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# Destructive Disassembly of Bolts and Screws by Impact Fracture

**Kyung Geun Pak** and **Raj Sodhi,** Mechanical Engineering Dept., New Jersey Institute of Technology, Newark, New Jersey, USA. *E-mail:* Sodhi@admin.njit.edu

## Abstract

Disassembly, the process of removing components from a product at the end of its useful life, is complex due to a variety of fastener shapes and variability in their damage during use. Mechanical impact has been suggested as a cost-effective method for destructive disassembly of joining elements. This research analyzes the process of impact disassembly by studying the characteristics of elastic waves caused by the impact. Elastic waves are modeled in a one-dimensional bar, which transfers the impact energy to a protruded bolt head mounted in an infinite elastic medium or structure. Stress wave equations are presented for each period when they bounce back and forth between the two ends of the bar. It is determined that the maximum stress occurs when the wave front reflects in the second or later period. This higher stress can be applied to shear off the bolt head with the same amount of energy invested as in a single wave impact. The results can be used to design effective destructive disassembly procedures and new demanufacturing tools, resulting in an increase of disassembly efficiency and a reduction of recycling cost.

***Keywords:*** *Destructive Disassembly, Impact Fracture, Stress Wave, Demanufacturing*

## Introduction

Unlike the assembly process, the disassembly process of products at the end of a life cycle is generally manual and involves complex problems. The process is especially difficult when dealing with a large variety of products and uncertainty in product condition or damage after use. The aim of the disassembly process is cost minimization, hazardous materials isolation, and opportunities to reuse or reutilize materials and components. Therefore, an appropriate disassembly technology must combine flexibility and robustness to be able to deal with these issues. The disassembly process involves three activities: (1) unfastening to separate the components, (2) destructive disassembly where either the fastener and/or the component may be destroyed, and (3) partly destructive or semidestructive disassembly where the fastener can be destroyed during the disassembly process with no damage to the components. The efficiency of the disassembly process employing destructive means such as cutting, sawing, or breaking the fasteners is presently not well understood. Because efficiency in recycling is an important issue, there have been some recent researches on destructive disassembly techniques. In general, a product is a combination of parts or subassemblies assembled by joining elements. The determination of effort needed to disassemble these joining elements has become a vital issue in demanufacturing research. Most disassembly-line situations encounter a wide range of fastener geometry and variable fastener damage during use. Hence, it is advantageous to destroy the fastener element to achieve economic viability of the process. In this study, the focus is on the destructive approach and on developing a robust method for breaking protruded heads of joining elements such as screws and bolts by applying side impact. Specifically, analytical equations are derived representing the impact stress in a transmitter bar and the shear stress in the bolt head mounted on an elastic medium. The efficiency of the developed method is evaluated in comparison with the conventional method involving a single stress wave impact. This research should lead to an accurate determination of destructive disassembly effort for commonly used fasteners and subsequently assist in developing an efficient destructive disassembly procedure.

## Literature Review

Researches in the disassembly area are relatively recent activities. Recently, researchers have begun to address and devise general and standard solutions for many aspects of product disassembly. Most of the researches related to this issue involve models to compute the disassembly effort and to facilitate the economic analysis of the disassembly activities. In an effort to standardize disassembly operation times, Dowie and Kelly (1994) conducted a series of disassembly experiments with simple operations, including destructive operations. Kroll (1996) and Kroll, Beardsley, and Parulian (1996) developed a method for estimating the ease of disassembly using work measurement analysis. Hanft and Kroll (1995) proposed a metric for evaluating disassembly activities such as drilling, sawing, and grinding and proposed a metric for evaluation. There have been many researches on design for disassembly (DFD), which involves the development of products that are easy to take apart to enable recycling. Research related to DFD has increased in recent years, but a major thrust of the work on disassembly has been focused on disassembly sequencing, disassembly path planning, and the evaluation tool development. Gupta and McLean (1996) and Penev and Ron (1996) have provided an overview of the ongoing research in disassembly and the trends for future activities. Hrinyak, Bras, and Hoffman (1996) have examined the existing disassembly software tools presently available to designers. Recently, Shyamsunder, Ashai, and Ghad (1996) have initiated work to build a three-dimensional virtual disassembly tool. Although the overall economics of the disassembly process is still not well understood, the destructive disassembly approach is adopted in many recycling processes for fast and efficient separation of products (Henstock 1988; Chen, Navin-Chandra, Prinz 1993; Kirby and Wandehra 1993). Feldmann and Meedt (1995) adopted the use of destructive or partially destructive processes for efficient and flexible disassembly of electronic devices. They presented methods for the planning of optimal disassembly and the calculation of the most economic disassembly depth and reported the need for highly flexible tools designed for disassembly operations. They proposed devices (drill-driver and drill-gripper) that combine transmission of torques or forces with drilling for unscrewing various types of screwed fasteners and for increasing transmittable forces. In contrast to the literature mentioned above, Studny, Rittel, and Zussman (1999) proposed to apply the destructive disassembly procedure based on impact mechanics and provided rigorous investigation on the mechanical aspect of the disassembly process. In their research, they applied lateral impact on the head of a joining element to cause shear-fracture on the neck area. They adopted an "instrumented bar" (Kolsky 1963) between the striker and screw head to measure and apply impact load (see *Figure 1*). By comparing the pulses before and after the reflection at the screw head, Studny, Rittel, and Zussman evaluated the energy invested in fracturing the screw head. Having this instrumented bar between the screw and the striker is advantageous in practical design application because it ensures a predictable contact surface to transmit stress waves and to position where the impact is applied.

