Experimental Investigation on Bolt Tension of Flanged Pipe Joint Subjected to Bending

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Experimental Investigation on Bolt Tension of Flanged Pipe Joint Subjected to Bending

by

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A thesis submitted to the Department of Civil Engineering of Bangladesh University of Engineering and Technology, Dhaka, in partial fulfillment of the requirements for the degree of

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LIST OF SYMBOLS AND ABBREVIATIONS

AISC	American Institute of Steel Construction
RCSC	Research Council on Structural Connections
ASTM	American Society for testing and Materials
FEM	Finite Element Method
2-D	Two dimensional
3-D	Three dimensional
f_y	Yield strength of steel
l_p	Pipe length
d_p	Nominal diameter of pipe
d_b	Nominal bolt diameter
f_w	Flange width
f_t	Flange thickness
p_t	Pipe wall thickness
n	Number of bolt
b_{ht}	Thickness of bolt head
T	Maximum bolt tension
CF	Applied clamping force
M	Applied Moment
D	Deflection
F_c	Compressive force
F_t	Tensile force
K_c	Stiffness in compression
K_t	Stiffness in tension

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ABSTRACT

Analysis of a flanged pipe joint subjected to bending and determination of maximum bolt tension is a rather difficult task due to absence of a rational method of analysis. Conventionally, in absence of specific guideline or code provisions, a method analogous to beam flexure is adopted to determine the bolt tension. It is assumed that bolt force (tension or compression) is linearly proportional to the distance from the neutral axis. But this methodology may not always give the accurate result since the assumptions made are not practically valid. In this experimental research, an investigation has been made to find out the effect of various parameters relating to flanged pipe joint connection.

To carry out the investigation, 36 flanged pipe joint specimens are prepared using 125 mm, 150 mm and 200 mm diameter pipe. Flanges are made from different thickness (6mm to 24mm) of mild steel plates having 30 mm, 36 mm and 48mm width. Flanges are connected using ASTM A325M bolts of 10 mm, 12 mm and 16 mm diameter. Flange is connected to the pipe using slip on flange method. During experiment two points load is applied by a custom made load spreader using the universal testing machine. To measure bolt force special strain gauge BTM-6C is inserted inside bolt shank and reading is taken using data logger. The pipe joint has been subjected to moment by applying force at a certain distance from the flange joint and under this moment the maximum bolt tension has been measured.

Based on analysis of experimental results it may be suggested that for a flanged pipe joint subjected to pure bending moment, when flange thickness is same as pipe thickness bolt tension value obtained from conventional analysis should be doubled to obtain a reasonable estimate of bolt tension. When flange thickness is thrice the pipe thickness bolt tension calculated by conventional method may be assumed reasonable. For intermediate thickness of flange a linear interpolation may be performed.

CHAPTER 1
INTRODUCTION

1.1 GENERAL

Different types of structures are built all over the world. Steel pipe (Tube or circular hollow section [CHS]) is used as a structural frame member in different structures. When length of such a member is more than the length of available sections, pipes are typically joined through bolted flanged connections. In flanged pipe joint structure, pipes are connected with flanges which are then bolted together in order to make a strong joint. Bolted circular flange connections are used for tubular members in buildings as well as in structures such as chimneys, pylons for lighting, communication, wind turbines and ski-lift installations. When a flanged pipe joint is subjected to bending, the whole structure tends to bend and some reaction forces are developed on the bolts. In many cases the pipe joint may fail due to higher loading. Therefore, in order to ensure the safety of flanged pipe joint structure, the maximum bolt reaction needs to be known. Adequate design of such a bolted connection requires proper estimation of the force (tensile) developed in bolts. Conventionally, the linear distribution method of bolt force is generally used for determining bolt reaction force. But this method does not always give the accurate result since there are some limitations. So far, only a few research works has been carried out on the design of such a connection. However, none of these works has been verified against experimental results, so there are significant scopes to investigate this matter. The present experimental work aims at investigating the behavior of flanged pipe joints through laboratory testing.

1.2 BACKGROUND AND RESEARCH SIGNIFICANCE

Pipe is one of the most widely used structural steel products. It is also used as a structural frame member in different structures all over the world. In flanged pipe joint structure, pipes are connected with flanges which are then bolted together in order to make a strong joint. Adequate design of such a bolted connection requires proper estimation of the force (tensile) developed in bolts. Generally no design guideline or code provision is available to rationally determine the bolt tension in such a joint. So far, only a few research works has been carried out on the design of such a connection. Long ago, Waters and Taylor explored the stress condition on flanged pipe joint, with the objectives of determining the location and magnitude of maximum stresses acting on the flanges. More recently, Hwang and Stalling using 2-D finite element model investigated the stressed behavior of such a connection. Later, Choudhury and Azim performed a nonlinear finite element analysis, to determine the maximum bolt tension acting in flanged pipe joints subjected to bending. Recently, Tahfeem using finite element analysis, developed an equation to determine maximum bolt tension in such a connection.

1.3 OBJECTIVES OF THE PRESENT STUDY

The objective of this research work is to experimentally investigate the behavior of a flanged pipe joint subjected to pure bending. The investigation shall include the study of the effect of various parameters on bolt tension under certain range of parametric conditions:

- Number of bolts.
- Flange thickness.
- Diameter of bolt.
- Flange width.
- Pipe diameter.

Finally to determine the maximum bolt tension in flanged pipe joint subjected to pure bending.

1.4 METHODOLOGY OF THE STUDY

For the purpose of carrying out the experimental investigation, an experimental rig has been setup to study flanged pipe joint. Three basic diameter of pipe are used for preparing the specimen flanged joints. Flanges are prepared using different thickness of mild steel plates. For bolting purpose ASTM A325M bolts with washer are used. Special strain gauge is inserted inside bolt shank to measure the strain. After cutting the pipes in required length, flanges are welded using slip on flanges method. Application of two points load is achieved by the use of the universal testing machine (UTM) together with a custom-built load spreader. This arrangement applied load to the upper portion of the specimen and this load is reacted by two frictionless knife edged loose saddles, which allowed the joint to rotate in the axial plane. Details of the experimental layout can be viewed in Figures 3.6. Applied load reading is recorded from UTM machine load cell and strain gauge reading is recorded in data logger. Finally, applied moment vs bolt force graphs are prepared. The specimen of flanged pipe joint is subjected to applied moment until tearing of bolts. Under this moment, the maximum bolt tension is being measured. The effect of various parameters on bolt tension such as diameter of pipe, flange thickness, flange width, diameter of bolt and number of bolt has been investigated.

1.5 ORGANIZATION OF THE THESIS

The report has been organized so as to best describe and discuss the problem along with findings that came out from the experiment performed. Chapter 1 introduces the problem and presents an overall idea about present experiment. Chapter 2 is literature review which represents the work performed so far in connection with it collected from different references. It also describes the strategy of advancement for the present problem to a success. Chapter 3 presents the material properties and experimental setup used in the present study. This chapter presents the mechanism in flanged pipe joint, research strategies, material properties, experimental setup, allowable stress and flange joint configuration, and design of pipe joint specimen, specimen preparation, equipment used for testing and data acquisition, and specimen designation. Chapter 4 is the core of this thesis write up, which describes the experimental results with many tables and figures followed by some definite remarks. The last chapter is Chapter 5, which summarizes the entire work as well as makes some recommendations for future research.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

Pipe is one of the most widely used structural steel products. It is found in every modern home for plumbing and heating, in industrial building, into cross-country oil lines, in water and gas systems, in transmission towers, chimneys and pylons for lighting, communication, wind turbines and ski-lift installations and in countless other places. It is also used as a structural frame member in different structures all over the world. This research paper focuses on the pipe element that is used as a structural element in different structures. Most importantly, these structural pipes offer better resistance to torsional stresses and sagging. Moreover, their comparatively small exterior surface area, without sharp angles, ensures ease of maintenance.

Steel pipes are used in different structures as structural frame members. Some of the most widely used examples where pipes are used as structural frame members with bolted joint connections are shown below:

- Buildings.
- Foot over bridges.
- Structures in entertainment parks.
- Water tanks.
- Bill board columns.
- TV masts.
- Transmission towers.
- Steel pipe piles.
- Wind turbine.

Buildings

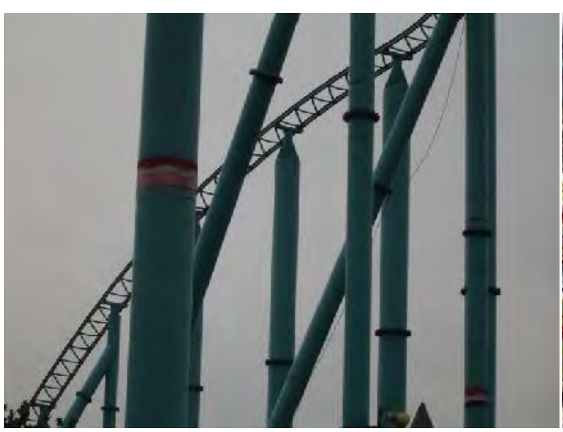
Tubular steel pipe columns are used in many building systems. For rapid construction and high salvage value in industrial building steel frame structures are widely used. Among various steel elements pipe element is one of the most popular elements. For example, Buckling-restrained braces connect the core to concrete-filled pipe columns and, through the action of steel yielding within concrete confinement, act as a seismic

(a)

(b)

(a)

(b)



(a)

(b)

fuse during an earthquake. Similarly, pipe column is used for the conveyor truss in the Aggregate Product, Inc. (API) Gateway Project, designed to span the United States-Mexico Border (Figure 2.1).

Foot Over Bridges

Generally several foot-over bridges are constructed for the pedestrian at different road crossing to facilitate uninterrupted and safe movement of the pedestrian. These over bridges are supported by means of various types of column system, such as concrete, steel I-section, steel pipe columns, etc. The foot over bridge in BUET campus, is supported by steel pipe columns (Figure 2.2).

Structures in Entertainment Parks

Appearance is the first criteria for the structures in an entertainment park and pipes are the most lucrative structural elements by any means. It takes least space, can take both static and dynamic forces, and above all it makes the structure attractive to the tourists. Such an example can be observed in Fantasy Kingdom Entertainment Park and Jamuna Future Park where the track of the roller coaster is supported by steel pipe columns (Figure 2.3).

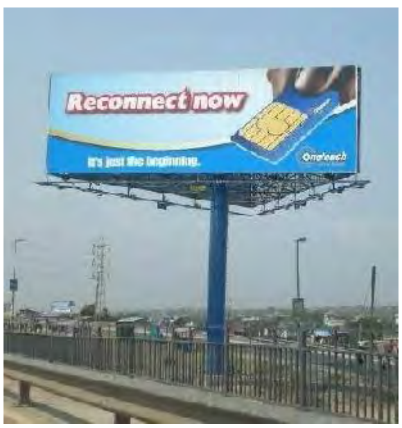
Overhead Water tanks

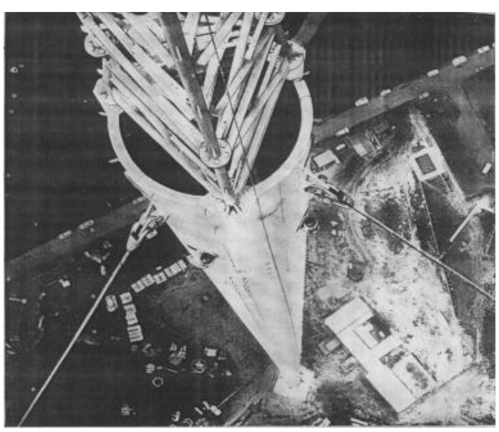
Overhead water reservoir can be built on pipe columns. A 500,000 gallon ellipsoidal tank was built at Florida, USA. This tank is elevated by a series of steel columns around its perimeter. The multi-column, leg type elevated tank design is common in the standpipe design. In Dhaka several overhead water tanks are built on circular pipe columns. At Lalmatia, there is a huge overhead water tank, supported on nine pipe columns (Figure 2.4).

Billboard Columns

Different types of support systems are used for the advertisement billboards. Among them steel pipe column is the most current practice. In Dhaka city a great number of bill boards are observed where steel pipe columns are used to support those structures (Figure 2.5).

(a) (b)





(a) (b)

TV Mast

Most of the tv mast column are made with steel pipe. The 1,265 foot T.V. mast constructed at Emley Moor, Yorkshire, mainly consists of cylindrical steel columns of 9 feet in diameter has been shown in Figure 2.6.

Transmission Towers

For different purposes transmission towers are used such as telecommunication network, electricity network, etc. And like many other elements, steel pipes are used as frame element in transmission towers. Three legged BTTB transmission tower at Katabon has been built using steel pipe elements (Figure 2.7).

Steel Pipe Piles

Steel pipe piles are mostly used in deep foundation for example in bridge foundation whose diameter is of more than 300mm.Steel pipe pile types differ from each other according to the structure of the pile point, pile shaft and the pile driving method to be used. Some driven steel pipe piles at a construction site are shown in figure 2.8.

Wind Turbine

A wind turbine is a device that converts kinetic energy from the wind into electrical power. Hydraulic towers, is applied to wind turbine generators ranging from 1kW to 50KW. It’s composed by three sections of steel pipe with different diameters, the hydraulic towers are supplied by tower base, hypocylinder, and hydraulic pump. Flanges are welded to the both sides of each tower for connections. Land and offshore based wind tower is shown in figure 2.9.

Beside these, there are several others examples where pipes are used as structural frame elements. Pipes are usually connected through flanges using bolts. The joints are the weakest element in most structures. In spite of their importance, bolted joints are not well understood. There are widely used design theories and equations for liquid transmission pipe joints, but they are not involved in the design and construction of the structural pipe frame systems. Pipe joints must be designed to resist moment because the pipes are the parts of a rigid frame. For example, pipes, which are parts of the wind-bracing system of a tier building, must resist end

(a) (b) (c)

(a) (b)

(a) (b)

moments resulting from both wind forces and gravity loads. In the usual case, moment resistance of bolted or riveted connections in such frameworks depends upon tension in the fasteners. When the pipe is subjected to bending or a moment acts on the pipe, some tensile force is produced in the bolt. Hence it is of utmost importance to determine the tension value of the bolt. Maximum bolt tensions are required for the safe design. In this research paper factors influencing determination of the bolt tension under a certain range of different parameters such as diameter of pipe, number of bolts, thickness of the pipe or flange, flange width, diameter of bolts etc are examined.

2.2 TYPES OF PIPE JOINTS

There are different types of steel and alloy steel pipe joints. Among them few names are given below:

- Butt weld joint
- Socket weld joint
- Threaded joint
- Flanged joint
- Compression sleeve coupling
- Grooved segment-ring coupling.

This research paper concentrates on flanged joint that is connected using a number of bolts (Figure 2.10).

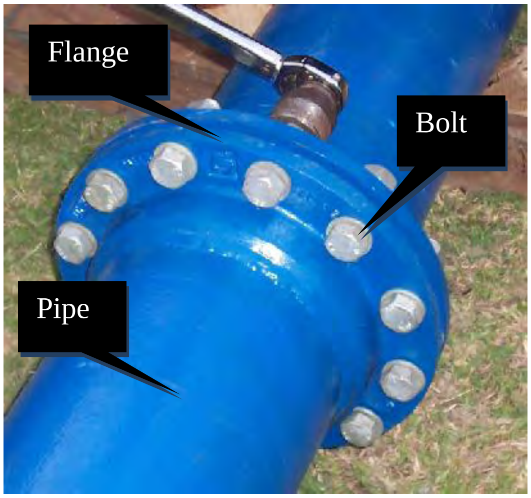


Figure 2.10 A flanged pipe joint with different components.

2.3 TYPES OF FLANGE

Flanges are most often used to connect pipes that have a diameter greater than 2 inches. A flange joint consists of two matching disks of metal that are bolted together to achieve a strong seal. The flange is attached to the pipe by welding, brazing, or screwed fittings. A number of the most common types of flanges are shown below:

(a) Blind or Blank Flange

(b) Lap Joint Flange

(c) Slip-On Flange

(d) Socket Welding Flange

(e) Threaded Flange

(f) Welding Neck Flange

Figure 2.11 Different types of flange.

- **Blind or Blank Flanges**

  These flanges are commonly used to seal or blank off the ends of valves and pipes. They are always rounded and have bolt holes around their perimeter. They are economical.

- **Lap Joint Flange**

  For pipes that are frequently taken apart for repair or replacement, lap joint flanges are used. These flanges are used with a universal stub-end insert that is easily rotated and makes it easy to line up bolt holes.

- **Slip-on Flange**

  Slip-on flanges are thinner than most flanges as they are slide over pipe ends, easily aligned with bolt holes, and then permanently welded into place. They are stronger than other flanges.

- **Socket Welding Flange**

  The socket welding flanges are created specifically for small piping that must withstand high pressure.

- **Threaded Pipe Flange**

  This popular type of flange not only comes in a wide range of materials and sizes, it can also be attached to pipes without welding, as it is a threaded flange. Extremely useful in connecting small pipes, it is a low cost flange that can be safely used in highly explosive and high pressure areas of projects and machinery.

- **Weld Neck Flange**

  The most popular type of high pressure pipe flange is the weld neck flange. The tapered hub design provides an extremely strong connection and these flanges can be repeatedly manipulated and bent without compromising the quality of the material and/or connection. The weld neck flange is highly resistant to extreme high and low temperatures.

2.4 METHODS OF WELDING FLANGES

The methods used in welding flanges to pipe will vary according to the type of flange and are basically the same as those used in standard welding practices (Figure 2.12). For welding Neck and Lap Joint Stub and Flange applications an initial root gap spacing between pipe and flange should be 1/16” to 1/8”. The first bead must penetrate uniformly to the inside wall of the assembly thus assuring a strong joint. The final bead should be built up above the pipe O.D. approximately 1/16”.

- **Welding neck flanges**

  This type preferred for severe services because it provides the greatest factor of safety and fatigue strength. These features make this type of flange suitable for all pressures and temperatures for which flanges are rated. Welding cannot cause distortion of the flange face.

- **Lap joint stub flanges**

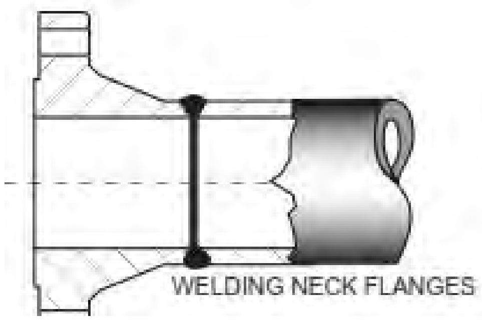
  This type assembly has a lower factor of safety in resisting pressure and fatigue than welding neck flanges. It is available for all pressures and temperatures for which flanges are rated and because the flange swivels freely on the stub, the alignment of both holes is independent from the welding operation. Welding cannot cause distortion of the gasket face.

- **Slip-on flanges Type-I**

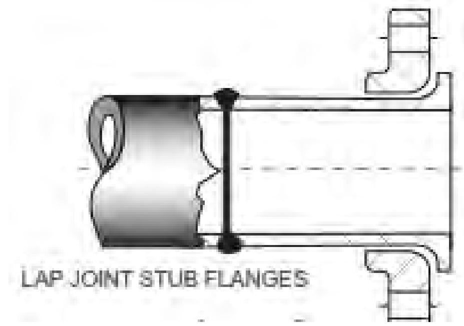
  This Slip-On Flange construction necessitates refacing after welding thus reducing the economic advantage this type flange has over the welding neck type. It is used where smooth bores free from pockets are desired. Slip-On Flanges are standard in the 150 lb. and 300 lb. classes only and because of their lower factor of safety in resisting pressure and fatigue, they are not recommended for service above 750ºF.

- **Slip-on flanges Type-II**

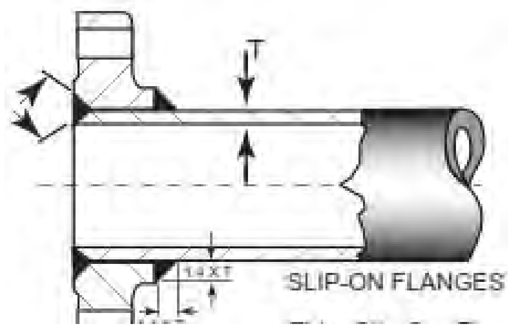
  This Slip-On Flange is often chosen over Welding Neck Flanges because of its lower initial cost and is widely used because it requires less accuracy in cutting pipe to length and permits alignment of bolt holes and squaring of flange faces with less difficulty. Refacing to repair warpage or weld spatter



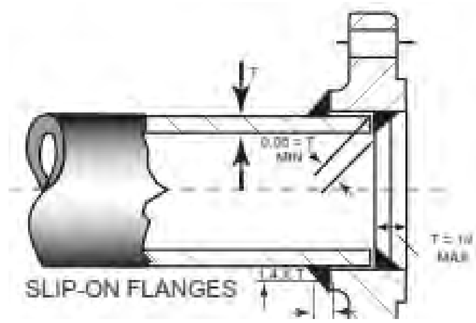
(a) Welding neck flanges



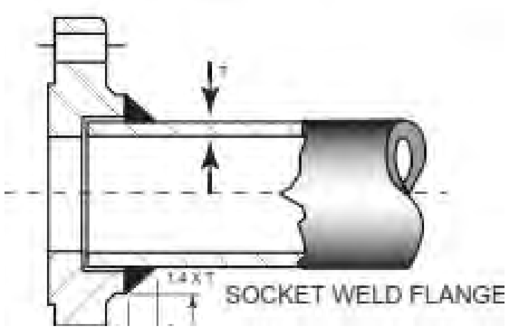
(b) Lap joint stub flanges



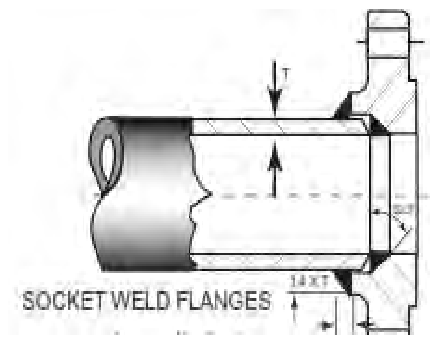
(c) Slip-on flanges Type-I



(d) Slip-on flanges Type-II



(e) Socket weld flanges



(f) Socket weld flanges

- **Socket weld flanges Type-I**

  For those applications in which the running of an internal weld is particularly difficult, this Socket Weld construction is recommended. In sizes 4” and smaller this construction has approximately the same resistance to internal pressure and fatigue as the Slip-On Flanges pictured above.

- **Socket weld flanges Type-II**

  This sock weld construction eliminates internal pockets while avoiding warpage from welding heat and weld spatter damage to the flange face. It has the same resistance to internal pressure as that of Slip-On Flanges and better fatigue life. This construction is recommended for all sizes in 50 lb. and 300 lb. pressure classes.

2.5 CONVENTIONAL ANALYSIS AND DESIGN

Bolt tension of a flanged pipe joint can be calculated by using the conventional method. This linear force distribution method is similar to the flexural stress distribution of a beam section subjected to bending. In fact, when a beam is subjected to bending, flexural stress generally develops along the axis of the beam which is normal to the plane section. This flexural stress distribution on the cross section of a beam due to bending has been shown in figure 2.13. From the figure, it can be seen that tensile stress develops at the lower half of the plane and compressive stress develops at the upper half of the plane section. It is also observed that the stress is zero at the midlevel but it tends to increase towards the extreme fiber of the beam section. This similar phenomenon also happens in the flanged pipe joint when it is subjected to bending.

A typical front view of a 200 mm diameter pipe joint specimen with 12 bolts have been shown in figures 2.14. Figure 2.15 shows the plan and linear force distribution system of the same flanged pipe joint. When a flanged pipe joint is subjected to bending; the reaction force normally develops on the bolts. From figure 2.15, it is observed that the tensile reaction force develops on the lower bolts and compressive force develops on the upper bolts whose phenomenon is almost similar to the flexural stress distribution of beam section mentioned earlier.

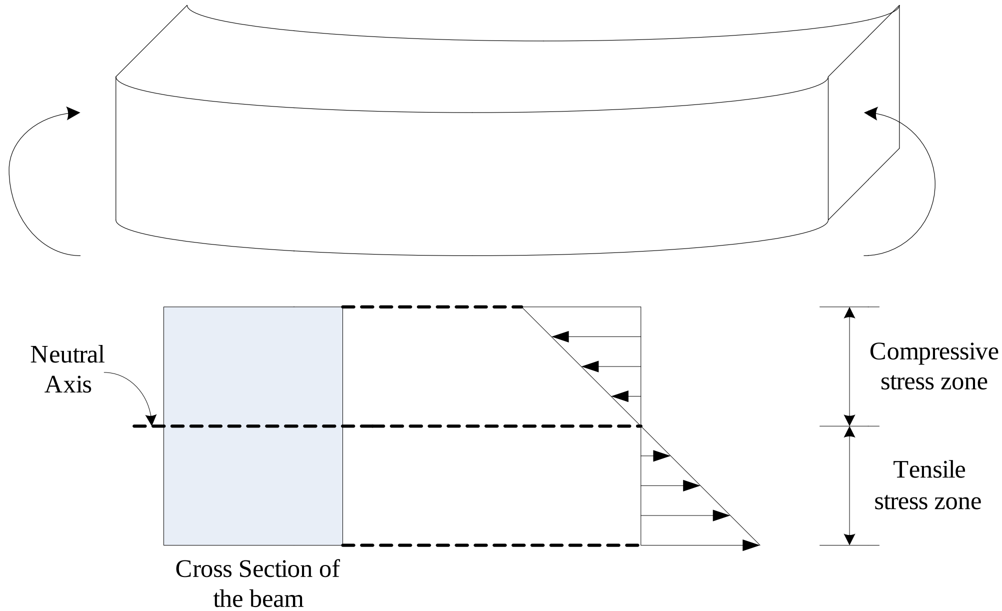


Figure 2.13 Flexural stress distribution in a beam section subjected to bending

Figure 2.14: Front view of a 200 mm diameter flanged pipe joint specimen with 12 bolts.

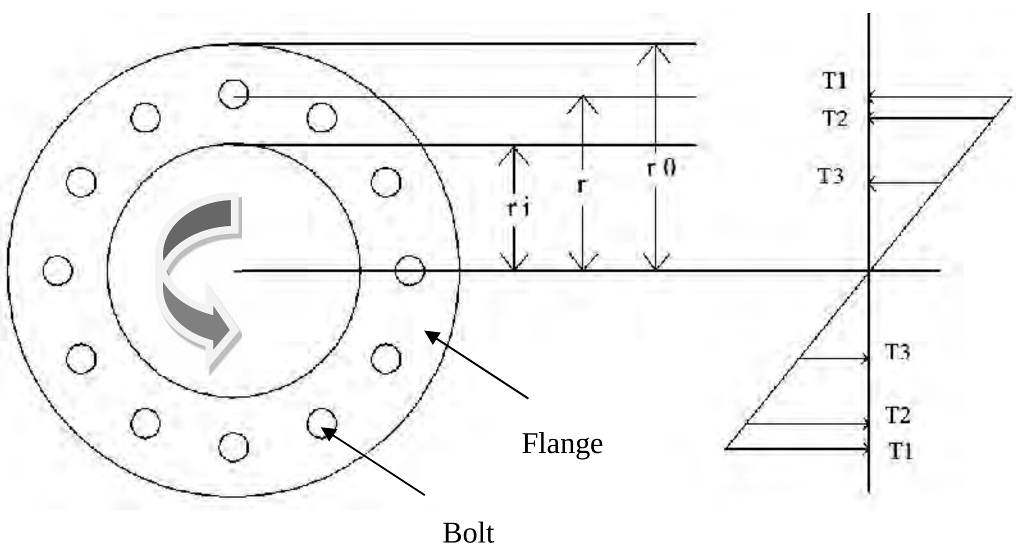


Figure 2.15 Plan and force distribution of a typical flanged pipe joint

Sample Calculation:

Assume,

The pipe radius = r_i in.

The outer radius of flange = r_o in.

The distance from the center of the pipe to the center of the bolt = r in.

Here,

$$T_2 = T_1 \times sin60°$$
$$T_3 = T_1 \times sin30°$$

Taking moment at the center of the pipe,

$$2 \times T_1 \times r + 4 \times T_2 \times r\, sin60° + 4 \times T_3 \times r\, sin30° = M$$

$$or, 2 \times T_1 \times r + 4 \times T_1 r \times sin^2 60° + 4 \times T_1 r \times sin^2 30° = M$$

$$or, T_1 = \frac{M}{\left(2r + 4r\, sin^2 60° + 4r\, sin^2 30°\right)}$$

By using this method, the maximum tension of a bolt can be determined which is farthest from the centre of the pipe for various pipe diameter and for different number of bolts.

2.6 PREVIOUS WORKS

2.6.1 Works on Stress in Bolted Flanged Connection

The earliest method of calculation to receive wide attention was the so-called "*Locomotive*" method, (*The Locomotive*) generally credited to the late Dr. A.D. Risteen, 1905. The section abcdefg in figure 2.15 is assumed to rotate counterclockwise, but without distortion. The final equation is in effect the conventional flexure formula, the external moment being the total bolt moment per radian angle and the section modulus being that taken about the axis X-X" through the center of gravity. For ring flanges this gives the tangential stress on either face, and for hubbed flanges it gives the tangential stress at the free end of the hub.

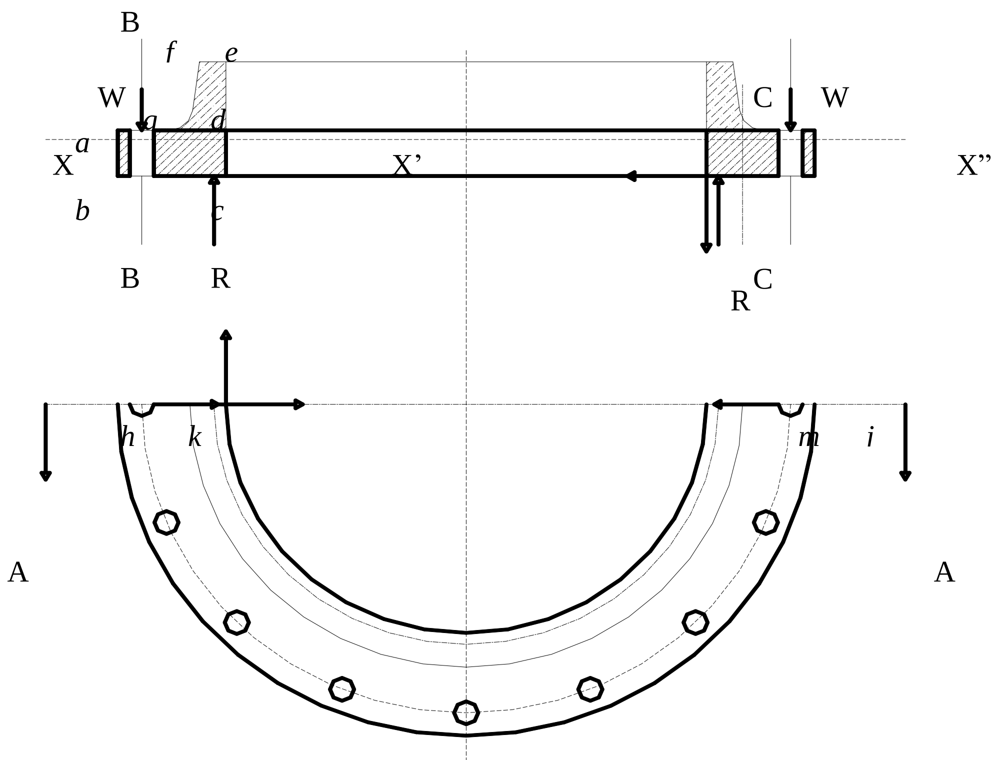


Figure 2.16 Illustration of earlier methods of calculating stress in a bolted flange connection.

Crocker and Sanford developed a method (Discussion of paper by Waters and Taylor , 1927) whereby the flange is analyzed as a beam, in which bending about the neutral axis X-X " takes place on the section A-A, and the external loads are one half the bolt load W and one half the reaction R, each concentrated at the center of gravity of their respective half-circles (The location of the bolt-load circle, however was assumed tangent to the inner edge of the bolt holes, and not along the bolt circle in figure 2.15. This method likewise gives the tangential stress on either face of a ring, or at the end of the hub. Den Hartog (Discussion of paper by Waters and Taylor, 1927) showed by vector analysis that although the Locomotive and Crocker-Sanford methods are derived in different ways, they are fundamentally identical.

A method devised by Tanner for ring flanges, and discussed by Waters and Taylor (Taylor-Waters, 1927), is to assume the ring to be fixed at the section B-B around the bolt circle and to be equivalent to a cantilever beam of length L I with the "concentrated" load R uniformly distributed across a width equal to the circumference of the ring. This method gives the radial stress assumed to be present at section B-B. In the application of the method, Tanner took account of the tangential stresses by using suitable factors derived from experiments on rings of the proportions in which he was interested. The Tanner method was modified by Crocker (Discussion of paper by Waters and Taylor, 1927) for application to hubbed flanges (and presumably adaptable to ring flanges also) by assuming the fixed section to be the weakest section C-C in the ring at the base of the hub, with the load W "concentrated" at the distance L2 at the free end and distributed along the bolt- loading circle. This likewise results in a calculation of the radial stress assumed to exist, in this case at section C-C.

None of the foregoing methods took into account all the conditions present in the flange under load, and so the Waters and Taylor paper in 1927 based on a combination of the flat plate and the elastically supported beam theories, was probably the first instance in which the stress conditions in a flange the three principal directions - tangential, radial, and axial - were explored with the object of determining the location and magnitude of the maximum stress. Formulas were included for the deflection of the ring, and the calculated deflections were compared with those actually obtained in several series of tests, the data of which were also reported. Because in flange proportions considered at that time the tangential stress in the ring at the inside diameter was the controlling factor, the formulas for stresses elsewhere in the flange were generally over-looked by designers.

The Waters-Taylor evoked extensive discussion in the course of which Timoshenko presented an analysis for both ring flanges and hubbed flanges, including a method of dealing with hubs shorter than the so called "critical" length. Most of these formulas can be found also in his work on "Strength of Materials"(S. Timoshenko, 1930).

During 1934 the rules for flanges in the A.S.M.E. and the A.P.1-A.S.M.E. Unified Pressure Vessel Codes were first published. The wide range of their application made it necessary to use formulas based on a rational and complete theory. As the Waters-Taylor equations met this requirement and had been checked by experiment, they were adopted with auxiliary charts to simplify the calculations. The radial-stress formula was omitted, however, because it was not believed that it would be the critical factor in any practical design.

A research paper was published by Waters et al., 1937 which outlines a revised analysis based on the ring, tapered hub, and shell being considered as three elastically coupled units loaded by a bolting moment, a hydrostatic pressure, or a combination of the two. Design formulas and charts were developed for the computation of stresses that are likely to be critical.

2.6.2 FEA Based Work by Chowdhury (2008)

A research study was conducted on bolt tension in a bolted flanged pipe joint by Choudhury, 2008. In that study, a nonlinear finite element analysis was performed on flanged pipe joint subjected to bending. After that, the effect of various parameters such as pipe diameter, flange width, flange thickness, number of bolts had been observed on the bolt tension of the pipe joint. The finite element mesh of the pipe joint with applied load has been shown in figure 2.17 from which his modeling can be easily understood. A typical deformed shape of the pipe joint subjected to bending is also shown in figure 2.18 which gives an idea of the properties of the contact elements at the pipe joint.

From these figures it is observed that the contact lines along the bolt location take more tension than other contact lines. It is to be noted here that only one bolt diameter (25 mm) was used in his analysis. To determine the maximum bolt tension, five empirical equations had been proposed by Chowdhury, 2008 for five different flange thickness to pipe wall thickness relationships (i.e. flange thickness = pipe wall thickness, flange thickness = 2 × pipe wall thickness, flange thickness = 3 × pipe wall thickness, flange thickness = 4 × pipe wall thickness and flange thickness = 5 × pipe wall thickness), using the finite element analysis results. The first four equations (2.1 to 2.4) were logarithmic equation with constant values related to the diameter of the pipe. The fifth equation (2.5) was the second order polynomial equation with all the constant values, depended on the diameter of the pipe. The proposed equations were developed to give bolt tension values as close as possible to the nonlinear finite element analysis results. There only one equation was generalized for a single case of

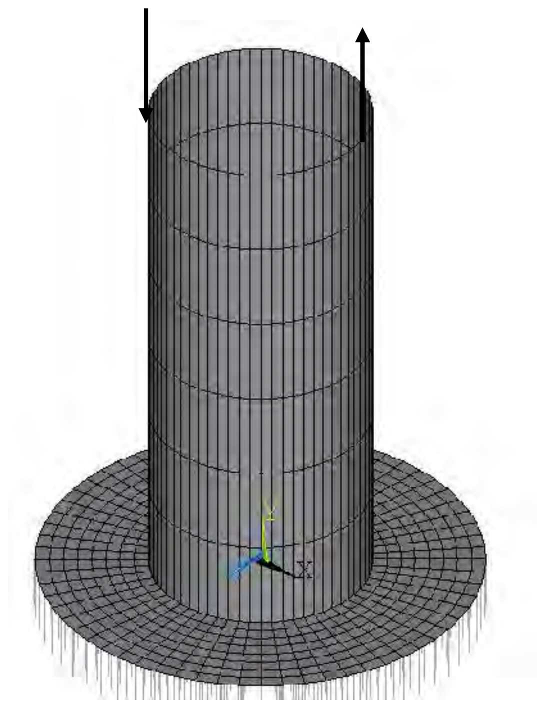


Figure 2.17 Finite elements mesh of flanged pipe joint with load developed by Chowdhury

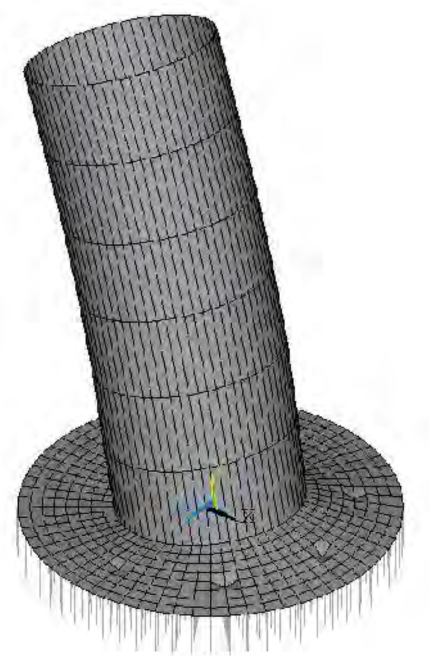


Figure 2.18 Typical deflected shape of flanged pipe joint model developed by Chowdhury

flange thickness. So the bolt tension obtained from the equations varied a little from the finite element analysis values. The five proposed empirical equations derived by Chowdhury, 2008 are shown in the following pages with the limiting values.

In case of Flange thickness = Pipe thickness

Proposed Equation:

$$T = p + a \times f_w \times \ln(n) + \frac{b}{(f_w)^2} \tag{2.1}$$

Where,

T = Bolt tension in Newton

p = Bolt tension by conventional analysis in Newton

f_w= Flange width, ($50 \le d \le 75$) mm.

$a = 1.0906 \times e^{0.02\, d}$

$b = (-2775) \times e^{0.03\, d}$

d = Pipe diameter ($150 \le d \le 300$) mm.

n = Number of bolts ($4 \le n \le 16$)

F_y = Yield strength of pipe

In Case of Flange thickness = 2×Pipe thickness

Proposed Equation:

$$T = p + a \times f_w \times \ln(n) + b \times f_w \tag{2.2}$$

Where,

T = Bolt tension in Newton

p = Bolt tension by conventional analysis in Newton

f_w= Flange width, ($50 \le d \le 75$) mm.

$a = a_1 d^2 + a_2 d + a_3$

$b = b_1 d^2 + b_2 d + b_3$

$a_1 = 0.006$ $b_1 = 0.012$

$a_2 = 0.7$ $b_2 = -3.6$

$a_3 = -250$ $b_3 = 300$

d = Pipe diameter ($150 \le d \le 300$) mm.

n = Number of bolts ($4 \le n \le 16$)

F_y = Yield strength of pipe

In Case of Flange thickness = 3 × Pipe thickness

Proposed Equation:

$$T = p + a \times f_w \times \ln(n) + b \times f_w + (f_w - 50) \times (300 - \mathrm{d}) \times n \qquad (2.3)$$

Where,

T = Bolt tension in Newton

p = Bolt tension by conventional analysis in Newton

f_w= Flange width, $(50 \leq \mathrm{d} \leq 75)$ mm.

$a = a_1 d^2 + a_2 d + a_3$

$b = b_1 d^2 + b_2 d + b_3$

$a_1 = 0.016$	$b_1 = 0.014$
$a_2 = -3.6$	$b_2 = -3.3$
$a_3 = 40$	$b_3 = 50$

d = Pipe diameter $(150 \leq \mathrm{d} \leq 300)$ mm.

n = Number of bolts $(4 \leq \mathrm{n} \leq 16)$

F_y = Yield strength of pipe

In Case of Flange thickness = 4 × Pipe thickness

Proposed Equation:

$$T = p + a \times f_w \times \ln(n) + b \times f_w + (f_w - 50) \times (300 - \mathrm{d}) \times 2 \times (n - 7) \quad (2.4)$$

Where,

T = Bolt tension in Newton

p = Bolt tension by conventional analysis in Newton

f_w= Flange width, $(50 \leq \mathrm{d} \leq 75)$ mm.

$a = a_1 d^2 + a_2 d + a_3$

$b = b_1 d^2 + b_2 d + b_3$

$a_1 = 0.022$	$b_1 = 0.0192$
$a_2 = -8.9$	$b_2 = -8.2$
$a_3 = 830$	$b_3 = 832$

d = Pipe diameter $(150 \leq \mathrm{d} \leq 300)$ mm.

n = Number of bolts ($4 \le n \le 16$)

F_y = Yield strength of pipe

In case of Flange thickness = 5 × Pipe thickness

Proposed Equation:

$$T = p + a\,n^2 + bn + c + \{(f_w - 50) \times 5 \times d\} \tag{2.5}$$

Where,

T = Bolt tension in Newton

p = Bolt tension by conventional analysis in Newton

f_w= Flange width, ($50 \le d \le 75$) mm.

$a = a_1 d^2 + a_2 d + a_3$

$b = b_1 d^2 + b_2 d + b_3$

$c = c_1 d^2 + c_2 d + c_3$

$a_1 = 0.281$	$b_1 = -6.45$	$c_1 = 36.4252$
$a_2 = -109.28$	$b_2 = 2526.78$	$c_2 = -14537.44$
$a_3 = 9765.75$	$b_3 = -224851.5$	$c_3 = 1299991$

d = Pipe diameter ($150 \le d \le 300$) mm.

n = Number of bolts ($4 \le n \le 16$)

F_y = Yield strength of pipe

2.6.3 FEA Based Work by Azim (2010)

More recently, another research study was carried out on bolt tension of pipe joint by Azim, 2010. In his research work, a nonlinear finite element analysis was performed on flanged pipe joint under bending. The effect of various parameters such as pipe diameter, flange width, flange thickness and number of bolts had been observed on the bolt tension. The finite element mesh of the pipe joint with applied load has been shown in figure 2.19. A typical deformed shape of the pipe joint due to bending is also shown in figure 2.20 to observe the behavior of contact elements.

From the deflected shape of the pipe joint as shown in figure 2.19, it can be seen that the contact lines along the edges of bolt holes took more tension compared to other surrounding contact lines.

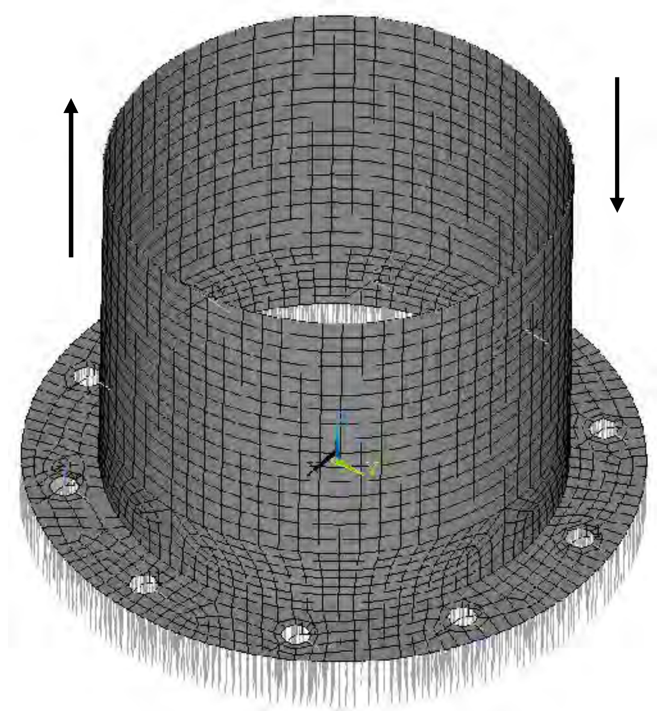


Figure 2.19 Finite elements mesh developed by Azim for flanged pipe joint with load

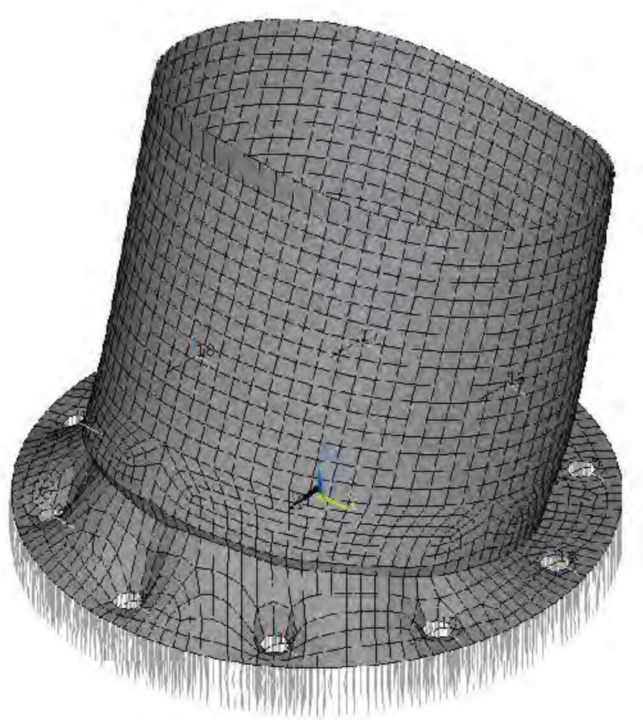


Figure 2.20 Typical deflected shape of flanged pipe joint developed byAzim

On the basis of the analysis results, one empirical equation had been suggested for determining the maximum bolt tension for three different flange thicknesses (flange thicknesses = pipe wall thickness, flange thicknesses = 2 × pipe wall thickness, flange thicknesses = 3 × pipe wall thickness) of the pipe joint within the specified range stated in this article. Also, this equation did not require conventional analysis to be done mandatorily. This equation had been developed through extensive statistical analysis with trial and error. More importantly, variables of this equation were taken as pipe nominal diameter, flange thickness, pipe wall thickness, flange width and number of bolts. Since bolt diameter was set at 20 mm for all cases, so bolt diameter was not incorporated in this equation. After fine tuning a number of times, one equation was reached which gave values very close to finite element analysis results (within ±15%). This equation is applicable for all AISC standard steel pipe sections of 40 ksi yield strength with diameter of 125 mm and higher. The empirical equation derived by Azim is given as follows:

$$T = \frac{1}{2}d^2 + \frac{300}{d}{f_t}^2 f_w + 3000\, b^k \tag{2.6}$$

where,

T = Maximum Bolt tension in Newton.

f_w = Flange width, (f_w = 75, 100, 125 mm)

d = Nominal Pipe diameter (d = 125, 150, 200, 250, 300 mm)

n = Number of bolts (n = 4, 6, 8, 10, 12, 14, 16)

f_t = Flange thickness (1, 2 or 3 times of pipe wall thickness)

p_t = Pipe wall thickness

$b = \frac{\sqrt{f_t}\,{p_t}^{1.5}}{n}$

k = refers to following table:

Table 2.1: Values of k to be used in the above proposed equation

f_w (mm)	f_t/p_t	k*				
		d=125mm	d=150mm	d=200mm	d=250mm	d=300mm
75	1	1.51	1.55	1.62	1.65	1.68
	2	1.30	1.21	1.47	1.45	1.49
	3	1.10	1.02	1.40	1.31	1.39
100	1	1.45	1.46	1.61	1.64	1.53
	2	1.20	1.28	1.44	1.51	1.51
	3	0.97	1.10	1.31	1.43	1.49
125	1	-	-	1.57	1.64	1.53
	2	-	-	1.40	1.48	1.49
	3	-	-	1.20	1.35	1.39

*linear interpolation can be used for intermediate values of f_w and f_t/p_t

2.6.4 FEA Based Work by Tafheem (2012)

More recently, another research study was carried out on bolt tension of pipe joint by Tafheem, 2012. In her research work, a nonlinear finite element analysis was performed on flanged pipe joint under bending. The effect of various parameters such as pipe diameter, flange width, flange thickness and number of bolts had been observed on the bolt tension. For meshing, pipe is divided along length and along periphery. Individual division is rectangular. The load is applied to the flanged pipe joint in two consecutive load steps. In load step one; the initial clamping force has been applied at the bottom of all bolt shanks in order to simulate the actual condition. Actually the applied clamping force in a bolt is calculated as the product of applied clamping stress (which is 5% of the yield strength of bolt that is 90 ksi) and the effective tensile stress area of a bolt. In the second load step, the model is subjected to yield moment of steel pipe. In this model, the moment is applied by a pair of parallel and opposite forces representing a couple. In the following figure 2.20, finite element mesh of the flanged pipe joint with applied load has been shown. A typical deformed shape of the pipe joint due to bending is also shown in figure 2.21 to observe the behavior of contact elements.

