DESIGN AND DEVELOPMENT OF A TENSILE SPLIT HOPKINSON GAS GUN

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Declaration

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List of Nomenclature

A_B	Cross-sectional area of the bar
A_{Barrel}	Cross-sectional area of barrel bore
A_S	Cross-sectional area of the specimen
A_{st}	Cross-sectional area of the striker
A_t	Area of the threads experiencing a shearing force
C	Factor depending on method of attachment for ASME pressure vessel
C_B	Speed of sound in the bar material
C_{st}	Speed of sound in the striker material
C_t	Circumference of the minor diameter of the thread
D	Distance between optical sensors
d_o	Outer diameter
d_i	Inner diameter
D_P	Diameter of projectile
D_{major}	Major diameter of the thread
D_{minor}	Minor diameter of the thread
E_B	Young's modulus of the bar
F	Force
F_{air}	Force of the pressurized air
F_b	Force exerted by the bolt
F_i	Force on incident bar
F_{st}	Force on striker
FS	Safety Factor
H	Height between the theoretical peaks of the internal and external threads

J	Joint efficiency
L	Length
L_{bar}	Length of gun barrel
L_{st}	Length of the striker
P	Pressure of the air pressure tank
P_{air}	Pressure of the air pressure tank
R	Inner radius of cylinder
r_c	Radius of crown of the pressure vessel head
S	Allowable stress from ASME boiler and pressure vessel code\
$\sigma_{Yield\ bar}$	Bar material yield strength
d	Bolt circle diameter
m	Mass
t	Time
T	Thickness
t_c	Thickness of thin-wall cylinder of gas gun
t_b	Thickness of barrel cylinder
t_{ec}	Thickness of the end cap of gas gun
v_i	Velocity of the incident bar
v_{st}	Velocity of the striker
X	Distance through the thread at the effective pitch diameter
ε	Strain
ε_{avg}	Average engineering strain
$\dot{\varepsilon}$	Average engineering strain rate
ε_R	Reflected strain pulse

ρ_B	Density of the bar material
ρ_{st}	Density of the striker
ρ_P	Density of the projectile
σ	Normal stress
σ $_{avg}$	Average engineering stress
σ $_i$	Stress in incident bar
σ $_s$	Yield strength of the specimen
σ $_{st}$	Stress in the striker
σ $_{true}$	$true$ True stress
σ $_t$	$thread$ Shear stress in the thread

Abstract

This is a project of Design and Development of the Split Hopkinson Tensile Bar Gas Gun apparatus to launch a striker up to designed velocity to perform tensile strain test to specimen at strain rate of 100/s to 1000/s. The gas gun consists of a barrel, striker, incident bar, pressure tank, valve components and pressure regulator. The objective of this project is to demonstrate the functionality of the gas gun by performing tests with different pressure. The performance of this Gas gun was tested by launching the striker 5 times with the same air pressure stored in the air pressure tank. Then, the procedure is repeated using higher pressure in the pressure tank and data are recorded. The result is that the gas gun is capable to launch the striker up to 13 m/s with 7 bar of air pressure by theoretically producing strain rate of 866.5/s to specimen made from DP6600.

Abstrak

Ini adalah projek berkenaan tentang Reka bentuk kejuruteraan penembak gas Tensile Split Hopkin Pressure Bar. Penembak gas ini di reka bentuk untuk penempak peluru keatas spesimen bagi bertujuan untuk mendapatkan kadar tegangan daripada 100/s hingga ke 1000/s. Penembak gas ini terbentuk daripada beberapa komponen iaitu laras, peluru, incident bar, tangki tekanan, komponen injap dan juga penyelaras tekanan. Projek ini adalah untuk menunjukkan fungsi penembak gas dengan menggunakan tekanan angin yang berbeza.Kebolehan penembak gas itu telah diuji sebanyak 5 kali dengan tekanan angin yang sama. Selepas itu, prosedur itu diulang semua dengan tekanan angin yang lebih tinggi. Keputusan yang telah didapati daripada ujikaji yang telah dijalankan mendapati bahawa peluru itu mampu mencapai kelajuan sehingga 13 m/s dengan tekanan angina yang dikenakan sebanyak 7 bar. Secara tidak langsung, penembak gas itu mampu melakukan kadar tekanan sebanyak 866.5/s terhadap specimen yang diperbuat daripada besi plat DP6600.

1) Chapter 1

Introduction

Split Hopkinson Pressure Bar is an apparatus for testing the dynamic stress-strain response of a material. This technique which is named after the Hopkinson Family due to their pioneering and research to high strain rate testing. The technique is classified into dynamic high-strain rates and can be conducted in several difference ways: compression, tension, torsion, shear, and triaxial.[1]

Bertram Hopkinson [2] suggested a way to measure stress pulse propagation in metal bar. Hopkinson used long steel rod, a short steel billet, a ballistic pendulum and device to generate impulse pressure. The basic principle of this technique is that by impacting one end of the rod by the short steel billet with detonation of explosives which then creates a compressive pressure wave inside the rod. As the compressive wave travel down the bar it would reflected at far end as a tensile wave.

