

Technical Report ARAEW-TR-08005

A Study of Dynamic Impact Models for Pile-Driver Breech Fatigue System

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March 2008



ARMAMENT RESEARCH, DEVELOPMENT AND ENGINEERING CENTER
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REPORT DOCUMENTATION PAGE				<i>Form Approved</i> <i>OMB No. 0704-0188</i>	
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1. REPORT DATE (DD-MM-YYYY) xx-03-2008		2. REPORT TYPE Final		3. DATES COVERED (From - To) 2008	
4. TITLE AND SUBTITLE A Study of Dynamic Impact Models for Pile-Driver Breech Fatigue System				5a. CONTRACT NUMBER	
				5b. GRANT NUMBER	
				5c. PROGRAM ELEMENT NUMBER	
6. AUTHOR(S) Cho, Changki ESEP Engineer South Korea				5d. PROJECT NUMBER	
				5e. TASK NUMBER	
				5f. WORK UNIT NUMBER	
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) U.S. Army ARDEC, AETC Benét Laboratories, AMSRD-AAR-AEW, B. 40 1 Buffington Street Watervliet Arsenal, NY 12189-4000				8. PERFORMING ORGANIZATION REPORT NUMBER ARA EW-TR-08005	
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES) U.S. Army ARDEC, AETC Benét Laboratories, AMSRD-AAR-AEW, B. 40 1 Buffington Street Watervliet Arsenal, NY 12189-4000				10. SPONSOR/MONITOR'S ACRONYM(S)	
				11. SPONSOR/MONITOR'S REPORT NUMBER(S)	
12. DISTRIBUTION/AVAILABILITY STATEMENT Approved for public release; distribution is unlimited.					
13. SUPPLEMENTARY NOTES					
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15. SUBJECT TERMS Impact modeling • Contact stiffness • Hysteresis damping					
16. SECURITY CLASSIFICATION OF:			17. LIMITATION OF ABSTRACT U	18. NUMBER OF PAGES 19	19a. NAME OF RESPONSIBLE PERSON
a. REPORT U/U	b. ABSTRACT U	c. THIS PAGE U			19b. TELEPHONE NUMBER (Include area code)

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A Study of Dynamic Impact Models for Pile-Driver Breech Fatigue System

Abstract

A lumped parameter approach to simulate the impact behavior of the impact pile-driver breech fatigue system is hereby presented. The goal is to investigate the fundamental characteristics of current system and find further improvement of hardware. To model and simulate the highly nonlinear behavior of the system, a good contact force model is essential. With this analysis, a hysteresis damping function is incorporated in the impact force model in order to achieve a desirable impact behavior of an impact pile driver and it allows accurate impact force modeling to simulate a realistic pressure time pulse pattern. Using the nonlinear contact force model, it is found that reliable results for the impact force can be achieved as the impact velocity increases. Finally, the numerical studies are compared with experimental measurements. It is shown that by selecting the proper contact stiffness and damping the chamber pressure pattern, which means the actual chamber pressure in the live firing of gun tube, can be created and controlled.

Key Words: Impact modeling • Contact stiffness • Hysteresis damping

1. Introduction

In many mechanical systems, typical dynamic loading is expressed by the form of impulsive force, which is exerted on the structure. The Impact phenomenon is the most common type of dynamic loading conditions that give rise to impulsive forces. The impact is characterized by abrupt changes in the values of system variables. Therefore, some appropriate contact-impact force model must be introduced in order to simulate and design these impacts in mechanical systems adequately.

In some cases, the estimation of the patterns and maximum values of contact forces for design or safety reasons are required. But the impact models based only on simple Hertzian contact law do not account for any energy loss due to impact because impacts are assumed to be perfectly elastic.

Hunt and Crossley [1] showed that linear spring-damper model did not represent the physical nature of energy transfer process and proposed the contact forces by the Hertz force displacement law. They estimated the energy dissipated due to impact was obtained in terms of a nonlinear damping force that was proportional to the n th power of elastic indentation between the contact surfaces. However, this analysis is confined to two bodies impacting at low velocities. Khulief and Shabana [2] developed a spring-dashpot model with a

nonlinear stiffness and a nonlinear damping in order to account for the energy loss due to impact. Yigit et al. [3] showed that a spring-dashpot model with nonlinear stiffness and damping law gives good results for the impact response of a radially rotating beam. The disadvantage of the spring-dashpot model is that the parameters of a damping model are difficult to obtain. Generally, the coefficient of restitution together with a simplified lumped parameter model of the system have to be used to obtain the damping parameters. Lankarani and Nikravesh [4][5] extended a contact force model with hysteresis damping to impact in multibody systems. Schwab et al. [6] investigated the dynamic response of mechanisms and machines affected by revolute joint clearance. They showed the procedure to estimate the maximum contact force during impact. Zhang and Vu-Quoc [7] studied the modeling of the coefficient of restitution as a function of the incoming velocity in elasto-plastic collisions. Anping Guo et al. [8] estimated the restitution coefficient straightforwardly from impact force history. Marhefka and Orin [9] studied the contact model with nonlinear damping for simulation of robotic systems and provided a detailed discussion of a collision model that was originally proposed by Hunt and Crossley.

For many years, the determination of safe service life for large caliber cannon has been experimentally verified using hydraulic pressurization to economically replicate the firing pressures of large caliber cannon without firing live ammunition. Hydraulic simulation of breech assemblies has included a dynamic pressure pulse for increased fidelity of the experimental simulation. In particular, a new gun system will require tens of thousands of cycles to determine fatigue life. However, it is complicate to simulate the impulsive force such as the chamber pressure of large caliber gun barrel in order to estimate the endurance life of gun barrel or breech mechanism.

Up to now, impact models have been widely studied. But most researches are restricted within the impact analysis between elastic bodies and structures. Few analyses of the pile driver impact to replicate high pressure in a large caliber gun barrel have been published by Lassel [10], Kathe [11], and Berggren [12]. It is the purpose of this paper to investigate the dynamic characteristics of the pile driver fatigue tester regarding how it reproduce the chamber pressure that we want to simulate.

2. A Continuous Contact Force Model

The overview of the contact force model focuses on Hertzian contact force model with dissipation and the process of energy transfer in modeling impact.

When two solid bodies are in contact, indentation occurs in the local contact zone subjected to the contact force. In the analysis of continuous contact force model, it is very important to determine the relationship between the contact force and the relative indentation between the two bodies.

The impact is generally considered to occur in two phases separately, the indentation phase and the restitution phase. During the indentation phase, the two bodies were indented in the normal direction to the impact surface, and the relative velocity of the contact points on the two bodies in that direction is reduced to zero. The end of the indentation phase is referred to as the instant of maximum indentation. The restitution phase starts at this point and lasts until the two bodies separate completely.

The basic contact force model starts from the impact of two spheres. If the contact area between the colliding objects is assumed to be small, the simplest and the best known contact force model between two spheres of isotropic materials is the non-linear Hertz law based on the theory of elasticity [1]. It is represented as a polynomial dependence of the contact force F on the indentation δ :

$$F(\delta) = \begin{cases} k_h \delta^p & \delta > 0 \\ 0 & \delta \leq 0 \end{cases} \quad (1)$$

where the generalized stiffness constant k_h depends on the material properties and the radii of the spheres and δ is the local relative normal indentation between the surfaces of the two spheres. So the indentation δ is computed as the difference between the displacements at contact point of two spheres. Therefore, the condition $\delta > 0$ represents that there is actual indentation, while the complementary condition $\delta \leq 0$ states that the two spheres are not in contact. For the frictionless Hertzian contact between two spheres, the exponent p is set to 1.5 for metallic surfaces and the stiffness parameter k_h [4] is given by

$$k_h = \frac{4}{3\pi(h_1 + h_2)} \sqrt{\frac{R_1 R_2}{R_1 + R_2}}, \quad (2)$$

where the material parameters are

$$h_i = \frac{1 - \nu_i^2}{\pi E_i}, \quad i = 1, 2 \quad (3)$$

with radius R_i , Poisson's ratio ν_i and Young's modulus E_i associated with each sphere.

The most complicated part of impact modeling is the process of energy transfer. Because the original Hertzian contact force model does not represent the energy dissipation, it can not cover both of the two phases, compression and restitution, during impact.

Some important conclusions can be drawn from the study presented in the works [1][2][3]. The Hertz relation besides its nonlinearity does not account for the energy dissipation during the impact process.

The contact force models given by Equations (1) and (2) are only valid for colliding bodies with circular contact areas. Some authors suggest the using of the more general force-displacement relation given by equation (4) but with a lower exponent, p , between 1 and 1.5.

Therefore, the Hertz relation along with the modification to explain the energy dissipation in the form of internal damping can be adopted for modeling contact forces in vibroimpact[1] and in a multibody system[4][5][6]. Under the hypothesis that the contact surface is small, Hunt and Crossley proposed the following form for the contact force F :

$$F(\delta, \dot{\delta}) = \begin{cases} k_h \delta^p + c_h \dot{\delta} & \delta > 0 \\ 0 & \delta \leq 0 \end{cases} \quad (4)$$

$$c_h = \mu \delta^p, \quad (5)$$

where δ is the indentation depth, $\dot{\delta}$ is the indentation velocity, and k_h and p are defined as above. The parameter c_h is called a nonlinear damping coefficient or a hysteresis form for the damping coefficient and μ is called the hysteresis damping factor. Similarly to Equation (1), the value of the exponent p depends only the local geometry around the contact surface. The force model of Equation (4) includes both an elastic component $k_h \delta^p$ and a dissipative term $\mu \delta^p \dot{\delta}$, and this dissipative term depends on both δ and $\dot{\delta}$, and is zero for zero indentation.

The so-called hysteresis damping factor μ can be estimated from a comparison of the energy loss at a central impact of two spheres with coefficient of restitution e after one hysteresis loop, yielding

$$\mu = \frac{3k_h(1-e^2)}{4\dot{\delta}^{(-)}}, \quad (6)$$

with $\dot{\delta}^{(-)}$, the indentation velocity just before impact, i.e., the initial impact velocity.

This model is expressed as,

$$F(\delta, \dot{\delta}) = k_h \delta^p \left[1 + \frac{3(1-e^2)}{4} \frac{\dot{\delta}}{\dot{\delta}^{(-)}} \right] \quad (7)$$

where k_h is the generalized stiffness constant, e is the coefficient of restitution, $\dot{\delta}$ is the relative indentation velocity and $\dot{\delta}^{(-)}$ is the initial impact velocity.

A major drawback of this model is the dependency of the hysteresis damping factor μ on the initial impact velocity $\dot{\delta}^{(-)}$. In finding this we have to track down the precise moment of impact which makes the continuous model partly non-smooth. Furthermore it can be shown that the approximate model underestimates the amount of dissipated energy, and consequently result in a high velocity after impact. For a restitution factor e of 0.75 and above, the error in the velocity after impact is less than 10%, while the error in the dissipated energy is less than 25%.

The coefficient of restitution mainly depends on the nature of the two materials of which the colliding objects are made. It is also affected by the impact velocity, the shape and size of the colliding objects, the location on the colliding objects at which the collision occurs, and their temperatures. It can be represented by the ratio of the relative velocities of the colliding objects before and after impact. For impacts of a ball(object 1) off a fixed surface(object 2), it can be simplified as velocity of ball after impact over velocity of ball before impact because the velocity of the fixed surface before and after impact is zero. And also it may be described by the ratio between decreasing heights of a bouncing ball.

However, Anping Guo et al. [8] estimated the coefficient of restitution straightforwardly from impact force history. Because the maximum force occurs when the deformation of the impacted body at the contact is at a maximum, the coefficient of the restitution is expressed as magnitude of restoring impulse over magnitude of deforming impulse defined as:

$$e = \frac{\text{magnitude of restoring impulse}}{\text{magnitude of deforming impulse}} \quad (8)$$

3. A Mathematical Model

3.1 Description of the existing system

The general view of pile driver breech fatigue system used to test the breech components are represented in Fig. 1. Pile driver breech fatigue system is composed of a hammer, a loading piston, a seal assembly, an hydraulic oil and a filler bar, the stub tube, the breech assembly, a fixed receiver ring and the main frame. Hammer drops vertically from fixed height and impacts on the loading piston. The loading piston transfers the impacted force to the test object

connected to the main frame through a seal assembly, a filler bar and hydraulic oil.



Fig.1. General view of the pile driver breech fatigue system

The sectional view of impact force transfer mechanism showing the basic structural elements of this system is described in Fig. 2.

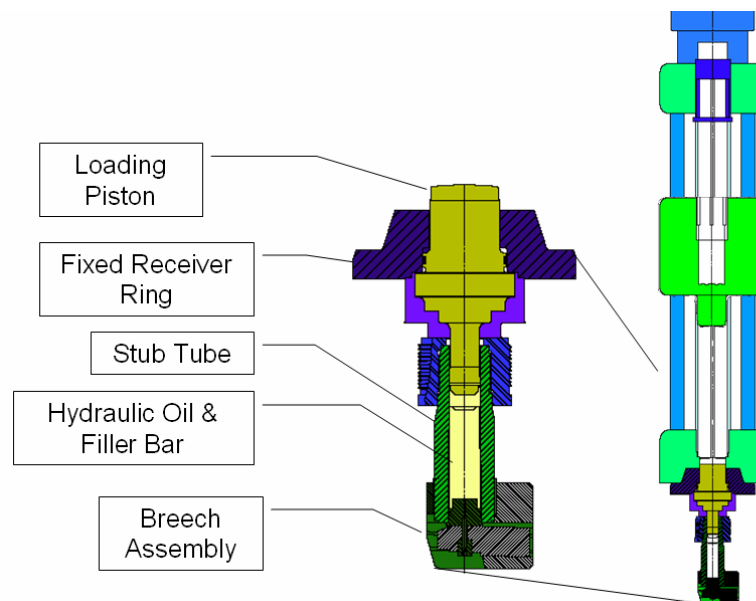


Fig.2. Sectional view of the impact force transfer mechanism

3.2 Equations of Motion

Adopting a very complex model may be considered as poor a judgment as an adopting an oversimplified model because the energy and time required to study complex models is wasteful. A reasonably simplified model makes the analysis much simpler as the result of reducing the number of variables and the complexity of the resulting equations of motion.

Therefore, a mathematical model for the pile driver breech fatigue system is represented by a simplified model which has equivalent inertia, damping, and stiffness characteristics. The fundamental case of a Hertzian contact law and a non-linear damping law in relation to the modeling of the impact system are introduced.

Because the interest is focused only on the reproduction of the magnitude and pattern of oil pressure exerted on the test specimen during impact, a simpler model can be used. Fig. 3. shows a three degree of freedom lumped model for the pile driver breech fatigue system.