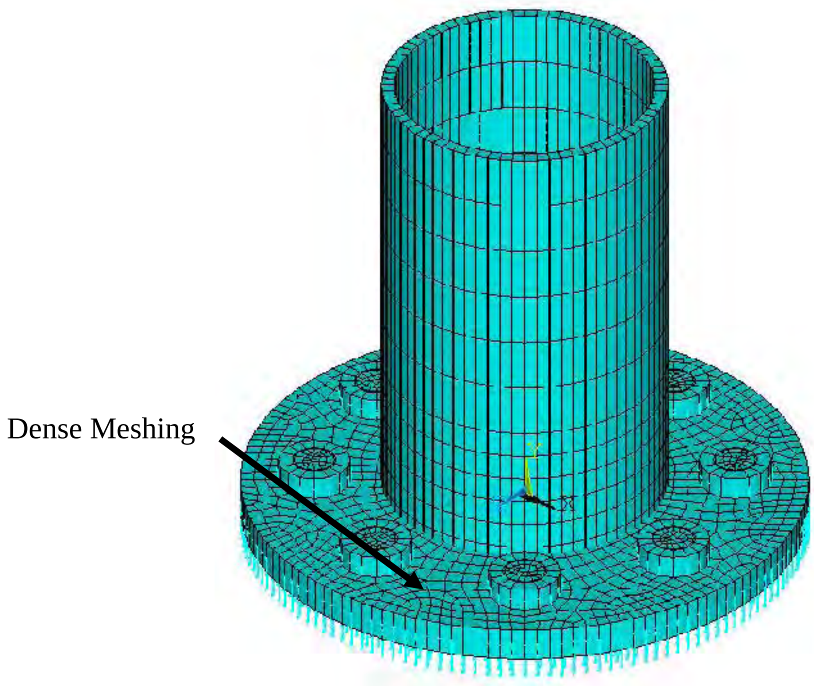


Figure 2.21 Finite element meshes showing all elements of the flanged pipe joint

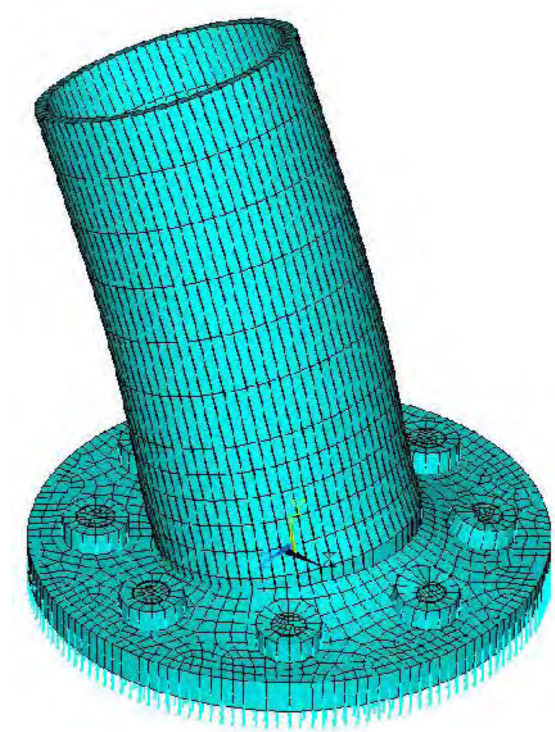


Figure 2.22 Typical deflected shape of flanged pipe joint

Proposed equation for determining bolt tension

***T* = Applied clamping force, *CF* +**

$$\frac{0.143\ n\,(f_t/p_t)\ f_w\ d_b\ d_p}{\{1+0.679\,(n)^{1.685}\}\{1+2.655(f_t/p_t)^{1.112}\}\{1+0.039(f_w)^{1.183}\}\{1+0.382(d_b)^{0.493}\}\{1+2.33(d_p)^{-0.471}\}} \qquad (2.7)$$

Where, Applied Clamping Force on bolt, $CF = \left\{0.75\left(\frac{\pi}{4}\right)\ {d_b}^2 f_y\ N\%\right\}$

The symbols used in above equation indicate the following parameters:

T = Maximum bolt tension in Kilo Newton

$\{0.75\left(\frac{\pi}{4}\right)\ {d_b}^2\}$ = Net bolt tensile stress area in mm^2

f_y = Yield stress of steel bolt = 90 ksi = 0.621 KN/mm^2

$N\%$ = % of yield stress of steel bolt for applied clamping stress = 5%

d_p = Nominal pipe diameter (50 mm $\leq dp \leq$ 304.8 mm)

p_t = Pipe wall thickness (4.75 mm $\leq p_t \leq$ 9.525 mm)

f_t = Flange thickness (2 to 4 times of pipe wall thickness p_t)

f_w = Flange width (50mm $\leq f_w \leq$ 150 mm)

n = Number of bolts (4 $\leq n \leq$ 22)

d_b = Bolt diameter (12.7 mm $\leq d_b \leq$ 25.4mm)

In this present study, only a single equation is put forward for different parameters within a specified range. Variables of the equation 2.7 are taken as pipe nominal diameter d_p, flange to pipe thickness ratio f_t /p_t, flange width f_w, diameter of bolt d_b and number of bolts n. In this research work, pipe length has been taken as twice the pipe's nominal diameter in case of all pipes, so pipe length is not incorporated in this equation. As a result, this equation fails to show the effect of variation in pipe length on bolt tension and in this way this equation lacks flexibility. Apart from that, the bolt head thickness is not included in the equation since this parameter changes with the bolt diameter.

2.6.5 Experimental Investigation of External Load on Flanges by Nash and Abid (2000)

David H Nash and Muhammad Abid, 2000 carried out an experiment on "Combined External Load Tests for Standard and Compact Flanges". The recognised standard method of gasketed flanged joint design contained within most pressure vessel codes is that based on the Taylor Forge procedure. This has, as its basis, bolt load calculations, which are designed to apply sufficient load to both seat and initialise the gasket, and to ensure sealing via a gasket when the operational pressure load is present. The flange ring and hub transmit the bolt load to the gasket and must therefore be stiff and flat. However, there are many real situations where additional loads arise through external pulling and bending. This is commonly seen in piping systems and other flanged pressure equipment.

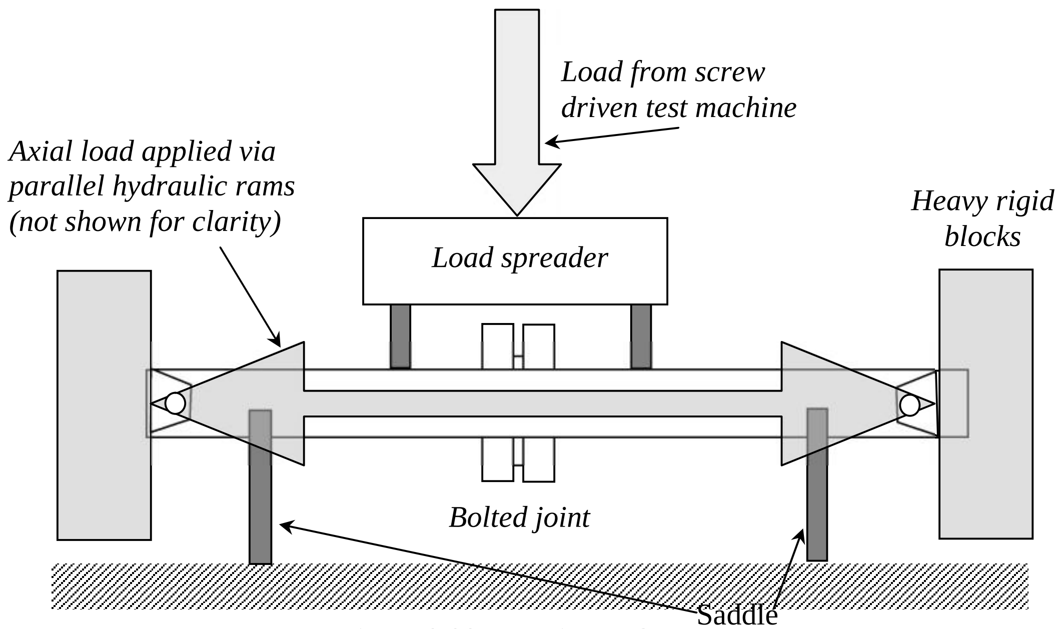
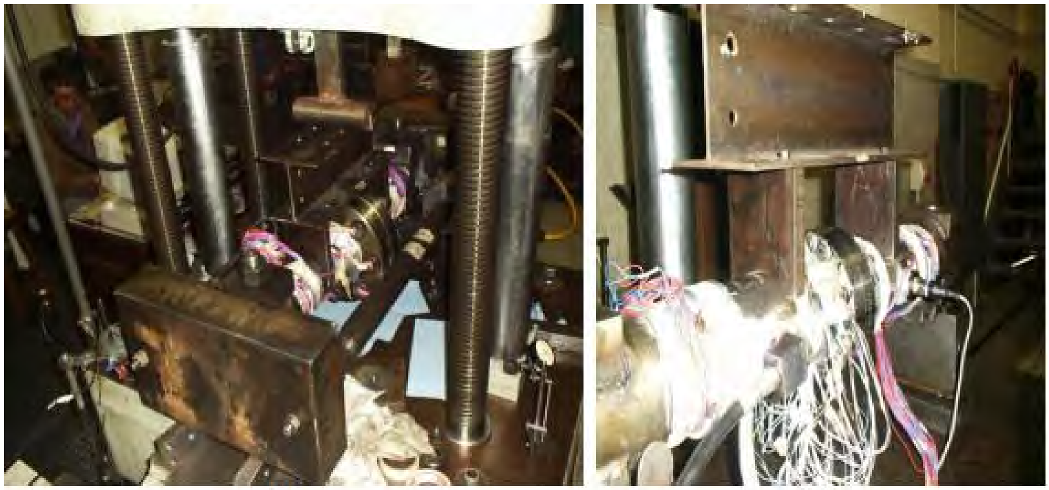


Figure 2.22 Experiment Setup

Although the codes do not specifically address the 'combined load' problem, the normal method for considering this additional load is to form an equivalent pressure. This over-pressure is calculated by making the stress generated in the pipe or vessel wall, by the external load, equal to a longitudinal pressure stress which may be tensile or compressive, depending on the nature of the load. This results in an over-pressure which can therefore be added to the operating pressure.

To examine the effect of differing combinations of externally applied load, a test rig was designed to work in tandem with an existing Instron testing machine. A number of bolted flange assemblies examined including standard ANSI joints and compact VERAX VCF joints (Figures 2.23). Loading is applied using a combination of



(a) (b)

(a) (b)

systems incorporating a pump, rams and a screw driven test machine. The application of the tensile and four point bending loads is shown in figure 2.22.

Pressure loading is applied to the assembled joint via a manually operated hand pump, with a 500 Bar capacity. Pressure gauges on the pump and on the assembly record the internal pressure. Axial tension load is applied via two symmetric parallel shafts loaded by hydraulic cylinders. This tensile load is transferred to the pipe by the use of heavy end plates and a pin-type connector, which locates the assembly and the loaded shafts. The end plates are deemed rigid enough to transfer the load from the shafts to the pipe assembly. However, these were strain gauged as a precaution, and load levels monitored. Four point bending was achieved by the use of the testing machine cross head together with a custom-built load applicator. This arrangement applied load to the upper portion of the joint using a load spreader device and this reacted by two frictionless loose saddles, which allowed the joint to rotate in the axial plane. Tests have been carried out using hydraulic fluid as the main pressurizing medium.

2.7 REMARKS ON PREVIOUS WORKS

2.7.1 Remarks on the Work by Chowdhury (2008)

No AISC specified standard pipe wall thicknesses were used for the pipe modeling by Chowdhury (2008). Rather a constant pipe thickness (i.e. 7.112 mm for all pipes), irrespective of pipe diameter, was used in his modeling. Another important thing is that the bolt holes were not made on the flange**.** In addition to that, no bolt head and bolt shanks were created for the bolt modeling.

The variation of diameter of bolt had not been used in his analysis which limited the scope of the result outputs. In addition, the effect of initially applied clamping forces on bolts was not taken into account which could make the whole study more realistic. Next, he considered only 50 mm flange width for comparatively smaller diameter pipes and 50 mm & 75 mm flange width for comparatively larger diameter pipes which did not provide adequate tool clearance and edge distance of flange for construction. In case of proposed equations, it would be more convenient in use if only one generalized equation instead of five had been developed for a certain range of different parameters. In spite of such limitations, the research work is a very significant one since few research works were carried out related to the evaluation of bolt tension of flanged pipe joint.

2.7.2 Remarks on the Work by Azim (2010)

The most important thing is that, though bolt holes were made on the flange in the work of Azim (2010) but no bolt head and bolt shanks were created in his bolt modeling. Beside that only one bolt diameter (20mm) was used in his analysis which limits the scope of the result outputs. In addition, the initial clamping force on the bolts was not applied at the foot of the contact lines at bolt locations. Rather in his model, the bottom of all contact lines was restrained in all directions which did not represent the actual condition.

In spite of having a number of limitations, this research work is a very significant one since some limitations of the previous researcher Chowdhury, 2008 had been tried out to minimize in a practical way. For instance, in the research study of Azim (2010), AISC specified pipe thicknesses had been used for respective pipe diameters. In addition to that, proper tool clearance and edge distance on flange had been maintained while modeling the pipe joint structures for construction.

2.7.3 Remarks on the Work by Tafheem (2012)

This research work is a very significant one since some limitations of previous two research works had been minimize in a practical way. For instance, in the work done by Chowdhury, 2008 no AISC specified pipe thicknesses had been used for respective pipe diameters. In addition to that, a constant pipe thickness, irrespective of pipe diameter, was used in his modeling. Also the bolt holes were not made on the flange**.** Then, in the work of Azim, 2010 no bolt head and bolt shanks were created in his bolt modeling. Beside that only one bolt diameter (20 mm) was used in his analysis which limits the scope of the result outputs. In addition, the initial clamping force on the bolts was not applied at the foot of the contact lines at bolt locations. All limitations of previous works are taken into consideration by Tafheem in her study. Moreover, one simple formula is proposed basing on FEA for various configuration of flange joint like pipe diameter, pipe wall thickness, flange width, flange thickness, bolt size and diameter etc.

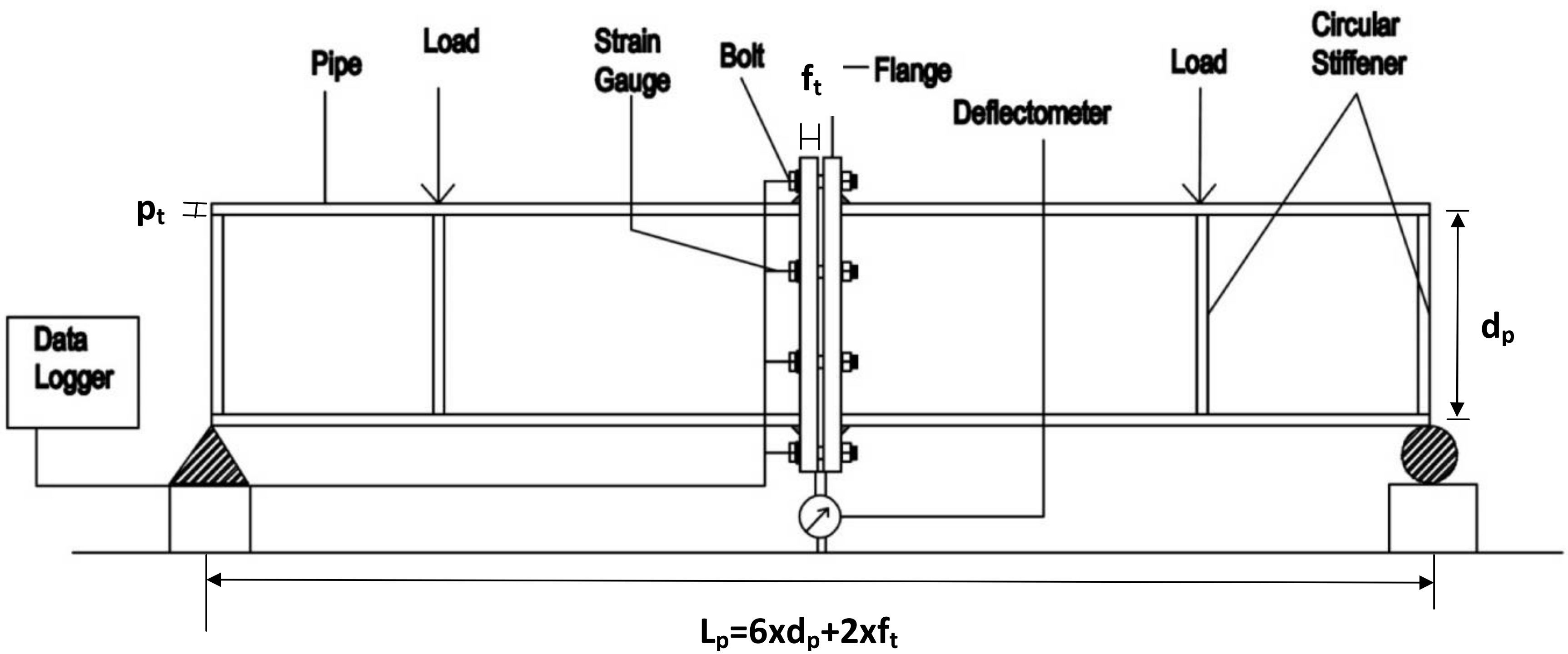


Figure 3.6: Typical pipe joint specimen and loading scheme

CHAPTER 3

MATERIAL PROPERTIES AND EXPERIMENTAL SETUP

3.1 INTRODUCTION

Flexural capacity of steel pipe joint largely depends on the pipe diameter, pipe thickness, flange thickness, size and number of bolts. In many countries, lot of efforts are undertaken to get new code standards for the calculation of flanged pipe joints under various loads. Most of the works are based on finite element analysis. More recently, Tafheem, 2012 in her study developed a finite element model to describe the real behavior of the pipe joint. On the other hand, very few experimental works were carried out to determine bolt force in flange pipe joint subjected to bending. As such, this experimental investigation aimed at determining bolt tensile forces of a flanged pipe joint subjected to bending. This chapter presents mechanism in flanged joint subjected to bending, research strategies, material properties, experimental setup, specimen preparation, equipment used for data acquisition and processing.

3.2 MECHANISM IN FLANGED PIPE JOINT DUE TO APPLIED MOMENT

In a structural pipe frame system, some pipes are connected with flanges which are then bolted together in order to make a strong joint. A typical front view of a flanged pipe joint has been shown in figure 3.1.When a flanged pipe joint is subjected to bending due to applied moment, the whole structure tends to bend. The deflected shape of the pipe joint due to bending has been shown in figure 3.2. From the figure, same deflected shape has been observed at both sides of the line of symmetry. In addition, no side of the flanges crosses the line of symmetry. Actually when the pipe joint is subjected to bending, some stresses are developed on the flange. The bolts connected to the flange take most of the tension as the flange is very much weak in tension.

3.3 RESEARCH STRATEGIES

The objective of the research is to experimentally investigate bolt tension of flanged pipe joint subjected to bending. For experiment purpose, few selected pipe (125 mm, 150 mm and 200 mm diameter) are used with flanges of varying thickness and different number of

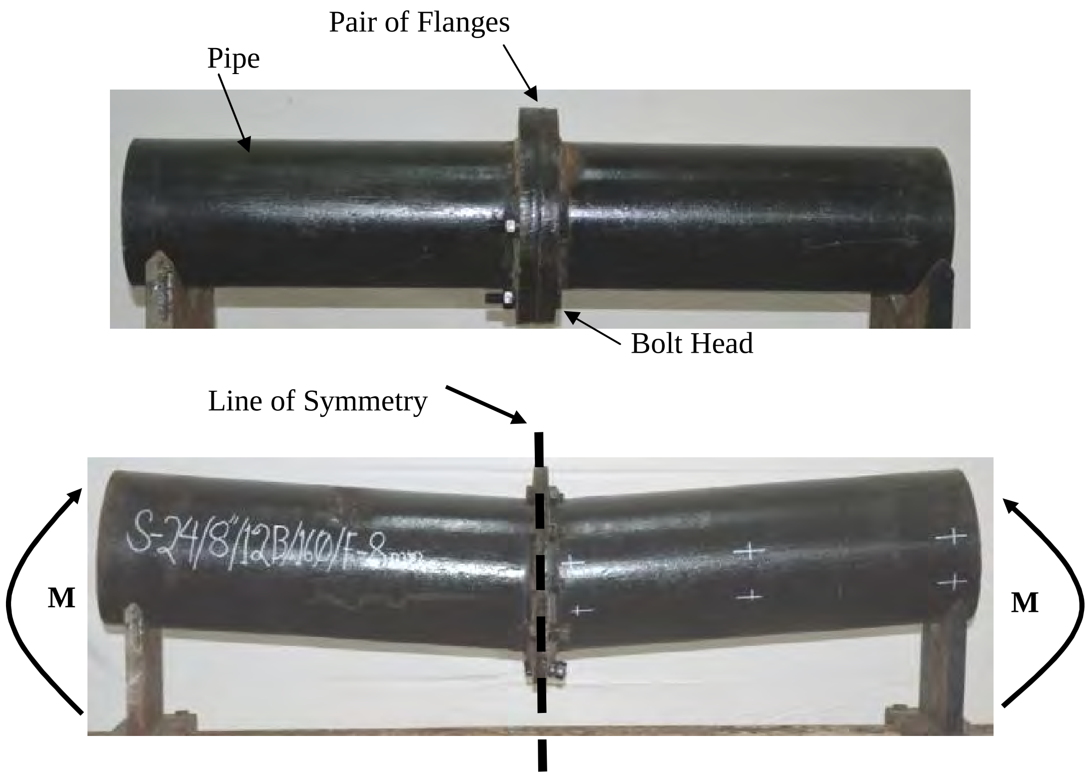


Figure 3.1: Front view and deflected shape of a flanged pipe joint specimen with 8 bolts

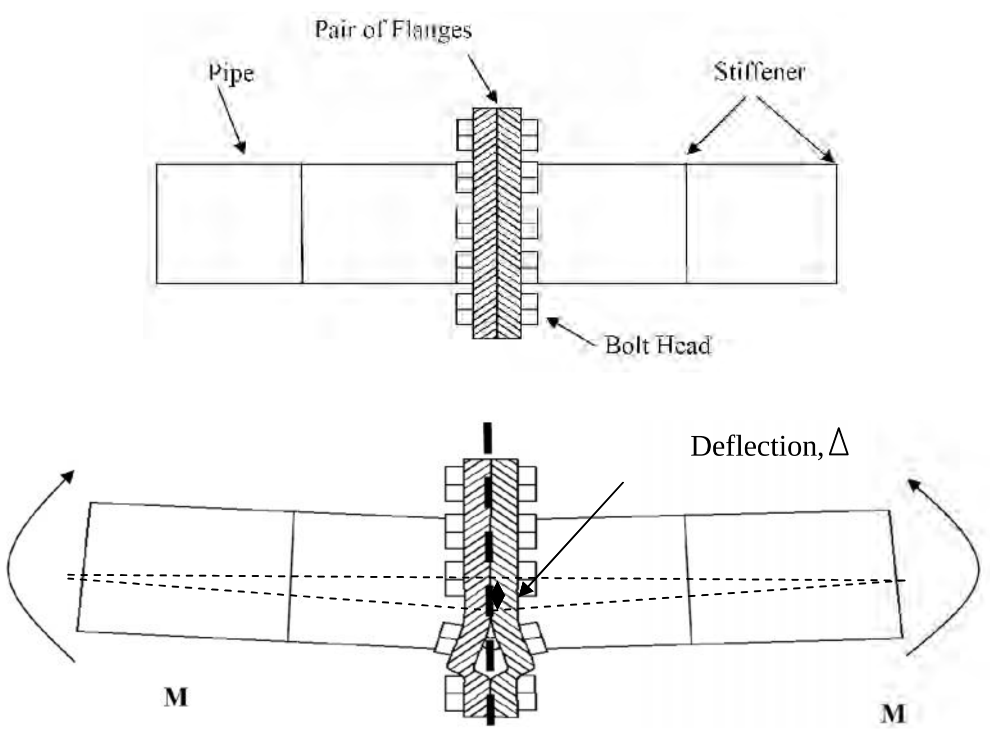


Figure 3.2: Front view and deflected shape of a schematic flanged pipe joint

bolt configuration. To apply moment at joint a custom made two points load spreader is used in UTM. For data acquisition two alternative means is used. Firstly, special strain gauge is used inside the bolt shank and data logger is used to record the strain readings. Secondly, Digital Image Correlation (DIC) method is used to measure the deflection. For this, video extension meter with specialized software is used. Finally a correlation between applied moment and bolt tensile force is established.

3.4 MATERIAL PROPERTIES

3.4.1 Pipe

For construction of steel pipe joint mild steel seamless pipe of ASTM A 53 specification is used. Material, chemical and mechanical properties of the pipe is shown in Table 3.1 and 3.2 respectively.

Table 3.1: Material properties of steel pipe

Ser	Nominal Pipe Size	Outside Dia	Wall Thickness	Nominal Weight		Weight Class	Test Pressure			
							Grade A		Grade B	
	in	mm	mm	lb/ft	kg/ft		psi	kPa	psi	kPa
1.	5	141.3	6.55	14.62	6,64	STD	1670	11510	1950	13440
2.	6	168.3	7.11	18.97	8.61	STD	1520	10480	1780	12270
3.	8	219.1	8.18	28.55	12.9	STD	1340	9240	1570	10820

Table 3.2: Chemical and Mechanical properties of steel pipe

Specification/ Classification		Chemical Composition				Mechanical Properties			
		C	Mn	P	S	Tensile Strength(Min)		Yield Point(Min)	
ASTM A53	A	0.25	0.95	0.05	0.045	48ksi	415MPa	30ksi	205MPa
	B	0.30	1.20	0.05	0.045	60ksi	330Mpa	35ksi	240MPa

3.4.2 Flange

In general term, flange refers to a circular disk of metal surrounding open end with several fixing holes for connecting the other things. So the flange is a disk-shaped parts, the most common in the pipeline project, the flanges are used in pairs. For experiment purpose flange with 6 mm to 24 mm thickness and 30 mm to 48 mm width are used. Number of holes for bolt varied from 8 to 12 holes (Figure 3.3).

(a) (b)

Applied load condition	Nominal strength per unit area, F_n (MPa)
Tension	620

3.5 EXPERIMENTAL SETUP

To investigate the behavior of bolted flanged pipe joint subjected to bending following procedure is followed:

a. For experimental investigation total 36 specimen flange pipe joints with varying parametric conditions, like different flange thickness, flange width, number of bolt and different diameter of bolts etc is used.

b. To observe the effect of various parameters on bolt tensile force BTM-6C strain gauge is embedded into a hole drilled at the center of bolt shank with A-2 adhesive. Bolt strain gauge reading is measured using data logger.

c. Bolted joint of the sample is subjected to two points loading by custom made load spreader to develop pure moment using UTM machine. Deflection of pipe joint is measured using deflectometer and video extensometer.

d. Stored data of strain gauge and video extensometer shall act as a basis for developing a reliable design guideline for bolted flanged pipe joints subjected to bending. General experimental setup is shown in figure 3.6.

3.6 ALLOWABLE STRESSES AND FLANGE JOINT CONFIGURATION

Allowable stresses and material properties for flange, pipe and bolt are given in Table 3.4. Bilinear kinematic hardening for elasto-plastic material properties is used during analysis. A bilinear material model consists of two sections each having a linear gradient. For the first section, an elastic material is used which is valid until the yield stress and the gradient of this section is the Young's Modulus of Elasticity (E). The second section functions beyond the yield stress, and gradient (plastic modulus) is 10% of the Young's Modulus of Elasticity. A flange with varying thickness from 6 mm to 24 mm and number of bolts varying from eight to twelve with 10 mm to 16mm diameter is used.

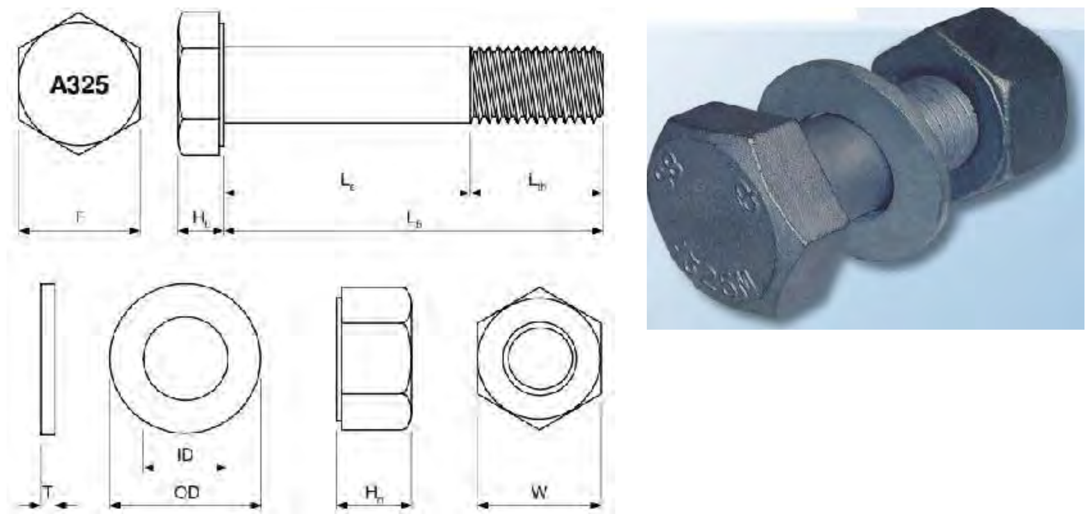
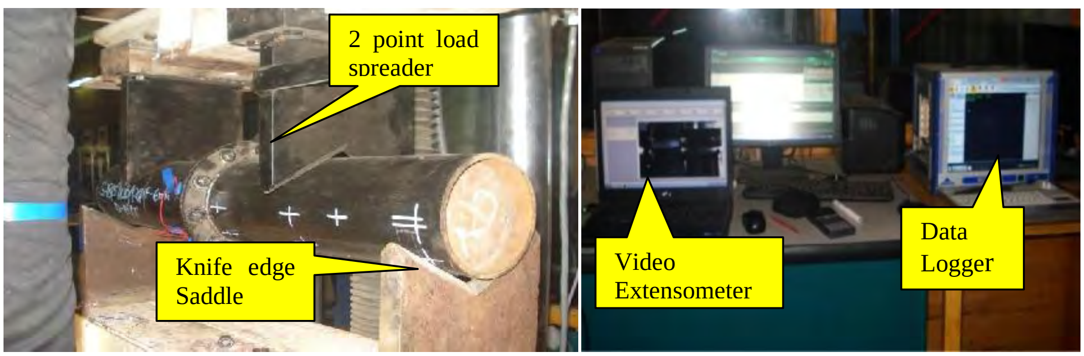


Figure 3.4: Nomenclature for Heavy Hex Bolt Dimensions



(b) (c)

Figure 3.5: (b) Side view of a simply supported specimen with two points loading system, (c) Data acquisition by UTM machine, video extensometer and data logger.

Table 3.4 Allowable stresses and material properties for flange pipe and bolt

Non-gasketed Joint	E (MPa)	υ	Stress (MPA)	Thermal Conductivity K(W/m-k)	Co-eff of Thermal Expansion α (m/m-k)
Pipe (ASTM A 508, Grade III)	191670	0.3	155(0.4σ_y)	47	12.5E^{-6}
ASTM A325M Bolt	210000	0.3	640	38	14.1E^{-6}
Washer	210000	0.3	640	38	14.1E^{-6}

Material Properties and Experimental Setup

3.7 DESIGN OF PIPE JOINT SPECIMEN

The steel pipe joints are designed employing the ASTM A508, Grade III specification. Material properties for flange are as per ASTM A105, ASME 1998a & b, for the bolt and washer is as per ASTM A325M.

3.7.1 Size and Geometry of Pipe Joint Specimen

The test specimens are comprised of 34 steel pipe joints. Three basic diameter of pipe are selected for experiment, those are 125mm, 150mm and 200mm.

Length of pipe joint specimen = 6 x Diameter of pipe + 2x Flange thickness.

Diameter of flange = 3 x bolt diameter.

To determine bolt forces under certain range of parametric conditions like flange thickness, flange width, number of bolts and diameter of bolts etc are varied. A list of different size and geometry of pipe joint specimen are given below:

Table 3.5: Size and geometry of pipe joint specimens.

Ser	Pipe nominal diameter, mm	Pipe thickness, mm	Flange Width, mm	Flange Thickness, mm	Size and number of bolts
1.	125	6.55	30, 36	6, 12,18	8,10(10&12mm)
2.	150	7.11	36, 48	7, 14, 21	10, 12(12&16mm)
3.	200	8.18	36, 48	8, 16, 24	10, 12(12&16mm)

3.8 SPECIMEN PREPARATION

3.8.1 Cutting, Refacing of Steel Pipe

All pipe of 6 meter standard length is cut into required length using gas cutter and mechanical saw machine. To remove the rough edge due to gas cutting grinding is done in machine shop. Circular disks of 10 mm thickness are welded at mid point and end.

3.8.2 Welding

For preparing flange steel plates of 6mm to 24 mm thickness are used. Gas cutter is used to cut the flanges out of steel plate. Then drill machine is used to prepare the holes. The diameter of hole is 1/16 in. larger than the diameter of bolt. For welding flanges with pipe ASTM A105 is followed (Table 3.6).

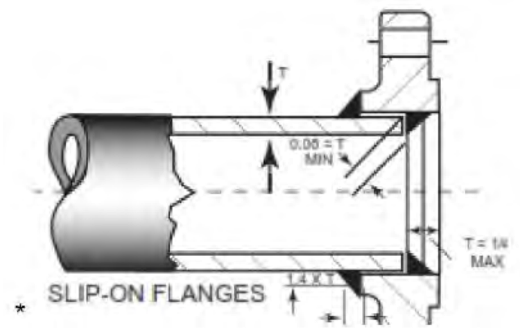
(a)

(b)

(c)

(d)

	ASTM Specification A105	
Grade Symbol	III	IV
Type of Steel	Carbon	
Chemical Composition	C-0.35 Max	
Minimum Physical Requirements	TS-60,000,YP - 30,000	TS-70,000,YP - 36,000
Heat Treatment	Hot forged with finishing temperature above upper critical. Cooled in still air	



Type	Gauge (mm)		Base (mm)		Gauge Center		Resistance (Ω)	Hole Diameter (mm)
	Length	Width	Length	Width	a	b		
BTM-6C	6	1	12	1.7	5	7	120	2.0

embedded a clearance between the end of the gauge and the bottom of hole should be 3~5mm. Before inserting gauge we have to clean the inside (inner surface) of the bolt with solvent such as acetone. If any moisture, grease, rust, solvent, etc. remains in the hole, the adhesive may not cure well. After preparation, mount bolt strain gauge before the prepared surface makes oxidizing membrane or is not contaminated. Determine an embedment position precisely and measure the embedment length. Mark the gauge leads according to the length. Bend the gauge leads rectangularly at the mark without injuring the insulation material (Figure 3.8).

3.8.5 A-2 Adhesive

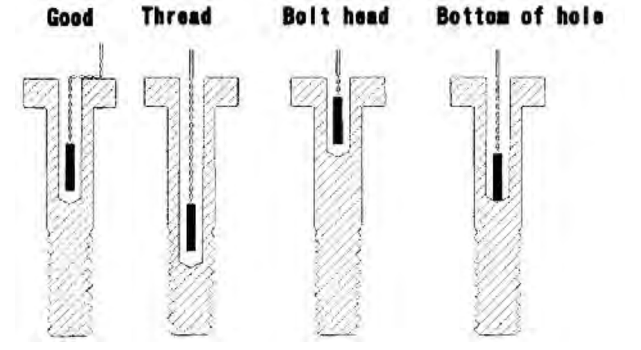
The A-2 is two-component heat curing epoxy adhesive which consists of drug A (main agent) and drug B (hardener) in each tube. The mixing ratio of A-2 is Drug A:B = 10:1 in weight. Mix both drugs well with a spatula, etc. to offer uniformity. If air bubbles are mixed while mixing, heat the A-2 at 50 ~ 60°C for about 30 minutes using a heat gun etc. to reduce the viscosity and remove the bubbles as much as possible. The mixed A-2 is in service for approximately 2- 3 hours.

3.8.6 Mounting Strain Gauge and Curing

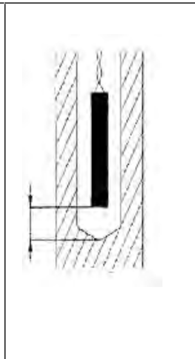
Enter the mixed A-2 into a syringe to which an exclusive needle is attached. Insert the needle carefully into the bolt hole so as the needle to hit the bottom of the hole. Fill the hole fully with the A-2 while pulling the needle up. Adjust the speed of pulling the needle and also of injecting the A-2 in order not to remain air bubbles in the hole. Apply the A-2 adhesive to the surface of the BTM gauge and insert it into the hole. Keep the bolt in upright position and allow the A-2 adhesive to cure for 12 hours at room temperature. Cure the A-2 with bolt in upright position at 140°C for 3 hours. Be sure to raise the temperature from room temperature. Quick temperature elevation may cause air bubbles or cracks.

3.9 TESTING AND DATA ACQUISITION

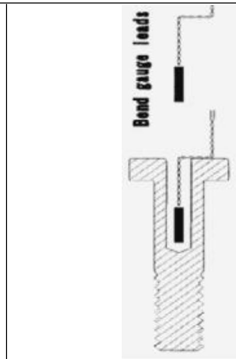
The experimental work consisted of testing 36 simply supported bolted flanged pipe joints. The specimens are tested under two points loading system (Figure 3.10).All specimens are tested using a 2000 KN universal testing machine (UTM) at Structural Mechanics Laboratory of Bangladesh University of Engineering and Technology.



a)

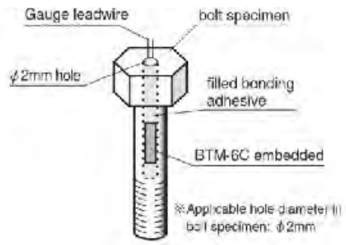


(b)



(c)

Figure 3.8: Mounting BTM-6C sensor in bolt (a) Correct drilling of bolt, (b) Position of sensor from bottom of hole, (c) Correct position of sensor lead wire.



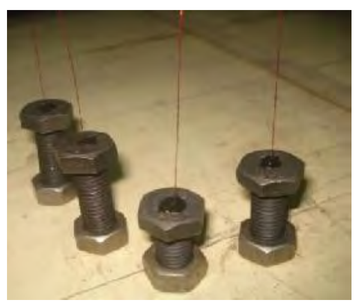
(a)

(b)

(c)

(d)

(e)



(f)

Figure 3.9: (a) Schematic diagram of a bolt with sensor, (b) Sensor BTM-6C,
(c) Bolt with 2mm hole and A2 Adhesive in syringe,(d) Preheating of bolt and adhesive,
(e) Inserting adhesive inside hole, (f) After heat curing of bolt with sensor inside.

The machine is also equipped with sophisticated computer control and data acquisition system (Video Extensometer). The specimens are tested under pure axial compression and the displacement rate maintained is 03 mm/min. 25 mm thick steel plate with knife edge v notch is used as simple support. Similar arrangement is used to apply two point loads from the top at midpoint between flange joint and support. Axial load and vertical displacement data are recorded from the load cell of a computer controlled UTM. But the deflection history is recorded by using video extensometer which is attached to computer. For measuring strain value of bolts another video camera is used, and then by post processing using video extensometer we obtain strain values. Failure pattern of specimen is obtained from high definition video recorded during the test. Also 3 TML BTM-6C sensors are embedded inside steel bolts for 24 specimens. These sensors are connected to Data Logger 2601. It recorded strain data of bolts.

3.9.1 Data Logger 2601

A data logger is an electronic device that records data over time or in relation to location either with a built in instrument or sensor or via external instruments and sensors. They generally are small, battery powered, portable, and equipped with a microprocessor, internal memory for data storage, and sensors. Data logger 2601 is interfaced with a personal computer and utilizes software to activate the data logger and view and analyze the collected data. One of the primary benefits of using data loggers is the ability to automatically collect data on a 24-hour basis. Data loggers are typically deployed and left unattended to measure and record information for the duration of the monitoring period. This allows for a comprehensive, accurate picture of the environmental conditions being monitored, such as air temperature and relative humidity. Given the extended recording times of data loggers, they typically feature a mechanism to record the date and time in a timestamp to ensure that each recorded data value is associated with a date and time of acquisition in order to produce a sequence of events. As such, data loggers typically employ built-in real-time clocks. Data logger 2601 is 20 input multi-channel instruments.

3.9.2 Video Extensometers

3.9.2.1 Camera System

All the measurements and outputs from the Video Extensometer are time stamped and the uncompressed video output from the camera can be recorded for post-test. Tinius Olsen’s

(a)

(b)

(c)

(d)

(e)

(f)

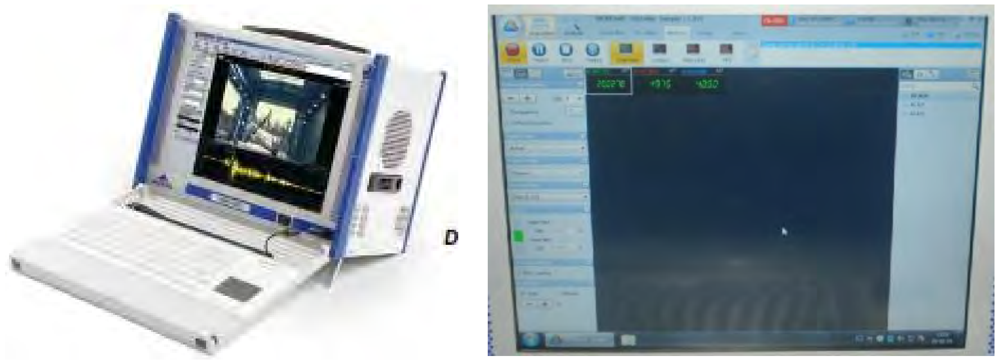
Video Extensometer uses a high resolution monochrome camera and advanced high speed image processing such that its point- to-point real-time video processing technology is capable of achieving accuracy for both low and/or high strain materials. The Video Extensometer can be further enhanced by using multiple camera systems to measure specimen strain.

3.9.2.2 Lighting and Marking of Targets

The extensometer is more than capable of following chosen targets in regular daylight conditions; however using additional lighting prevents any tracking loss of target as a result of changes to ambient lighting conditions. Any visible marking can be used for pattern recognition, and these can be natural patterning on the specimen surface, pen marks, blob markers, punched gauge marks or a spray paint speckle pattern. These targets can be defined by the user, allowing setting these to any gauge length as required.

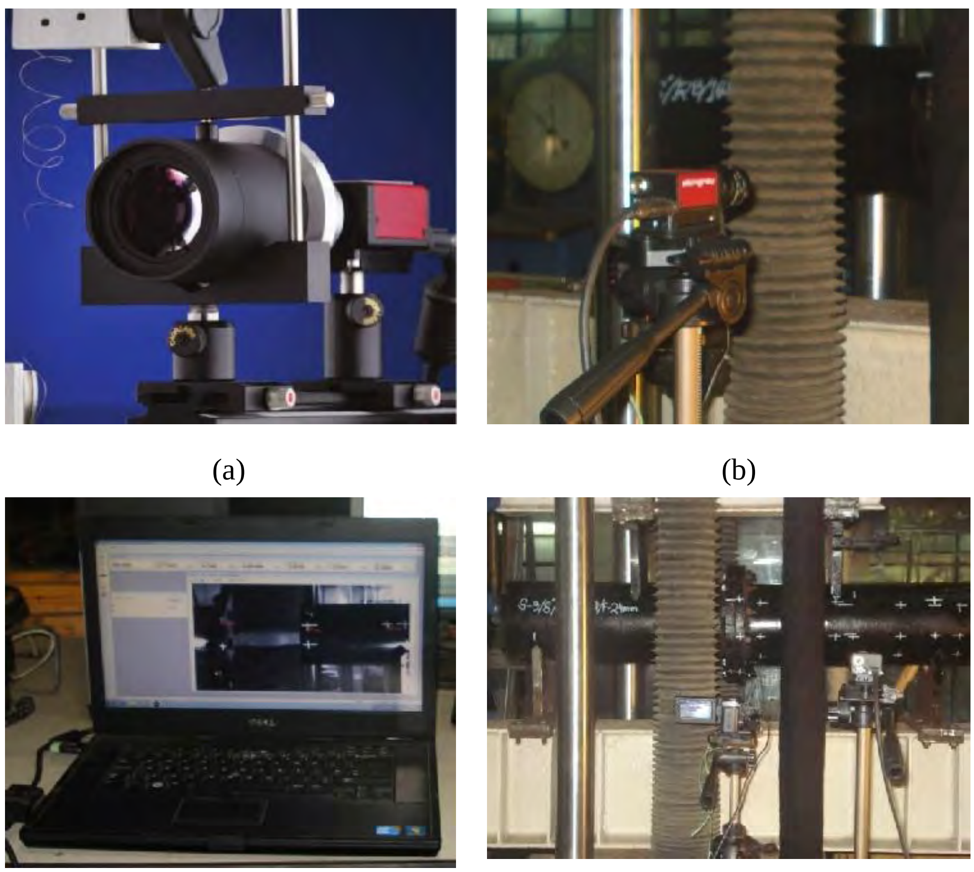
3.9.2.3 Software

Tinius Olsen has developed Horizon, a comprehensive software program that makes testing simple, precise, and efficient. Our Video Extensometer software monitors the object under test. By fitting an appropriate lens to the high resolution cameras, the Video Extensometer will measure objects smaller than 1 mm to larger than 100 m. Tinius Olsen’s Video Extensometer software uses patented technology to precisely measure 2D position of targets in images from the video camera; special targets are not required. As the specimen is tested, the software tracks the point-to-point movement of these targets from camera frame to frame, and strain data is calculated in real time. The high system resolution required to calculate these results is achieved using subpixel interpolation algorithms and with which the system can resolve to submicron levels of movement. The system can precisely measure each target’s position in every image from the video camera. Up to 100 targets can be measured in real-time at 15 Hz. From the measured positions of the targets, the system can calculate displacement, velocity, acceleration, angular rotation, 2D Strain. Setup of video extensometer is shown in figure 3.12.



(a) (b)

Figure 3.11 (a) A Data logger 2601, (b) Software outlook of data logger 2601.



(a) (b)

(c) (d)

Figure 3.12: (a) Camera with high resolution materials testing lens, (b) Camera with a general purpose lens, (c) Software outlook, (d) Additional gen purpose video camera used to measure bolt strain.

3.10 Specimen Designations and Legends

A special designation is used for the entire specimen flange pipe joints. The designation system provides the information about pipe size, number of bolts, bolt diameter and flange thickness. The specimen designation is shown in Figure 3.13. Details of all specimens are given in appendix A.