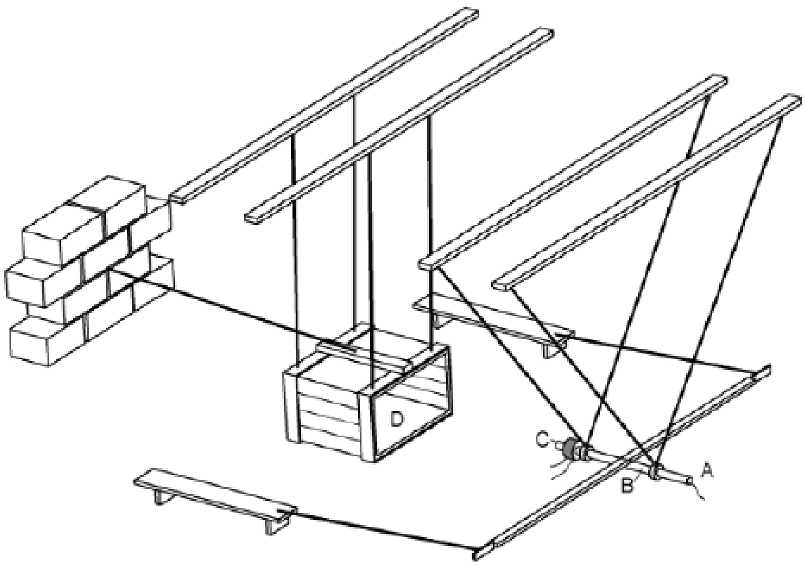

Figure 1-1: Bertram Hopkinson's experiment to measure pressure waves [1]

Later in 1949, Kolsky[3] extended the development of the Hopkinson Pressure Bar technique by using two elastic bars instead of using one. The specimen of the experiment was place between both bars where one of the bar becomes the transmitter bar and the other is the incident bar. The basic principle of this technique is that the striker (bullet) is propelled at specified velocity, striking the one end of the incident bar and caused compression on the specimen sandwiched between the two bars and then using condensers to measure the strains existing in the pressure bar. Technique has since been known as the split Hopkinson pressure bar (SHPB) or Kolsky bar that become the most widely used testing procedure today.

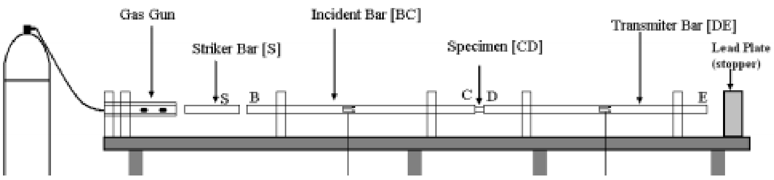

Figure 1-2: Schematic of split-Hopkinson pressure bar.

From the previous technique, it has been initially used for compression only. Then, Harding J., Wood, E. D., Campbell, and J.D [4] further the development of the split Hopkinson bar to be used for tensile test. The compression pulse in a tube is transferred to an inner rod via a mechanical joint. As compression waves are reflected as tension waves on free surface, the specimen is loaded in tension.

The tensile Split Hopkinson Pressure bar technique is used to determine the dynamic tensile stress-strain curve of materials. The difference between tensile test and the compressive test are the technique to generate pulse. There are three different technique that have been developed throughout the years.

The first design is created by Hauser [5] in 1966. A direct acting tensile SHPB uses a pair of concentric cylinders, with the inner cylinder being made up of two solid bars called the incident and transmission bar and with a specimen threaded between the incident and transmission bars. A hollow outer cylinder is attached to the incident bar via a transfer connection.

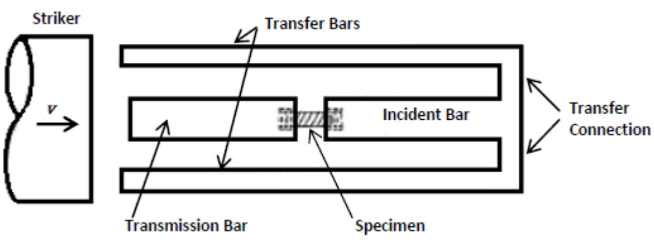

Figure 1-3: Direct tension loading by Hauser [5]

The second technique is designed by Lindholm and Yeakley [6] in 1968 where it used the top hat specimen and uses a hollow transmission bar of the same cross-sectional area as the incident bar.

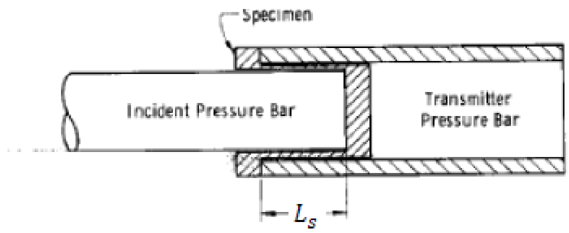

Figure 1-4: Technique by Lindholm and Yeakley [6]

The third technique is designed by Nicholas [7] in 1981 where a dumbbell-shaped specimen with threads on each end-side was used. A collar is applied to protect the specimen from the compressive pulse, thus it will propagate to the output bar, while the specimen will only receive pulse being reflected from the free-end of the output bar as a tension pulse.

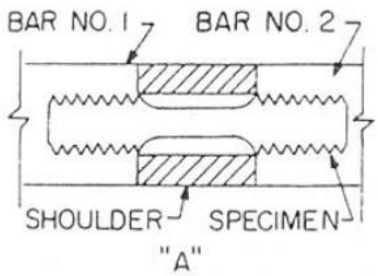

Figure 1-5: Technique by Nicholas [7]

1.1 Problem Statement of Project

Although there are many design of a Split Hopkinson Pressure Bar that have already been published, most of them are compression type of a Split Hopkinson Pressure Bar and very limited design for the tensile type of a Split Hopkinson Pressure Bar that have been created or published. Furthermore, the tensile Split Hopkinson Pressure Bar is still has not been standardized yet. The Split Hopkinson Pressure Bar technique used an apparatus with the dimensions and the materials chosen by its researcher without any specific explanation. Consequently, the results of the tests are slightly different one from another.

1.2 Objectives of Project

The main objective of this project is to develop a gas gun of the Split-Hopkinson Pressure bar apparatus in order to be able to perform the tension of a high strain rate testing of a material. The development work includes the following task

1. To design the mechanical component of the gas gun
2. To fabricate the mechanical component of the gas gun
3. To demonstrate the functionality of the gas gun by performing test with different pressure
4. To compare the result of the striker velocity to establish results

1.3 Scope of Work

For the scope of work, all the component of the gas gun need to be designed and drawn into a 3D parts in the Solid work. From the design, the pressure chamber, end caps, barrel, valve support and the valve will be fabricated using machine found in workshop of the school and collars of the valve will be machined using the CNC machine. The fabricated components are then need to be assemble to form the gas gun. The performance of the gas gun will be evaluated based on the velocity of the striker relative to the pressure applied to the pressure chamber.

