force of the machine foot A with one three-level factor and two two-level factors as shown in Table 1.

The plots of main and interaction effects with respect to the three system parameters are presented in Fig. 10 as the results of the design of experiment analysis. Results show that the single activation of the auxiliary brake system under empty load condition has the most dominant influence on the longitudinal reaction force of machine foot A. When the auxiliary brake system is activated, the moment of inertia of the flywheel attached on the fast rotating motor shaft creates a huge impact on machine foot A during a short period of brake time. The impact is transferred from the flywheel to the main drive assembly through the lower strand of the main drive chain. However, passenger loading conditions and the activation of the operational brake system reduce this effect.

The influence of the main drive chain tension and passenger loading conditions on the braking performance differs to a certain extent. The longitudinal force on machine foot A increases in the case of a loose main drive chain tension under an auxiliary braking event but decreases under an operational braking event. The longitudinal force on machine foot A decreases in the case of full passenger loading condition under an auxiliary braking event, but increases under an operational braking event in that case. The longitudinal force on machine foot A does not change considerably in different passenger loading conditions under high main drive chain tension. However, the force increases in the case of full passenger loading condition under a loose main drive chain tension.



(a) Main effects plot



(b) Interaction effects plot

Fig. 10. Analysis results of design-of-experiment study.

6. Conclusions

In this article, a comprehensive and efficient multibody dynamics simulation modeling approach is introduced to develop a complete system level escalator simulation model for braking performance analysis. The approach considers a drive machine with gear train, main drive chain band, operational brake system, and auxiliary brake system, which have never been considered in previous simulation studies. The simulation model is validated with actual measurement data within a 5.5% error range, considering maximum deceleration peak value. The study of the braking performance is undertaken with respect to three defined system parameters to reach the following conclusions using the verified simulation model and statistical data analysis approach.

- The moment of inertia of the flywheel is originally intended to ensure passenger safety by keeping the maximum deceleration and stopping distance within a certain range. However, the flywheel also increases the internal impact force in the escalator system when the brake systems are activated. Therefore, an auxiliary braking under an empty loading condition should be taken as the worst case scenario considering the braking performance of an escalator system.
- The main drive chain tension has a more considerable effect on the escalator braking performance under an auxiliary braking event than an operational braking event. In particular, loose main drive chain tension has a negative influence on the system. The main drive chain tension changes during an operation depending on the lubrication, alignment, and other operating conditions. Therefore, maintaining the proper level of main drive chain tension is important in preventing the additional dynamic impact caused by the loose chain band.

References

[1] CEN, *EN115:1995+A1:1998: Safety rules for the construction and installation of escalators and passenger conveyors*, European Norm (1998).

[2] CEN, *EN115-1:2008: Safety of escalators and moving walks –Part 1: Construction and Installation*, European Norm (2008).

[3] ASME, *ASME A17.1-2004: Safety code for elevators and escalators*, An American National Standard (2004).

[4] L. Al-Sharif, Experimental investigation into the relationship between passenger comfort on a stopping escalator and the kinematics of the stop, *Lift Report*, 6 (2007).

[5] C. J. Park and T. W. Park, Computational modeling and vibration reduction of an escalator system using robust design, *Computer Modeling and Simulation in Engineering*, 4 (4) (1999) 255-259.

[6] C. J. Park, Dynamic characteristics analysis of an escalator using the computational dynamics program, *ASME Design Engineering Technique Conference*, Atlanta, USA (1998) MECH-5813.

[7] C. J. Park, Improvement of handrail slippage characteristics using multi-body dynamic analysis technique, *KSME A*, 28 (9) (2004) 1422-1428.

[8] P. E. Nikravesh, *Computer-aided analysis of mechanical systems*, Prentice-Hall, New Jersey, USA (1988).

[9] FunctionBay Inc., *RecurDyn V8R1 Tutorial*, Republic of Korea (2012).

[10] J. L. Meriam and L. G. Kraige, *Engineering mechanics volume 1 statics*, Second Ed., Wiley & Sons, New Jersey, USA (1987).

[11] D. C. Threlfall, The inclusion of Coulomb friction in mechanisms programs with particular reference to DRAM, *Mechanism and Machine Theory*, 13 (1978) 475-483.

[12] Physical Measurement Technologies Inc., *EVA-625 Manual*, New Hampshire, USA (2009).

[13] Minitab Inc., *MINITAB R14 Tutorial*, Pennsylvania, USA (2005).

Chan Jong Park obtained a Ph.D. in Multi-Flexible Body System Dynamics from Ajou University in Republic of Korea. He is currently working for Escalator System and Innovation Centre as a project manager for technology in OTIS Vienna, Austria. His interests include dynamics, vibration and optimization for heavy duty machinery, transportation and vehicle systems.

Gero Gschwendtner has nearly 20 years of experience in the elevator and escalator industry. He has been working as director of escalator engineering and codes for OTIS and responsible for escalator engineering centers in Austria, Czech Republic and China since 2005. Moreover, he is globally in charge of codes and standards for elevator and escalator since 2013. He is a representative at several CEN and ISO Code Committees and is in charge of the escalator safety standards within CEN.