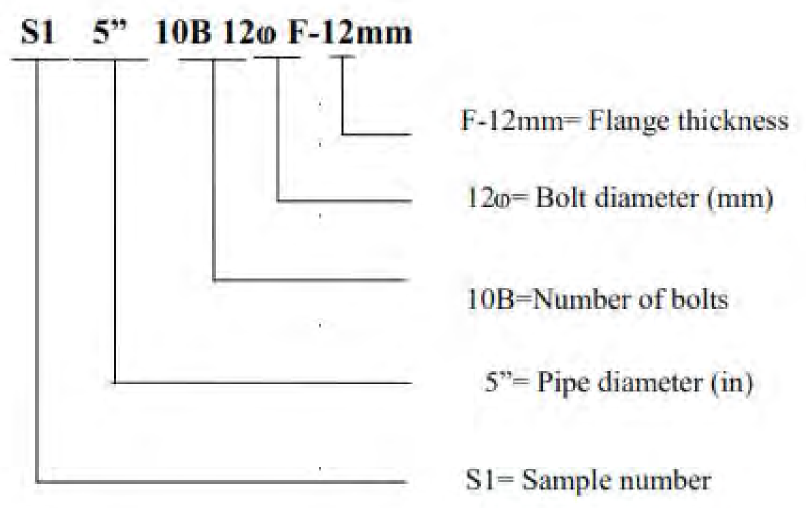


Figure 3.13: Specimen designation

CHAPTER 4

EXPERIMENTAL INVESTIGATION RESULTS OF FLANGED PIPE JOINT SUBJECTED TO BENDING

4.1 INTRODUCTION

Detailed experimental investigation and data acquisition procedure of a flanged pipe joint subjected to bending is described in the previous chapter. In this chapter, effect of various controlling parameters on bolt tension of a flanged pipe joint is described with supported data and graphs.

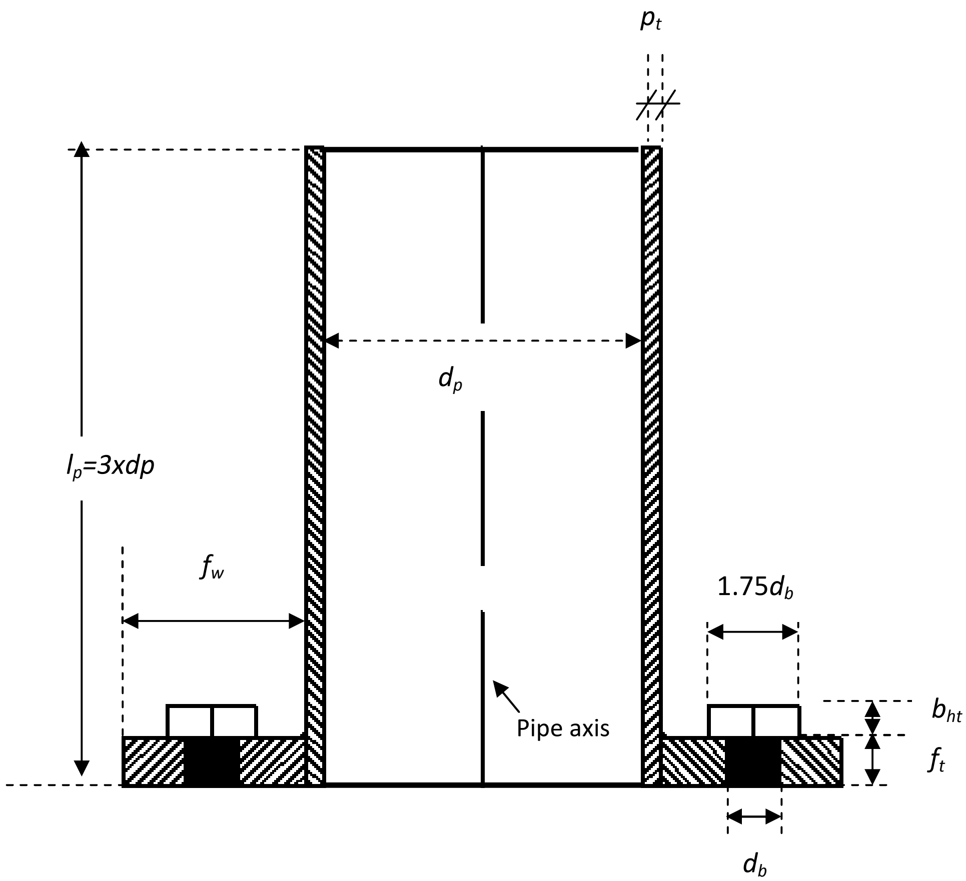
4.2 STUDY PARAMETERS

The study parameters of bolted flanged joint specimen under investigation are described in the following Table 4.1. While one of the parameters is varied, the other values of parameters are maintained equal to the reference values as mentioned below. Bolt tensions are measured for different flange thickness, flange width, diameter of bolt and number of bolts for different diameter of pipes. For each category of pipe diameter, the maximum value of bolt tension is considered.

Table 4.1 Study parameters for a flanged pipe joint

Parameter	**Symbol**	**Variable Data**
Nominal diameter of pipe [*AISC specified nominal pipe	d_p	125 mm, 150 mm and 200 mm
Pipe wall thickness [*AISC specified pipe thickness for above diameter pipes]	p_t	6.55 mm, 7.11 mm and 8.18 mm
Pipe length	l_p	3 times of nominal diameter of pipe ($l_p=3d_p$)
Flange thickness	f_t	1 to 3 times of the pipe wall thickness [$f_t = 1p_t$; $f_t = 2p_t$; $f_t = 3p_t$]
Flange Width	f_w	30 mm, 36 mm and 48 mm
Nominal Bolt Diameter [**RCSC specified bolt diameter]	d_b	9.75 mm, 12.7mm and 15.875mm
Number of bolts	n	8, 10 and 12


3 d_b
1.5d_b
1.5d_b



p_t
d_p
l_p=3xdp
f_w
1.75d_b
b_ht
f_t
Pipe axis
d_b

(a) (b)

Figure 4.3 (a) Tinious Olsen universal testing machine (UTM), (b) Direct tension test of bolt and Control panel for monitoring UTM operation the bolt is done using special coupling holder. Also 3 TML BTM-6C sensors are embedded inside steel bolts for each specimen. These sensors are connected to Data Logger 2601. It recorded strain data of bolts.

4.4 FORCE DISTRIBUTION CHARACTERISTICS OF BOLT

ASTM A325M steel bolts have been used for bolting the pipe joint. The diameter of the bolts of for flanged pipe joint has been selected according to RCSC. It is of utmost importance that bolts of adequate diameter should be provided in the pipe joint. Inadequate bolts may result in overstressing of the bolts at low loading levels and will possibly lead to brittle failure by fracture of the bolts. It is thus imperative that bolts are not stressed to their ultimate stress levels even at the ultimate load capacity of the pipe joint. Bolt fracture should be avoided at all cost. During experiment mainly two types of bolt failure is observed, those are thread slip failure and rupture of bolt (Figure 4.4). From the economic point of view, the bolt diameters should be such that the bolts are effectively used and not over designed to the extent to be considered uneconomic. We observed from conventional force distribution method (figure 2.15), that compressive forces are developed on the upper bolts and tensile forces are developed on lower bolts; and no reaction forces are developed on bolt located at midsection. During experiment we observed that maximum tension develops on extreme bolts, but no compression force develops on other side extreme bolts. From conventional analysis of determining bolt tension method, it is assumed that there is no effect of flange thickness on bolt tension, moreover all forces due to bending are taken by bolts and compressive force develops on upper bolts. But in reality, we observed that compressive forces in upper flanges are transferred from one part to another by contact stress and bearing stress. Therefore in reality, the bolts on compressive side does not experience any force due to applied moment. Slenderness effect of the flanged pipe joint specimen has no significant effect here, because pipe is subjected to pure bending without axial force and shear, moreover having circular cross section. The distribution of bolt reaction forces among bottom and side bolts of the flanged pipe joint (pipe diameter of150 mm, flange width of 36 mm, bolt diameter of 12 mm, flange thickness = 1×pipe thickness, number of bolt of 10), which is subjected to bending due to applied moment of 17.11 kN.m, has been shown in figure 4.5.

4.5 CALIBRATION OF STRAIN GAUGE

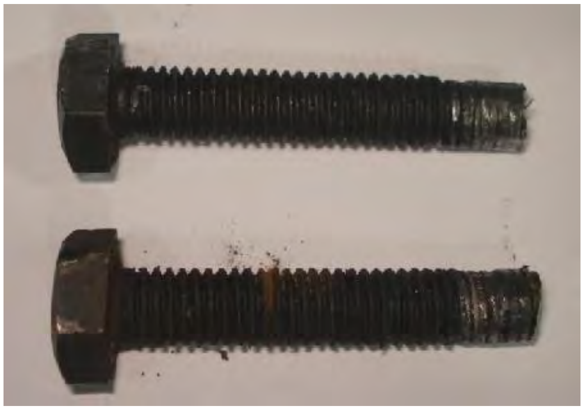
For measuring strain value inside bolt we have used BTM-6C which provided the strain value in mm/mm. For converting this into actual bolt force, we needed to carry out calibration test. For all different type of bolt we conducted direct shear test (Figure 4.3 (b)) for three bolts with strain gauge inside. From which we generated actual load vs bolt strain graph as shown in appendix A. From actual load vs strain we developed the equation to convert strain value into actual force using trend line. For 10 mm, 12 mm and 16 mm diameter bolt this calibration test is conducted. During test all strain values as recorded in data logger are converted into bolt force using this calibration chart of strain gauge.

4.6 DEFLECTED SHAPE OF SPECIMENS

The deflected shapes obtained during experiment resembles similar deflected shape pattern as predicted by Tafhem (2012) in her research paper using FEA model. From all deflected shape it's clear that deflection is dependent on mainly pipe diameter and flange thickness. We also observed that with increase in flange thickness the deflection of specimen gets reduced, since thicker flange means stiffer flange which is capable of carrying more loads. Thus, in case of thick flange we observe less bolt tension than thin flange, since some of the loads are shared by flange. Figure 4.6 shows the deflected shape of a specimen flanged pipe joint. Rest of the deflected shapes is shown in appendix B at the end of this paper.

4.7 PROCESSING OF RAW DATA

For preparing bolt tension vs applied moment graph, data are collected from three different sources. Those sources are load cell of UTM for calculating applied moment, video extensometer for measuring deflection and data logger for recording gauge value of bolt shank. In UTM load cell, data are recorded in the format of time (second) vs load (kN) for each second. Similarly, in data logger machine bolt strain is recorded in the format of time (second) vs strain value in mm/mm. Now strain values are converted into bolt tension (kN) with respect to time, similarly applied load (P) is converted into moment ($M=P/2*l_1$). Then a bolt tension vs applied moment



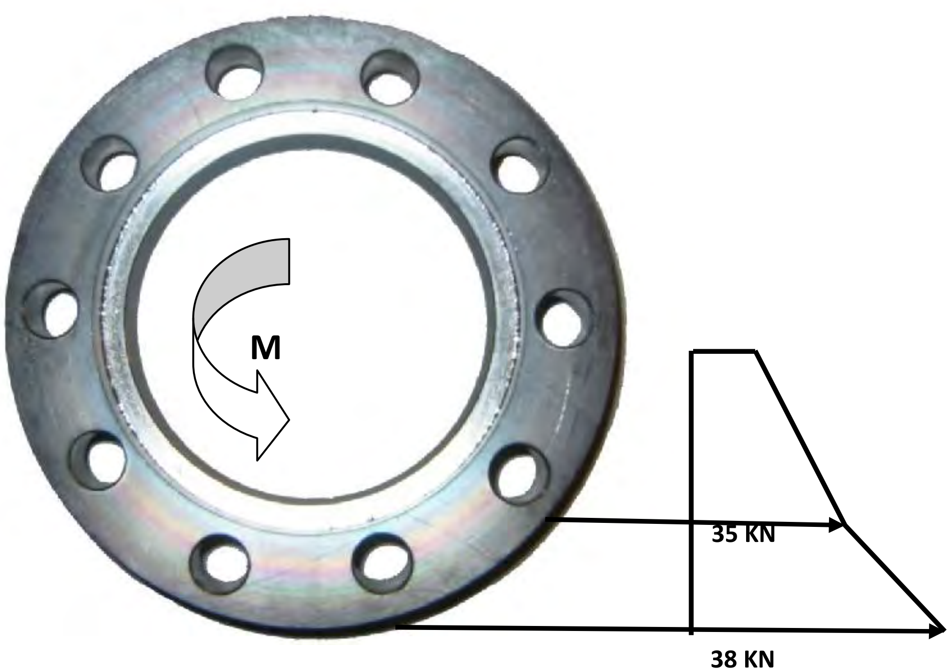

M
35 KN
38 KN

Figure 4.6 (a) Deflected shape of a 200 mm diameter pipe with 16 mm flange thickness (S-26) showing effect of flange thickness on deflection

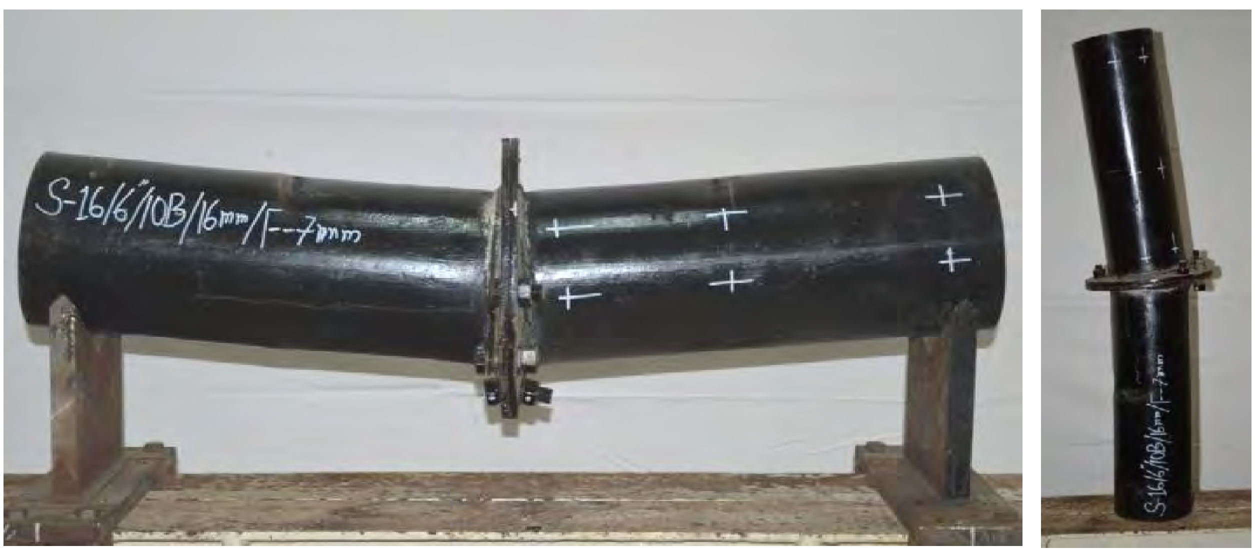


Figure 4.6 (b) Deflected shape of a150 mm diameter pipe with 7 mm flange thickness (S-16) showing effect of flange thickness on deflection

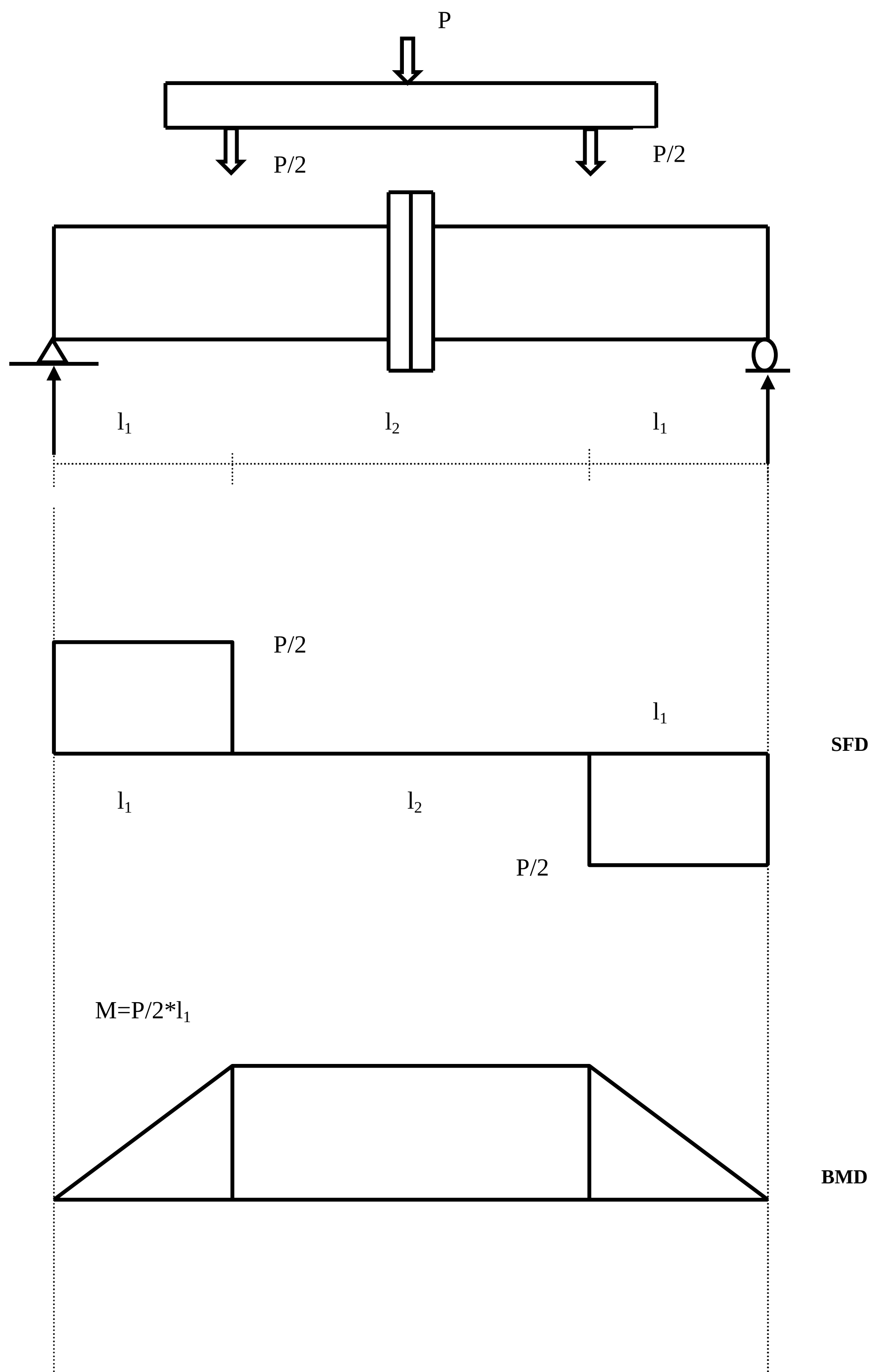


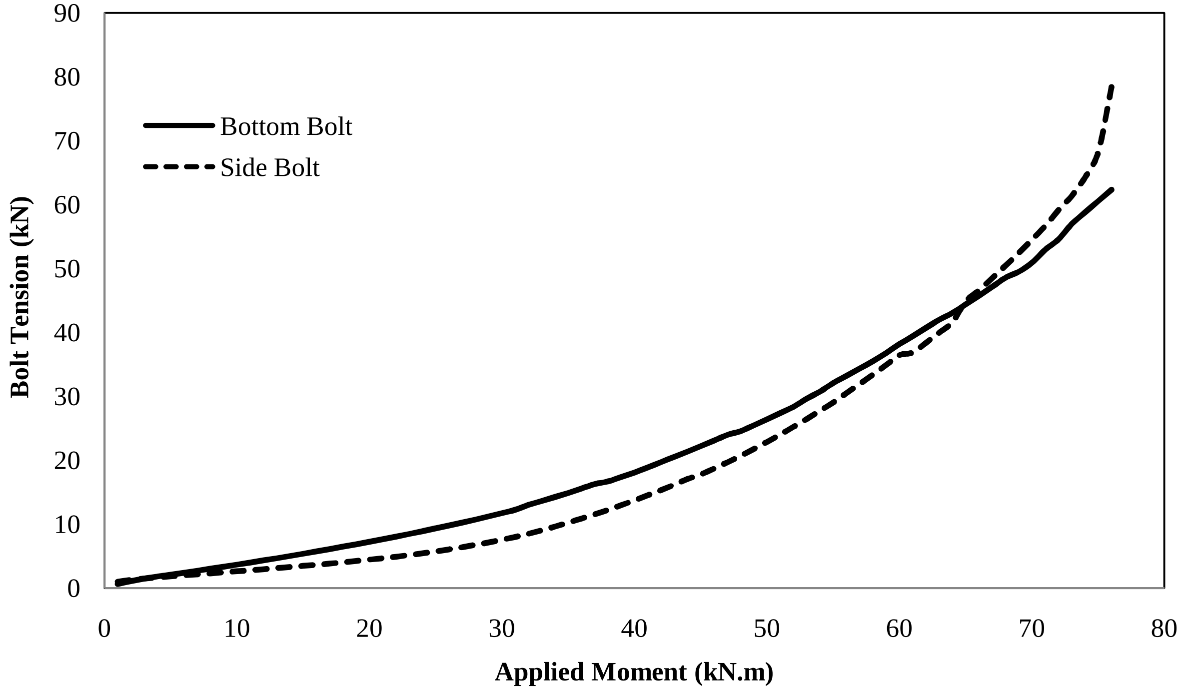
Figure 4.7: Line diagram of loading arrangement of flanged pipe joint, SFD and BMD

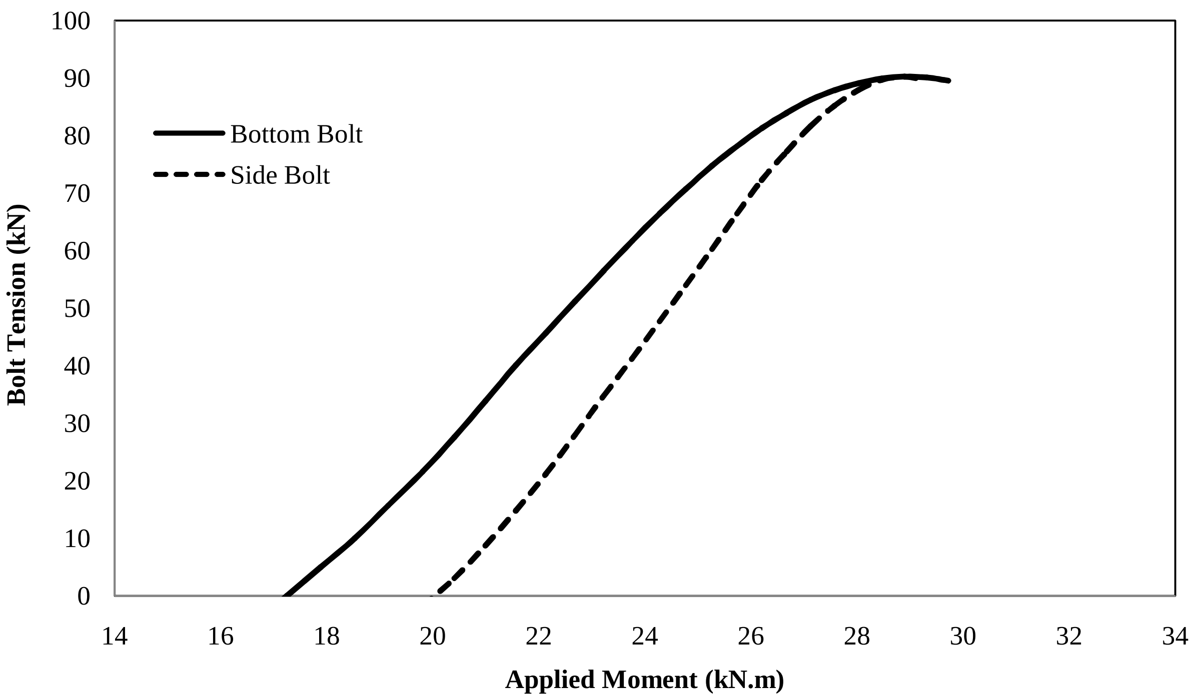
graph is developed. For preparing bolt tension vs specimen deflection graph, video extensometer deflection (Δ) datas are converted from pixel to mm. Now, plotting applied load data from UTM and deflection data from video extensormeter our applied load vs specimen deflection graph is prepared. Bolt tension vs applied moment graphs are shown in article 4.8.1. Selected applied force vs deflection graph are shown in article 4.8.2 rest are given in appendix C.

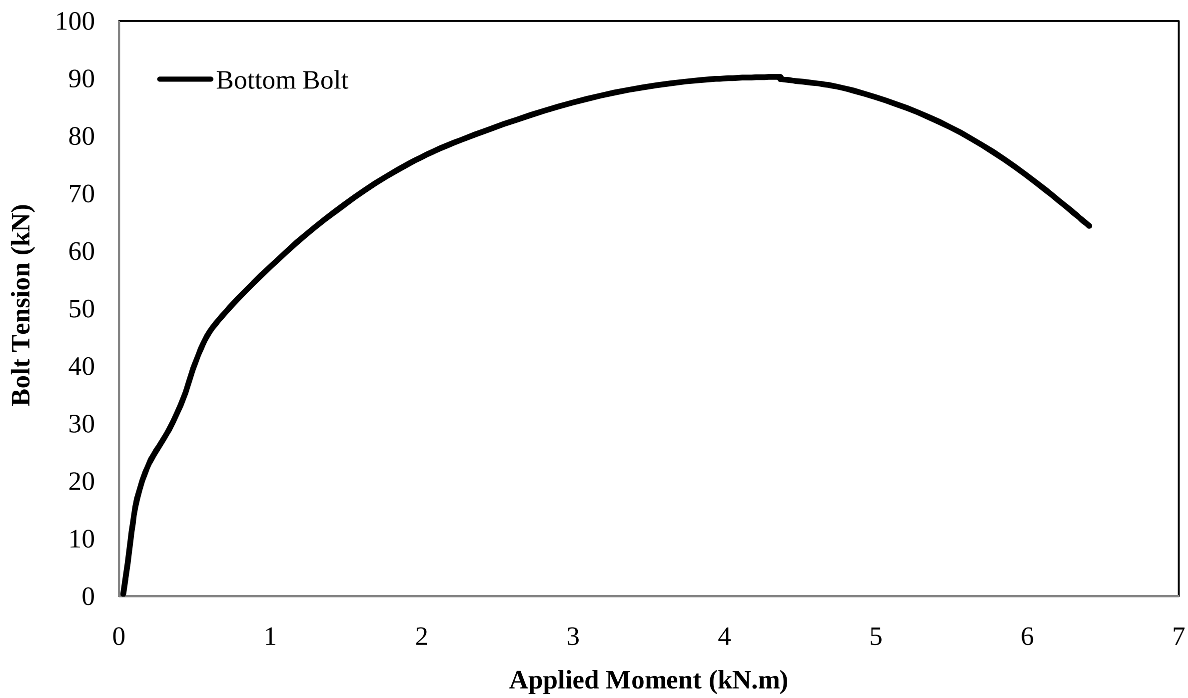
4.8 PRESENTATION OF THE RESULTS

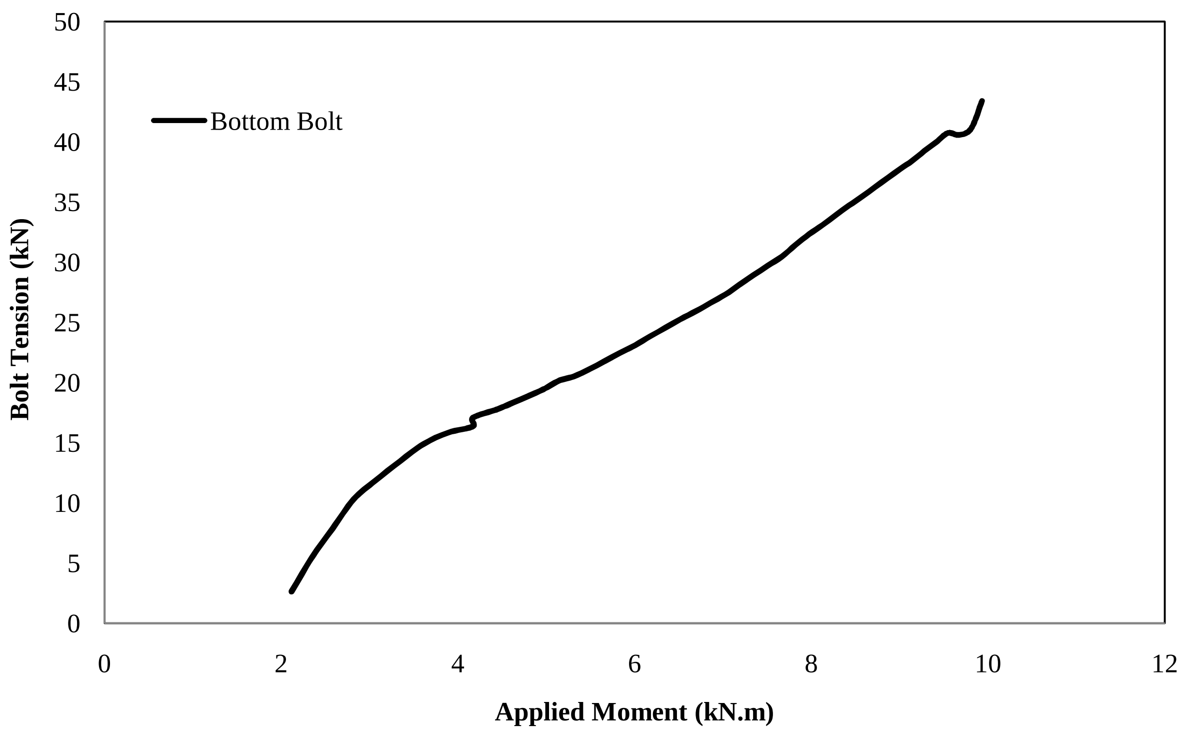
4.8.1 Bolt Tension vs Applied Moment Graph

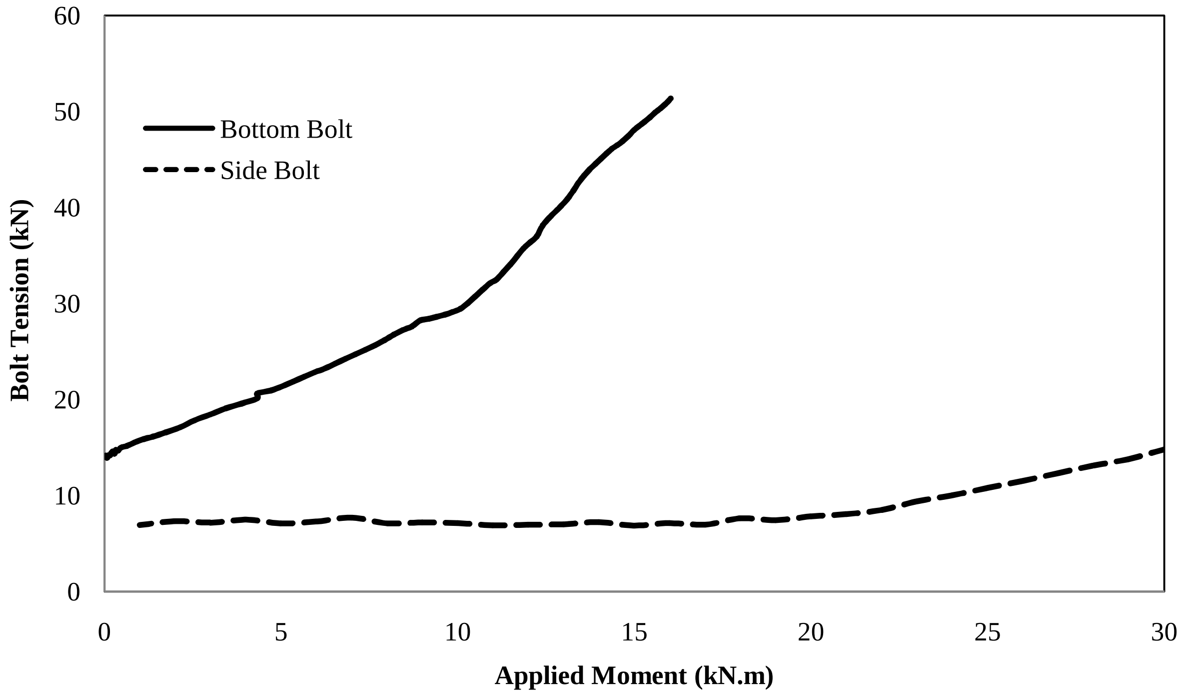
For structural safety we always need to know the likely bolt force that may develop in the bolted flange joint during its service life. These forces in the bolts developed due to applied load or applied moment on the structure. For developing our bolt tension vs applied moment graph, we applied two points load using a custom made load spreader in midsection of pipe as shown in figure 3.5. Due to this applied force at certain distance from flange moment being induced on the flange joint. From load cell of UTM we obtained the applied force value and multiplying it with distance gives us the moment value. For obtaining bolt tension value we have used strain gauges inside bolt shank. Using data logger recorded value we converted bolt strain into bolt tension. Plotting applied moment in X-axis and bolt tension in Y-axis we obtained our desired graph (Figure 4.8 to figure 4.28). From graph we observed that after certain point the side bolt force is higher than the bottom bolt force this happens may be after yielding/tearing of bottom bolts.


90
80
70
60
50
40
30
20
10
0
0
10
20
30
40
50
60
70
80
Bottom Bolt
Side Bolt
Bolt Tension (kN)
Applied Moment (kN.m)

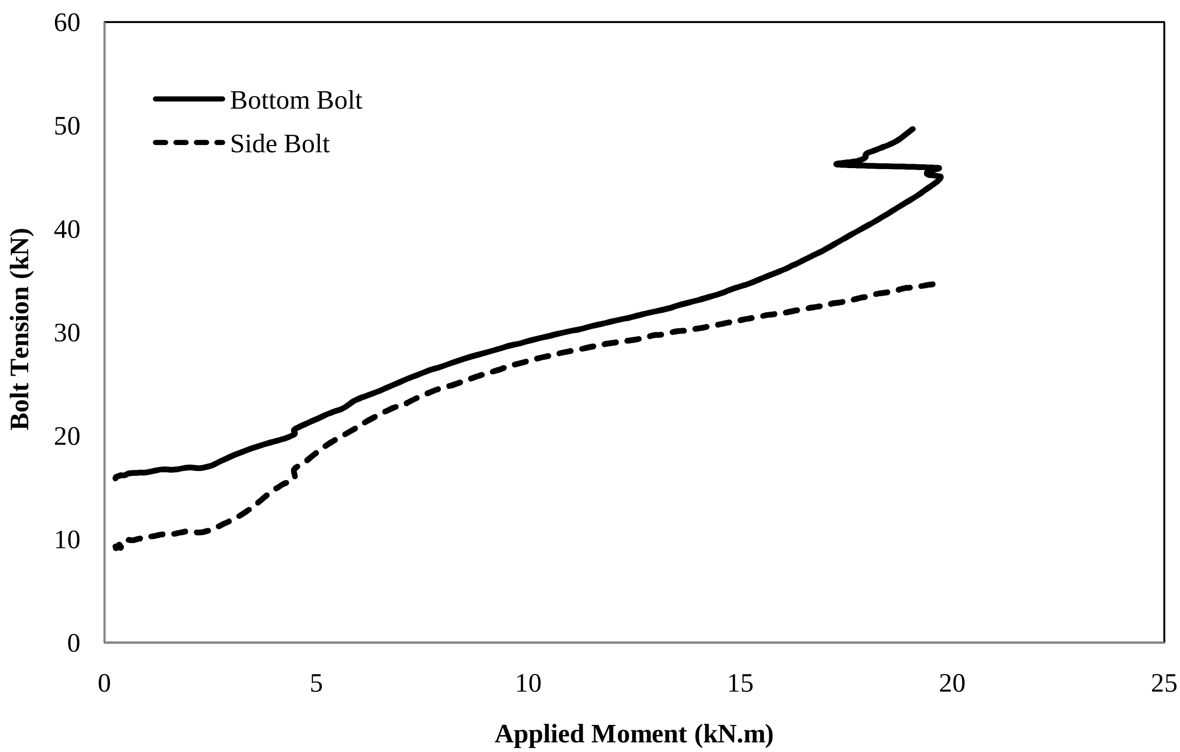

100
90
80
70
60
50
40
30
20
10
0
14
16
18
20
22
24
26
28
30
32
34
Bottom Bolt
Side Bolt
Bolt Tension (kN)
Applied Moment (kN.m)


Bottom Bolt
Bolt Tension (kN)
Applied Moment (kN.m)
0
10
20
30
40
50
60
70
80
90
100
0
1
2
3
4
5
6
7

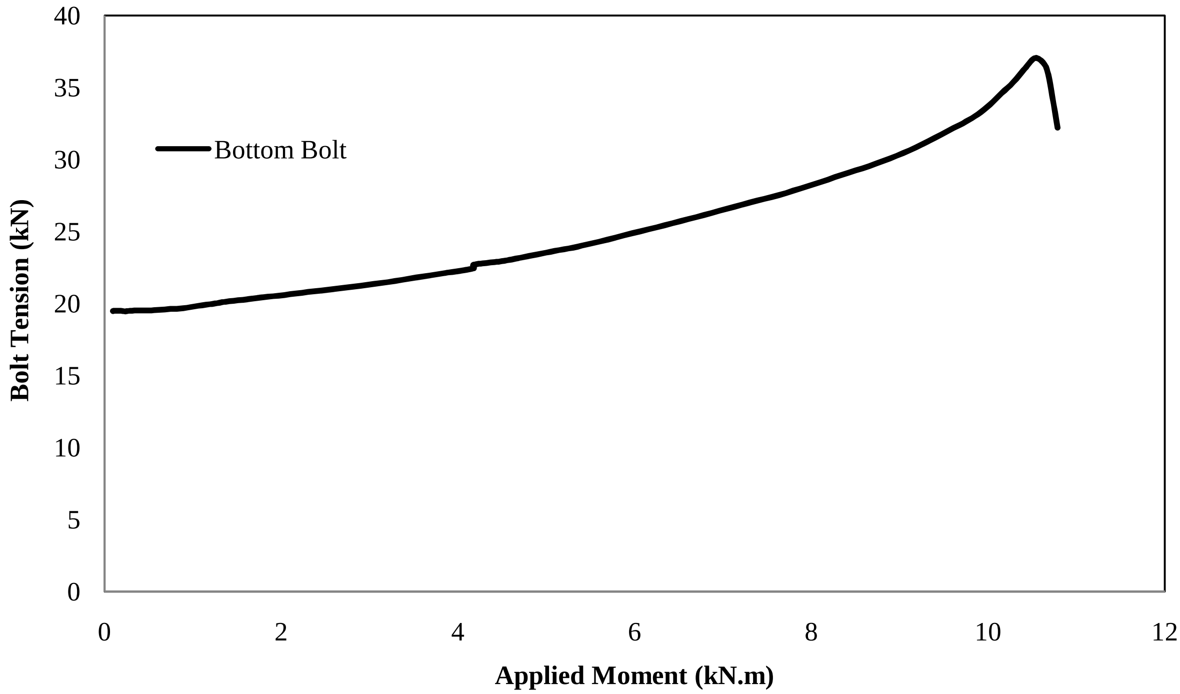

Bottom Bolt
Bolt Tension (kN)
Applied Moment (kN.m)
0
5
10
15
20
25
30
35
40
45
50
0
2
4
6
8
10
12



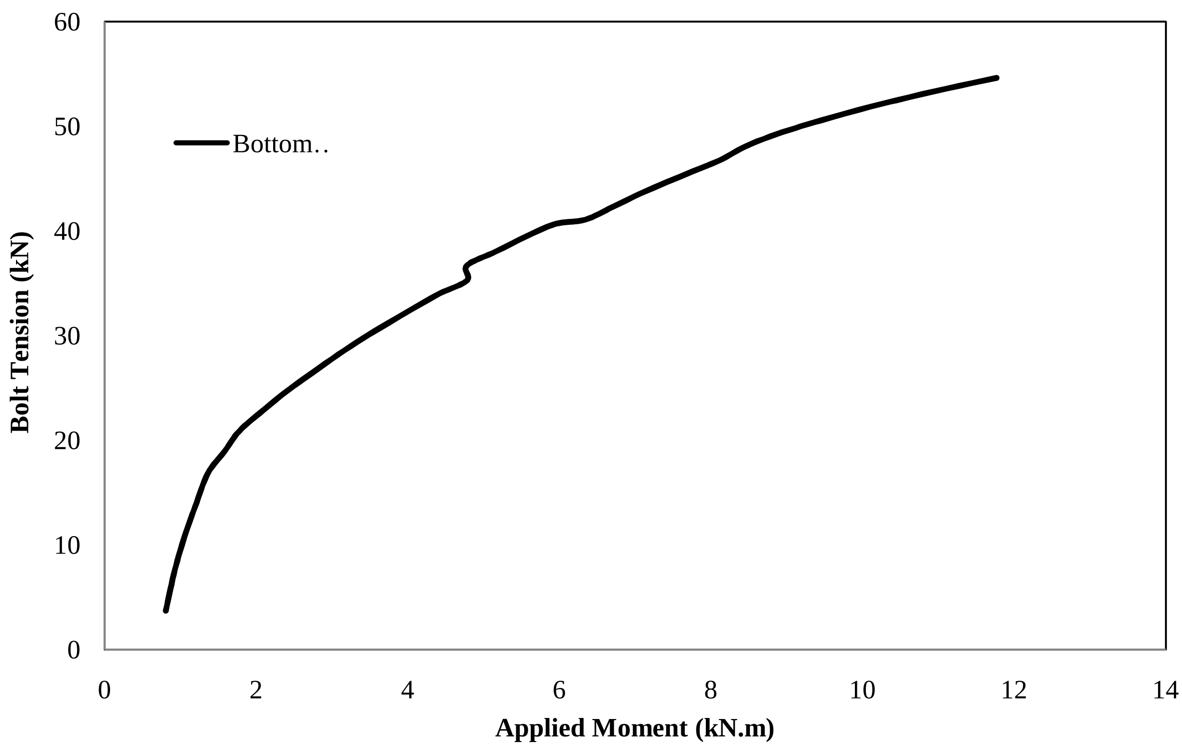
60
50
40
30
20
10
0
Bottom Bolt
Side Bolt
Bolt Tension (kN)
0
5
10
15
20
25
30
Applied Moment (kN.m)



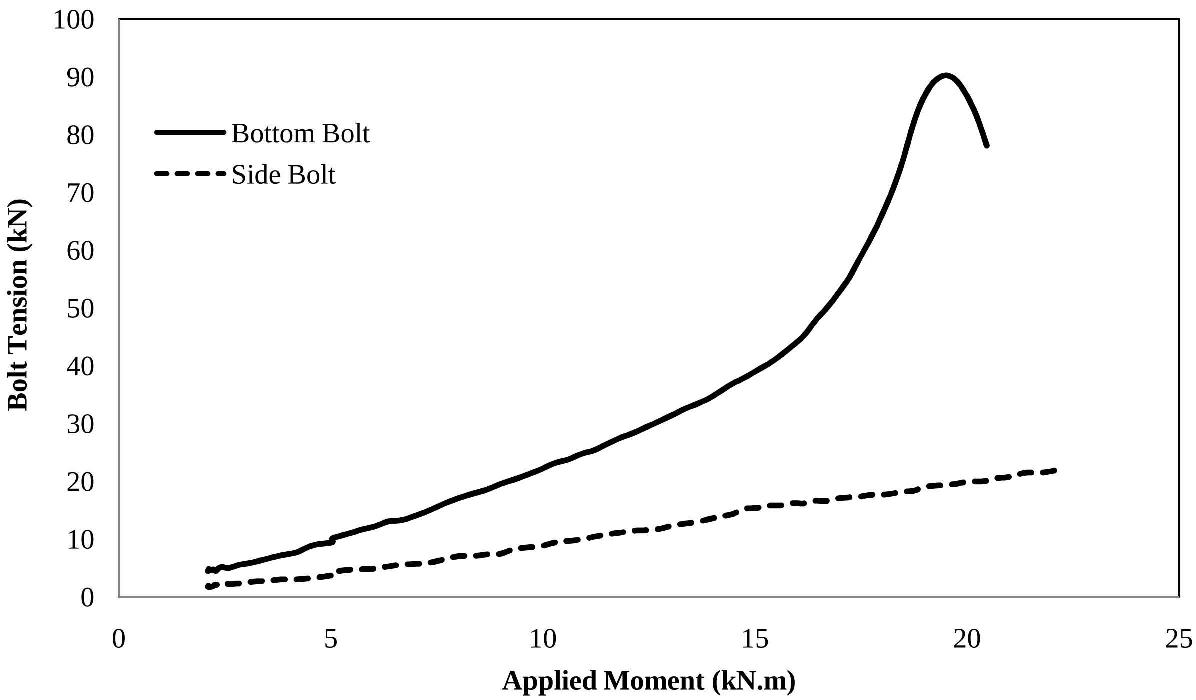
60
50
40
30
20
10
0
Bottom Bolt
Side Bolt
Bolt Tension (kN)
0
5
10
15
20
25
Applied Moment (kN.m)



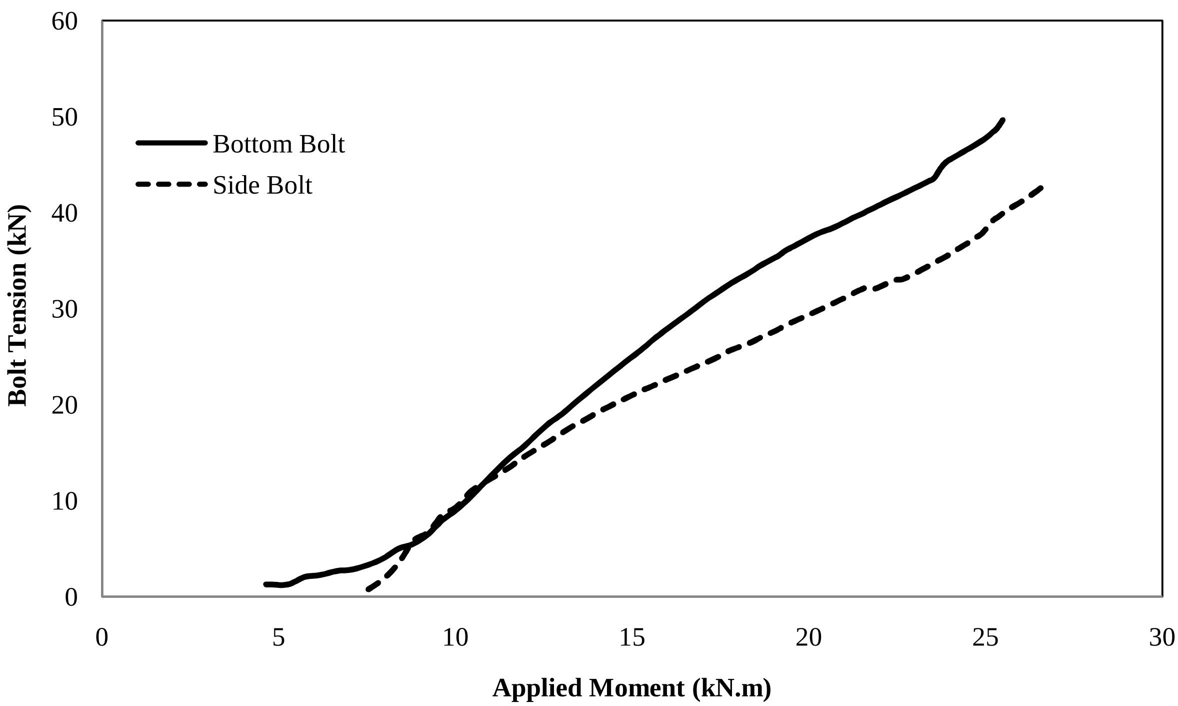
Bottom Bolt
Bolt Tension (kN)
Applied Moment (kN.m)
0
5
10
15
20
25
30
35
40
0
2
4
6
8
10
12



