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Optimization of the industrial guide-way vehicle to improve the running stability †

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Abstract

This paper presents an optimization of the industrial guide-way vehicle that aims to improve running stability at increased speeds. A guide-way vehicle was used to transfer products in various manufacturing industries. Using Design Of Experiment(D.O.E.), the design prototype was optimized. The improved design prototype and its design parameters were obtained by a case study determined by the engineering discussion. The computational model for the optimization was validated by correlation with the test results. Through this procedure, the optimization method presented in this paper has been proven to be effective.

Keywords: Optimization; Design of Experiment (D.O.E.); Case study; Multibody dynamics

1. Introduction

A guide-way vehicle is used to transfer products in automobile, semiconductor and LCD manufacturing industries. To improve the productivity, the transportation capacity for the guide-way vehicle is continually increasing. To increase the transportation capacity, the vehicle speed can be raised. Although this is an easy and efficient solution, the running stability of the vehicle can be affected negatively by the increased operating speed.

This paper presents an optimization of the industrial guide-way vehicle that aims to improve running stability at increased operating speeds.

Optimization is achieved by the Design Of Experiment(D.O.E.) method. [1] First, a reliable computational model should be obtained by correlation with test results. A reliable simulation model can replace the real system to shorten the design time and improve the optimization efficiency. Using the obtained computation model, case studies were carried out with respect to seven design parameters, determined by engineering discussion. Through the case studies, an improved design prototype and three design parameters (change of the driving wheel base, change of the upper wheel base and change of the lower guide wheel base) that largely affect the running stability were selected.

Using D.O.E., the improved design prototype is optimized. Applying the optimized design parameters to the computer model, dynamic simulation was carried out under the current and target speed. Comparison of the simulation results show that the running stability was improved. Consequently, the optimization method presented in this paper was proven to be efficient.

2. Dynamic model and validation

2.1 Dynamic Model of a Guide-Way Vehicle

Fig. 1 shows the dynamic model of the guide-way vehicle to be optimized. The vehicle is composed of four bodies, namely the driving front part, driving

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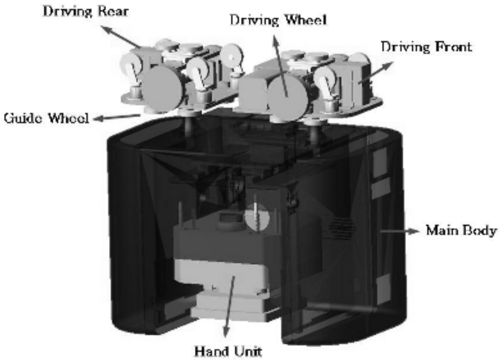

Fig. 1. Dynamic model of the guide-way vehicle.

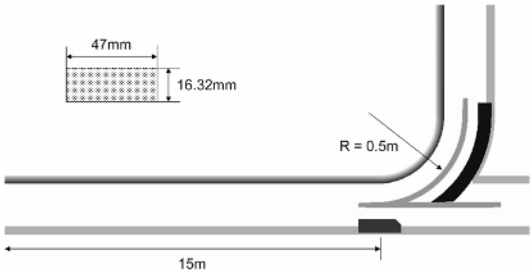

Fig. 2. Rail profile and trajectory.

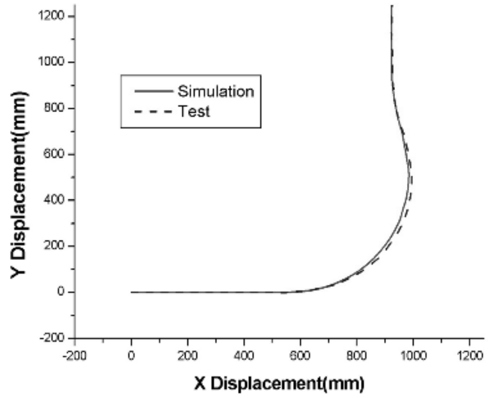

Fig. 3. Comparison of xy trajectories.

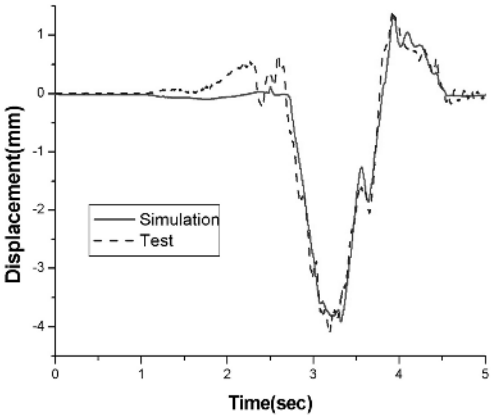

Fig. 4. Comparison of vertical displacement.

rear part, main body and hand unit. The main body contains the electronic unit and the hoist unit.

There are four driving wheels. Two are located at the front part and two at the driving rear parts. There are 18 guide wheels: two at each upper part, four that guide the inside rail at each lower part, six to prevent the vehicle being lifted at the cornering and the divergence sections.

Fig. 2 shows the shape of a section and the course of the rail, which are used in the driving simulation.

2.2 Validation of the dynamic model through a driving test on the cornering section

A reliable computational model is essential for optimization. Therefore, a driving test was done to validate the dynamic model.

The running stability of the vehicle under roll and lateral acceleration is affected more when the vehicle is running on a curved section than on a straight section. Therefore, the reliability of the dynamic model was verified through correlation between the test and simulation results for the cornering section.

In the test, a high precision and resolution 3-dimensional laser tracker was used to acquire the displacement of a specific point of the vehicle. Lateral acceleration by cornering imposes more displacement outside of the turning radius upon the vehicle than the designed rail geometry does, as shown by the dotted line in Fig. 3. The dotted line in Fig. 4 shows vertical displacement at the cornering section. Figs. 3 and 4 show that results of the test and simulation are comparable. Error at early entrance in the rising and dropping direction is thought to be caused by the setting rail in the field. Consequently, a reliable computational model was obtained through correlation with the results of the 3-dimensional displacement test.

3. A case study to increase the speed at the cornering section

A case study was carried out to develop an improved design prototype of the current guide-way vehicle and to identify the design parameters that affect the vehicle performance. Contents of the case study are given in Table 1. Fig. 5 shows the procedure for the investigation of the running stability. While a

Table 1. Contents of case study.

Case study No.	Contents
1	Driving wheel base change (-30mm, +30mm)
2	Driving wheel tread change (- 10mm)
3	Turning radius change
4	The number of upper wheels
5	Rail geometry change on the crossroad
6	Change of the lower guide wheel base
7	Double line arrangement of the upper wheels

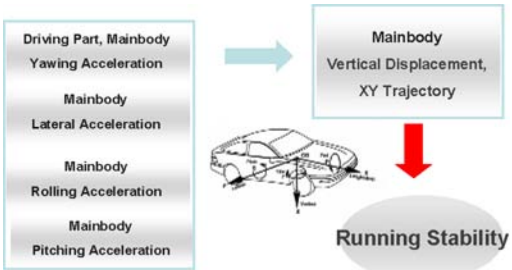

Fig. 5. Procedure for investigating the driving stability of the guide-way vehicle.

vehicle is running, displacements can be obtained by integrating the accelerations in each direction according to the vehicle coordinates. With the displacement obtained, the running stability can be estimated. The velocity of the vehicle is 0.5m/s at the cornering section.

4. Optimization

Optimization to guarantee the running stability of a guide-way vehicle at increased speeds is achieved by D.O.E. [1]. In this study, the base model for the optimization of a guide-way vehicle was a prototype with double line arrangement of the upper wheels, which was selected through the above case study. In the case study, three design parameters(change of the driving wheel base, change of the upper wheel base and change of the lower guide wheel base) were found to largely affect the performance. The objective function for optimization was the maximum deflection of the vehicle running on the cornering section, chosen as such because among all the running sections, the cornering section affected the running stability the most. Furethermore, the larger the deflection of the vehicle, the more unstable the vehicle becomes. Considering

Table 2. Experiment table.

No.	DP1	DP2	DP3	Response
1	-30	-1.5	-4	-573.87
2	-30	-1.5	12.5	-575.13
3	-30	7.5	-4	-573.2
4	-30	7.5	12.5	-574.73
5	30	-1.5	-4	-573.95
6	30	-1.5	12.5	-574.8
7	30	7.5	-4	-573.28
8	30	7.5	12.5	-574.78
9	0	3	4.25	-574.29
10	-36.48	3	4.25	-574.27
11	36.48	3	4.25	-574.32
12	0	-2.47	4.25	-574.76
13	0	8.47	4.25	-573.82
14	0	3	-5.78	-573.41
15	0	3	14.28	-574.81

Table 3. ANOVA Table.

Factor	S	φ	V	F0	F(0.01)
Regression Variation	5.2384	3	1.7461	210.1083	6.22
Residual Variation	0.0914	11	0.0083		
Sum	5.3298	14			

Table 4. Optimized parameters.

DP1	DP2	DP3
-15.72	7.29	-3.92

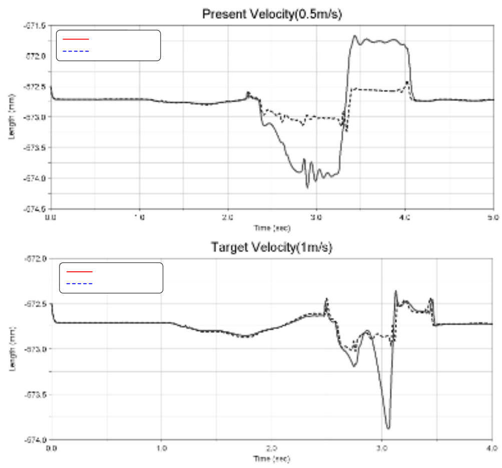

Fig. 6. Comparison of vertical displacement for present and optimized model.

the kinematic interferences of the vehicle, the ranges of the design parameters are determined by Eq. (1).

$$\begin{aligned} &-30 < DP1 < 30 \\ &-1.5 < DP2 < 7.5 \\ &-4 < DP3 < 12.5 \end{aligned} \tag{1}$$

Table 2 shows the experiment results obtained by use of the central composite orthogonal array [1] for the three design parameters.

Response values in Table 2 are the maximum deflections according to the change of the parameters of the dynamic model. Using these results, the response surface function of the second order recursive model can be estimated as Eq. (2).

$$\begin{aligned} \text{Vertical\_Displacement} = &- 574.2648 + 0.0054 \times DP1 \\ &+ 0.2649 \times DP2 - 0.6245 \times DP3 \\ &- 0.0261 \times DP12 \\ &- 0.0227 \times DP22 \\ &+ 0.0991 \times DP32 \\ &- 0.0475 \times DP1 \times DP2 \\ &+ 0.115 \times DP2 \times DP3 \\ &- 0.055 \times DP1 \times DP3 \end{aligned} \tag{2}$$

To check the reliability of the model function, ANOVA(ANalysis Of VAriance) table [2] is obtained as shown in Table 3. Table 3 shows the recursive model function has reliability at the one percent significance level since F0 is greater than F(0.01). Table 4 shows the results of each optimized parameter.

Using the optimized model, running simulation on the cornering section is carried out according to the current velocity(0.5m/s) and target velocity(1m/s). Fig. 6 shows the vertical deflections at each velocity. This result shows that the maximum deflection, which is the objective function, was improved.

5. Conclusions

The conclusions of this study are as follows.

(1) The simulation result for the cornering section was comparable to the test result obtained by the use of the laser tracker. Therefore, the reliability of the dynamic model was verified.

(2) To evaluate running stability for increased speeds, case studies were done with respect to seven design parameters, determined by engineering discussion. Through the case studies, an improved design prototype was obtained and three design parameters that most affected the running stability were determined. With the three parameters, optimization was processed.

(3) By applying the optimized parameters to the dynamic model, the running stability of the vehicle was improved.

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