International Journal On Engineering Technology and Sciences – IJETS™
ISSN(P): 2349-3968, ISSN (O): 2349-3976
Volume III, Issue IV, April- 2016

DESIGN AND FABRICATION OF HYDRAULIC MULTI EXTRACTOR

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ABSTRACT

The hydraulic power source is used in many industrial applications. The hydraulic power is mainly employed where the payload is more. In industries, the bearings are removed from the machine part is performed by manual or single purpose puller. So it will take more time and risky job. In our project, the design of hydraulic multi extractor mainly focuses on to overcome this disadvantage. The hydraulic multi extractor works on the principal of hydraulic working medium. By using this extractor complicated bearings and shaft arrangements can be dismantled easily.

Traditional method of bearing removal or installation is hammering, but unnecessary hammering causes several problems. The unsafe and excessive hammering cause's damage of bearing surface or sometimes chance to failure and excessive human effort required. The modification made in easy removing and installing bearing. The purpose of modification are required less human effort , simplicity of operation , Removing and installing bearing done without damaging bearing surface, compact, portable and well suited. The hydraulic multi extractor based on hydraulic system on the principle of Pascal's law which states that "Pressure distribution in enclosed cylinder is uniform in all direction. The hydraulic multi extractor can also perform operations like removal of different size of bearings, separation of two misalignment shafts from the machine part and punching operations. The design will be helpful for the industries to overcome their problem in future.

Keywords:

Hydraulics, Principles Of Hydraulics, Description of parts, Working of Hydraulic multi extractor, Bearing Pulling and Pushing operations, Cad diagrams

INTRODUCTION

HYDRAULIC FLUIDS:

The fluid, which is selected for use in hydraulically actuated equipment, will have a considerable effect on its performance, maintenance costs and service life. The primary function of the liquid in a hydraulic system is to transmit power to perform useful work. The hydraulic fluid must transmit an applied force from one part of the system to another and must respond to reproduce any change in magnitude or direction of the applied force. In-order to perform this primary function, the fluid must be relatively incompressible and have flow ability.

Three types of fluids are generally accepted as adaptable for use in hydraulic circuits:

- Petroleum oils
- Synthetic fluids
- Water

DESCRIPTION OF EQUIPMENT

HYDRAULIC CYLINDER:

A hydraulic cylinder is a device, which converts fluid power into linear mechanical force and motion. It usually consists of a movable element, a piston and a piston rod operating within a cylindrical bore.

SINGLE ACTING CYLINDER:

Single acting cylinder can deliver in a force in only one direction .The single acting cylinder is the simplest type of all hydraulic cylinders. Single acting cylinder has only one port at one end of the cylinder barrel to allow the hydraulic fluid.

DOUBLE ACTING CYLINDER:

In the double acting cylinders, liquid pressure can be applied to the either side of the piston, thereby providing a hydraulic force in both the directions. The double acting cylinder is mostly used in application where larger stroke length is desired.

DIRECTION CONTROL VALVES (INDUSTRIAL):

This type of direction control valves can be shifted by applying pressure against a piston at either end of the valve spool. When pressure of fluid is applied at the left end, it pushes the spool to the right. Removal of this left end fluid supply and introduction of fluid through the right end passage causes the spool to shift to the left.

ACCUMULATORS:

Hydraulic accumulators are used to store the hydraulic fluid under pressure and release pressurized fluid to the system on demand. The potential energy is stored in the accumulator and act as a secondary (or) auxiliary power source to do useful work whenever required by the system.

PRESSURE CONTROL VALVES:

The most widely used type of pressure control valve is the pressure relief valve. It is normally a closed valve. Its function is to limit the pressure to a specified maximum valve by diverting pump flow back to the tank. It is also employed as a backup device when the main pressure control device fails.

CHECK VALVE:

The simplest type of one direction flow valve is check valve. Check valve is a one way value because it permits flow in only one direction and prevents any flow in the opposite direction.

MOTORS:

An electric motor is a machine which converts electric energy into mechanical energy. Its action is based on the principle that when a current carrying conductor is placed in a magnetic field, it experiences a mechanical force

PUMPS:

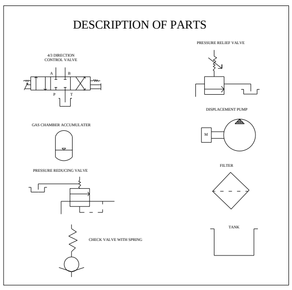
It transfers the energy to the cylinder and it pulls the piston to do the operation of the hydraulic multi extractor. It needs to pull the puller to remove the bearing from the shaft. The pump has many types and classifications. Some of them are explained in the following lines.