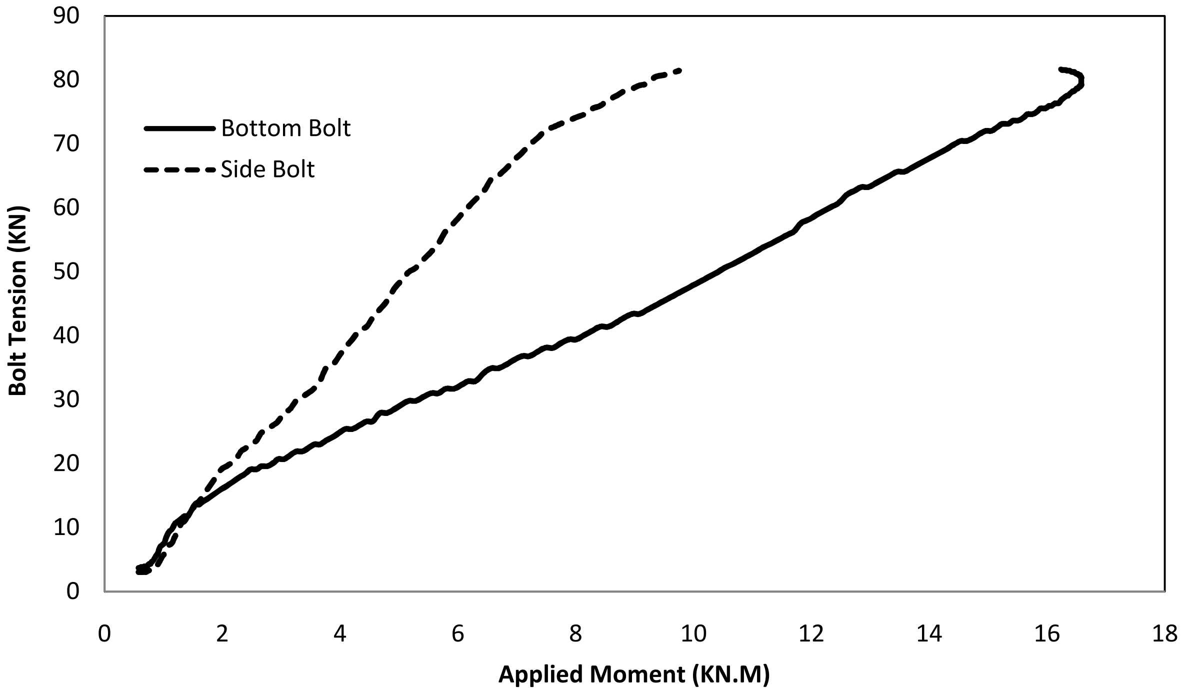
Bottom..
Bolt Tension (kN)
Applied Moment (kN.m)
0
10
20
30
40
50
60
0
2
4
6
8
10
12
14



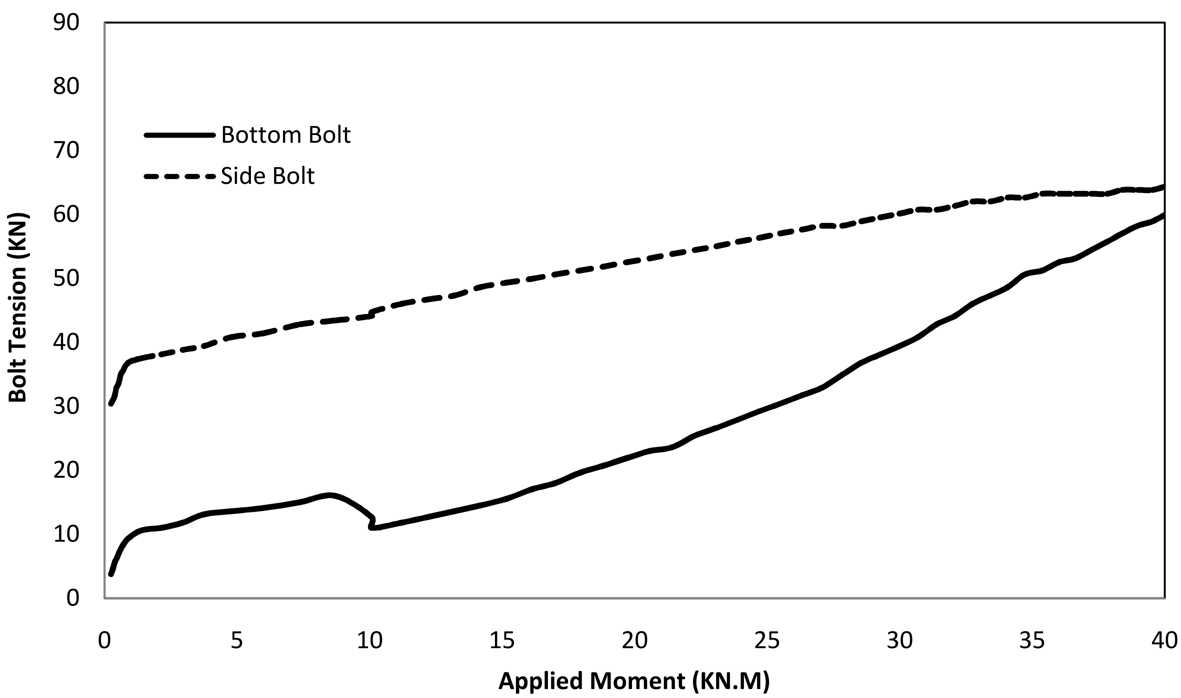
Bottom Bolt
Side Bolt
Bolt Tension (kN)
Applied Moment (kN.m)



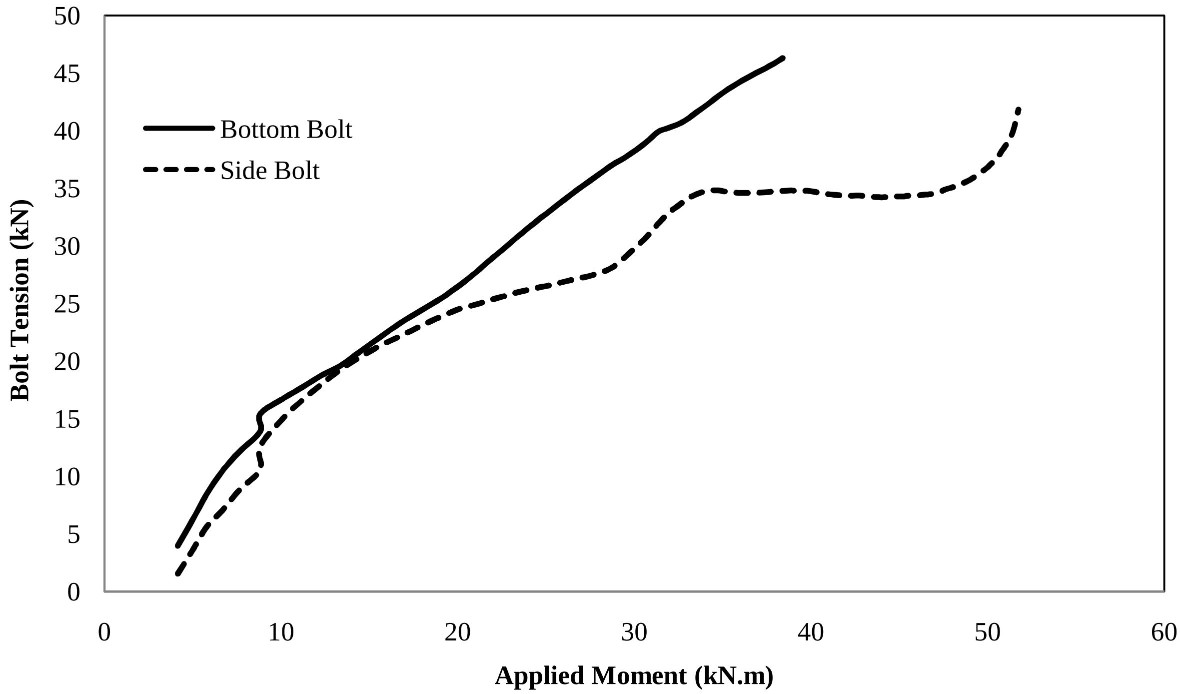
Bottom Bolt
Side Bolt
Bolt Tension (kN)
Applied Moment (kN.m)

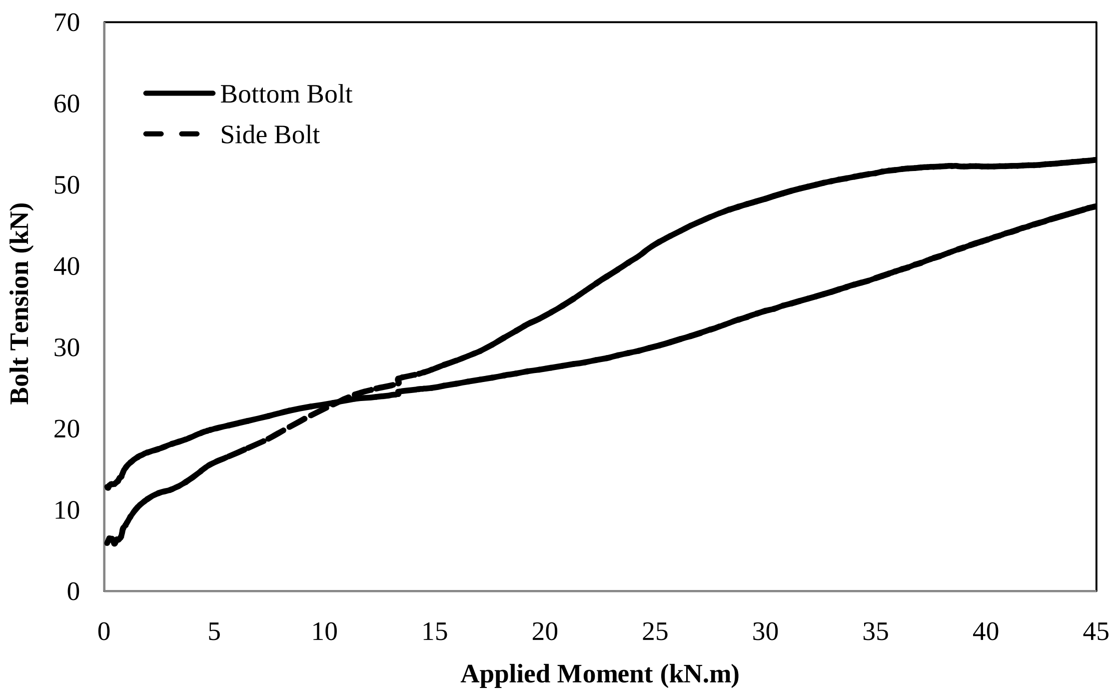


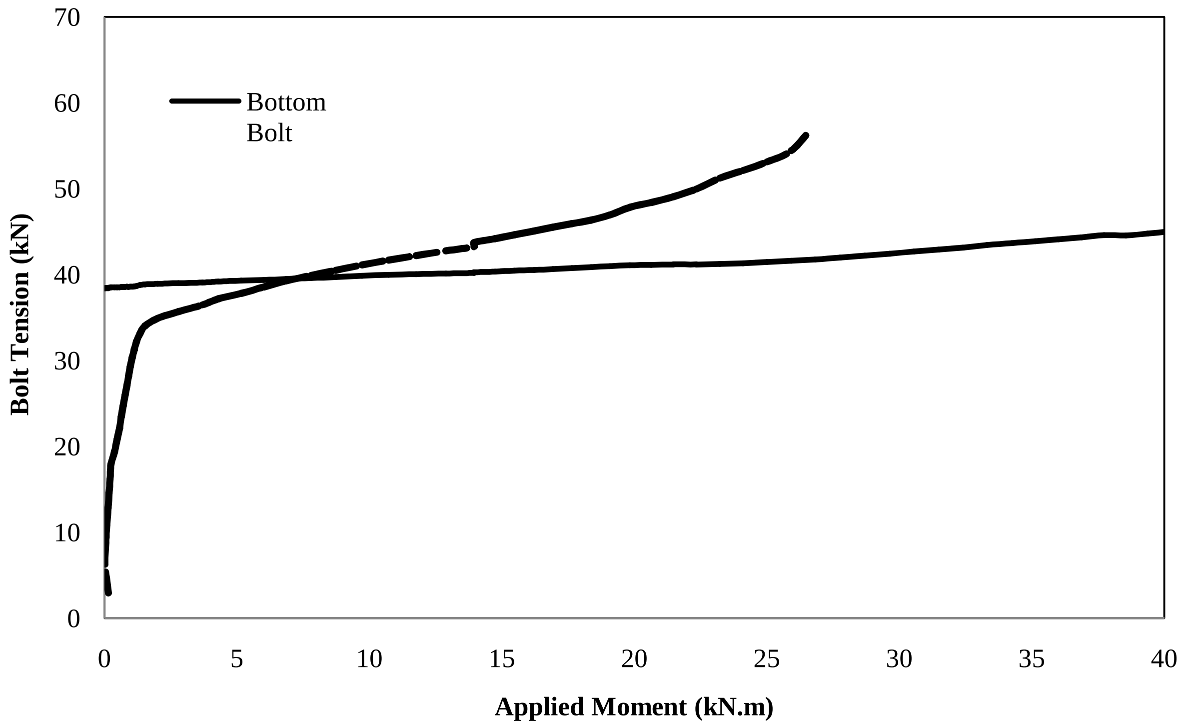
Bottom Bolt
Side Bolt
Bolt Tension (KN)
Applied Moment (KN.M)
0
10
20
30
40
50
60
70
80
90
0
2
4
6
8
10
12
14
16
18

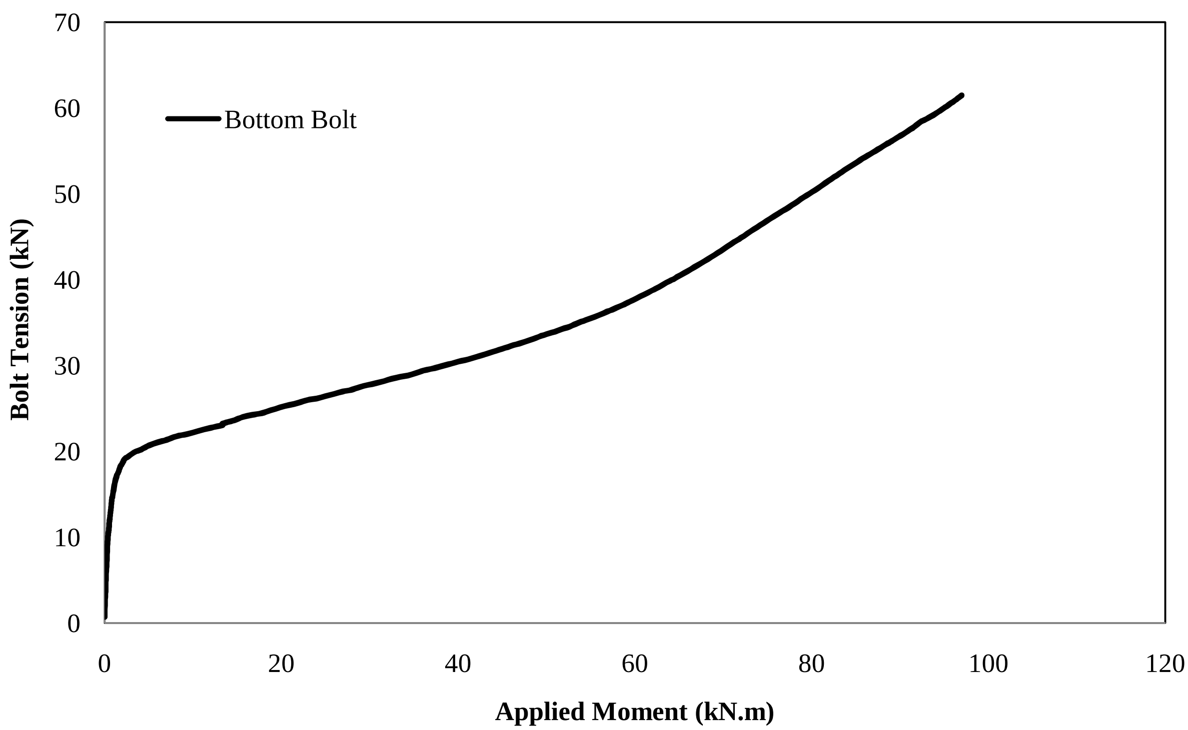


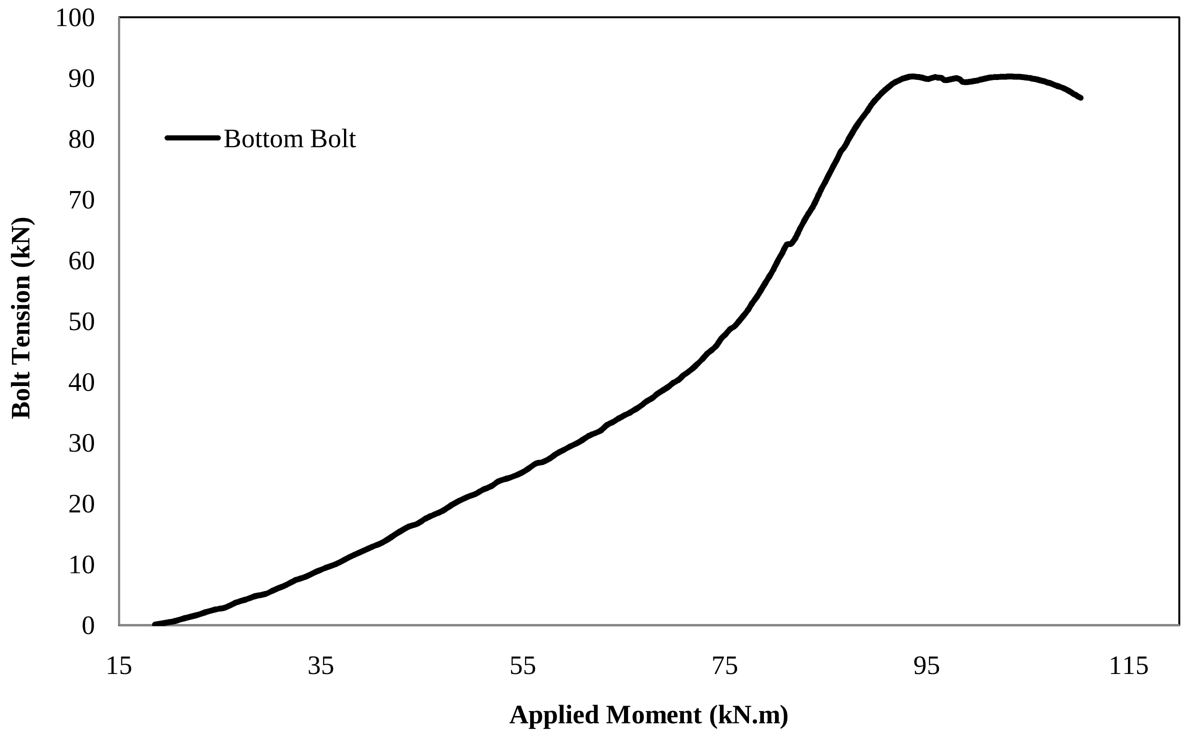
Bottom Bolt
Side Bolt
Bolt Tension (KN)
Applied Moment (KN.M)
0
10
20
30
40
50
60
70
80
90
0
5
10
15
20
25
30
35
40

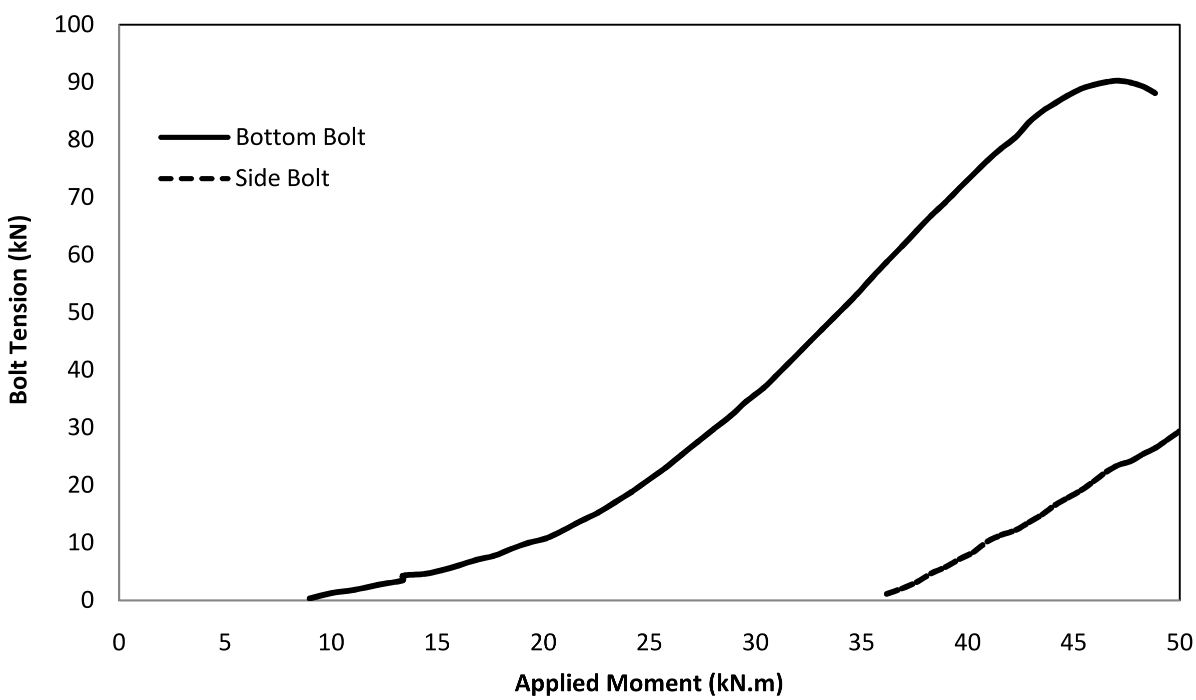

50
45
40
35
30
25
20
15
10
5
0
Bolt Tension (kN)
Bottom Bolt
Side Bolt
0
10
20
30
40
50
60
Applied Moment (kN.m)

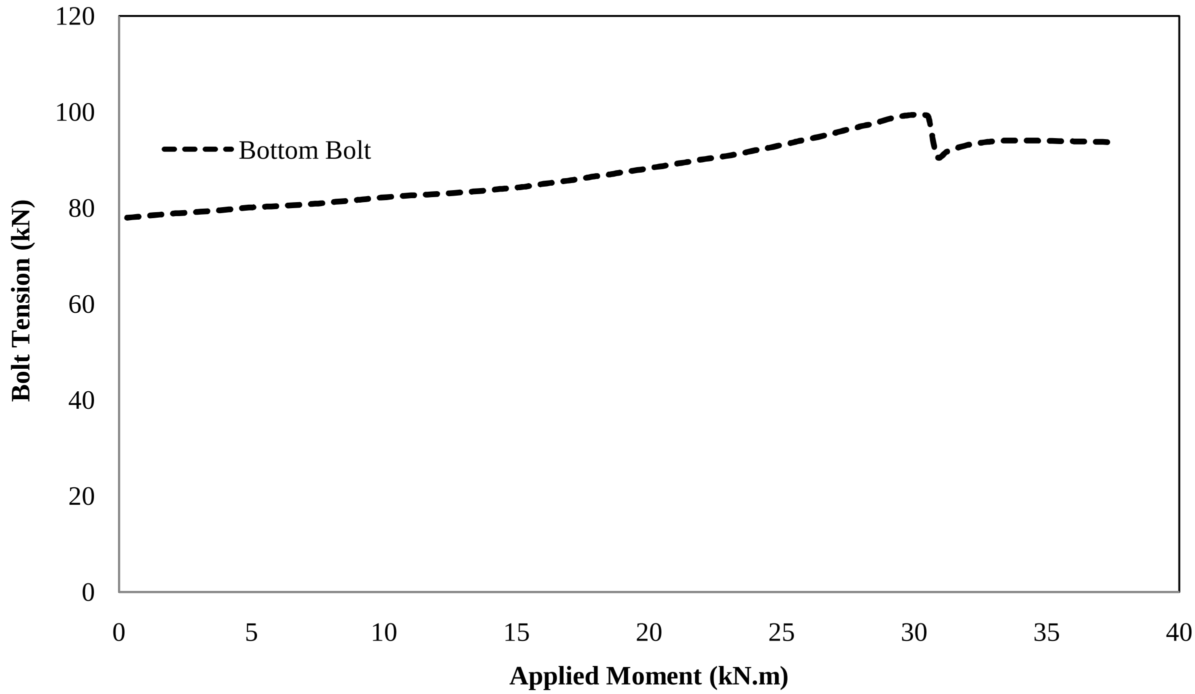

70
60
50
40
30
20
10
0
Bolt Tension (kN)
Bottom Bolt
Side Bolt
0
5
10
15
20
25
30
35
40
45
Applied Moment (kN.m)

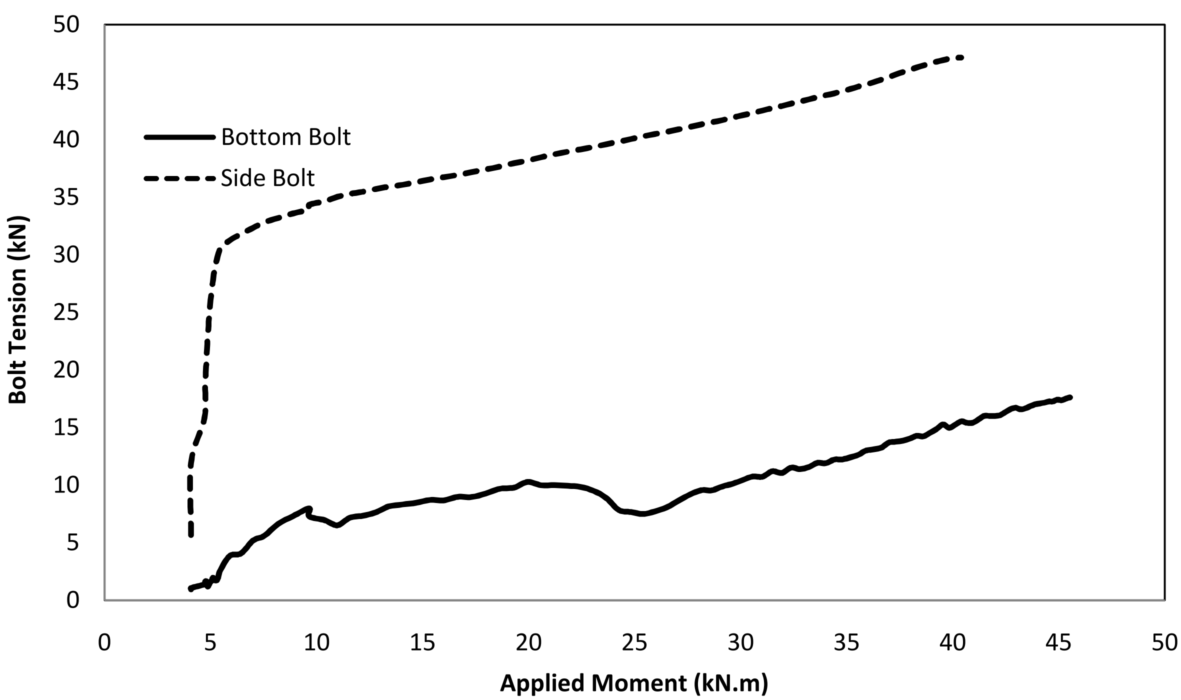

Bottom Bolt
Bolt Tension (kN)
Applied Moment (kN.m)
0
10
20
30
40
50
60
70
0
5
10
15
20
25
30
35
40

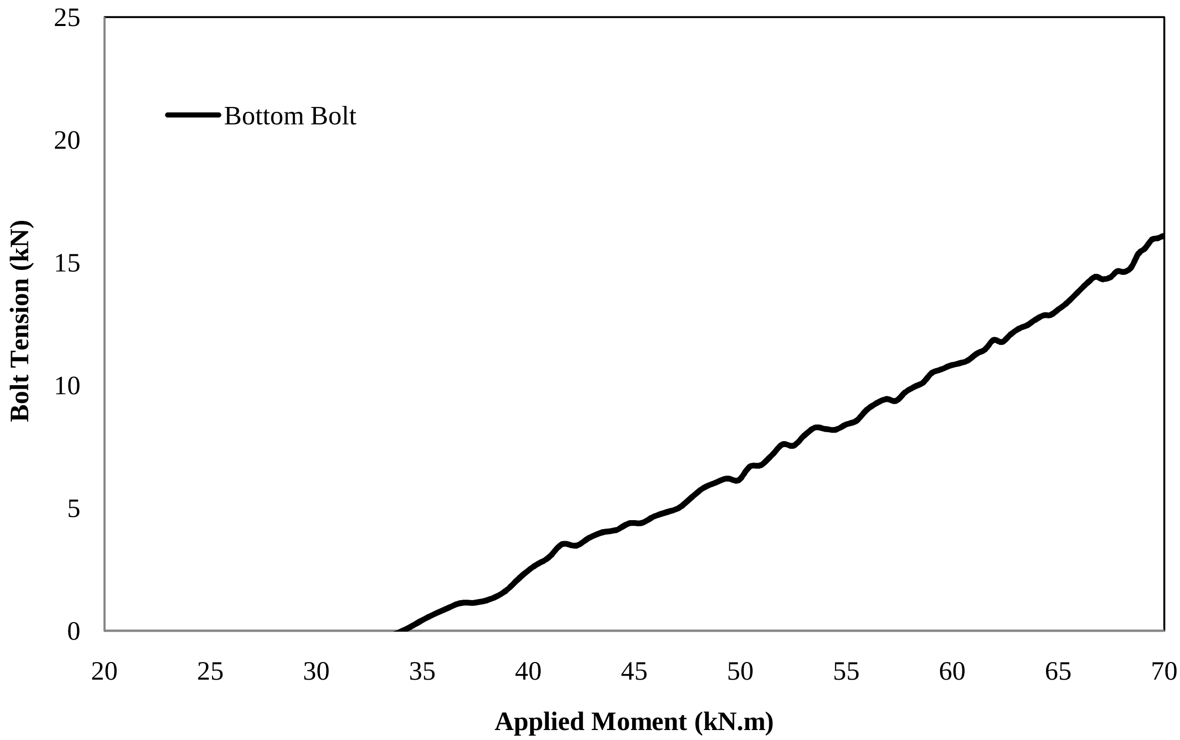

Bottom Bolt
Bolt Tension (kN)
Applied Moment (kN.m)
0
10
20
30
40
50
60
70
0
20
40
60
80
100
120

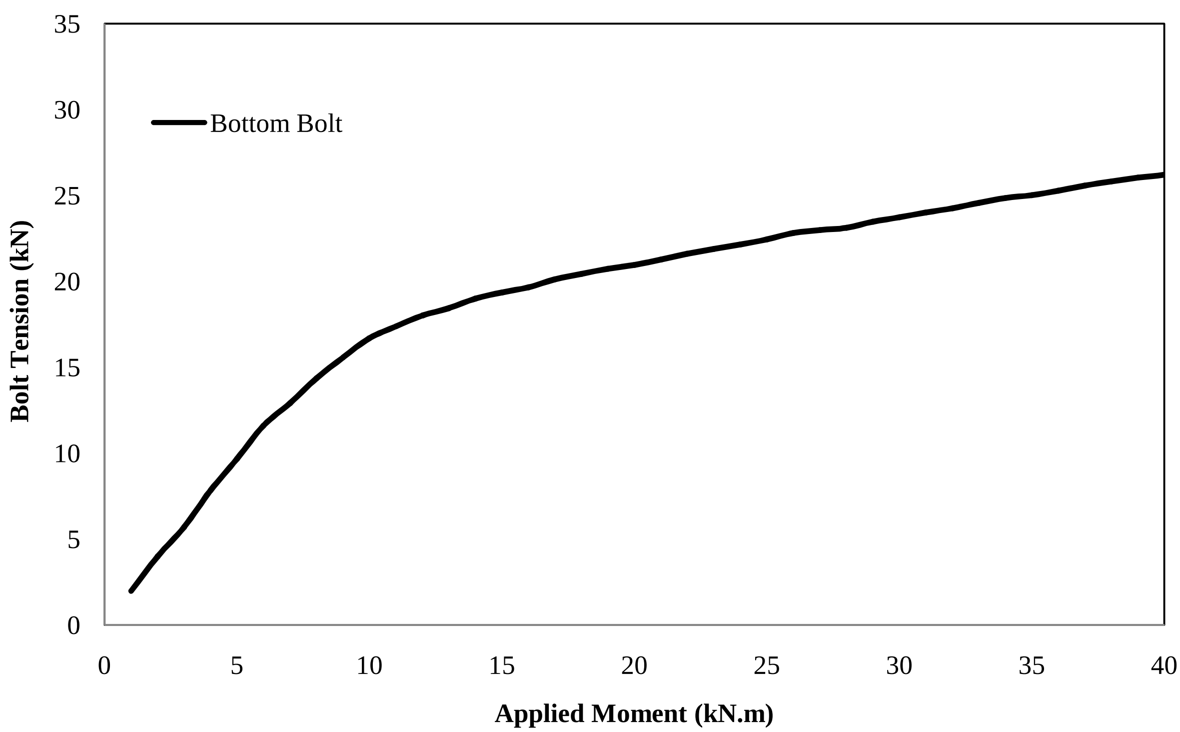

Bottom Bolt
Bolt Tension (kN)
Applied Moment (kN.m)
0
10
20
30
40
50
60
70
80
90
100
15
35
55
75
95
115

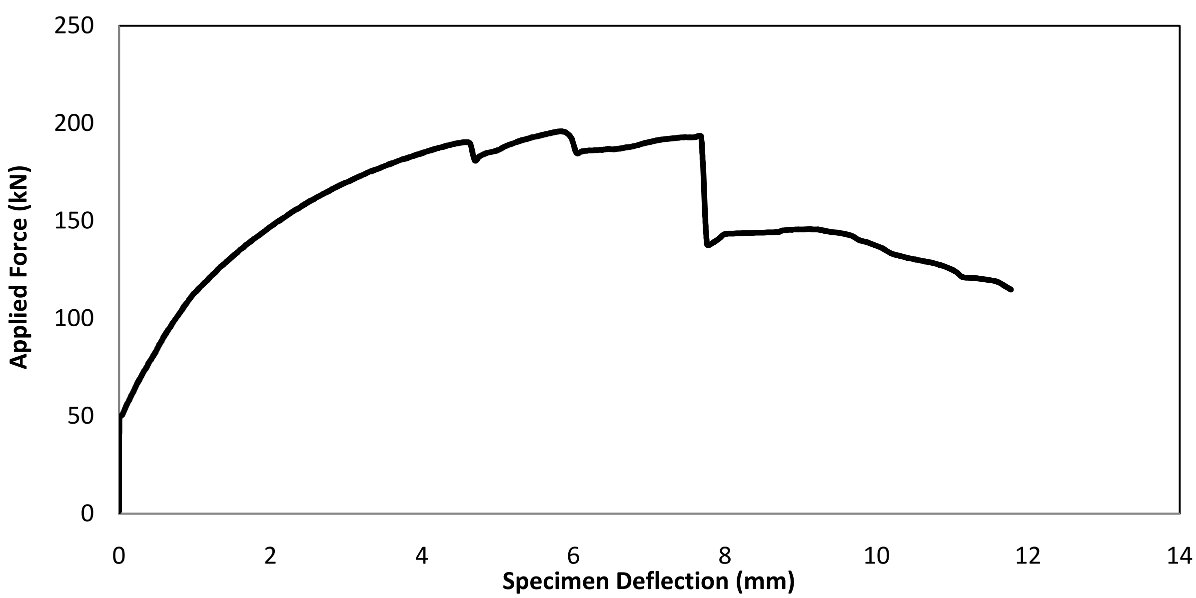

Bottom Bolt
Side Bolt
Bolt Tension (kN)
Applied Moment (kN.m)
0
10
20
30
40
50
60
70
80
90
100
0
5
10
15
20
25
30
35
40
45
50

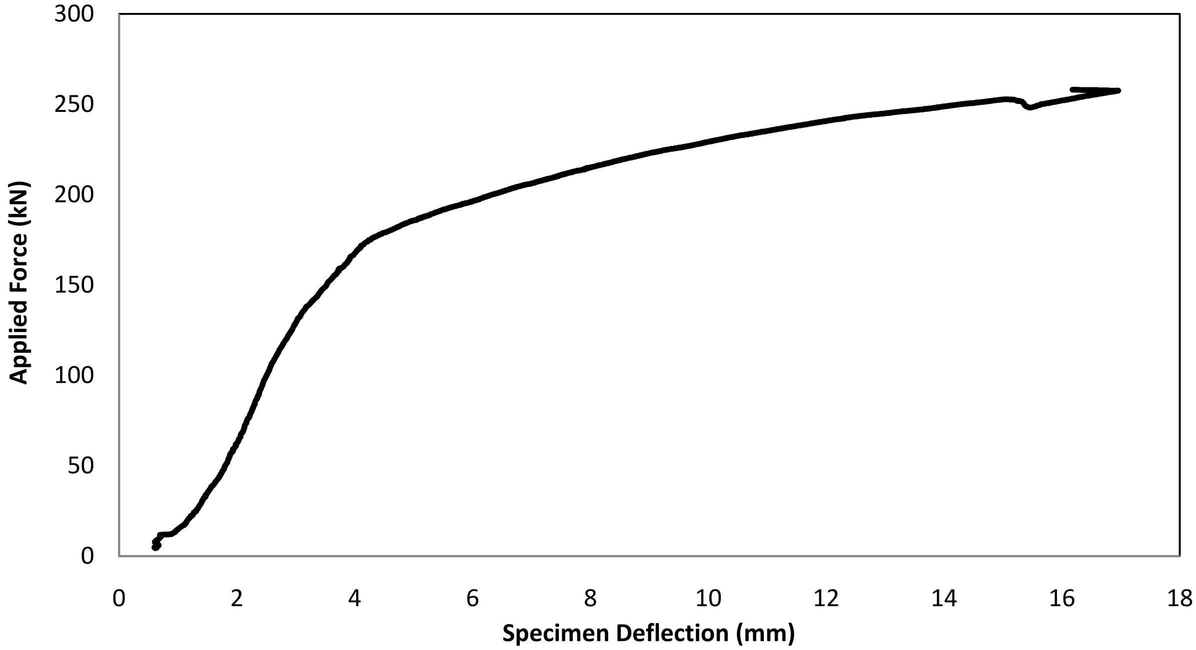

120
100
80
60
40
20
0
Bolt Tension (kN)
Bottom Bolt
0
5
10
15
20
25
30
35
40
Applied Moment (kN.m)


50
45
40
35
30
25
20
15
10
5
0
Bolt Tension (kN)
Bottom Bolt
Side Bolt
0
5
10
15
20
25
30
35
40
45
50
Applied Moment (kN.m)

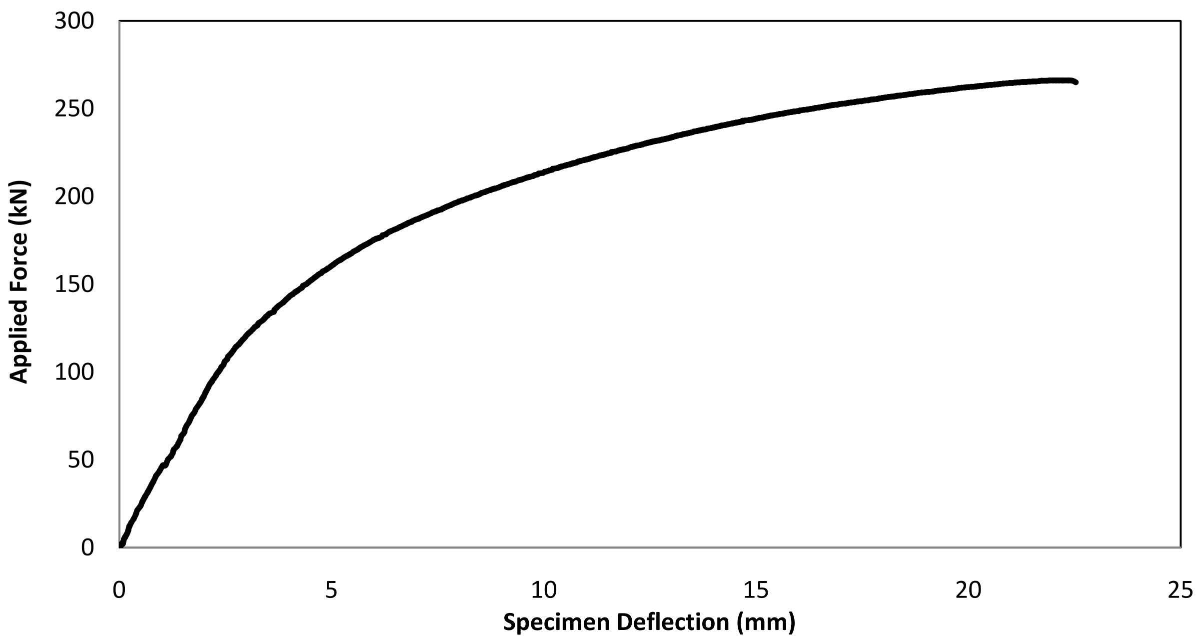

25
20
15
10
5
0
Bottom Bolt
Bolt Tension (kN)
20
25
30
35
40
45
50
55
60
65
70
Applied Moment (kN.m)


35
30
25
20
15
10
5
0
Bottom Bolt
Bolt Tension (kN)
0
5
10
15
20
25
30
35
40
Applied Moment (kN.m)

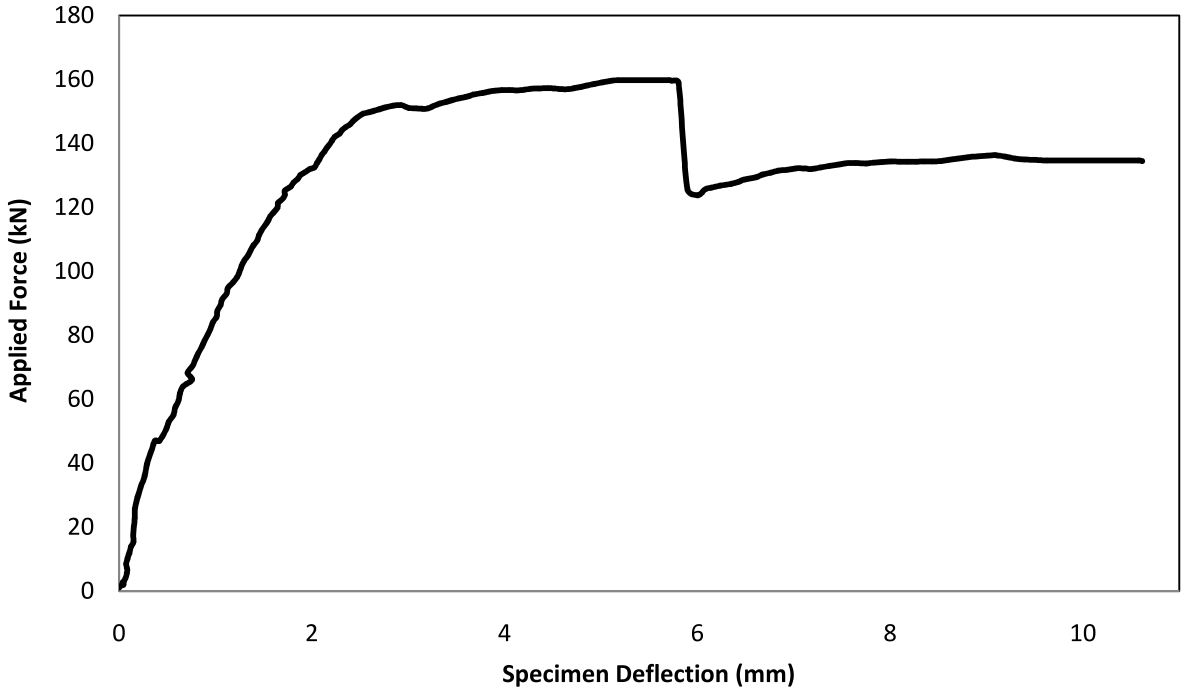

250
200
150
100
50
0
0
2
4
6
8
10
12
14
Applied Force (kN)
Specimen Deflection (mm)



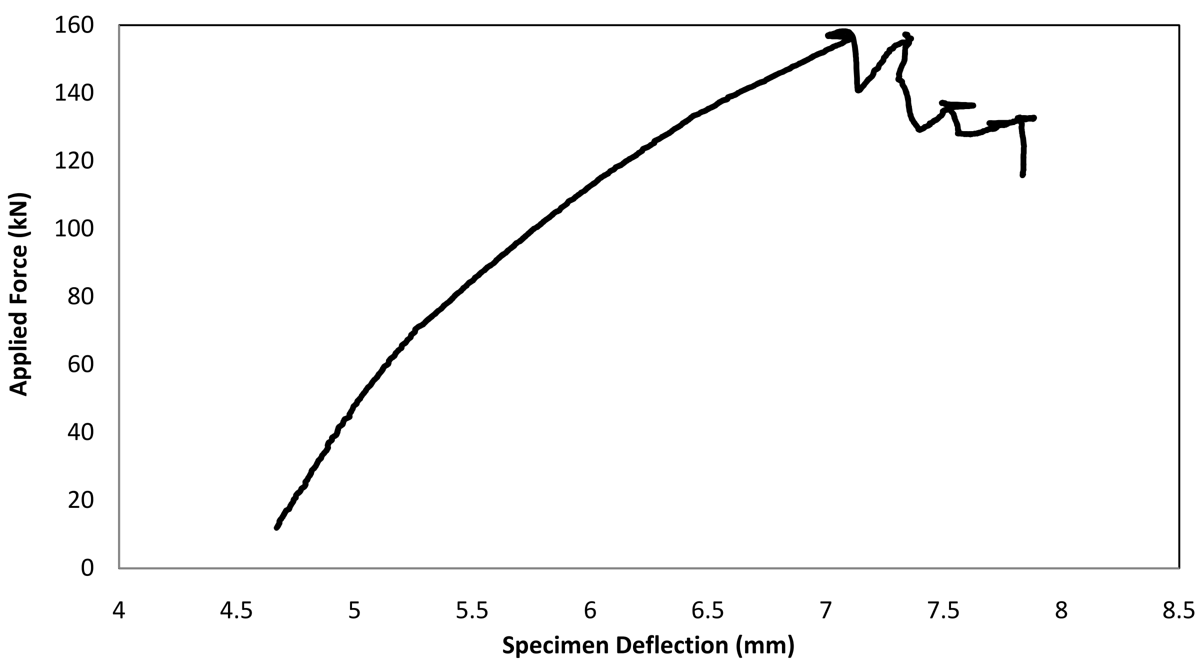
300
250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
14
16
18
Specimen Deflection (mm)



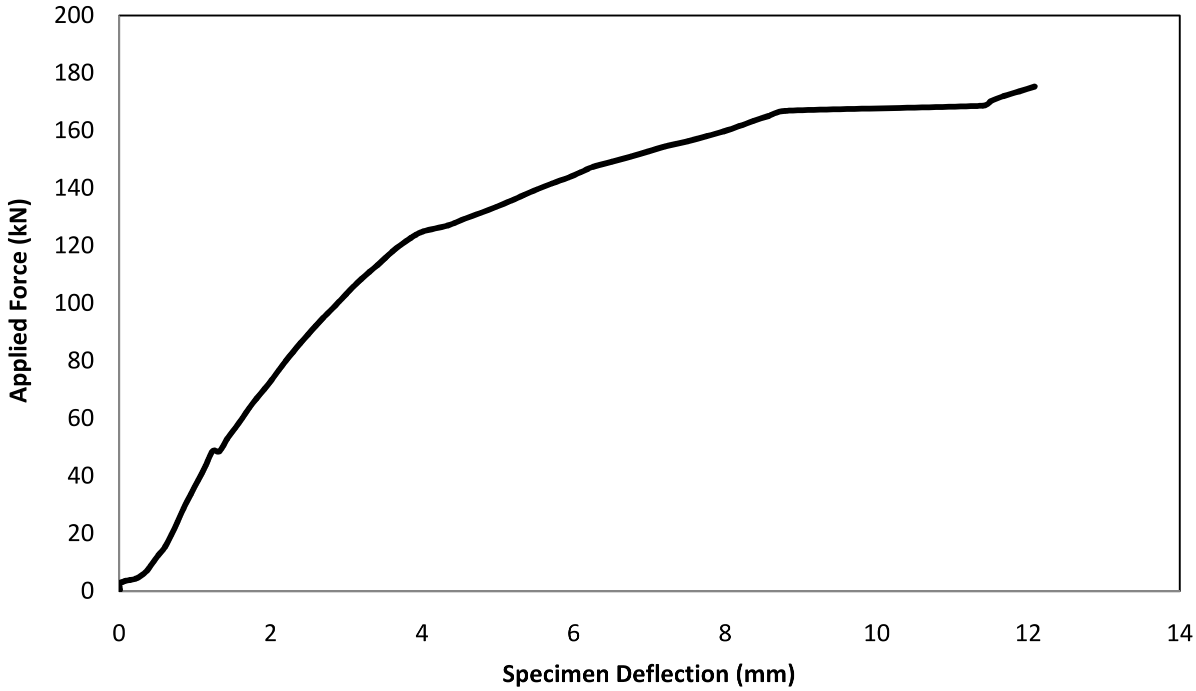
300
250
200
150
100
50
0
Applied Force (kN)
0
5
10
15
20
25
Specimen Deflection (mm)



