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Research Article / Araştırma Makalesi

THE DYNAMICAL BEHAVIOUR OF THE DOUBLE - SHOES BRAKE USED IN ELEVATORS

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ABSTRACT

The double-shoe braking system used in elevators of buildings with the purpose of mechanical braking behavior are used an experimental method for the preparation of the dynamic model in this study. The double-shoe braking system is used in many mechanisms. The double –shoe braking system is a complex structure consisting of many parts, so that it is difficult to create a model with a theoretical approach to make the right contact definitions. Therefore, experiments were performed by creating experimental model in this study and the results of these experiments have shown that the structural noise and vibrations were significantly affected by the speed, and braking and take-off acceleration of the engine.
Keywords: Elevators, elevator safety systems**,** double-shoe brakes, vertical transportation systems.

ASANSÖRLERDE KULLANILAN ÇİFT PABUÇLU FRENLERİN TASARIMI

ÖZ

Bu çalışmada, binalarda kullanılan asansörlerde mekanik frenleme amacıyla kullanılmakta olan çift pabuçlu fren sistemlerinin dinamik modellerinin hazırlanması için deneysel bir yöntem kullanılmıştır. Çift pabuçlu fren sistemi bir çok sistemde kullanılmaktadır. Çift pabuçlu fren sistemi bir çok parçadan oluşan kompleks bir yapıdadır. Bu durumda doğru kontak tanımları yapıp teorik yaklaşımlarla model oluşturmak oldukça güçtür. Bu nedenle, bu çalışmada deneysel model oluşturularak, deneyler yapılmış ve deney sonularına göre yapısal gürültü ve titreşimlerin, tahrik motoru hızından, duruş ve kalkış ivmelerinden oldukça etkilendiği gözlenmiştir.
Anahtar Sözcükler: Asansörler, asansör güvenlik sistemleri, çift pabuçlu frenler, düşey transport sistemleri.

1. INTRODUCTION

Elevators are used to transport passenger one floor to another and load in vertical directions. Hence, they are expected to offer a comfortable and safe service to their passengers in the buildings. Brakes are used as safety systems in elevators. There are more than one safety systems in elevators which one of them is double –shoe brake system. The brakes used in elevators have two basic tasks. Firstly, they are responsible for holding the elevator stable when it is not in

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operation, and secondly for stopping the cab in the desired floor as commanded by the user. According to the standards, the electromechanical brakes that intervene should be used in elevators' drive mechanism. For this aim, when the electromechanical brakes are not using any reasons, double-shoe brake system is capable of stooping the car.

In elevators, the brakes are active when the system is not in operation, which is not the case in other systems. The brake opens and starts its movement once the drive engine actuates. In case of a breakdown or power disruption, the brake can be activated manually by operating the lever on the braking system and automatically deactivated when this lever is released.

The literature has few studies regarding the double- shoe braking systems used in elevators. These systems are often considered as same with the double-shoe braking systems used in vehicles. Although the overall structure is same, the brakes in vehicles operate while the vehicle is in motion. The brakes on the elevators, however, operate when the system stops.

To summarize some of the studies in the literature on braking systems: Hirasata et al. have measured the frictional force, wear rate and changes in temperature in experimentally and given a relationship between these properties [1]. Xiao et al. have examined the tribological properties of the C/C-SiC composite material by using a disc-disc type test mechanism. It has been shown that this composite had perfect frictional properties [2]. Zhuan et al. were used the composites braked [3]. Their study investigated the tribological characters mated with self and steel counterparts, respectively for the friction counterparts at low brake speed [4].

Fan et al. have experimentally examined the frictional characteristics of the sandwich structured C/SiC composite brake shoes used as the braking shod material in planes. These properties have been determined by using a disc-disc type test mechanism [5]. Their other study indicated that the wearing mechanism of this material was in grain abrasion form [6].

Yokoi et al have shown experimentally that the noise levels have strong correlation with both surface roughness and sliding speed [7,8,9,10,11]. Lindberg et al. have conducted experimental studies on the noise and vibration in a vehicle's interior caused by the braking system in the vehicle and shown that the vibrations were fast and weak according to the braking pressure and speed [12].