FILTERS:

A filter may be defined as a device for the removal of solids from a fluid where in the resistance to motion of such solids is in a tortuous path

RESERVOIR TANK:

The fluid reservoir is the storage tank in which the hydraulic fluid is contained. They are usually made of steel sheets, welded at the joints and it is vital to remove all the scale, rust from the inside tank. The inner side of the tank should be painted with a scalar to minimize oxidation.



Description of hydraulic parts

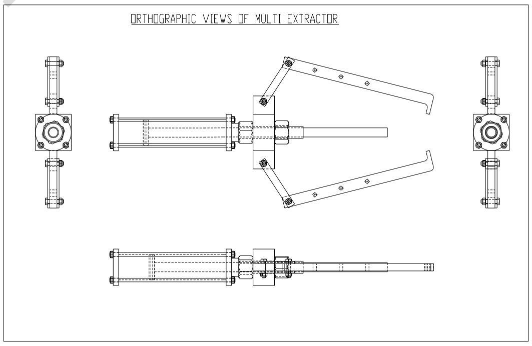
WORKING PRINCIPLE

Hydraulic multi extractor consist of 4/3 DC valve. The valve consists of two spools. When the left spool activated fluid from the pressure reducing valve enter into the valve and to the cylinder. Then the piston starts moving, during the extension of the piston the leg arms get gripped manually and the piston rod hits the bearing and dismantles it. During the activation of the right spool fluid the cylinder flows to the DC valve and then to the tank. Hence the piston retracts back to the initial position.

There are five operations can be prepared in the multi extractor. They are

- Removing the bearing from engine block.
- Ordinary puller operation.
- Removing bearing from the fixed wheel.
- Removing the rod from the hallow shaft.
- Punching operation.

Different diameter of bearing can be removed from the shaft, by using type 1 bushes. Type 2 bushes can be also used for removing bearing from the fixed wheel. Piston rod cap type2 is use for the ordinary puller operations.



Orthographic view

Bearing removal operation 1

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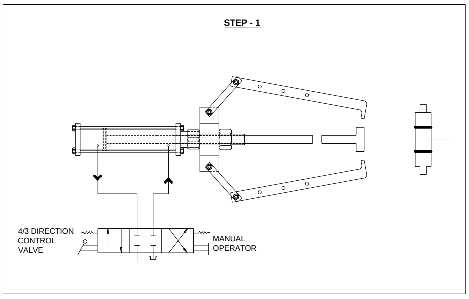
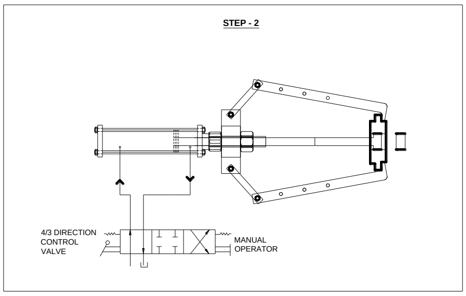
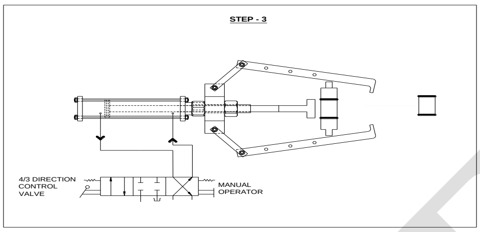
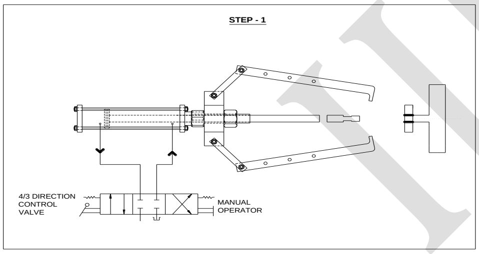
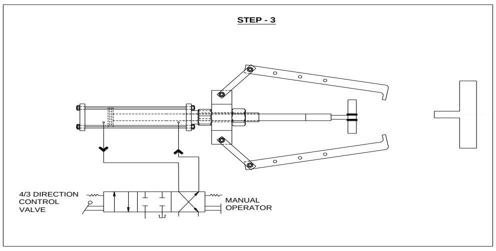