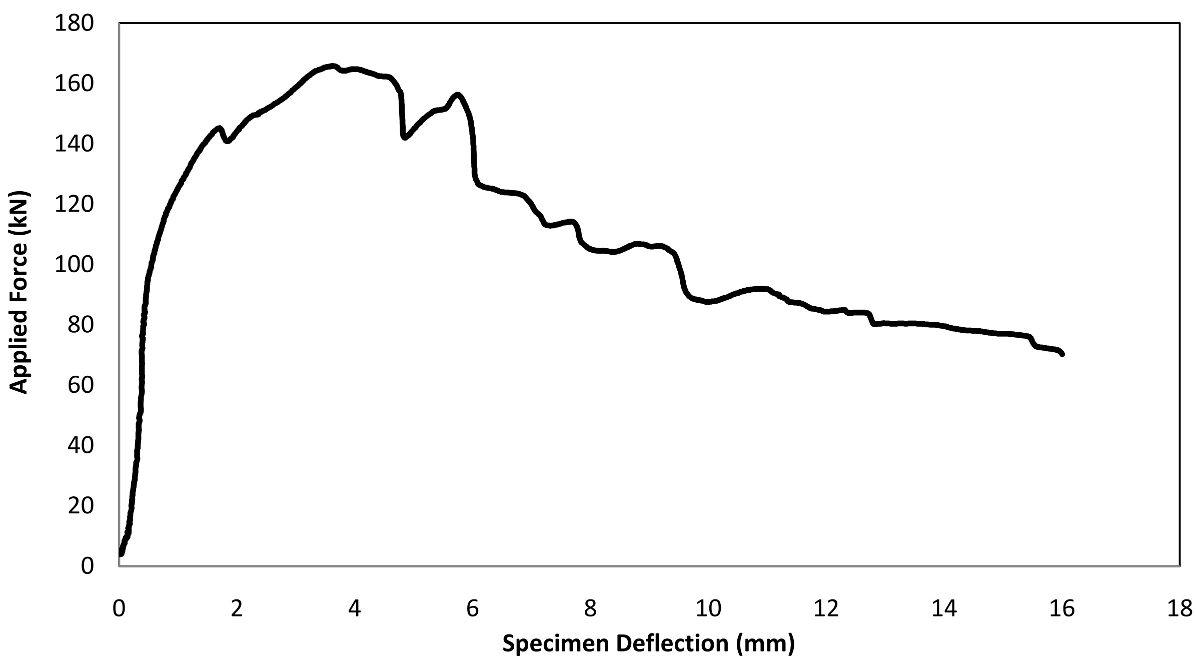
Applied Force (kN)
Specimen Deflection (mm)
180
160
140
120
100
80
60
40
20
0
0
2
4
6
8
10



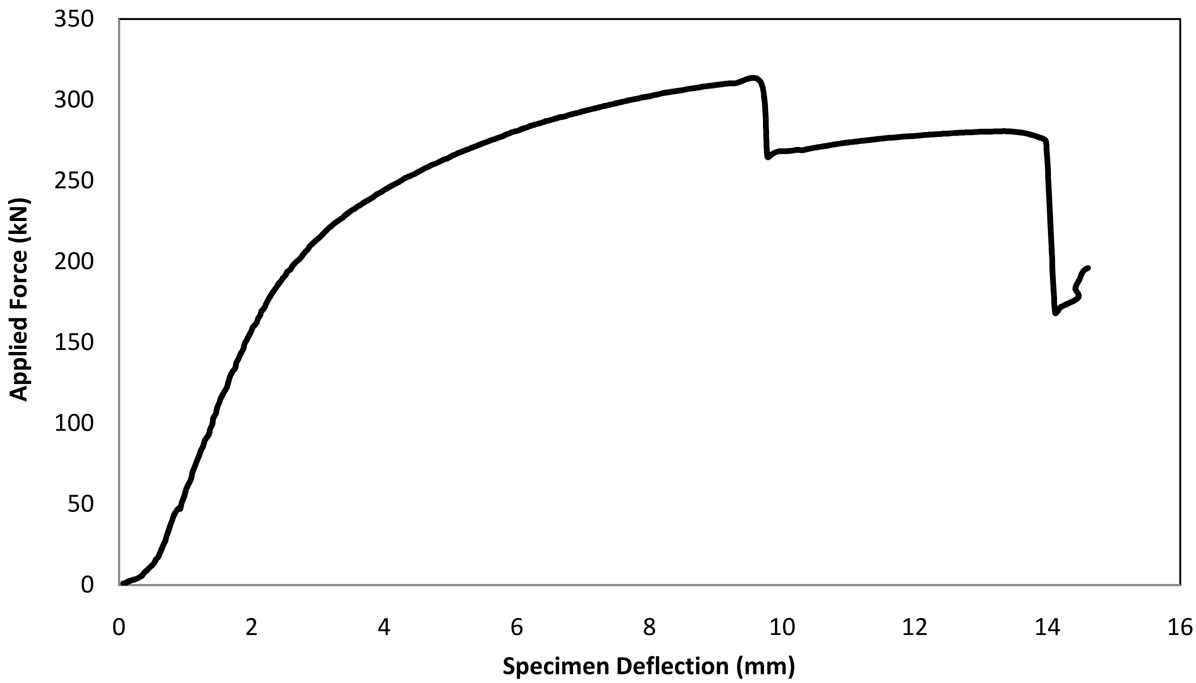
Applied Force (kN)
Specimen Deflection (mm)
160
140
120
100
80
60
40
20
0
4
4.5
5
5.5
6
6.5
7
7.5
8
8.5



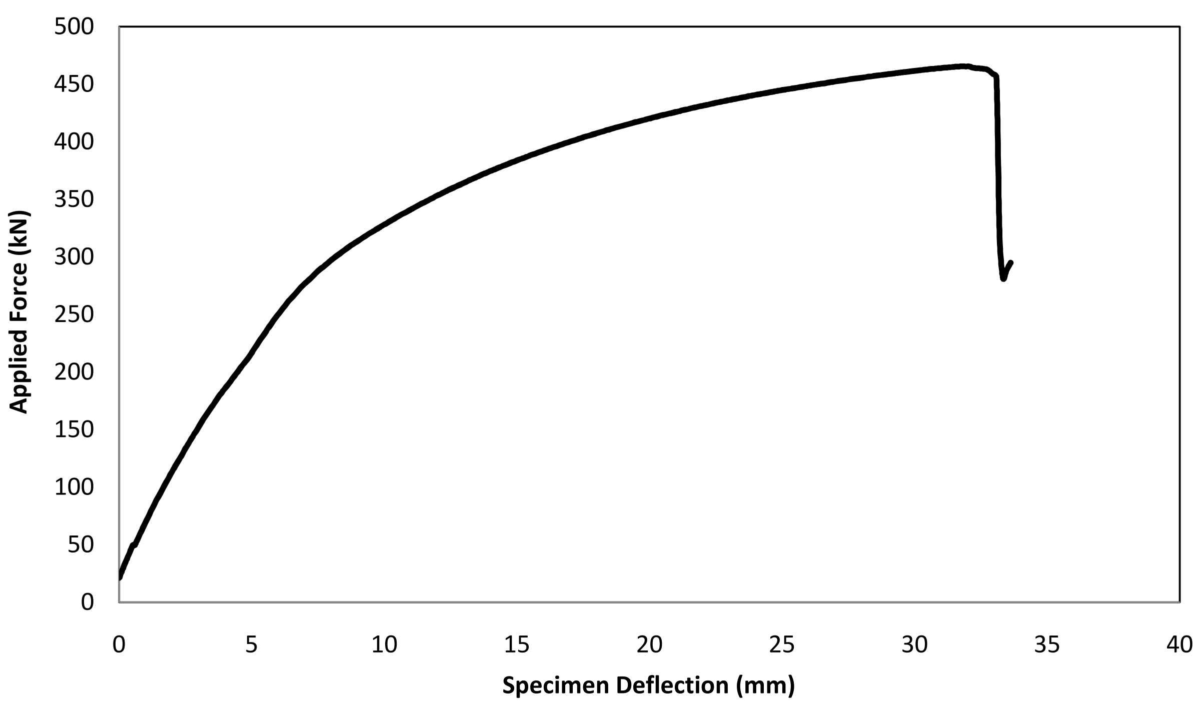
Applied Force (kN)
0
20
40
60
80
100
120
140
160
180
200
0
2
4
6
8
10
12
14
Specimen Deflection (mm)



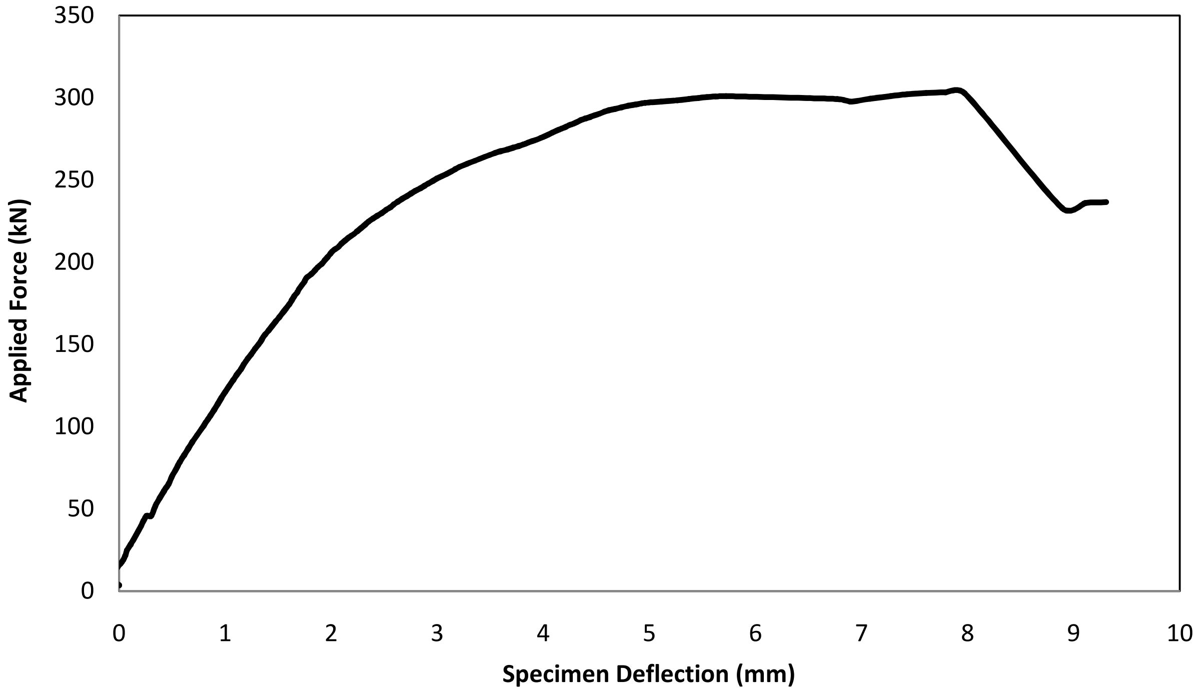
Applied Force (kN)
0
20
40
60
80
100
120
140
160
180
0
2
4
6
8
10
12
14
16
18
Specimen Deflection (mm)



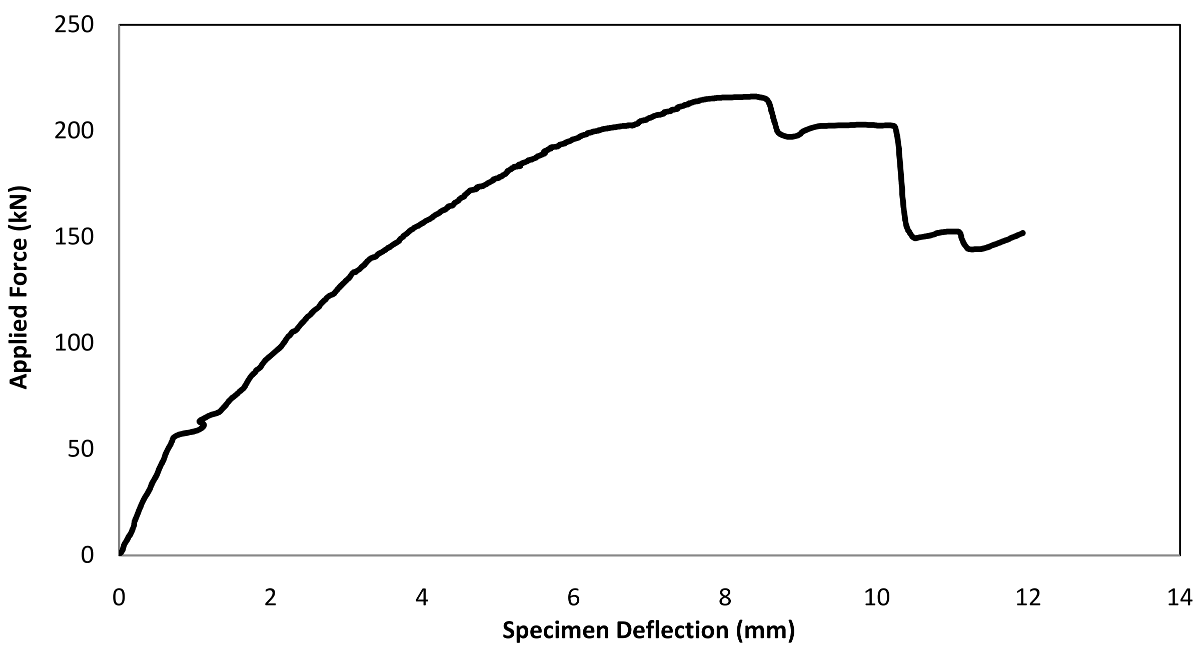
Applied Force (kN)
350
300
250
200
150
100
50
0
0
2
4
6
8
10
12
14
16
Specimen Deflection (mm)



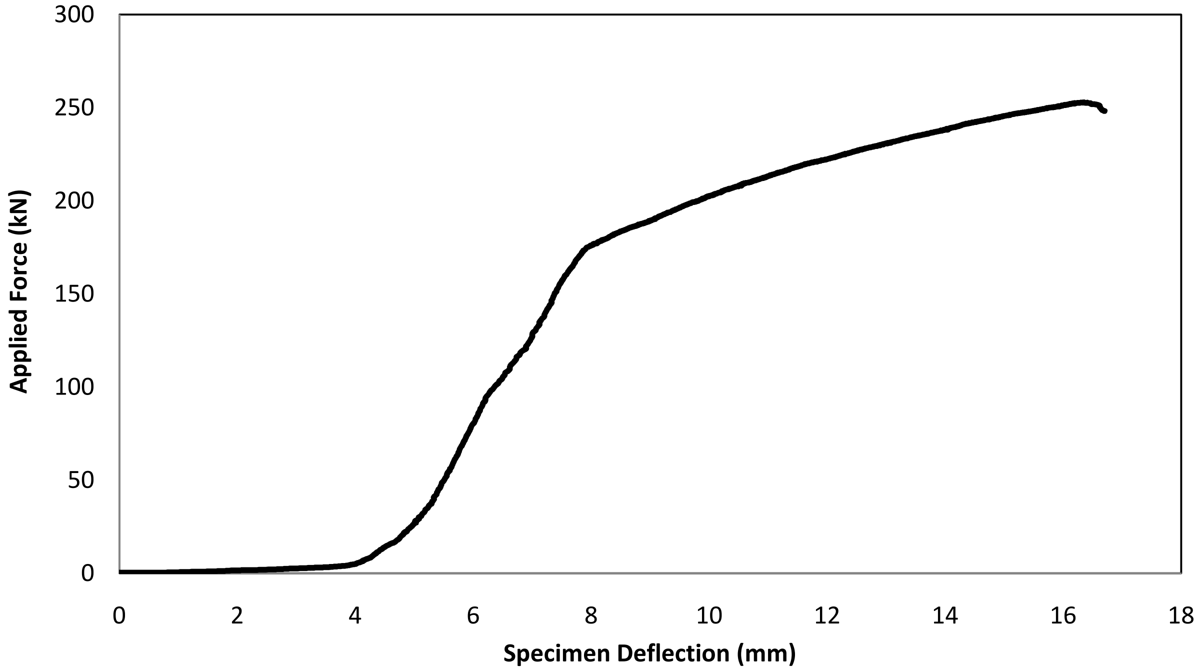
Applied Force (kN)
500
450
400
350
300
250
200
150
100
50
0
0
5
10
15
20
25
30
35
40
Specimen Deflection (mm)



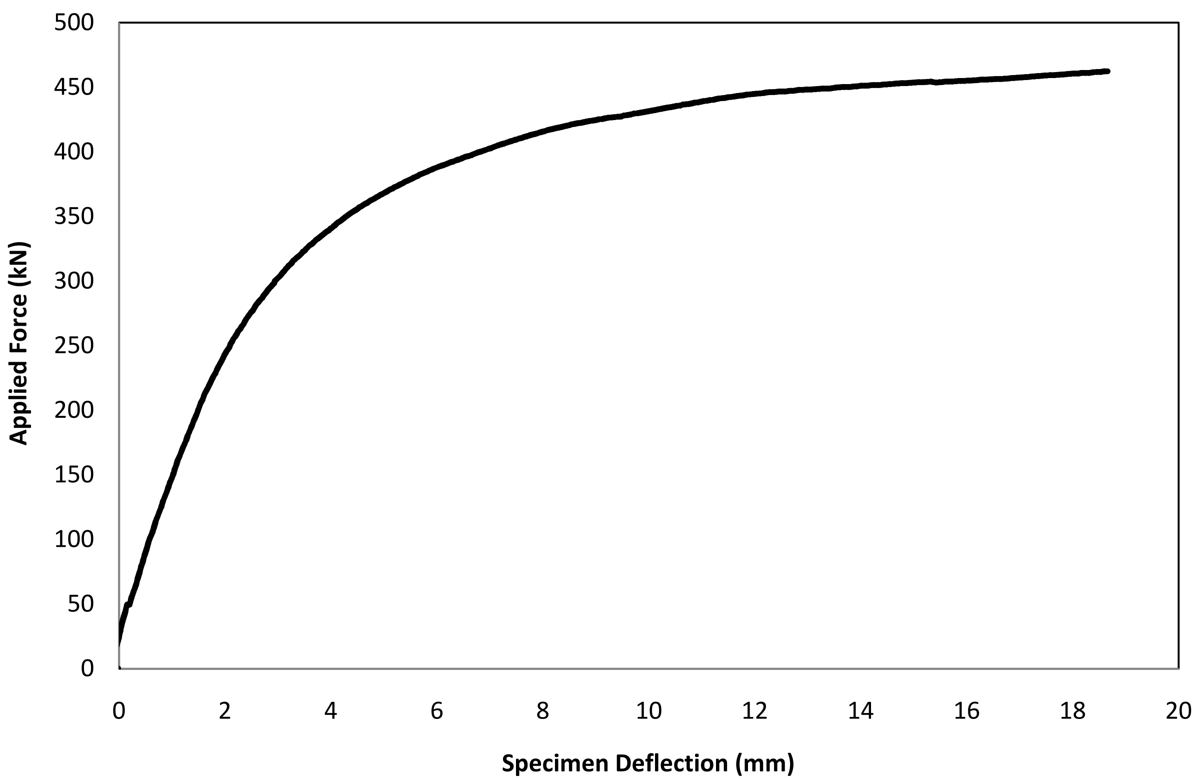
350
300
250
200
150
100
50
0
Applied Force (kN)
0
1
2
3
4
5
6
7
8
9
10
Specimen Deflection (mm)



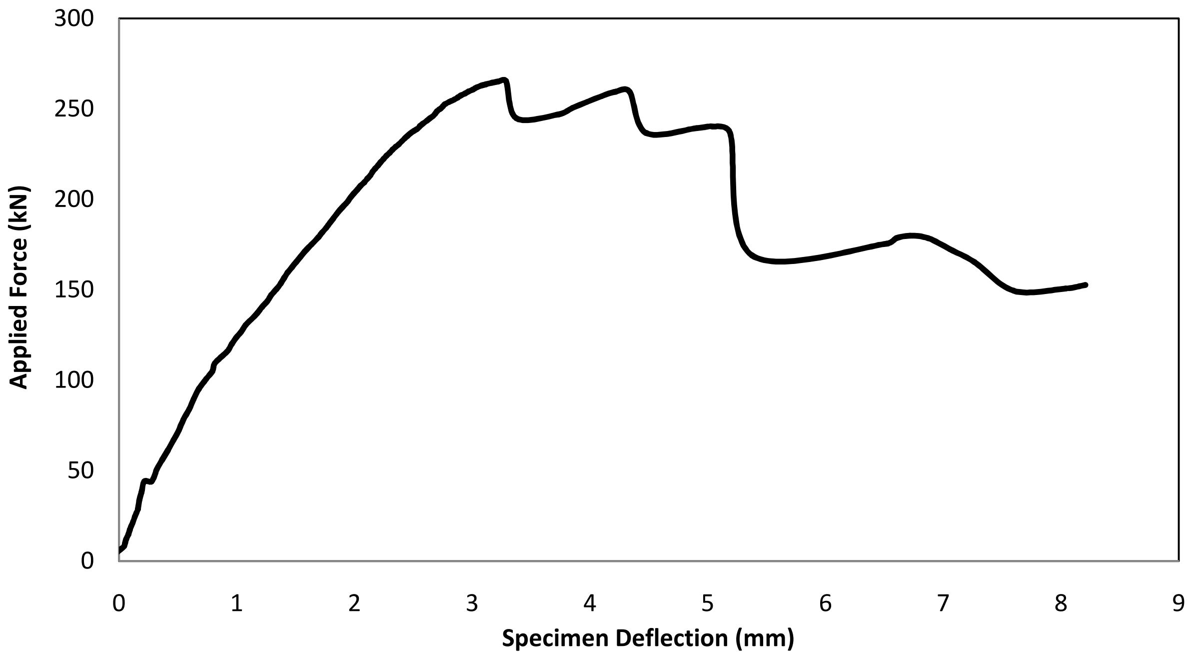
250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
14
Specimen Deflection (mm)



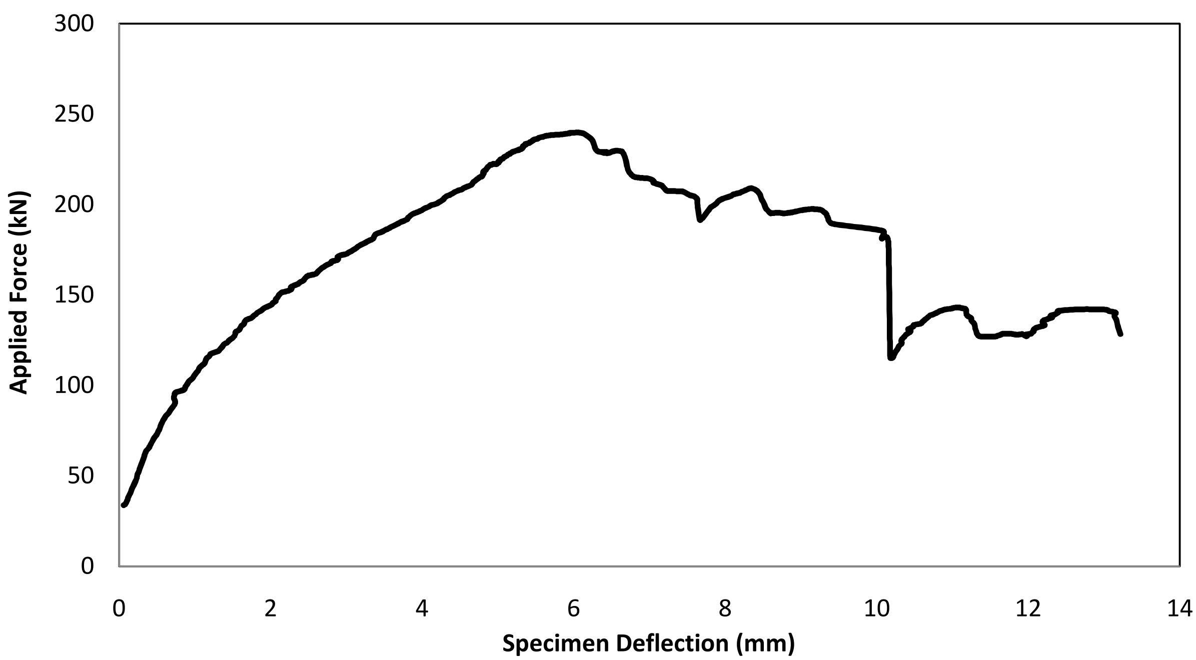
Applied Force (kN)
0
50
100
150
200
250
300
Specimen Deflection (mm)
0
2
4
6
8
10
12
14
16
18



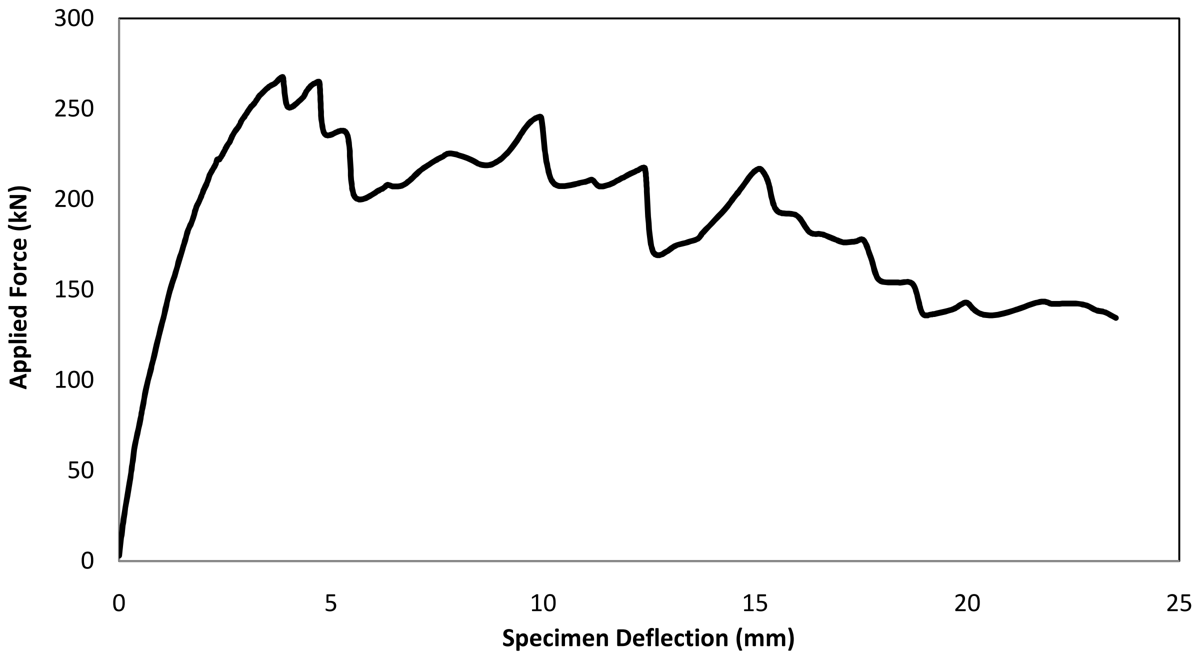
Applied Force (kN)
0
50
100
150
200
250
300
350
400
450
500
Specimen Deflection (mm)
0
2
4
6
8
10
12
14
16
18
20



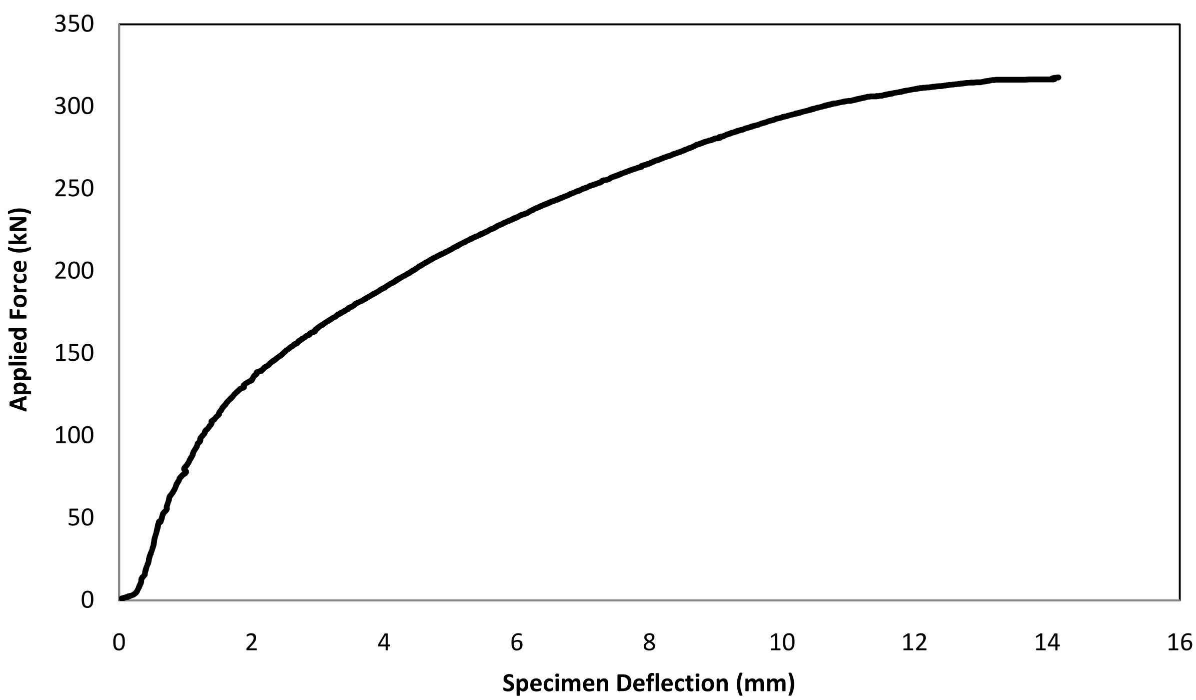
300
250
200
150
100
50
0
0
1
2
3
4
5
6
7
8
9
Applied Force (kN)
Specimen Deflection (mm)



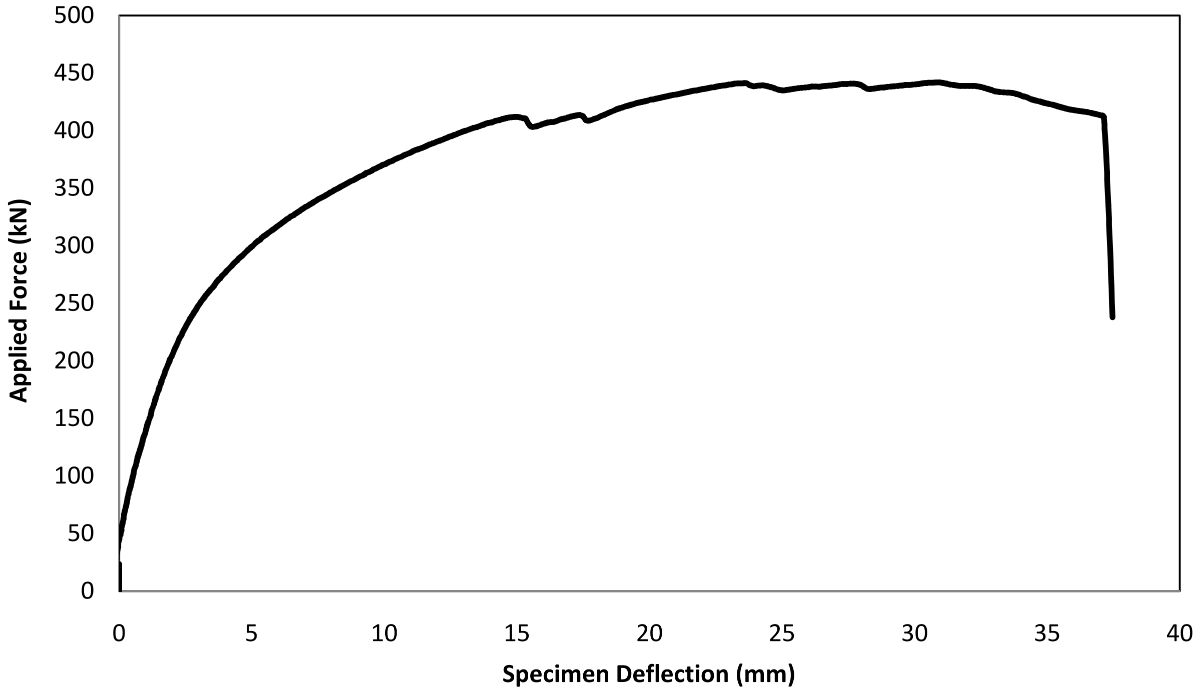
300
250
200
150
100
50
0
0
2
4
6
8
10
12
14
Applied Force (kN)
Specimen Deflection (mm)



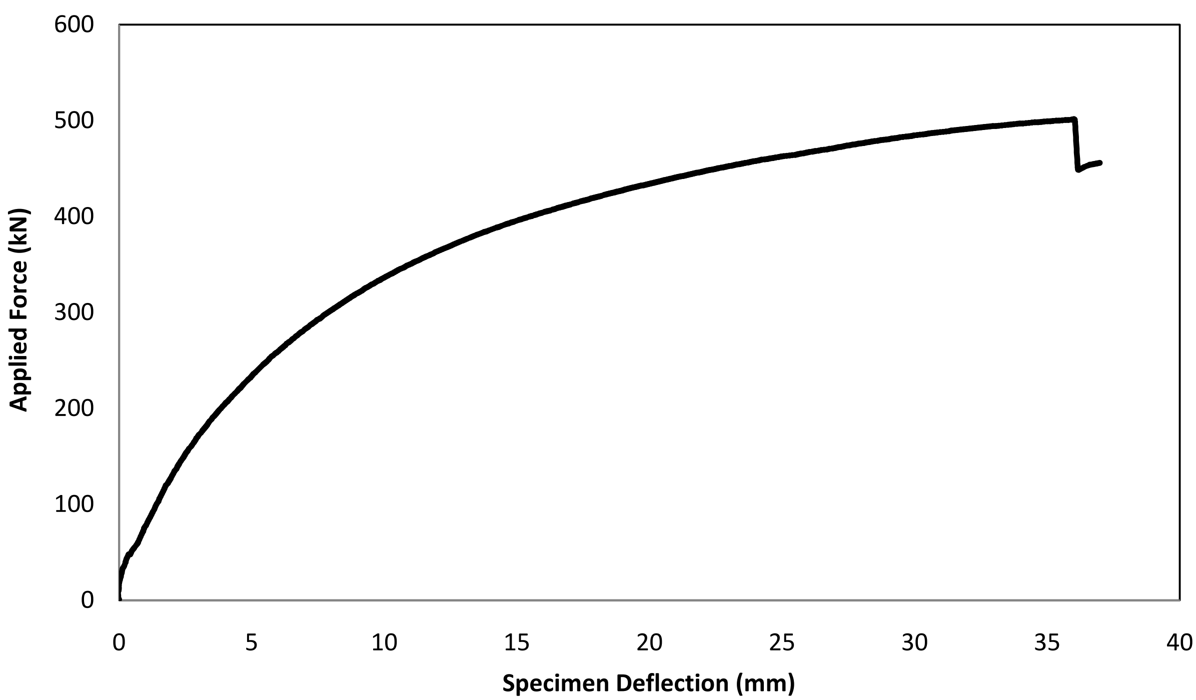
300
250
200
150
100
50
0
Applied Force (kN)
0
5
10
15
20
25
Specimen Deflection (mm)



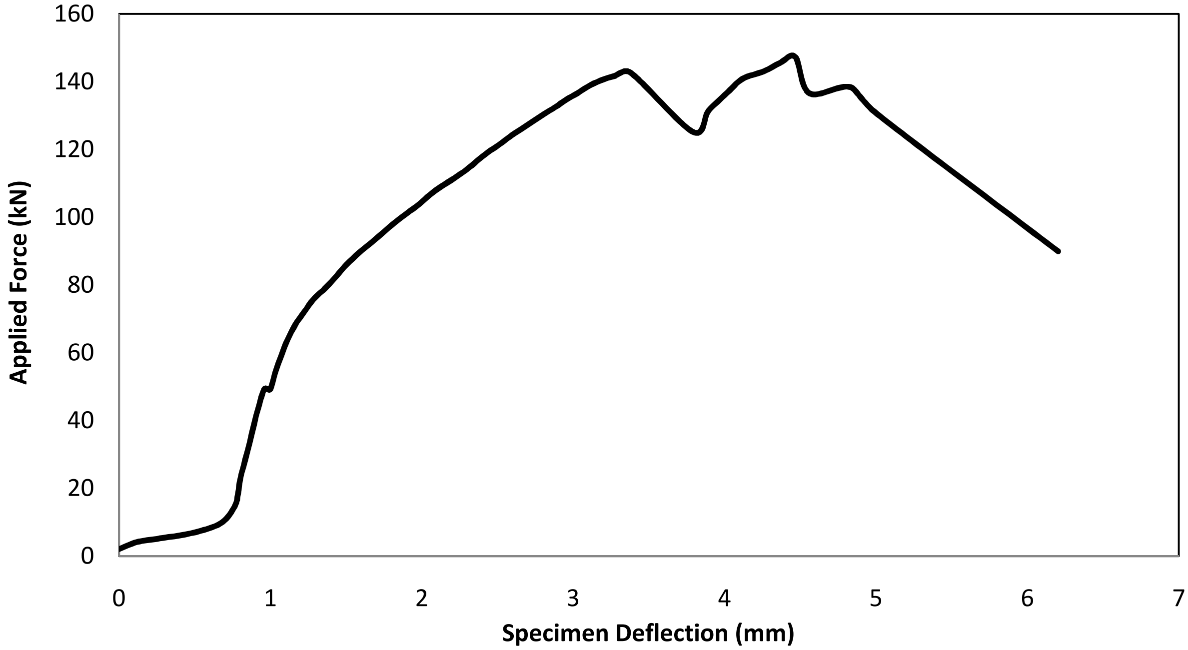
350
300
250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
14
16
Specimen Deflection (mm)



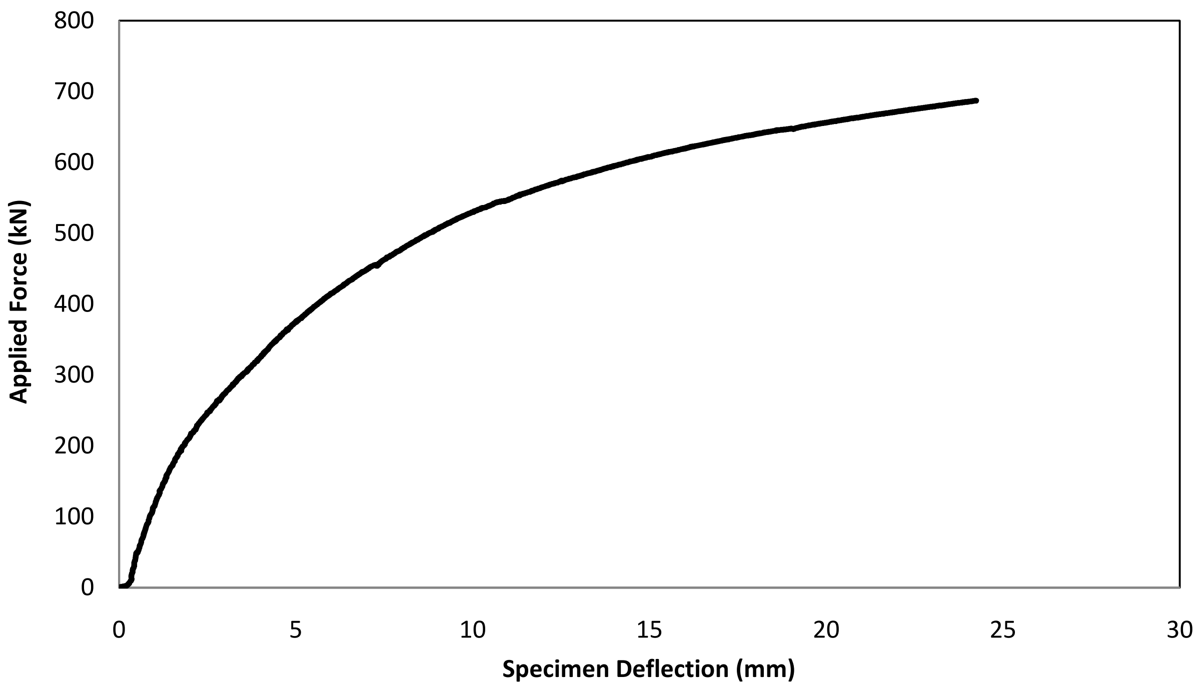
Applied Force (kN)
Specimen Deflection (mm)
0
50
100
150
200
250
300
350
400
450
500
0
5
10
15
20
25
30
35
40



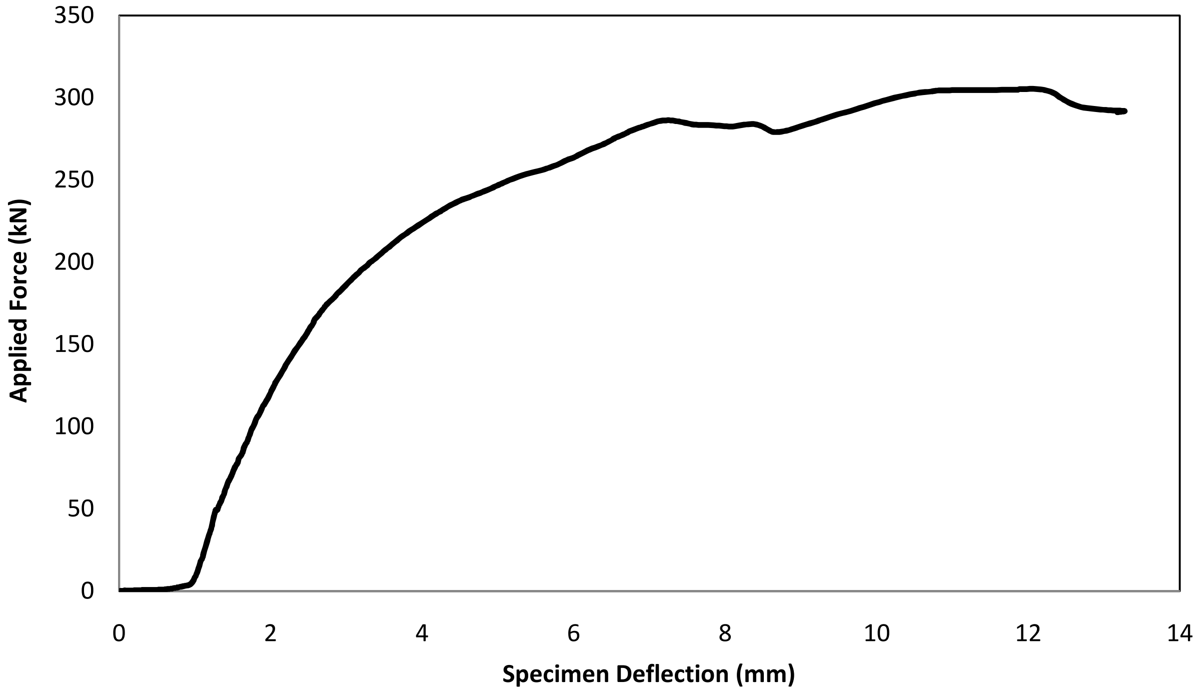
Applied Force (kN)
Specimen Deflection (mm)
0
100
200
300
400
500
600
0
5
10
15
20
25
30
35
40

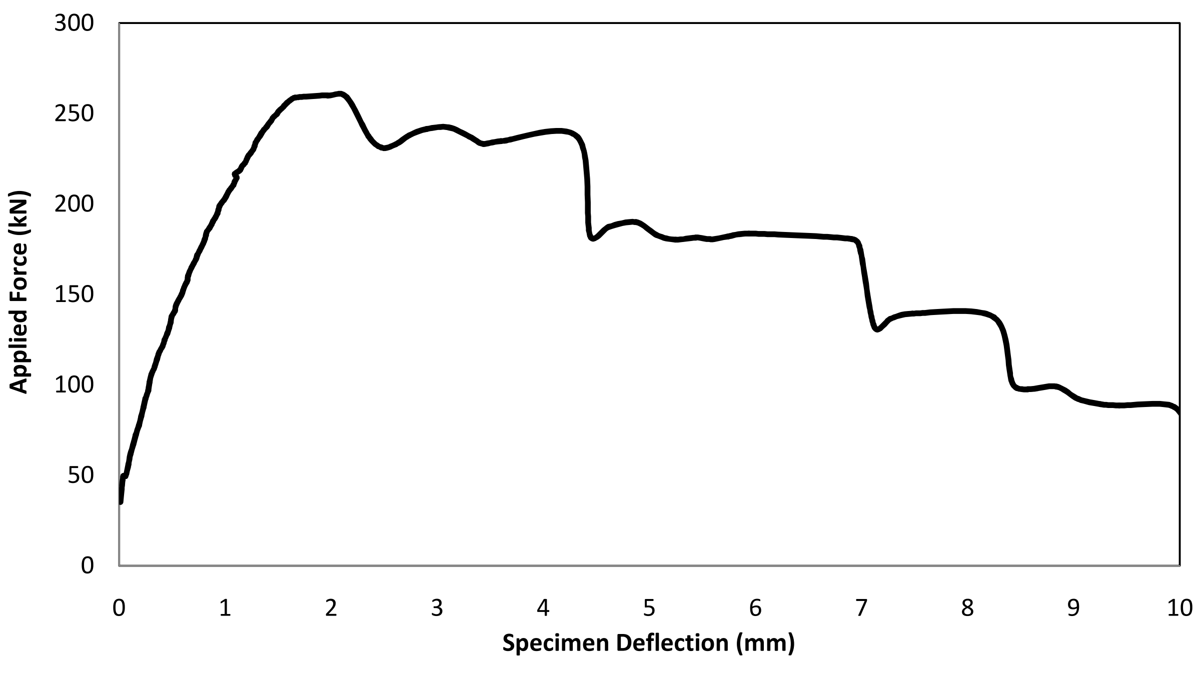


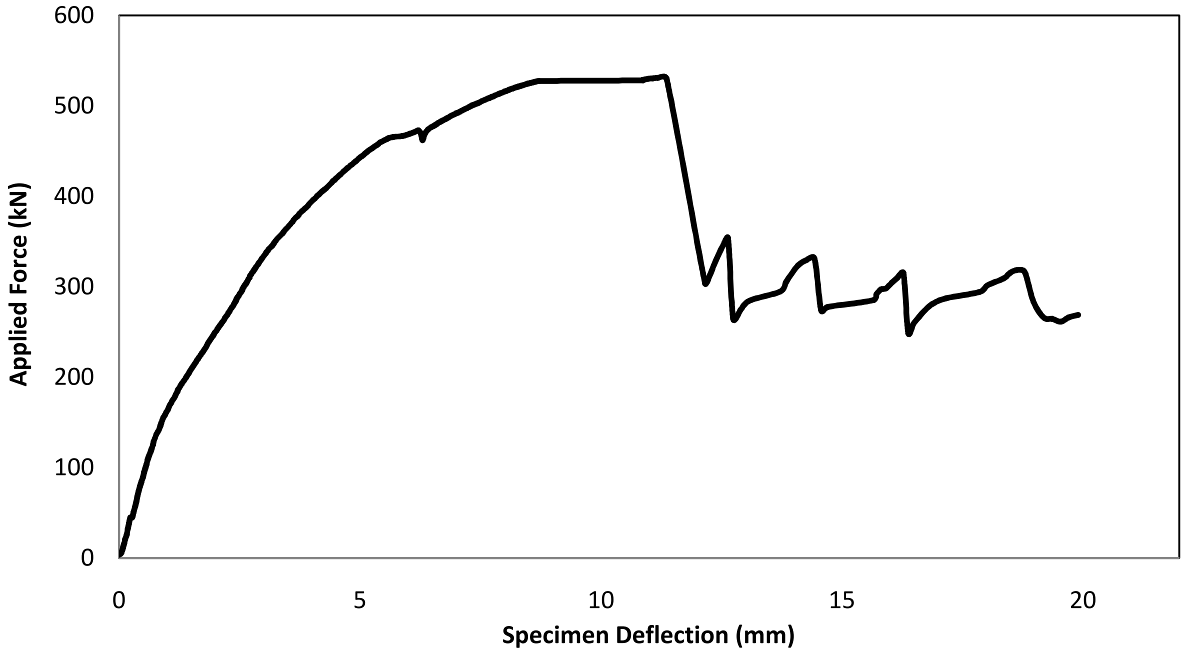
160
140
120
100
80
60
40
20
0
0
1
2
3
4
5
6
7
Applied Force (kN)
Specimen Deflection (mm)



800
700
600
500
400
300
200
100
0
0
5
10
15
20
25
30
Applied Force (kN)
Specimen Deflection (mm)