Yurtseven et al. have examined the tribological properties of the brake pads [13]. Bettge and Starceviz have examined the topographical properties of contact areas on wearing surface of disc brakes and the results of the study have shown a wide contact inequality at high braking forces [14].

Ostermeyer has examined the friction and wearing in braking systems in this study [15]. Severin and Dörsch have examined the impacts of the friction materials and friction layer on braking in a study that they conducted on the friction materials in brakes and it has been shown that the friction coefficient decreased as the ratio of iron decreased [16]. Hohmann et al. have performed contact analysis in drum brakes and disc brakes [17]. Ripin has worked on contact pressure and shown that the maximum amount of wearing occurred on the leading area of the pad [18]. Tirovic and Day have examined the effects of piece geometry, material properties and contact characteristics on pressure distribution [19].

Verma et al. focused on the tribological behavior of a commercial friction pad material dry sliding against a cast iron disc [20]. Camacho et al presented their study and description of the wear mechanism implied in disk and shoe pads [21]. Another study is made thermal performance of brake by Abdulwahab et al. [22]. All these studies mentioned about, are related to the materials of brake and tribological features.

The studies regarding the safety systems used in elevators are generally about the parachute systems and no studies directly on braking systems have been found in the literature. Çavdar et al. have analyzed the parachute braking mechanism used in elevators for speed control and made their proposals on design [23]. Bedir has experimentally examined the forces and strains that the cylinder type sudden braking safety systems are exposed during parachute brake [24]. Ersavas has conducted a modeling study and analysis on parachute systems used in elevators [25]. De

Jong has reviewed the calculation and design criteria of the sudden braking and sliding safety elements in elevators and presented their effects on guide rails in his study [26].

We see that the literatures contains a very few studies conducted on double-shoe braking systems used in elevators. For the purpose of closing this gap, the dynamic behaviors of the double-shoe brakes that have a very complex structure has been examined and experimental modeling method has been used in this study.

The outline of the paper is as follows. Section 2 describes the experimental methods and experimental procedure. Section 3 gives data reduction. Section 4 presents results and discussion. Finally, it gives the conclusion in Section 5.

2. EXPERIMENTS

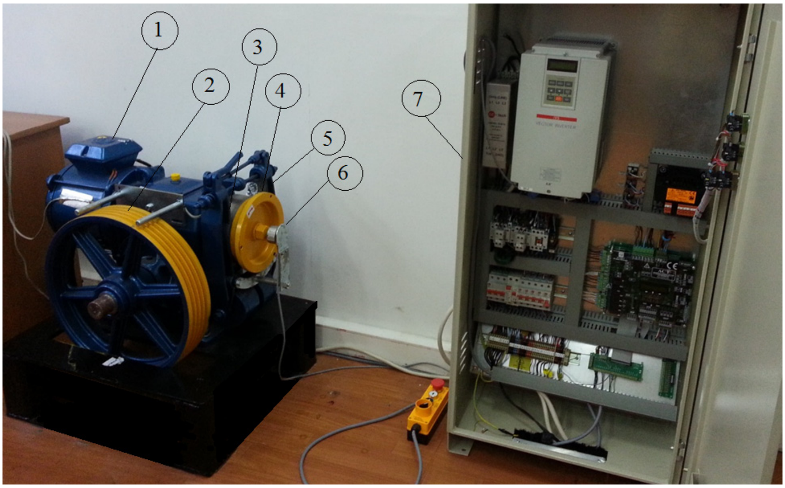
In this study, some experiments have been conducted in laboratory environment on the double-shoe brake systems of the elevator engines used in buildings. AKAR brand DAK135 type elevator engine has been used as elevator engine. The tests have been conducted on an elevator engine with 400 kg lifting capacity, 1 m/sec can speed, 480 mm roller diameter, 4 ropes with 10 mm diameter each and 4 kW engine output. During the set-up of test environment in the laboratory, the engine was placed on an elevator table positioned on a vibration-insulated floor in order to protect the engine against outside environment.

An AC Series command system compliant with EN 81-1 and EN81-2 standards has been used in the elevator. This system is controlled by a 32-bit microprocessor. The system has all the features that an elevator should have and is flexible and modular to be directly connected to any computer. Fig. 1 shows the strain gauge and acceleration sensor fitted to the test setup.