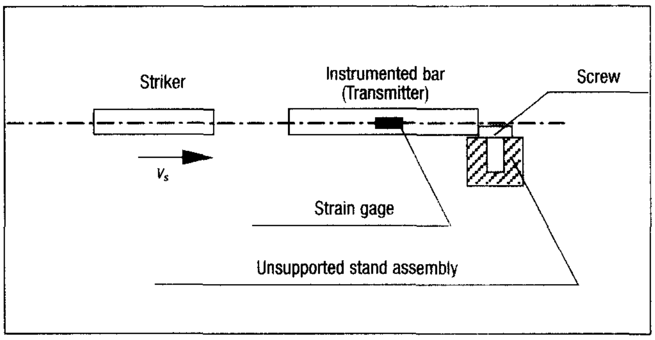


***Figure 1***
**Side Impact for Destructive Disassembly**

The striker launched by an air gun hits one end of the instrumented bar and generates a stress wave. This stress wave travels toward the other end, where the instrumented bar and the screw head are contacting each other. At the contacting surface, the stress wave reflects after losing some of its energy transmitted to the screw and travels back toward the struck end. The energy transmitted to the screw head causes fracture in the neck area of the screw if it generates higher stress than the dynamic strength of the bolt neck. With the strain gage attached in the instrumented bar, Studny, Rittel, and Zussman measured the stress wave before and after the reflection and calculated the force and energy invested into fracture. They showed the feasibility of using dynamic impact to fracture screwed fasteners and integrated their results into the preliminary design of a robotic disassembler. However, in addition to their rigorous study on the fracture energy and quantitative test results, more investigation is needed in the theoretical aspect of impact to improve the efficiency of the process. This paper studies impact mechanisms between the parameters of the striker, transmitter (instrumented bar), and so on, and suggests a method that will improve the efficiency of the impact-destructive disassembly process.

The impact-destructive method is categorized according to the number of reflections of the stress wave at the bolt-contacting surface. If the method is designed to create maximum stress after the wave reflects at the bolt-contacting surface several times, it is called the Multiple Reflection of Stress Wave (MRSW) Method. If the maximum stress occurs only at the first reflection at the bolt-contacting surface, and this should cause fracture at the bolt, the method is called the Single Reflection of Stress Wave (SRSW) Method.

## SRSW Method

The experiment conducted by Studny, Rittel, and Zussman (1999) falls into the Single Reflection of Stress Wave (SRSW) case. In their setup, the material and diameter of the striker are identical to those of the transmitter (instrumented bar), but the length of the striker is set relatively shorter than that of the transmitter. Therefore, no matter how fast the striker hits, the pulse duration ends before the front end of the reflected wave reaches back to the struck end of the transmitter. The pulse duration varies not with the speed but with the length of the striker, and the magnitude of the stress depends only on the speed of the striker. Therefore, the maximum stress occurs at the front of the wave and this should cause the fracture on the bolt neck when it first reflects at the bolt-contacting surface. In this case, the maximum stress in the transmitter is given by the following:

$$\sigma = \frac{1}{2}\rho c_L v_s \tag{1}$$

where

$\sigma$: stress in transmitter

ρ: material density of transmitter

$c_L$: longitudinal wave velocity in transmitter

$v_s$: speed of striker

If the cross section area of the striker is much larger than that of the transmitter bar, and the material of both the transmitter and the striker is identical, the stress in the transmitter bar can be written as follows (Johnson 1972, Spotts 1964):

$$\sigma = \rho c_L v_s \cong \rho c_L v_s \frac{1}{\left(1 + \frac{A_t \rho_t c_t}{A_s \rho_s c_s}\right)} \tag{2}$$

where

$A_t, A_s$: section area of transmitter and striker, respectively (in this case, $A_t << A_s$)

$c_t, c_s$: longitudinal wave velocity in transmitter and striker, respectively

$\rho_t, \rho_s$: material density of transmitter and striker, respectively.

If the bolt is mounted on a rigid body, the shear stress on the bolt neck is as follows:

$$\tau_b = 2\frac{A_t}{A_b}\sigma \tag{3}$$

where $A_b$ is the section area of the bolt neck.