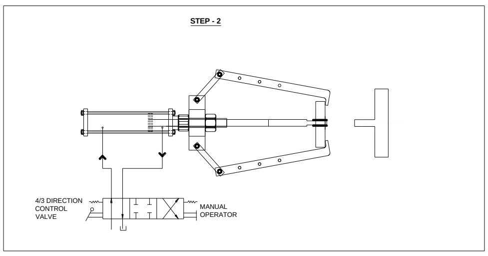




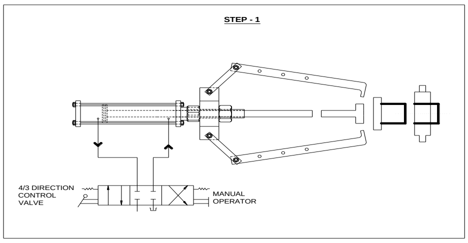
Bearing removal operations 2

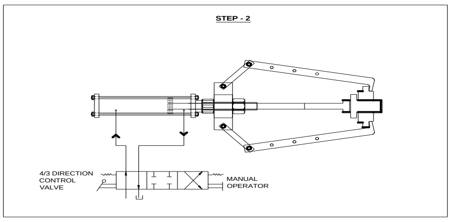


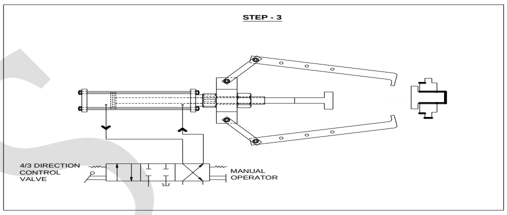




Bearing removal operation 3







DESIGN CALCULATION

SELECTION OF CYLINDER AND PISTON:

Our required maximum force [F] = 350KN
Required maximum pressure [P] = 400 bar
By applying formula F = P x A
350 x 10 = 400 x 10^5
Area [A] = 12 x $10^{-3}$$m^2$
For cylinder [A] = (π/4) d^2
d = 150 mm
Cylinder bore diameter = piston diameter
d = 150 mm

From basic values, by using piston diameter we have to select the cylinder diameter and rod diameter

Piston diameter [d_p] = 150mm
According to this,
Cylinder diameter [d_c] = 125mm
Piston rod diameter [d_{pr}] = 65mm
Piston rod length [L_{pr}] = 1200mm
Total stroke length [L_s] = 650mm
Square plate area [A_s] = 430 x 500 mm
Thickness of square plat [T_s] = 18mm
Piston width [W_p] = 45mm

6.2 SELECTION OF BASE PLATE PIPE:

Material for pipe = steel C 35
Allowable tensile strength [σ_t] = 140N/mm^2
Modulus of elasticity [E] = 2.060 x 10^5 N/mm^2
Modulus of rigidity [G] = 0.790 x 10^5 N/mm^2
Melting point [°C] = 1400°C
Based on our design requirement,
External diameter [D_0] = 80mm.

International Journal On Engineering Technology and Sciences – IJETS™
ISSN(P): 2349-3968, ISSN (O): 2349-3976
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Internal diameter $[D_i]$ = 60mm
Length of pipe $[L_{bp}]$ = 450 mm.
Thread pitch [P] = 4.257mm
Thread length of pipe $[L_{tbp}]$ = 400mm
From PSG data book pg.no:5.43,
Size designation E x t-FP $2^1/_4$
Major diameter = 85mm
Minor diameter = 64mm
Height of the pitch thread [H] = 0.961 x P
= 0.961 x 4.25
= 4mm
Radius of curve thread [r]= 0.137 x 5.257
= 0.7202mm
= 0.7mm
Area of single plate = L_p x W_p
= 0.040 x 0.050
= 2 x 10^{-3} m^2

Therefore the dimension for using our requirement is conditionally satisfied.