(a) (b)

Figure 1. The position of sensors used in the test system.(a) Acceleration sensor, (b) Strain gauge.

The strain and acceleration data at different speeds have been recorded with a data logger at the test conducted on the system shown in Fig. 2.



(1) Elevator engine and gearbox, (2) Flywheel, (3) Accelerometer, (4) Straingauge, (5) Brakes pad, (6) Tachometer, (7) Elevator control system.

Figure 2. Elevator experimental setup

The experiments in this paper were performed under laboratory conditions by creating the system which is similar to real elevator used in buildings. The objective of this experiment is to investigate the effects of the resonances and structural vibrations while the elevator is in operation. The planned tests aimed the data is collected by strain gage and acceleration sensors which are located on system brakes arms. Firstly, electrical brake is deactivated. The equivalences of effects of load and acceleration of the system are provided by flywheels which are located to the motor shaft. For the performed measurements, sensor installation and hardware setup mounted on the brake levers carefully. The features of strain gauge used in test KYOWA type KFG-5-120-C1-11N2C2, and gage factor 2.10±1.0%, gage length 5mm which is shown in Fig.1-b. The surface where stick the strain gauge, is cleaned from rust and dirt by using Isopropyl alcohol and abraded by using different size of sandpaper. For bonding operation CC-33A cyanoacrylate based adhesive is used. The T-F13 self-bonded gauge terminals are used on the each side of the gage for wiring. Three wire connections used on each strain gage to cancel the wire resistance effect. The wires soldered to D-Sub 9 pin malev connectors to make the connection to the data acquisition system. For the measurement of acceleration, an accelerator is used in Fig.1-a. Acceleration measured by DJB Instruments A/120/EPE accelerometer which has 98,51mV/g sensitivity. To ensure that there is not a temperature, a K-type thermocouple is bonded close to installed gauge area and finally to ensure about the changes on the strain gauge and accelerometer data whether they are caused by position change, stress or any outside force during the measurement a simple direct camera is used. For data acquisition 8 channel 200 kS/s DEWE 43 hardware is used with 120Ω,2,5 V quarter strain gauge bridge completion adapters. Below a screenshot from the measured area is given Fig.4. After preparing these conditions meticulously, the experiment is done with different motor speed at 600d/d, 800d/d and 1400d/d. The measurements were made by author precisely, all data were recorded simultaneously. Due to the complexity of the experiment, the measurement was made at different times and the optimum time for the test was set as 60 seconds in order to adequately observe the resonance cases. The test has been completed within the same week as a measure against the possible disruptive effects of environmental conditions such as temperature and humidity. By changing the frequency of engine over driving system which functions by frequency control on elevator control unit, first

600 d/d is arranged and the tests are done. The system, which was send the call floor, operated for 60 seconds and this process was repeated fussily for800d/d, 1000d/d and 1400d/d.