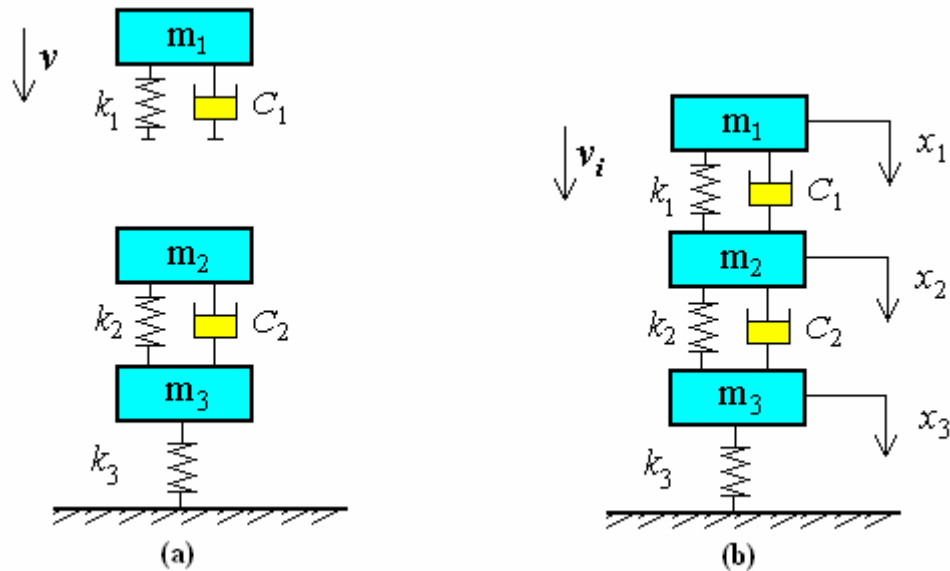


Fig.3. A lumped-parameter model of the pile driver breech fatigue system
(a) Before impact (b) During impact

It is known that the linear spring-damper model does not represent the physical nature of energy transfer process, therefore a hysteresis damping function, which represents the dissipated energy during the impact, is introduced in the impact force model. Hertzian modified contact model on the impact only between hammer and loading piston is considered, but not considered in the contact between the loading piston and the seal assembly.

Applying Newton's 2nd law, $\Sigma F = ma$, to each mass in Fig. 3, the governed equations are derived from the free-body diagram as the following three coupled equations:

$$\begin{aligned} m_1 \ddot{x}_1 &= -\rho k_1 (x_1 - x_2)^p - \mu (x_1 - x_2)^p (\dot{x}_1 - \dot{x}_2) \\ m_2 \ddot{x}_2 &= +\rho k_1 (x_1 - x_2)^p + \mu (x_1 - x_2)^p (\dot{x}_1 - \dot{x}_2) - k_2 (x_2 - x_3) - c_2 (\dot{x}_2 - \dot{x}_3) \\ m_3 \ddot{x}_3 &= +k_2 (x_2 - x_3) + c_2 (\dot{x}_2 - \dot{x}_3) - k_3 (x_3) \end{aligned} \quad (9)$$

$$\rho = 1, \text{ for } (x_1 - x_2) > 0$$

$$\rho = 0, \text{ otherwise}$$

where m_1 is the lumped mass of the hammer, m_2 is the equivalent mass of the loading piston and seal assembly and m_3 is the equivalent mass of the fixed receiver ring, the stub tube and the breech assembly connected to the main frame, and their corresponding displacements are expressed as x_1 , x_2 and x_3 , respectively. In this model, the coefficient k_1 represents the nonlinear contact stiffness k_h from Hertzian contact model, $F(t) = k_h \delta(t)^p$, $p=1.5$, where δ is the indentation of the bodies at the contact points. It means the difference between the displacement of hammer, x_1 , and of loading piston, x_2 . The coefficient k_2 represents the bulk modulus of hydraulic oil. The coefficient k_3 represents the equivalent stiffness of the joints between stub tube including breech assembly and fixed receiver ring. In this model, the damping coefficient c_1 represents a nonlinear damping coefficient $c_h = \mu \delta^p$ and is proposed to account for the energy loss during impact. The damping coefficient c_2 is introduced to account for the dissipation of energy as the result of friction of seal assembly.

The nonlinear differential equations (2) are solved for the initial conditions.

$$\begin{aligned} x_1(0) &= x_2(0) = x_3(0) = 0 \\ \dot{x}_1(0) &= v_i \\ \dot{x}_2(0) &= \dot{x}_3(0) = 0 \end{aligned} \quad (10)$$

However, the validity of model in representing the real system depends on how to tune well the inertia, damping and stiffness characteristics of the system.

4. Model Simulation and Validation

4.1 Model Simulation

The solution of equation (1) is obtained numerically using the 4th-order Runge-Kutta method. To solve the equation (1), the Runge-Kutta method is implemented by a windows supported MATLAB package. The calculation algorithm can be described as follows: (1) Assign the initial conditions and the total iteration time for calculation. (2) Assign the parameter values of m_1 , m_2 , m_3 , k_1 , k_2 , k_3 , c_1 , c_2 , μ , p and v_i . (3) Compute the state variables using the Runge-Kutta method. These steps were iterated until the final time is satisfied.

4.1.1 Determination of input parameters

Since we are interested in the shape of pressure, it is important to select the contact force model correctly. But major difficulty in the present design is how to define indefinite stiffness and damping parameters. At first, the estimated values in conjunction with the input parameters such as the stiffness, the damping coefficient and the damping frictional coefficient were used because the exact values were unknown and later were compared to the experimental values of the system.

First, the stiffness parameter k_1 of contact area needs to be determined. It is assumed that the contact area keeps circular contact and is much smaller than the radii of curvature of the hammer and the loading piston. The materials in contact are steels with Young's modulus $E = 2.1 \times 10^{11} \text{ N/m}^2$, Poisson's ratio $\nu = 0.3$. The coefficient of restitution is assumed to be $e = 0.95$. The radii of curvature of the hammer and the loading piston are 300 mm . Therefore the stiffness parameter k_1 can be calculated from equation (2) to be $5.958 \times 10^{10} \text{ N/m}^{1.5}$. Hysteresis damping factor μ can be calculated from equation (6) to be 1.595×10^9 . But here because it is value by calculation, not by experiment, a little lower value 5.958×10^6 is used.

Second, the stiffness k_2 represents the characteristics of hydraulic oil dynamically connected to the stub tube and the breech assembly. It can be replaced by the bulk modulus of hydraulic fluid because the fluid is incompressible in general. Bulk modulus of the oil (water-glycol) is $2.189 \text{E}9$ and that of water at 100°C is $2.07 \text{E}9 \text{ N/m}^2$. Here it is assumed to be $2.228 \times 10^8 \text{ N/m}^2$.

All the parameters used for the calculations in this model are listed in Table 1. Especially, an impact velocity, $\dot{\delta}^- (\text{m/s})$, of the hammer just before impact is calculated from $v_0 = \sqrt{2gh}$, which depends on the drop height(h).

Table 1 The parameters of the pile driver breech fatigue system

Mass of the hammer	m_1	2,948.4 kg
Mass of the loading piston and seal assembly	m_2	241.7 kg
Mass of the stub tube and breech assembly	m_3	500 kg
Contact stiffness of the hammer and loading piston during impact	k_1	$5.958 \times 10^{10} \text{ N/m}^{1.5}$
Stiffness of the hydraulic oil	k_2	$2.228 \times 10^8 \text{ N/m}$
Stiffness of the combined stub tube and breech assembly	k_3	$2.500 \times 10^{10} \text{ N/m}$
Nonlinear damping coefficient	c_1	$c_1 = \mu(x_1 - x_2)^p$
Damping frictional coefficient of the seal assembly	c_2	$2.556 \times 10^4 \text{ Ns/m}$
Cross-sectional area of the piston	A_p	$2.27 \times 10^{-2} \text{ m}^2$
Impacting velocity(depends on the hammer drop height)	v_i	$2.73 \sim 6.73 \text{ m/s}$
Arbitrary exponent	p	1.5
Hysteresis damping factor	μ	1.595×10^6

4.1.2 Simulation

The impact phase generally consists of two stages, indentation stage and restitution stage. After determining the impact force history, estimation of the coefficient of restitution is straightforward. In the impact force history, the maximum force occurs when the indentation of the loading piston at the contact is at a maximum, i.e., the displacement of the hammer is at a maximum. It also illustrates where the indented and final impulses occur. The boundary between these stages is the point of maximum indentation and the relative contact velocity in the normal direction vanishes in this point. The end of the indentation stage is characterized by the indented normal impulse P_i and it occurs when the impact velocity in the normal direction v_i momentarily equals zero. Once these events have been quantified, energy relationships are used to find the final normal impulse P_f , which characterizes the end of the restitution stage. The right area of the vertical axis is generally smaller than the left half area. Some of the energy is lost during impact.

During the duration of impact or contact, the impact force, $F(\delta, \dot{\delta})$, is always positive. Once it becomes negative in the progress of computation, the contact phase is complete.

Fig. 4 shows the impact force history of hammer exerted on the loading piston during impact at impact velocity $v_i = 2.73 \text{ m/s}$ using currently developed

model. Since the loading piston is initially at rest, the coefficient of restitution, e , can be estimated straightforwardly from the impact force history of Fig. 4. The coefficient of the restitution in this example is expressed as magnitude of restoring impulse ($area S_2$) over magnitude of deforming impulse ($area S_1$) based on the equation (8). Calculated coefficient of restitution ranges from 0.90 to 0.95.

Fig. 5 shows the hysteresis effects according to adding the nonlinear damping term to the nonlinear stiffness at impact velocity $v_i = 5.5$ m/s. It represents the nonlinear characteristics both in the contact force model with nonlinear stiffness and with the nonlinear stiffness plus nonlinear damping.

Fig. 6 shows pressure versus time history at the oil chamber during Impact with the impact velocity $v_i = 2.73$ m/s by simulation in the contact force model with nonlinear stiffness and nonlinear damping.

4.2 Model Validation

The presented simulation model has been validated by comparing the pressure results obtained by the simulation with the experimental results of the real system conducted by Benet Labs. It is the oil pressure exerted on the stub tube or breech assembly through the loading piston during impact.

4.2.1 Experimental results

The impact experiment was performed by using the 06 Vulcan impact pile driver system in order to validate the simulation model. The hammer is dropped freely on the loading piston through the guide-rail from the required height as shown in Fig. 1. It has the mass $m_1 = 2948.4$ kg and is suspended at the top of a frame. It is released at a vertical height $h = 0.254$ m above the loading piston and the initial impact velocity of the hammer $v_0 = \sqrt{2gh}$ depends on the height(h).

The oil pressures in chamber were measured with Kistler pressure sensor (Type 6203). The data acquisition system of model Odyssey made by Nicolet Instrument Technologies was used. A dual mode amplifier (Type 5010, Kistler) and a signal conditioning amplifier (Type 2310, Instruments Division) were used. The pressures were picked up at three locations, Kistler pressure sensors at two locations and HAT gage at one location of the oil chamber. HAT gage is made of strain gages and calibrated by Benet Labs. But acceleration and displacement were not measured.

Fig. 7 shows the oil pressure vs. time relation in the chamber by the experiment. It means the pressure exerted on the stub tube or breech assembly during impact. It is noted from Fig. 7 that the pressure exerted on the stub tube or breech assembly shows the harmonic or the higher modes of vibration. But

additional data such as the force and acceleration of hammer during impact the displacements of masses need to be measured to overcome the lack of data. These data can be acquired by the load cell, the accelerometer and the laser-type displacement sensor, respectively. And the impact hammer behavior can also be recorded on the high-speed video.

4.2.2 Comparison between experimental and simulation results

In order to validate this method, the numerical result of the pile driver system with nonlinear contact force model needs to be compared with the experimental ones. Fig. 8 shows the comparison of numerical result and experimental one, where experimental (solid) and calculated (dashdot) lines of pressure exerted on the stub tube or breech assembly through the loading piston and the oil in the chamber during impact were compared. It is found that the result obtained by simulation is almost in good agreement with that of the experiment.

The current modeling method offers a reasonable practical accuracy. The equivalent mass of the loading piston and seal assembly, m_2 , needs to be split into several separated masses in order to obtain more detailed information about transferring process of impact force. And also spring constant or damping coefficient corresponding to each mass has to be used.

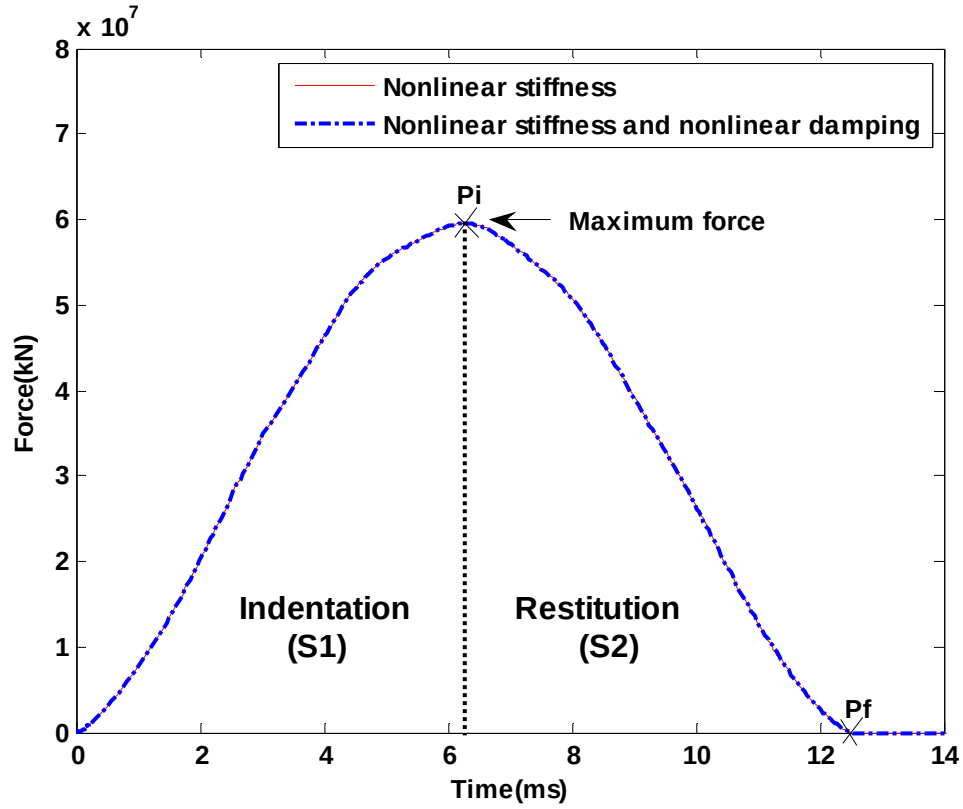


Fig. 4 Impact force history of hammer exerted on the loading piston during impact at impact velocity $v_i = 2.73 \text{ m/s}$: P_i is indented impulse and P_f is final impulse:

