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MECHANICAL ENGINEERING

Design project I

Hydraulic Bottle Jack Design

SECTION II

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ABSTRACT

Car jack is a device used to lift up the cars while changing the tires during an emergency. Car jacks are available at the market has some disadvantages such as requiring more energy to operate, are not suitable for women and cannot be used on the uneven surface. The purpose of this project is to modify the design of the existing car jack in terms of its functionality and also human factors considerations. In this project, the scopes of research were on the designing 1.5-ton at 20cm lifting capacity of car jack by using optimization concept. To optimize the existing design, the hand lifter has been replaced by the use of pedal lever as it can reduce energy usage. In the process of obtaining a suitable design, the customer needs will be translating to the engineering characteristic to obtain the concepts that need to be modified and fabricated. From the house of quality, the best concept will be valued based on the weighted rating method. The morphology chart had been used to obtain the best concept solution. After the best concept had been selected using the weighted rating method, next step was to determine the part and component that can be modified by arrange the part into chunks and clustering with the component according to the function or system. From this step, it can be determined which component can be reduced or modified. Then the configuration design was analyzed. According the function factor and critical issue so that the design that had been implementing was according to the specification and customer requirement. The last step for this project was parametric design. In this topic, the new design concept will be calculated to obtain the required force and compared with the theoretical calculation in the table of human factor.

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CHAPTER I

1. INTRODUCTION

1.1 Background

A jack is a mechanical device which uses a screw thread or a hydraulic cylinder to lift heavy loads or apply great linear forces. The most common forms of jacks available in the market are Scissor car jacks, House jacks, Hydraulic jacks, Pneumatic jacks and Strand jacks that are extensively used in Construction, Industrial, Automobile and Engineering segments. In most of the powerful jacks, hydraulic power is used to provide more lift over greater distances. Mechanical jacks usually have lifting capacity ranging from 1.5 tons to 3 tons. Hydraulic jacks are mechanical devices used to lift heavy loads, vehicles, weight equipment or apply great forces using hydraulic fluid as the main source of power. These are widely used in automotive, industrial and construction industries. These are sturdy in construction, compact in size, portable and capable of exerting great forces. It consists of two cylinders of different sizes which are connected together by a pipe and a hydraulic fluid or oil. The hydraulic fluid is incompressible and using a pump plunger is forced into the cylinder of the jack. Oil is used because of its stable and self-lubricating nature. When the plunger pulls back, oil is drawn out of the reservoir and it goes inside the pump chamber. When the plunger moves forward, the oil is pushed back into the cylinder. This oil movement builds up pressure in the cylinder. And it is this pressure which leads to the working of the hydraulic jack. It also finds usage in workshops and also lifts elevators in low and medium rise buildings. These can be segmented into two types: Bottle Hydraulic Jack and Floor Hydraulic Jack. Bottles are portable in design; in these the piston is in a vertical position and it supports a bearing pad which touches the object being lifted. Bottle Hydraulic are most appropriate for lifting vehicles (cars, trucks, SUVs, trailers), houses and other heavy objects. In a Floor Jacks, the piston is in a horizontal position and there is a long arm which provides the vertical motion to a lifting pad. There are wheels and castors in floor jacks. The working principle of all hydraulic jacks is common but these differ in their shapes and sizes. Hydraulic jacks with varied sizes and specifications are used to lift different types of heavy equipment and vehicles such as bulldozers, forklifts, elevators, trolleys & trailers and excavators. These can also be found in household equipment's as well like door stoppers, cars, bikes etc. Hydraulic Jacks are high in demand across the globe owing to their sturdy construction, reliable & hassle free operation, unparalleled performance, user-friendly design and less maintenance [11].

Hydraulic jack can be classified into three, based on the application of the jack that are:

1. Hydraulic service

2. Hydraulic bottle jack

3. Session hydraulic bottle jack

Fundamental features of using hydraulics compared to mechanics for force and torque increase/decrease in a transmission.

Hydraulic machinery is machines and tools which use fluid power to do work. Heavy equipment is a common example. In this type of machine, high-pressure liquid called hydraulic fluid is transmitted throughout the machine to various hydraulic motors and hydraulic cylinders. The fluid is controlled directly or automatically by control valves and distributed through hoses and tubes.

The popularity of hydraulic machinery is due to the very large amount of power that can be transferred through small tubes and flexible hoses, and the high power density and wide array of actuators that can make use of this power.

Hydraulic machinery is operated by the use of hydraulics, where a liquid is the powering medium. Pneumatics, on the other side, is based on the use of a gas as the medium for power transmission, generation and control [11].

1.1.1 Definition

A bottle jack is a hydraulic jack which resembles a bottle in shape, having a cylindrical body and a neck, from which the hydraulic ram emerges. In a bottle jack the piston is vertical and directly supports a bearing pad that contacts the object being lifted. With a single action piston, the lift is somewhat less than twice the collapsed height of the jack, making it suitable only for vehicles with a relatively high clearance. For lifting structures such as houses the hydraulic interconnection of multiple vertical jacks through valves enables the even distribution of forces while enabling close control of the lift [8].



Figure 1 types of Hydraulic jacks

Hydraulic jacks, the bottle hydraulic jack as consists of a cylinder, a piston and lever operated pump and their capacities is to up to 50 tones and lifting height is e. The device

pushes against a piston, pressure built in the jack container up to 22in. Large hydraulic jack may be provided with two pumps. In other word is a device used invariably in the machinery and equipment. The device itself is light and portable but the device is capable of exerting great force.

Hydraulic jacks are typically used for shop work, rather than as an emergency jack to be carried with the vehicle. Use of jacks not designed for a specific vehicle requires more than the usual care in selecting ground conditions, the jacking point on the vehicle, and to ensure stability when the jack is extended. Hydraulic jacks are often used to lift elevators in low and medium rise buildings [9].

1.1.2 Application of hydraulic jack

- For changing of tires of vehicles while in emergency e.g. when the tire of the vehicle is damaged by sharp object [9].
- Building of field erected storage tanks.
- Increasing tank capacity by adding shell rings or courses.
- Dismantling of old tanks.
- Repair or replacement of tank bottom plate.
- Repair to tank foundation.
- Erection of other circular structures such as reactor shields in nuclear power stations, etc.

1.1.3 Material selection

Table 1 bottle jack materials

NO	PARTS	MATERIAL
1	O-RING (SEALING)	NITRIDE BUTADIENE
2	SOLID RAM	Alloy steel, AISI1015
3	SPRING FOR SAFETY VALVE	beryllium copper C1720
4	HOLLOW RAMS	milled steel ASIM, A487
5	CYLINDER(CYLINDER RAM)	Milled steel ASTM , A487
6	PLUNGE CYLINDER	Milled steel ASTM , A487
7	PLUNGE WITH PLUNGE ROD	Alloy steel, AISI 1002
8	NUT	malleable cast iron ASTM, A47
9	TOP-CUP WITH SQUARE THREAD (SADDLE)	Gray Cast iron, ASTM60
10	HANDLE AND HANDLE SOCKET	Carbon steel, AISI1030
11	HANDLE SOCKET PINS AND RELEASING SCREWS	Alloy steel ANSI1318
12	BASE PLATE	Gray cast iron ASM60
13	BALL FOR RELEASING AND SAFTEY VALVE RISERVIOR	stainless steel AISI 900
14	SCREW	Alloy steel AISI 1040
15	HANDLE SOCKET LINK	alloy steel ANSI1318

Material selection for housing

Because housings are intricate shaped, they are mostly cast.

Gray cast iron;

- it has good cast ability & machinability,
- corrosion resistance,
- good wear resistance (b/c it is impregnated with graphite) [8].

Table 2 properties for housing material

Mechanical Properties of Some Housing Materials

	Material	S_y (Mpa)	E (Mpa)	α 1/°C
Rolled plate	SAE 1020 annealed	250	$2.00 \cdot 10^5$	$12.2 \cdot 10^{-6}$
	SAE 4130 normalized	490	$2.00 \cdot 10^5$	$12.2 \cdot 10^{-6}$
	Hardox 400, Creusabro 4000	900	$2.06 \cdot 10^5$	$12.2 \cdot 10^{-6}$
	Cast steel	240–260	$1.70 \cdot 10^5$	$12.2 \cdot 10^{-6}$
Casting	Gray iron	$S_u = 170\text{--}250$	$(0.9\text{--}1) \cdot 10^5$ See Eq. (9.1)	$(10\text{--}12) \cdot 10^{-6}$
	High-duty cast iron	320–370	$1.70 \cdot 10^5$	$12 \cdot 10^{-6}$
	Aluminum alloys	150–180	$0.72 \cdot 10^5$	$(20\text{--}26) \cdot 10^{-6}$
	Magnesium alloys	80–120	$0.45 \cdot 10^5$	$(26\text{--}30) \cdot 10^{-6}$

1.1.4 Operating principle of hydraulic bottle jack

A Hydraulic Jack uses a fluid, which is incompressible, that is forced into a cylinder by a pump plunger. Oil is used since it is self-lubricating and stable. When the plunger pulls back, it draws oil out of the reservoir through a suction check valve into the cylinder. The suction valve ball is within the chamber and opens with each draw of the plunger. The discharge valve ball is outside the chamber and opens when the oil is pushed into the cylinder. At this point the suction ball within the chamber is forced shut and oil pressure builds in the cylinder. When the handle is moved upward, ball (1) is seated and fluid is drawn from the reservoir through the screen into the valve assembly. From there fluid enters the plunger cylinder [11].

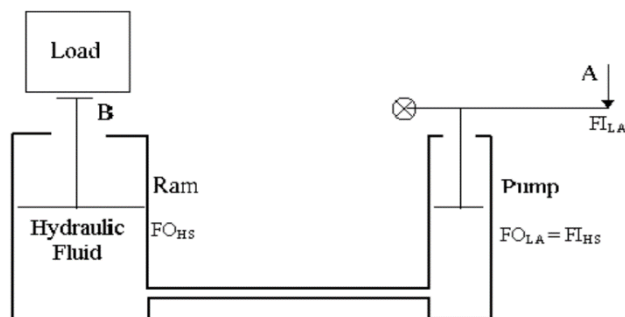


Figure 2 Hydraulic principle

Other down ward stroke of the handle or pressure stroke, fluid pressure to the master cylinder. Continues operations of the pump will create sufficient pressure to raise the first stage ram. Until it hits the stop of the second stage ram with the additional pressure. The second stage ram will be raised until it hits the stop of the third stage ram. If the pressure is still continued, the third stage ram will be raised to the full height of lift until it hits the stop of the cylinder.

At the end of the travel of the rams or with an over load applied to the jack. Excessive pressure from the continued pump unseats ball and fluid is by passed to the reservoir (with this occurs the handle will suddenly drop or go soft during the pumping stroke). If the releasing screw is opened pressure is released through unseated ball and fluid to the reservoir [5].

The following figure shows how hydraulic bottle jack works.

- a) Represents base dimension
- b) represents maximum height.
- c) represents minimum height.
- d) Represents hydraulic stroke (travel)
- e) represents extension screw (travel)

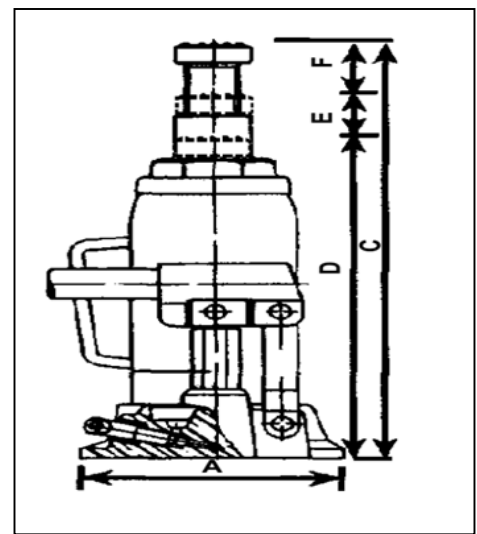


Figure 3 shows principle of hydraulic bottle jack.

Operating instruction

Prior to its first use, make sure to check for proper hydraulic oil level in the bottle jack. Then roughly test the bottle jack for proper prior to its actual size. If the jack appears not to be working properly it may be necessary to bleed its hydraulic air of excess air. To do so [5].

- ❖ Remove the oil filler plug
- ❖ Assemble the complete handle assembly and lower the saddle of the jack fully.
- ❖ Insert on the lower portion of the handle assembly over the release screw.
- ❖ Turn the handle assembly clock wise to close the valve release screw.
- ❖ Insert the handle into the plunger or fulcrum assembly.
- ❖ Apply pressure to the saddle and pump the handle assembly slowly several times.

- ❖ The trapped air is pressurized and will be audible when released in the manner.
- ❖ Once bled air check oil fill hole & if necessary, top of the oil fill hole with hydraulic oil.
- ❖ Then replace the oil filler plug.
- ❖ Turn the jack several times for proper operation before attempting to lift a load.

The following guide line are help full;

- ❖ Assemble handle, ensure that spring clips align with slot.
- ❖ With an emergency break place the vehicle in the park.
- ❖ Locate and close release valve by turning handle clock wise.
- ❖ Verify lift point center jack saddle under lift point.
- ❖ Insert handle into sleeve and pump to contact lift point. To lift continuous pumping until load reaches desired.

1.1.5 components

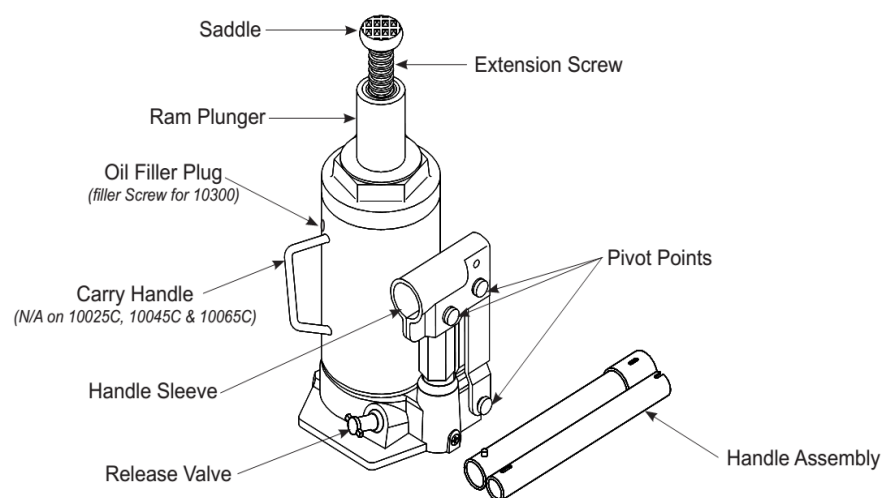
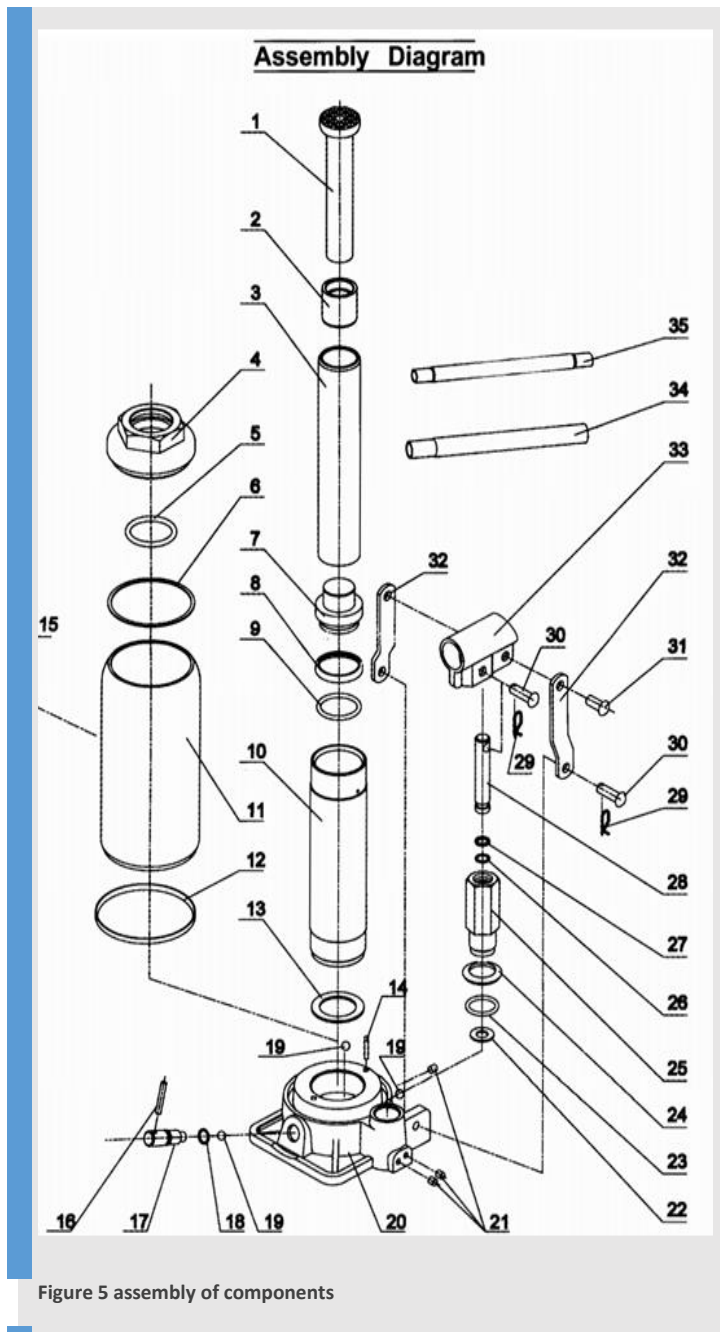


Figure 4 Main components of hydraulic bottle jack

- i. O-ring (sealing)
 - a. O-ring are torus-shaped seals of circular cross-section made from a variety of elastomeric and are used in static and dynamic condition.
- ii. Solid ram
- iii. Spring for safety valve

- iv. Hollow rams
- v. Cylinder (cylinder ram; just like hollow ram it is subjected to fluid friction, compressive load, tensile load, buckling and corrosion.
- vi. Plunge cylinder: - again its characteristics are the same as that of ram cylinder
- vii. Plunge with plunge rod
- viii. Nut
- ix. Top-cup with square thread (saddle)
- x. Handle and handle socket
- xi. Handle socket pins and releasing screws
- xii. Base plate
- xiii. Ball for releasing and safety valve reservoir: - it is all the same character to that of cylinder.