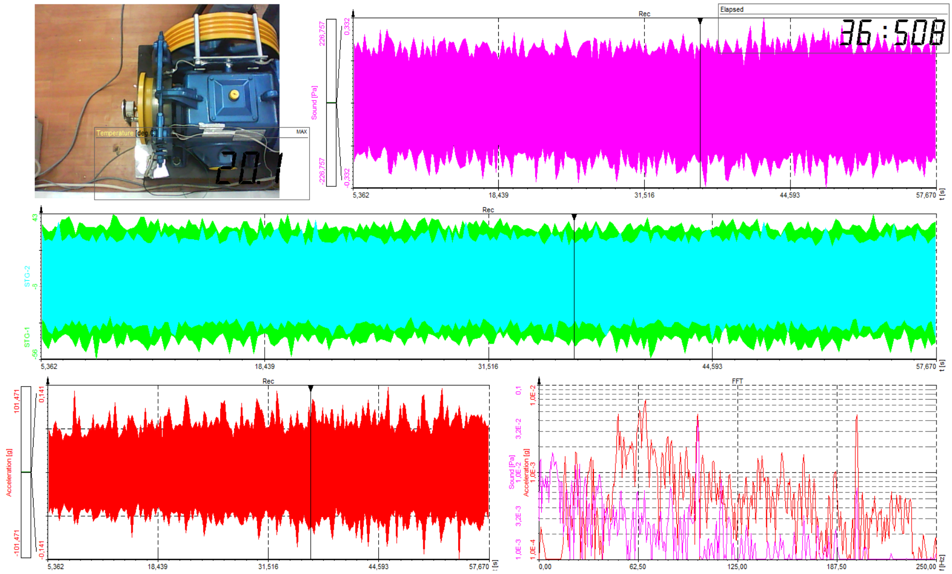


Figure 3. Screen capture from the measurement

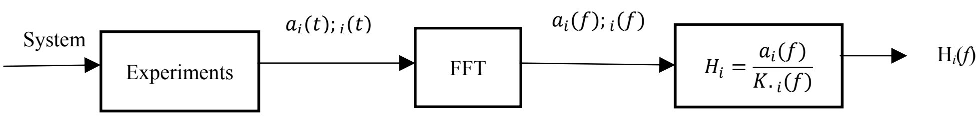
3. DATA REDUCTION

According to the measurements made on the brake system, the acceleration values of the system's structural vibrations have been obtained in the time section as against the braking force input. These data have been indicated in the frequency area with the Fast Fourier Transformation (FFT). It is shown that the measurement of strain gauge 1 and strain gauge 2 and acceleration and FFT for different speeds of engine in Fig.5-12. The most important resonance frequency area which shows in the figures is 100 Hz. The most important parameters in the model are breaking force, rigidity value of the structure and the damping ratio. The change in braking force has been measured with strain gauge. The rigidity value *(k)* and damping ratio *(c)* have been obtained by the following processes.

The transfer function of a system with 2nd order may be expressed as follow according to the equation 3.1.

$$H = \frac{a}{F} = \frac{a}{K.\varepsilon_{xx}} \tag{3.1}$$

The equation that expresses the dynamic behaviors of the braking system in the most correct way (H_f) is obtained by taking the average of the previously obtained experimental transfer functions.



ε_i = *The measurement of strain* a_i = *The acceleration measurements*
K =Coefficient factor between strain and force H_i = *Transfer functions*

Figure 4. The flow chart to create dynamic behavior equation experimentally

The unknown values of the second order equation as well as the c and k values are obtained by using the previously obtained experimental transfer function and equation 3.1 via an optimization process (Fig.4).

Here ; the coefficient factor (*K*) between strain and force are found in the below.

The straingauge was settled on the brake arm in the model as following.

At the point placed on brake arm, the strain value measured by strain gauge section surface,

$$\varepsilon_{xx} = -x_s \frac{d^2 v}{dz^2} \tag{3.2}$$

Here;

x_s: The distance to the weight axis of linking point of strain gauge bending strain of the cross section at measured point;

$$\sigma = E.\varepsilon_{xx} = \frac{M_y x_s}{I_{yy}} \tag{3.3}$$

Bending moment,

$$M_y = F.L_z \tag{3.4}$$

L_z is the constant value which expresses the distance of momentum.

In these equations, the relationship of strength and strain is written as following;

$$F = \frac{E.I_{yy}}{L_z.x_s} \varepsilon_{xx} \tag{3.5}$$

Strength x displacement and beam stiffness according to force can be written as following;

$$F = K.\varepsilon_{xx} \tag{3.6}$$

In this case we can write K as following.

$$K = \frac{EI_{yy}}{L_z.x_s} \tag{3.7}$$

As a result of process; the transfer functions which give the vibration amplitude are obtain to response to the braking force. It is expressed the curve which is obtained from the result of test continues with black line and the curve which is obtained from transfer function with cutting red line in this figure. It is understood that the test results are quite close when the failure of model which is gained by the result of optimization.

4. RESULT AND DISCUSSION

The brake systems of the elevators have a complex structure consisting of several parts. Therefore, it will be more appropriate to find the dynamic behavior equation of such a structure with an experimental approach. The structure contains several material groups such as metals and composites. This makes it difficult to make correct contact definitions and create a model with mathematical methods. Therefore, the system model has been created with experimental data in this study. For this purpose, the elevator system has been operated at different speeds and amplitude output functions have been produced according to the different speed inputs.