(a) $k_1 = 5.958 \times 10^{10} \text{ N/m}^{1.5}$, $k_2 = 2.228 \times 10^8 \text{ N/m}$, $k_3 = 2.500 \times 10^{10} \text{ N/m}$,

$c_2 = 2.556 \times 10^4 \text{ Ns/m}$, $p = 1.5$

(b) $\mu = 1.595 \times 10^6$, the others are the same in (a)

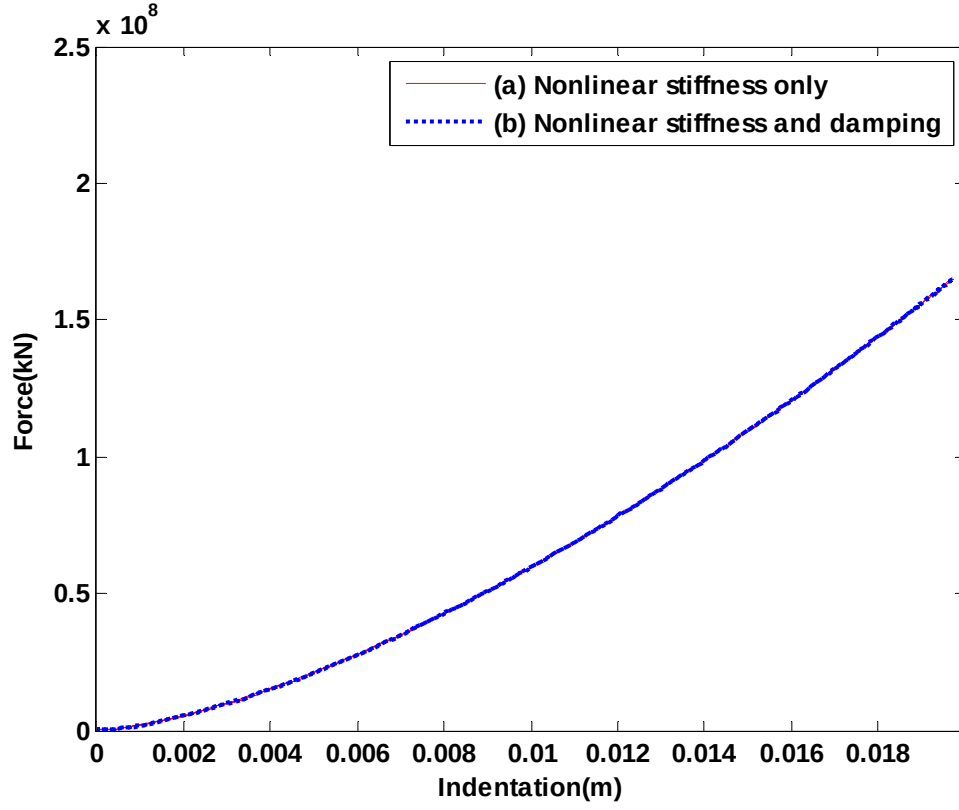


Fig. 5 Comparison of force-displacement curve according to adding the nonlinear damping term at impact velocity $v_i = 5.5 \text{ m/s}$:

(a) $k_1 = 5.958 \times 10^{10} \text{ N/m}^{1.5}$, $k_2 = 2.228 \times 10^8 \text{ N/m}$, $k_3 = 2.500 \times 10^{10} \text{ N/m}$,
 $c_2 = 2.556 \times 10^4 \text{ Ns/m}$, $p = 1.5$

(b) $\mu = 1.595 \times 10^6$, the others are the same in (a)

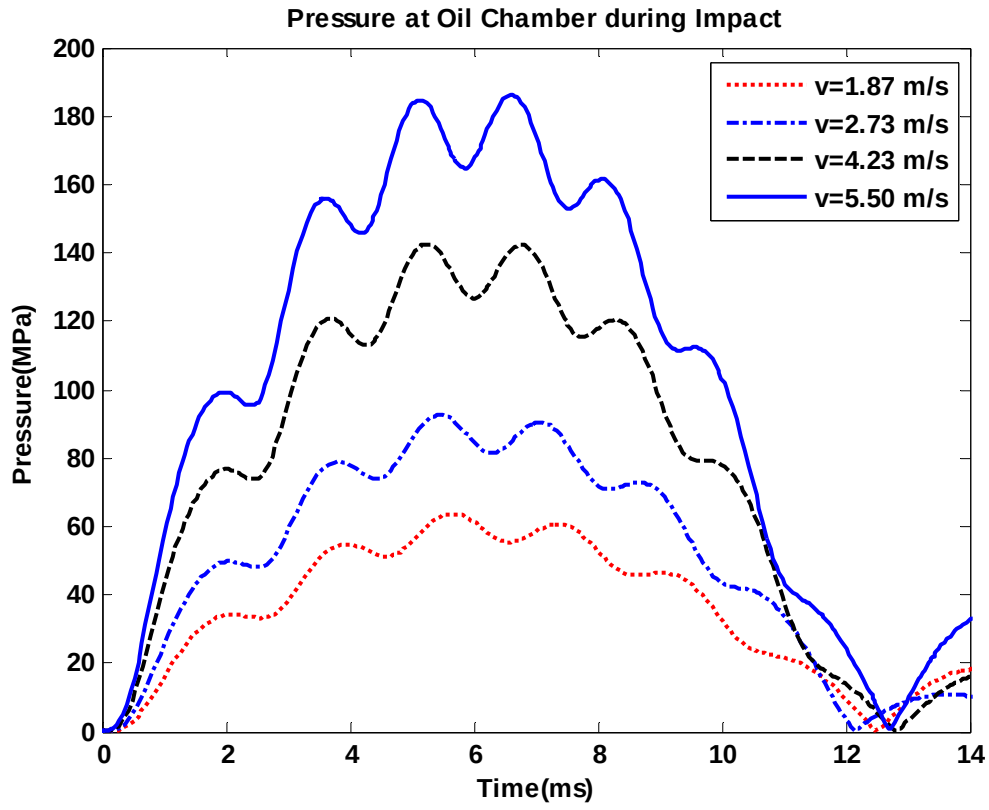


Fig. 6. Pressure vs. time history at Oil Chamber during Impact for different impact velocities $v_i = 1.87, 2.73, 4.23, \text{ and } 5.50 \text{ m/s}$ (Nonlinear stiffness only)

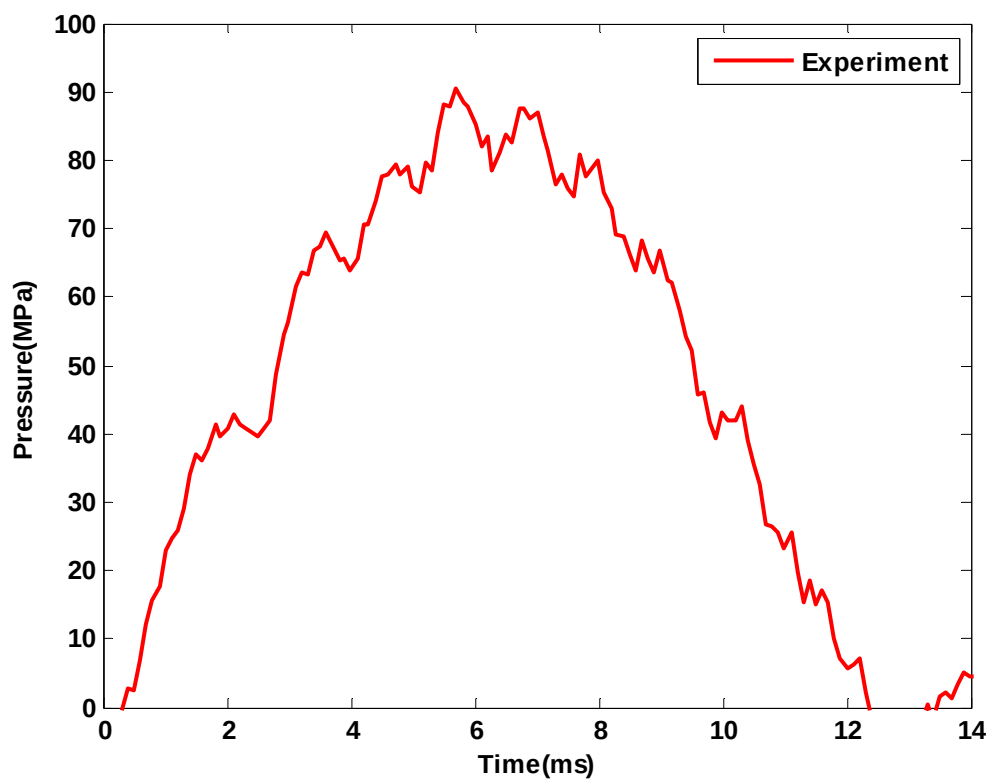


Fig. 7 Oil pressure vs. time history in the oil chamber by experiment

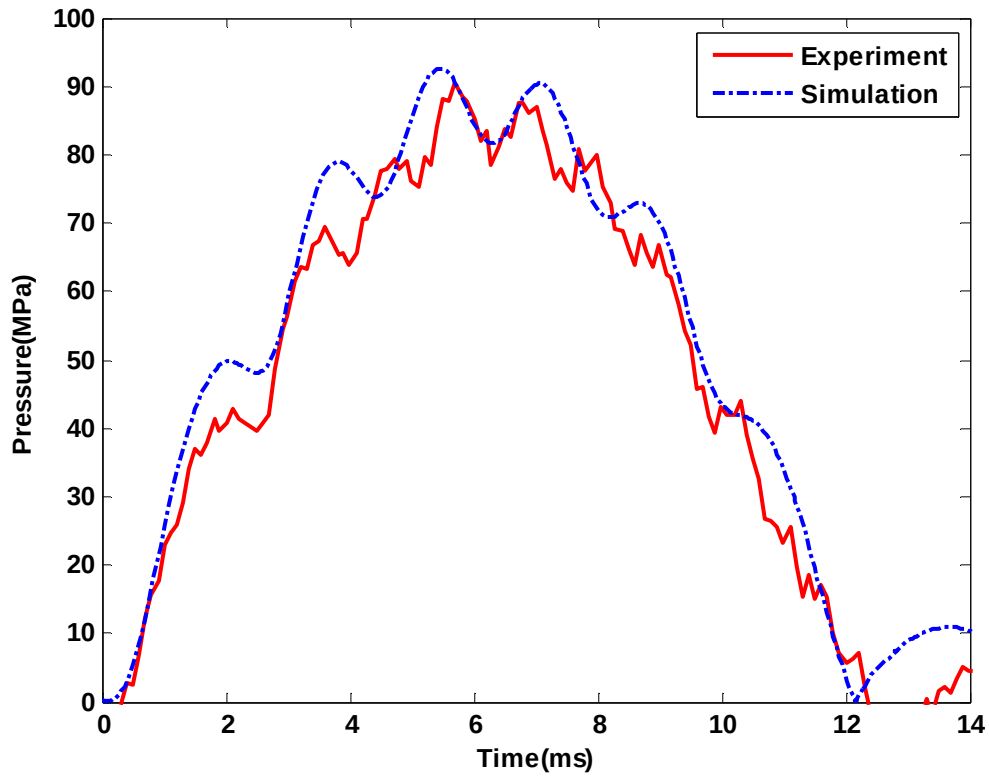


Fig. 8 Comparison between experimental and simulation result of impact force exerted on the stub tube or breech assembly. Parameters in simulation are as follows:

$$k_1 = 5.958 \times 10^{10} \text{ N/m}^{1.5}, k_2 = 2.228 \times 10^8 \text{ N/m}, k_3 = 2.500 \times 10^{10} \text{ N/m},$$

$$c_2 = 2.556 \times 10^4 \text{ Ns/m}, p = 1.5, \mu = 1.595 \times 10^6, v_i = 2.73 \text{ m/s}$$

5. Discussion

The comparison between the contact force model with nonlinear stiffness only and with the combination of nonlinear stiffness and nonlinear damping, for impact velocity $v_i = 2.73 \text{ m/s}$ is given in Fig. 9. It shows that the contact force model with the combination of nonlinear stiffness and nonlinear damping differs little from the model with nonlinear stiffness only for the given parameters.

Fig. 10 shows the variation of the contact force and indentation during impact for different impact velocities $v_i = 1.87, 2.73, 4.23, \text{ and } 5.50 \text{ m/s}$. Fig. 9(a) and (b) represent the variation of the contact force and indentation in the contact force model with the nonlinear stiffness only and Fig. 9(c) and (d) are the results of the contact force model with the combination of nonlinear stiffness and nonlinear damping. It shows that contact force and indentation increase as the velocity increases in both cases.

Fig. 11 shows the contact force versus indentation depth during impact according to the variation of velocity in the contact force model with nonlinear stiffness and damping. Fig. 12 shows the contact force versus time during impact according to the variation of the oil stiffness k_2 in the contact force model with nonlinear stiffness and nonlinear damping. It is found that the duration time of impact reduces as the oil stiffness k_2 increases, but is not influenced by the velocity increase.

Fig. 13 shows the pressure versus time history at oil chamber during impact for different impact velocities in the contact force model with nonlinear stiffness and nonlinear damping. It means the pressure exerted on the stud tube or breech assembly through the oil during impact.

Fig. 14 represents the effects of the variation of various parameters on m_2 during impact. It includes the displacement histories of m_1 , m_2 , and m_3 for $k_2 = 2.228 \times 10^8 \text{ N/m}$ and $v_i = 2.73 \text{ m/s}$ and displacement histories of m_2 according to k_2 , v_i , and μ variation. It is found that the displacement of m_2 decreases and the duration time shortens as oil stiffness k_2 increases, but the displacement of m_2 increases as the velocity increases. The effects of μ in the current parameters are not found. Fig. 15 shows the effects of the variation of parameters k_1 and p on m_2 during impact.

Fig. 16 shows the effects of the variation of parameter k_3 on the impact characteristics during impact. It represents that the contact force and

displacement reduces as the stiffness k_3 increases, but the pressure increases and the duration time shortens.

Fig. 17 shows the effects of the variation of damping coefficient c_2 on the impact characteristics during impact.

Fig. 18 shows the pressure time histories exerted on the stud tube or the loading piston through the oil in the chamber during impact with $v_i = 2.73 \text{ m/s}$ according to the variation of k_2 value.