Detail part list



Part	Description	Qty
1	Valve Block	1
2	copper washer	1
3	Cylinder	1
4	O-Ring	1
5	O-Ring Retainer	1
6	Ram	1
7	Top Cup	1
8	O-Ring	1
9	Scaling Gasket	1
10	Oil Tank	1
11	Oil Plug	1
12	Seal Ring	1
13	Tube	1
14	Bolt	1
15	Screw	2
16	Ball	1
17	Seal Ring	1
18	Relesse Spindle	1
19	Ball	1
20	Copper Washer	1
21	Pump Cylinder	1
22	U-Cup Seal	1
23	Back Up Ring	1
24	Plunger	1
25	Pump Link	1
26	Handle	1
27	Handle Socket	1
28	Shaft	2
29	Cutter Pin	1
30	O-Ring	1
	Seal Kit	1

1.1.6 Factor of safety

$S_Y = \sigma_{c \text{ direct}} * C$ where s_y =yield strength

$\sigma_{c \text{ direct}}$ = direct compressive stress

C = constant

This constant decides how much more the strength should be as compared to the induced stress. It assures the safety of machine member from failure and hence is termed as factor of safety and may be designated as N or $F.S.$

The right hand side of the above equation is termed as permissible or safe allowable stress, while left hand is induced stress. F.S may be arbitrary selected as 3 to 5 based on yielding or 5 to 7 based on fracture failure [9].

1.1.7 Housing design

Housing encloses and supports the machine parts. It is most commonly made of cast iron or cast steel although welding is also used as a means of fabrication.

Design of housing begins when the other elements of a mechanism have been designed in every detail. Housing is just one of the components of the, as a shell is an integral part of a tortoise's skeleton. Housing is almost always the biggest part of a mechanism. It is usually made as a closed box, so that the rest parts are mounted inside it. In this respect the housing also serves as a guard [5].

Function of housing

SERVING AS AGUARD; Housing isolates a mechanism from the surrounding to create conditions inside that are comfortable for the mechanism. On the other hand, the housing isolates surrounding, including personal, from the mechanism. So that we are protected from hearing the noise and from being sprinkled with hot oil; besides we can't easily put our finger into it. The housing is an assembly basis for the rest of parts and unit of the mechanism. There for it should meet some obligatory requirement;

Because mechanism is assembled inside the housing, the latter must be designed so as to enable assembly operation and the control of parameters to be checked after assembly. i.e the housing should have;

- Parting or sufficiently sized openings that enables convenient assembly &
- Control of the mechanism.
- Dimensions should be accuracy.
- Material should be;
- Heat resistance
- Ability to clump noise and vibration.
- Low weight
- Ability to with stand aggressive environment such as; sea water & others which can be achieved on any material by suitable coating [5].

Alignment of pieces and shapes of housing

Means must be provided to get the various pieces of the housings to line up properly when the machine is assembled. This lining up cannot be done by the bolts which hold the

housing together since these bolts cannot be very accurately located themselves. Because of this, and to aid assembly, the holes through which the bolts pass are considerably larger than the bolts completely incapable of accurately aligning the housing parts [5].

Supporting the housing

The housing, which commonly supports the machine parts, must itself be supported on a floor or foundation surface. This is generally accomplished by forming a base as part of the housing. The base is on only one piece of the housing and the housing should be designed with this in mind. If more than one piece of the housing were extended downward to rest on the floor or foundation, difficulty would be encountered in trying to get the bottom surfaces of the pieces to line up together. Hence the base should be made of one piece wherever possible [5].

1.1.8 Manufacturing process

For the plunger of the pump.

STEPS;

- I. First select with the appropriate diameter.
- II. Then face and turn the work piece according to the dimensions.
- III. Make internal groove for O-ring with the required thickness and depth.
- IV. Finally, with the precise finishing machine make finishing.

For solid ram

STEPS;

- I. Initially select the blank (work piece) with the required dimension and faces the work piece by using lathe machine for the required and exact dimension
- II. Secondly drill the internal blind with the required dimension (diameter) by using lathe machine without missing the center
- III. Finally: - steep turn the external part of the solid ram and finishing process with high surface finishes required so that use precise surface finish machine.

For hollow ram

STEPS;

- I. Make the blank *by casting with* a dimension of the nearest to the exact dimension and bore and turn the internal and external part by using lathe machine.
- II. Make groove for the O-ring seal with the appropriate dimension.
- III. Make some finishing with précised surface.

- IV. Finish material or machine like internal and external grinding machine.

For the base plate

STEPS;

- I. By using casting process make the lower base plate blanks which is suitable for machining
- II. Correct the casting blank for the appropriate dimensions.
- III. Drill the upper surface for the fluid passage to the required diameter and make thread for releasing screw with 1mm pitch.
- IV. Make thread the upper inner surface to tight the cylinder and the reservoir with a metallic thread engagement part with 1mm pitch by using lathe machine.
- V. Finally make the necessary surface finish.

For extension screw

STEPS;

- I. Select the blank (work piece) with a required dimension and by using lathe machine face and turn for the required dimension.
- II. Chamfer the lower part of the screw by 45° for easy of losing.
- III. Make a thread with a square thread by using the appropriate pitch and use milling machine for easiness, when using indenting to this thread, use the appropriate pitch.
- IV. Finally use the necessary fining by using surface finishing machine

For the handle

The handle is made from the round bar.

STEPS;

- I. Turn the round bar and face with the required specification.
- II. Make to some length knurling for good griping ability.
- III. Finally make the required surface finish. It is usually made by extrusion process.

For the socket

STEPS;

- I. The socket is made from socket metal of the required thickness.
- II. Cut the sheet metal with the correct lay out by using cutter machines.
- III. Drill the socket metal for pin connection with a drilling machine.

Roll the sheet metal with accurate dimension.

for the cylinder of the pump

STEPS;

- I. Prepare hollow round bar with the required dimension.
- II. Bore the round bar by using lathe machine to the required diameter.
- III. Make thread by using lathe machine on the lower part of it.
- IV. And make hexagonal head by using milling machine on the upper part of it.
- V. Finally make the necessary finishing.

For ram cylinder

STEPS;

- I. Make the blank by casting with the nearest dimension that enables as to make the exact dimension bore and turn the dimension by using lathe machine with the required dimension.
- II. Make the groove for O-ring seal with the appropriate depth with by using lathe machine on the top inner surface of the cylinder.
- III. Make precise surface finishing by using surface grinding machine.
- IV. Finally, on the lower part and on the upper part make a thread for thigh ting with the top cup and base plate

For octagonal top cup

STEPS;

- I. By using casting process make the top cup handle which is suitable for machining.
- II. Make some correction the casting for the appropriate dimension.
- III. Next make a thread at the lower and upper inner surface to tighten the cylinder and the reservoir with a metallic thread arrangement part with 1mm pitch by using lathe machine
- IV. Finally make necessary surface finishing process.

O-ring

O - Ring seal made from rubber compound by the process vocalizations.

First make mold for the vocalizations process then, insert the compound on it.

For the reservoir

STEPS;

- I. The reservoir can be manufactured from the sheet metal with the appropriate thickness.
- II. First prepare a sheet metal with appropriate thickness, width and height. Then, roll the sheet metal by using rolling machine.
- III. Next weld the rolled metal using welding machine. then make thread at the lower part and the make groove at the lower and upper inner surface to the required length, width and height.
- IV. Finally make a necessary surface finish

LINK: can be manufactured by using sheet metal with the appropriate length and thickness.

- First cut a sheet metal in a strip then drill at the required position for pin connection.
- Finally bond with the required dimension.

Spring

If spring is of small diameter & the wire diameter is also small, then the spring is normally manufactured by cold drawn processes through mangle. However, for a spring having large coil diameter & large coil diameter one has to go for manufactured by hot process. First one has to heat the wire & then use proper mangle to wind the coil [8].

1.1.9 Assembly and disassembly process

The jack assembly is portable, self-contained unit with a single speed manually operated pump at the lift unit consisting of three rams and on extension screw or cased within an anchored cylinder mounted on the base. A stamped shell enclosed the ram and cylinder to form the fluid reservoir [9].

Assembly procedure

The following steps will help us to assemble the hydraulic bottle jack.

- i. First put all components of HBJ in order pair and identify the dimension of all components.
- ii. Put the base on the on the flat table.
- iii. Assemble the cylinder with the base together with the base plate.
- iv. Assemble the reservoir to base together with sealing part.
- v. Insert O-ring and seal of nut to piston.
- vi. Assemble piston to the piston.
- vii. Assemble nut to piston.

- viii. Insert adjusting screw and adjusting screw socket to piston rod to get ram.
- ix. Assemble O-ring (pump, plunger).
- x. Assemble plunger, spring bolt, short pin, link plate, and swing socket.
- xi. Assemble oil fill plug, pin, release valve seal, ball, and spring with base.
- xii. Install rubber handle holder onto the top nut & slide down ward until it rests upon the reservoir of the jack.
- xiii. Finally assemble the handle section by lining up the spring detent in the rubber grip section with the slot in the other section.

Disassembly procedure

Steps;

1. Clean table (keep the work area clean and well it)
2. Use manuals
3. Open the jack base.
4. Drain the hydraulic oil.
5. Open the bottle jack unit.
6. Safely over load the check valve.
7. Disassemble the plunger.
8. Clean the bottle jack unit passage ways.
9. Release the valves (O-ring).
10. Check valves & safety over load valve.
11. Disassemble the ram seal.

1.2 Problem Statement

Nowadays in this country, most of the cars were equipped with the hydraulic bottle jack. We found that the hydraulic bottle jack was very difficult to be used because this types of jack needed more strength and energy to operate this jack by hydraulic. Thus, we want to develop a product based from the problem faced by the users who drive a car regarding to this issue. To overcome this problem, a research has been conducted to find the solution on how to design a hydraulic scissor jack for the car using the simplest and cheapest way while it is energy saved.

Although there were many ways to solve this problem, we recommend that the design this hydraulic scissor jack system is the practical way when as considered all the factors and consequences especially about the analysis to develop this product. During the research, we have found that most of the car user has difficulties in maintaining their vehicles breakdown especially cars in the scope of changing tires.

In general: First, to minimize the labor force that west when lifting hydraulic in hydraulic bottle jack by changing the system from mechanical to hydraulic Because, mechanically operated scissor jack is difficult for a person who have a hand problem when they went to lift something.

Second position, the most common problem encountered while using mechanical scissor jack is the minimum life time due to familiarity of jack components. This project will have to minimize this by designs hydraulic bottle jack. B/c, the load lifted by the fluid; this fluid additionally used for as lubrication and coolant to protect the material from corrosion and wear between piston and fluid cylinder.

1.3 Objectives

1.3.1 General objective

The main objective of the project is to Design of hydraulic bottle jack.

1.3.2 Specific objective

The project relates to the designing of simple hydraulic scissor jack, which have 15KN carrying capacity with 200 mm lifting height. And its analysis along with structural improvements to make such a modified jack that is very stable and can take enough loads on uneven surfaces and somewhat inclination is also allowed.

The project also aims at designing of extension screw, nut and cup screw, solid and hollow ram, pump cylinder, reservoir, plunger, top cup and base, handle & socket for pump, Handle socket, Handle socket pins, spring at discharge valve, valves, o - ring seals (for pump piston /cylinder, hollow ram& solid ram plunger, releasing screw and refiling screw and finding stresses, efficiency, expected life of lifting members. This project used to develop a hydraulic bottle jack such that it is cost effective, having a long life and can be handled roughly for the following date considering in design process.

CHAPTER II

2. LITERATURE

2.1 General machine design procedure

It is difficult if not possible to lay down any set rules of procedure for designing machinery. This situation encountered are too varied to allow this. However, it is possible to point out a general procedure which will in the majority cases, prove to be help full to the designer. This general procedure can be stated in the form of steps as follow [8].

1. Before starting work on the design of any machine, get thoroughly familiar with what the machine is intended to accomplish and what special requirements must be considered.
2. Make free hand sketches of various ways the machine might be constructed at the same time making any preliminary calculation which might be required to substantiate your ideas or to establish approximate sizes. Having established what seems to be feasible construction, make a lay out drawing of the machine paying particular attention to the necessary details of construction.
3. Analyze the lay out forces, stress etc., & make whatever calculations are necessary to be certain that the parts will perform satisfactorily.
4. Revise the lay out drawing as necessary for the finished design.

2.1.1 Things to be considered while designing

- i. Proper functioning of designed machine.
- ii. Cost.
- iii. Ease of manufacture and assembly.
- iv. Strength and rigidity of parts.
- v. Wear of parts.
- vi. Ease of service and replacing parts.
- vii. Proportion of parts.

2.1.2 Designer should think the following

- A designer thinks on the drawing board or sketch pad. It is impossible to visualize all the details of machine without putting it on paper.
- Sketches should be made large, and enough to picture the construction of the, rather than merely showing a schematic of operation.

- Drawing should be as complete & detailed as possible. It is only being seeing the machine pictured in its true size & shape that the designer can make use of his own judgment. Designing of machine elements involves the fallowing steps;
 - Specifying the problem.
 - Selection of proper mechanism.
 - Analysis of forces.
 - Selection of material.
 - Selection of F.S.
 - Calculation of cross sectional dimensions using basic design equations.
 - Modifying and finalizing dimensions with proper tolerances and preparing drawing with proper instruction for manufacturing [8].

2.2 Selection of hydraulic oil

Hydraulic fluids, also called hydraulic liquids, are the medium by which power is transferred in hydraulic machinery. Common hydraulic fluids are based on mineral oil or water.

Examples; of equipment that might use hydraulic fluids include excavators, brakes, power steering systems, transmissions, backhoes, garbage trucks, aircraft flight control systems and industrial machinery.

Hydraulic systems like the ones mentioned above will work most efficiently if the hydraulic fluid used has low compressibility

A fundamental feature of hydraulic systems is the ability to apply force or torque multiplication in an easy way, independent of the distance between the input and output, without the need for mechanical gears or levers, either by altering the effective areas in two connected cylinders or the effective displacement between a pump and motor. In normal case hydraulic ratios is combined with a mechanical force or torque ratio for optimum machine designs, like in boom movements and track drives for an excavator.

Use only good grade hydraulic jack oil. Avoid mixing different types of fluid and never use brake fluid oil, transmission fluid, motor or glycerin. Improper fluid will cause premature failure of the jack potential for sudden and immediate loss of load. [11].

2.2.1 Function of hydraulic oil & properties

The primary function of a hydraulic fluid is to convey power. In use, however, there are other important functions of hydraulic fluid such as protection of the hydraulic machine components. The table below lists the major functions of a hydraulic fluid and the properties of a fluid that affect its ability to perform that function [9].