The experimental dynamic behavior function that contains the vibration amplitude outputs according to the speed input has been produced in the frequency area by taking the average of the test data obtained at different speeds (Fig. 5-12).

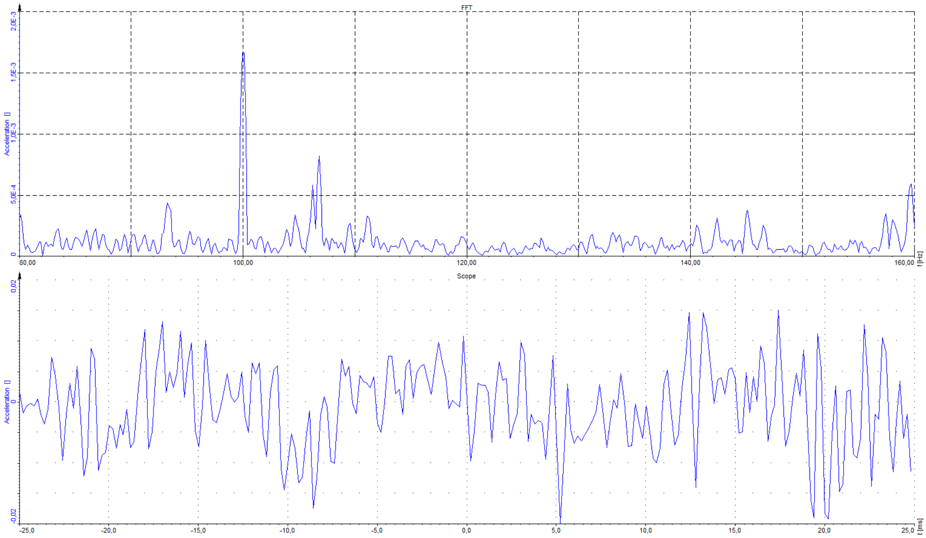


Figure 5. FFT and Scope (0.5sn), 100 Hz, 0.02g and acceleration for 600 d/d

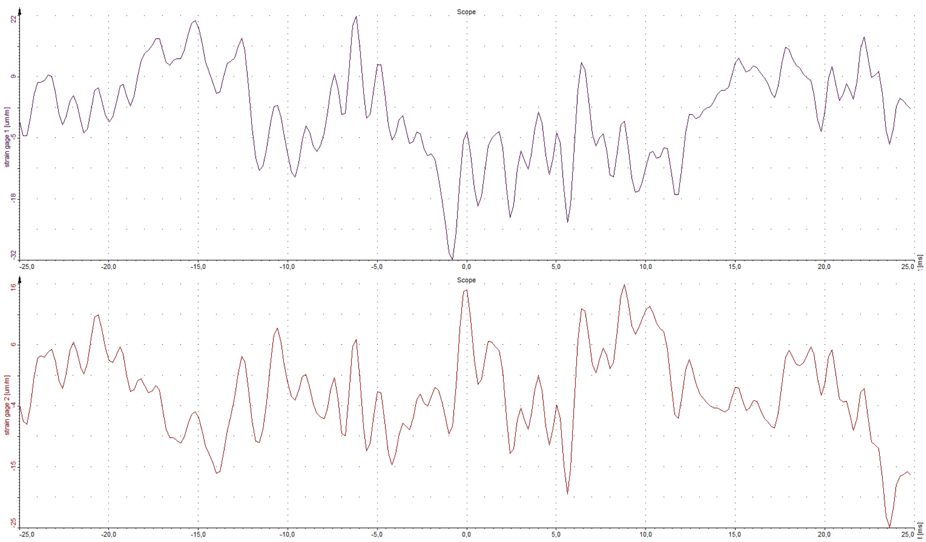


Figure 6. Strain gauge 1 - 32 microstrain, Strain gauge 2 - 25 microstrain for 600 d/d, Scope (0.5sn)