Fig. 19 shows the phase-space plots according to impact velocities on $(\delta, \dot{\delta})$ plane during impact in the contact force model with the nonlinear stiffness and nonlinear damping. It represents the phase trajectories on $(\delta, \dot{\delta})$ plane of a hammer which strikes the loading piston with various impact velocities. The trajectories traveled in clockwise direction. The restitution velocities, $\dot{\delta}_{res}$, after impact are always smaller in magnitude than the corresponding indentation velocities, $\dot{\delta}_{ind}$. Moreover, for increasing $\dot{\delta}_{ind}$ the resulting $\dot{\delta}_{res}$ converges to the limit value. The hammer rebounds from the loading piston, while its velocity decreases in magnitude, due to the dissipative term in the contact force $F(\delta, \dot{\delta})$. The coefficient of restitution in this example can also be calculated purely from the phase-space plots of Fig. 19. It ranges from 0.88 to 0.92.

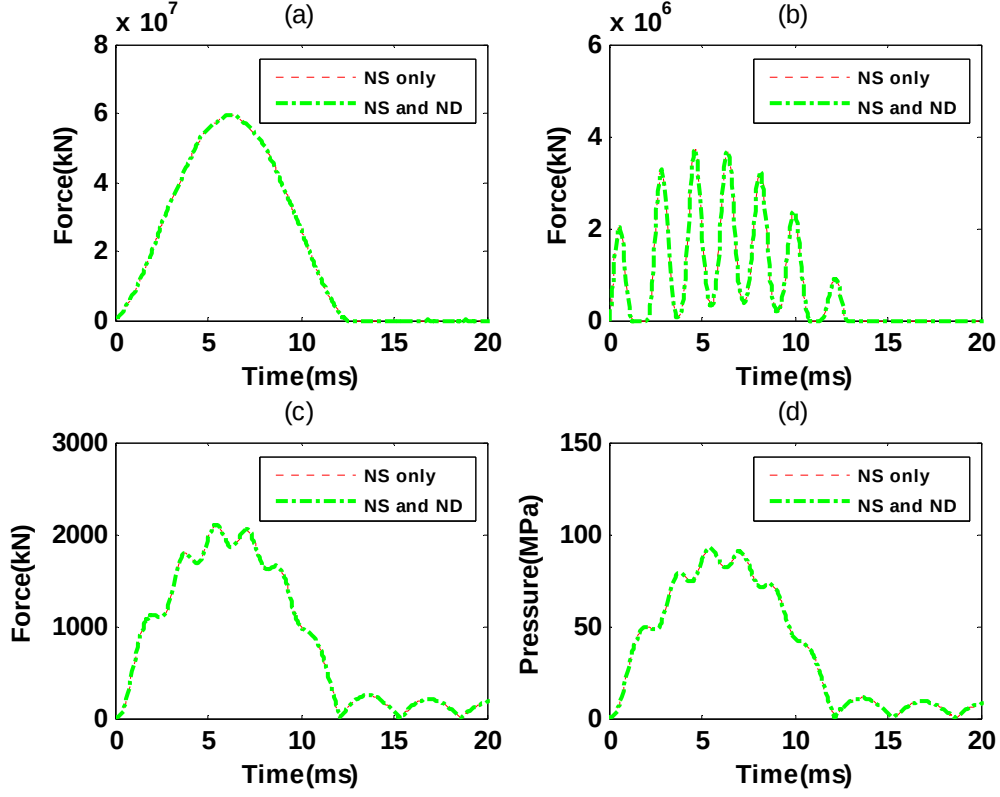


Fig. 9 Comparison between the contact force model with nonlinear stiffness and the contact force model with nonlinear stiffness and nonlinear damping for $v_i = 2.73 \text{ m/s}$. Parameters in simulation are as follows: $k_1 = 5.958 \times 10^{10} \text{ N/m}^{1.5}$, $k_2 = 2.228 \times 10^8 \text{ N/m}$, $k_3 = 2.500 \times 10^{10} \text{ N/m}$, $c_2 = 2.556 \times 10^4 \text{ Ns/m}$, $p = 1.5$, $\mu = 1.595 \times 10^6$

(a) Contact force $F = k_1(x_1)^p + \mu x_1^p(\dot{x}_1 - \dot{x}_2)$ versus time

(b) Contact force $F = k_1(x_1 - x_2)^p + \mu(x_1 - x_2)^p(\dot{x}_1 - \dot{x}_2)$ versus time

(c) Force at the loading piston $F = k_2(x_2 - x_3) + c_2(\dot{x}_2 - \dot{x}_3)$ versus time

(d) Pressure in the oil chamber

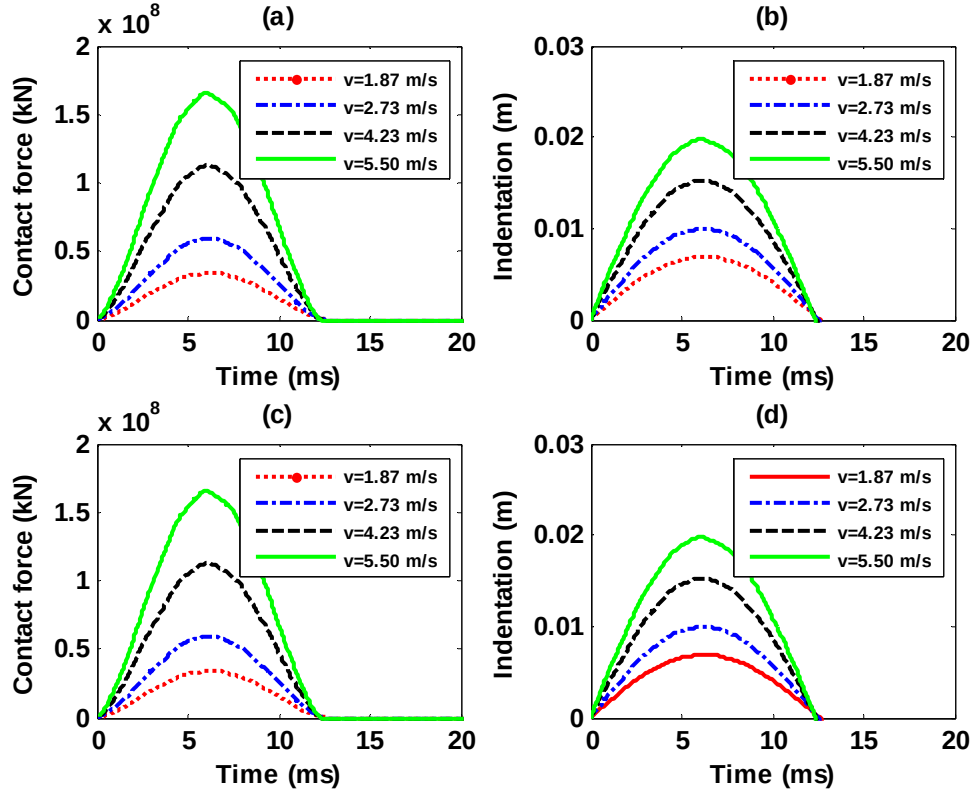


Fig. 10 Contact force and indentation during impact for different impact velocities

$v_i = 1.87, 2.73, 4.23, \text{ and } 5.50 \text{ m/s}$. Parameters in simulation are as follows:

$$k_1 = 5.958 \times 10^{10} \text{ N/m}^{1.5}, k_2 = 2.228 \times 10^8 \text{ N/m}, k_3 = 2.500 \times 10^{10} \text{ N/m}, c_2 = 2.556 \times 10^4$$

$$\text{Ns/m}, p = 1.5, \mu = 1.595 \times 10^6;$$

(a) Contact force versus time due (Nonlinear stiffness only)

(b) Displacement versus time due (Nonlinear stiffness only)

(c) Contact force versus time due (Nonlinear stiffness and nonlinear damping)

(d) Displacement versus time due (Nonlinear stiffness and nonlinear damping)

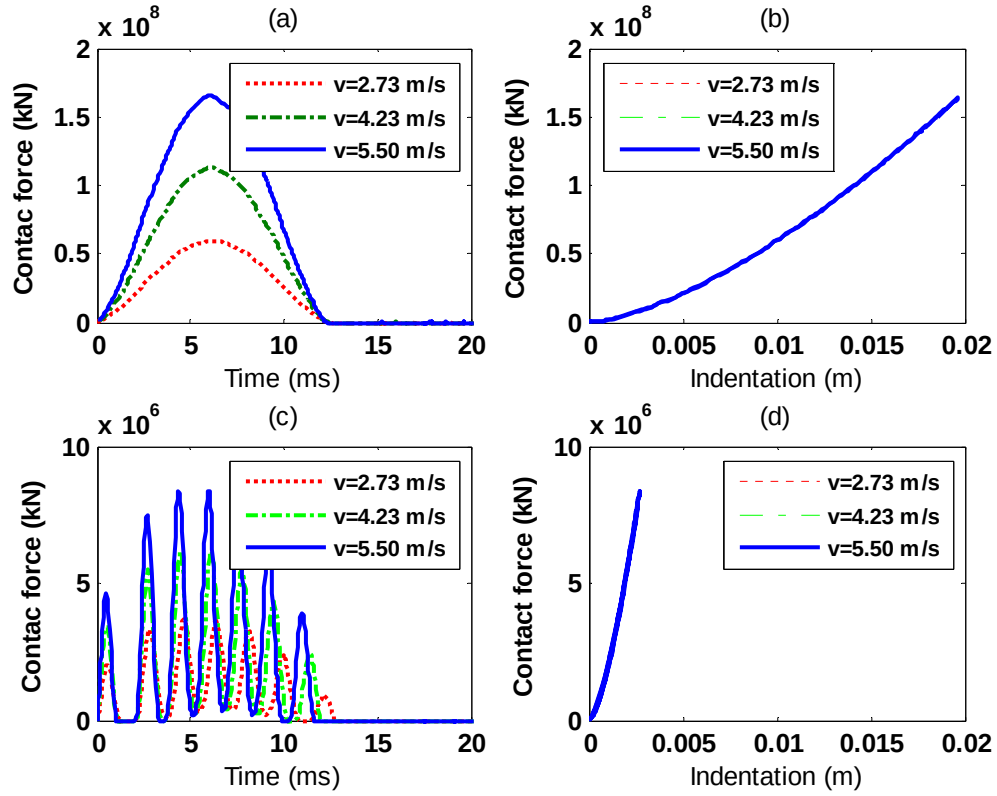


Fig. 11 Contact force versus indentation depth according to the variation of velocity in the contact force model with nonlinear stiffness and damping.

Parameters in simulation are as follows: $k_1 = 5.958 \times 10^{10} \text{ N/m}^{1.5}$, $k_2 = 2.228 \times 10^8 \text{ N/m}$, $k_3 = 2.500 \times 10^{10} \text{ N/m}$, $c_2 = 2.556 \times 10^4 \text{ Ns/m}$, $p = 1.5$, $\mu = 1.595 \times 10^6$

(a) & (b) Contact force: $F = k_1(x_1)^p + \mu x_1^p(\dot{x}_1 - \dot{x}_2)$, Indentation: x_1

(c) & (d) Contact force: $F = k_1(x_1 - x_2)^p + \mu(x_1 - x_2)^p(\dot{x}_1 - \dot{x}_2)$, Indentation: $(x_1 - x_2)$

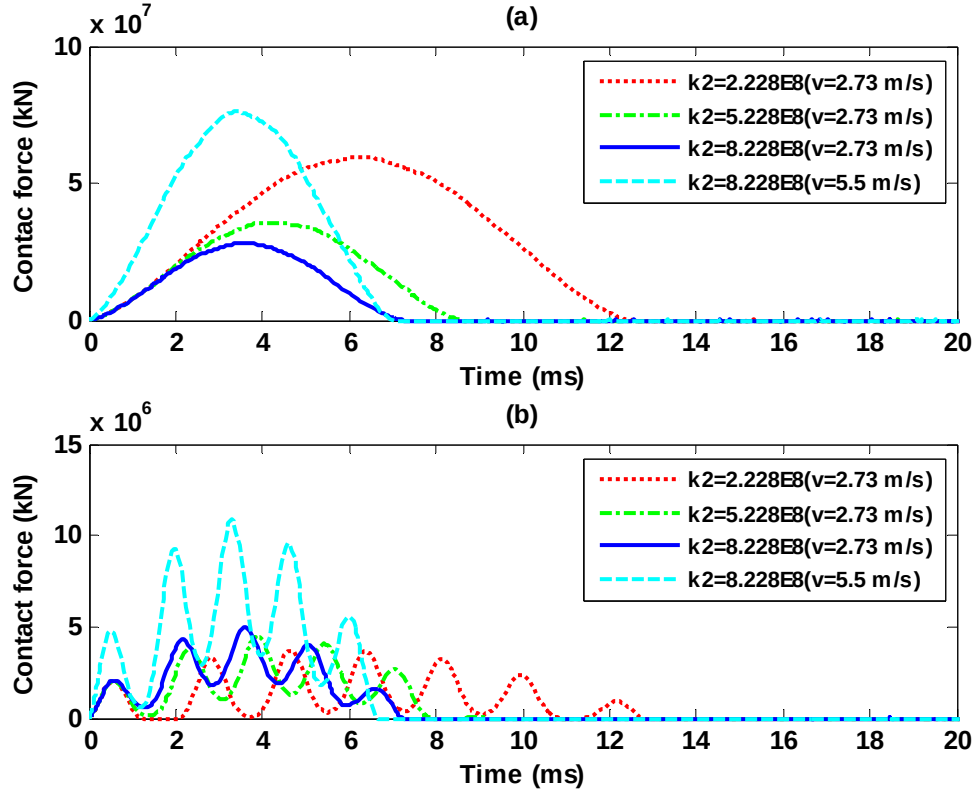


Fig. 12 Contact force versus time during impact according to the variation of the oil stiffness in the contact force model with nonlinear stiffness and nonlinear damping. Parameters in simulation are as follows:

$$k_1 = 5.958 \times 10^{10} \text{ N/m}^{1.5}, \quad k_2 = 2.228 \times 10^8 \text{ N/m}, \quad k_3 = 2.500 \times 10^{10} \text{ N/m}, \\ c_2 = 2.556 \times 10^4 \text{ Ns/m}, \quad p = 1.5, \quad \mu = 1.595 \times 10^6$$

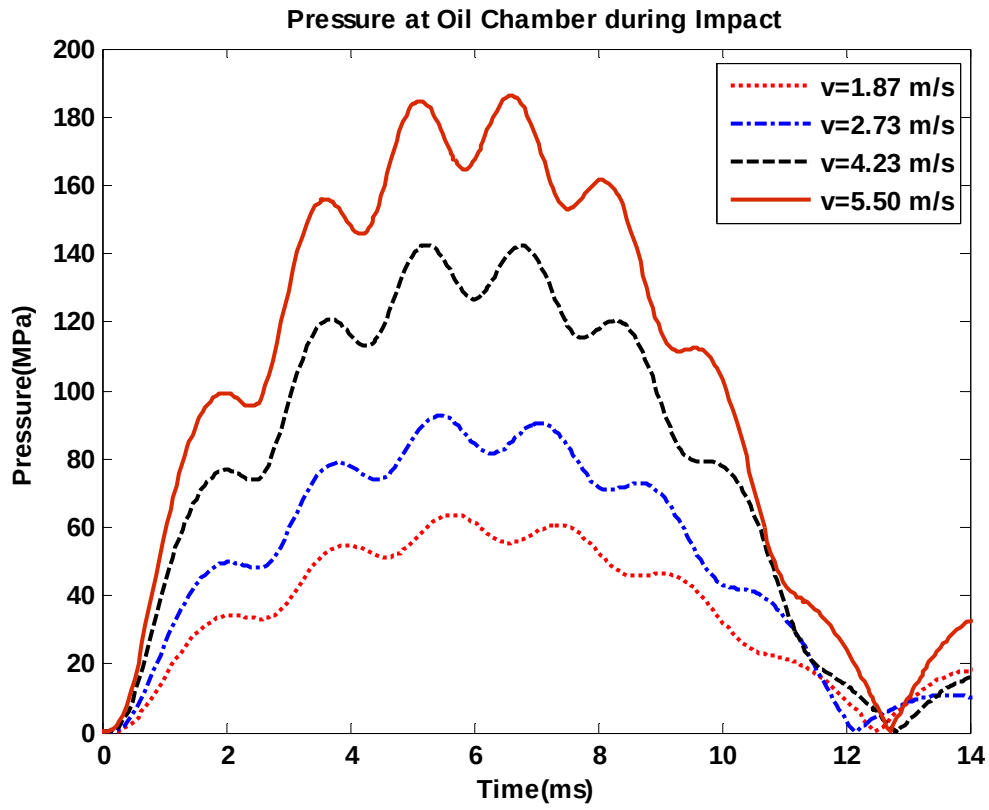


Fig. 13 Pressure vs. time history at Oil Chamber during Impact for different impact velocities $v_i = 1.87, 2.73, 4.23, \text{ and } 5.50 \text{ m/s}$ (Nonlinear stiffness and nonlinear damping)

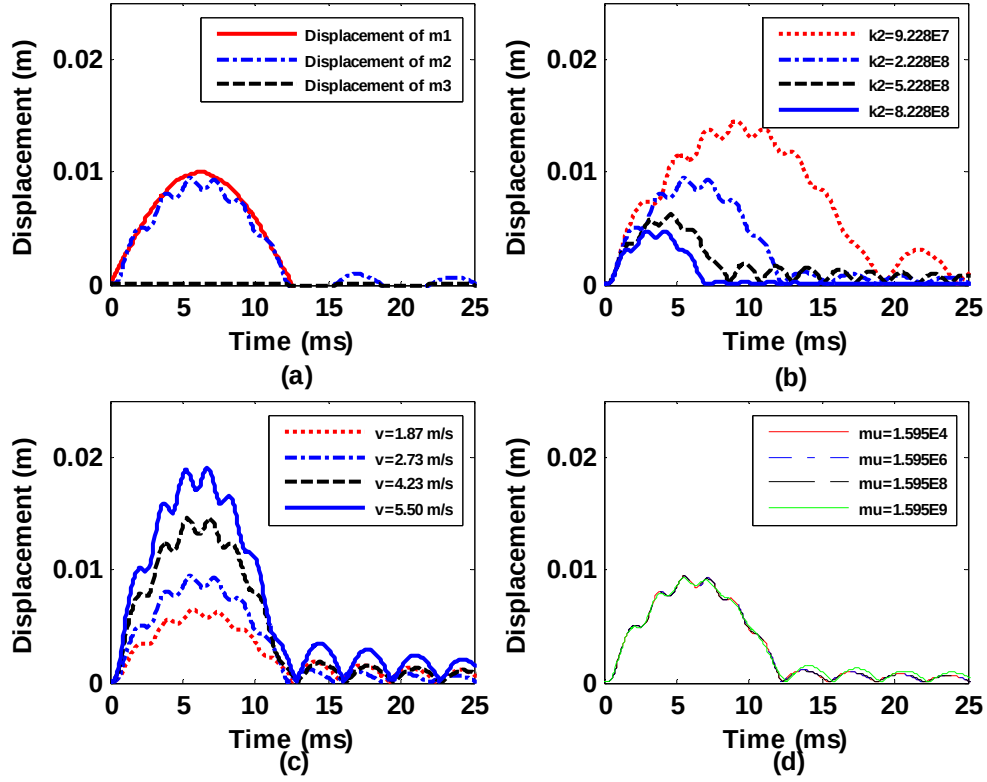


Fig. 14 Effects of the variation of parameters k_2 , v_i , and μ on m_2 during impact
(a) Displacement histories of m_1 , m_2 , and m_3 at $k_2 = 2.228 \times 10^8 \text{ N/m}$ and $v_i = 2.73 \text{ m/s}$

(b) Displacement histories of m_2 according to k_2 variation

(c) Displacement histories of m_2 according to v_i variation

(d) Displacement histories of m_2 according to μ variation

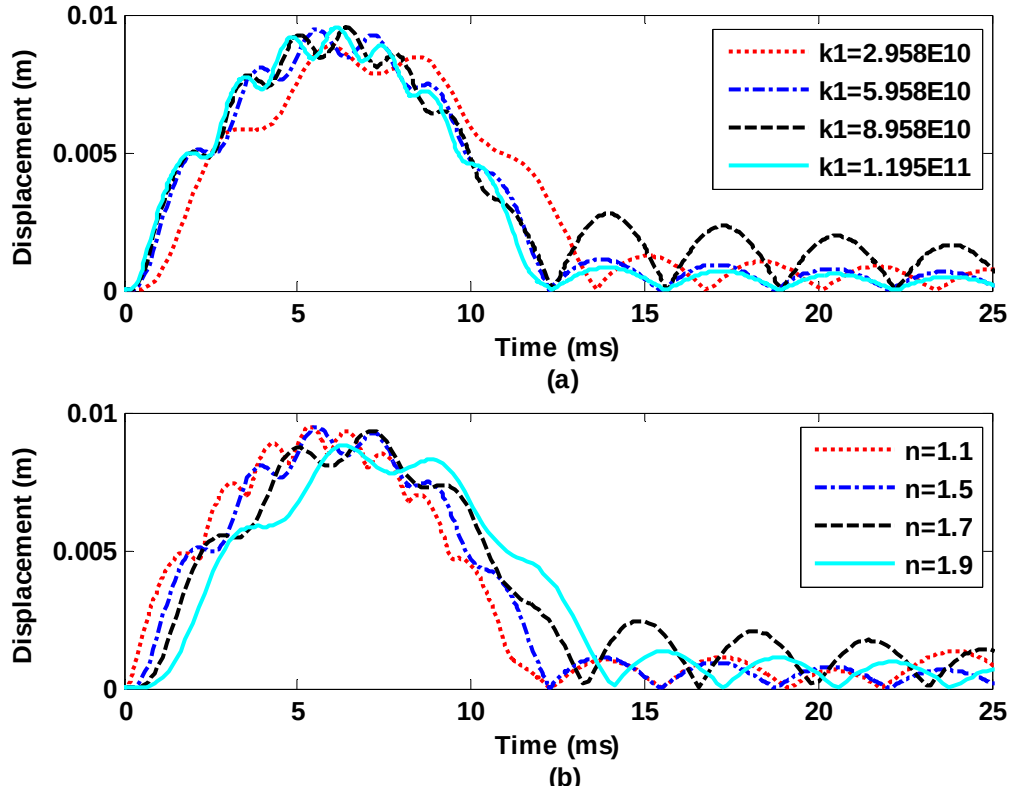


Fig. 15 Effects of the variation of parameters k_1 and p on m_2 during impact
(a) Displacement histories of m_2 according to k_1 variation
(b) Displacement histories of m_2 according to p variation

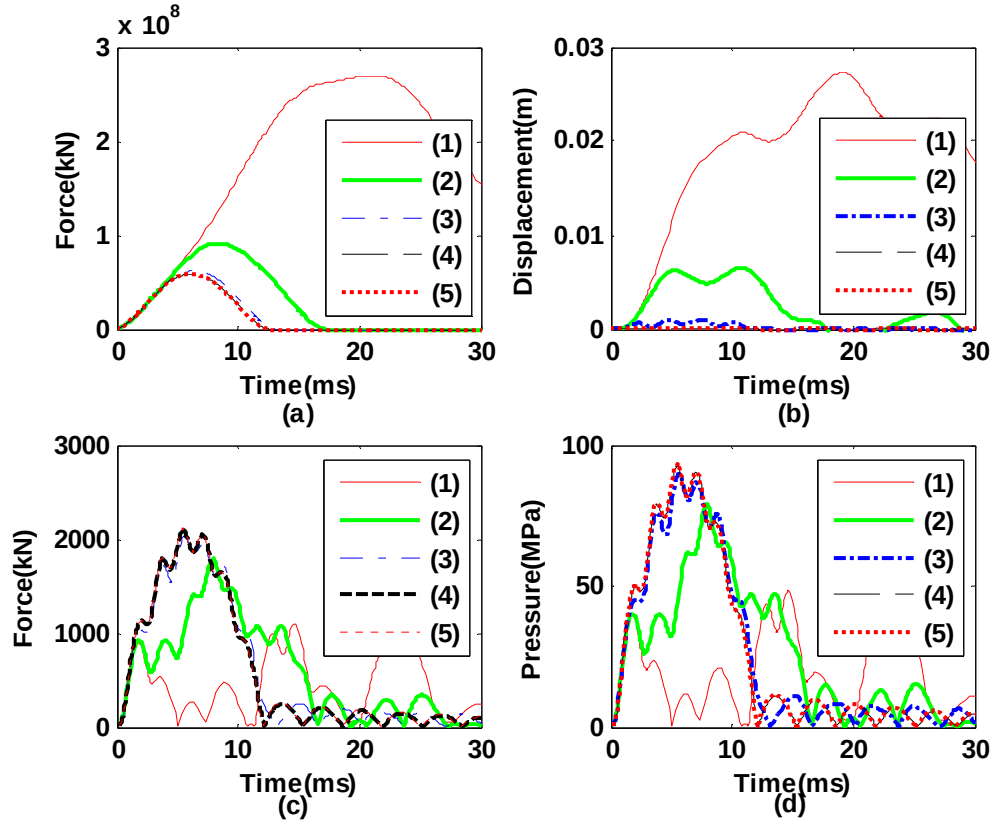


Fig. 16 Effects of the variation of parameter k_3 on the impact characteristics during impact. Parameters in simulation are as follows: $k_1 = 5.958 \times 10^{10} \text{ N/m}^{1.5}$, $k_2 = 2.228 \times 10^8 \text{ N/m}$, $c_2 = 2.556 \times 10^4 \text{ Ns/m}$, $p = 1.5$, $\mu = 1.595 \times 10^6$, $v_i = 2.73 \text{ m/s}$
(1) $k_3 = 2.500 \times 10^7 \text{ N/m}$, (2) $k_3 = 2.500 \times 10^8 \text{ N/m}$, (3) $k_3 = 2.500 \times 10^9 \text{ N/m}$,
(4) $k_3 = 2.500 \times 10^{10} \text{ N/m}$, (5) $k_3 = 2.500 \times 10^{11} \text{ N/m}$

- (a) Contact force versus time
- (b) Displacement of m_2 versus time
- (c) The force exerted on the loading piston versus time
- (d) The pressure in the oil chamber versus time

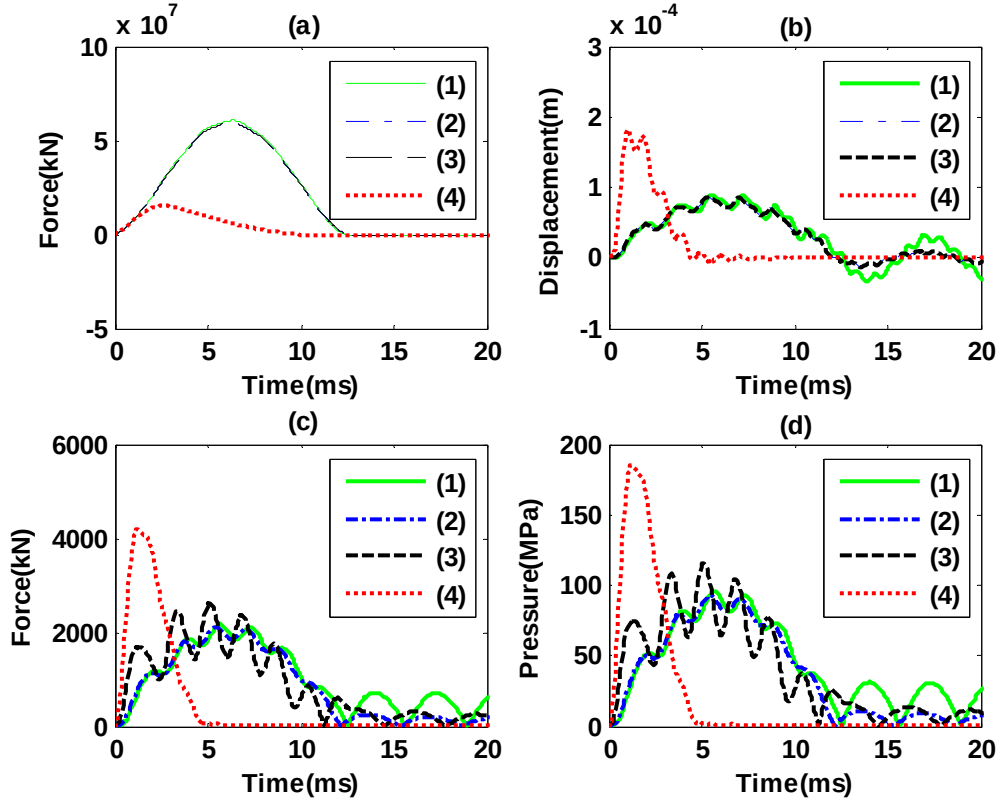


Fig. 17 Effects of the variation of damping coefficient c_2 on the impact characteristics during impact. Parameters in simulation are as follows: $k_1 = 5.958 \times 10^{10} \text{ N/m}^{1.5}$, $k_2 = 2.228 \times 10^8 \text{ N/m}$, $k_3 = 2.500 \times 10^{10} \text{ N/m}$, $p = 1.5$, $\mu = 1.595 \times 10^6$, $v_i = 2.73 \text{ m/s}$

(1) $c_2 = 2.556 \times 10^3 \text{ Ns/m}$, (2) $c_2 = 2.556 \times 10^4 \text{ Ns/m}$, (3) $c_2 = 2.556 \times 10^5 \text{ Ns/m}$,
(4) $c_2 = 2.556 \times 10^6 \text{ Ns/m}$

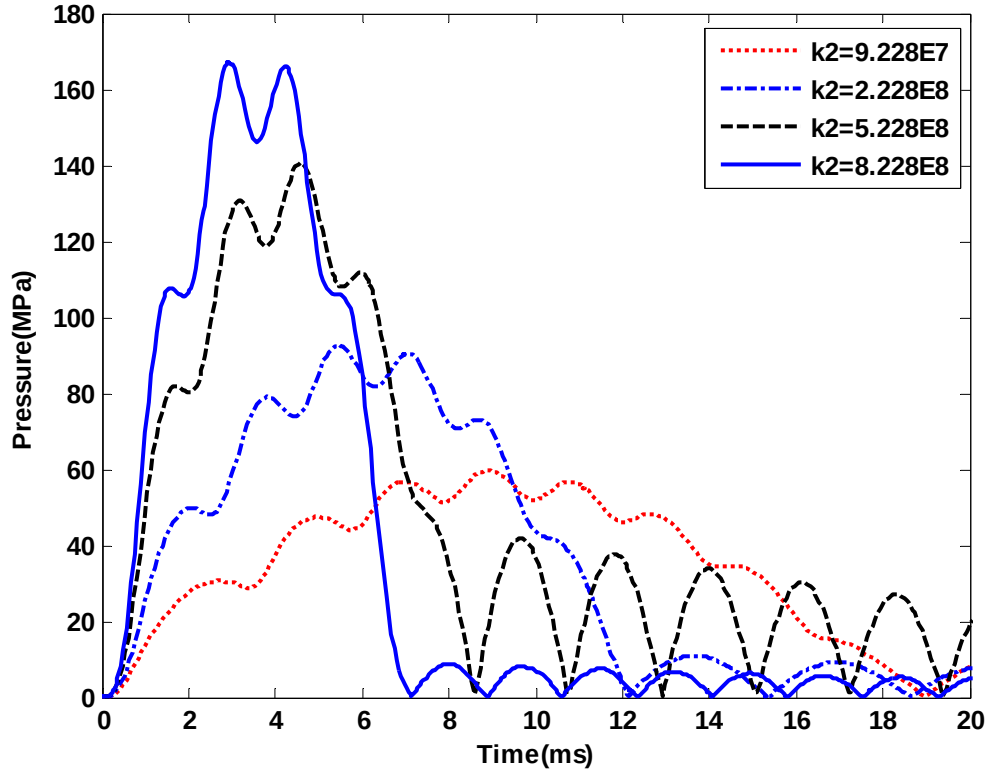


Fig. 18 Pressure time histories exerted on the stud tube or the loading piston through the oil in the chamber during impact with $v_i=2.73 \text{ m/s}$ according to the variation of k_2 value. Parameters in simulation are as follows: $k_1 = 5.958 \times 10^{10} \text{ N/m}^{1.5}$, $k_3 = 2.500 \times 10^{10} \text{ N/m}$, $c_2 = 2.556 \times 10^4 \text{ Ns/m}$, $p = 1.5$, $\mu = 1.595 \times 10^6$, $v_i = 2.73 \text{ m/s}$

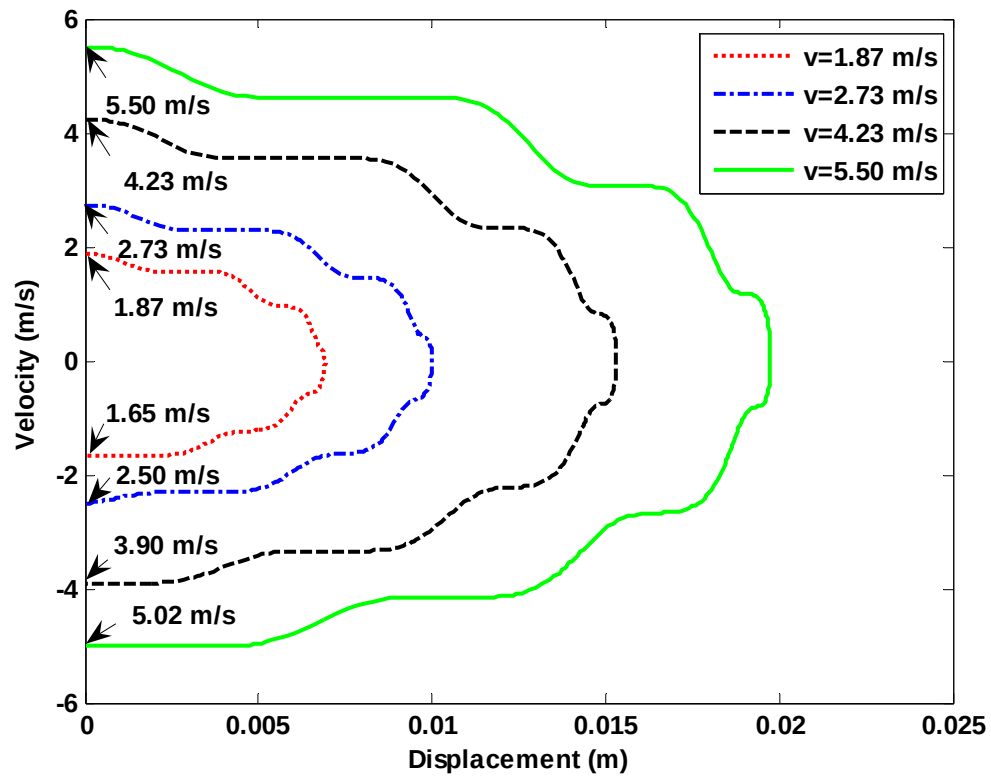


Fig. 19 Phase-space plots according to impact velocities on $(\delta, \dot{\delta})$ plane with nonlinear stiffness and nonlinear damping

5. Conclusions

In this study, the impact behavior of the impact pile driver breach fatigue system was investigated using a lumped parameter approach with a simple nonlinear spring-mass-damper model. The model includes the contact model of the hammer, the influence of energy losses on the contact area, and the effects of stiffness and damping parameters by introducing nonlinear stiffness and damping parameters.

From the results of this study the following conclusion may be drawn.

1. Current simulation model by a lumped parameter method using MATLAB is valid in order to obtain the effects of parameters as well as the contact force histories of pile drive breach fatigue system during impact.

2. Good agreement between the experiment and simulation result was confirmed. The accuracy of current impact simulation model using nonlinear contact force model is almost dependent on the stiffness of contact area, frictional damping coefficient, and the stiffness of oil chamber.

6. Future Works

Further research efforts are required in order to obtain more detailed information of the interactions between the contact mechanism and fluid or oil mechanism:

1. Parameters such as the nonlinear stiffness, k_h , exponent, p , and relative displacement, δ , have to be obtained by the experiment rather than by the calculation.

2. Current model needs to be refined using MATLAB-SIMULINK, which has modular concept of Simulink® environment,

ACKNOWLEDGEMENT

This work was supported by the ARDEC International Office through the efforts of Dr. Jerome Rubin and Lu Ting, who arranged Dr. Cho's assignment at Benét Labs. The Republic of Korea's Agency for Defense Development generously funded Dr. Cho's labor throughout this endeavor. Diana Arduini of the Watervliet Arsenal, Kevin Perry of Picatinny Arsenal, and Joe Driscoll of Benét's Automation Management Branch are acknowledged for their efforts to provide a user-friendly working environment for Dr. Cho while meeting stringent operational security requirements. The entire Modeling and Simulation Branch is appreciated for receiving Dr. Cho and educating him on experimental and numerical simulation issues related to his work. Supervisory and consulting support throughout his assignment at Benét Labs was provided by Dr. Eric Kathe and Robert W. Berggren.

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- [12] Robert W. Berggren, "BFSIM Analysis and Validation," ARDEC Modeling and Simulation Forum, October, 2006



A D D



ARDEC (Benet Labs)

A Study of Dynamic Impact Models for Pile-Driver Breech Fatigue System

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Agency for Defense Development, Korea

Exchange Engineer(2006-2007), Benet Labs, ARDEC, U.S. Army

Dr. Eric Kathe

Supervisor, Benet Labs, WS&T, AETC, ARDEC, U.S. Army

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 - Related Research Works
 - Impact Pile Driver Fatigue System
 - A Continuous Contact Force Model
 - A Mathematical Model
 - The Governing Equation
 - Model Simulation and Validation
 - Discussion
 - Summary
- ☐ **Experiences**

I (Chang-ki Cho)



I. Personal Data

- Title : Principal researcher
- Date of Birth : [REDACTED]
- Scientific or Technical Specialty:
 - Dynamic analysis of automatic gun for AIFV or turreted gun for attack helicopter (Medium caliber weapon)
- Marital Status : [REDACTED]

II. Education

- B.S.(1980), M.S.(1991), Ph.D(1998)

III. Professional Employment

- ADD : 1979.12 ~ Present

IV. Motivation as a Exchange engineer for ESEP

- To learn about more advanced concepts and technologies in the advanced research institute such as ARDEC (Benet Labs)
 - “ better, faster and cheaper” R&D process ?

My Family



Description of Duties

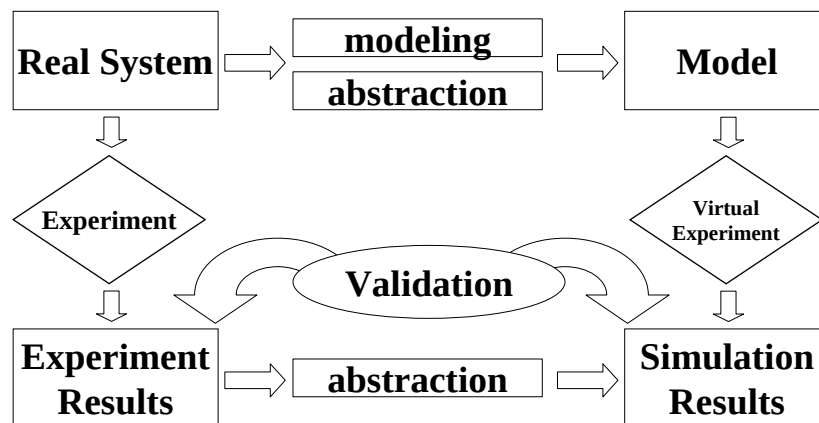
- ❑ Conduct R&D work in the area of shock-isolated horizontal breech fatigue actuator to replace current vertical pneumatic pile driver based system

- ❑ Conduct R&D work in the area of automated loading of a vented erosion simulator.
Perform mechanism and cartridge concept refinement as required

What can I do for it ? How can I do for it?

What results can I get from it?

Modeling and Simulation Concept



A real system exists and operates in time and space

A simplified representation of a system at some particular point in time or space

The manipulation of a model in such a way that it operates on time or space to compress it, thus enabling one to perceive the interactions that would not otherwise be apparent because of their separation in time or space

A Study of Dynamic Impact Models for Pile-Driver Breech Fatigue System

Objective

- ❑ To investigate the fundamental dynamic characteristics of current pile driver fatigue system**
 - Building of the lumped parameter model to replicate or mimic the high pressure data**
 - Program using Matlab SW tool**
 - Validation of model**
 - Discuss the effects of the variation of key parameters on the model**

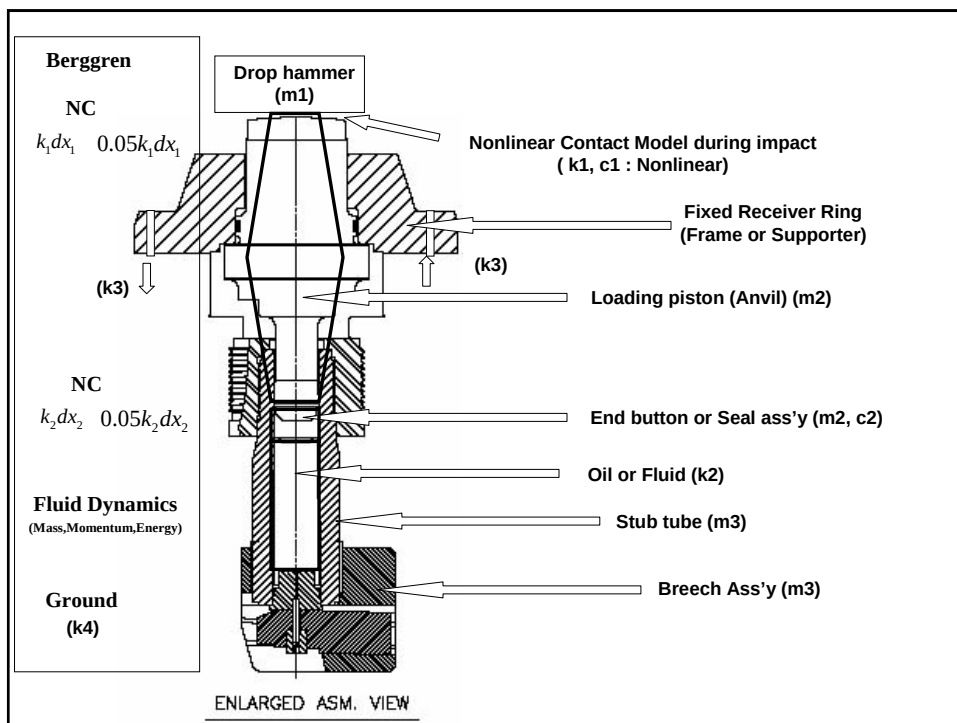
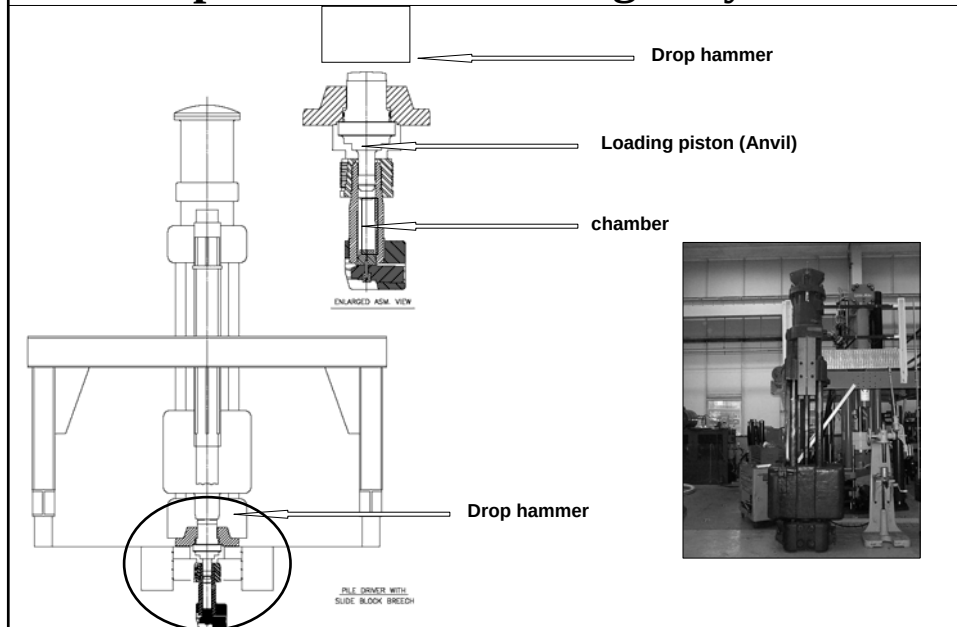
Related Research Works(1)

- ❑ K.H. Hunt et al. : “Coefficient of Restitution Interpreted as Damping in Vibroimpact,” J of Applied Mechanics (1975)
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- ❑ A. L. Schwab et al. : “A Comparison of revolute joint clearance models in the dynamic analysis of rigid and elastic mechanical systems,” Mechanism and machine Theory (2002)
- ❑ Zhang, X et al. : “Modeling the Dependence of the Coefficient of Restitution on the Impact Velocity in Elasto-Plastic Collisions,” J of Impact Engineering, (2002)

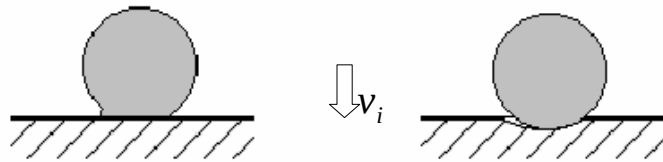
Related Research Works(2)

- ❑ Anping Guo et al. : “Substructure Analysis of a Flexible System Contact-Impact Event ,” J of Vibration and Acoustics (2004)
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- ❑ Eric Kathe et al. : “Using a six Ton Pile Driver to Mimic Firing Pressures of Large Caliber Cannon-Softly ,” the 76th Shock & Vibration Symposium (2005)
- ❑ Bob Berggren : “BFSIM Analysis and Validation,” ARDEC Modeling and Simulation Forum (2006)

Impact Pile Driver Fatigue System



A Continuous Contact Force Model(1)

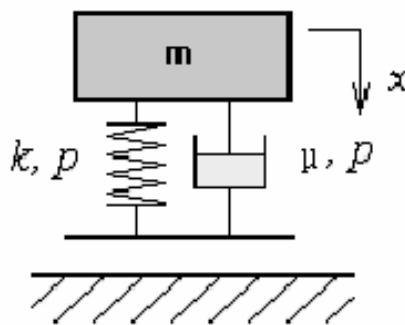


(a) Soft impactor, hard surface (b) Hard impactor soft surface

Extreme scenarios of relative hardness

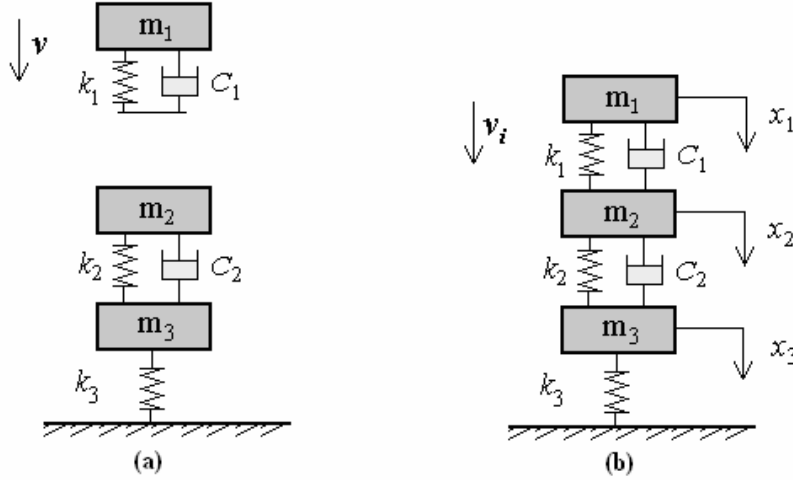
(Hertz's contact theory)

A Continuous Contact Force Model(2)



Mathematical model of 1 d.o.f. impact system for the contact mechanism of impact system (soft impactor, hard surface)

A Mathematical Model



(a) Before impact

(b) During impact

A lumped-parameter model of the pile driver breach fatigue system

The Governing Equation

$$m_1 \ddot{x}_1 = -\rho k_1 (x_1 - x_2)^p - \mu (x_1 - x_2)^p (\dot{x}_1 - \dot{x}_2)$$

$$m_2 \ddot{x}_2 = +\rho k_1 (x_1 - x_2)^p + \mu (x_1 - x_2)^p (\dot{x}_1 - \dot{x}_2)$$

$$-k_2 (x_2 - x_3) - c_2 (\dot{x}_2 - \dot{x}_3)$$

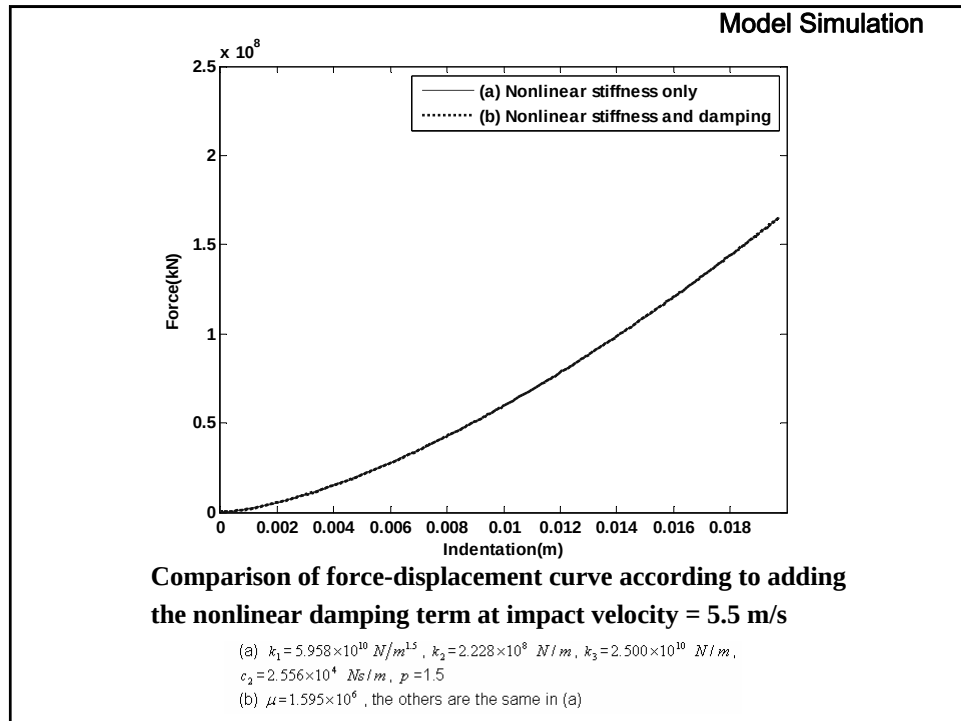
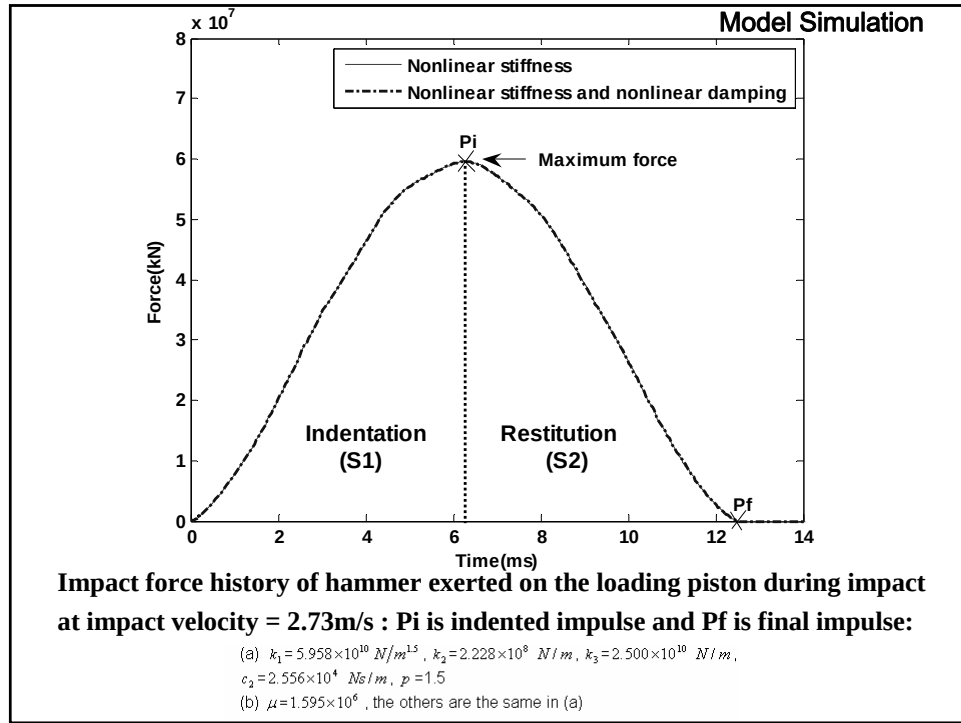
$$m_3 \ddot{x}_3 = +k_2 (x_2 - x_3) + c_2 (\dot{x}_2 - \dot{x}_3) - k_3 (x_3)$$

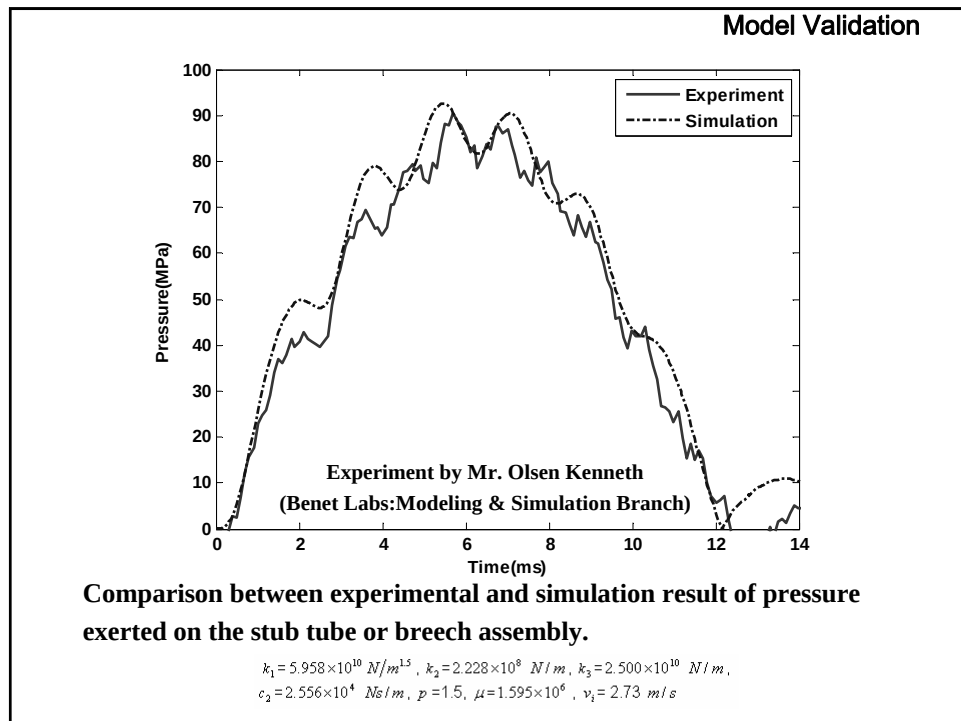
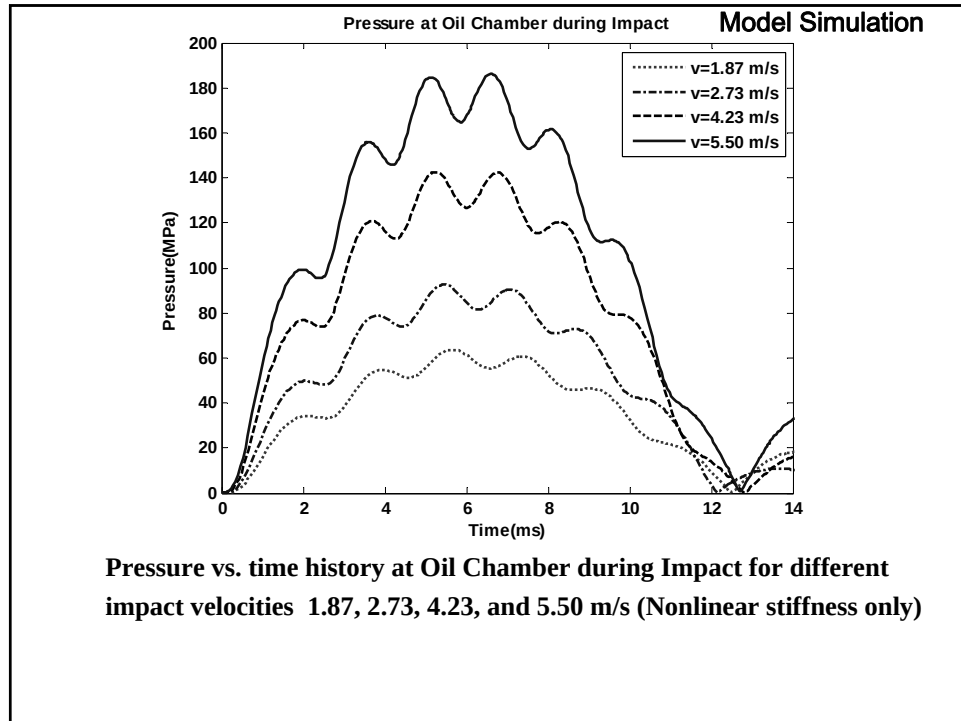
$$\rho = 1, \text{ for } (x_1 - x_2) > 0$$

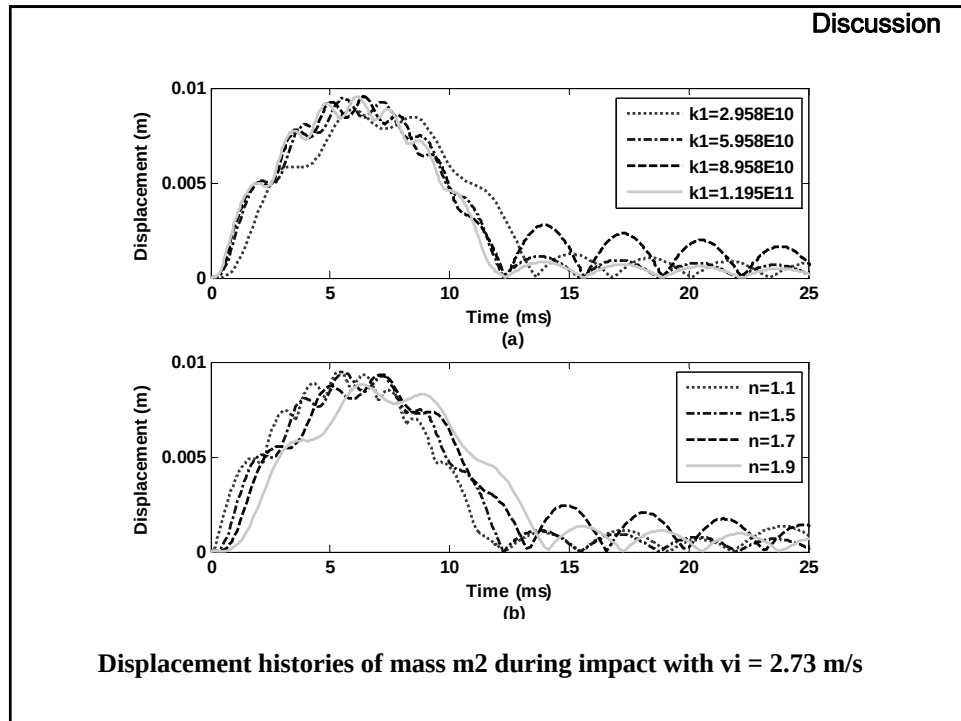
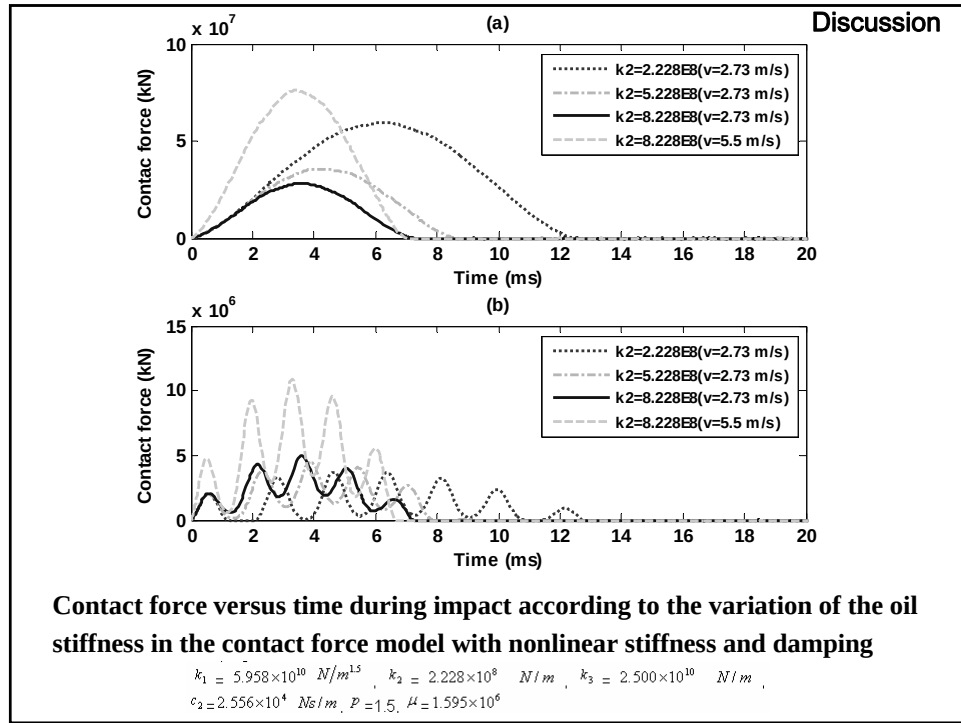
$$\rho = 0, \text{ otherwise}$$

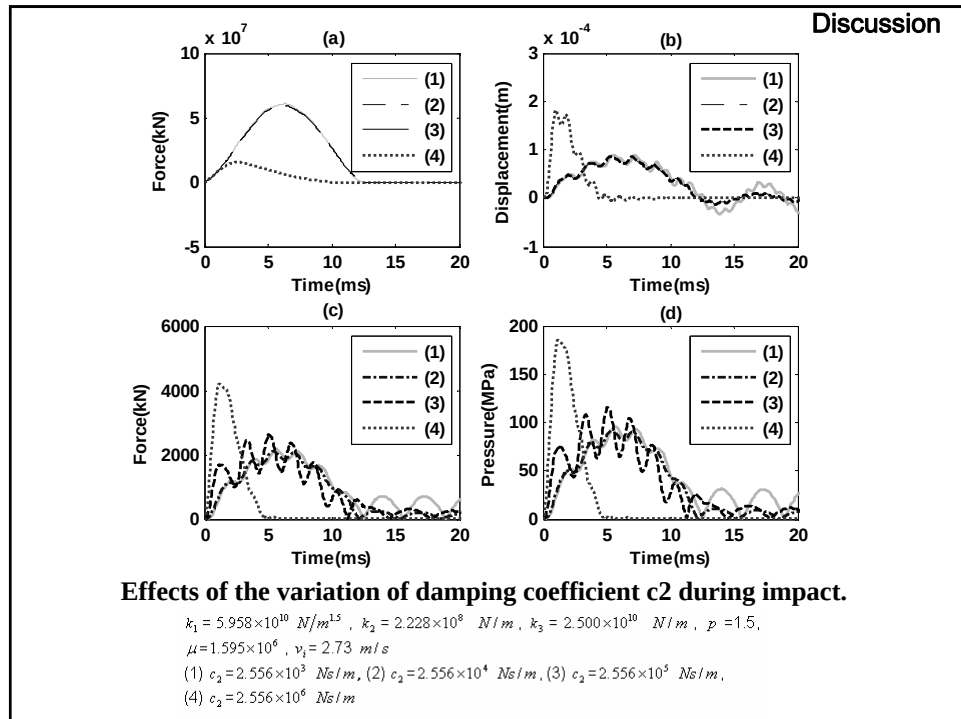
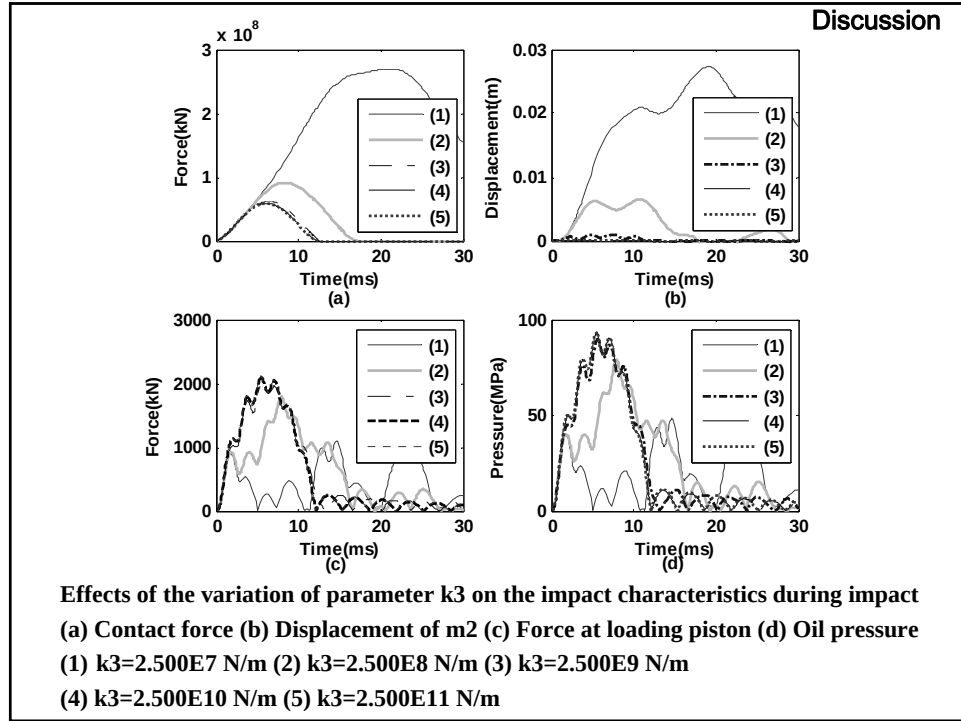
$$I.C. : x_1(0) = x_2(0) = x_3(0) = 0$$

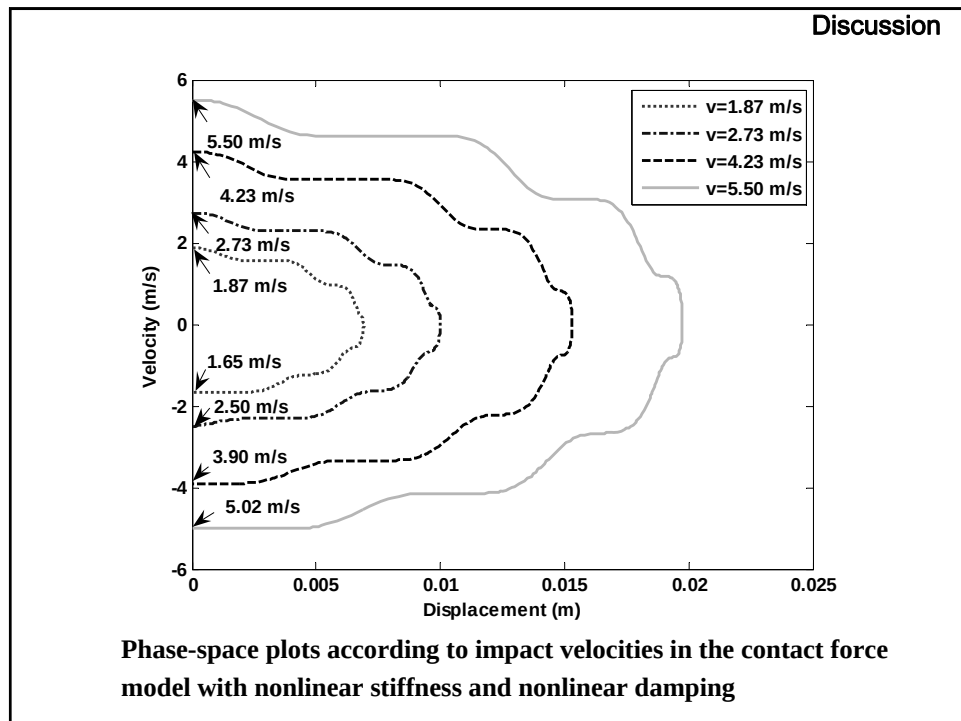
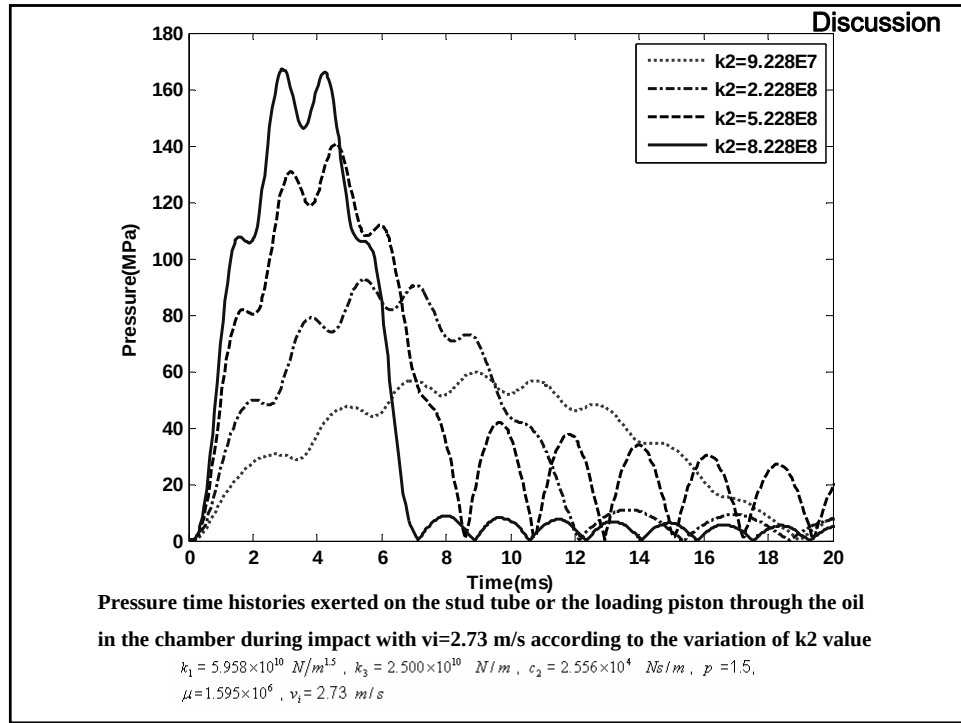
$$\dot{x}_1(0) = v_i, \quad \dot{x}_2(0) = \dot{x}_3(0) = 0$$











Summary

- Impact behavior of the pile driver breach fatigue system
 - Simulation model by a lumped parameter method using Matlab programming
 - The accuracy of current model using nonlinear contact force model (stiffness of contact area, stiffness of oil)
- Further research efforts
 - Tuning of input parameters by experimental values
 - Refined using Matlab-Simulink, modular concept of Simulink® environment(contact mechanism, oil mechanism)

Experiences

- ✓ Understanding the Modeling and Simulation
 - Many Reference Papers
- ✓ Excursions
 - NYC, Niagara Falls, Orlando & Miami, Boston, Philadelphia, Washington D.C., Las Vegas, Some cities of Canada
- ✓ Language Ability
 - Voices (Albany, every Mon. & Wed. after work)
- ✓ Understanding American Culture
 - Horizons
 - Gathering, Visiting Home, Hiking
 - Learning the Bible
- ✓ Strengthen my religious faith
 - Catholic Church of Albany Korean Community
 - American Catholic Church

Thank you for your attention.

I love Benet Labs (ARDEC)