Reference:

1) A textbook of machine design, R.S. Khurmi.
2) PSG data book pg. 5.43

SELECTION OF BOLTS AND NUTS:

Total load or force [F] = 200KN
This load is shared by 4 bolts and nuts
Load/bolt $[P_b]$ = 200 x10^3/4
= 50000 N
By using formula,
Stress area of bolt$[A_c]$ = $(60P_b/\sigma_y)^{2/3}$ mm^2
Material for bolt and nut = 30 C8 steel
Yield strength of material $[\sigma_y]$= 350 – 450 N/mm^2
= 400 N/mm^2
A_c = $(60 \times 50000/400)^{2/3}$
= 383.1547 mm^2
From PSG data book, pg. 5.42,
For A_c = 383.15 mm^2
We can choose M30 x 2 bolt and nut
Pre load $[P_i]$ = 2860 x d
= 2860 x 30
= 85800 N
Tightening torque[T] = 0.2 x d x P_i
= 0.2 x 30 x 85800
=514800 N mm
We choosing Fine series thread,
Size of the bolt = M30 x 2
Pitch [P] = 2mm
Major diameter = 30mm
Pitch hole diameter = 15mm
Minor diameter of bolt = 27.54mm
Minor diameter of nut = 27.835mm
Depth of the thread = 1.857mm ~2mm
Maximum depth of engagement = 1.653mm
Length of bolt = 69.86mm ~ 70mm
Thread length of bolt = 62.05mm ~ 60mm
Grade = S
Property class = 5.6
Width of the bolt shank = 15mm
Width of the nut = 24.83mm ~ 25mm
Required number of bolt and nuts is 4
Designation of bolt with a nut: Hex Bolt M30 x 2 x 70N – IS: 1364-S-5:6

Reference:

1) Design of machine elements, G.K. Vijayaraghavan.
2) Handbook of mechanical design, Gitin M Maitra.
3) PSG data book pg.5.42

6.4 SELECTION OF LEG ARM AND ARM SUPPORT PLATE:

Material of leg arm = 30 C8 steel
Capacity tons = 20 Tons
Type of leg arm = 2way leg arm
Based on our design requirement:
Total load [P] = 200 KN
This load is shared by 2 leg arm and 4 arm support plate.

That is 1 leg arm and 2 leg arm support plate taken as a load or force of 100 KN.

Total area $[A_T]$ = 100000/250 x 10^5
= F/P
= 4 x $10^{-0.3}$
= 0.004 m^2
Based on our design dimension:
One leg arm length $[L_1]$ = 600mm= 0.6m
Leg arm width $[W_i]$ = 0.06m
Area of leg arm $[A_i]$ = L_1 x W
= 0.6 x 0.06
= 0.036 m^2
1 arm support plate width $[L_s]$ = 260mm = 0.26m
Arm support plate width $[W_S]$ = 0.050m
Area of support plate $[A_2]$ = $L_s \times W_s$
= 0.26 x 0.050
$[A_2]$ = 0.013m^2
2 arm support area plate $[A_{2T}]$ = 0.026 m^2
Area oh hole $[A_3]$ = $(\pi/4)$ d^2
= π x 0.018^2
= 2.5446 x 10^{-4}
Totally 4 holes are there, $[A_{3T}]$ = 2.5446 x 10^{-4} x 4
= 1.01784 x 10^{-3}
Total area of leg arm and support plate $[A_{T1}]$ = A_1 + A_{2T} – A_{3T}
= 0.036 +0.026+1.0178 x 10^{-3}
$[A_{T1}]$ = 0.063 m^2
Thickness of leg arm $[T_1]$ = 40mm = 0.04m
Thickness of the leg support plate $[T_S]$ = 20mm = 0.02m

To compare A_T and A_{T1}

So, based on the results the design dimensions of leg arm and support plate is conditionally satisfied.

Reference:

1) Hi-Force Hydraulics [P] Ltd, Chennai

6.5 SELECTION OF BASE PLATE PIPE NUTS:

Total load [P] = 200 KN

This is shared by two hexagonal flat single chamfered nuts.

Load per nut $[P_n]$ = 200 x 10^3/2
= 100000 N

Based on our design dimension of base plate pipe we take,

Material of nut = 30 C8 steel
Yield strength of the materia = 350 to 450 N/mm^2
= 400 N/mm^2
Size of the nut = M172 x 5
Pitch = 5.263 mm ~ 5mm
Type of nut = hexagonal flat nut single chamfered
Normal width of nut = 110mm
Maximum width of nut = 110mm
Minimum width of nut = 108mm

According to load, the M72 x 5 nut is enough
That is from PSG data book pg.5.42
For A_c = 957.886mm^2
Size of the nut is M72 x 5