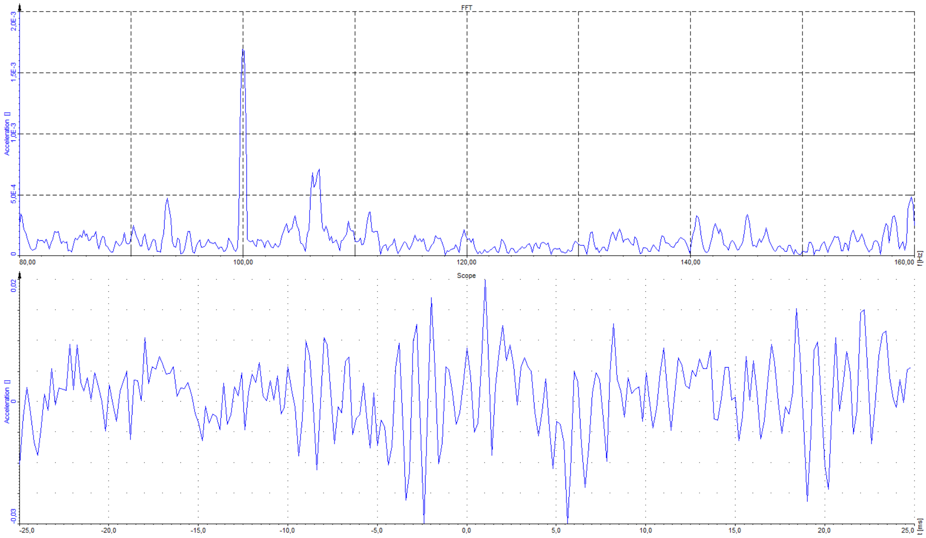


Figure 7. FFT and Scope (0.5sn), 100 Hz, 0.03g and acceleration for 800 d/d

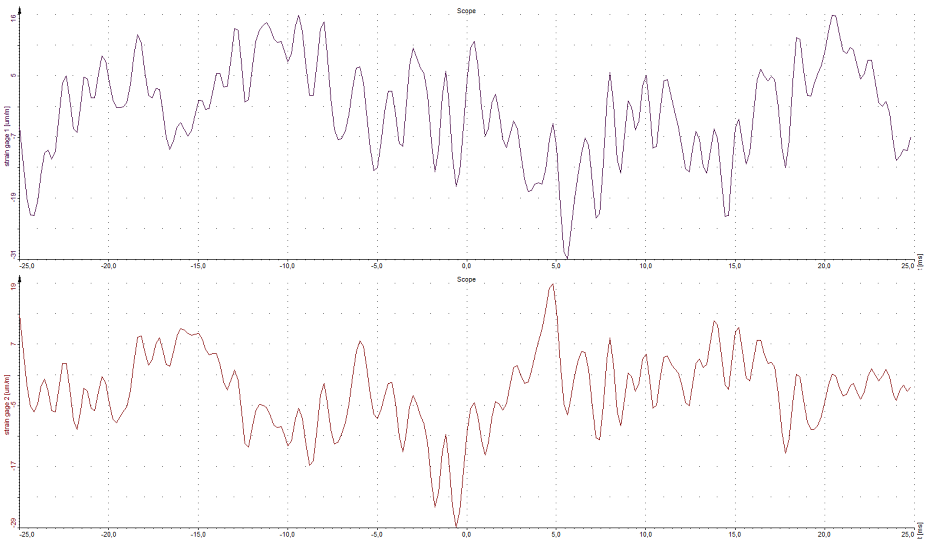


Figure 8. Strain gauge 1 - 31 microstrain, Strain gauge 2 - 29 microstrain for 800 d/d, Scope (0.5sn)