2) Chapter 2

Literature Review

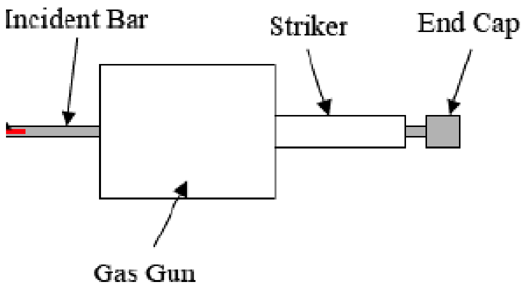

Figure 2-1: Layout of a gas gun of a tensile SHPB apparatus

This project is focused on the design of the gas gun of the tensile Split Hopkinson Pressure Bar. The gas gun is one of the major component that play a big role for the working principle of the Split Hopkinson Pressure Bar. It is a device which launch the striker to strike the anvil in order to create high strain rate tensile stress. The different between this designs of the gas gun from Hauser, Lindholm and Nicholas technique is that it need to launch a hollow striker. Therefore, the design must be a direct method where the striker (which is hollow) will fit perfectly around the incident bar and slide along it at high velocity. Figure 2-2 shows that how the striker slide on to the incident bar where the gas gun is located between the striker and the specimen. Furthermore, there is a switch where it will open the valve after is reaches a certain pressure to release the pressurize air to launch the striker.

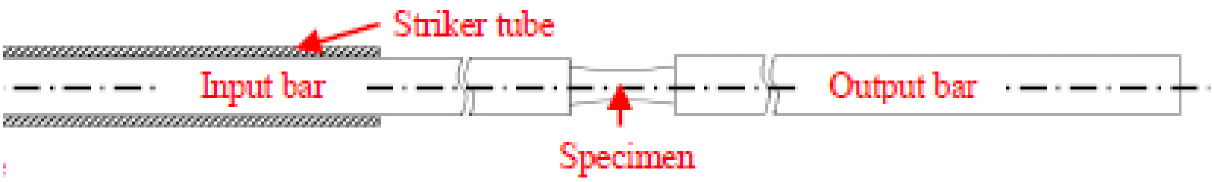

Figure 2-2: Schematic of tensile type split-Hopkinson pressure bar. [1]

There are three main components of the design of the gas gun which are the barrel, pressure chamber and valve components. The principle of the mechanism of the gas gun is that the striker is positioned in the barrel. Then, the pressure chamber will be pressurize to the required pressure. After the pressure reaches the required pressure, a switch then is turn on to release to pressurized air. As the gas enters the gun barrel and the gas volume expands, the striker is accelerated along the incident bar up to the required velocity.

2.1.1 Barrel

A gun barrel is a crucial part of this project where it is the straight shooting tube, usually made of rigid high-strength metal, through which a rapid expansion of high-pressure gases is released (via propellant combustion or via mechanical compression) behind a projectile in order to propel it out of the front end (muzzle) at a high velocity. The hollow interior of the barrel is called the bore. The barrel length will be selected, so the striker would not exceed the velocity of the striker that would induce plastic deformation to the material used.

Another component the barrel is the end cap. The end cap involve in two function which is to block the one end of the barrel so that striker can be pushed and as a support for the incident bar to slide on.

2.1.2 Air pressure tank

Air Pressure tank is the place where it hold the required high amount of pressurized air before it will be released into the barrel. The wall of the air pressure tank must thick enough so that it can capable to hold high pressure without fail or defect. It also need to be made from high tensile strength in order to withstand the pressure. The pressure tank then assembly with valve component for the pressurized to be release to the barrel

2.1.3 Valve Assembly

The assembly consist solenoid valve, ball valves, pressure gauge, air fitting coupler, pneumatic PU-tube and air hose. The function of the valve is to release the pressurized air in the pressure tank into the barrel. After the pressure tank reaches the required pressure, the solenoid valve is mechanical actuated using a switch. The actuators will move the valve uncovering the openings in the solenoid valve.

2.2 Types of Tensile Split Hopkinson Pressure Bar

The main functionality of a Gas Gun is one thing which is to shoot a projectile. However, different gas gun have different mechanism and components.

One of the recent design of a gas gun is developed by Young [1]. Young's design composed of three main components: a barrel, a pressure chamber and a valve assembly. Using mechanically actuated valves the pressure chamber is filled to the pressure required to achieve the desired striker velocity. Once the gas gun pressure chamber reaches the required pressure, the pneumatic actuators move the valve toward the specimen uncovering two openings in the gas gun barrel. The valve blocks one end of the gas gun barrel, leaving the opposite side as the path of least resistance for the gas to escape. As the gas enters the gun barrel and the gas volume expands, the striker is accelerated along the incident bar up to the required velocity.

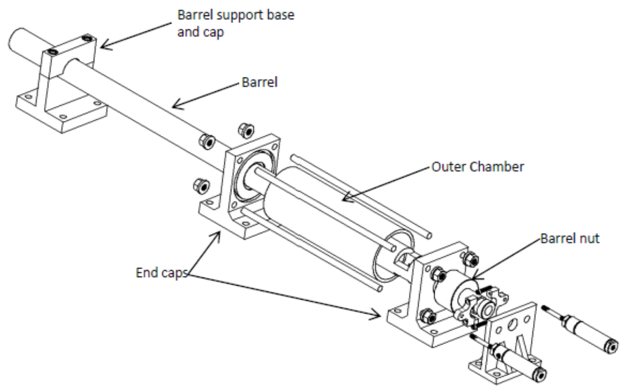

Figure 2-3: The 3D model of the Young's Gas Gun design [1]

For the Young's design [1], the barrel is fabricated from AISI 304 stainless steel bar which turned into a hollow barrel by gun drilling along the central axis of the bar to an inside diameter of 27mm, then using a boring bar the valve area was enlarged to the final diameter of 38mm. Two openings were then machined to allow the pressurized gas to transfer from the pressure chamber into the gun barrel. The length of one meter is selected so the striker would not exceed the velocity of the striker that would induce plastic deformation in the C350 maraging steel and also kept the incident bar supports a reasonable distance apart.

For the End caps of the pressure chamber, Young [1] machined the 2 end plates from solid blocks of AISI 1018 steel into an inverted “T” shape with a hole in the centre of the upright to allow the barrel to pass through, and a circular groove machined into the upright to have a transition fit with the ends of the thin-wall cylinder. The thickness of the two end caps was also calculated from Pressure vessels the ASME code Section VIII, Division 1, UG-34 [8] according to the following relationship:

$$t_{ec} = d\sqrt{\frac{\mathrm{CP}}{\mathrm{SJ}}}$$

Where:

d = Bolt circle diameter

C = Factor depending on method of attachment, code Section VIII, Division 1, Fig. UG-34 ASME boiler and pressure vessel code [8]

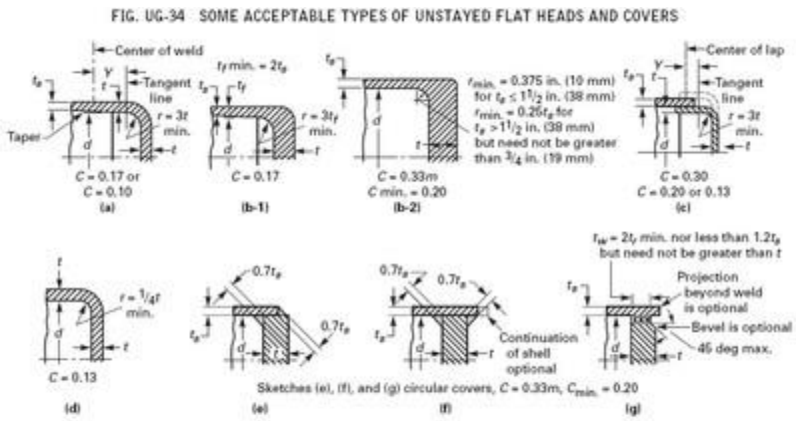

Figure 2-4: Factor depending on method of attachment, fig. UG-34 2013 ASME boiler and pressure vessel code

From the ASME boiler and pressure vessel code [8], the value for, C, was determined to be 0.30 and, d, was calculated to be 141mm. With these values the thickness of the flat end caps was calculated to be 20.78mm, however a thickness of 25.4mm was used since this is a common size available from steel suppliers.

From Young’s design [1] , The thin-wall cylinder was fabricated from a 114.5mm diameter, solid AISI 4140 bar stock by boring out the centre material to achieve a final inside diameter of 101.8mm. The thin-wall cylinder was fabricated this way to avoid having a weld seam in the pressure chamber. The thin-wall cylinder was placed concentrically around the barrel and held in place by the two end caps. The thickness of the thin-wall cylinder was calculated using the following equation from Pressure vessels the ASME code Section VIII, Division 1, UG-27 [8] for a thin-wall cylinder was used:

$$t = \frac{PR}{SJ - 0.6P}$$

This equation is valid for "use when, t, is less than 1/2 R or, P, is less than 0.385 SJ"[8]. The joint efficiency is equal to 1, since there is no weld joint in the thin-wall cylinder, and the allowable stress has a value of "S = 95 MPa"[8]. This gave a wall thickness of 3.85mm, but a thickness of 6.35mm was used on the thin-wall cylinder for ease of machining the thin-wall cylinder.

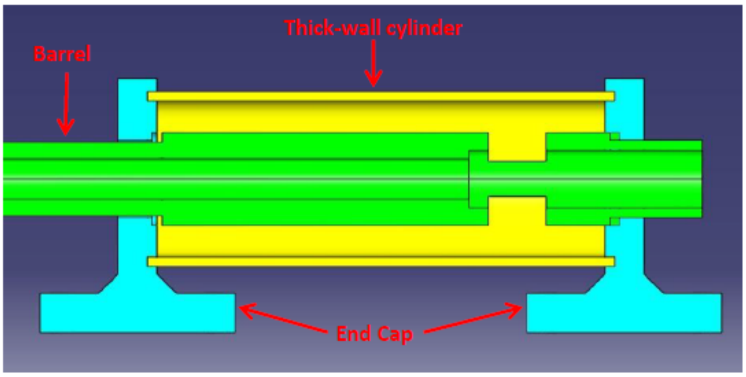

Figure 2-5: The cross section view of the pressure chamber of Young's design [1]

For the valve assembly, Young's design [1] use cylindrical valve that is fabricated from Teflon with a tight fit with the barrel, with two O-ring grooves to seal the barrel off from the pressure chamber, and a groove for locating the collar which is attached to the pneumatic cylinders. The valve has a hole in the centre to concentrically locate the incident bar in the barrel.

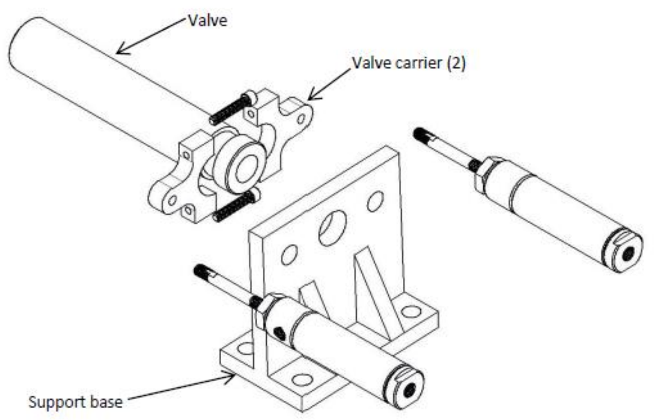

Figure 2-6: The 3D model of the valve assembly of Young's design [1]

Other than that, other design of a gas gun is developed by Owens's research [9]. Owen's design used a normal gas gun which was modified such that the chamber and valve are off axis from the barrel. The air enters the barrel at a 45º angle. When the valve is opened, the gas can flow through the elbow and wye fittings and into the barrel, thus propelling the striker forward. The aft end of the barrel has a bronze bushing surrounding the location where the incident bar passes through to prevent air from leaking out.

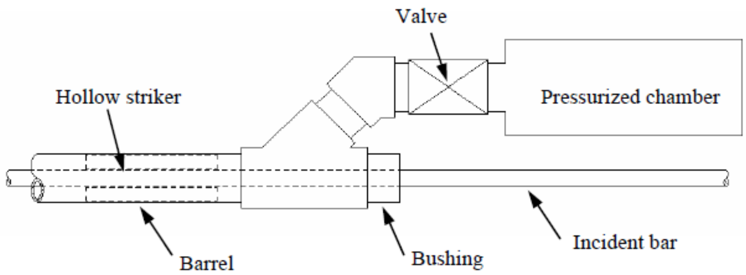

Figure 2-7: The schematic diagram of the Owen's design [9]

As for the Owen's design [9], the gun barrel is the gun barrel is fabricated from seamless steel tubing with 50.8 mm outside diameter and 38.1 mm inside diameter. Several 12.7 mm diameter exhaust holes are drilled into the fore end of the barrel.

For Owen’s design [9], it is not stated on the parameter of the end caps or the theory on how to find the thickness of the end caps. Owen also does not stated on how the end cap was fabricated.

For Owen’s design [9], The gas chamber is constructed from a 152 mm diameter cast iron pipe. It is sealed to the aluminium bulkheads with a room temperature vulcanized rubber sealant. Four, 25.4 mm diameter threaded rods are used to maintain a compressive load on the chamber. A pressure gage and quick release fitting is attached through the bulkhead to allow the chamber to be filled with gas.

However, Owen’s design on valve assembly is relative simpler where he uses cast iron pipe fittings with diameter of 38.1 mm are to assemble the barrel to the valve and also to connect the valve to the aluminium bulkhead as figure 2-7.

The third design of a gas gun is developed by G.C Ganzenmuller’s research[10] study in 2017. The mechanism of the design is that he used an external pneumatic tube which is mounted off-axis and connects to the striker via pulling rod. The pneumatic tube may be constructed with a large piston diameter to operate with low pressure compressed air. As the compressed air move in to the pneumatic tube, the air will push the piston and the piston will pull the connecting rod and push the striker to the anvil.

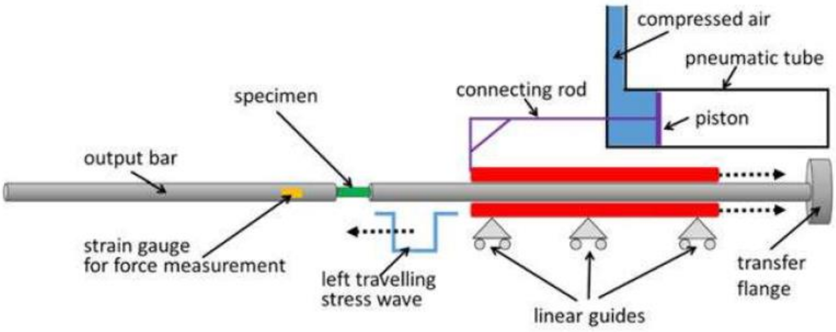

Figure 2-8: The schematic diagram of the G.C Ganzenmuller’s diagram [10]

Fourth design of the gas gun was developed by Juan Felipe Acosta research from Wichita State University in 2002[11]. The mechanism of the design is almost similar to Kevin Young’s design where it used compressed air tank, gun barrel and valve assembly. The gas gun is designed so that incident bar the incident bar goes through its barrel as well as the trigger box allowing mobility but capable of holding air pressure. The incident bar is concentric with the gun barrel. This allows for a hollow striker bar to ride along the incident bar and inside the gun barrel. The hollow striker bar is manufactured from the same material as the bars. The bars are supported approximately every 254 mm using frelon-lined linear plane bearings. The air gun operated on compressed air and could fire striker bars of different lengths ranging from 38.1 mm to 304.8 mm. The air intake was controlled with a single-stage pressure regulator rated for a 1 MPa drop. A transfer flange is attached to the end of the incident bar to generate the loading pulse. The design of the Acosta's Gas Gun is shown in figure 2-9 below.

For the Acosta's design [11], is not mentioned on what the material is made of and also some of the parameters of the design of the Gun Barrel. The only known parameter of the Gun Barrel is the inner diameter of the bore hole which is 31.75mm.

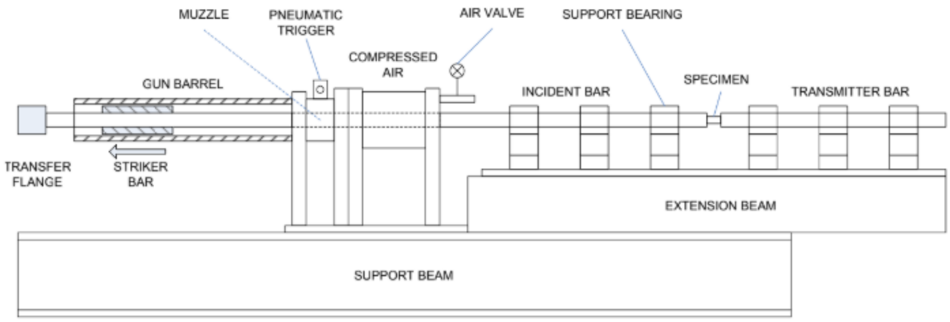

Figure 2-9: Acosta's design on the tensile Split Hopkinson Pressure Bar [11]

For Acosta's design [11], It is also have not stated on the fabrication of the end cap. However, the thickness of the end cap used is 31.75mm contained circular groove of 168.4mm of diameter. Also, there is a circular hole 31.75mm for the incident bar to pass through which is concentric to the circular groove. Figure 2-10 shown below is the Acosta end cap design.

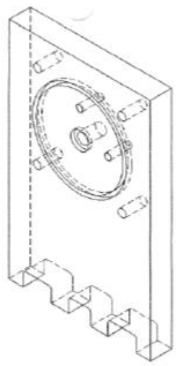

Figure 2-10: Acosta's 3D CAD drawing of pressure vessel End Cap [11]

For Acosta’s pressure vessel design, the thin-wall cylinder designed have a thickness of 6.35mm with outer diameter of 168.3mm and inner diameter of 155.6mm. The length of the cylinder is constructed to length of 261.6mm. The material of the thick wall cylinder is not known as it is not mentioned in Acosta’s thesis. Also, the way of the cylinder is fabricated is also not stated in the thesis.

For Acosta’s design, the valve assembly is quiet similar to Young’s assembly design but using a pneumatic trigger. The pneumatic trigger is assembled between at one end of the gun barrel and the pressure vessel to hold the pressurized air in the vessel before releasing it into the barrel. Not only that, the pneumatic trigger to allow mobility which in other word to ease to assembly and disassembly of the component of the gas gun.

The major difference between Young’s design and Acosta’s design with Owen’s design is that the valve of Young’s and Acosta’s design is located inside the pressure chamber. However, for Owen’s design, the valve is located outside the pressure chamber where it is manually actuated by an operator. Furthermore, for Young’s and Acosta design, the incident bar goes through the barrel, both of the end caps, the whole pressure chamber and the whole valve. However, the incident bar for Owen’s design only goes through the barrel and end caps. Another difference is the way all three of the design actuated their pressurized air into the Gun barrel. Young’s design used pneumatic cylinder to allow pressurized air to be released into the barrel as the air pressure tank reached the desired value. However, for Acosta’s design, it actuated the air pressure using pneumatic trigger to release the air in the pressure vessel in to the barrel while Owen's design use a ball valve to release the pressure in air pressure tank into the barrel.

2.3 Theory

2.3.1 Thin-wall pressure vessel

This thin-wall pressure vessel theory is applicable to the air pressure tank thick-wall and the Gun Barrel thin-wall. The reason on why the Gun Barrel used the same calculation as the air pressure tank is because of safety of the barrel where the barrel need to be capable to withstand the same amount of the pressure as the pressure tank if the striker is stuck in the barrel.

To calculate the thin-wall thickness, pressure vessels equation are used in the calculation so that the wall is capable to withstand the high amount of the pressure. The derivation of the pressure vessel equation starts from the force exerted by the wall of the barrel cylinder due to the internal pressure. Total force on one half of the cylinder owing to the internal pressure 'p':

$$Force = Pressure \times Projected\ Area$$

$$F = PDL \qquad 2\text{-}1$$

The total resisting force owing to hoop stresses set up in the cylinder walls:

$$F = 2\ \sigma \mathrm{Lt} \qquad 2\text{-}2$$

Equate the total resisting force and the Total force on one half of the cylinder owing to the internal pressure 'p':

$$2\sigma \mathrm{Lt} = PDL \qquad 2\text{-}3$$

$$\sigma = \frac{PD}{2t} \qquad 2\text{-}4$$

Therefore, the Circumferential or hoop Stress is the following equation:

$$\sigma = \frac{PR}{t} \tag{2-5}$$

Which resulted in thin-wall equation:

$$\mathrm{t} = \frac{PR}{\sigma} \tag{2-6}$$

Thus, from the hoop stress equation, the ASME code Section VIII, Division 1, UG-27 [8] thin-wall equation can be used.

$$t = \frac{PR}{SJ - 0.6P} \tag{2-7}$$

The calculation of the allowable stress:

$$S = \frac{yield\ strength}{FS} \tag{2-8}$$

Note: his equation is valid for “use when, t, is less than 1/2 R or, P, is less than 0.385 SJ”

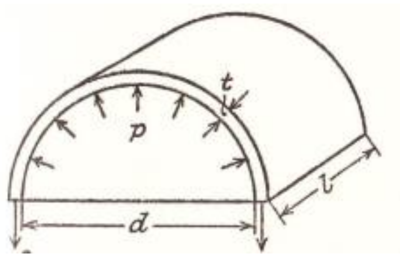

Figure 2-11: Cross-sectional of thin-wall cylinder exerted by hoop stress

To calculate the thickness of the hemispherical head thin-wall, formula of ASME Code Section VIII, Division 1, Fig. UG-27 is used which are:

$$t = \frac{Pr_c}{2SJ - 0.2P} \tag{2-9}$$

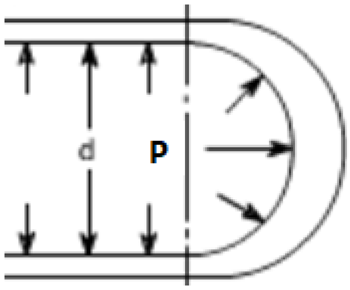

Figure 2-12: Cross-sectional of the hemispherical head cap exerted by pressure

2.3.2 Bar Design

The stresses exerted on the thread on the bars is calculated by calculating the ratio of the impact force to shear area of the threads. The stress exerted can be calculated using the following equation:

$$\sigma_t = \frac{F_{st}}{A_t} \tag{2-10}$$

In order to find the force exerted on the thread, area of the striker and yield stress of the striker need to be determine, which using the following equation:

$$F_{st} = A_{st}\sigma_{st} \tag{2-11}$$

The area of the striker is calculated using the following equation:

$$\mathrm{A}_{st} = \frac{3.142}{4}(d_o^2 - d_i^2) \tag{2-12}$$

In order to find the Shear area of the thread, number of thread, circumference of the thread and the distance through the thread at the effective pitch diameter is calculated as the following equation:

$$A_t = No.of\ thread\ \times \mathrm{C}_t \times 2X \tag{2-13}$$

The thread circumference is calculated using following equation:

$$\mathrm{C}_t = 3.142 \times D_{min} \tag{2-14}$$

The distance through the thread at the effective pitch diameter is calculated as the following equation:

$$X = \frac{\frac{\left(D_{maj} + \frac{H}{4}\right) - D_{min}}{2}}{\tan 60} \qquad 2\text{-}15$$

Figure 2-13 below shows the basic M-profile parameter of a thread which was taken from myodesie.com website[12]

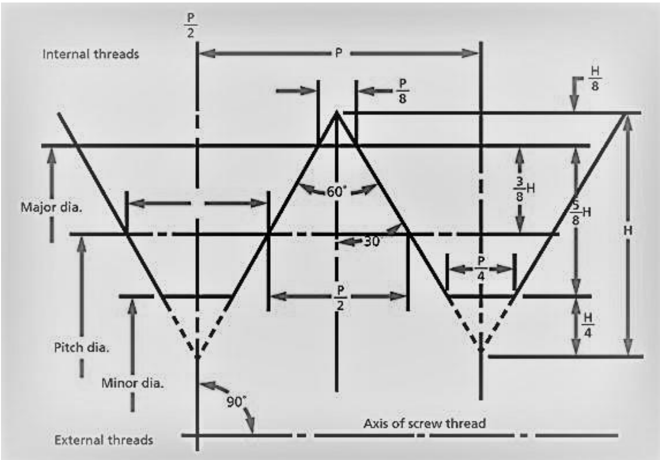

Figure 2-13: M-profile parameter of a standard thread

Then, using a Von Mises equation, the shear stress is converted to normal stress as following equation:

$$\sigma = \sqrt{3}\sigma_t < yield\ strengh\ of\ the\ bar\ material \qquad 2\text{-}16$$

The maximum stress exerted by the thread of the bar must be lower than the yield strength of the bar material so that the thread of the bar is capable to withstand the maximum stress exerted without yielding.

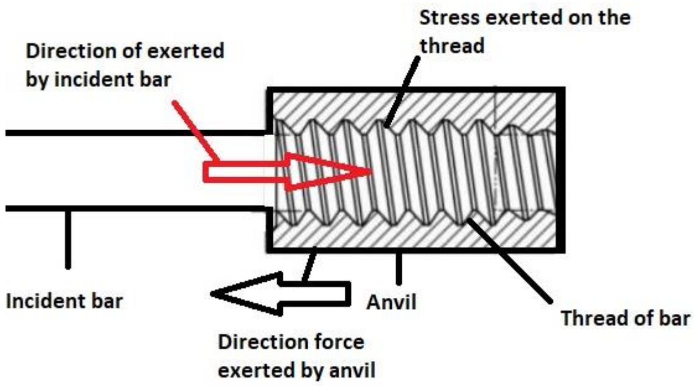

Figure 2-14: Diagram of the incident bar/anvil thread interface

2.3.3 Bolt requirement for vertical plate of barrel support

The stress exerted of thread on the bolt for the vertical plate of the second type barrel support is calculate by the ratio of the force to shear area of the threads:

$$\sigma_t = \frac{F_{bolt}}{A_t} \tag{2-17}$$

The assumption is that the force of the bolts is equal to the force of pressurized air to launch the striker. Therefore, force of pressurized air is the product of the maximum air pressure used and the area of the barrel bore. The following is the equation of the force exerted by the bolt.

$$F_{air} = A_{barrel} \times P_{air} \tag{2-18}$$

Thus, the exerted force of the bolts:

$$F_{air} = F_{bolt} \tag{2-19}$$

For the shear area of the thread:

$$A_t = No.of\ thread\ \times \mathrm{C}_t \times 2X$$

The thread circumference is calculated using following equation:

$$C_t = 3.142 \times D_{min}$$

The distance through the thread at the effective pitch diameter is calculated as the following equation:

$$X = \frac{\frac{\left(D_{maj} + \frac{H}{4}\right) - D_{min}}{2}}{\tan 60}$$

Therefore, the equation of shear stress of the thread on the bolt:

$$\sigma_t = \frac{F_{bolt}}{A_t} \tag{2-20}$$

Then, using a Von Mises equation, the shear stress is converted to normal stress as following equation:

$$\sigma = \sqrt{3}\sigma_t < yield\ strengh\ of\ the\ material \tag{2-21}$$

The maximum stress exerted by the thread of the bolt on the vertical plate of the barrel support must be lower than the yield strength of the material so that the material of the bolt is capable to withstand the maximum stress exerted without yielding.

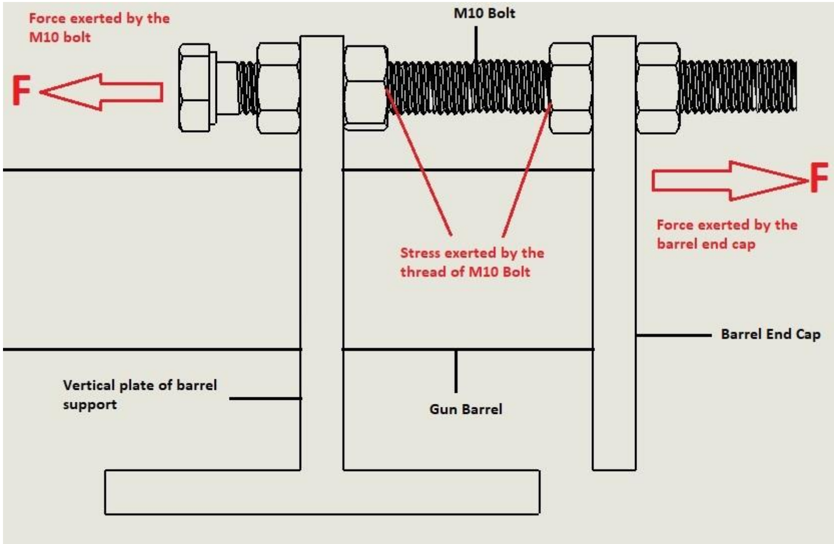

Figure 2-15: Force exerted by the bolt at vertical plate of barrel support

2.3.4 Bolt requirement due to shear force

This theory is applied to the bolt of the second type of barrel support and the bolt of stopper as both of it experience a shear force. The normal stress exerted on the bolt start from the shear stress exerted of bolt on the horizontal plate of the barrel support is calculated by the ratio of the shear force exerted by a bolt and the shear area of the bolt:

$$\tau = \frac{F_{bolt}}{A_{bolt}} \qquad \text{2-22}$$

The assumption is that the maximum shear force exerted on the bolt is equal to the maximum force of pressurized air to launch striker which is the product of Maximum air pressure used and the area of the barrel.

$$F_{air} = F_{bolt} \qquad \text{2-23}$$

For the calculation of the shear area of the bolt:

$$\mathrm{A}_{bolt} = \frac{3.142}{4}(d^2) \qquad \text{2-24}$$

Thus, the shear stress exerted on each of the bolt:

$$\tau = \frac{F_{bolt}}{A_{bolt}} \qquad \text{2-25}$$

To calculate the normal stress, Von Mises equation is used:

$$\sigma = \sqrt{3}\tau < yield\ strengh\ of\ the\ material$$

The maximum stress exerted by the bolt must be lower than the yield strength of the material so that the material of the bolt is capable to withstand the maximum stress exerted without yielding.

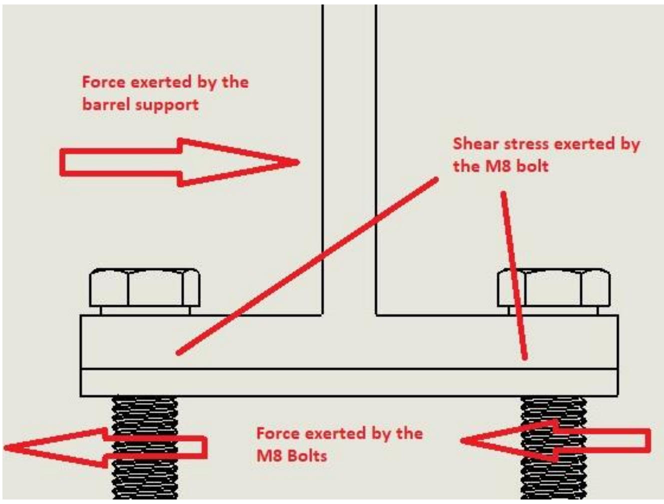

Figure 2-16: Shear stress exerted by the bolt

2.3.5 Determination of Striker Velocity

For the Gas Gun of the Split Hopkinson Pressure Bar, one of the factor that determine the strain rate of a specimen material is the striker velocity. The striker velocity equation below was taken from Young research [1].

The stress in the striker, σ_{st}, and incident bar, σ_i, are "related with the velocity in the common interface" [13].

$$\sigma_{st} = \rho_{st} C_{st} (v_{st} - v_i) \qquad \text{2-26}$$

$$\sigma_i = \rho_B C_B v_i \qquad \text{2-27}$$

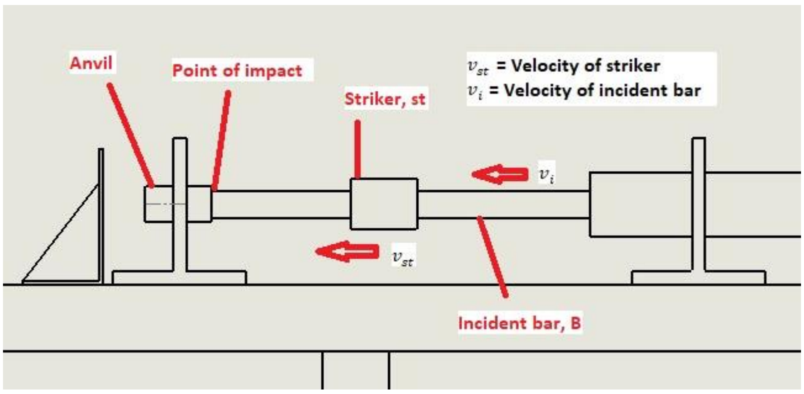

Figure 2-17: Variables used for the determination of striker velocity

Also at impact, the force exerted onto the incident bar is equal to the reaction force exerted on the striker:

$$\mathrm{F}_{st} = \mathrm{F}_{i} \qquad 2\text{-}28$$

$$\sigma_{st}\mathrm{A}_{st} = A_B\sigma_i \qquad 2\text{-}29$$

Substituting equation 2.26 and 2.27 into equation 2.29 gives:

$$\rho_{st}C_{st}(v_{st} - v_i)\mathrm{A}_{st} = A_B\rho_B C_B v_i \qquad 2\text{-}30$$

$$\beta = \frac{A_B\rho_B C_B}{\mathrm{A}_{st}\rho_{st}C_{st}} \qquad 2\text{-}31$$

In order to ensure one dimensional wave propagation, the properties and cross-sectional areas of the striker bar and incident bar must be equal, which yields β = 1. Therefore equation 2.30 can be rearranged as:

$$v_i = \frac{\beta v_{st}}{1 + \beta} = \frac{v_{st}}{2} \qquad 2\text{-}32$$

Substituting equation 2.32 into equation 2.26 and equation 2.27 gives:

$$\sigma_{st} = \rho_{st}C_{st}(v_{st} - \frac{v_{st}}{2}) = \frac{v_{st}\rho_B C_B}{2} \qquad 2\text{-}33$$

$$\sigma_i = \frac{v_{st}\rho_B C_B}{2} \qquad 2\text{-}34$$

Similar to the interface between the striker and incident bars, the force exerted onto the specimen is equal to the reaction force exerted on the incident bar, assuming no losses:

$$\mathrm{F}_{s} = \mathrm{F}_{i} \qquad 2\text{-}35$$