The force that will cause fracture of the bolt head can be calculated from the stress equations given above. In both cases, the maximum stress occurs when the stress wave reflects at the bolt-contacting end in the first period, and this maximum stress should be higher than the dynamic shear strength of the bolt head to cause fracture.

## MRSW Method

If the transmitter in *Figure 1* has much smaller mass than that of the striker (shorter length and smaller diameter), the wave front reflects several times at the bolt-contacting surface until the kinetic energy of the striker ceases. The maximum stress occurs when the wave front reflects in the second or later period. The maximum stress in this case is higher than in the SRSW method even if the impact was generated by the same striker with the same speed. Therefore, higher stress can be applied to the bolt head with the same amount of energy invested on launching the striker if the mass and the length of the transmitter are reduced. This method requires the condition that the body on which the bolt is mounted has enough inertia so that it supports the bolt until the maximum stress occurs without any fixture. If the body has not much inertia to withstand the transmitted force form the transmitter, a fixture would be needed. Assuming the bolt is mounted on a rigid body, the problem can be solved as "a bar with a fixed end struck by a moving mass at the other end" introduced in many literatures (Kolsky 1963, Timoshenko and Goodier 1970, Johnson

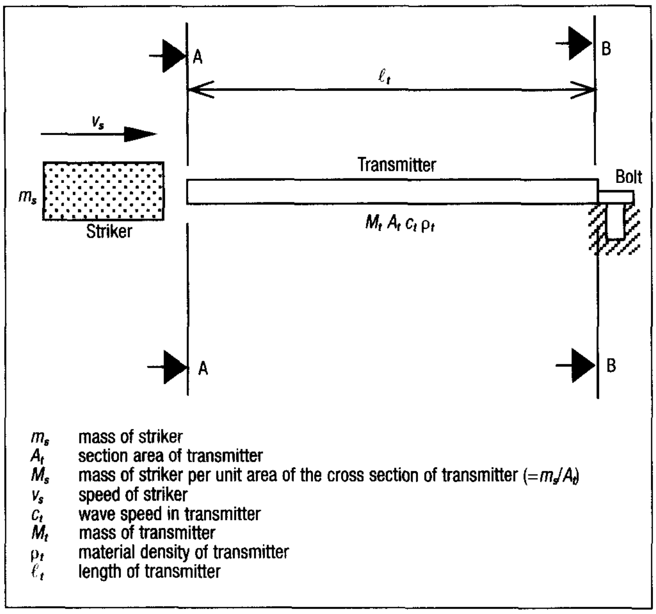


***Figure 2***
**Schematic of Applying Impact to Fracture Bolt Head**

1972, Spotts 1964). However, with a real situation, the bolt-supporting body would be nonrigid. In other words, when the stress wave reflects at the bolt-contacting surface, the magnitude of the reflected wave will be reduced, and this will affect the forming of the wave equation of the second period, and so on. Therefore, new equations are required for the stress wave forming after the first reflection.

## Stress Wave Equation When Bolt is Mounted on a Rigid Body

If the bolt is mounted on a rigid body, it can simply be analyzed as a bar with a fixed end struck by a rigid mass (Johnson 1972, Spotts 1964). With one-dimensional stress wave theory, the relationship of the fracture of the bolt is investigated with the geometrical and material characteristics of the instruments, including striker and transmitter. Instrument setup is similar to the one shown in *Figure 1* except that the relative size of the striker is much larger than that of the transmitter, as shown in *Figure 2*.

When a rigid striker of mass $m_s$ impinges with speed $v_s$ on one end of transmitter, the stress $\sigma_0$ at the head of the stress wave is given by the following equation:

$$\sigma_0 \cong \rho_t c_t v_s \tag{4}$$

According to the analysis by Johnson (1972) and Spotts (1964), the stress wave generated by the impact at the struck end of the transmitter during time $0 < t < T$ can be derived from the equation of motion for $M_s$ as follows:

$$M_s \frac{dv_1}{dt} = -A_t \sigma_1 \tag{5}$$

where $\sigma_1$ is the stress intensity at the interface A-A at time $t$ after first contact and $v_1$ is the particle speed of the transmitter at the interface A-A. Solving the above equation, the stress wave in the transmitter during $0 < t < T$ is as follows:

$$\sigma_1(t) = \sigma_0 e^{-\frac{2\alpha t}{T}} \tag{6}$$

where $\alpha = M_t / M_s$ and $T = 2\ell_t / c_t$.

This compressive stress wave propagates from the struck end to the contacting surface B-B. The stress at B-B is zero until $t = \ell_t / c_t$, and it suddenly becomes $2\sigma_1$ right after the front end of the wave is reflected at this surface. This means that the stress exerted on the bolt head is twice the stress on the struck end.