Table 3 lists the major functions of a hydraulic fluid

FUNCTION	Property
Medium for power transfer and control	▪ Low <u>compressibility</u> (high bulk modulus) ▪ Fast air release ▪ Low foaming tendency ▪ Low volatility
Medium for heat transfer	▪ Good thermal capacity and conductivity
Sealing Medium	▪ Adequate <u>viscosity</u> and <u>viscosity index</u> ▪ Shear stability
Lubricant	▪ Viscosity for film maintenance ▪ Low temperature fluidity ▪ Thermal and oxidative stability ▪ Hydrolytic stability / water tolerance ▪ Cleanliness and filterability ▪ Demulsibility ▪ Antiwear characteristics ▪ Corrosion control
Pump efficiency	▪ Proper viscosity to minimize internal leakage ▪ High viscosity index
Special function	▪ Fire resistance ▪ Friction modifications ▪ Radiation resistance

Environmental impact	▪ Low toxicity when new or decomposed ▪ Biodegradability
Functioning life	▪ Material compatibility

2.2.2 Composition of hydraulic oil

The original hydraulic fluid, dating back to the time of ancient Egypt, was water. Beginning in the 1920s, mineral oil began to be used more than water as a base stock due to its inherent lubrication properties and ability to be used at temperatures above the boiling point of water. Today most hydraulic fluids are based on mineral oil base stocks.

Nature Brake fluid such as rapeseed (also called canola oil) are used as base stocks for fluids where biodegradability and renewable sources are considered important. Other base stocks are used for specialty applications, such as for fire resistance and extreme temperature applications. Some examples include: glycol, esters, organophosphate ester, polyalphaolefin, propylene glycol, and silicone oils.

Hydraulic fluids can contain a wide range of chemical compounds, including: oils, butanol, esters, (e.g. phthalates, like DEHP, and adipates, like bis (2-ethylhexyl) adipose), polyalkylate glycols (PAG), phosphate esters (e.g. tributyl phosphate), silicones, alkylated aromatic hydrocarbons, polyalphaolefins (PAO) (e.g. poly isobutene), corrosion inhibitors, etc.

Brake fluid is a subtype of hydraulic fluid with *high* boiling point, both when new (specified by the equilibrium boiling point) and after absorption of water vapor (specified by wet boiling point). Under the heat of braking, both free water and water vapor in a braking system can boil into a compressible vapor, resulting in brake failure. Glycol-ether based fluids are hygroscopic, and absorbed moisture will greatly reduce the boiling point over time. Silicone based fluids are not hygroscopic, but their inferior lubrication is not suitable for all braking systems [9].

SAFETY

Because industrial hydraulic systems operate at hundreds to thousands of PSI and temperatures reaching hundreds of degrees Celsius, severe injuries and death can result from component failures and care must always be taken when performing maintenance on hydraulic systems [8].

2.2.3 Biodegradable hydraulic fluids

Environmentally sensitive applications (e.g. farm tractors and marine dredging) may benefit from using bio degradable hydraulic fluids based up on rapeseed (Canola) vegetable oil when there is the risk of an oil spill from a ruptured oil line. Typically these oils are available as ISO 32, ISO 46, and ISO 68 specification oils. ASTM standards ASTM-D-6006, Guide for Assessing Biodegradability of Hydraulic Fluids and ASTM-D-6046, Standard Classification of Hydraulic Fluids for Environmental Impact are relevant [12].

2.2.4 Selection hydraulic oil for hydraulic bottle jack

A hydraulic bottle jack uses a fluid which is incompressible, that is forced in a cylinder by a pump plunger. Oil is used since it is self-lubricating and stable.

When a plunger pulls back, it draws oil out of the reservoir through the suction check valve in to the chamber. When the plunger moves fore ward, it pushes the oil through the discharge check valve into the cylinder.

The suction valve ball is within the chamber and opens with each draw of the plunger. The discharge valve ball is outside the chamber and opens, when the oil is pushed in to the cylinder. At this point the suction ball within the chamber is forced and oil pressure builds in the cylinder [5] [9].

Due to the above function and high viscosity high temperature and selected SDT fluid is ATF (Automatic Transmission Fluid) and designation is AES NO 10, Part number H008& MOBIL DTE 13

GENERALLY HYDRAULIC OIL IS;

- Used to reduce the friction between the milling surfaces.
- Used to dissipate the heat developed inside the system.
- Used to clean the hydraulic part.
- It prevents parts from corrosion

2.2.5 How to add oil

- Position the jack on level ground in an upright (vertical) position & fully lower the saddle.
- Remove filler plug.
- Oil should be filled to the level of the oil filler plug hole. If low add oil as needed.

REPLACING OIL

- Plug hole to drain oil, remove oil filler plug & open release valve.
- Turn jack on its side & drain old oil out through the oil filler [13].



Figure 6 shows how to fill oil.

CHAPTER III

3. METODOLOGY

3.1 Input parameter

- ❖ Lifting Distance; =20cm = 200mm
 - ✓ Maximum lifting height; =360mm
 - ✓ Minimum lifting height; =160mm
- ❖ Load capacity; =15KN
- ❖ Driving mechanism; HYDRAULIC

In order to get numbers of rams, thickness of neck;

$$A) \text{ Number of rams} = \frac{H_{\max} - H_{\min}}{H_{\min}}$$

$$NR = \frac{(360\text{mm} - 160\text{mm})}{160\text{mm}}$$

$$NR = 1.25$$

Since N_R should be whole number then, **NR=2**

THICKNESS OF NECK

$$\text{But, } y = \frac{(H_{\max} - H_{\min})}{2} = \frac{(360\text{mm} - 160\text{mm})}{2}$$

$$y = 100\text{mm}$$

Since $H_{\min} = 100\text{mm}$, therefore $l = y + 2t$

$$t = \frac{(H_{\min} - y)}{2}$$

$$t = \frac{(160\text{mm} - 100\text{mm})}{2} = 30\text{mm}$$

CHECKING

$$H_{\max} = H_{\min} + 2y$$

$$= 160\text{mm} + 2(100\text{mm}) = H_{\max} = 360\text{mm} \dots\dots\dots \text{Therefore, the analysis is correct.}$$

3.2 Force analysis FOR JACK HANDLE AND PUMP FORCE

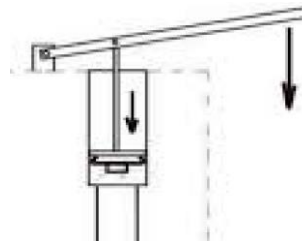


Figure 7 shows mechanisms of jack handle and pump force

Where

- ✚ F_p =pin force
- ✚ F_H =human force (usually it is taken to be (100 to 200))
- ✚ F_F =fulcrum force
- ✚ L =overall handle length (again most of the time to be taken (300-500mm))
- ✚ A =pin to pin length (it also taken (30-50mm))

Using equilibrium condition;

$$\curvearrowright \Sigma M_A = 0$$

$$(F_H \times L) - (F_F \times a) = 0$$

$$F_F = \frac{F_H \times L}{a} \dots \dots \dots (1)$$

$$2) \quad \Sigma F_y = 0$$

$$F_F - F_p - F_H = 0$$

$$F_p = F_F - F_H \dots \dots \dots (2)$$

*Substitute equation (1) into (2) we have;

$$F_p = F_F (1 - L/a) \dots \dots \dots (3)$$

3.3 Pressure distribution

By using Pascal's principles, that is intensity of pressure is transmitted equally in a direction through a mass of fluid at a plunger.

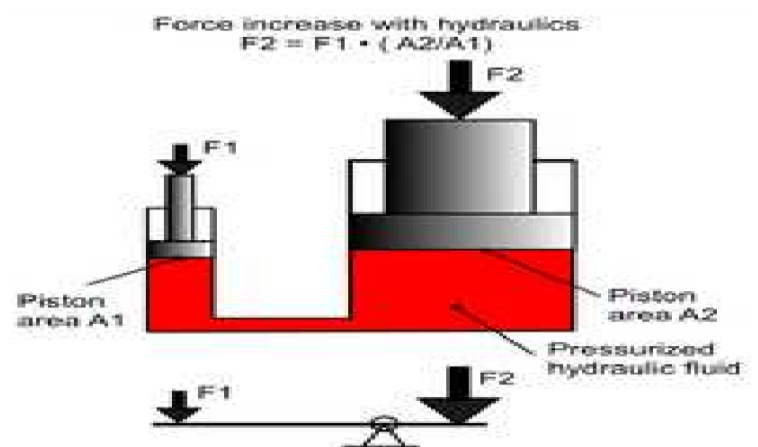


Figure 8 hydraulic principle

A force F_P is applied to the plunger and a force F_R is applied to the rams. Since the force at two cylinders will be proportional to their respective cross-sectional areas. Pressure is the same everywhere it is easy to solve the force applied on ram and plunger

Therefore $F = PA = P(\pi r^2)$

But $P_{ram} = P_p$Pascal's principle

$$\Leftrightarrow \frac{F_{ram}}{A_{ram}} = \frac{F_{plunge}}{A_{plunge}} \quad \text{where,}$$

$$A_{ram} = \pi d_r^2 / 4$$

D_R = diameter of the ram

$$A_P = \frac{\pi d_p^2}{4}$$

D_p =diameter of the plunger

$$\text{Since, } F_R / A_R = F_P / A_P \Rightarrow \frac{F_R}{\frac{\pi d_r^2}{4}} = \frac{F_P}{\frac{\pi d_p^2}{4}}$$

$$F_R / d_r^2 = F_P / d_p^2$$

$$F_R = (F_P / d_p^2) * d_r^2 \dots\dots\dots(4)$$

$$\text{But, } A_P = \frac{\pi D_P \times D_P}{4} \dots\dots\dots \text{cylinder bore area (circular area)}$$

D_P plunger diameter varies (10 to 16mm) it is standard.

From this range let use $D_P = 12\text{mm}$

And, therefore $A_p = \frac{\pi D_p \times D_p}{4}$

And pressure of the pump is given by,

$P_{\text{pump}} = \frac{F_p}{A_p}$ where, A_p plunge cross-sectional area and

P_{pump} pump pressure

This implies that $P_{\text{pump}} = \frac{F_p}{A_p}$ (5)

* Using Pascal's law,

$P_{\text{pump}} = P_{\text{PLUNGE}} = P_{\text{ram}} = P_{\text{CY}}$

Therefore, $P_p = P_c$ $P_c = F_c / A_c$ where P_c =cylinder pressure

A_c =cylinder cross-sectional area

F_c =cylinder force

$F_c / A_c = (\text{N/mm}^2) \text{ or Mpa}..... (5)$

3.4 Pressure range selection

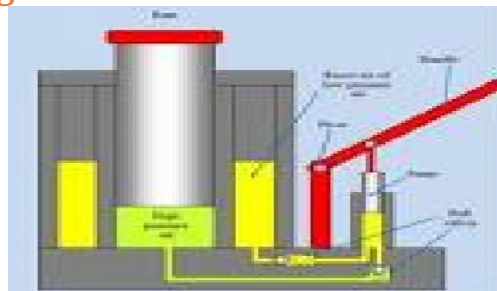


Figure 9 pressure range with handle

Assumption: assume that the internal cylinder diameter and the hollow ram diameter are equal. Therefore, $D_{\text{CR}} = D_{\text{HR}}$

Where D_{CI} = internal cylinder diameter

D_{HR} =hollow ram diameter

Let assume that $D_{\text{HR}} = D_{\text{CI}} = 60\text{mm}$

Therefore, $A_c = D_{\text{HR}}^2 / 4$

$A_c = \pi \times 60^2 / 4 = 2827.43\text{mm}^2$

Hence the minimum pressure to raise the given weight is:

$$P_{CMIN} = \frac{F}{A_c} \quad \text{where: } F = 15\text{KN} \dots\dots\dots \text{ is the given design force, } P_{CMIN} = \frac{15 \times 10^3 \text{N}}{2827.43 \text{mm}^2} = 5.3052 \text{MPa}$$

Therefore

$P_{CMIN} = 5.3052 \text{ MPa}$, the internal pressure to be designed should be greater than the minimum pressure since there are parts to be lifted with this pressure.

Hence $P_{CDESIGN} > P_{c \min} = 5.3052 \text{MPa}$

In order to find the internal design pressure, length of the handle, force of fulcrum, plunge force and pin to pin length first let us assume the human force and length range for handle.

Assuming $F_h = 200\text{N}$ from the range of (100N to 200N)

And $L = (300\text{mm to } 500\text{mm})$, assuming, $L = 450\text{mm}$

$a = 30\text{mm to } 50\text{mm}$, and also assume, $a = 40\text{mm}$

By using equation (1) to equation (2)

$$F_F = \frac{F_H * L}{a}$$

$$F_F = \frac{200 * 450}{40}$$

$$F_F = 2250 \text{KN}$$

$$F_P = F_F - F_H$$

$$F_P = (2250 - 200) \text{ KN}$$

$$F_P = 2050 \text{KN}$$

$$P_C = \frac{F_P}{A} = \frac{(2050) * 4}{3.14 * 12 * 12} \dots\dots\dots \text{ where } A_P \text{ is area of the plunge}$$

$$P_C = 18.125 \text{MPa}$$

Finally, from the above, $P_{cylinder}$ must be greater than $P_{c \min} = 5.3052 \text{MPa}$. For this reason, the selected quantities are: -

$$L = 450 \text{ mm}, \quad F_f = 2250 \text{N}, \quad F_p = 2050 \text{N}$$

$$a = 40 \text{mm}, \quad P_c = 18.125 \text{MPa}$$

3.5 Stress analysis

3.5.1 Design of extension screw



Figure 10 screw

We have selected square thread screw. Because of the following reason;

- ✓ They can easily manufactured
- ✓ They have high efficiency.
- ✓ Widely used for transmission of power in either direction.
- ✓ They offer less frictional resistance to motion

STEP 1) CALCULATE ALL DIAMETER, PITCH, DEPTH OF THE TREAD

$S_y = 480 \text{ Mpa}$ yield strength..... from the range of (430-585)

$F = 15 \text{ KN}$given load

By taking, $\sigma_{all} = 0.65 * S_y = 312 \text{ Mpa}$

$\sigma_{all} = 480 \text{ Mpa} / 1.5 = 320 \text{ MPa}$ FS=1.5

To get core diameter (dc) of the screw

$$\Sigma_{all} = \frac{4W}{\pi dc^2} = \frac{F}{A} \quad \dots\dots\dots F=15 \text{ KN}$$

$$dc = \sqrt{\left(\frac{4F}{\pi \sigma_{all}} \right)}$$

$$dc = \sqrt{\left(\frac{4 * 15000}{\pi * 312 * 10^6} \right)} = dc = 7.8 \text{ mm}$$

* To be more safe let us take **dc = 10mm**

* Pitch of the thread (p) = $\frac{dc}{4}$

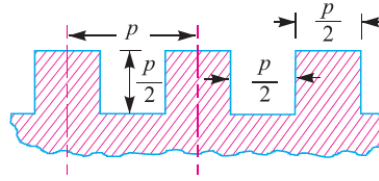


Figure 11 screw analysis

$$p = \frac{10\text{mm}}{4}$$

$$p = 2.5\text{mm}$$

$$h = \frac{p}{2} = 1.25\text{mm} \dots \dots \dots \text{depth of screw thread}$$

$$d_b = p + dc = 10\text{mm} + 2.5\text{mm} = 12.5\text{mm}, \text{ where } d_b = \text{diameter of the screw}$$

$$d_n = d_b + 2(a) \dots \dots \dots \text{where } a = 0.25 \text{ constant}$$

$$d_n = 12.5\text{mm} + 2*(0.25\text{mm}) = 13\text{mm}$$

$$H = 0.5P + a, \text{ where } H = \text{depth of the nut thread}$$

$$H = 0.5 (2.5) + 0.25\text{mm} = \mathbf{1.5\text{mm}}$$

STEP2) DETERMINE CORE AREA OF THE SCREW AND MEAN DIAMETER, HELIX ANGLE, ANGLE OF FRICTION AND FORCE NEEDED TO RAISE THE LOAD.

$$A_c = \frac{\pi d_c^2}{4} = \dots \dots \dots \text{core area of the screw.}$$

$$A_c = \frac{\pi*(10\text{mm})^2}{4} = \mathbf{78.5\text{mm}^2}$$

Mean diameter of the screw (d_m)

$$D_M = \frac{d_b + dc}{2}$$

$$D_M = \frac{12.5\text{mm} + 10\text{mm}}{2}$$

$$\mathbf{D_M = 11.25\text{mm}}$$

Helix angle (α):

$$= \tan \alpha = \frac{p}{\pi dm}$$

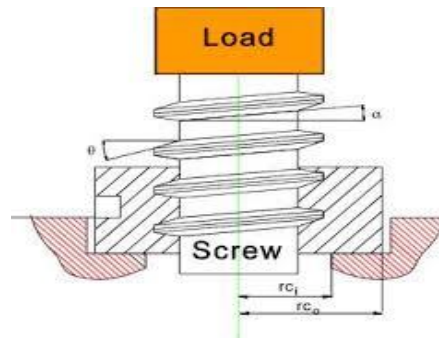


Figure 12 screw with load

$$\alpha = \tan^{-1} \frac{p}{\pi dm}$$

$$\alpha = \tan^{-1} \frac{2.5mm}{\pi 11.25mm}$$

$$\alpha = 4.05^\circ$$

Frictional angle (β)

$$\tan \beta = \mu$$

Where (μ) is the coefficient of friction between the thread is equal to 0.125 (for carbon alloy steel).

$$\beta = \tan^{-1} \mu$$

$$\beta = \tan^{-1} (0.125) = 7.125^\circ$$

When the frictional angle (β) is greater than the helix angle (α) the arrangement is self – locking. $\beta > \alpha$

Therefore, this arrangement is self –locking.