But, according to base plate pipe dimension we have to choose the stress area
A_c = 1915.77mm^2
From PSG data book pg. 5.43
Size of the nut is M172 x 5
Required number of nut is 2

Reference:

1) Handbook of mechanical design, Gitin M Maitra
2) PSG data book pg.5.43

6.6 SELECTION OF BOLTED JOINTS:

Total load [P] = 200000N
Induced pre load by means of spanner for tightening
$[P_i]$ = 200000/2
= 100000N
We have to assume,
Material of bolt and nut = 30 C8 steel
Yield strength of material $[\sigma_y]$ = 350 to 450N/mm^2
= 400N/mm^2
Factor of safety [n] = 2.5

The effective stiffness of the parts held together by the bolt is 2.5 times the stiffness of the bolt.

$q_p = 2.5\ q_b$

Load on each belt $[P_E]$ = P/Number of bolted joints
= 200000/4
= 50000N
Bolt load on one bolt joint $[P_b]$ = P_i +P $\{q_b/(q_b+q_p)\}$
= 100000+50000 {1/ (1+2.5)}
$[P_b]$ = 114285.72N
The stress area $[A_c]$ of the bolt and nut is given by,
$P_b/A_c = \sigma_y/n$
$A_c = (P_b \times n) / \sigma_y$
= (114285.71×2.5)/400
A_c = 714.29mm^2
From PSG data book pg.5.42,
We choose M36×3 bolt and nut
Tightening torque [T] = 0.2×d×P_i
= 0.2×36×100000
= 720000Nmm
We have to choose fine series thread only.
Size of the bolt and nut = M36×3
Pitch = 3mm
Hole diameter of nut = 18mm
Minor diameter of bolt = 32.32mm
Depth of thread = 2.840mm~3mm
Maximum depth of engagement = 2.426mm
Length of bolt shank = 128.36mm~130mm
Thread length of bolt shank = 79.86mm~80mm
Grade = S
Property class = 6.4
Width of the nut = 29.86mm~30mm
Designation of bolted joints:
Required number of bolted joints 4
Hex Bolt M36×3×130N.IS:1364-S-6.4

Reference:

1) Handbook of mechanical design, G.M. Maitra
2) Design of machine elements, G.K. Vijayaraghavan
3) PSG data book, Pg.no: 5.42

6.7 SELECTION OF SUPPORT PIPE:

Material for pipe = Steel C 35
Allowable tensile stress $[\sigma_t]$ = 140N/mm^2
Modulus of elasticity [E] = 2.060×10^{5}N/mm^2
Modulus of rigidity [G] = 0.790×10^{5}N/mm^2
Melting point [°C] = 1490°C
Based on our design parameter
External diameter $[D_o]$ = 86mm
Internal diameter $[D_i]$ = 54mm
Length of pipe $[L_{BP}]$ = 400mm
Thread length of pipe $[L_{TBP}]$ = 400mm
External thread pitch $[P_{ET}]$ = 5.257mm~5mm
Internal thread pitch $[P_{IT}]$ = 2.867mm~3mm
From PSG data book Pg.5.43
Size designation E×-FP 2¼
Major diameter = 85.87mm~86mm
Our required minor diameter = 53.864mm~54mm
Height of the pitch External thread $[H_{ET}]$ = 0.961 × P
= 0.967 × 5.257
Height of the pitch internal thread $[H_{IT}]$ = 5.051~5mm
= 0.961 × P
= 2.75~3mm
Radius of external curve thread $[R_{ET}]$ = 0.137×5.257
= 0.7mm
Radius of internal curve thread $[R_{IT}]$ = 0.137×2.867
= 0.39mm~0.4mm

Reference:

1) A text book of machine design, R.S. Khurmi

6.8 DESIGN CALCULATION OF WELDING JOINTS:

By using fillet weld, the bolt should be joined.
Diameter of shaft (or) Bolt [D] = 30mm
Tensile load [P] = 50KN
Length of bolt [L] = 70mm
Assume permissible tensile stress is not to exceed
$[\tau_{tmax}]$= 610mpa

This joint is subjected to directed shear stress and tensile stress

Area of the weld [A] = tπD
= 0.707×h×π×D
= 0.707×h×π×3
[A] = 66.63h mm^2
Direct tensile stress = P/A
= 50000/66.63h

International Journal On Engineering Technology and Sciences – IJETS™
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$= 750.41/h\ N/mm^2$