The head of the reflected wave travels back to the common contacting surface A-A and generates more stress between the striker and the transmitter from $t = 2\ell_t / c_t$, and again reflects toward the contact surface B-B. Therefore, the striker faces more resistance after $t = 2\ell_t / c_t$, and because this affects the striker initiating stress, the stress wave equation for $T < t < 2T$ needs to be redefined. When the head of the reflected wave is being reflected at the struck end A-A during this period, it provides a compressive stress of $2\sigma_1$. Then the total stress $\Sigma\sigma$ at the struck end during this period will be as follows:

$$\Sigma\sigma = 2\sigma_1(t-T) + \sigma_2(t) \tag{7}$$

where $\sigma_2(t)$ is the stress wave equation during $T_t < t < 2T_t$ and is to be determined. Again, the motion equation during this period is written as follows:

$$M_s A_t \frac{dv_2}{dt} + A_t \Sigma\sigma = 0 \tag{8}$$

Solving Eq. (8) gives the following:

$$\sigma_2(t) = \sigma_0 e^{-\frac{2\alpha t}{T}} \left[ 1 + 4\alpha e^{2\alpha} \left( 1 - \frac{t}{T} \right) \right] \tag{9}$$

This process repeats for $\sigma_3(t)$, $\sigma_4(t)$, and for longer intervals, but the equations become very lengthy.

From the equations for $\sigma_1(t)$, $\sigma_2(t)$, $\sigma_3(t)$, and $\sigma_4(t)$, and so on, the stresses at the struck end and fixed end can be calculated. It can be shown that the plots of the empirical equations give very good approximation to the theoretical curves (Spotts 1964). Therefore, to calculate the shear stress on the bolt neck mounted on a rigid body, the above equations are used and converted to the expression for the shear stress on the bolt. Comparison of the calculated shear stress with the dynamic shear strength of the bolt can predict the fracture.

## Stress Wave Equation When Bolt is Mounted on an Elastic Body

In most real situations, the material on which the bolt is mounted is not rigid. Thus, some of the impact energy is transmitted and lost into the material. Therefore, when the stress wave is being reflected at the contact surface B-B, as in *Figure 3*, the magnitude of the reflected stress becomes smaller than that of the stress before the reflection. Usually in a high-velocity impact analysis, overall structural behavior can be ignored because deformation is concentrated at the impact point. However, in the case of a relatively low velocity of the striker or a relatively low density of the material on which the bolt is mounted, loss of impact energy dissipated into the material needs to be considered (Macaulay 1987). Wada (1984) has shown that the reflection characteristics of longitudinal waves in a semi-infinite cylindrical rod change with the thickness of the elastic plate to which the rod is connected. The reflected stress wave changed from tensile to compressive as the plate thickness is increased. Therefore, depending on the characteristics of the material or the structure on which the bolt is mounted, the reflected stress wave at the bolt-contacting surface (B-B) would have different magnitudes. A schematic diagram of the stress wave situation is shown in *Figure 3*. The stress wave equation at the struck end for $0 < t < T$ is identical to Eq. (6) because the front of the reflected stress wave does not return to the struck end in this period and therefore does not affect the equation. However, for the second period of $T < t < 2T$, the wave equation for this period will be different from Eq. (9) because the returning wave that was reflected at B-B has been reduced in its magnitude. Therefore, the striker faces less resistance during this time interval than in the rigidly fixed-end case.

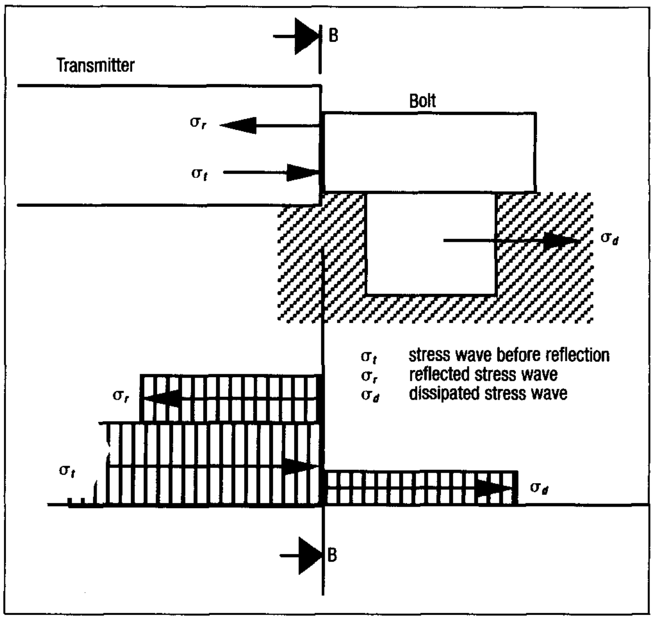


***Figure 3***
**Reflection and Dissipation of Compressive Stress Wave with Elastic Medium**

Let us assume that there exists a ratio $R$ ($-1 \le R \le 1$) between the stress before reflection and after reflection. It is reasonable to assume that the elastic wave reflects with a constant ratio $R$ if the transmitted elastic load remains in the elastic boundary of the supporting medium or structure. Then the reflected stress wave $\sigma_r$ and the transmitted stress wave before reflection can be related as follows:

$$\sigma_r = R\sigma_t \tag{10}$$

Following the procedure described in the literature (Timoshenko and Goodier 1970, Johnson 1972, Spotts 1964), the equations for stress waves in second and subsequent periods can be derived with $R$.