** The force required raise the load is given by;

$$W = F (\tan \alpha + \beta)$$

$$W = 15KN [(0.0707 + 7.125)]$$

$$W = 107.9355KN$$

There for the force required raise the load of 15KN is; $W = 107.9355KN$

STEP3) DETERMINE FRICTIONAL TORQUE

FRICTIONAL TORQUE

The torque which resists the frictional force is called frictional torque.

Therefore, thread surface frictional torque;

$$T_f = \frac{F \cdot d_m \cdot \tan(\alpha + \beta)}{2} = \frac{15000 \cdot 11.25 \cdot \tan(4.05 + 7.125)}{2}$$

$$T_f = 16668.44 \text{ Nmm}$$

STEP4) COMPARE WITH DIRECT MAXIMUM AND SHEAR STRESS

Now let us check the screw for the principal stress on it

$$\text{Direct compressive stress} = \frac{4 \cdot F}{\pi d_c^2}$$

$$\sigma_{c \text{ direct}} = \frac{4 \cdot 15000}{\pi 10^2} = 190.98 \text{ MPa}$$

Maximum torsional shear stress

$$\tau_{\text{Max}} = 16 T_f / \pi d_c^3$$

$$\tau_{\text{Max}} = 16 \cdot 16668.44 / \pi 10^3$$

$$\tau_{\text{Max}} = 84.89 \text{ MPa}$$

$$\text{Maximum principal stress} = \frac{\sigma_c + \sqrt{\sigma_c^2 + 4\tau^2}}{2}$$

$$\sigma_{\text{Max}} = \frac{191 \text{ MPa} + \sqrt{(191 \text{ MPa})^2 + 4(84.89 \text{ MPa})^2}}{2}$$

$$\sigma_{\text{Max}} = 223.3 \text{ MPa}$$

Since σ_{max} is much less than $\sigma_{\text{all}} = 321.3 \text{ MPa}$

That is, $\sigma_{\text{all}} > \sigma_{\text{max}}$Therefore, the screw is safe from crushing.

With efficiency of;

$$\text{Efficiency } (\eta) = \frac{1 - \sin \beta}{1 + \sin \beta}$$

$$= \frac{1 - \sin 7.125}{1 + \sin 7.125}$$

$$\eta = 0.779 = 77.9\%$$

STEP4) FINALLY CHECK FOR BUCKLING

Rankin's formula is applicable for columns, ranging from very long to short ones for this reason WE use this formula to check buckling.

$$\text{Crippling load} = \frac{\text{crushing load}}{1 + a\left(\frac{L}{K}\right)^2}$$

$$\text{Therefore, } W_{CR} = \frac{\sigma_c \cdot A}{1 + a\left(\frac{L}{K}\right)^2} \quad \text{where } \sigma_c = \text{crushing stress or yield stress in compression.}$$

A = cross-sectional area of the column

a = Rankine constant

L = Equivalent length of the column

K = radius of gyration.

$$\text{Radius of gyration (K)} = \sqrt{\frac{I}{A}}$$

$$K = \sqrt{\frac{(\pi d_c^4 / 64)}{\frac{\pi d_c^2}{4}}}$$

$$K = \sqrt{(d_c^2) / 16}$$

$$K = d_c / 4 = 10 / 4 = 2.5 \text{ mm}$$

Let us assume the effective length L is = 80 mm

Since a = 1/9000

$$\text{From; } a = \frac{\sigma_c}{\pi^2 E}$$

$$E = \frac{\sigma_c}{\pi^2 a} = \frac{191 \text{ MPa}}{\pi^2 \left(\frac{1}{9000}\right)}$$

$$E = 174 \text{ GPa}$$

$$W = \pi^2 EI / L^2 \dots\dots\dots \text{Where}$$

$$I = K^2 A = 2.5^2 (78.5) \text{ mm}$$

$$I = 490.625 \text{ mm}^4$$

$$W = \pi^2 174 \text{ GPa} (490.625) / 80^2$$

$$W_{CR} = 131.2 \text{ kN}$$

$$\text{Permissible buckling load} = \frac{\text{rankine load}}{\text{safety factor}}$$

$$P_B = W_{CR} / F. S = 131.2 / 2 \text{ kN}$$

$$= 65.6 \text{ kN}$$

CONCLUSION

Since our load which is 15KN is much less than permissible buckling load, this screw is safe from buckling

$$P_B > F$$

3.5.2 Design of nut and cup screw

DESIGN OF NUT

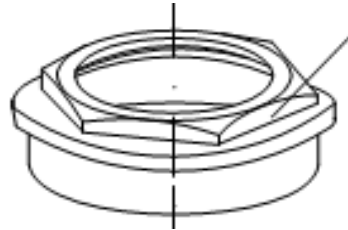


Figure 13 nut

to find number of thread in the nut;

Assume bearing pressure is $P = 12 \text{ MPa}$ from the range of (10 TO 15) MPa.

$$\text{From } W = Nt * \frac{Pb\pi(D^2 - D_c^2)}{4}$$

$$Nt = 4 * \frac{F}{Pb * \pi(D^2 - D_c^2)}$$

Were $F = 20 \text{ KN}$

$$D = 11.25 \text{ mm}$$

$$D_c = 10 \text{ mm}$$

$$P = 12 \text{ MPa}$$

Therefore;

$$Nt = 4 * \frac{15000 \text{ N}}{12 \text{ MPa} * \pi(11.25^2 - 10^2)}$$

$$Nt = 59.9$$

Height of the nut is;

$$H = N_T * P = 59.9 * 2.5 \dots \dots \dots P = \text{Pitch}$$

$$H = 149.8 \text{ mm}$$

The outer diameter of the nut

$$D_N = 1.5d_c$$

$$D_N = 1.5 \times 10\text{mm}$$

$$D_N = 15\text{mm}$$

Say $D_N = 20\text{mm}$

Design of cup

Maximum diameter of the cup = $2 \times d_c$

$$D_c = 2 \times 10\text{mm} = 20\text{mm}$$

Let us assume that thickness of the cup

$$t = 15\text{mm}$$

Shear area of the cup can be calculated as follows

Area that resists shearing of the cup = $\pi d_b \times T_c$

$$= \pi \times 11.75\text{mm} \times 15\text{mm} = 553.4\text{mm}^2$$

The induced shear stress: -

$$\tau_{\text{Indused}} = \frac{W}{A_{\text{cup}}}$$

$$\tau = \frac{15000\text{N}}{553.4\text{mm}^2} = \tau = 27.105\text{MPa}$$

CONCLUSION

Since the induced shear stress is much less than the allowable shear stress.

$$\text{I.e. } \tau_{\text{all}} > \tau_{\text{indused}}$$

3.5.3 Design of solid and hollow ram

DESIGN OF SOLLID RAM

Because of it is subjected compressive load, fluid friction and corrosion, the selected material is should be: -

Alloy steel, AISI1015, $\sigma_u = 420.6\text{MPa}$, $S_y = 313.7\text{Mpa}$

$$\sigma =, S_y/2 = 313.7/2\text{Mpa} = 156.85$$

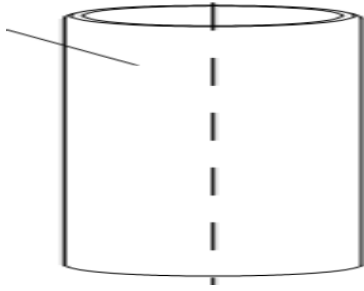


Figure 14 solid ram

PARAMETERS

$$D_c = D_R = 60 \text{ mm}$$

$$P_c = 18.125 \text{ MPa}$$

$$F = 15 \text{ kN}$$

CHECK FOR SHEARING AND CRUSHING STRESS

Since the pressure applied on the bottom of the solid ram when it hits the stop of the second stage ram subjects the neck for shearing and crushing, we have to check for;

Maximum force acting on the neck

$$F_{\text{at neck}} = P_{\text{cylinder}} * A_c, \text{ where } A_c = \frac{\pi}{4} d_c^2$$

$$A_c = \frac{\pi}{4} 60^2$$

$$A_c = 2827.33 \text{ mm}^2$$

$$F_{\text{at neck}} = 18.125 \text{ MPa} * 2827.33 \text{ mm}^2$$

$$F_{\text{at neck}} = 51.25 \text{ kN}$$

The induced shear force (T_{induced})

$$\tau = \frac{F_{\text{at neck}}}{A_{\text{shear}}}, \quad \text{but } A_{\text{(shear)}} = \pi D_i * l$$

$$A_{\text{(shear)}} = \pi 20 \text{ mm} * 15 \text{ mm} = 942 \text{ mm}^2$$

$$\tau = \frac{51.25 \text{ kN}}{942 \text{ mm}^2}$$

$$\tau = 54.4 \text{ MPa}$$

CONCLUSION

Since $\tau_{\text{all}} > \tau_{\text{induced}}$ then this ram is safe from shearing

$$\tau_{\text{All}} = 0.5 * 313.7 = 156.85 \text{ MPa}$$

The induced crushing stress (σ_c)

$$\sigma_c = \frac{F@neck}{A_{crushing}} \quad \text{but } A_{(crushing)} = \frac{\pi}{4}(D_o^2 - D_i^2)$$

Where D_o = outer diameter of the ram

D_i = inner diameter of the ram

$$A_{(crushing)} = \frac{\pi}{4}(40^2 - 20^2) = 1225 \text{ mm}^2$$

$$\sigma_c = \frac{51.25 \text{ kN}}{1225} = 41.83 \text{ MPa}$$

CONCLUSION

Since $\sigma_{all} > \sigma_c$ have our ram is save from crashing

CHECK FOR BUCKING

Again using Rankine formula, we can check our solid ram from bucking

$$W_{CR} = \frac{\sigma_c A}{1 + Q \left(\frac{L}{K}\right)^2}$$

Where W_{CR} = crippling load

σ_c = Crushing stress on field stress on compression

A = Cross –sectional area of the column

Q = Rankine constant

L = equivalent length of the column

K = least radius of gyration

For carbon steel

$$\sigma_c = 250 \text{ Mpa and fixture coefficient} = \frac{1}{0.25}$$

$$\text{Rankine constant} = \frac{1}{9000}$$

Rankine constant for this ram will be:

$$Q = \frac{1}{9000} * \frac{1}{0.25} = 0.00044$$

And the equivalent length (L) = 100mm

$$\text{Radius of gyration (K)} = \sqrt{\frac{I}{A}}$$

$$K = \sqrt{\frac{(\pi d_i^4 / 64)}{\frac{\pi d_i^4}{4}}}$$

$$K = \sqrt{(d_i^2)/4}$$

$$K = \frac{d_i}{4} = \frac{40}{4} = 10\text{mm}$$

$$\left[\frac{L}{K}\right]^2 = \left[\frac{95\text{mm}}{10}\right]^2 = 90.25\text{mm}$$

Finally, the crippling load

$$W_{CR} = \frac{\sigma_c A}{1 + Q\left(\frac{L}{K}\right)^2} \quad \text{but } A_c = \pi D_i^2 / 4 = \pi 40^2 / 4 = 1256.64\text{mm}^2$$

$$W_{CR} = \frac{\sigma_c A}{1 + Q\left(\frac{L}{K}\right)^2} = \frac{250 \times 1256.64}{1 + 0.00044 \times 90.25^2}$$

$$W_{CR} = 75.6\text{KN}$$

$$\text{Permissible buckling load} = \frac{\text{Rankine load}}{S.f}$$

$$W_{CR} = \frac{W_{cr}}{S.f} = W_{CR} = \frac{75.6}{3} = 24.2\text{KN}$$

CONCLUSION

Since the permissible buckling load is much greater than the given load i.e $W_{CR} > F = 15\text{KN}$, hence the ram is safe from buckling

Design of hollow ram

Here it is also subjected to fluid friction, compressive load, buckling and corrosion and also from economic point of view, the selected material is: -

Milled steel ASIM, A487 and $S_y = 90\text{MPa}$

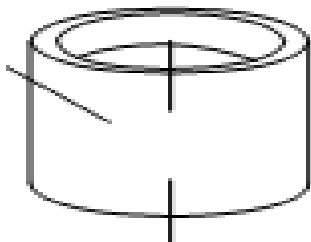


Figure 15 hollow RAM

Material; Mild steel

$$S_Y = 90\text{MPa}$$

$$\tau_{\text{All}} = 0.5 * S_Y = 0.5 * 90\text{MPa}$$

$$= 45\text{MPa}$$

$$\sigma_{\text{All}} = 0.65 * S_Y = 0.65 * 90\text{MPa} = 58.5 \text{ MPa}$$

Due to given load, let assuming the hollow ram is thin walled cylinder.

Therefore, from hoop stress (σ_h)

$$\sigma_h = \frac{P_i * D_i}{2 t} \quad \text{where } P_i = \text{internal pressure.}$$

D_i = internal diameter

t = hollow ram thickness

C_f = corrosion factor

Therefore, $P_i = 5.3052\text{MPa}$

$$D_i = 40\text{mm}$$

Let us assume that

$$\sigma_h = \sigma_{\text{all}} = 58.5\text{MPa}$$

$$\text{Therefore, } \sigma_h = \frac{P_i * D_i}{2 t}$$

$$t = \frac{P_i * D_i}{2 \sigma_h}$$

$$t = \frac{5.3052 * 40}{2 * 58.5}$$

$$\mathbf{t = 2.721\text{mm}}$$

Due to corrosion effect we have to add corrosion factor $C_f = 1.66\text{mm}$

$$\text{Therefore, } t_{\text{actual}} = 2.721\text{mm} + 1.66\text{mm}$$

$$\mathbf{T_{\text{actual}} = 4.381\text{mm}}$$

Assuming stop length (L) = **10mm**

$$\text{And } D_o = D_i + 2t = 40\text{mm} + (2 * 4.381)$$

$$\mathbf{D_o = 68.76\text{mm}}$$

Most outer diameter of the ram becomes: -

$$D_{MO} = D_O + 2(L) = 68.76 + 20 = 88.76\text{mm}$$

CHECK FOR SHEAR AND CRUSHING STRESS AT THE STOP (NECK)

Maximum force acting on the neck (stop)

$$F_{\text{at neck}} = P_c * A_c \quad \text{but,} \quad A_c = \frac{\pi * D_{MO}^2}{4}$$

$$A_c = \frac{\pi * 90.56^2}{4} = 6446.8\text{mm}^2$$

$$\text{Therefore, } F_{\text{at neck}} = 5.3052\text{MPa} * 6446.8\text{mm}^2 = 34202\text{N}$$

Induced shear force at neck (stop) is:

$$\tau = \frac{F_{\text{at neck}}}{A_{\text{shear}}} \quad \text{where } A_{\text{shear}} = \pi * D_O * h$$

$$A_{\text{shear}} = \pi * 68.76\text{mm} * 10\text{mm}$$

$$A_{\text{shear}} = 2161\text{mm}^2$$

$$\tau = \frac{34202\text{N}}{2161\text{mm}^2} = 15.83\text{MPa}$$

Crushing stress at neck (stop)

$$\sigma_c = \frac{F_{\text{at neck}}}{A_{\text{crush}}} \quad \text{where}$$

$$A_{\text{crush}} = \frac{\pi}{4}(D_{MO}^2 - D_O^2)$$

$$= \frac{\pi}{4}(88.76^2 - 68.76^2) \text{mm}^2$$

$$A_{\text{crush}} = 2474.3\text{mm}^2$$

$$\text{Therefore, } \sigma_c = \frac{34202\text{N}}{2474.3\text{mm}^2} = \sigma_c = 13.82\text{MPa}$$

CONCLUSION

Since the induced shear of the crushing stress are much less than the allowable shear of normal stress, our hollow ram is safe from this induced stress. i.e,

$$\sigma_c = 13.82\text{MPa} < \sigma_{\text{all}} = 58.5\text{MPa} = \tau = 15.83\text{MPa} < \tau_{\text{all}} = 45\text{MPa}$$

CHECK FOR BUCKLING

Here again the Rankin formula is applicable to check buckling. That is;

$$W_{\text{CRP}} = \frac{\sigma_c A}{1 + Q\left(\frac{L}{k}\right)^2} \quad \text{where } W_{\text{CR}} = \text{cripling load}$$

$\sigma_c = 320\text{MPa}$ for mild steel crushing stress

$$Q = \frac{\sigma_c}{C\pi^2 E} = \frac{1}{0.25} * \frac{1}{7500} = 0.00053 \dots \dots \dots \text{rankine constant}$$

$L = 100\text{mm}$ effective length

$$\text{Radius of gyration (K)} = \sqrt{\frac{I}{A}}$$

$$K = \sqrt{\frac{(\pi(D_{Mo}^4 - D_o^4)/64)}{\frac{\pi(D_{Mo}^2 - D_o^2)}{4}}}$$

$$K = \sqrt{\frac{D_{Mo}^2 - D_o^2}{16}}$$

$$K = \sqrt{\frac{88.76^2 - 68.76^2}{16}}$$

$$K = 14.5\text{mm}$$

$$\left[\frac{L}{K}\right]^2 = \left[\frac{100\text{mm}}{14.5}\right]^2 = 47.56$$

$$\text{And Area (A)} = \frac{\pi}{4} (DM^2 - DO^2)$$

$$A = \frac{\pi}{4} (88.76^2 - 68.76^2)$$

$$A = 2474.32\text{mm}^2$$

$$W_{CRP} = \frac{320\text{MPa} * 2474.32\text{mm}^2}{1 + 0.00053 * 47.56}$$

$$W_{CRP} = 772.3\text{KN}$$

$$\text{Permissible buckling} = \frac{\text{rankine load}}{S.f}$$

$$W_{CRP} = \frac{772.3\text{N}}{4} \dots \dots \dots \text{SF}=3$$

$$W_{CRP} = 193.1\text{KN}$$

CONCLUSION

HOLLOW RAM is also safe from buckling. SINCE $W_{CRP} > F$

3.5.4 Design of pump cylinder

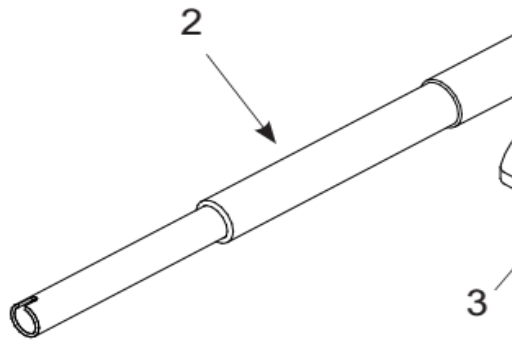


Figure 16 pump cylinder

Hydraulic pump supply fluid to the components in the system, Pressure in the system develops in reaction to the load.

Just like hollow ram it is subjected to fluid friction, compressive load tensile load, buckling and corrosion and also from economical point of view, the selected material is:

Milled steel ASTM, A487, $S_y=90\text{MPa}$

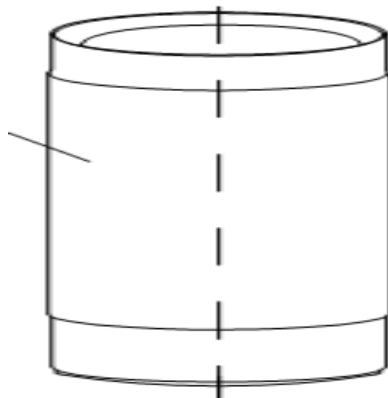


Figure 17 hollow RAM 2

Material: mild steel

$S_Y = 90\text{MPa}$.

Assume $S.F = 2$

$\tau_{All} = 0.5 * S_Y = 45\text{MPa}$

$\sigma_{All} = 0.65 * S_Y = 58.5\text{ MPa}$

THICKNESS OF THE WALL

Again for the same reason which is less internal pressure applied on the cylinder, we can treat as thin walled cylinder.

Therefore, from hoop stress (σ_h)

$$\sigma_h = \frac{P_c \cdot D_i}{2t} \quad \text{where } P_c = 18.125 \text{ MPa} \dots \dots \dots \text{cylinder pressure}$$

$$\Rightarrow t = \frac{P_c \cdot D_i}{2\sigma_h} \quad D_i = 12 \text{ mm} \dots \dots \dots \text{internal diameter of the cylinder}$$

$$\sigma_h = \sigma_{\text{all}} = 58.5 \text{ MPa} \dots \dots \dots \text{assumption}$$

$$\text{Therefore, } t = \frac{18.125 \text{ MPa} \cdot 12 \text{ mm}}{2 \cdot 58.5 \text{ MPa}} = \mathbf{1.86 \text{ mm}}$$

To be more safe let us take corrosion factor $C_F = 1.44 \text{ mm}$

$$T_{\text{actual}} = t + C_f$$

$$T_{\text{actual}} = 1.86 \text{ mm} + 1.44 \text{ mm}$$

$$T_{\text{actual}} = 3.3$$

Therefore, the outer diameter of the cylinder becomes

$$D_o = D_i + 2(t_{\text{actual}})$$

$$D_o = 12 \text{ mm} + 2 \cdot 3.3 \text{ mm} = \mathbf{D_o = 18.6 \text{ mm}}$$

II) CHECK FOR BUCKLING

Therefore, for mild steel;

$$\sigma_c = 320 \text{ MPa} \dots \dots \dots \text{crushing stress.}$$

$$Q = \frac{\sigma_c}{C \pi^2 E} = \frac{1}{0.25} \cdot \frac{1}{7500} = 0.00053 \quad \text{where } C = \text{the end fixture coefficient}$$

$$Q = \text{Rankine constants}$$

Crippling load

$$W_{\text{CR}} = \frac{\sigma_c A}{1 + Q \left(\frac{L}{k}\right)^2} \quad \text{where } \sigma_c = 320 \text{ MPa}$$

$$A = \frac{\pi}{4} (D_o^2 - D_i^2) \dots \dots \dots \text{cross-sectional area}$$

$$A = \frac{\pi}{4} (18.6^2 - 12^2)$$

$$A = 158.5 \text{ mm}^2$$

$$Q = 0.00053$$

L = effective length of the cylinder

$$K = \sqrt{\frac{I}{A}}$$

$$K = \sqrt{\frac{(\pi(D_0^4 - D_i^4)/64)}{\frac{\pi(D_0^2 - D_i^2)}{4}}}$$

$$K = \sqrt{\frac{D_0^2 - D_i^2}{16}}$$

$$K = \sqrt{\frac{18.6^2 - 12^2}{16}}$$

$$K = 3.55\text{mm}$$

$$\left[\frac{L}{K}\right]^2 = \left[\frac{60\text{mm}}{3.55\text{mm}}\right]^2 = 293.5$$

$$W_{CR} = \frac{\sigma_c A}{1 + Q\left(\frac{L}{K}\right)^2}$$

$$W_{CR} = \frac{320 \times 158.5}{1 + 0.00053 \times 293.8}$$

$$W_{CR} = 43.2\text{KN}$$

$$\text{Permissible buckling load} = \frac{\text{buckling load}}{S.f}$$

$$W_{CR} = \frac{43.4\text{KN}}{3} \dots W_{CR} = 14.6\text{KN} \dots \text{SF}=3$$

CONCLUSION

Since the permissible buckling load is greater than *the fulcrum* load ($F_F=2.25\text{kN}$) our PUMP cylinder is safe from buckling.

$$W_{CR} = 15.46\text{KN} > F_F=2.25\text{kN}$$

3.5.5 Design of reservoir

The hydraulic fluid reservoir holds excess hydraulic fluid to accommodate volume changes from: cylinder extension and contraction, temperature driven expansion and contraction, and leaks. The reservoir is also designed to aid in separation of air from the fluid and also work as a heat accumulator to cover losses in the system when peak power is used. Design engineers are always pressured to reduce the size of hydraulic reservoirs, while equipment operators always appreciate larger reservoirs. Reservoirs can also help separate dirt and other particulate from the oil, as the particulate will generally settle to the bottom of the tank.

Some designs include dynamic flow channels on the fluids return path that allow for a smaller reservoir.

It is all the same character to that of cylinder and from economical point of view and also availability, the selected material is:

Milled steel ASTM, A487 and $S_y = 90\text{MPa}$

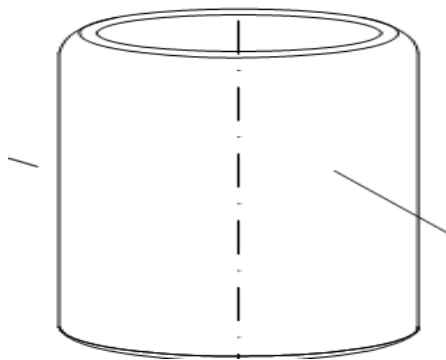


Figure 18 shows reservoir

Material: mild steel

$S_y = 90\text{ MPa}$

$\sigma_{ALL} = 0.65 * S_F = 0.65 * 90\text{MPa} = 58.5\text{MPa}$

Volume determination

The volume of the fluid occupied by the ram and the cylinder should be equal to that of the reservoir volume.

I.e ram volume (V_R) = $V_1 + V_2$

Where $V_1 = A_1 L_1$

$$= \frac{\pi}{4} (D_1^2) * L_1 \dots\dots\dots L=65\text{mm}$$

$$= \frac{\pi}{4} 60^2 * 65$$

$$V_1 = 183783.17\text{ mm}^3$$

$$V_2 = A_2 L_2 \dots\dots\dots L_1 = L_2$$

$$V_2 = \frac{\pi}{4} (D_2)^2 * L = \frac{\pi}{4} * 88.76^2 * 65 = 402,196\text{ mm}^3$$

Therefore $V_{RAM} = V_1 + V_2$

$$= 183783.17 + 402196\text{ mm}^3$$

$$V=585,979.3 \text{ mm}^3$$

Inner diameter of the reservoir

In order to carry this amount of volume of fluid in the height of the reservoirs volume should be equal to the volume of the rams.

$$\text{i.e. } V_{\text{RAM}} = V_{\text{RESERVOIR}}$$

$$\text{Therefore, } V_{\text{RESERVOIR}} = \frac{\pi}{4}(D_{\text{inner of reservoir}}^2 - D_{\text{outer of the cylinder}}^2) * \text{height}$$

$$V_{\text{RESERVOIR}} = \frac{\pi}{4}(D_i^2 - 88.76^2) * 100 = D_i = \mathbf{123.8mm}$$

To be more safe and reservoir STD let us take $D_i = \mathbf{130mm}$

Thickness of the reservoir

Since only atmospheric pressure is applied on the inside and outside of the reservoir, we can treat the reservoir as thin walled cylinder.

Therefore from hoop stress (σ_h)

$$\sigma_h = \frac{P_c * D_i}{2t} \quad \text{where, } P_c = 1 \text{ bar} = 10^5 \text{ Pa} = 0.1 \text{ MPa}$$

$$t = \frac{P_c * D_i}{2 \sigma_h} \dots \dots \dots D_i = 130 \text{ mm} \dots \dots \text{internal diameter of the reservoir.}$$

$$\sigma_h = \sigma_{\text{all}} = 58.5 \text{ MPa} \dots \dots \dots \text{assumption and, } t = \text{thickness}$$

$$t = \frac{0.1 \text{ MPa} * 130 \text{ mm}}{2 * 58.5 \text{ MPa}} = 0.11 \text{ mm}$$

To be more safe let us add corrosion factor $C_F = 2 \text{ mm}$

$$T_{\text{actual}} = t + C_f$$

$$T_{\text{actual}} = 0.11 \text{ mm} + 2 \text{ mm} = \mathbf{2.11 \text{ mm}}$$

3.5.6 Design of plunger

Since it has reciprocating motion, it is subjected to fatigue load compressive load, tensile load, fluid friction and from economic point of view and availability of material the selected material is:

Alloy steel, AISI 1002, $S_{tu} = 90 \text{ MPa}$, $S_Y = 46 \text{ MPa}$

Material: alloy steel

$$S_Y = 40 \text{ MPa.}$$

Assume, $N = 2 \dots \dots \dots$ since it is subjected tensile and compressible stress

$$\sigma_{\text{all}} = 0.65 * S_Y$$

$$= 0.65 * 40 \text{ MPa},$$

$$\sigma_{\text{all}} = 26 \text{ MPa}$$

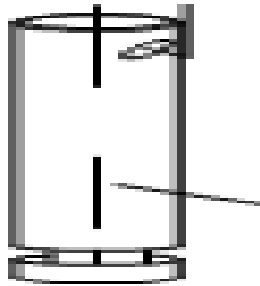


Figure 19 plunger rod diameter

In order to determine the plunger rod diameter (D_R) from compressive stress (σ_c) it can be equated as follows. $\sigma_c = \sigma_{\text{all}} = \frac{Ff}{A_{\text{rod}}}$ assumption.

$$A_{\text{rod}} = \frac{FR}{\sigma_{\text{all}}} = \frac{(\pi DR)^2}{4} = \frac{FF}{\sigma_{\text{all}}} \text{Where } F_R \text{ IS Fulcrum force}$$

$$D_R = \sqrt{\frac{4 * Ff}{\pi * \sigma_{\text{all}}}}$$

$$D_R = \sqrt{\frac{4 * 2250}{\pi * 26}} \text{ mm}$$

$$D_R = 10.4 \text{ mm}$$

✓ CHECK FOR BUCKLING

For alloy steel

$$\sigma_c = 250 \text{ MPa} \text{ crushing stress}$$

Q = Rankine constants

$$Q = \frac{\sigma_c}{C \pi^2 E} = \frac{1}{0.25} * \frac{1}{9000} = 0.00044 \text{ where C is fixture constants}$$

And, cross-sectional area A_c ;

$$A_c = \frac{(\pi DR)^2}{4} = 86.4 \text{ mm}^2$$

Active length (L) = 65mm

Radius of gyration (K)

$$K = \sqrt{\frac{I}{A}}$$

$$K = \sqrt{\frac{\pi D_R^4 / 64}{\frac{\pi D_R^2}{4}}}$$

$$K = \sqrt{\frac{D_R^2}{16}}$$

$$K = \frac{D_R}{4} = \frac{10.4}{4} = K = 2.6 \text{ mm}$$

$$(L/K)^2 = (65/2.6)^2 = 675$$

Finally, the crippling load becomes

$$W_{CR} = \frac{\sigma_c A}{1 + Q \left(\frac{L}{K}\right)^2} = \frac{250 * 86.4}{1 + 0.00044 * 675}$$

$$W_{CR} = 18 \text{ KN}$$

$$\text{Permissible buckling load} = \frac{\text{Rankine load}}{S.f}$$

$$W_{CR} = \frac{18 \text{ KN}}{3} = \dots\dots\dots S.f = 3$$

$$W_{CR} = 6 \text{ KN}$$

CONCLUSION

Since the permissible buckling load is greater than the fulcrum load ($F_F = 2.25 \text{ KN}$) this plunger rod is safe from buckling.

$$W_{CR} = 6 \text{ KN} > F_F = 2.25 \text{ KN}$$

3.5.7 Stroke calculation

The Fluid pushes against the face of the piston and produces a force.

$$F = PA \dots\dots\dots \text{Full area of piston.}$$

$$F = P (A - a) \dots\dots\dots \text{on the rod.}$$

This force on the load is often less, because of friction between the seals and piston rod and piston.

Speed; the speed of the piston depends upon the flow rate. The volume per second entering cylinder must be the change in volume per second.

$$Q \text{ (m}^3/\text{s)} = \text{area} * \text{distance moved per second}$$

$$Q \text{ (m}^3/\text{s)} = \text{area} * \text{velocity} \dots\dots\dots \text{full side}$$

$$Q \text{ (m}^3/\text{s)} = (A - a) * \text{volume} \dots\dots\dots \text{rod side}$$

Finally, POWER = volume * F

Number of stroke (N.s) = volume of the ram/volume of the plunger

But $V_{RAM} = 585979.3 \text{ mm}^3$

And volume of the plunger = $A \times H = \pi d_p^2 / 4 \times L_p$

SINCE $d_p = 10.4 \text{ mm}$we assume that $L_p = 100 \text{ mm}$

$$V_p = \pi (10.4)^2 \times \frac{100}{4} = 8490 \text{ mm}^3$$

$$(N.s) = 601151.8 \text{ mm}^3 / 8490 \text{ mm}^3 = \therefore (N.s) = 7$$

So in order to raise the load 7 stroke is needed.

3.5.8 Design of top cup and base *design of base*

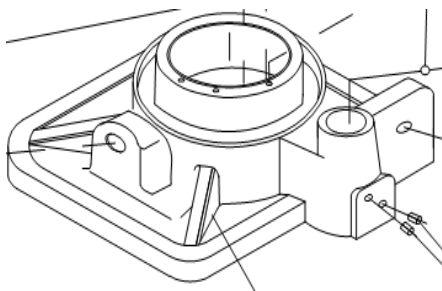


Figure 20 base

The base is used to support the jack system and also protect forgive particles from going into the system. the material used for the basement is most of the time cast iron.

This is b/c of the following reasons:

- ❖ It is cheap and easily available
- ❖ It can easily resist wear when we compared to steel and
- ❖ It is highly resist compressive load.

This base is *casted in the molding system and drilled top for fluid comes in* and out of the pump into system and from the system into the reservoir and also from the reservoir into the pump and so on.

The material is gray cast iron ASTM60 and $S_Y = 43 \text{ Mpa}$

design of second stage ram
Material specification: mild steel

$$S_Y = 90 \text{ MPa.}$$

Assume $N = 1.25$

$$\tau_{\text{All}} = 0.5 * S_Y$$

$$= 45 \text{ MPa}$$

$$\sigma_{\text{All}} = 0.65 * S_Y = 58.5 \text{ MPa}$$

Again from the given load and the applied internal pressure, it is possible to assume the ram cylinder as a thin walled cylinder.

Hence: from hoop stress (σ_h)

$$\sigma_h = \frac{P_c * D_i}{2 t} \quad \text{where } P_c = 18.125 \text{ MPa}$$

$D_i = D_{\text{MO}}$ of the second stage ram

$$D_i = 90.56 \text{ mm}$$

Assume that $\sigma_h = \sigma_{\text{all}} = 58.5 \text{ MPa}$.

$$\sigma_h = \frac{P_c * D_i}{2 t}$$

$$t = \frac{P_c * D_i}{2 \sigma_h}$$

$$t = \frac{18.125 * 90.56}{2 * 58.5}$$

$$t = 14.3 \text{ mm}$$

To be more safe let us added corrosion factor (C_F) = 0.85mm on it.

Therefore, $t_{\text{(actual)}} = t + C_F$

$$t_{\text{(actual)}} = 14.3 \text{ mm} + 0.85 = t_{\text{act}} = 15.2 \text{ mm}$$

1) CHECK FOR THE SHEARING AND CRUSHING STRESS AT THE STOP (NECK)

Since the material of the diameter of the ram cylinder is the same as that of the third stage ram, it is safe from crushing and shearing stress induced on the neck (stop) of the cylinder.

2) CHECK FOR BUCKLING

$$W_{\text{CR}} = \frac{\sigma_c A}{1 + Q \left(\frac{L}{k} \right)^2} \quad \dots\dots\dots \text{Crippling load}$$

But from the text, for mild steel

$\sigma_c = 320\text{MPa}$ and

$$Q = \frac{\sigma_c}{C\pi^2 E} = \frac{1}{0.25} * \frac{1}{7500} = 0.00053$$

And cross sectional area of the cylinder becomes

$$A = \frac{\pi}{4} (D_{MO}^2 - D_i^2)$$

But $D_{MO} = D_i + 2t = 88.76\text{mm} + 2*(30\text{mm}) = 148.8\text{mm}$

$$A = \frac{\pi}{4} (D_{MO}^2 - D_i^2)$$

$$A = \frac{\pi}{4} (148.8\text{mm}^2 - 88.76\text{mm}^2) = 11,197\text{mm}^2$$

$L = 130\text{mm}$ effective length of the cylinder

$$\text{Radius of gyration (K)} = \sqrt{\frac{I}{A}}$$

$$K = \sqrt{\frac{(\pi(D_{MO}^4 - D_o^4)/64)}{\frac{\pi(D_{MO}^2 - D_o^2)}{4}}}$$

$$K = \sqrt{\frac{D_{MO}^2 + D_o^2}{16}}$$

$$K = \sqrt{\frac{148.8^2 + 88.76^2}{16}}$$

$$K = 43.32\text{mm}$$

$$(l/k)^2 = \left[\frac{130\text{mm}}{43.32\text{mm}}\right]^2 = 9$$

FINALLY;

$$W_{CR} = \frac{320\text{MPa} * 11197\text{mm}^2}{1 + 0.00053 * 11.89}$$

$$W_{CR} = 3.583 \times 10^6 \text{N}$$

$$\text{Permissible buckling} = \frac{\text{rankine load}}{S.f}$$

$$W_{CR} = \frac{3.583 \times 10^6 \text{N}}{2}$$

$$W_{CR} = 1.79\text{MN}$$

CONCLUSION

Since the permissible buckling load is much greater than the given load, which is 20K N, this ram cylinder is safe from buckling.

$$W_{CR} = 1.79MN > F=15KN$$

Design of top cup

Since it subjected to compressive load the selected material is:

Gray Cast iron, ASTM60 and $S_Y=86MPa$

Material: GRAY CAST IRON

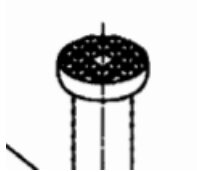


Figure 21 cup

$$S_Y = 86MPa$$

Assume that $S.f = 2$

$$\sigma_{All} = 0.65 * S_Y = 55.9MPa$$

$$\tau_{All} = 0.5 * S_Y = 43MPa$$

I) CHECK FOR SHEARING AT CRITICAL POINTS

$F_{at neck} = P_c * A_c$where $P_c = 18.125 MPa$ internal cylinder pressure

$$Area (A_c) = \frac{\pi}{4} (D_o^2 - D_i^2)$$

$$= \frac{\pi}{4} (148.8^2 - 68.76^2)$$

$$A_c = 13,767.54mm^2$$

$$F_{at neck} = 18.125MPa * 13767.54 mm^2$$

$$F_{at neck} = 247.9KN$$

II) INDUCED SHEAR STRESS

$$\tau = \frac{F_{at neck}}{A_{shear}} \quad \text{Where } A_{SH} = \pi D_i * L \dots\dots\dots L=20mm$$

$$A_{SH} = \pi * 68.76mm * 20mm$$

$$A_{SH} = 4320.32mm^2$$

$$\tau_{IN} = \frac{247.9KN}{4320.32mm^2}$$

$$\tau_{IN} = 57.4 \text{ MPa}$$

CONCLUSSION

Hence the induced shear stress at the critical point is much lower than the allowable shear stress, our cup is safe.

3.5.9 Design of handle & socket (for pump), links, & pins (at plunger & links)

Design of handle & socket for pump

Due to compressive load and tensile load and also bending force on it, the selected material is:

Carbon steel, AISI1030, $S_{tu} = 400 \text{ MPa}$ and $S_Y = 237 \text{ MPa}$

I. DESIGN OF HANDLE

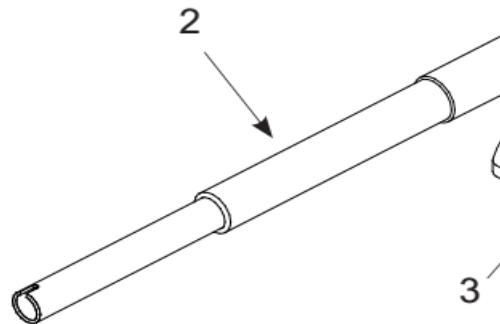


Figure 22 handle

Material: Carbon steel

$$S_Y = 237 \text{ MPa}$$

$$\tau_{ALL} = 0.65 * S_{ALL} = 154 \text{ MPa}$$

$$\tau_{ALL} = 0.5 * S_Y = 118.5 \text{ MPa}$$

To determine the diameter of the handle

Therefore to determine diameter let us calculate it from τ_{ALL}

$$\tau_{all} = (16 T_{MAX}) / \pi d^3 \dots\dots\dots \text{but } T_{MAX} = F * R \dots\dots\dots \text{for } R = 490 \text{ mm \&}$$

$$F = 200 \text{ N Then, } T = 200 * 490 = 98000 \text{ Nmm}$$

$$d = \sqrt[3]{\frac{16 * T_{max}}{\pi * \tau_{all}}}$$

$$d = \sqrt[3]{\frac{16 * 9800}{\pi * 118.5 \text{ MPa}}}$$

$$d = 16.14\text{mm} = 20\text{mm}$$

AGAIN FROM MAXIMUM BENDING THEORY

$$\sigma_{ALL} = \frac{32 M_{max}}{\pi d^3}$$

Where $M_{MAX} = 490 \text{ mm} \cdot 200\text{N}$

$$M_{MAX} = 98000\text{N}\cdot\text{mm}$$

$$d = \sqrt[3]{\frac{32 \cdot 98000}{\pi \cdot 154}}$$

$$d = 18.64\text{mm} \text{ to STD value } d = 20\text{mm}$$

Due to safety from τ_{all} of σ_{all} we have to select the greatest diameter from the above

Therefore, $d_{SHAFT} = 20\text{mm}$

Handle socket

Since it is subjected to compressive, tensile and bending load and also buckling, the material selected is:

Alloy steel ANSI1318, $S_Y = 70\text{MPa}$

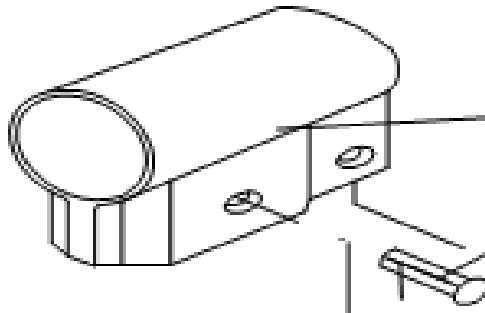


Figure 23 shows handle socket

Material – Carbon steel, $S_Y = 237\text{MPa}$

$$\tau_{ALL} = 0.5 \cdot S_Y = 0.5 \cdot 237\text{MPa} = 118.5\text{MPa}$$

$$\sigma_{ALL} = 0.65 \cdot S_Y = 154\text{MPa}$$

DETERMINATION OF DIAMETER

Again from allowable stresses we can determine the outer diameter of the socket

$$\sigma_{ALL} = \frac{4 \cdot F_f}{\pi (D_2 - d)} \quad \text{where } F_f = \text{fulcrum force} \quad F_f = 2250\text{N}$$

D_1 outer side diameter of the handle socket

d inside side diameter of the handle socket

$$D_1 = \sqrt{\frac{4Ff}{\pi \sigma_{all}} + d^2}$$

$$D_1 = \sqrt{\frac{4 \cdot 2250}{\pi \cdot 154} + 20^2}$$

$$D_1 = 20.78 \text{ mm}$$

$$\text{And, } \tau_{all} = (4 \cdot F_F) / (\pi (D^2 - d^2))$$

$$D_2 = \sqrt{\frac{4 \cdot 2250}{\pi \cdot 118.5} + 20^2} = D_2 = 20.59 \text{ mm}$$

In order to determine the diameter, let us check the thickness from the two results,

$$\text{Therefore, } t_1 = D_1 - d = 20.7 \text{ mm} - 20 \text{ mm} = 0.7 \text{ mm}$$

$$t_2 = D_2 - d = 20.6 \text{ mm} - 20 \text{ mm} = 0.6 \text{ mm}$$

This implies that it can be possible to use a material thickness greater than 0.7 mm therefore by adding correction factor $C_f = 3 \text{ mm}$

For convenience it becomes.

$$D = d + 2(t_{actual})$$

$$D = 20 \text{ mm} + 2 \cdot (0.66 \text{ mm} + 3 \text{ mm})$$

$$D = 27.32 \text{ mm. BUT FROM STD } D = 30 \text{ mm}$$

Handle socket pins

Hence, it is subjected to torsion shearing compressive load the selected material is:

Alloy steel ANSI1318, $S_y = 75 \text{ MPa}$.

i) FULCRUM PIN

Material: Alloy steel, $S_y = 75 \text{ MPa}$

$$\tau_{all} = 0.5 \cdot 75 \text{ MPa} = 37.5 \text{ MPa}$$

DETERMINATION OF THE DIAMETER

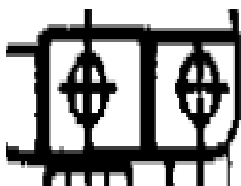


Figure 24 pin

Since the pin is subjected to double shear,

$$2\tau_{all} = \frac{Ff}{4\pi d^2} \dots\dots\dots \text{Where } F_f = 2250\text{N}$$

$$d = \sqrt{\frac{Ff}{2\pi\tau_{all}}}$$

$$d = \sqrt{\frac{2*2500}{\pi*37.5}}$$

$d = 6.18\text{mm}$, to be more safe and standard let us take $d_{PIN} = 10\text{mm}$

i. PIVOT PIN



Figure 25 pivot

Material – Alloy steel, $S_Y = 75\text{ MPa}$;

$$\tau_{all} = 0.5 * S_Y = 37.5\text{MPa}.$$

DETERMIATION OF THE DIAMETER

Again it subjected to double shear

$$\tau_{All} = \frac{Fp}{2A_{shear}}$$

$$\tau_{All} = \frac{Fp}{2\pi d^2}$$

$$d = \sqrt{\frac{Fp}{2\pi\tau_{all}}}$$

$$d = \sqrt{\frac{2*2050}{\pi 37.5}}$$

$$d = 5.9\text{mm}$$

To be more safe and standard let us take $d_{PIN} = 8\text{mm}$

3.5.10 Design of spring at discharge valve Definition;

A spring is an object used to store mechanical energy. A spring are elastic material or bodies (generally metal) that can ba twisted, pulled or stretched by some force. They can return to their original shape when force is removed. In other words it is also termed as a

resilient member. A spring is a flexible element used to exert a force or torque at the same stored energy.

Function of spring;

- ✓ To provide cushioning, to absorb or to control the energy due to shock & vibration.
- ✓ To control motion.
- ✓ To measure force.

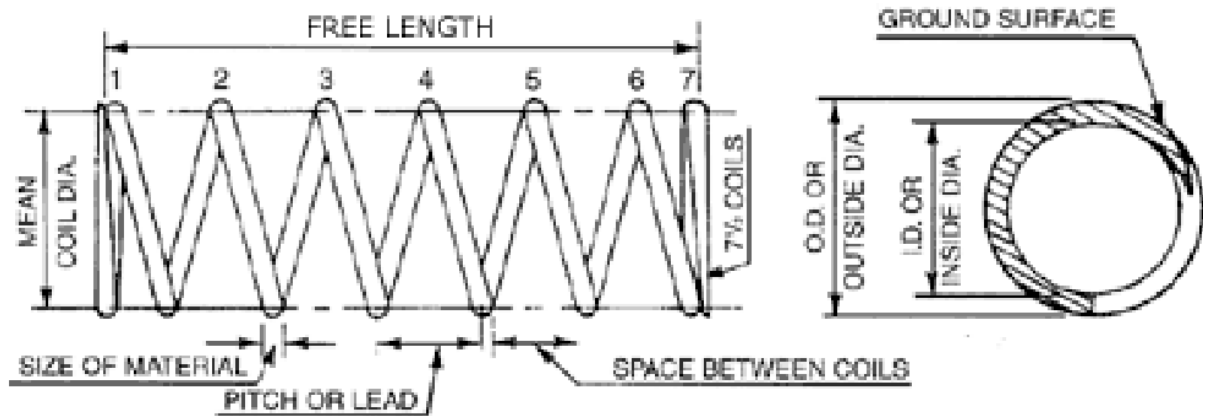


Figure 26 helical spring

I) MATERIAL for spring material, we have that for discharge valve from text book called Engineering Material Beryllium Copper (C17200)

$S_Y = 860 \text{ MPa}$From the range of (690-860)

$\tau_{all} = 0.5 * S_Y = 430 \text{ MPa}$

Modulus of elasticity (G) = 131 GPa

A) DESIGN PROCEDURE

STEP1 CHECK THE SAFETY FACTOR

Assume a wire diameter of $d = 3 \text{ mm}$ and spring index $C = 5$

$D_M = d * C = 3 * 5 = 15 \text{ mm}$ D_M , is mean diameter &

$D_O = D_M + d = 15 + 3 = 18 \text{ mm}$ D_O , is outside diameter.

Shear stress factor (K_s), Neglecting effect of curvature

$$K_s = 1 + \frac{1}{2C} = 1.1$$

Maximum shear stress induced

$$\tau_{IN} = K_S * \frac{8W D_{mean}}{\pi d^3}$$

Where $F_A = P \times A$ Axial load

And $A = \frac{\pi d^2}{4}$ assume of the fluid line with $d = 4\text{mm}$

$$A = \frac{\pi 4^2}{4} = 12.57\text{mm}^2$$

And $P = 18.125\text{ N/mm}^2$ is the cylinder pressure

Therefore, $W = P \times A = 18.125\text{N/mm}^2 \times 12.57\text{mm}^2 = 227.83\text{N}$

$$\tau_{IN} = K_S * \frac{8W D_{mean}}{\pi d^3}$$

$$= 1.1 * \frac{8 * 227.83 * 15}{\pi * 3^3}$$

$$\tau_{IN} = 354.55\text{MPa}$$

Let us check the safety of factor (N)

$$FS = \frac{\tau_{all}}{\tau_{induced}} = \frac{430}{354.55} = 1.213$$

CONCLUSION

Since the safety of factor is greater than one, our discharge spring is safe.

STEP2. CALCULATE SPRING CONSTANT

SPRING CONSTANT (K)

Assume the deflection of the spring (δ) = 2mm

And assume the end details of the spring is square ends

$$\delta = \frac{F}{K} \Rightarrow K = \frac{F}{\delta} = \frac{W}{\delta} = \frac{227.8\text{N}}{2 * 0.001\text{m}} = 113.8 * 10^3 \text{ N/m}$$

Number of active length (N_a)

$$K = \frac{d^4 G}{8 * D^3 * m * N_a} \dots\dots\dots C = D/d$$

$$N_a = \frac{d^4 G}{8 * D^3 * m * K} = \frac{G * \delta}{8 * K * C^3}$$

$$N_a = \frac{131 * 10^9 * 10^{-3} * 2}{8 * 113.7 * 10^3 * 5^3} = 4.6$$

$$N_a = 5$$

Number of total turn of the spring (N_a')

$N_a' = N_a + 2$ for square end

$$N_a' = 5 + 2 = 7$$

STEP3. Free length of the spring

We know that free length of the spring

$$L_f = (N_a' * d) + (\delta) + (0.15 \times \delta)$$

$$= 7 * 20 + 2 + (0.15 * 2)$$

$$L_f = 30.05$$

Pitch of the spring

$$P = \frac{\text{Free length}}{\text{Total number of turn} - 1} = \frac{L_f}{N_a' - 1} = 30.05 / 6 = 5.008\text{m}$$

STEP4 CHECK FOR BUCKLING

The critical load (W_{CR}) that causes buckling may be calculated by using the following relation

i.e $W_{CR} = K * K_B * l_f$ where K = spring rate or stiffness of the spring

$$K = 113.8 \text{MPa}$$

l_f = free length of the spring

K_B = Buckling factor that depending up on the ratio = $\frac{l_f}{D_{mean}}$

$$K_B = \frac{l_f}{D_{mean}} = \frac{30.05}{20} = 1.5025$$

And for built in end spring, $\mu_B = 0.71$

$$K = 113.8 * 10^3 \text{N/m from the above}$$

Therefore. $W_{CR} = K * K_B * l_f$

$$= 113.8 * 10^3 \frac{\text{N}}{\text{m}} * 0.71 * 30.05 * 10^{-3}$$

$$W_{CR} = 2508.7 \text{ N}$$

CONCLUSION

Since the critical axial load is greater than the applied axial load ($W = 277.8 \text{ N}$) then this spring is safe from buckling. $W_{CR} = 2508.7 \text{ N} > W = 277.8 \text{ N}$

3.5.11 Design of valves

Design of ball discharge valve

Material due to the application of the fluid friction on it and we have selected stainless steel and diameter of the ball = mean diameter of the spring

Therefore, $D_B = D_{\text{mean}} = 15\text{mm}$

Design spring for relief valve

MATERIAL; this spring is come to in operation, if the jack is over landed to 1KN large than the designed (require) load. Carbon stainless- steel

$\tau_{\text{All}} = \text{for average service} = 350 \text{ MPa}$

DESIGN STEPS

STEP1; CALCULATE ALL DIAMETER AND PITCH OF SPRING.

Assume for standard wire gauge (SWG) =7, a wire diameter of $d = 4.47\text{mm}$ and spring index $C = 5$

Mean diameter $D_M = C \times d = 22.35\text{mm}$

Outer diameter $D_o = D + d = 22.35 + 4.47 \text{ mm}$

$$D_o = 26.35\text{mm}$$

Shear stress factor (K_s), neglecting the effect of curvature

$$K_s = 1 + \frac{1}{2C} = 1 + \frac{1}{2} \times 5 = 1.1$$

STEP2; CHECK FOR MAXIMUM SHEAR STRESS INDUCED

$$\tau_{\text{Indused}} = K_s * \frac{8FD}{\pi d^3}$$

Where $F = P \times A$ applied load

But internal pressure developed will be increase and the applied load also increases by 1KN

$$\text{I.e } W_{\text{app}} = F_{\text{Ram}} + 1\text{KN} = 15\text{KN} + 1\text{KN} = 16\text{KN}$$

D = mean diameter

d = wire diameter

K_s = shear stress factor

Therefore, the internal pressure (P)

$$P = \frac{W_{app \text{ on the Ram}}}{A_{ram}} = \frac{F_{app}}{n(dr)^2}$$

$$P = 4 * \frac{16KN}{n(60mm)^2} = 5.6588MPa \quad \dots\dots\dots \text{assume ram diameter is } d_R = 60mm$$

$$F_{ax} = P * A = 5.6588 * \frac{\pi}{4} * 4.47^2 \quad \dots\dots\dots \text{assume line diameter } d_l = 4.47mm$$

$$F_{ax} = 88.804N$$

$$\tau_{In} = K_s * 8 * w * D / \pi d^3 = 1.1 * 8 * 266N * 26.82mm / \pi (4mm)^3$$

$$\tau_{In} = 312.24Mpa$$

CONCLUSION

Since the induced shear stress is less than that of allowable shear stress therefore the safe region and acceptable.

STEP3; DETERMINE SPRING STIFFNESS & FREE LENGTH

Spring rate (k)

Assume deflection of the spring (δ) = 2mm and assume end details of the spring is square end

$$K = F / \delta = 88.804 / 2 = 44.402N/mm$$

Number of active coil (N_a)

$$N_a = \frac{G * \delta}{8 * k * C^3}$$

$$\text{Therefore } N_a = \frac{70 * 10^3 * 2}{8 * 44.402 * 5^3} = 3.153 \Rightarrow \text{for the sake of safety take } N_a = 4$$

Free length (L_f)

$$L_f = N_a' * \delta + \delta + 0.15d$$

$$N_a' = N_a + 2 = 8 = 5 * 2 + 2 + 0.15 * 2 = 12.3mm$$

STEP4; USING FREE LENGTH DETERMINE PITCH OF THE SPRING (P)

$$P = \frac{L_f}{N_a - 1} \quad \Rightarrow \text{but } N_a' = N_a + 2 = 4 + 2 = 6 \quad \dots\dots\dots \text{for square end}$$

$$P = \frac{12.3}{6 - 1} = 2.46mm$$

B) CHECK FOR BUCKLING

The critical axial load (F_{CR}) that causes buckling may be calculated by using the following relation

$$W_{cr} = k \cdot k_B \cdot L_f \quad \dots\dots\dots \text{Where } k = 58.25 \text{ N/mm}$$

K_b = buckling factor

L_f = free length

$$K_B = \frac{L_f}{D} = \frac{12.3}{22.24} = 0.55$$

$$F_{CR} = 58.25 \text{ N/mm} \cdot 0.55 \cdot 12.3 = 394 \text{ N}$$

CONCLUSION

Since the critical axial load is greater than that of applied axial load, the spring is safe from buckling.

3.5.12 Ball for relief valve

Material is Stain less steel due to application of fluid friction, this material is selected and diameter of the ball

$$D_B = \text{mean diameter of the spring} = 22.35 \text{ mm}$$

Design of spring for intake valve

Material again from the same text book the spring material is selected.

A. Design procedure

SEP1) Assume that wire diameter, $d = 2 \text{ mm}$ and index $c = 8$

STEP2) Mean diameter

$$D = cd = 8 \times 2 = 16 \text{ mm}$$

STEP3) Outer diameter

$$D_o = D + d = 16 + 2 = 18 \text{ mm}$$

STEP4) Shear stress factor $K_S = 1 + 1/2c = 1.0625$

STEP5) max shear stress that induced

$$\tau_{\text{Induced}} = K_S \times 8WD / \pi d^3$$

Where $w = PA \dots\dots\dots$ the force acting on spring (axial load)

Assume that the fluid in the reservoir is in the atmospheric pressure that is $P_{\text{atm}} = 1 \text{ bar}$.

$$A = \pi d^2 / 4 = \pi \cdot 3^2 / 4 = 7.03 \text{ mm}^2$$

$$W=PA = 0.1*12.57 =0.703\text{N}$$

$$D=\text{mean diameter}=16\text{mm}$$

$$D=\text{wire diameter} =2\text{mm}$$

$$K_S=\text{shear stress factor} =1.0625$$

$$\tau_{\text{Induced}} =K_S*w*D/\pi d^3 = 1.0625**16/\pi 2^3 =4.78\text{MPa}$$

Since the induced shear stress is less than the allowable stress

The spring is safe and acceptable

STEP6. Spring rate (k)

Assume deflection (δ) =2mm and end detail of the spring is square and $k=w/\delta$ =

$$K = 0.703/2 = 0.35$$

STEP7. Number of active turns (N_a)

$$N_A=GD/8KC^3=3.3 = N_a = 4$$

STEP8. Total turn of the coil (N_a')

$N_a'=N_a+2$ for square and ground ends

$$= 4 +2$$

$$N_a' = 6$$

STEP9) free length of the spring (L_f)

$$L_f = N_a'.\delta + \delta + .15\delta$$

$$L_f = (6 * 2) + 2 + (0.15 * 2) = 14.3 \text{ mm}$$

$$\text{Pitch} =L_f/N_a'-1 = 14.3/5= 2.8 \text{ mm}$$

Ball for intake valve

Material due to the same reason as the above the selected material is called stainless steel.

Diameter of the ball

$$D_{\text{ball}} =\text{mean diameter of the spring} =16\text{mm}$$

3.5.13 Design of o - ring seals (for pump piston /cylinder, hollow ram& solid ram plunger

O-ring are torus-shaped seals of circular cross-section made from a variety of *elastomeric* and are used in static and dynamic condition.

Material Nitride Butadiene (class A)

And we have selected from compatible for different purpose.

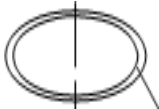
i. O- RING for pump piston /cylinder



O-ring piston with diameter of; $D=12\text{mm}$

Figure 27 O-ring piston

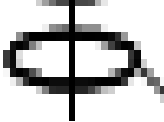
ii. O- RING for hollow RAM



$D_{mo}=90.56\text{mm}$

Figure 28 O-ring hollow ram

iii. O - RING solid ram plunger



O-ring of plunger with $d=10.4\text{mm}$

Figure 29 O - ring solid ram

iv. Design of the releasing screw

Material alloy steel, ANSI1318, $S_y=75\text{MPa}$

The selected STD screw M6

v. Design of refiling screw

Material: alloy steel and the selected STD screw is M3.

CHAPTER IV

COST ANALYSIS

RAM CYLINDER:

$$\text{Volume of A} = 2827.33 \times 95 = 268,597 \text{ mm}^3$$

$$\text{Volume of B} = 2474.3 \times 95 = 235,059 \text{ mm}^3$$

$$\text{Volume of C} = 158.5 \times 60 = 9510 \text{ mm}^3$$

$$\begin{aligned} \text{Total volume} &= \text{volume of A} + \text{volume of B} + \text{volume of C} \\ &= 268,597 \text{ mm}^3 + 235,059 \text{ mm}^3 + 9510 \text{ mm}^3 \\ &= 513,166 \text{ mm}^3 \end{aligned}$$

$$\begin{aligned} \text{Weight} &= \text{volume} \times \text{density} \\ &= 513,166 \times 10^{-9} \times 7.868 \times 10^6 \text{ gm/m}^3 \\ &= 4038 \text{ g} = 4.04 \text{ kg} \end{aligned}$$

RESERVOIR:

$$\text{Volume of A} = 183,783.17 \text{ mm}^3$$

$$\text{Volume of B} = 418,461.6 \text{ mm}^3$$

$$\begin{aligned} \text{Total volume} &= \text{volume of A} + \text{volume of B} \\ &= 183,783.17 \text{ mm}^3 + 402,196 \text{ mm}^3 \\ &= 585,979.3 \text{ mm}^3 \end{aligned}$$

$$\begin{aligned} \text{Weight} &= \text{volume} \times \text{density} \\ &= 585,979.3 \times 10^{-9} \times 7.868 \times 10^6 \text{ gm/m}^3 \\ &= 4611 \text{ g} = 4.61 \text{ kg} \end{aligned}$$

PLUNGER:

$$\text{Total volume} = 8490 \text{ mm}^3$$

$$\begin{aligned} \text{Weight} &= \text{volume} \times \text{density} \\ &= 8490 \times 10^{-9} \times 7.868 \times 10^6 \text{ gm/m}^3 \end{aligned}$$

$$=66.799\text{g}=0.0668\text{kg}$$

BASE PLATE:

$$\text{Volume of A} = l \times b \times h$$

$$= 148.8 \times 130 \times 15.2 = 292,448 \text{ mm}^3$$

$$\text{Weight} = \text{volume} \times \text{density}$$

$$= 292,448 \times 10^{-9} \times 7.868 \times 10^6 \text{ gm/m}^3$$

$$= 2.301 \text{ kg}$$

TOP CUP (NICK):

$$\text{Volume} = 432,032 \text{ mm}^3$$

$$\text{Weight} = \text{volume} \times \text{density}$$

$$= 432,032 \times 10^{-9} \times 7.868 \times 10^6 \text{ gm/m}^3$$

$$= 3399\text{g} = 3.4 \text{ kg}$$

HANDLE (LEVER):

$$\text{Volume} = 368.155 \text{ mm}^3$$

$$\text{Weight} = \text{volume} \times \text{density}$$

$$= 368.155 \times 10^{-9} \times 7.868 \times 10^6 \text{ gm/m}^3$$

$$= 2.87 \text{ kg}$$

$$\text{Net weight} = 4.04 \text{ kg} + 4.61 \text{ kg} + 0.0668\text{kg} + 2.301 \text{ kg} + 0.712\text{kg} + 3.4 \text{ kg} + 2.87 \text{ kg}$$

$$\approx 18 \text{ kg}$$

No	Material	Material Weight	cost	Total Cost
1	Mild Steel	18	26	468
2	Cast Iron	2.062	24	49.488
Total Direct Material Cost				517.488

If the money is converted to ethopian currency it becomes 11,747 birr but the cost must consider the following economical cost analysis

$11,747.00 + 15\% \text{ overhead cost} + 10\% \text{ depreciation cost} = \mathbf{14,683.722 \text{ Birr}}$