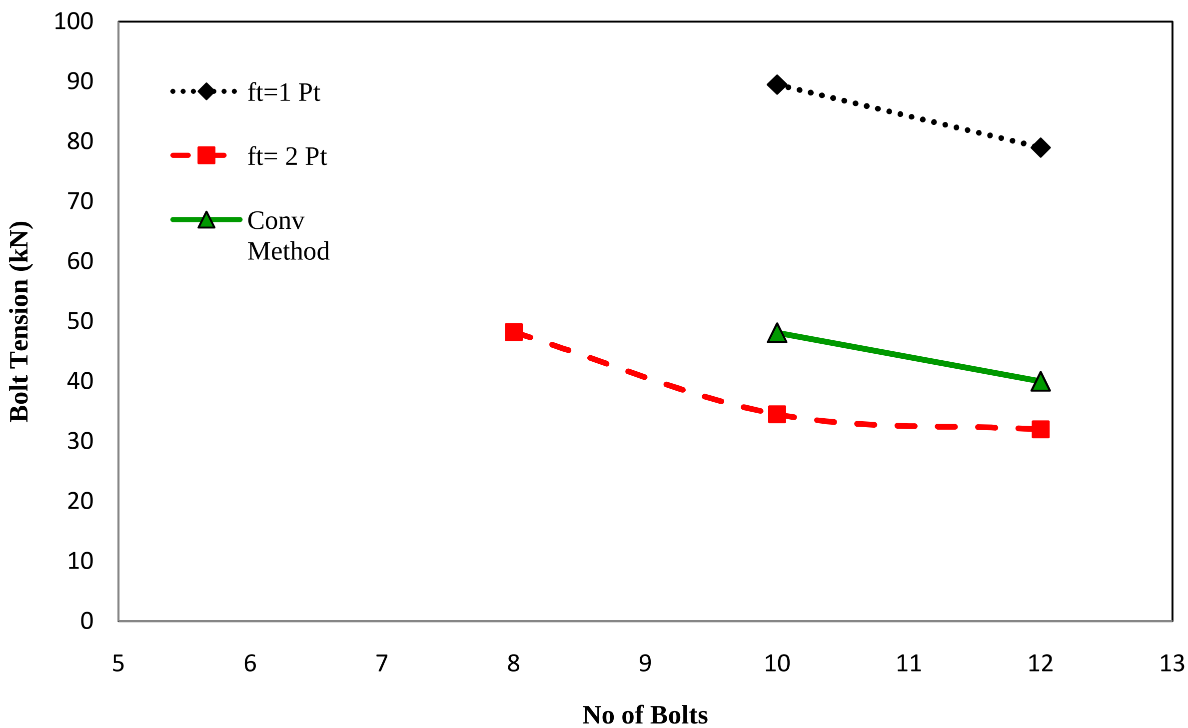

350
300
250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
14
Specimen Deflection (mm)


300
250
200
150
100
50
0
Applied Force (kN)
0
1
2
3
4
5
6
7
8
9
10
Specimen Deflection (mm)

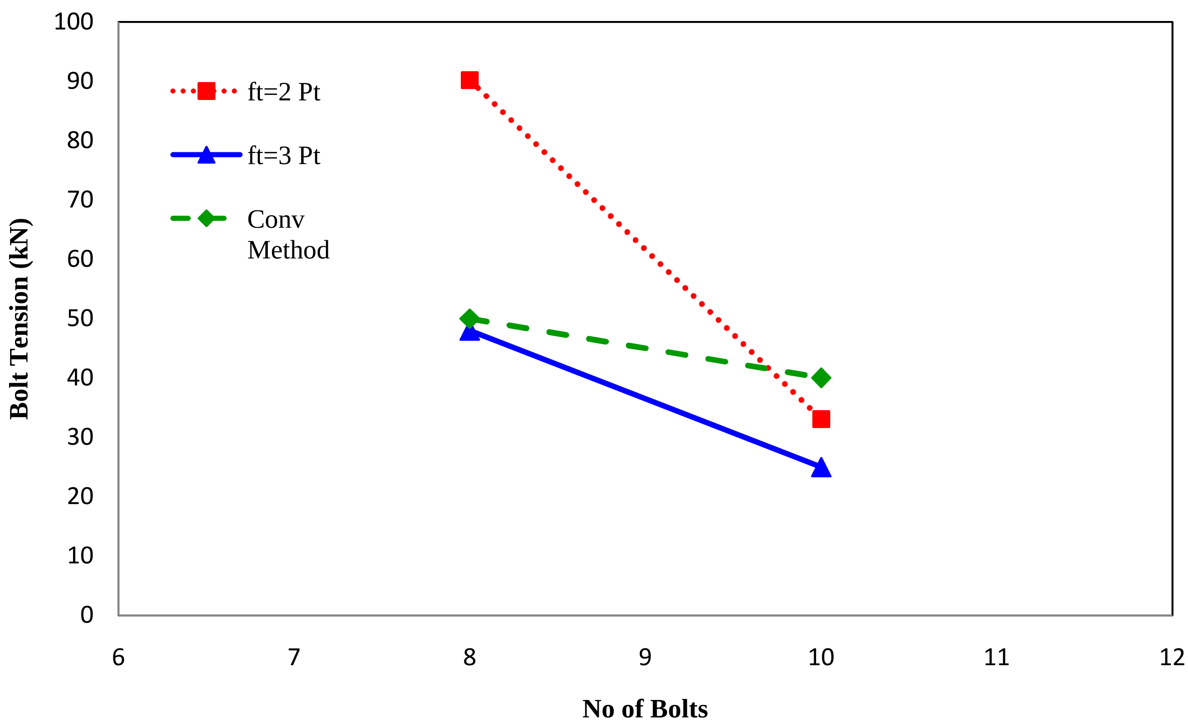


Applied Force (kN)
600
500
400
300
200
100
0
0
5
10
15
20
Specimen Deflection (mm)

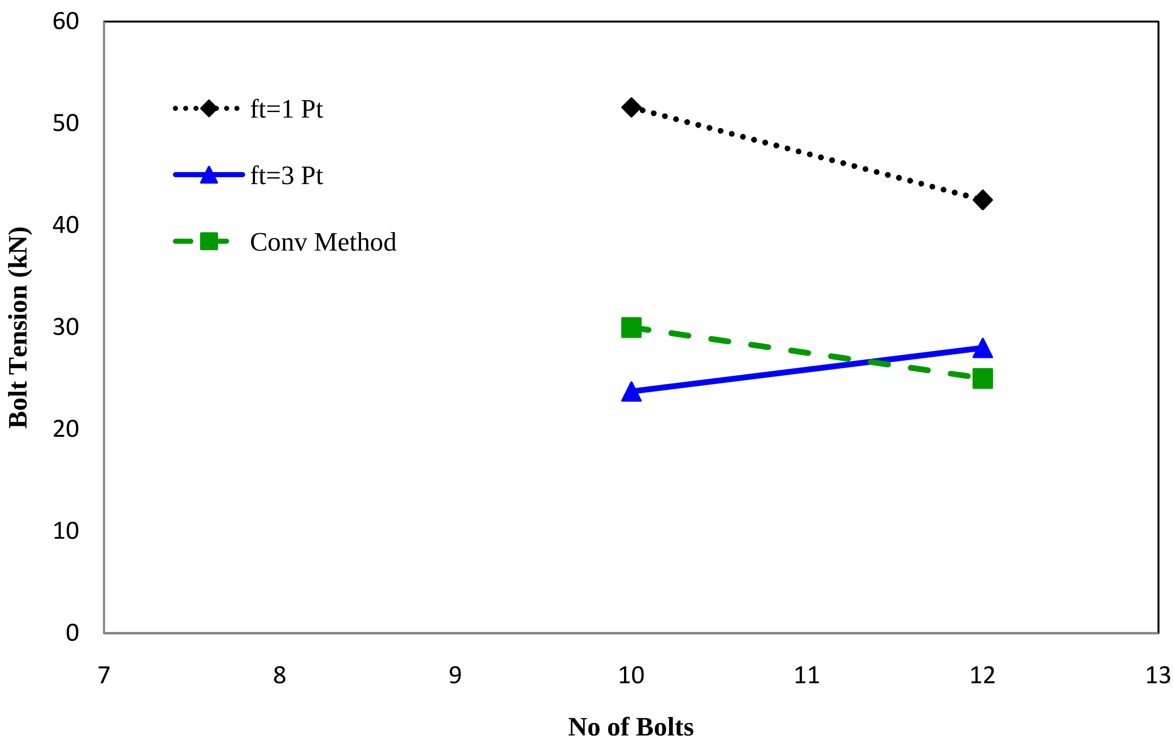
No of bolts	Bolt tension in kN $f_t=1P_t$	$f_t=2P_t$	Conventional Method
10	89.5	34.5	48.11
12	79	32	40



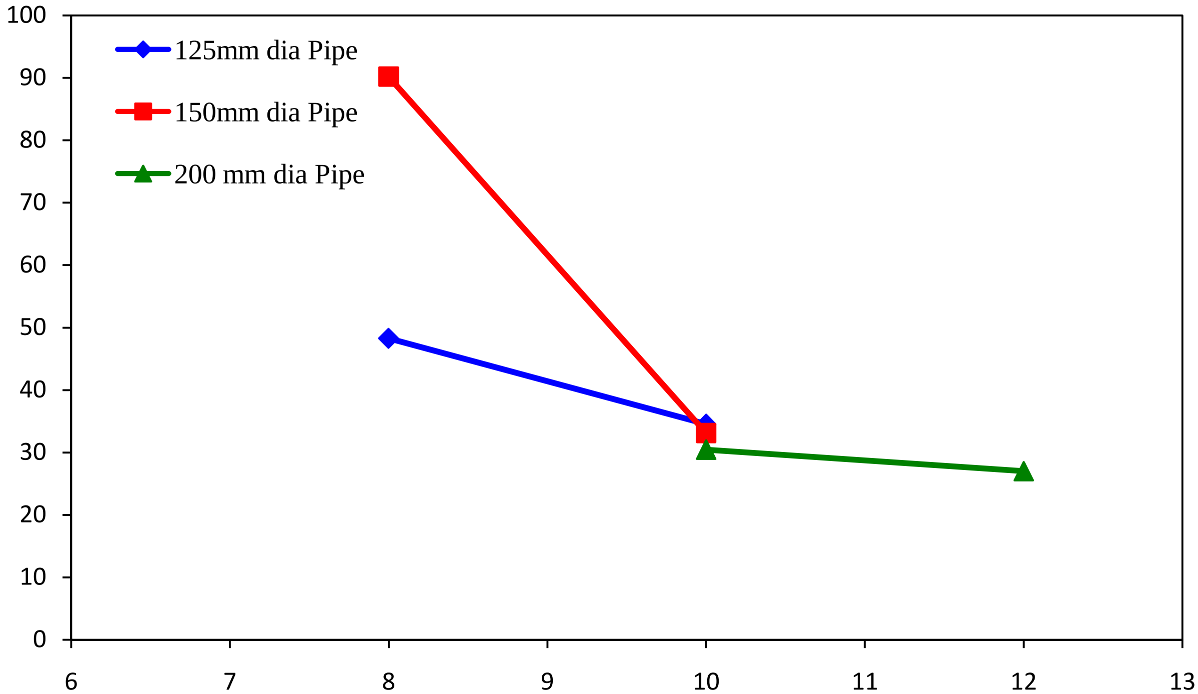
No of bolts	Bolt tension in kN			Conventional Method
	f_t=1P_t	f_t=2P_t	f_t=3P_t	
8	-	90.21	48	50
10	38	33.1	25	40



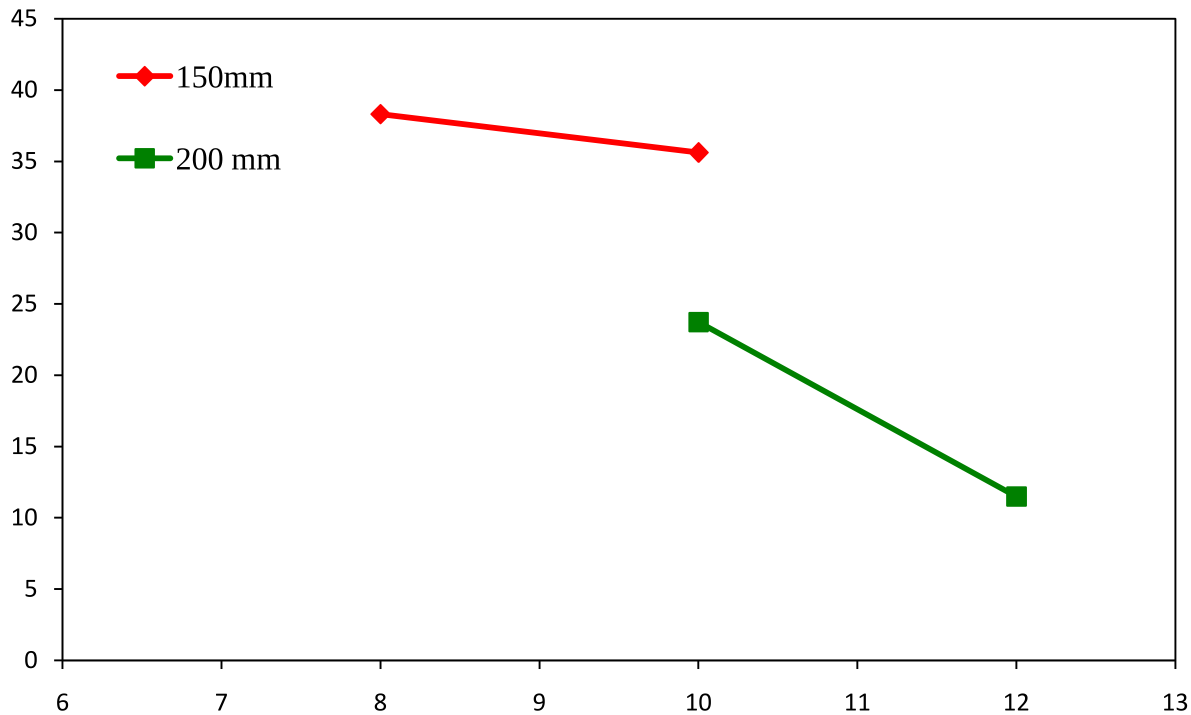
No of bolts	Bolt tension in kN			Conventional Method
	$f_t=1P_t$	$f_t=2P_t$	$f_t=3P_t$	
10	51.6	30	23.73	30
12	42.5	27	28	25



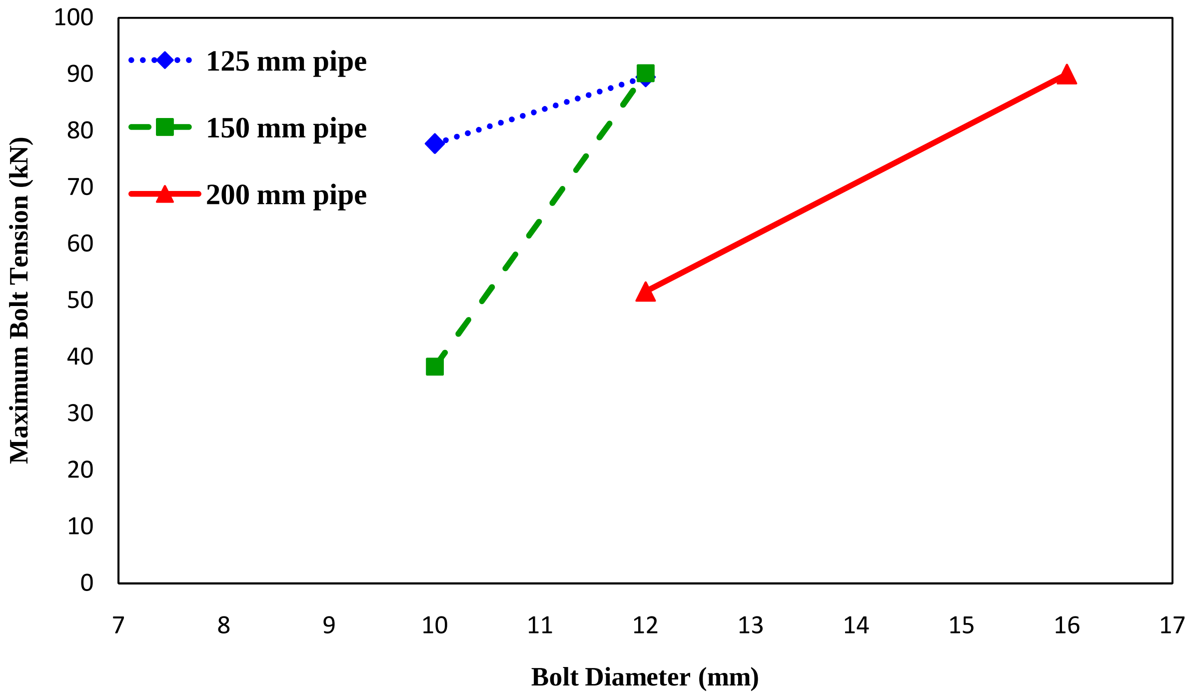
No of bolts	Bolt tension in kN		
	125 mm dia pipe	150 mm dia pipe	200 mm dia pipe
8	48.25	90.21	nil
10	34.5	33.1	30.4
	nil	nil	27



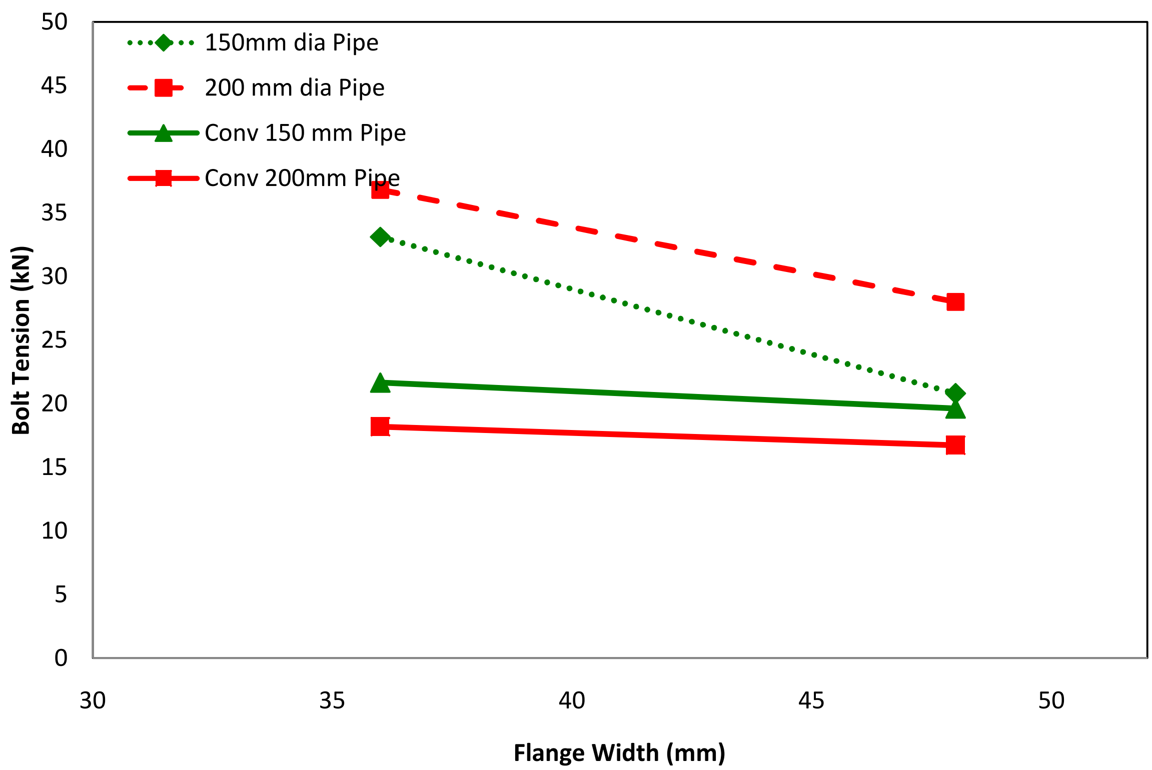
No of bolts	Bolt tension in kN		
	125 mm dia pipe	150 mm dia pipe	200 mm dia pipe
8	-	38.31	-
10	-	35.6	23.73
12	-	-	11.5



Diameter of bolt	Bolt tension in kN		
	125 mm dia pipe	150 mm dia pipe	200 mm dia pipe
10	77.73	-	-
12	89.5	38.31	51.6
16	-	90.21	90



Flange Width (mm)	Bolt tension in kN			
	150 mm dia pipe	200mm dia pipe	Conv for 150mm pipe	Conv for 200mm pipe
36	33.1	36.8	21.67	18.2
48	20.8	28	19.62	16.74



4.8.1 Effect of number of bolts on bolt tension

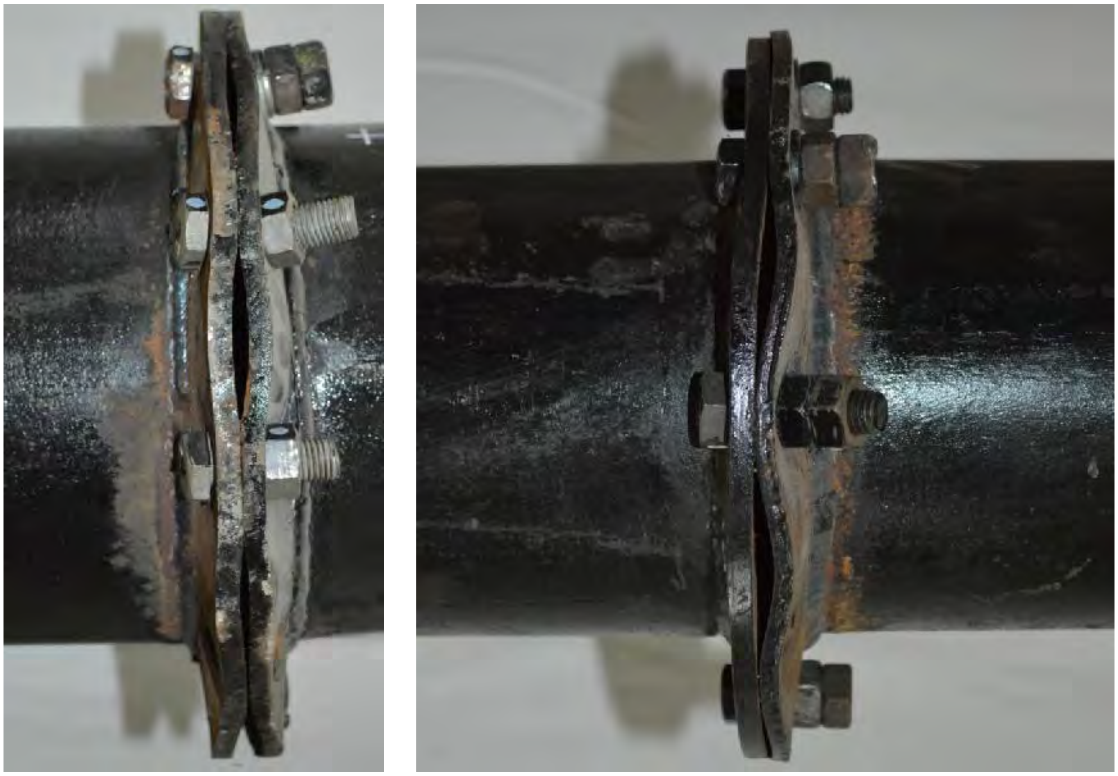
The curves, generated for bolt tensions against different number of bolts from the experimental results, are presented in figure 4.54 through figure 4.56. The trend lines of the curves are demonstrated a downtrend with a parabolic nature for increasing number of bolt under the study parameters. It means that bolt tension decreases with the increase in number of bolts as logically expected. Because for increasing the number of bolt for same applied load, the individual bolt tension reduces than previous tension, since load is distributed between more numbers of bolts.

4.8.2 Effect of flange thickness on bolt tension

With reference to figure 4.57 through figure 4.58, it may be suggested that, bolt tension also depends on flange thickness. But it does not follow any particular trend. From the presented graphs, both increase and decrease in tension value have been observed at random when flange thickness is increased or decreased. Apart from that it has been observed that, the band width of bolt tension values (ranging from $f_t = 1p_t$, $f_t = 2p_t$ to $f_t = 3p_t$) tends to decrease as the pipe diameter increases. This clearly indicates that the effect of flange thickness on bolt tension is reducing with the increase in pipe diameter. Generally if the flange thickness is relatively small (that is if the flange is less stiff) then considerable bending will occur on the flange and at the same time connected bolts to the flange will tend to be elongated. As a consequence the contact area between two flanges will be reduced and the developed prying force near the edge of the flange will increase as seen in figure 4.60. Therefore in case of thin flanges the bolt tension value is comparatively large. On the other hand, in case of thick flange, the flange is stiffer to resist the applied loading and the contact area between two flanges does not get more reduced due to applied load. In addition, the prying force develops small in comparison with the thin flange. Hence the bolt tension value develops relatively small in case of thick flange.

4.8.3 Effect of bolt diameter on bolt tension

With reference to figure 4.59, as the diameter of bolt increases an insignificant increase in bolt tension has been observed for any particular case of diameter of pipe, number of bolt, flange width and flange thickness. This occurs mainly due to the fact that if the diameter of bolt increases, the stiffness of the bolt generally increases. As a result, a little more tension value develops on greater diameter bolts.



)0, ıt

thic:ness t

$Q = \frac{M_p}{n}$, wh

CHAPTER 5

FINDINGS AND RECOMMENDATION

5.1 GENERAL

The experimental research work emanated with an aim to determine the maximum bolt tension of a flanged pipe joint subjected to bending. Initially some variable parameters are chosen for a flanged pipe joint. The bolt tension is then determined for different pipe diameter (basically 125mm, 150 mm and 200 mm pipe) by considering different flange thickness, flange width, diameter of bolt and number of bolts. After completing the experiment, a good number of graphs are drawn for different pipe diameter representing bolt tension against number of bolts (for different flange thicknesses) and bolt tension against bolt diameter (for different flange width) to find out precisely the effect of various parameters (i.e. pipe diameter, diameter of bolt, number of bolts, flange width and flange thickness) on bolt tension. These graphs demonstrated that pipe diameter, flange thickness and number of bolts have significant effect on bolt tension for a flanged pipe joint. From the experimental analysis results**,** these graphs shall act as a ready reconnoiter to determine bolt tension for different applied moment.

5.2 GENERAL FINDINGS OF THE RESEARCH WORK

The outcomes of the research work are summarized as follows:

a. The research study provided us with an opportunity to understand the behavior of a bolted flanged pipe connection subjected to pure bending.
b. The effect of various parameters on the bolt tension as reflected in this research work are followings:
    - It is observed that bolt tension decreases with increasing the number of bolts.
    - Flange thickness has significant effect on bolt tension and it is noticed that the bolt tension decreases with increasing in flange thickness.

- The maximum bolt tension increases as diameter of bolt increases for any particular case of other parameters.
- The maximum bolt tension decreases as the flange width increases in maximum cases.

5.3 RECOMMENDATION FOR DETERMINING BOLT FORCE

Based on analysis of experimental results it may be suggested that for a flanged pipe joint subjected to pure bending moment, when flange thickness is same as pipe thickness bolt tension value obtained from conventional analysis should be doubled to obtain a reasonable estimate of bolt tension. When flange thickness is thrice the pipe thickness bolt tension calculated by conventional method may be assumed reasonable. For intermediate thickness of flange a linear interpolation may be performed.

5.4 SCOPE FOR FUTURE INVESTIGATION

The following recommendations for future research work may be suggested:

a. In the present experiment, we used BTM-6C strain gauge which has difficulties in drilling tiny hole in bolt shank and proper placement. In future external force measuring gauge may be used instead of washer.
b. In the present experiment, only three types of pipe having 125mm, 150 mm and 200 mm diameter are used to prepare flanged pipe joints. In those flanges only bottom and side bolts forces are measured. In future, different diameter of pipes with strain gauges in each bolt may be used to develop bolt force distribution between different bolts having different pipe thickness.
c. Also further comprehensive investigation can be performed to compare results obtained from experimental investigation with FEA of similar model. In the process, formulating a general equation based on FEA and validated by experimental results.

APPENDIX A

Flanged Pipe Joint Specimen Data

A1: Different parameters used in the present study are listed below:

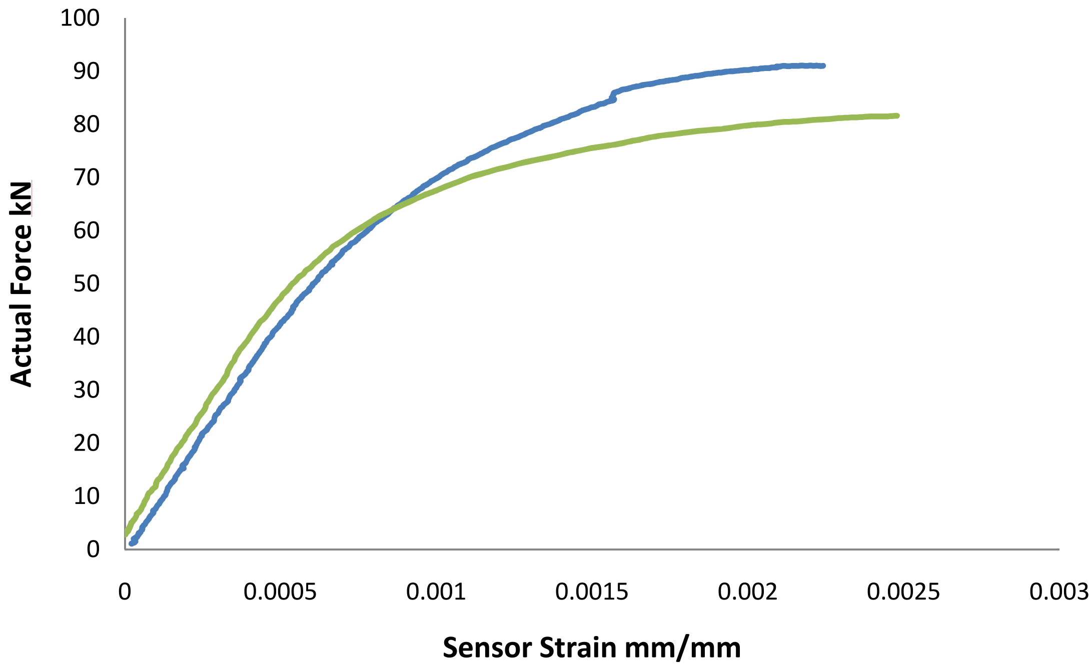
Nominal Diameter of Pipe, d_p (mm)	Pipe wall thickness, p_t (mm)	Flange Width f_w (mm)	Flange thickness, f_t (mm)	Diameter of Bolt, d_b (mm)	Number of bolts, n
125	6.55	30	1,2, 3 times of p_t	9.5	8,10
		36		12.7	8,10
150	7.11	36	1,2, 3 times of p_t	12.7	10, 12
		48		15.875	10,12
200	8.18	36	1,2, 3 times of p_t	12.7	10, 12
		48		15.875	10,12

A2: Specimen Designation Details

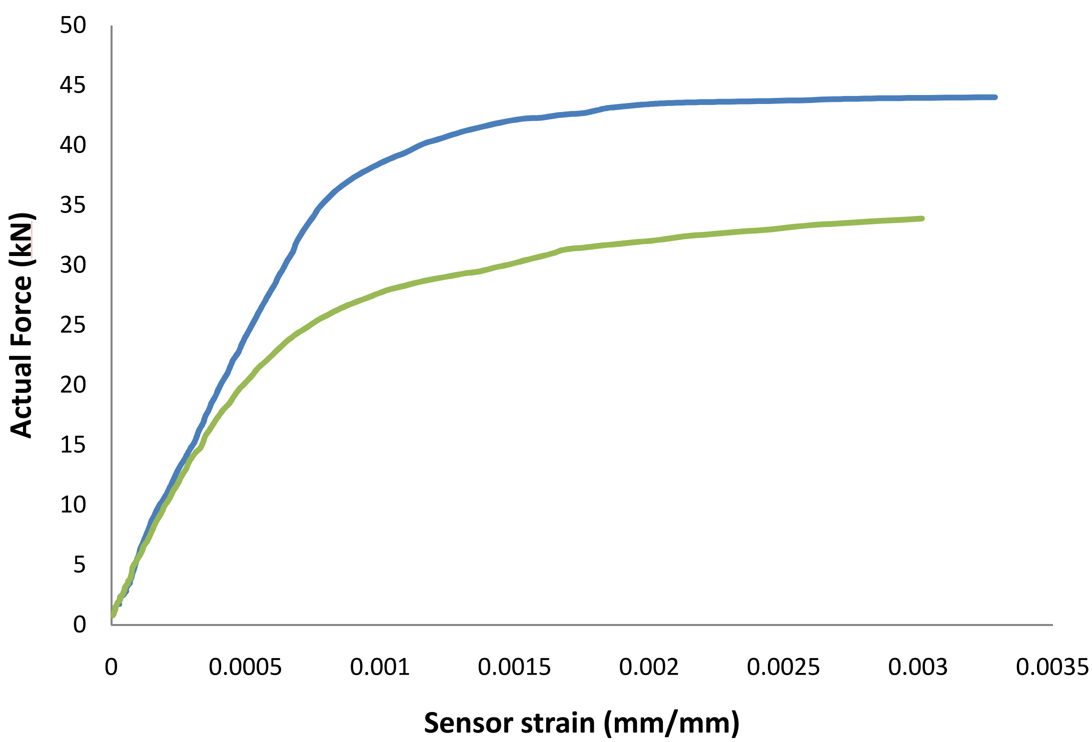
A special designation is used for the entire specimen flange pipe joints. The designation system provides the information about pipe size, number of bolts, bolt diameter and flange thickness. The specimen designation is shown in figure 3.13 of chapter 3.Below total 34 specimens' details are shown:

Ser	Specimen Designation	Pipe Dia, mm	No of Bolts	Bolt Dia, mm	Flange Thickness, mm
1.	S-1	125	10	10	6
2.	S-2	125	10	10	12
3.	S-3	125	8	10	18
4.	S-4	125	10	12	6
5.	S-5	150	8	16	14
6.	S-6	150	10	12	21
7.	S-7	150	10	12	14

Ser	Specimen Designation	Pipe Dia, mm	No of Bolts	Bolt Dia, mm	Flange Thickness, mm
8.	S-8	150	8	16	21
9.	S-9	200	12	16	24
10.	S-10	200	10	12	18
11.	S-11	200	10	16	24
12.	S-12	200	12	12	24
13.	S-13	125	8	10	6
14.	S-14	150	8	12	21
15.	S-15	150	10	12	14
16.	S-16	150	10	16	7
17.	S-17	150	10	16	14
18.	S-18	125	10	12	6
19.	S-19	125	10	12	6
20.	S-20	150	8	12	12
21.	S-21	150	8	16	14
22.	S-22	150	8	12	7
23.	S-23	200	12	12	16
24.	S-24	200	12	16	8
25.	S-25	200	10	16	8
26.	S-26	200	12	12	8
27.	S-27	200	12	12	8
28.	S-28	200	10	12	8
29.	S-29	200	10	12	24
30.	S-30	200	12	16	16
31.	S-31	125	8	12	6
32.	S-32	125	8	10	16
33.	S-33	125	10	12	12
34.	S-34	200	10	12	8



100
90
80
70
60
50
40
30
20
10
0
0
0.0005
0.001
0.0015
0.002
0.0025
0.003
Actual Force kN
Sensor Strain mm/mm



50
45
40
35
30
25
20
15
10
5
0
0
0.0005
0.001
0.0015
0.002
0.0025
0.003
0.0035
Actual Force (kN)
Sensor strain (mm/mm)





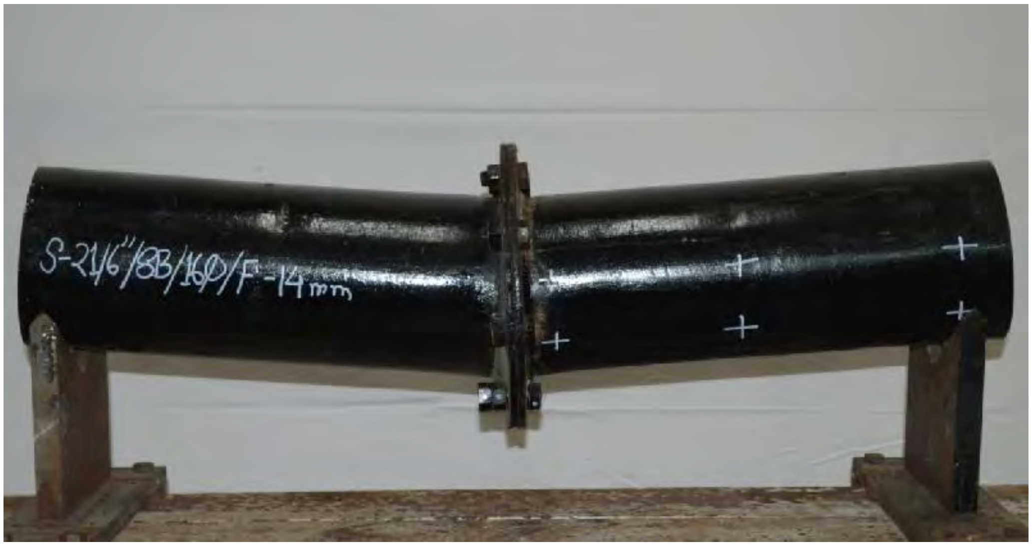
S-19/5"/10B/124/F-6mm


S-33/5"/10B/12Ø


S-3/5"/8B/10Ø/F-18mm


S-16/6"/10B/16mm/F--7mm


S-15/6"/10B/F-14mm


S-21/6"/8B/16Ø/F-14mm


S-22/6"/8B/12Ø/F-7

Figure B17: Deflection of 150 mm pipe with 10- 12 mm bolts and 14 mm flange(S-7)

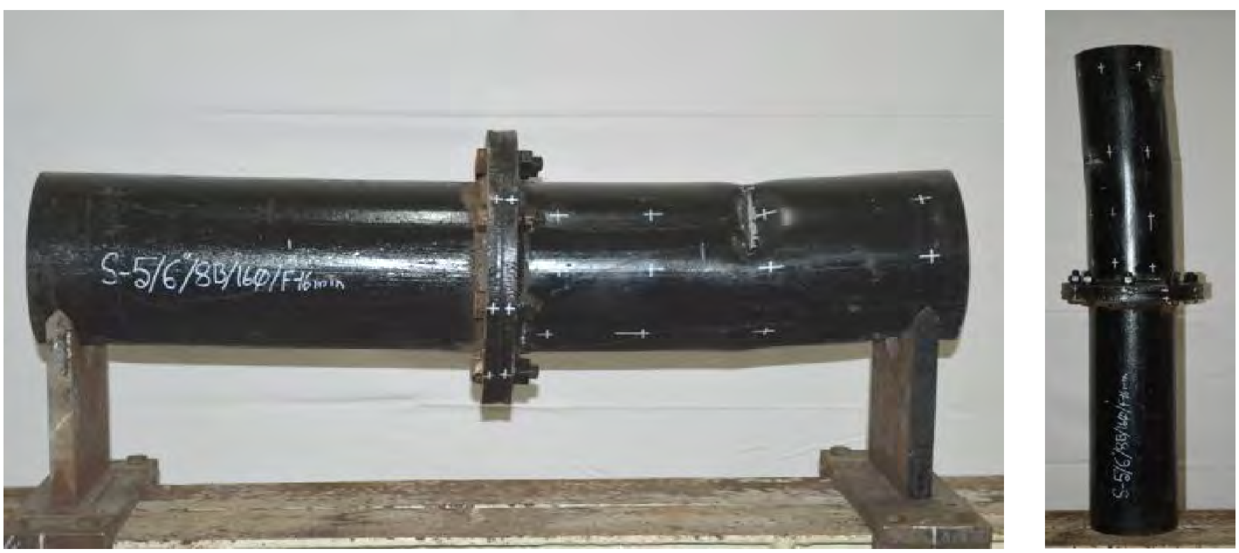


Figure B18: Deflection of 150 mm pipe with 8- 16 mm bolts and 14 mm flange(S-5)

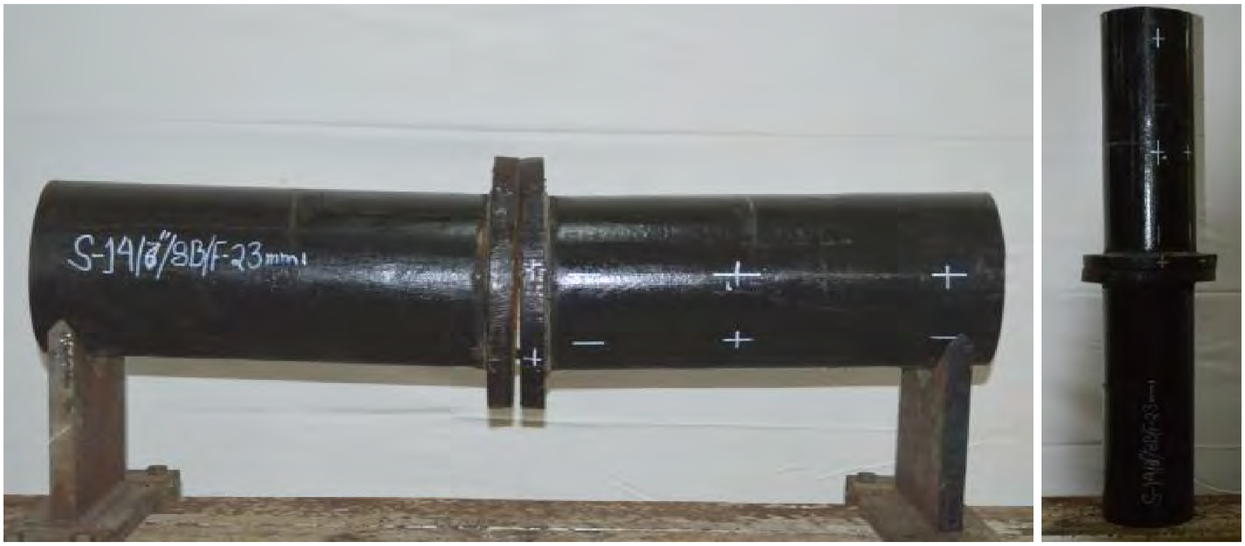


Figure B19: Deflection of 150 mm pipe with 8- 12 mm bolts and 21 mm flange(S-14)

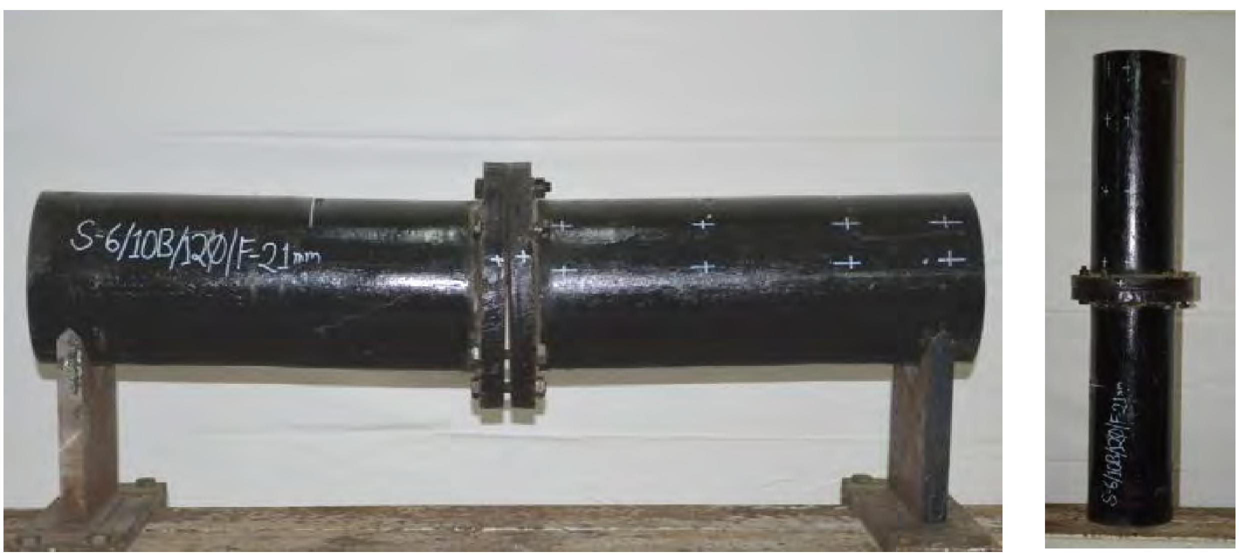


Figure B20: Deflection of 150 mm pipe with 10- 12 mm bolts and 21 mm flange(S-6)


S-8/6"/8B/16Ø/21mm


S-8/6"/8B/16Ø/21mm


S-27/8"/12B/12mm/F-8mm


S-25/8"/10B/16Ø/F-8mm


S-24/8"/12B/16Ø/F-8mm

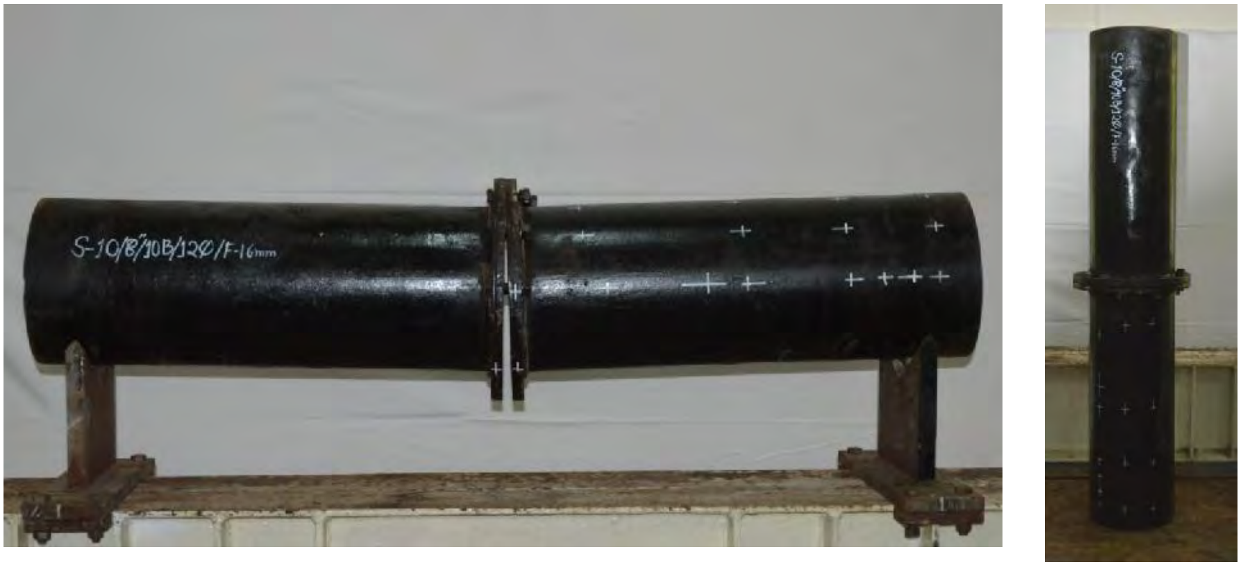
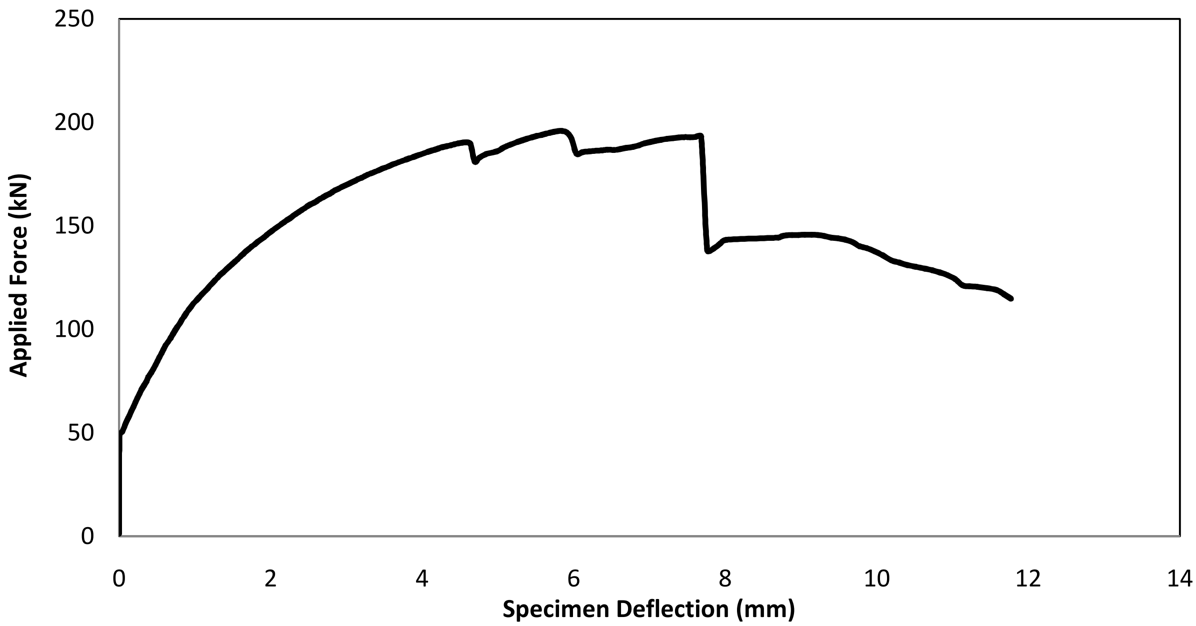