As in the fixed-end case, when the front end of this reflected stress wave returns to the struck end of the transmitter, it affects the stress wave equation for $T < t < 2T$. Then the total stress $\Sigma\sigma$ at the struck end during this period will be as follows:

$$\Sigma\sigma = 2R\sigma_1(t-T) + \sigma_2(t) \tag{11}$$

where $\sigma_2(t)$ is the stress wave equation during $T < t < 2T$ and is to be determined. The force equilibrium during this period is as follows:

$$M_s A_t \frac{dv_2}{dt} + A_t \Sigma\sigma = 0 \tag{12}$$

Solving the equation for the second period of $T$, the stress $\sigma_2(t)$ is as follows:

$$\sigma_2(t) = \sigma_0 e^{-\frac{2\alpha t}{T}}\left[1 + 4R\alpha e^{2\alpha}\left(1 - \frac{t}{T}\right)\right] \tag{13}$$

This process repeats for the stress wave equations of $\sigma_3(t)$, $\sigma_4(t)$, and so on. Then the solutions for $\sigma_3(t)$ and $\sigma_4(t)$ are obtained, as follows:

$$\sigma_3(t) = \sigma_0 e^{-\frac{2\alpha t}{T}}\left(A_3 t^2 + B_3 t + C_3\right) \tag{14}$$

where

$$A_3 = \frac{8\alpha^2 R^2}{T^2} e^{4\alpha}$$

$$B_3 = -\frac{4\alpha R}{T}\left\{e^{2\alpha} + R(1+8\alpha)e^{4\alpha}\right\}$$

$$C_3 = 8R^2\alpha(1+4\alpha)e^{4\alpha} + 4R\alpha e^{2\alpha} + 1$$

and

$$\alpha_4(t) = \alpha_0 e^{-\frac{2\alpha t}{T}}\left(A_4 t^3 + B_4 t^2 + C_4 t + D_4\right) \tag{15}$$

where

$$A_4 = -\frac{32\alpha^3 R^3}{3T^3} e^{6\alpha}$$

$$B_4 = \frac{8\alpha^2 R^2}{T^2}\left\{R(1+12\alpha)e^{6\alpha} + e^{6\alpha} + e^{4\alpha}\right\}$$

$$C_4 = -4R\alpha\left\{\begin{matrix}\left(12\alpha + 72\alpha^2\right)R^2 e^{6\alpha} + 12R\alpha e^{6\alpha} \\ +8R\alpha e^{4\alpha} + e^{6\alpha} + e^{4\alpha} + e^{2\alpha}\end{matrix}\right\}$$

$$\begin{aligned} D_4 = {} & 72(4\alpha+1)R^3\alpha^3 e^{6\alpha} + 72R^2\alpha^2 e^{6\alpha} \\ & + 4(8\alpha-1)R^2\alpha e^{4\alpha} - 8R\alpha e^{2\alpha} \\ & + 12\left(e^{6\alpha} + e^{4\alpha} + e^{2\alpha}\right)R\alpha + 1 \end{aligned}$$

From Eqs. (6), (13), (14), and (15), total stress at the struck end can be calculated for different values of $R$. *Figure 4* is the plots of the stress ratio $\sigma/\sigma_0$ versus $t/T$ at the struck end for different values of $R$ when $\alpha = 1/4$. The graph shows that the stress ratio $\sigma/\sigma_0$ for $R = 1.0$ with $\alpha = 1/4$ calculat-

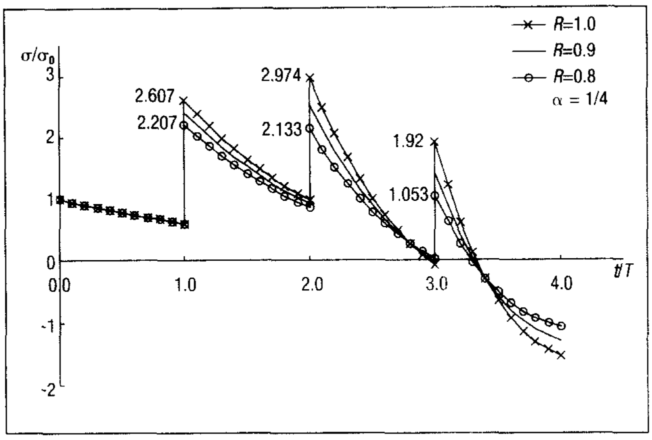


***Figure 4***
**Stress Ratio $\sigma/\sigma_0$ at Struck End for Different *R* Values**

ed from the equation is identical to the classical solution for the stress wave in a one-dimensional bar fixed to a rigid wall (Spotts 1964). In addition, it is noted that the time for maximum stress shifts from $2T$ to $T$ as $R$ decreases. On the other hand, the resultant stress $\sigma$ on the bolt-contacting end of the transmitter is of interest, and this stress will be converted as shear stress $\tau_b$ at the bolt-neck. $\sigma$ and $\tau_b$ are related as follows:

$$\tau_b = \frac{A_t}{A_b}\sigma \tag{16}$$

The shear stress $\tau_b$ should exceed the dynamic shear strength of the bolt to cause fracture. In the first period, the bolt head will receive no stress at all until $t = T/2$, the moment that the stress wave propagated from the struck end reaches it. The resultant stress then suddenly becomes $2\sigma_1(t) - \sigma_d$ and will last until $t = 3T/2$. Because $\sigma_d = \sigma_1(t) - R\sigma_1(t)$, the resultant stress can be expressed as follows:

$$\begin{aligned} & \sigma = \sigma_1(t - T/2) + R\sigma_1(t - T/2), \\ & \quad (T/2 < t < 3T/2) \end{aligned} \tag{17}$$

Similarly, the resultant stress that the bolt head is receiving from the transmitter for continuing periods can be derived, as follows:

$$\begin{aligned} & \sigma = \sigma_2(t - T/2) + R\sigma_2(t - T/2) \\ & \quad + R\sigma_1(t - 3T/2) + R^2\sigma_1(t - 3T/2), \\ & \quad (3T/2 < t < 5T/2) \end{aligned} \tag{18}$$

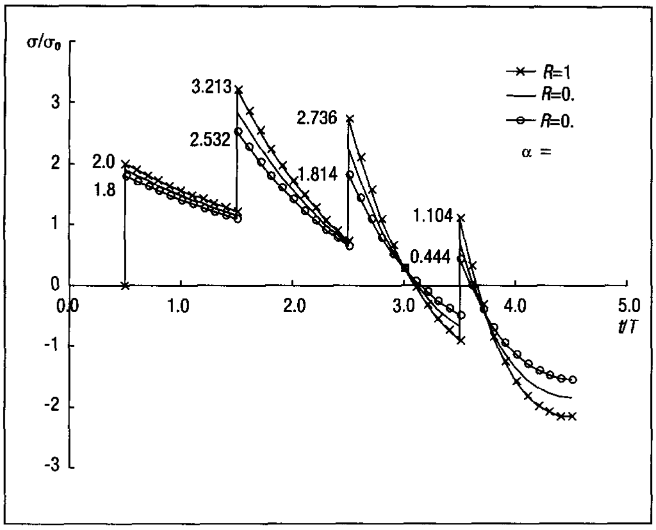


*Figure 5*
**Stress at Bolt-Contacting End for Different *R* Values (α is fixed)**

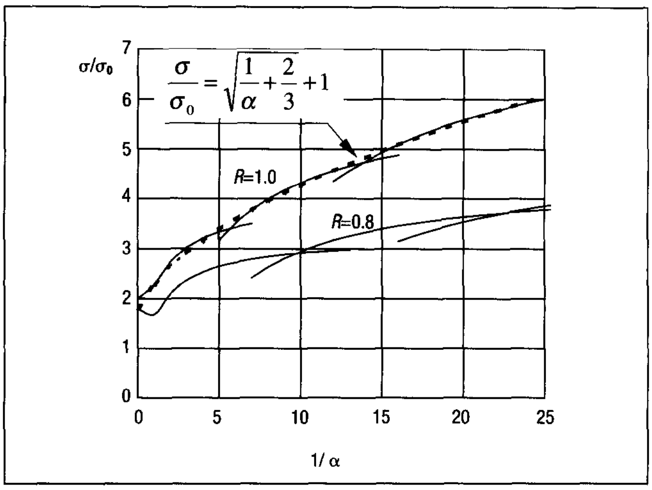


*Figure 6*
**Stress at Bolt-Contacting End for *R* = 1.0 and *R* = 0.8**

$$\begin{aligned}\sigma = {} & \sigma_3(t-T/2)+R\sigma_3(t-T/2) \\ & +R\sigma_2(t-3T/2)+R^2\sigma_2(t-3T/2) \\ & +R^2\sigma_1(t-5T/2)+R^3\sigma_1(t-5T/2), \\ & (5T/2<t<7T/2)\end{aligned} \tag{19}$$

and so on.

For the weight ratio α = 0.25, the plot of $\sigma/\sigma_0$ for different values of *R* is shown in *Figure 5*. By comparing *Figure 5* and *Figure 4*, it can be noticed that the maximum stress at the bolt-contacting end is higher than that at the struck end. By comparing the resultant stress from Eqs. (17), (18), and (19) with the dynamic strength of the bolt head, the fracture of the bolt head can be predicted.

Using Eqs. (17), (18), and (19), the ratio of maximum stress to initial stress ($\sigma/\sigma_0$) can be plotted for different values of α. The dashed curve in *Figure 6* is a plot of the empirical equation for the rigidly fixed-end case (Spotts 1964) that gives very good approximation to the theoretical solution for *R* = 1.0. The curves from Eqs. (15), (16), and (17) with *R* = 0.8 are shown also for only the portion of each curve that rises above the next lower curve. As noticed in *Figure 6*, the maximum stress decreases significantly as the reflection ratio *R* decreases.

The maximum stress shown in *Figure 5* can be compared with the maximum stress of the SRSW method. In this case, for α = 0.25, the increments of maximum stresses for different values of *R* are shown in *Table 1*. As indicated in the table, for α = 0.25, the maximum stress can be increased theoretically up to 60% using the MRSW method when the bolt is mounted on a rigid body.

Analytical determination of the value of *R* would be very difficult especially when the bolt is mounted on an arbitrarily shaped body. Because the elastic medium or structure on which the bolt is mounted is receiving stress in the transversal direction, there exists a combination of dilatation wave and distortion wave. Because the bolt is protruded, the elastic medium will also receive moment, which will cause motion perpendicular to its plane. Therefore, the stress wave equation of a one-dimensional bar should relate to three independent equations of motion of the elastic medium to find the analytic solution for *R*.

If needed, the value of *R* can be determined experimentally. This can be accomplished by generating an incident pulse with a small striker and measuring the magnitudes of the incident wave and the reflection wave separately, as Wada (1984) did in his experiment, and finding a ratio (*R*) between the magnitudes. In Wada's paper, he presented the ana-

*Table 1*
**Comparison of Maximum Stresses**

| *R* | Maximum Stress ($\sigma/\sigma_0$) | | Increase (%) |
|---|---|---|---|
| | SRSW Method | MRSW Method | |
| 1.0 | 2.0 | 3.213 | 60.7 |
| 0.9 | 1.9 | 2.862 | 50.6 |
| 0.8 | 1.8 | 2.532 | 40.7 |
| 0.7 | 1.7 | 2.221 | 30.6 |
| 0.6 | 1.6 | 1.930 | 20.6 |
| 0.5 | 1.5 | 1.660 | 10.7 |
| : | | | |

lytic solution and the experimental data for the reflection characteristics of longitudinal waves in a cylindrical bar contacted perpendicularly to an elastic plate. However, because the bolt-mounting medium has much more complex motion than a plate receiving force only in the $z$-direction, Wada's experimental method cannot be adopted directly in this research to determine reflection ratio of a stress wave. Therefore, only his experimental technique—separating incident wave and reflected wave—is adopted to investigate the reflection characteristics from a bolt head mounted on an arbitrarily shaped elastic body. The analytic determination of the value of $R$ is very difficult even with a bolt mounted on an ideally shaped geometry, such as an infinite plate or an infinite half space. Moreover, even if the solution were achieved for that kind of geometry, its implementation to practical situations, which involves arbitrarily shaped geometry, would be limited.

## Conclusions

Equations have been developed that represent stress waves caused by impact in a one-dimensional bar connected to an elastic medium. Literature on this issue provides the analytical stress wave equations for which a rigid striker with much heavier mass impacts an elastic bar supported by a rigid wall on the other end (Timoshenko and Goodier 1970, Johnson 1972, Spotts 1964). When the mass ratio of the striker to the bar gets larger, the classical analysis shows that the maximum stress on the wall-side end of the bar reaches much higher. This means that much higher stress can be produced on the wall-side end of the bar even with the same amount of mass and speed of the striker. Therefore, the fact that higher stress can be generated with the same amount of energy for launching the striker gives the possibility of improving efficiency of fracturing bolt heads using impact in the destructive disassembly process. The study shows that the maximum stress can be increased theoretically up to 60% in certain conditions when the MRSW method is applied. The equations have been developed for a transmitter bar contacting with a bolt mounted on an elastic product, which is virtually any product that has constant reflection value. The stress wave equations are developed based on the assumption that the stress waves would reflect in a constant ratio $R$ if the magnitude of the stress wave remains within the elastic boundary of the bolt-supporting medium. In addition, it is assumed that the product on which the bolt is mounted has enough inertia to withstand the force transmitted into the bolt-supporting medium. For a given value of $R$ and the parameters of the striker and transmitter, the developed equations give an estimated value of stress that will be accumulated on each end of the bar. From this value, the shear stress on the bolt head can be calculated and compared with the dynamic shear strength of the bolt to determine the fracture. The study shows that the equations developed here give identical results that classical equations give for the rigidly supported case ($R = 1$) (Johnson 1972, Spotts 1964, Burr 1995). The classical analysis for the rigidly supported case has been evaluated throughout many studies and proven well fitted with the empirical equation. Although the developed equations give the identical result that the classical equations give for $R = 1$, they need to be verified for different values of $R$. Therefore, future research will evaluate the developed equations through experiments to verify whether they correspond closely to the actual situation.

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### References

Burr, A.H. (1995). *Mechanical Analysis and Design*. Englewood Cliffs, NJ: Prentice Hall.

Chen, R.; Navin-Chandra, D.; and Prinz, F. (1993). "Product design for recyclability: A cost benefit analysis model and its application." IEEE Int'l Symp. on Electronics and the Environment.

Dowie, T. and Kelly, P. (1994). "Estimation of disassembly times." Unpublished technical report. Manchester, UK: Manchester Metropolitan Univ.

Feldmann, K. and Meedt, O. (1995). "Recycling and disassembly of electronic devices." *Proc.*, Prolomat '95, F-L. Krause, ed. Berlin, Germany, pp233-245.

Gupta, S.M. and McLean, C.R. (1996). "Disassembly of products." *Computers and Industrial Recycling* (v1-2), pp225-228.

Hanft, T.A. and Kroll, E. (1995). "Ease of disassembly evaluation in design for recycling." *Design for X-Concurrent Engineering Imperatives*, G.Q. Huang, ed. New York: Chapman & Hall.

Henstock, M. (1988). *Design for Recyclability*. The Institute of Metals.

Hrinyak, M.; Bras, B.; and Hoffman, W.F. (1996). "Enhancing design for disassembly: A benchmark of DFD software tools." 96-DETC/DFM-1271. *Proc.* of ASME DET and CIE Conf.

Johnson, W. (1972). *Impact Strength of Materials*. London: Edward Arnold.

Kirby, J. and Wandehra, I. (1993). "Designing business machines for disassembly and recycling." IEEE.

Kolsky, H. (1963). *Stress Waves in Solids*. Dover Publications.

Kroll, E. (1996). "Application of work-measurement analysis to product disassembly for recycling." *Concurrent Engg. Research and Applications* (v4, n2), pp149-158.

Kroll, E.; Beardsley, B.; and Parulian, A. (1996). "A methodology to evaluate ease of disassembly for product recycling." *IIE Trans.* (v28), pp837-845.

Macaulay, M.A. (1987). *Introduction to Impact Engineering*. New York: Chapman and Hall.

Penev, K.D. and Ron, A.J. (1996). "Determination of a disassembly strategy." *Int'l Journal of Production Research* (v34, n2), pp495-506.

Shyamsunder, N.; Ashai, Z.; and Ghad, R. (1996). "Virtual design for disassembly methodology." *Journal of Engg. Design and Automation*.

Spotts, M.F. (1964). *Mechanical Design Analysis*. Englewood Cliffs, NJ: Prentice Hall.

Studny, D.; Rittel, D.; and Zussman, E. (1999). "Impact fracture of screws for disassembly." *Journal of Mfg. Science and Engg.* (v121, Feb. 1999).

Timoshenko, S.P. and Goodier, J.N. (1970). *Theory of Elasticity*, 3rd ed. New York: McGraw-Hill.

Wada, Hiroshi (1984). "Reflection characteristics of longitudinal waves in a semi-infinite cylindrical rod connected to an elastic plate." *Int'l Journal of Mechanical Science* (v26, n5).

## Authors' Biographies

Raj S. Sodhi, PhD, PE, is an associate professor in mechanical engineering at the New Jersey Institute of Technology. He received his PhD in mechanical engineering from the University of Houston. He served as director of manufacturing programs at NJIT from 1990 to 1994. The NJIT manufacturing engineering program was recognized with the SME University LEAD Award in 1994. Dr. Sodhi is active in research and education related to mechanical design, mechanisms synthesis, design for recycling, and manufacturing engineering. He has received the NJIT Excellence in Teaching Award for Graduate Instruction, the N. Watrous Procter & Gamble Award from the Society of Applied Mechanisms and Robotics for significant contributions to the science of mechanisms and robotics, and the Ralph R. Teetor New Engineering Educator Award of the Society of Automotive Engineers. He is a member of SME, ASME, and SAE.

Kyung Pak received his PhD in mechanical engineering from the New Jersey Institute of Technology in May 2002. His research interests are in mechanical design, fracture mechanics, and manufacturing engineering.