4.1 Part and assembly drawing

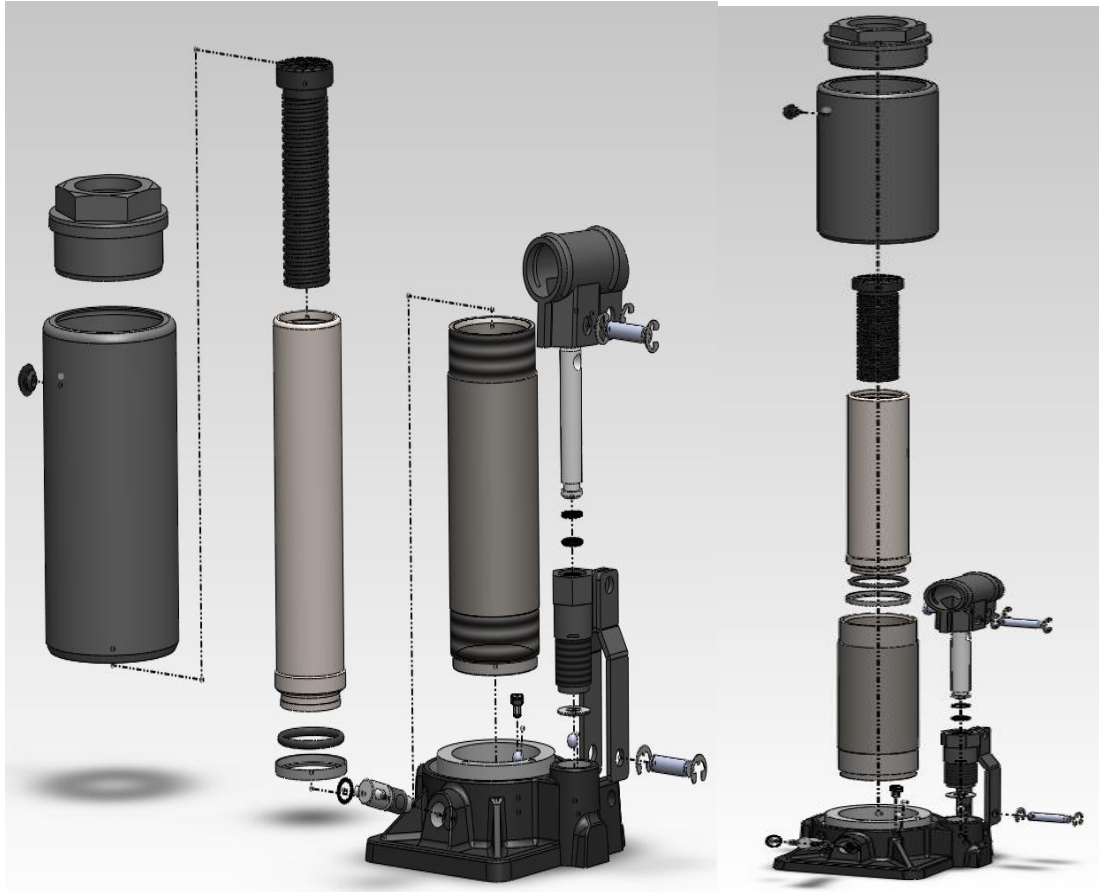


Figure 30 Assembly 3D Exploded

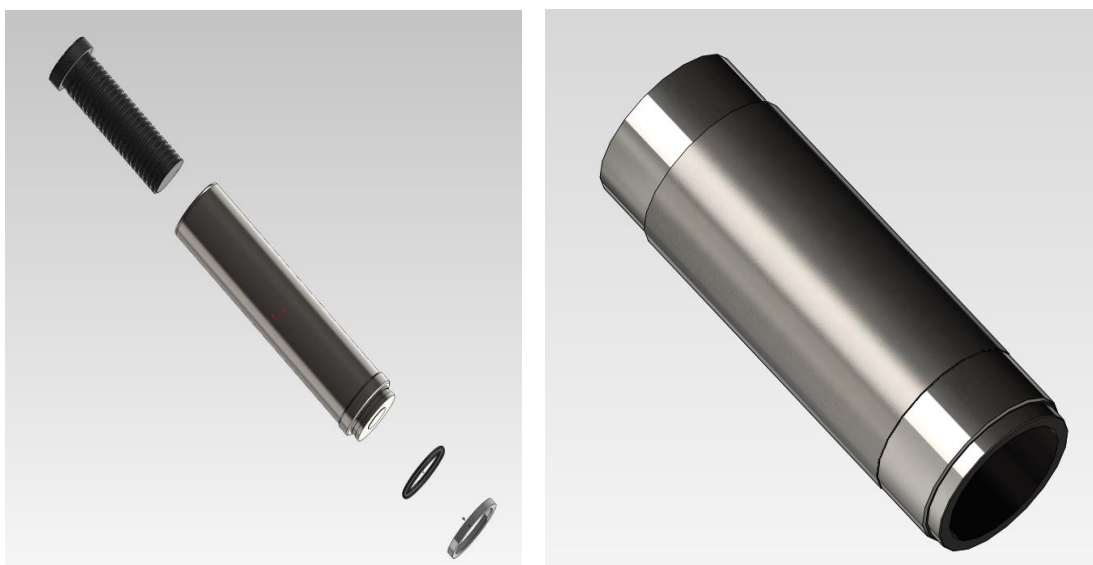


Figure 31 Piston Components



Figure 32 Lever/Pump Components

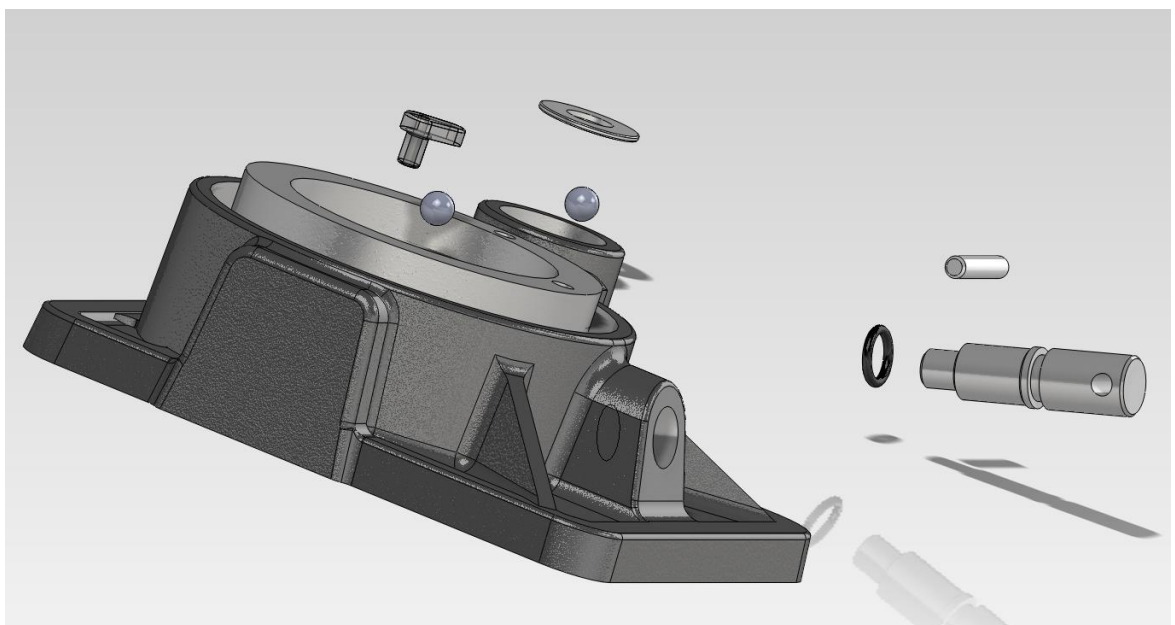


Figure 33 Bottle Jack Base Components



Figure 34 Bottle Jack Housing/Retaining Nut

CHAPTER V

CONCLUSIONS

Object lifting jacks are the ideal product to push, pull, lift, lower and position loads of anything from a couple of kilograms to hundreds of tones. The need has long existed for an improved portable jack for automotive vehicles. It is highly desirable that a jack become available that can be operated alternatively from inside the vehicle or from a location of safety off the road on which the vehicle is located. Such a jack should be light enough and be compact enough so that it can be stored in an automobile trunk, can be lifted up and carried by most adult's torso position of use, and yet be capable of lifting a wheel of a 4000-5000-pound vehicle off the ground. Further, it should be stable and easily controllable by a switch so that jacking can be done from a position of safety. It should be easily movable either to a position underneath the axle of the vehicle or some other reinforced support surface designed to be engaged by a jack. Thus, the product has been developed considering all the above requirements. This particular design of motorized automated object lifting jack will prove to be beneficial in lifting and lowering of heavy loads.

APPENDIX

Table 4 standard wire gauge (swg) number & corresponding diameter of spring wire

<i>SWG</i>	<i>Diameter (mm)</i>	<i>SWG</i>	<i>Diameter (mm)</i>	<i>SWG</i>	<i>Diameter (mm)</i>	<i>SWG</i>	<i>Diameter (mm)</i>
7/0	12.70	7	4.470	20	0.914	33	0.2540
6/0	11.785	8	4.064	21	0.813	34	0.2337
5/0	10.973	9	3.658	22	0.711	35	0.2134
4/0	10.160	10	3.251	23	0.610	36	0.1930
3/0	9.490	11	2.946	24	0.559	37	0.1727
2/0	8.839	12	2.642	25	0.508	38	0.1524
0	8.229	13	2.337	26	0.457	39	0.1321
1	7.620	14	2.032	27	0.4166	40	0.1219
2	7.010	15	1.829	28	0.3759	41	0.1118
3	6.401	16	1.626	29	0.3454	42	0.1016
4	5.893	17	1.422	30	0.3150	43	0.0914
5	5.385	18	1.219	31	0.2946	44	0.0813
6	4.877	19	1.016	32	0.2743	45	0.0711

Table 5 values of allowable shear stress

Table 23.1. Values of allowable shear stress, Modulus of elasticity and Modulus of rigidity for various spring materials.

Material	Allowable shear stress (τ) MPa			Modulus of rigidity (G) kN/m ²	Modulus of elasticity (E) kN/mm ²
	Severe service	Average service	Light service		
1. Carbon steel					
(a) Upto to 2.125 mm dia.	420	525	651	80	210
(b) 2.125 to 4.625 mm	385	483	595		
(c) 4.625 to 8.00 mm	336	420	525		
(d) 8.00 to 13.25 mm	294	364	455		
(e) 13.25 to 24.25 mm	252	315	392		
(f) 24.25 to 38.00 mm	224	280	350		
2. Music wire	392	490	612	70	196
3. Oil tempered wire	336	420	525		
4. Hard-drawn spring wire	280	350	437.5		
5. Stainless-steel wire	280	350	437.5		
6. Monel metal	196	245	306	44	105
7. Phosphor bronze	196	245	306	44	105
8. Brass	140	175	219	35	100

Table 6 Indian standard designation

Table 2.5. Indian standard designation of steel according to IS : 1570 (Part I)-1978 (Reaffirmed 1993).

<i>Indian standard designation</i>	<i>Tensile strength (Minimum) N/mm²</i>	<i>Yield stress (Minimum) N/mm²</i>	<i>Minimum percentage elongation</i>	<i>Uses as per IS : 1871 (Part I)-1987 (Reaffirmed 1993)</i>
Fe 290	290	170	27	It is used for plain drawn or enamelled parts, tubes for oil well casing, steam, water and air passage, cycle, motor cycle and automobile tubes, rivet bars and wire.
Fe E 220	290	220	27	
Fe 310	310	180	26	These steels are used for locomotive carriages and car structures, screw stock and other general engineering purposes.
Fe E 230	310	230	26	
Fe 330	330	200	26	
Fe E 250	330	250	26	
Fe 360	360	220	25	It is used for chemical pressure vessels and other general engineering purposes.
Fe E 270	360	270	25	
Fe 410	410	250	23	It is used for bridges and building construction, railway rolling stock, screw spikes, oil well casing, tube piles, and other general engineering purposes.
Fe E 310	410	310	23	
Fe 490	490	290	21	It is used for mines, forgings for marine engines, sheet piling and machine parts.
Fe E 370	490	370	21	
Fe 540	540	320	20	It is used for locomotive, carriage, wagon and tramway axles, arches for mines, bolts, seamless and welded tubes.
Fe E 400	540	400	20	
Fe 620	620	380	15	It is used for tramway axles and seamless tubes.
Fe E 460	620	460	15	
Fe 690	690	410	12	It is used for locomotive, carriage and wagon wheels and tyres, arches for mines, seamless oil well casing and drill tubes, and machine parts for heavy loading.
Fe E 520	690	520	12	
Fe 770	770	460	10	It is used for locomotive, carriage and wagon wheels and tyres, and machine parts for heavy loading.
Fe E 580	770	580	10	
Fe 870	870	520	8	It is used for locomotive, carriage and wagon wheels and tyres.
Fe E 650	870	650	8	

Table 7 typical application of copper alloy

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Table 11.6 Compositions, Mechanical Properties, and Typical Applications for Eight Copper Alloys

Alloy Name	UNS Number	Composition (wt%) ^a	Condition	Mechanical Properties			Typical Applications
				Tensile Strength [MPa (ksi)]	Yield Strength [MPa (ksi)]	Ductility [% EL in 50 mm (2 in.)]	
Wrought Alloys							
Electrolytic tough pitch	C11000	0.04 O	Annealed	220 (32)	69 (10)	45	Electrical wire, rivets, screening, gaskets, pans, nails, roofing
Beryllium copper	C17200	1.9 Be, 0.20 Co	Precipitation hardened	1140–1310 (165–190)	690–860 (100–125)	4–10	Springs, bellows, firing pins, bushings, valves, diaphragms
Cartridge brass	C26000	30 Zn	Annealed	300 (44)	75 (11)	68	Automotive radiator cores, ammunition components, lamp fixtures, flashlight shells, kickplates
			Cold-worked (H04 hard)	525 (76)	435 (63)	8	
Phosphor bronze, 5% A	C51000	5 Sn, 0.2 P	Annealed	325 (47)	130 (19)	64	Bellows, clutch disks, diaphragms, fuse clips, springs, welding rods
			Cold-worked (H04 hard)	560 (81)	515 (75)	10	
Copper–nickel, 30%	C71500	30 Ni	Annealed	380 (55)	125 (18)	36	Condenser and heat-exchanger components, saltwater piping
			Cold-worked (H02 hard)	515 (75)	485 (70)	15	
Cast Alloys							
Leaded yellow brass	C85400	29 Zn, 3 Pb, 1 Sn	As cast	234 (34)	83 (12)	35	Furniture hardware, radiator fittings, light fixtures, battery clamps
Tin bronze	C90500	10 Sn, 2 Zn	As cast	310 (45)	152 (22)	25	Bearings, bushings, piston rings, steam fittings, gears
Aluminum bronze	C95400	4 Fe, 11 Al	As cast	586 (85)	241 (35)	18	Bearings, gears, worms, bushings, valve seats and guards, pickling hooks

Table 8 design dimension of screw thread, bolt and nut

Table 11.1. Design dimensions of screw threads, bolts and nuts according to IS : 4218 (Part III) 1976 (Reaffirmed 1996) (Refer Fig. 11.1)

Designation	Pitch mm	Major or nominal diameter Nut and Bolt ($d = D$) mm	Effective or pitch diameter Nut and Bolt (d_p) mm	Minor or core diameter (d_c) mm		Depth of thread (bolt) mm	Stress area mm ²
				Bolt	Nut		
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Coarse series							
M 0.4	0.1	0.400	0.335	0.277	0.292	0.061	0.074
M 0.6	0.15	0.600	0.503	0.416	0.438	0.092	0.166
M 0.8	0.2	0.800	0.670	0.555	0.584	0.123	0.295
M 1	0.25	1.000	0.838	0.693	0.729	0.153	0.460
M 1.2	0.25	1.200	1.038	0.893	0.929	0.158	0.732
M 1.4	0.3	1.400	1.205	1.032	1.075	0.184	0.983
M 1.6	0.35	1.600	1.373	1.171	1.221	0.215	1.27
M 1.8	0.35	1.800	1.573	1.371	1.421	0.215	1.70
M 2	0.4	2.000	1.740	1.509	1.567	0.245	2.07
M 2.2	0.45	2.200	1.908	1.648	1.713	0.276	2.48
M 2.5	0.45	2.500	2.208	1.948	2.013	0.276	3.39
M 3	0.5	3.000	2.675	2.387	2.459	0.307	5.03
M 3.5	0.6	3.500	3.110	2.764	2.850	0.368	6.78
M 4	0.7	4.000	3.545	3.141	3.242	0.429	8.78
M 4.5	0.75	4.500	4.013	3.580	3.688	0.460	11.3
M 5	0.8	5.000	4.480	4.019	4.134	0.491	14.2
M 6	1	6.000	5.350	4.773	4.918	0.613	20.1

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
M 7	1	7.000	6.350	5.773	5.918	0.613	28.9
M 8	1.25	8.000	7.188	6.466	6.647	0.767	36.6
M 10	1.5	10.000	9.026	8.160	8.876	0.920	58.3

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