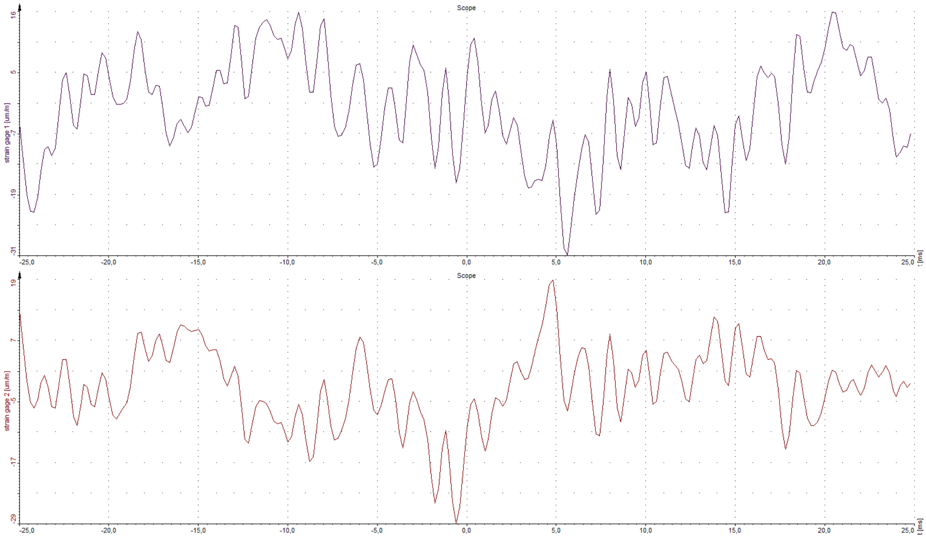


Figure 9. FFT and Scope (0.5sn), 100 Hz, 0.01g and acceleration for 1000 d/d

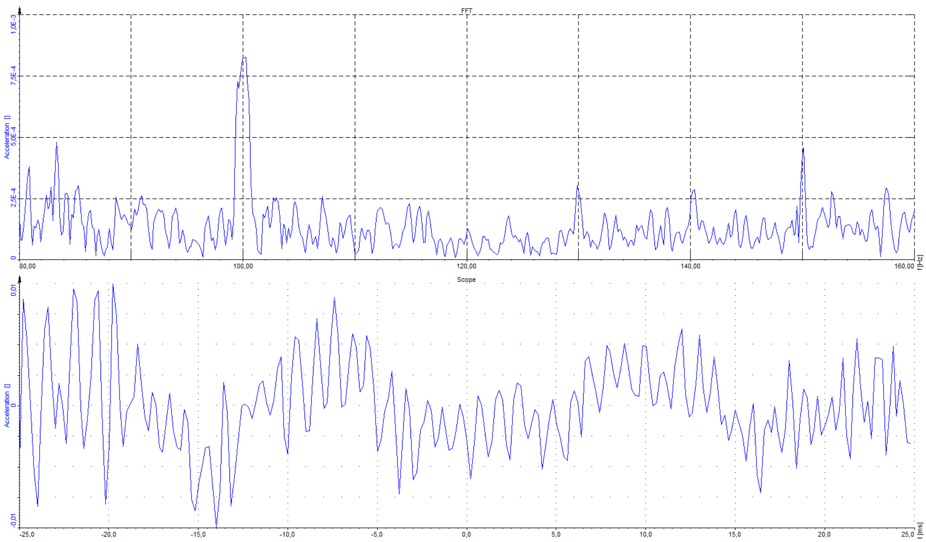


Figure 10. Strain gauge1 -34 microstrain, Strain gauge 2 - 30 microstrain for 1000 d/d, Scope (0.5sn)