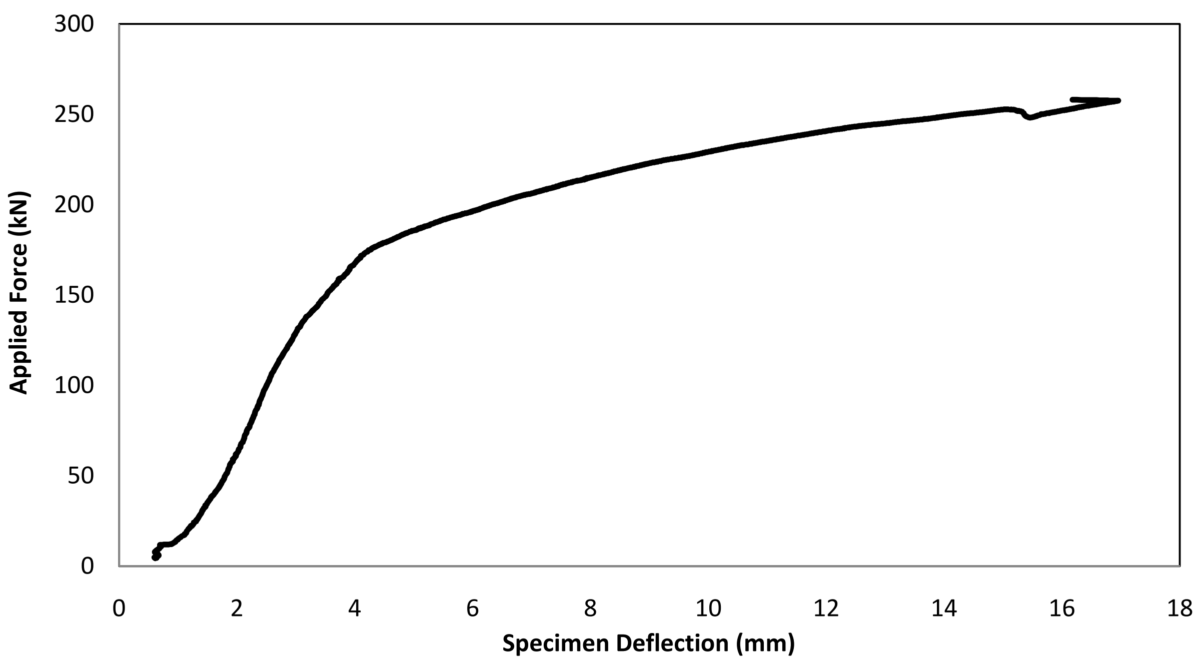
Figure B27: Deflection of 200 mm pipe with 10- 12 mm bolts and 16 mm flange(S-10)

Figure B28: Deflection of 200 mm pipe with 12- 12 mm bolts and 16 mm flange(S-23)

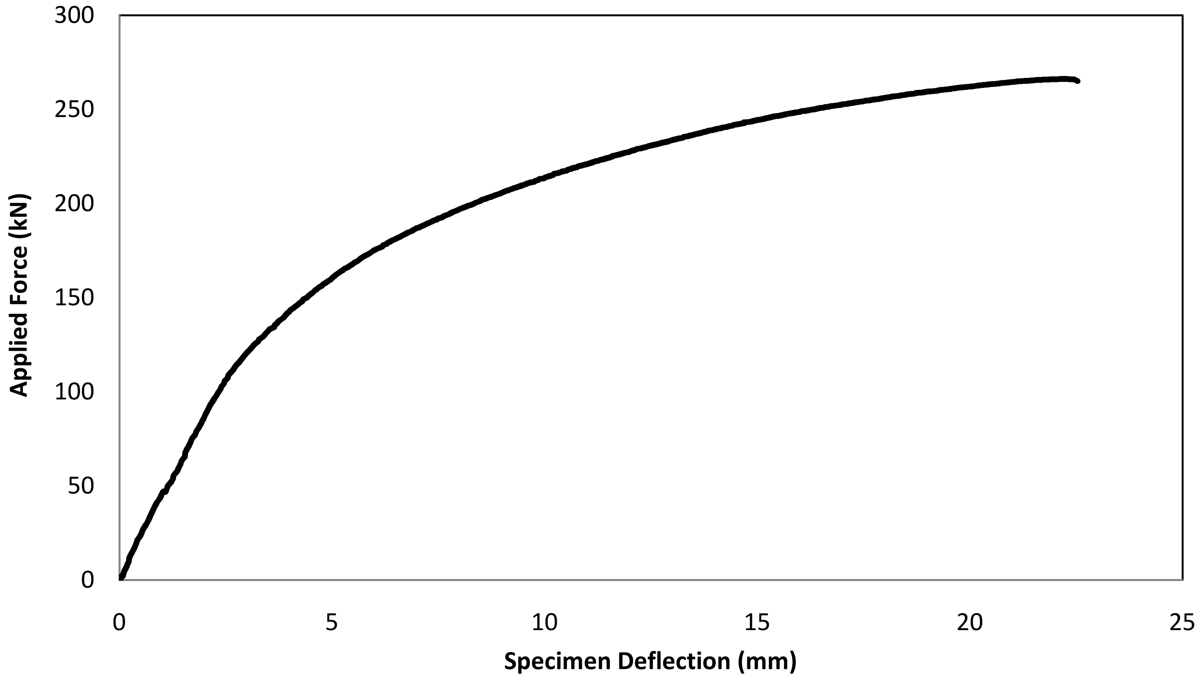

S-12/8"/12B/12Ø/F-24mm



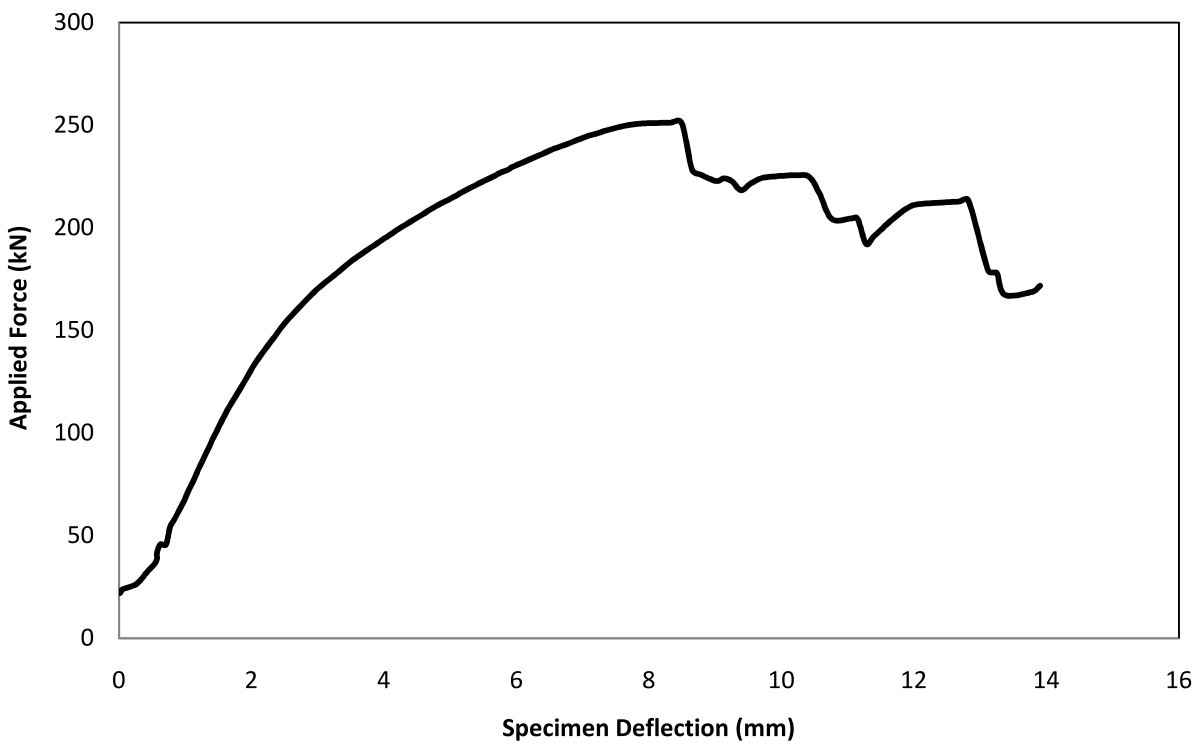
Applied Force (kN)
250
200
150
100
50
0
0
2
4
6
8
10
12
14
Specimen Deflection (mm)



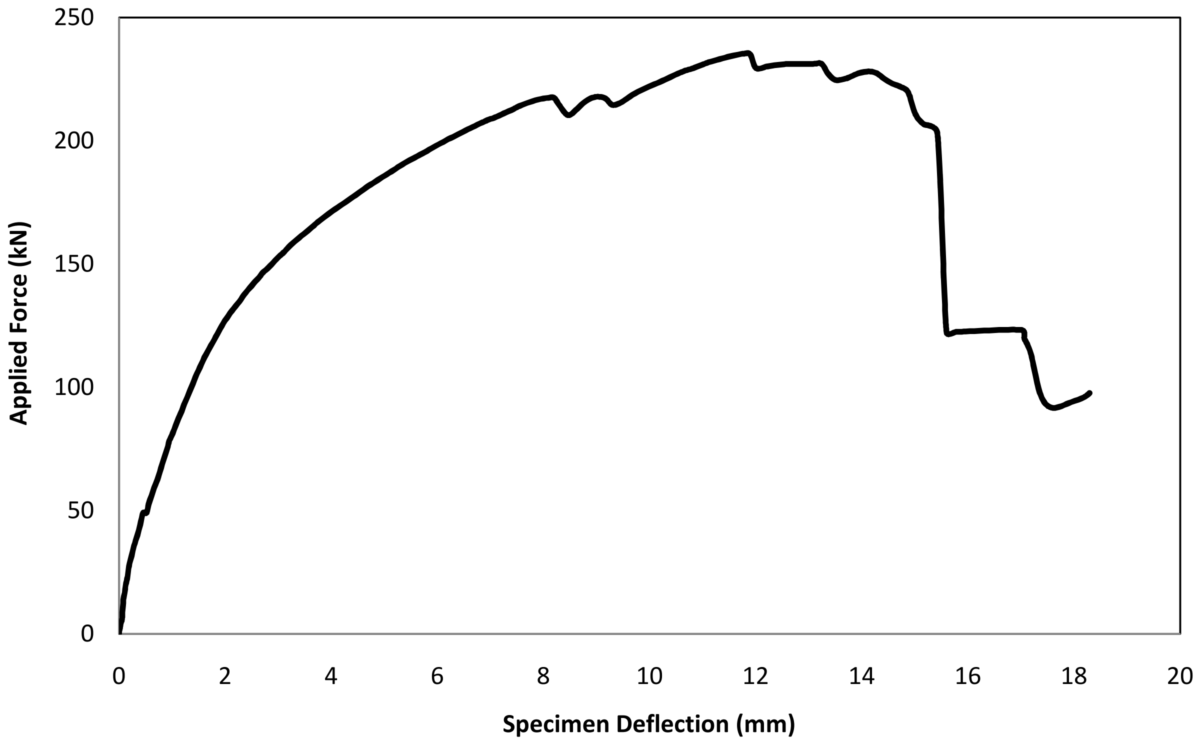
Applied Force (kN)
300
250
200
150
100
50
0
0
2
4
6
8
10
12
14
16
18
Specimen Deflection (mm)



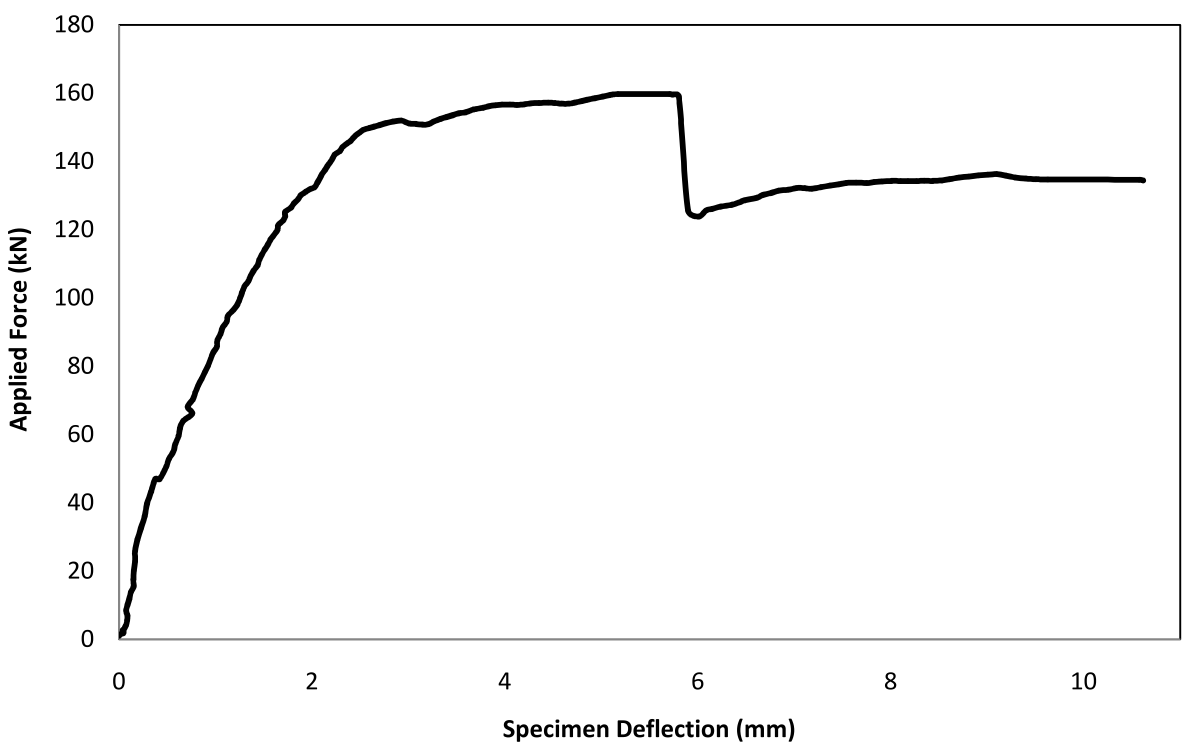
300
250
200
150
100
50
0
Applied Force (kN)
0
5
10
15
20
25
Specimen Deflection (mm)



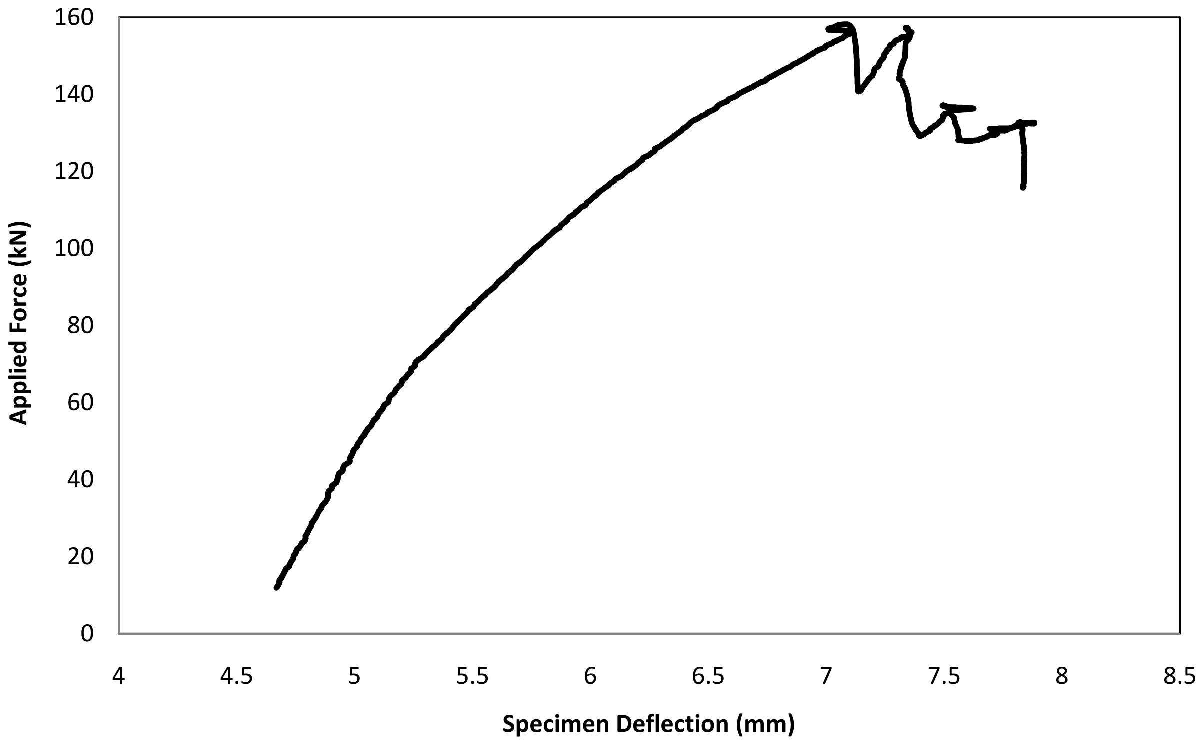
300
250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
14
16
Specimen Deflection (mm)



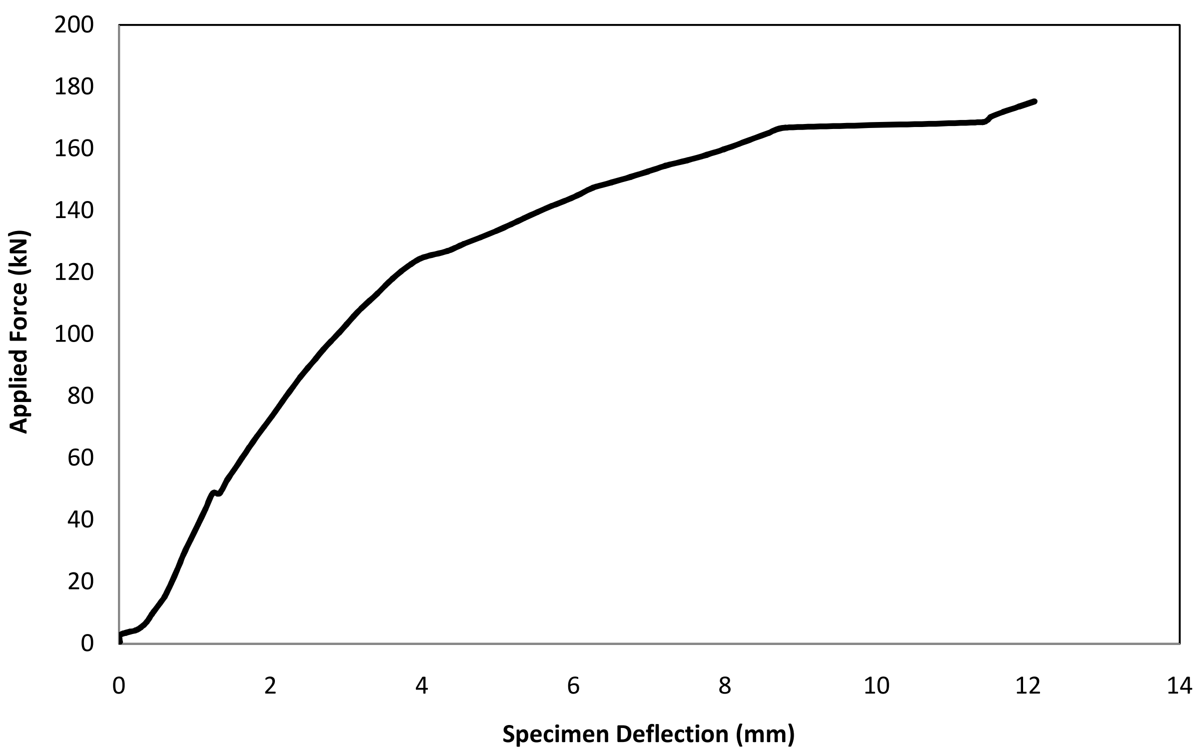
Applied Force (kN)
250
200
150
100
50
0
0
2
4
6
8
10
12
14
16
18
20
Specimen Deflection (mm)



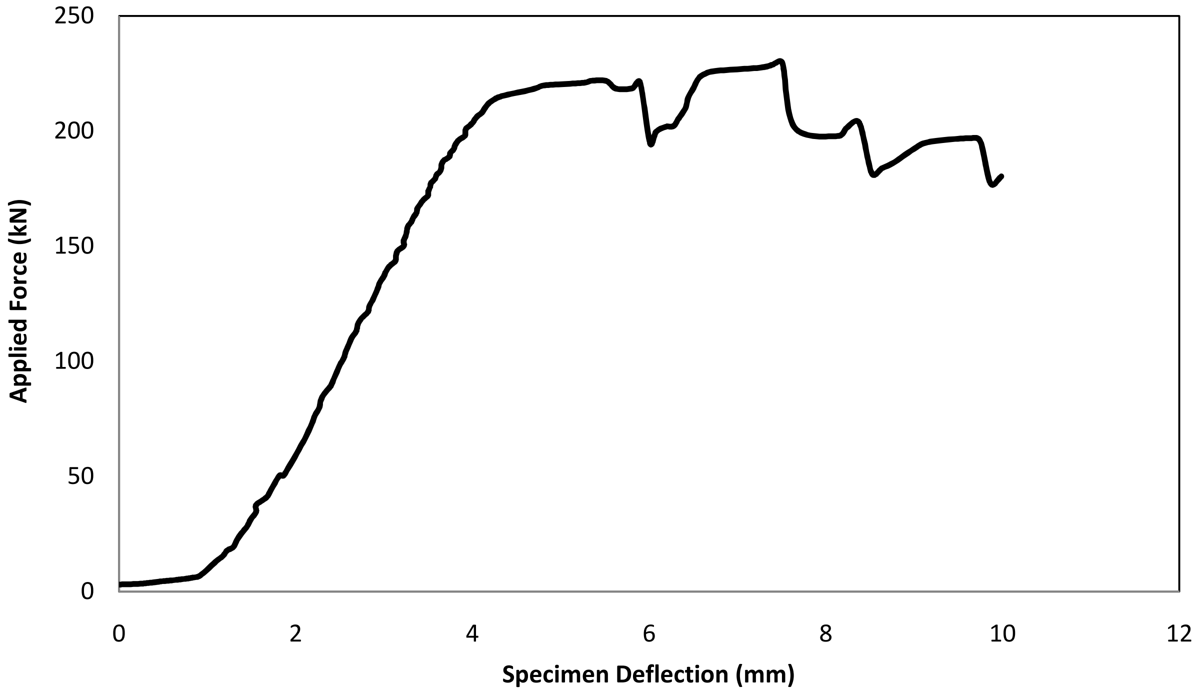
Applied Force (kN)
180
160
140
120
100
80
60
40
20
0
0
2
4
6
8
10
Specimen Deflection (mm)

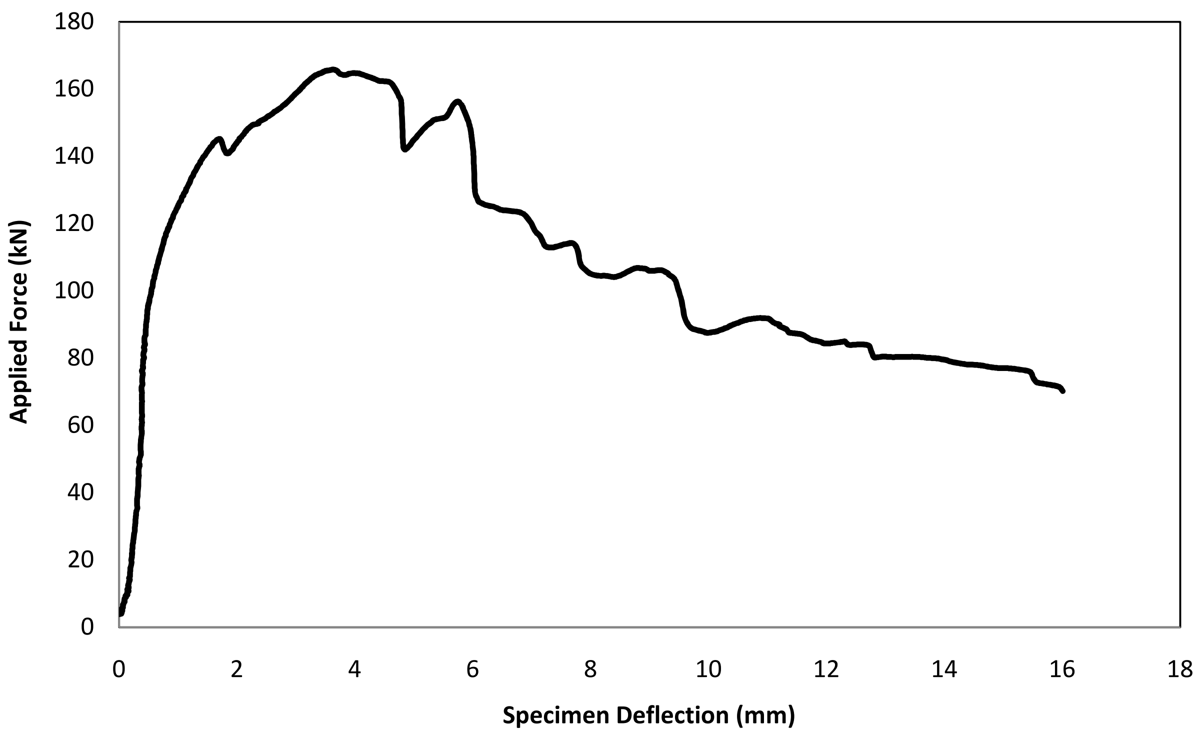


Applied Force (kN)
0
20
40
60
80
100
120
140
160
4
4.5
5
5.5
6
6.5
7
7.5
8
8.5
Specimen Deflection (mm)



Applied Force (kN)
0
20
40
60
80
100
120
140
160
180
200
0
2
4
6
8
10
12
14
Specimen Deflection (mm)


250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
Specimen Deflection (mm)


180
160
140
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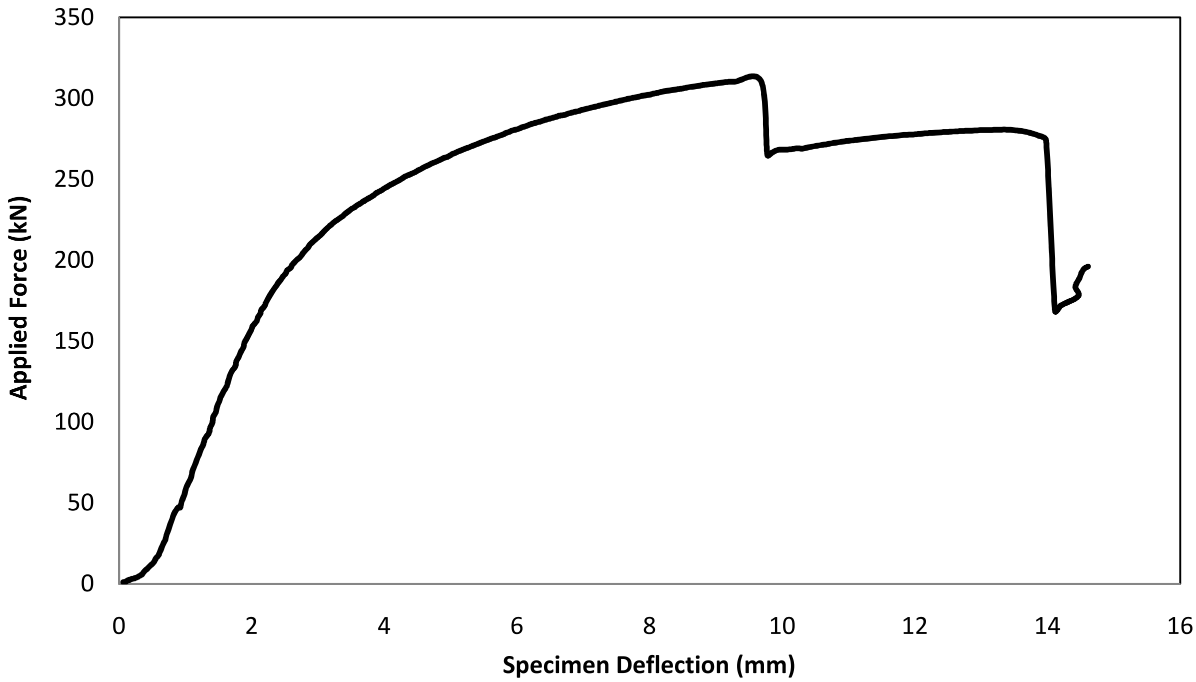
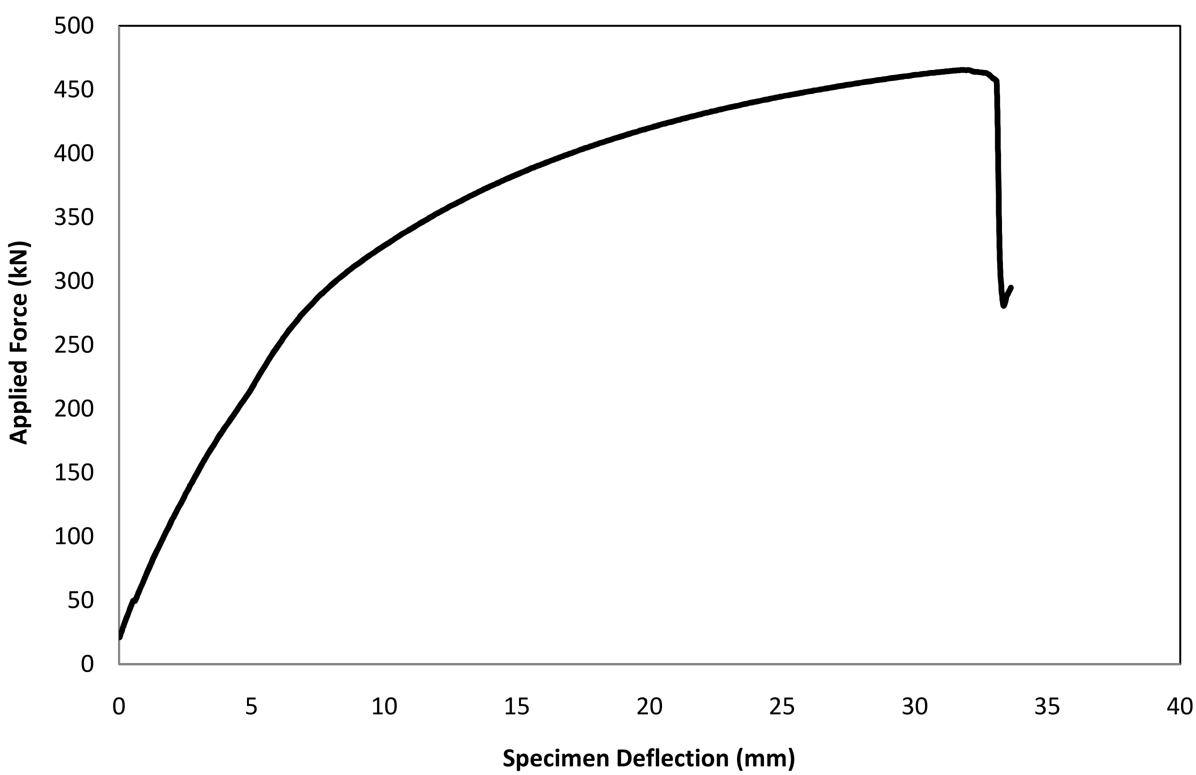
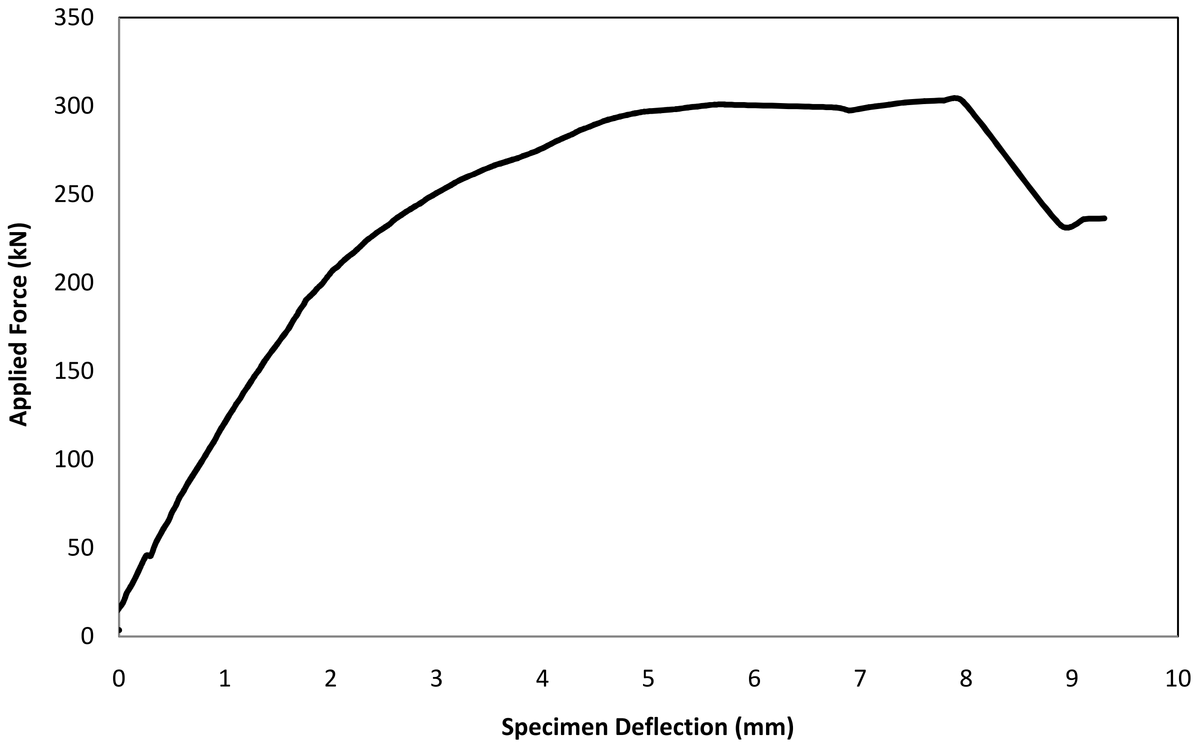
60
40
20
0
Applied Force (kN)
0
2
4
6
8
10
12
14
16
18
Specimen Deflection (mm)

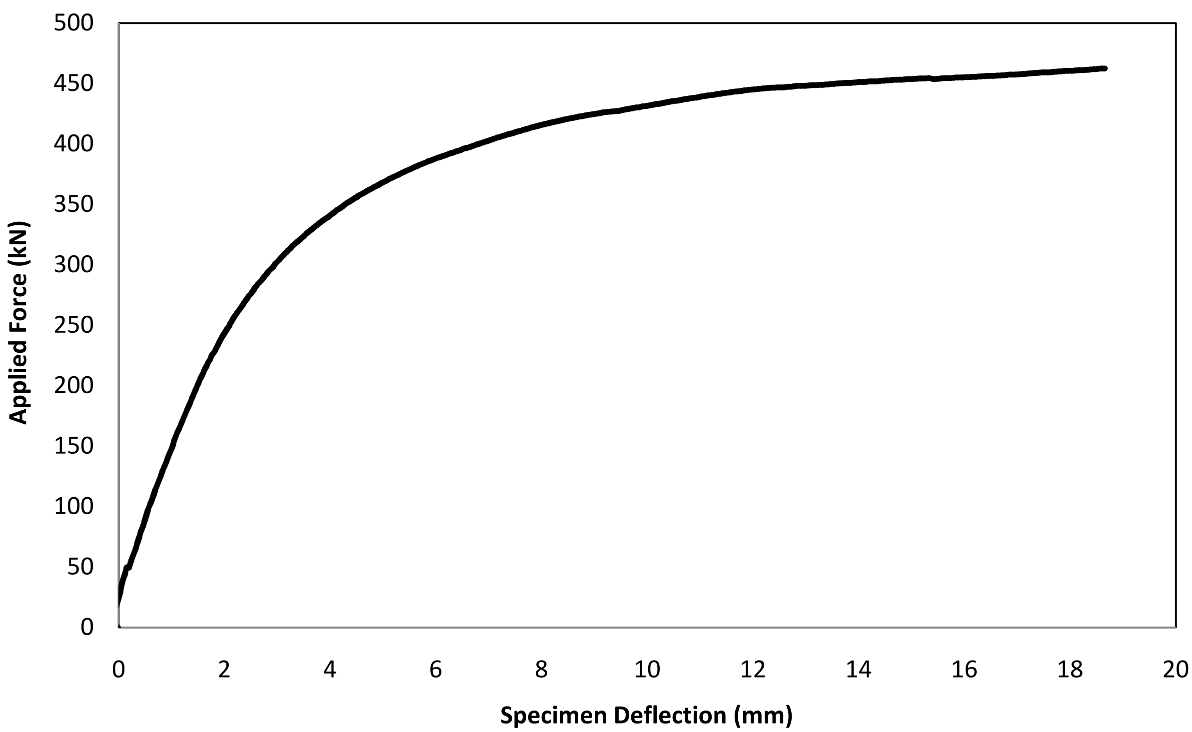


Applied Force (kN)
350
300
250
200
150
100
50
0
0
2
4
6
8
10
12
14
16
Specimen Deflection (mm)



Applied Force (kN)
500
450
400
350
300
250
200
150
100
50
0
0
5
10
15
20
25
30
35
40
Specimen Deflection (mm)


350
300
250
200
150
100
50
0
0
1
2
3
4
5
6
7
8
9
10
Applied Force (kN)
Specimen Deflection (mm)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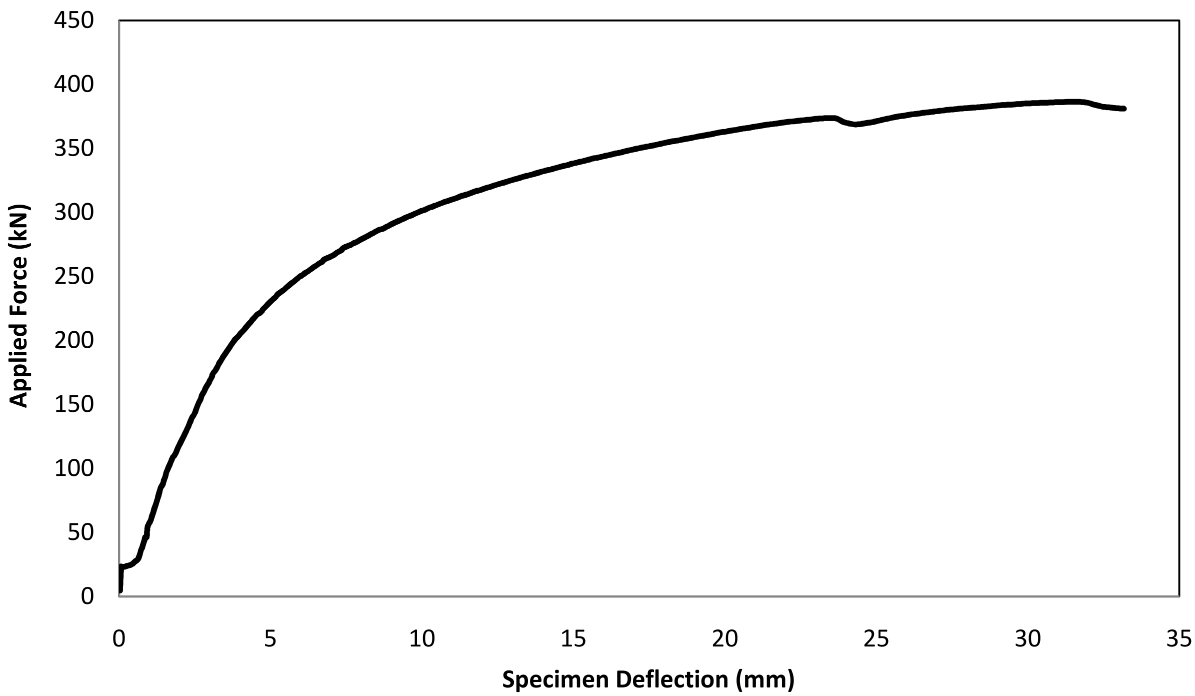
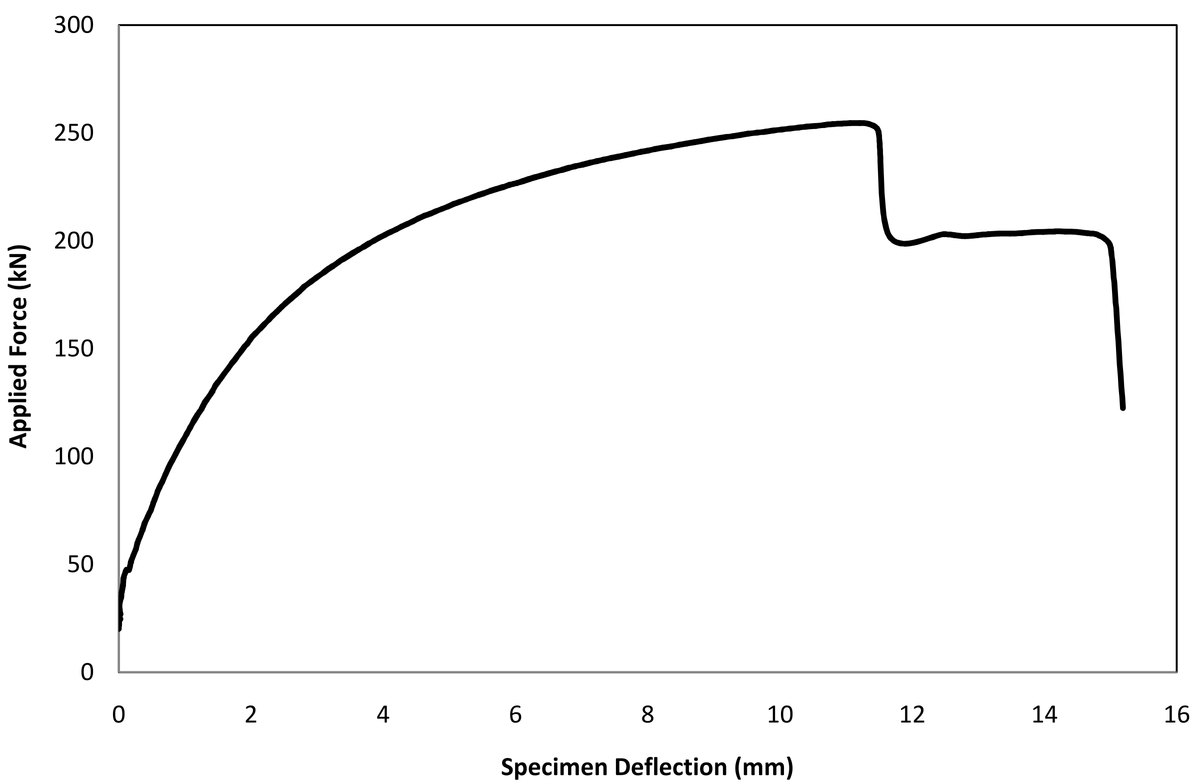
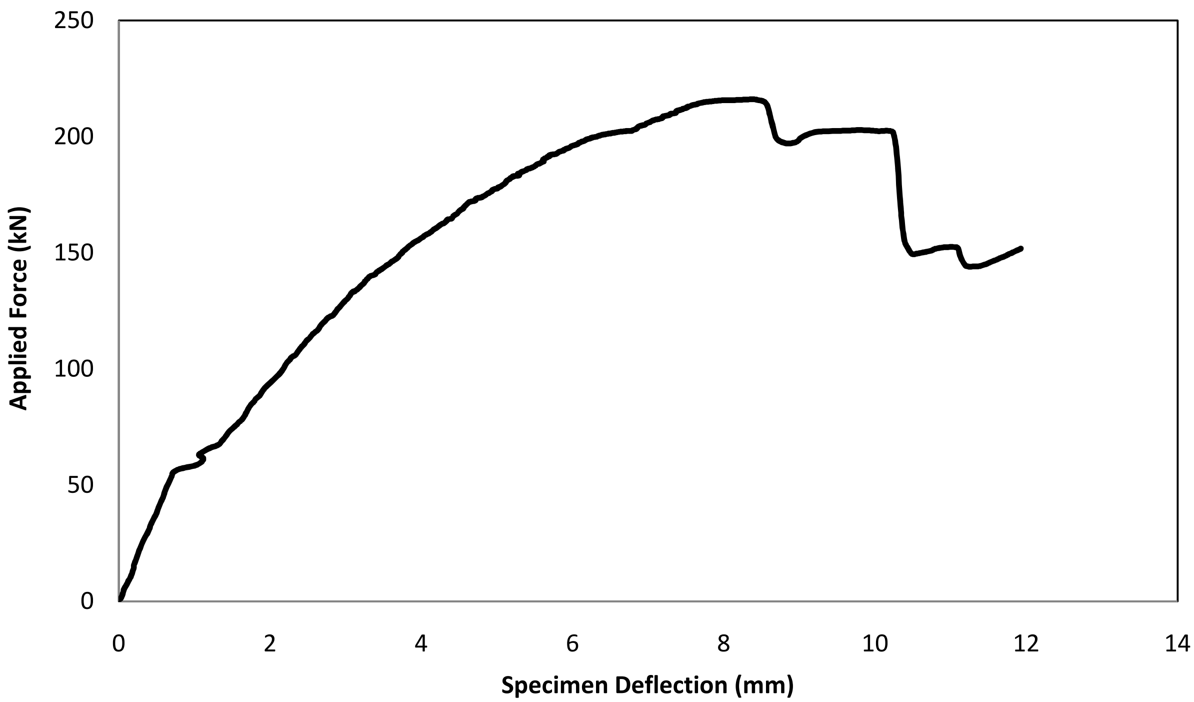
450
400
350
300
250
200
150
100
50
0
0
2
4
6
8
10
12
14
16
18
20
Applied Force (kN)
Specimen Deflection (mm)