Moment [M] $= P \times l$

$= 50000 \times 70$

$= 3500 \times 10^3\ N\ mm$

Section modulus [Z] $= \pi\ t\ D^2/4$

$= (\pi \times 0.707 \times h \times 30^2)/4$

$= 499.75\ h$

Normal stress $\sigma = M/Z$

$= 3500 \times 10^3/499.75\ h$

$= 7003.52/h$

Maximum tensile stress $\tau_{max} = {}^1/_2(\sigma^2 + 4[\tau_1]^2)$

$= {}^1/_2(7003.52/h)^2 + 4\ [750.41/h]^2)$

$= {}^1/_2\,(51301722.91/h^2)$

610 $= 3581.26/h$

h $= 5.87mm$

Size of the weld is [h] = 5.87 mm~6mm

By using PSG data book. Pg.11.4

According to the size of the weld = 6mm

Permissible static load per cm length is,

By choosing covered electrode,

In tension the static load = 420Kgf/cm

= 42Kgf/mm

In shear the tensile load = 335 Kgf/cm

= 33.5 Kgf/mm

Design stresses for fillet welded joints

Steady load $= 950\ Kgf/cm^2$

$= 9.50\ Kgf/mm^2$

Reversed load $= 350\ Kgf/cm^2$

$= 3.50\ Kgf/mm^2$

Selection of recommended welding processes,

Assume material= C38 stainless steel

Based on material,

Recommended process = inert gas metal arc welding

Based on fillet welded joint,

Recommended process = inert gas metal arc welding

Reference:

1) Design of machine elements, G.K. Vijayaraghavan
2) PSG data Book Pg.11.4,11.8,11.2

6.9 WELDING CALCULATION OF RECTANGULAR PLATES:

Here, by using fillet weld the rectangular plate should be joined.

Tensile load [P] $= 50KN = 50 \times 10^3 N$

Eccentricity [e] = 20mm

Cross section $= 10 \times 50 = 250mm$

Throat area of the rectangular section [A]= t [2b+2l]

$= 0.707 \times h\ [(2 \times 50) + (2 \times 10)]$

[A] $= 84.84h\ mm^2$

Direct tensile stress $[\sigma_t] = P/A$

$= 50000/84.84h$

$= 589.34/h\ N/mm^2$

Moment [M] $= P \times e$

$= 50 \times 10^3 \times 40$

$= 2000 \times 10^3\ N\ mm$

Section modulus $[Z] = t\ [bd + (d^2/3)]$

$= 0.707 \times h\ [(10 \times 50) + (50^2 + 3)]$

[Z] $= 942.67h\ mm^3$

Nominal stress = M/Z

$= (2000 \times 10^3)/\ (942.67h)$

$= 2121.63/h\ N/mm^2$

Assume, maximum permissible stress 310Mpa

Maximum tensile stress,

$T_{tmax} = {}^1/_2(\sigma^2 + 4[\tau_1]^2)$

$= ½(2121.63/h)^2 + (589.34/h)^2$

$= ½(2201.96/h)$

310 $= 1100.98/h$

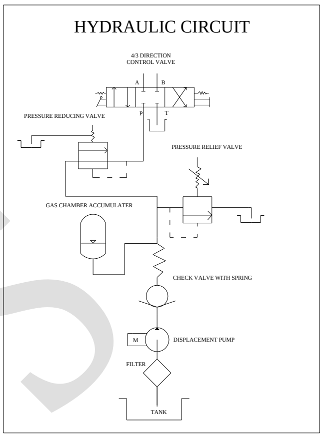
h = 3.67mm~4mm

Size of the weld is 4mm

Material of the plate is C38 steel.

Reference:

1) Design of machine elements, G.K. Vijayaraghavan
2) PSG data book Pg.11.4, 11.2, 11.8



Hydraulic circuit

CONCLUSION

The project undergone in Salem steel plant, Salem was very much useful to us. It thought us the practical knowledge about how to rectifying the problems like removal of bearing from the machine part by hydraulic multi extractor. Emerging due to poor knowledge about this report, we had faced a lot of difficulties has been overcome by sharing the ideas with the staff members of the concern. Hence the design will be helpful for the industries to overcome their problems in future.

REFERENCES

The following papers are being studied and are referred for the project. These papers belong to various authors, having various papers related to the hydraulic bearing multi extractor.

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APPENDICES: