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Investigation and design of a new shock absorbing device that cooperates between two colliding objects †

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Abstract

This paper deals with a new type of shock absorbing device that cooperates between two colliding objects. The new device utilizes a four-bar-chain-like articulated mechanism with some possible actuations. The devices are assumed to be deployed in the pre-crash phase (by sensing and identifying unavoidable collisions) so as to provide an extended deformable region between the two objects. Moreover, by functioning like a four-bar-chain mechanism, they produce a repulsive effect by pushing each other and sliding in the opposite lateral direction. To investigate the capacity of the proposed articulated shock absorbing mechanism, a standard numerical optimization technique called SQP and a new optimization technique called ALPSO are applied. ALPSO is an attractive method for solving multimodal optimization problems based on Particle Swarm Optimization and constraint treatment using an Augmented Lagrange Method. We demonstrate ALPSO and show its applicability to this problem. The optimization process automatically determines the mode of the operation and gives an estimation of the development potential of the new device.

Keywords: Active safety; Investigation and design; Multibody dynamics; Optimization

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1. Introduction

In this paper we propose a new shock absorbing device. The basic concept of the device is that an articulated bumper mechanism is deployed in the pre-crash phase to make an extended deformation zone. This device acts like a four-bar mechanism and produces a relative pushing and sliding motion between two colliding objects, see Fig. 1. The two main principles to alleviate the crash effects are the improved absorption of energy (using the extended deformable zone) and the reduction of this energy by avoiding a frontal collision. This avoidance is achieved by the inclined bumpers pushing the opposing object away.

The investigation of the design and its functionality is based on a simple rigid body model with penalty contact. It allows a straightforward analysis of the influence of the main parameters such as bumper spring stiffnesses and damping coefficients. These parameters determine the mode of operation, i.e., the interaction of the mechanical effects of energy absorption and pushing.

To investigate the capacity of the articulated bumper mechanism, the optimal parameter values are identified by numerical optimization. The optimization process automatically determines the mode of operation and gives an estimation of the development potential of the new device. A crash analysis is computed with RecurDyn, and optimization is conducted with MATLAB.

2. Modeling

Fig. 2 shows the top view of the 3D model of an

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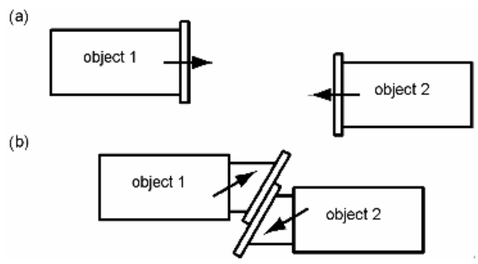

Fig. 1. Articulated bumper deployed in the pre-crash phase.

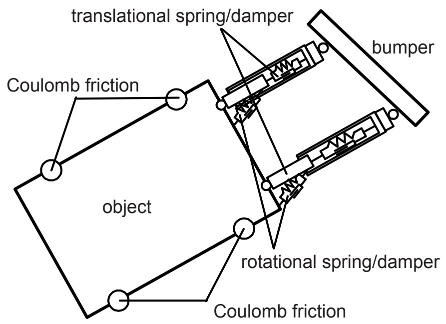

Fig. 2. Simple rigid body model of object and dumper.

object with a shock absorbing device. The object and the bumper are modeled by simple rigid bodies [1, 2]. The struts are assumed to be massless, and thus not considered as rigid bodies. Rigid body contacts (collisions) are considered between two (convex) bumpers. Contacts between two objects, or between object and bumper, are also considered. The analysis models are implemented in RecurDyn.

2.1 Contact model

The contact forces between the two colliding bumpers are computed based on a simple penalty approach that utilizes a virtual spring and damper between a vertex P of the penetrating body and the surface point C of the penetrated body, see Fig. 3. The contact force is given by

$$\mathbf{F_C} = \left(k_C D + d_C \dot{D}\right) \mathbf{e_N}, \tag{1}$$

where the spring stiffness k_C and the damping coefficient d_C are multiplied by the penetration depth D or its time derivate $\dot{D}$, respectively. The vector $\mathbf{e_N}$, which is normal to the penetrated surface, determines the direction of the contact normal force.

2.2 Strut model

The strut forces acting between the object and the bumper can be analytically derived using Newton's and Euler's law and by neglecting any mass of the struts, see Fig. 4. The applied torque at P is given by the rotational spring-damper element and the axial strut forces are determined by the translational spring and damper.

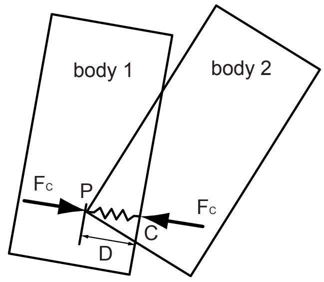

Fig. 3. Penalty contact model.

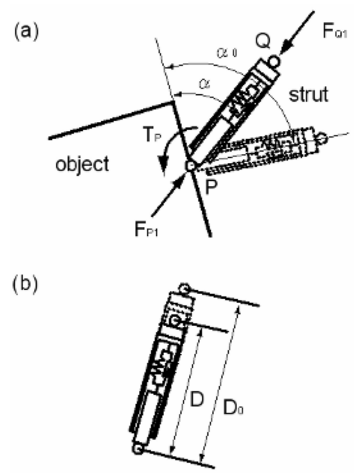

Fig. 4. Strut force and torque between bumper and object.

Thus, the axial forces can be written as

$$\mathbf{F_{P1}} = \mathbf{F_{Q1}} = \left(k_T\left(D - D_0\right) + d_T \dot{D}\right) \mathbf{e_{PQ}} \tag{2}$$

with the distance $D = \overline{PQ}$ and the unity direction vector $\mathbf{e_{PQ}}$. The applied torque is given by

$$\mathbf{T_P} = \left(k_R\left(\alpha - \alpha_0\right) + d_R \dot{\alpha}\right) \mathbf{e_{PQ-90^\circ}}. \tag{3}$$

The unstretched (natural) length D_0 and angle α_0, as well as the stiffness parameters and damping coefficients, are design dependent.

2.3 Friction model

The friction forces between object and ground are

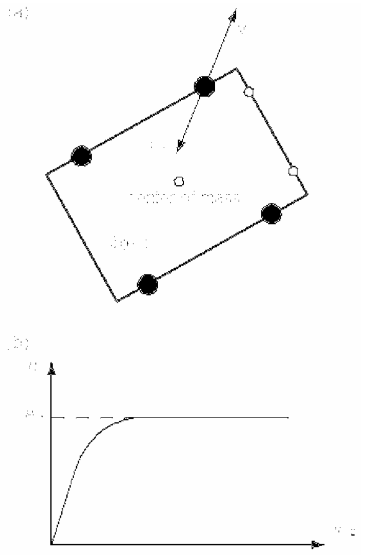

Fig. 5. Simple friction model.

calculated by Coulomb friction elements. It is assumed that each applied point of the friction forces is blocked, and thus leads to a simple sliding element that generates the Coulomb force

$$\mathbf{F_T} = -\mu(\mathbf{v}) f_N \mathbf{e_v} \tag{4}$$

with the friction coefficient μ and the vector of relative velocity $\mathbf{e_v}$. Since the force normal to the contact plane f_N cannot be computed based on the statically (kinetically) undetermined rigid body system, it is assumed that the gravitational force of the object is evenly distributed across the applied point of the friction forces,

$$f_N = \frac{1}{4} g m_{object}. \tag{5}$$

The transition from stick friction to slip friction is modeled with a continuous and smooth tangent hyperbolic function as shown in Fig. 5.

3. Optimization

The goal of this work is to investigate the capability of the safety device and the corresponding functionality. Therefore, the main task is to determine the unknown design parameters such as bumper stiffness and damping coefficients. This is done by applying numerical optimization procedures that automatically identify the optimal layout. The design variables considered in this work are

Table 1. Fixed paramters.

Initial object velocity	50 km/h
Offset of objects	0.72 m (half object width)
Object mass m	1210 kg
Object inertia I	1760 kgm^2
Penalty stiffness $k_{penalty}$	10^8 N/m
Penalty damping $d_{penalty}$	10^4 Ns/m

$$p = \lfloor k_{0,T1} \quad k_{0,T2} \quad k_{0,R1} \quad k_{0,R2} \quad d_T \quad d_R \rfloor, \tag{6}$$

see Section 2. The indices 1 and 2 correspond to the two different struts (spring-damper elements) of an object, such that each translational (index T) and rotational (index R) is considered separately, and the damping coefficients are identical for both struts.

The initial bumper inclination angle together with the predetermined fixed maximum strut length

$$\max\{D_{01} \quad D_{02}\} = 1 \tag{7}$$

determines the unstretched lengths D_{01} and D_{02} of the two translational bumper springs, as well as the initial rotation angles α_{01} and α_{02} of the unstretched rotational springs, see Section 2.

For all optimization procedures presented in this work, the most important non-variable (constant) system parameters are shown in Table 1.

The initial parameter values are

$$\mathbf{p_0} = \lfloor 1.00\times10^2 \quad 1.00\times10^2 \quad 1.0\times10^{10} \quad 1.0\times10^{10} \quad 5.00\times10^1 \quad 5.0\times10^2 \rfloor. \tag{8}$$

The parameters bounds for all optimization procedures are

$$\underline{\mathbf{p}} = \lfloor 1.00\times10^1 \quad 1.00\times10^1 \quad 1.0\times10^9 \quad 1.0\times10^9 \quad 5.00\times10^0 \quad 5.0\times10^1 \rfloor \tag{9}$$

and

$$\overline{\mathbf{p}} = \lfloor 1.00\times10^3 \quad 1.00\times10^3 \quad 1.0\times10^{11} \quad 1.0\times10^{11} \quad 5.00\times10^2 \quad 5.00\times10^3 \rfloor. \tag{10}$$

3.1 Minimizing maximum acceleration

We design the proposed device based on the minimization of the maximum acceleration at the object

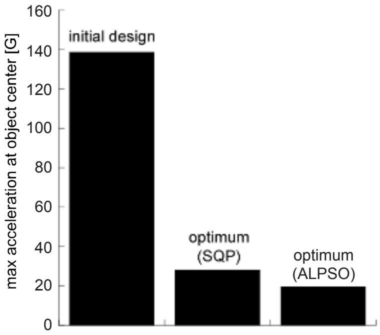

Fig. 6. Result of acceleration peak value.

center during the collision. The criterion is given by

$$A_{\max} = \max_t \; a(t) = \max_t \; \|\mathbf{a}(t)\|_2 \tag{11}$$

or

$$G_{\max} = A_{\max}/9.81, \tag{12}$$

where $\|\bullet\|_2$ means the Euclidean norm.

The optimization problem is formally written as

$$\begin{aligned} &\min_{\mathbf{p}} G_{\max}(\mathbf{p}) \\ &s.t. \quad \underline{\mathbf{p}} \le \mathbf{p} \le \overline{\mathbf{p}} \;. \end{aligned} \tag{13}$$

This problem is solved by two different optimization methods: SQP [3-5] and ALPSO [6, 7]. A standard SQP algorithm is provided in MATLAB as *fmincon()*. Since analytical sensitivity analysis is not available, the gradients are numerically approximated by finite differences. Starting from the same initial design, the following results are obtained

$$\begin{aligned} \mathbf{p}^*_{SQP} = \lfloor 7.33\times10^1 \quad 9.68\times10^2 \quad 1.00\times10^9 \\ 3.32\times10^{10} \quad 5.01\times10^1 \quad 4.69\times10^3 \rfloor \end{aligned} \tag{14}$$

and

$$\begin{aligned} \mathbf{p}^*_{ALPSO} = \lfloor 9.40\times10^2 \quad 1.06\times10^1 \quad 2.64\times10^9 \\ 7.48\times10^{10} \quad 1.17\times10^1 \quad 4.14\times10^3 \rfloor . \end{aligned} \tag{15}$$

Both optimized designs are non-symmetric. Note that there are differences in the translational and rotational spring-damper elements. Fig. 6 shows the maximum acceleration at the object center. In both cases, the maximum acceleration is remarkably reduced (by about a factor of seven). ALPSO seems to be slightly better than SQP in terms of the objective function. Fig. 7 shows the time history of the acceleration at the object center. The acceleration curve by

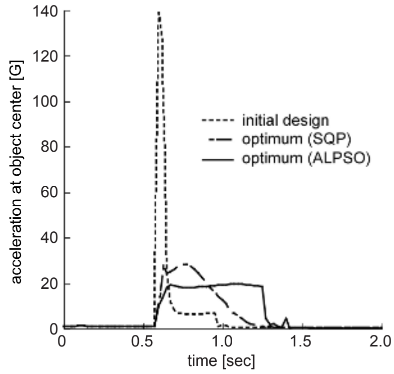

Fig. 7. Result of acceleration peak value minimization (time versus acceleration at object center).

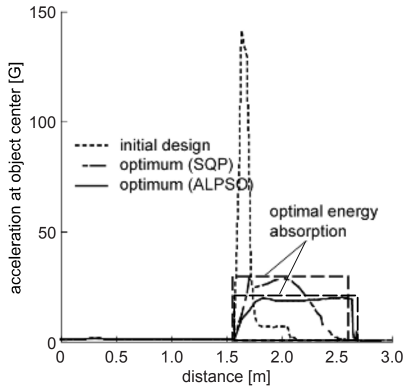

Fig. 8. Result of acceleration peak value minimization (distance versus acceleration at object center).

SQP has a peak, while the curve by ALPSO is almost flat. Fig. 8 illustrates the absorption of the kinetic energy by the optimized bumper struts. The areas under the plotted curves correspond to the amount of absorbed energy (acceleration integrated over the distance of the object center of mass). The rectangular areas depicted by the dashed lines mean the upper bounds that can be absorbed by the constant acceleration. ALPSO is also better than SQP in this measure by exploiting the possible rectangular area.

3.2 Effect of the shock absorbing device

Fig. 9 shows an effect of the initial design of the shock absorbing device. The inclined bumper causes the avoidance of a frontal collision. However, the maximum acceleration is very high. The struts of the initial design are too stiff. The initial design does not utilize the extended deformable zone and cannot reduce the energy of the collision. On the other hand,

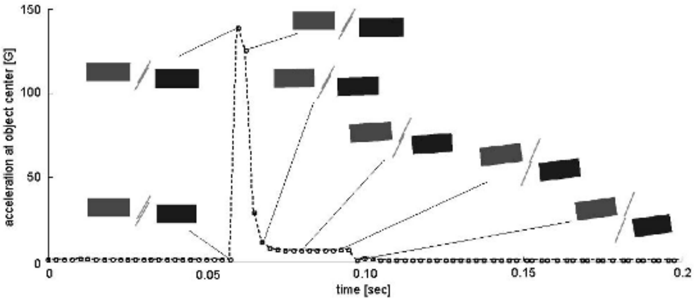

Fig. 9. Effect of the shock absorbing device (initial design).

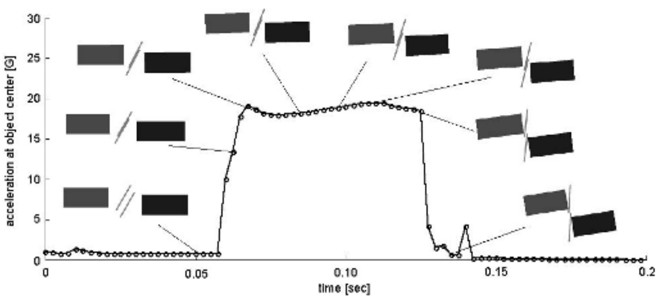

Fig. 10. Effect of the shock absorbing device (optimized design).

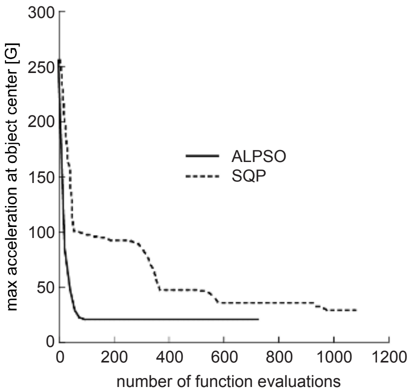

Fig. 11. Performance characteristic.

Fig. 10 shows an effect of the optimized shock absorbing device. The optimized device alleviates the crash effects. The inclined bumper leads to a relative sliding and pushing motion of the colliding objects. The maximum acceleration is remarkably reduced.

3.3 Performance characteristic

Fig. 11 shows the optimization histories of the two methods. Both methods are comparable in the reduction of the maximum acceleration. Regarding the convergence characteristic, ALPSO converged quicker in the early stage. In contrast, SQP converged in a staircase pattern. ALPSO produced a better convergence rate than SQP. An increment of function evaluations increases the calculation time. ALPSO is predictably effective in this problem.

4. Conclusions

In this paper, we have proposed a new shock absorbing device. Based on a simple rigid body model with penalty contact, the basic functionality and the development potential of the articulated bumper mechanism were investigated by two different optimization methods: SQP and ALPSO. The design variables, namely the stiffness and damping coefficients of the bumper struts were designed using the optimization.

It was demonstrated that the proposed articulated bumper mechanism may be able to considerably reduce impact accelerations. This was achieved by two main principles, namely the energy absorption by the extended deformable zone and the repulsive motion caused by the inclined bumpers thus reducing the amount of energy. However, if the repulsive motion is emphasized too much, this leads to an increase of the amount of energy that is left after the collision, which might cause another collision or further damage.

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