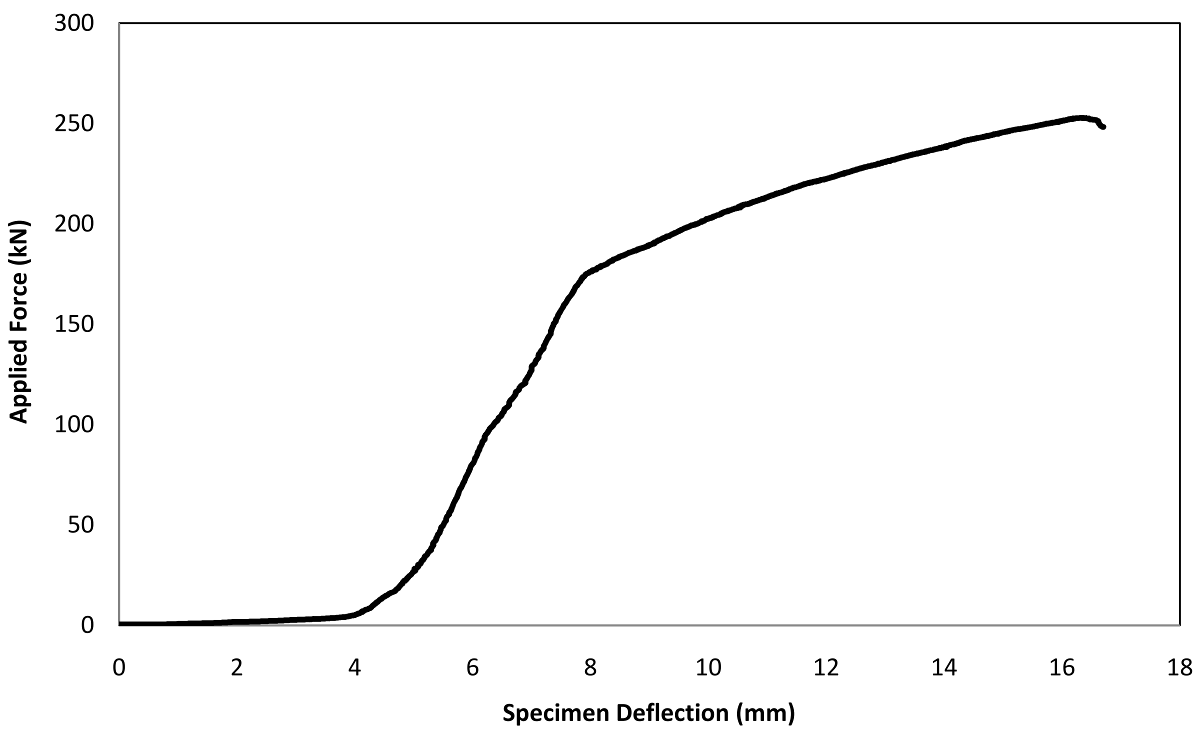


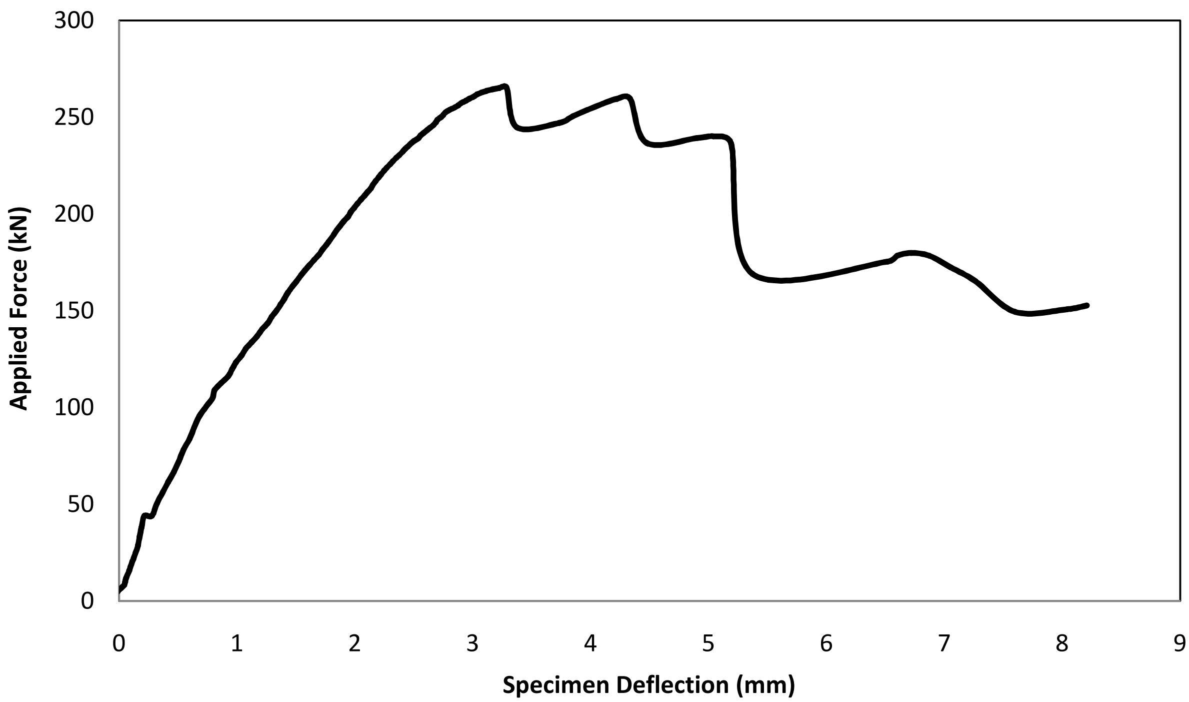
450
400
350
300
250
200
150
100
50
0
Applied Force (kN)
0
5
10
15
20
25
30
35
Specimen Deflection (mm)



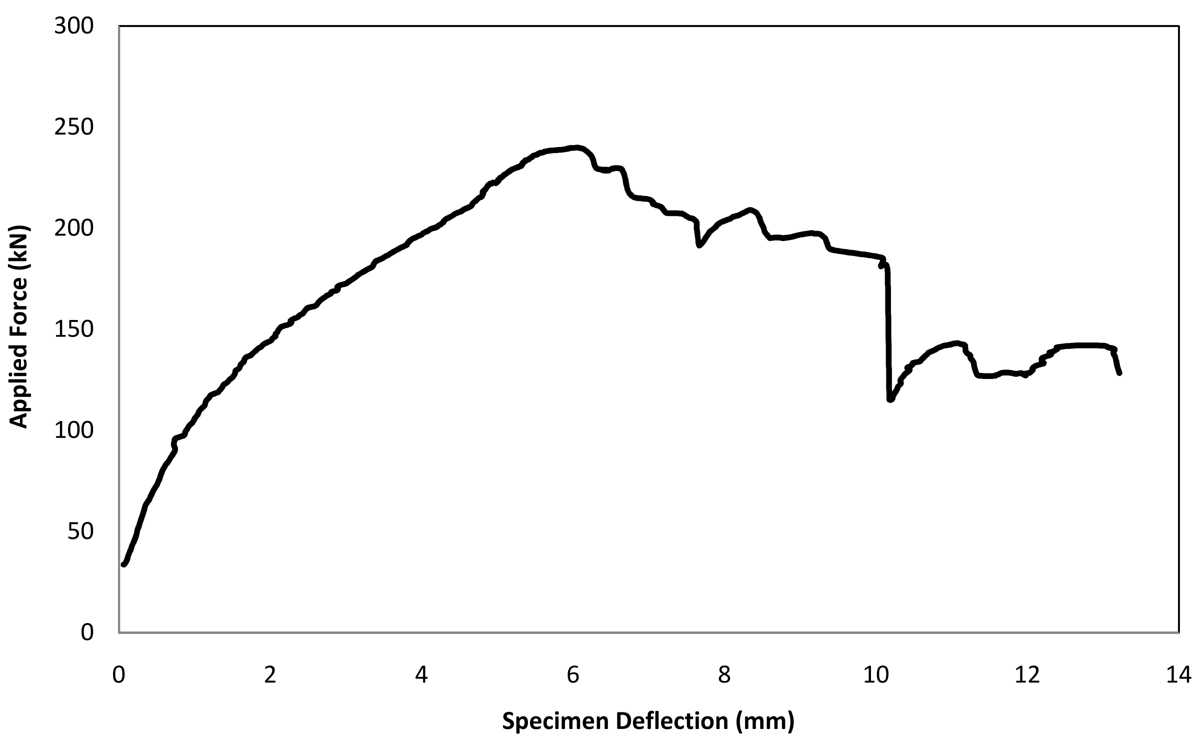
300
250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
14
16
Specimen Deflection (mm)


250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
14
Specimen Deflection (mm)

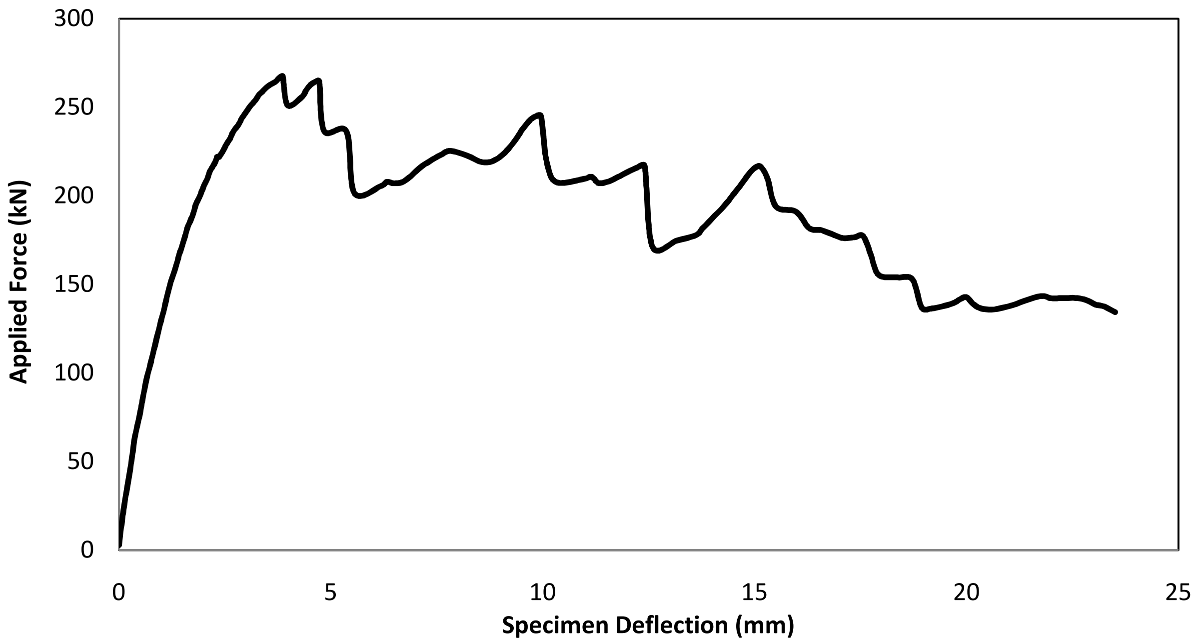

300
250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
14
16
18
Specimen Deflection (mm)



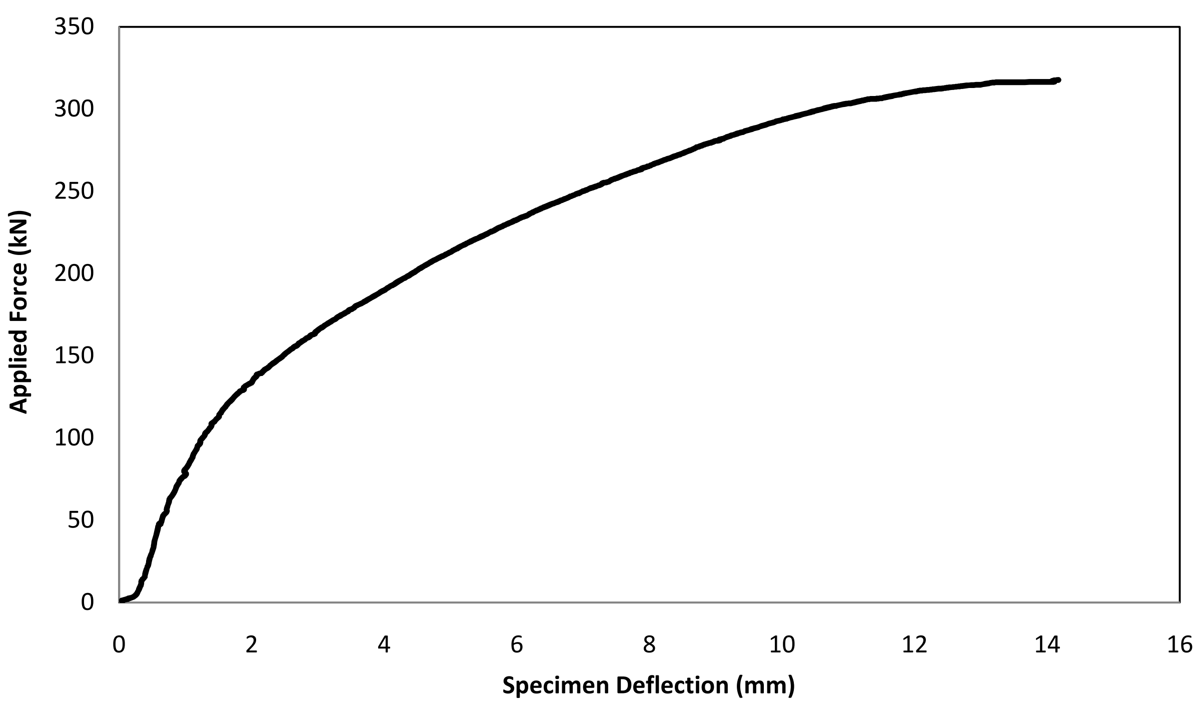
Applied Force (kN)
300
250
200
150
100
50
0
0
1
2
3
4
5
6
7
8
9
Specimen Deflection (mm)



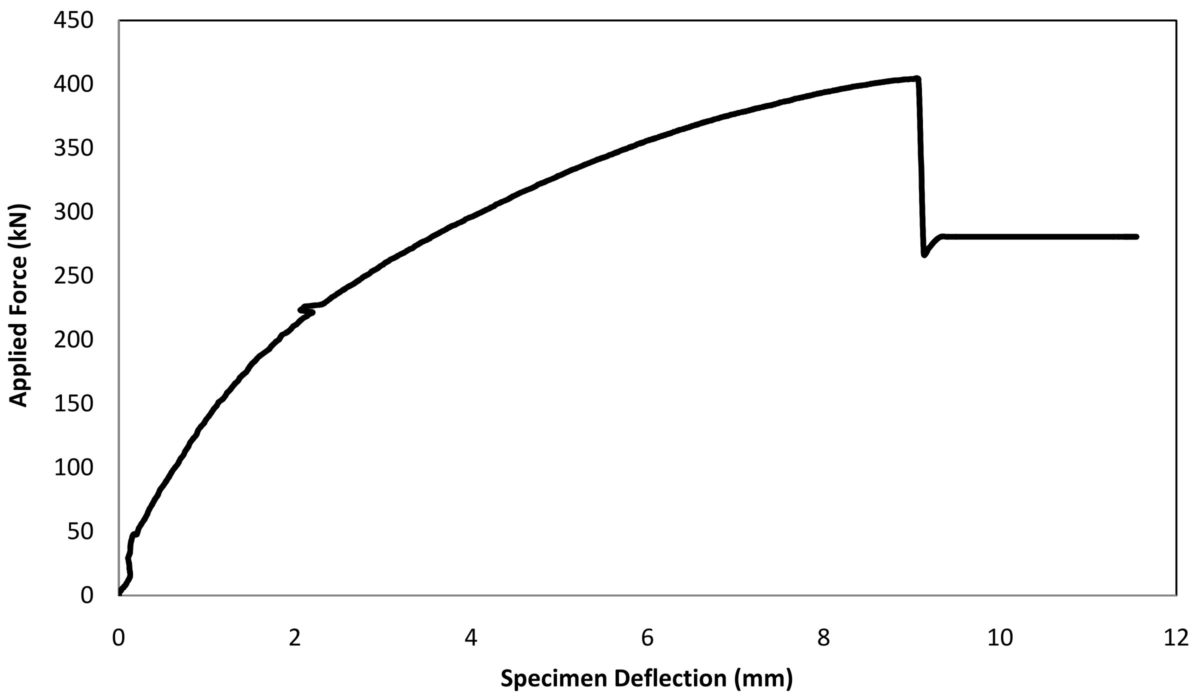
Applied Force (kN)
300
250
200
150
100
50
0
0
2
4
6
8
10
12
14
Specimen Deflection (mm)



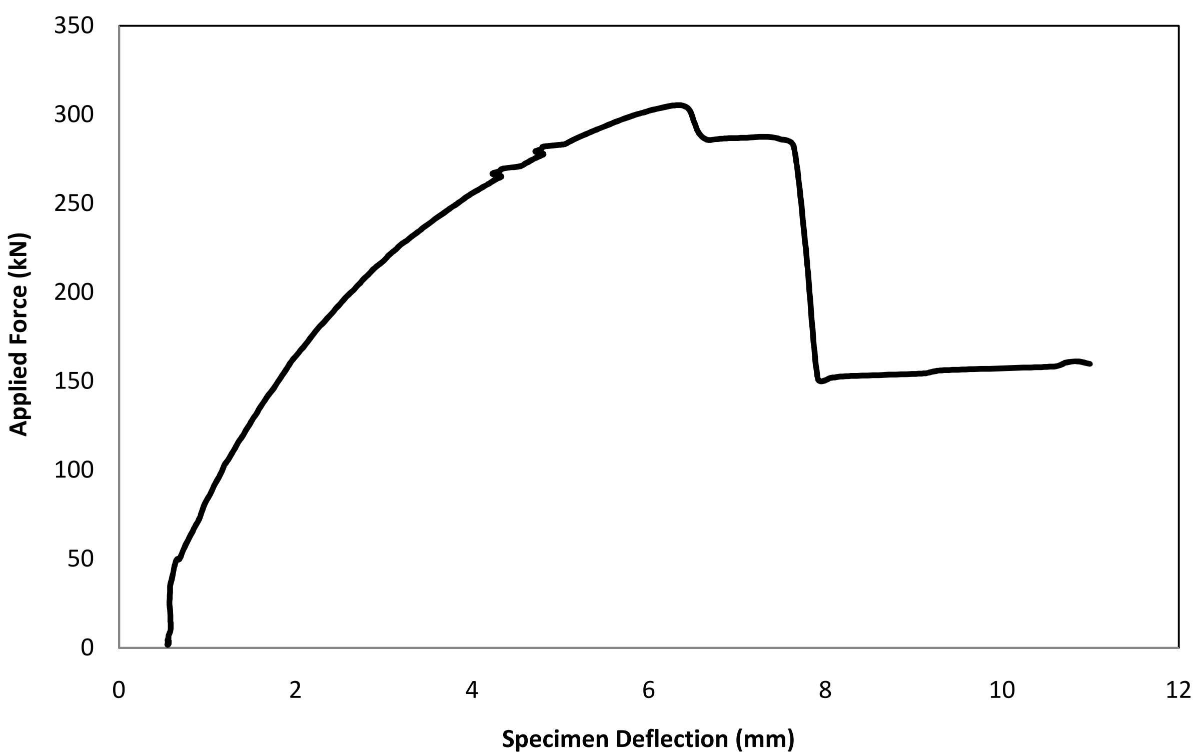
Applied Force (kN)
Specimen Deflection (mm)
0
50
100
150
200
250
300
0
5
10
15
20
25



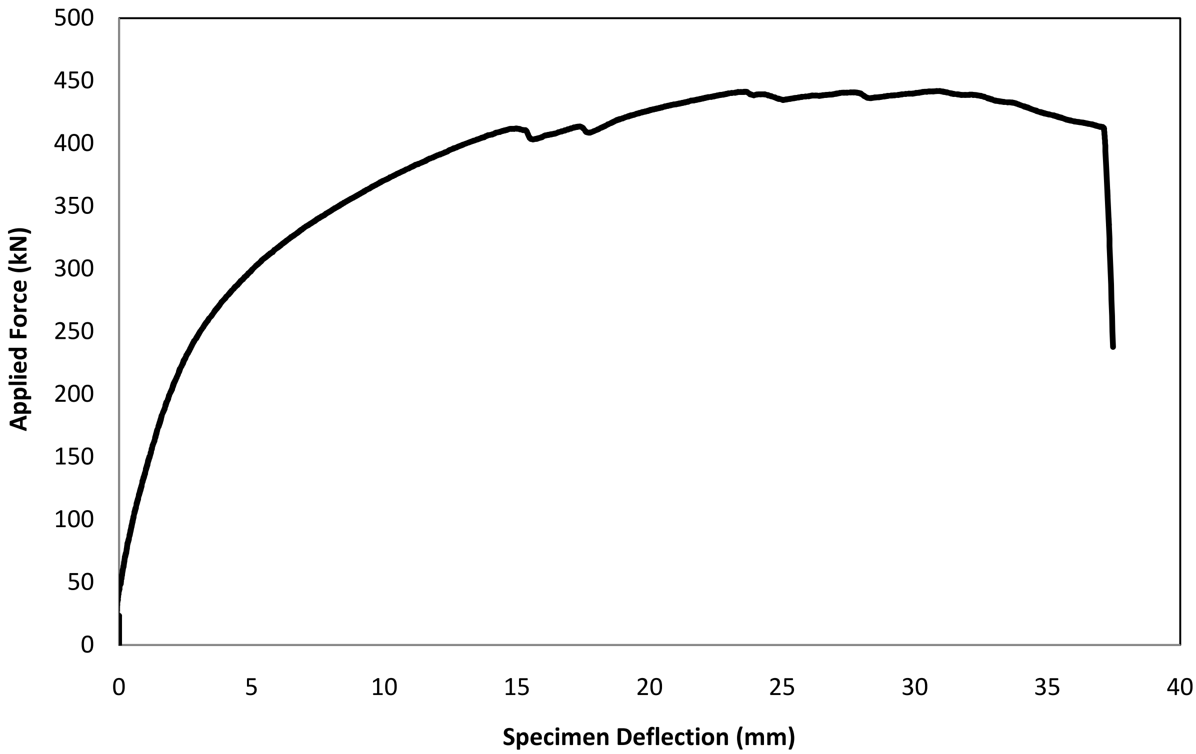
Applied Force (kN)
Specimen Deflection (mm)
0
50
100
150
200
250
300
350
0
2
4
6
8
10
12
14
16



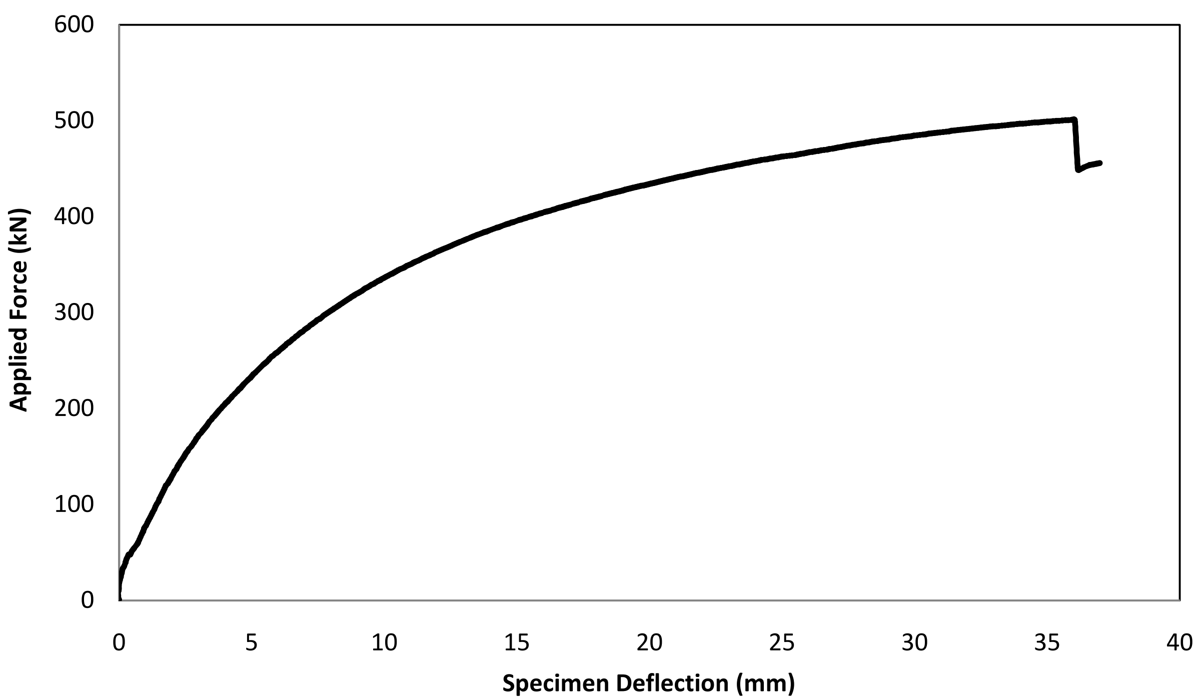
450
400
350
300
250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
Specimen Deflection (mm)



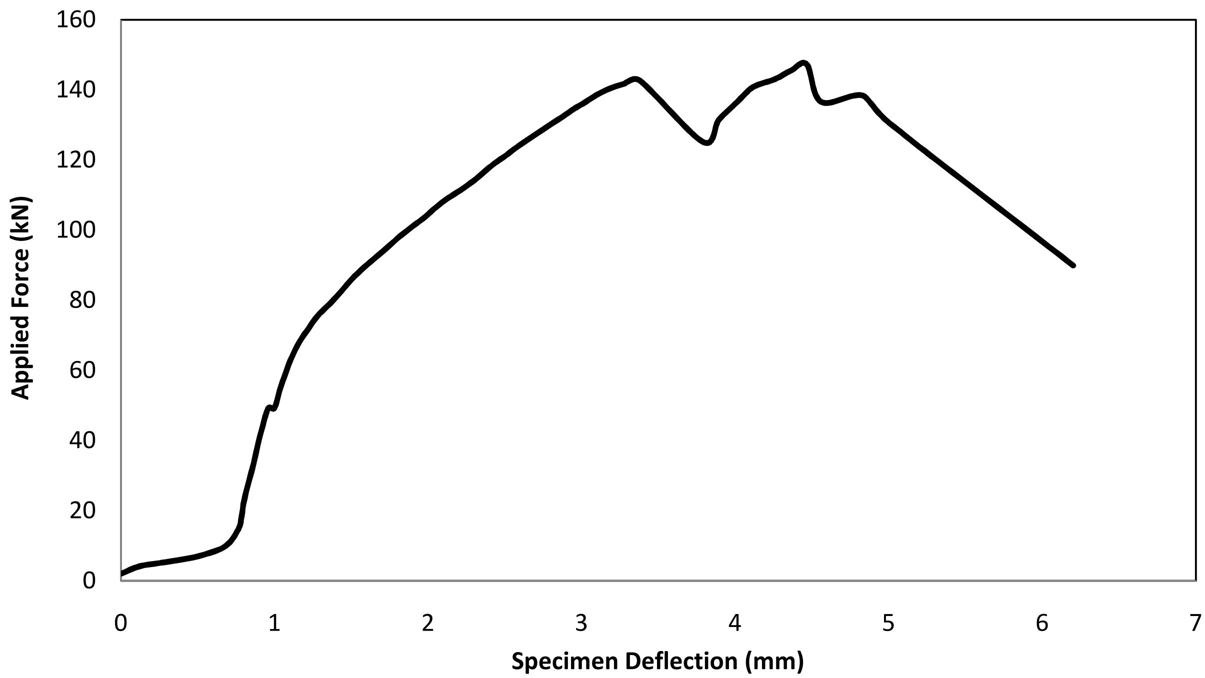
350
300
250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
Specimen Deflection (mm)



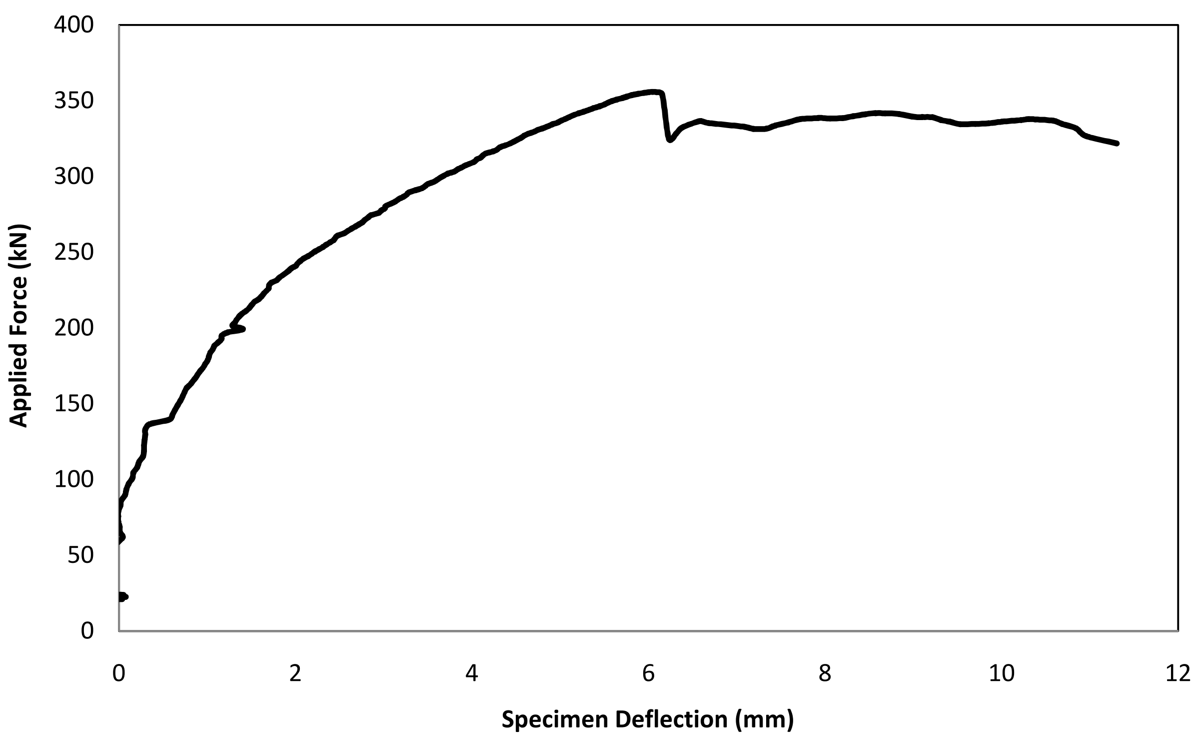
500
450
400
350
300
250
200
150
100
50
0
Applied Force (kN)
0
5
10
15
20
25
30
35
40
Specimen Deflection (mm)



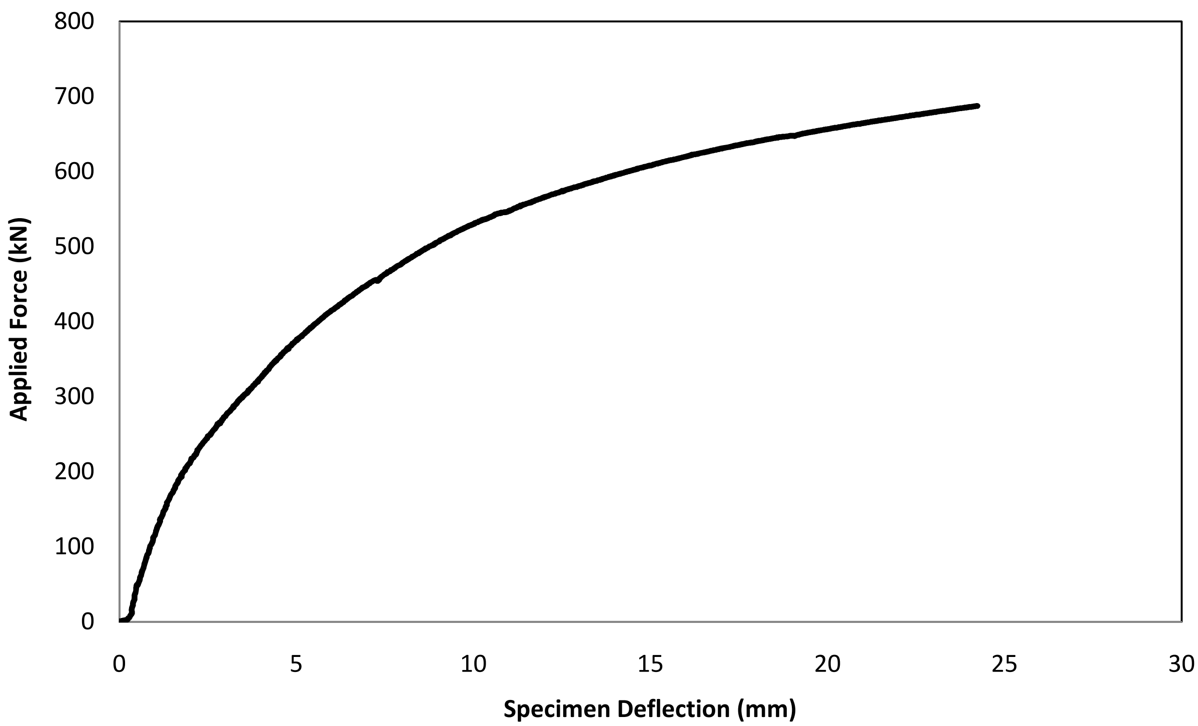
600
500
400
300
200
100
0
Applied Force (kN)
0
5
10
15
20
25
30
35
40
Specimen Deflection (mm)



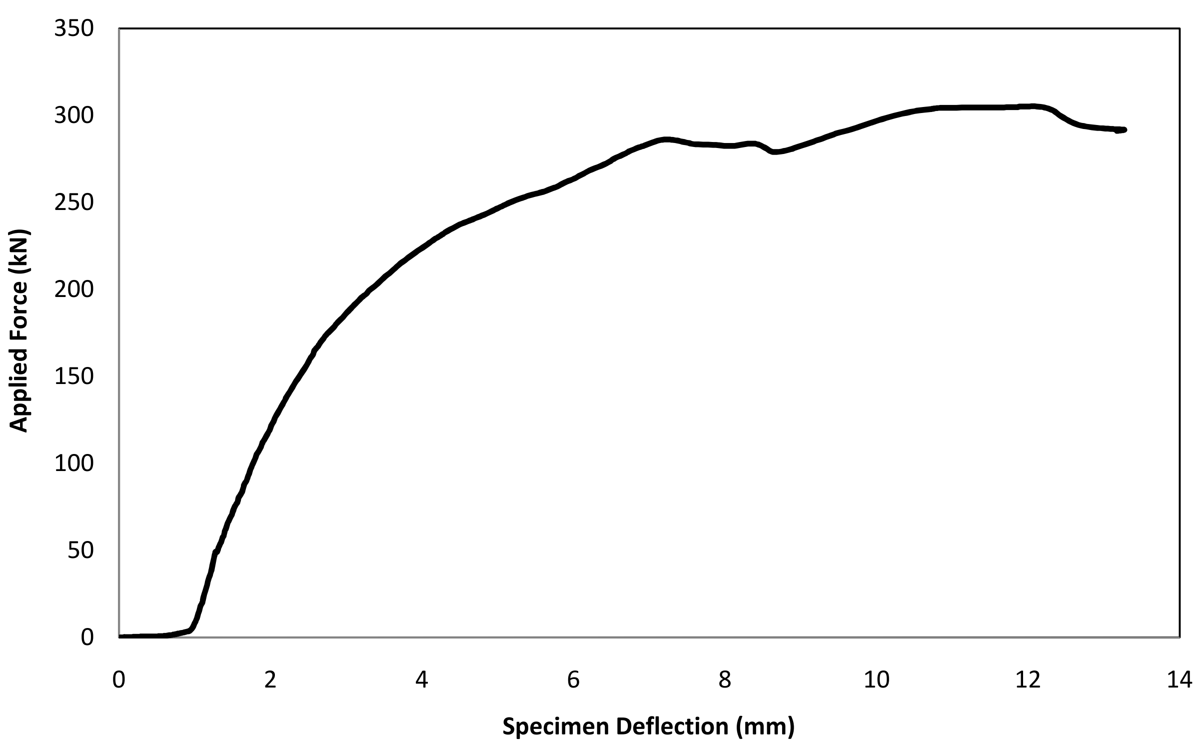
160
140
120
100
80
60
40
20
0
0
1
2
3
4
5
6
7
Applied Force (kN)
Specimen Deflection (mm)



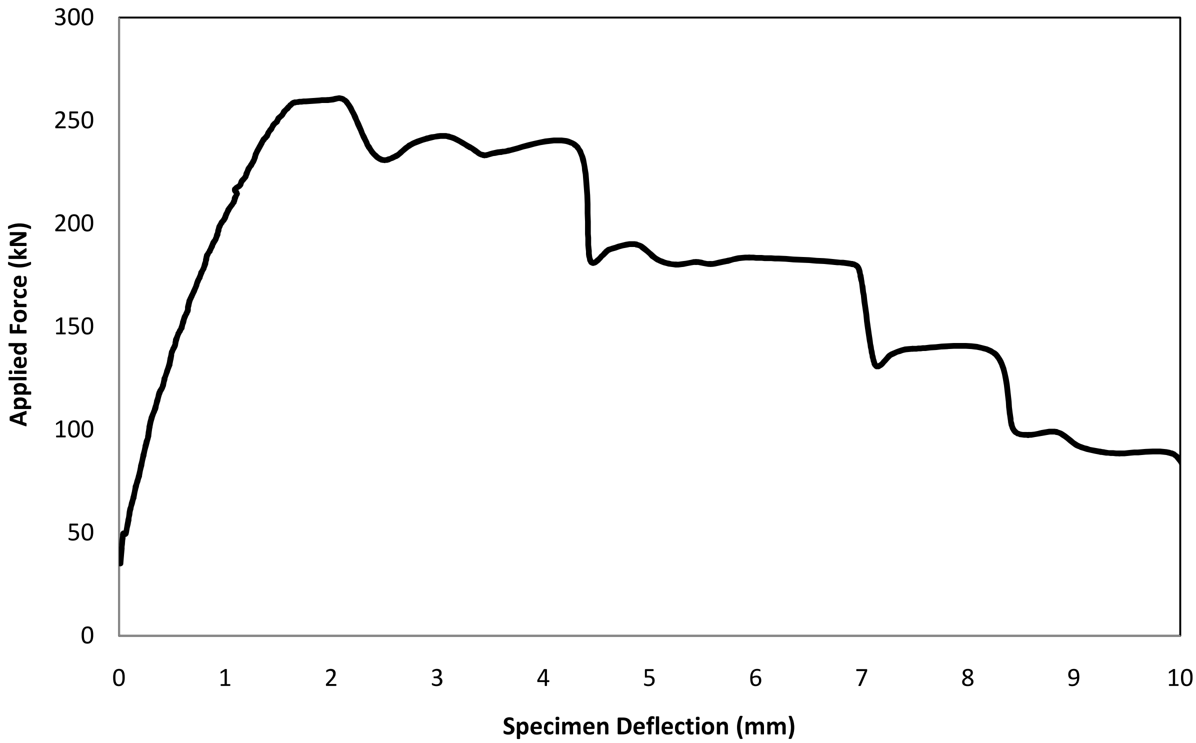
400
350
300
250
200
150
100
50
0
0
2
4
6
8
10
12
Applied Force (kN)
Specimen Deflection (mm)

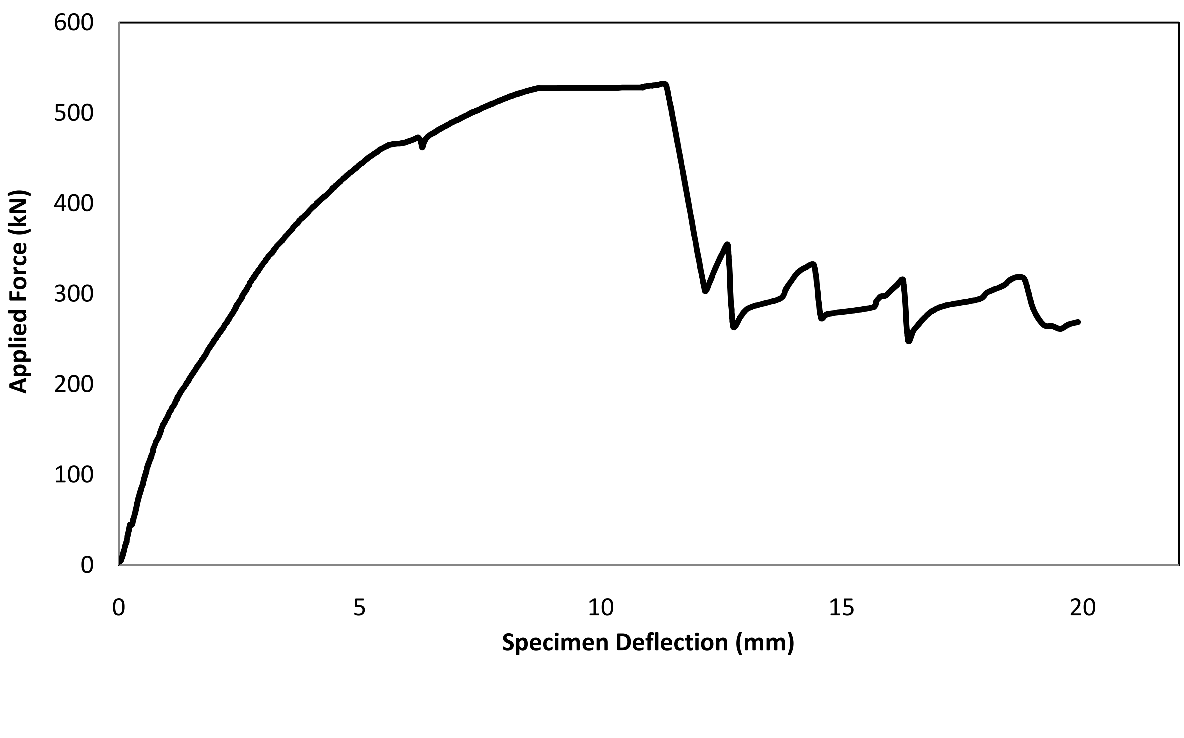


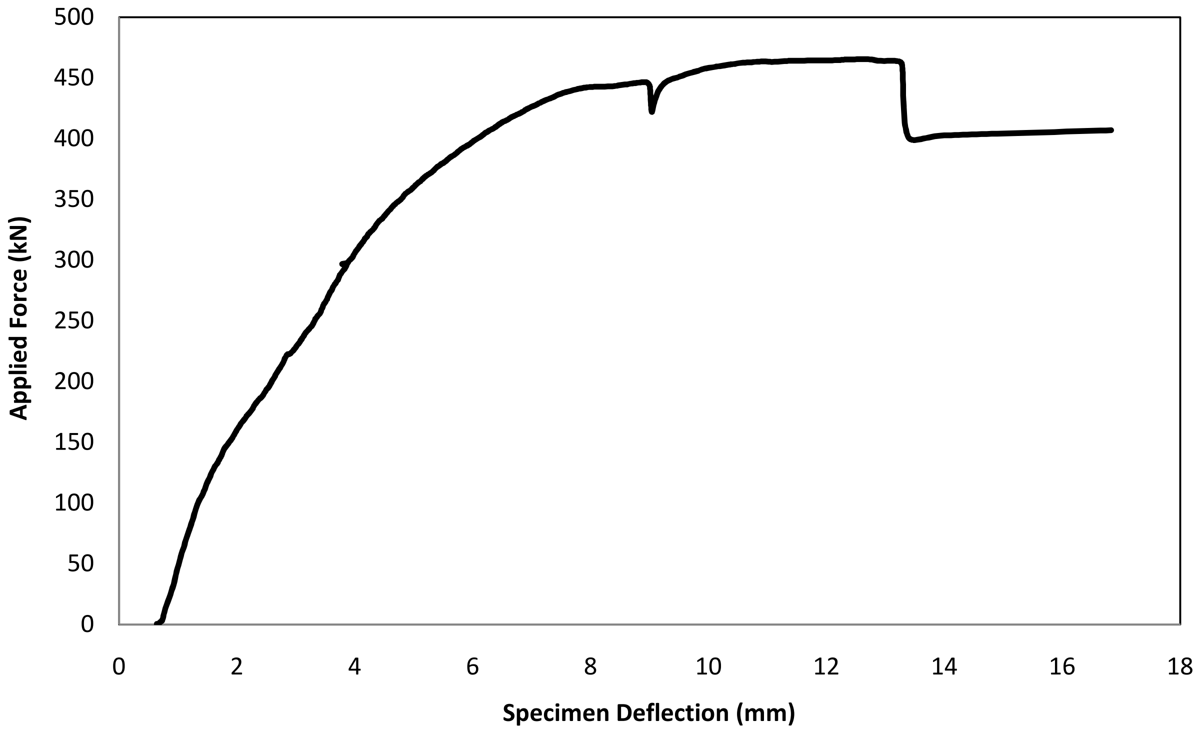
800
700
600
500
400
300
200
100
0
Applied Force (kN)
0
5
10
15
20
25
30
Specimen Deflection (mm)



350
300
250
200
150
100
50
0
Applied Force (kN)
0
2
4
6
8
10
12
14
Specimen Deflection (mm)


300
250
200
150
100
50
0
0
1
2
3
4
5
6
7
8
9
10
Applied Force (kN)
Specimen Deflection (mm)


600
500
400
300
200
100
0
0
5
10
15
20
Applied Force (kN)
Specimen Deflection (mm)



Applied Force (kN)
500
450
400
350
300
250
200
150
100
50
0
0
2
4
6
8
10
12
14
16
18
Specimen Deflection (mm)

APPENDIX D

VIDEO EXTENSOMETER

D1: Camera System

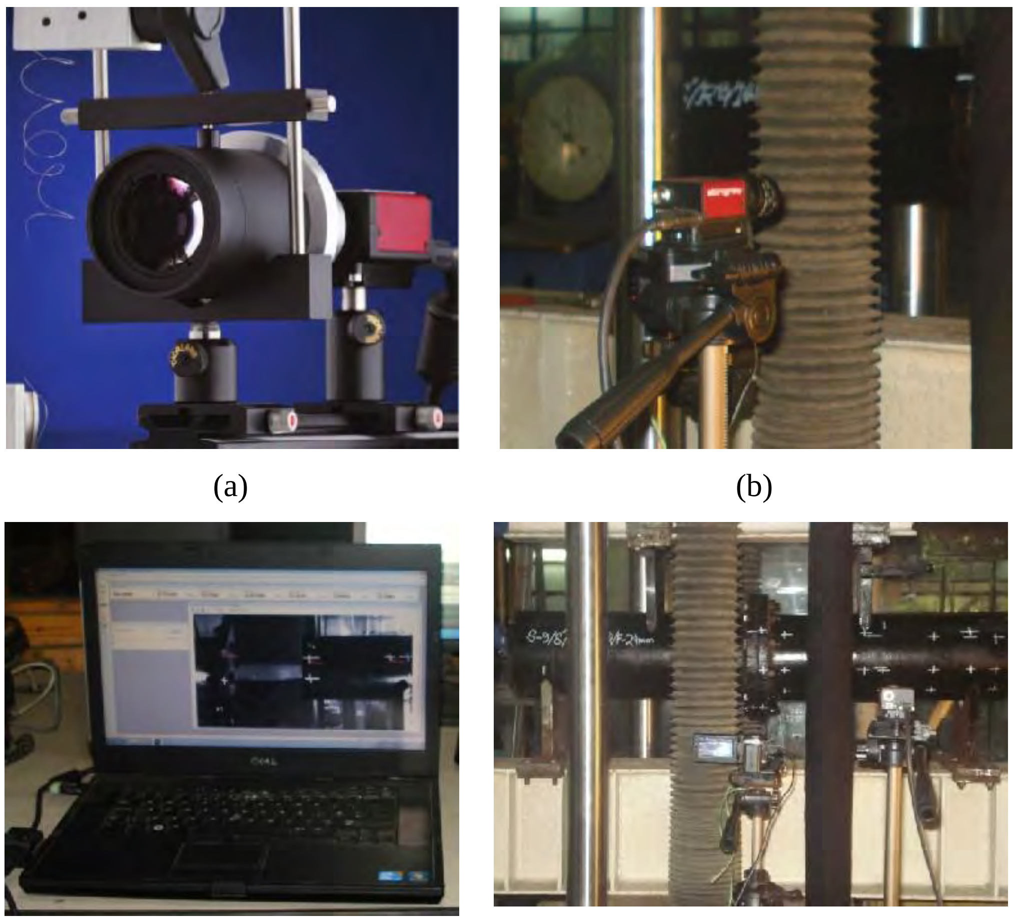
All the measurements and outputs from the Video Extensometer are time stamped and the uncompressed video output from the camera can be recorded for post-test. Tinius Olsen's Video Extensometer uses a high resolution monochrome camera and advanced high speed image processing such that its point- to-point real-time video processing technology is capable of achieving, and exceeding, ASTM E83 Class B1and ISO 9513 Class 0.5 accuracy for both low and/or high strain materials. The Video Extensometer can be further enhanced by using multiple camera systems to measure specimen strain. The system works by acquiring an image of the specimen and using Tinius's pattern recognition technology to lock onto a minimum of two targets, which can equate to a number of different gauge lengths.

D2: Lighting and marking of targets

The extensometer is more than capable of following chosen targets in regular daylight conditions; however using additional lighting prevents any tracking loss of target as a result of changes to ambient lighting conditions. Any visible marking can be used for pattern recognition, and these can be natural patterning on the specimen surface, pen marks, blob markers, punched gauge marks or a spray paint speckle pattern. These targets can be defined by the user, allowing setting these to any gauge length as required. As the specimen is tested, the software tracks the point-to-point movement of these targets from camera frame to frame, and strain data is calculated in real time. Since multiple gauge lengths are possible in both longitudinal and transverse directions, the determination of r and N values is simple and straightforward. The high system resolution required to calculate these results is achieved using subpixel interpolation algorithms and with which the system can resolve to submicron levels of movement.

D3: Software

Tinius Olsen has develop Horizon, a comprehensive software program that makes testing simple, precise, and efficient. Our Video Extensometer software monitors the object under test. By fitting an appropriate lens to the high resolution cameras, the Video Extensometer will measure objects smaller than 1 mm to larger than 100 m. Tinius Olsen's Video Extensometer software uses patented technology to precisely measure 2D position of targets in images from the video camera; special targets are NOT required. The system can precisely measure each target's position in every image from the video camera. Up to 100 targets can be measured in real-time at 15 Hz. From the measured positions of the targets, the system can calculate displacement, velocity, acceleration, angular rotation, 2D Strain. Setup of video extensometer is shown in figure D1.



(a) (b)

(c) (d)

Figure D1: (a) Video camera with high resolution materials testing lens, (b) Video camera with a general purpose lens, (c) Software outlook, (d) Additional gen purpose video camera used to measure bolt strain.

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