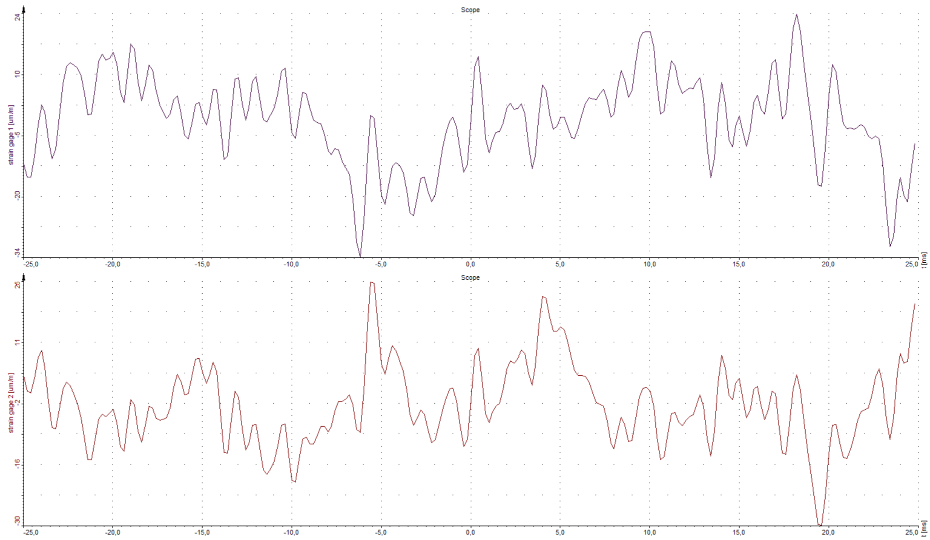


Figure 11. FFT and Scope (0.5sn), 100 Hz, 0.01g and acceleration for 1400 d/d

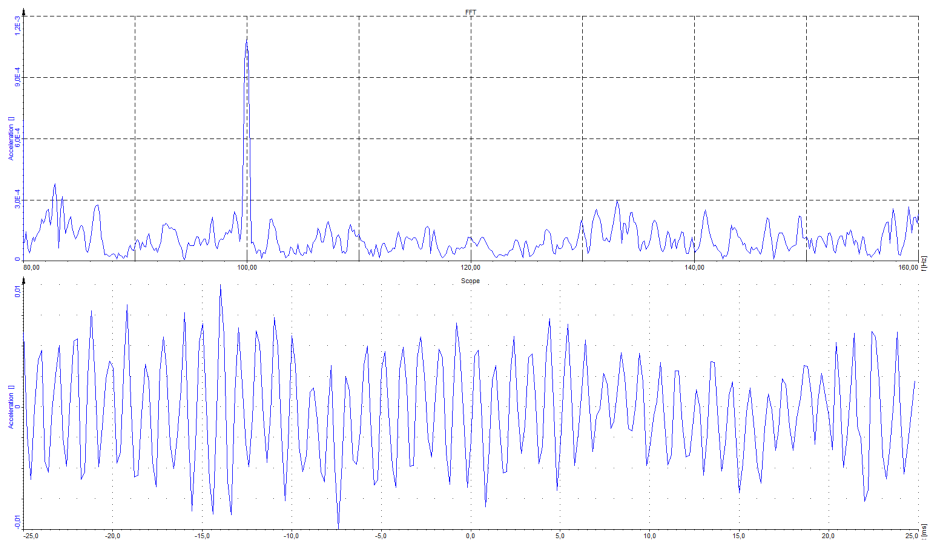


Figure 12. Strain gauge1-27 microstrain, Strain gauge 2 - 24 microstrain for 1400 d/d, Scope (0.5sn)

The most significant resonance frequencies are at around 100 Hz and 200 Hz frequencies. In fact, the speed of engine which is used in experiment is between 10-24 Hz. The reasons of this are the complex of materials contact with the different parts. In these contacts, the other conditions like the damping ratios, surface of stiffness, the pressure of brake force, the surface roughness of brake shoes and drum, heat, humidity and etc… cause high frequency resonances.

The parameters that determine the dynamic behaviors of the system can be defined for each resonance frequency by means of this method. We understand, when the system's dynamic behavior is examined, that the resonance frequencies should be considered when designing a brake in order to reduce noise and vibrations. Accordingly, the changes in several parameters in the design are observed by using experimental dynamic behavior model and the position of the resonance frequencies can be optimized.

4. CONCLUSION

There are insufficient data for the brake design in elevator industry. So that, the results obtained from the test is important for the design and manufacture of double-shoes brake. We can summarize main factors to be considered in brake design as follows:

(1) It's a fact that the noise and vibrations caused by the operation of the double-shoe brake systems, one of the safety equipment in elevators, are two of the most significant issues for both the elevator and the building. Increasing noise and vibration will damage the building, elevator system's structure as well as the people. The brakes should be designed by considering the frequencies in resonance areas in order to reduce such damaging effects.

(2) It is important to conduct modeling and simulation studies to find out the possible dynamic effects at conceptual design stage. However, conducting studies with experimental models on these types of structures will reduce the risk of possible errors.

(3) The most important two parameters in a brake structure are the average rigidity value and damping value. The rigidity value is related to the masses in the brake structure and can be increased by choosing appropriate structural elements. The damping values, on the other hand, vary significantly depending on the pad material used. Therefore, choosing the correct pad material during the designing stage will reduce noise and vibrations.

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