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Hydraulic Design of a Forklift for the Transfer of Light Loads or with a Maximum Weight of 3 Tons

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Abstract

This research contains the description of the design process and selection of the components of a forklift truck with a load capacity of 3 tons, the system is developed with commercial characteristics, being able to fulfill daily work functions.

Keywords: Hydraulic pump, hydraulic motor, bending, buckling, power, moment, forks, load.

I. INTRODUCTION

In network intrusion detection system (NIDS) research, there are three types of detection approaches misused or signature-

A forklift is a vehicle used to move objects from one place to another, the forklift sized in this work has a maximum load capacity of 3 tons, which is composed of a hydraulic system whose function is to position the load in a specific place, this system is composed of three cylinders, the first of them and the one that will run the greatest force is the lifting cylinder, the second of them is the lateral displacement cylinder which is intended to position the load on the horizontal axis streamlining the work, reducing equipment transfers. Finally, there is the tilt cylinder, which is designed to tilt the load to have a static balance between the load and the vehicle. On the other hand, the vehicle consists of five main parts as shown in Figure 1. The load, the forks, the wheels, the cabin, and the counterweight. These segments constitute the static part of the vehicle, which provides the center of gravity and balance between load and equipment, an important factor in the safe operation of the vehicle.

For the development of this research, other commercial designs were taken into account for the elaboration of the hydraulic dimensioning, the considerations that were taken were the maximum height of elevation, the maximum distance of lateral travel, and the degrees of inclination of the forks. These considerations gave us fundamental parameters for the sizing of the vehicle.

In the process of the work it became evident that static elements should be calculated in order to ensure the optimal operation of the equipment, these considerations were based on the search for the force that had to exert the lifting and displacement cylinders taking into account that we had the maximum force executed by the lifting cylinder.

A free body diagram was made with the objective of determining the force exerted on the tilt cylinder, for the static development, a plane was made with the load in vertical position in the most critical case which is that it occupies the entire area of the vertical forks, then tracing its component of inclination at 10 °, from there the centroid is located idealizing that the weight is distributed throughout the truck.

Taking into account that it is a development of a work to have as a reference in research will develop the step by step to reach the goal which is to have the calculations, the necessary components to get to develop the mechanism mentioned above is also important to note that in this report of the development of the process must have basic concepts in materials science, fluids and hydraulics.

The objective of this research is to study and design a hydraulic system to make the optimal selection of equipment, accessories and/or components necessary for the execution of a forklift truck. Conduct state of the art studies for the design of hydraulic systems in forklift trucks. Identify all the components of the system in order to locate a company with the capacity to supply them and calculate and select the elements that make up the system according to the operating requirements.

II. METHODOLOGIC AND RESULTS

In the development of this problem several stages of calculations were made for the optimal operation of the system, first, the lifting cylinder was selected, to continue with the lateral displacement cylinder, finally, the tilt cylinder.

The following operating considerations were taken into account when calculating the lifting cylinder:

System Pressure	
P max (psi)	2600
P oper (psi)	2400
P min (psi)	2000

Speed	
300	mm/s
0,3	m/s
11.811	in/s

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Lifting cylinder		
force	ton	lbf
	3	6613,86

$$P = \frac{F}{A} \tag{1}$$

$$A = \frac{\pi * D^2}{4} \tag{2}$$

$$D = \sqrt{\frac{fuerza*4}{\pi * P_{oper}}} \tag{3}$$

Taking into account the formula one we clear the diameter knowing that the area is implicit in it.

Knowing the theoretical diameter of the cylinder we refer to the Parker catalogue where we will select a commercial one.

With the obtained area we recalculate the pressure of the system. Using fórmula 1, we clear the real pressure.

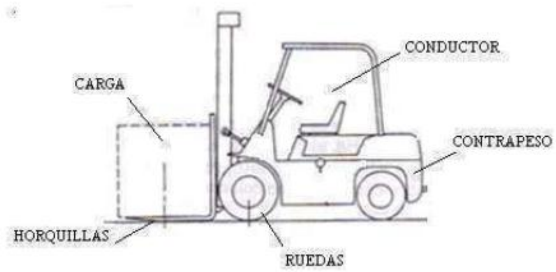

Fig. 1. Main elements of a forklift truck. Source:[1]

The type of cylinder fastening that best suits the application to be fulfilled must be selected (Figure 2). This type of mounting was selected because of its support points that minimize the bending of the element as it is supported along the entire length of the cylinder.

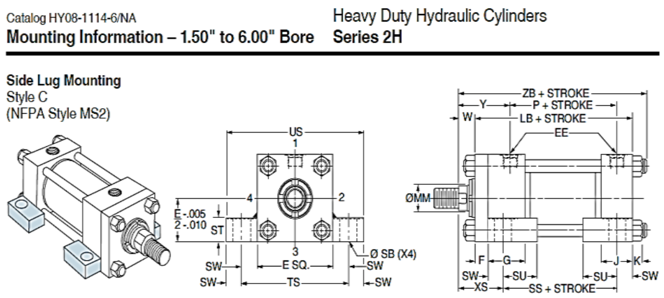

Fig. 2 Cylinder selection. Source: [1]

The cylinder that complies with the theoretically established calculations is selected.

Style C – Dimensional and Mounting D

Bore Ø	E	EE NPTF[1]	EE SAE[2]	F	G
1.50	2.50	1/2	10	0.38	1.75
2.00	3.00	1/2	10	0.63	1.75
2.50	3.50	1/2	10	0.63	1.75
3.25	4.50	3/4	12	0.75	2.00

$$I = \frac{\pi * \text{Ø}_{\text{vástago}}{}^{4}}{64} \tag{4}$$

$$\text{fuerza critica} = \frac{K * \pi^2 * E * I}{L_{Pandeo}{}^{2}} \tag{5}$$

$$\text{Ø}_{Critico} = \left(\frac{64 * L_{pandeo} * carrera_{cilindro}{}^{2}}{\pi^3 * K * E}\right)^{1/4} \tag{6}$$

To verify that the selected cylinder complies with the required capacity, the critical diameter is calculated with the formula 6.

To perform the calculation, several parameters must be taken into account, such as:

The critical force= maximum pressure*trade diameter.

$$\text{Fcr} = 2600\frac{lbf}{in^2} * \frac{\pi}{4} * 2.0in^2$$

Fcr= 8168.14 lbf. --- 3.64 Ton.

C=0.25 L=2LP =47.2 in

With the calculated piston rod diameter, the selected cylinder and the commercial piston rod diameter are searched in the PARKER catalogue.

The diameter of 1.375 in was selected as it meets the working conditions.

Maximum critical force	
8168,1409	lbf
3,64649147	ton

Career	
60	cm
23,6220472	in
c	
0,25	

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Modulus of elasticity

30000000 psi

Stem diameter

1,258532236

Style C – Dimensional and Mounting Data

Bore Ø	Rod No.	MM Rod Ø	W	XS	Y	Add Stroke ZB Max.
1.50	1 (std.)	0.625	0.63	1.38	2.00	6.25
	2	1.000	1.00	1.75	2.38	6.63
2.00	1 (std.)	1.000	0.75	1.88	2.38	6.69
	2	1.375	1.00	2.13	2.63	6.94

Sideshift cylinder

The following operating considerations were taken into account for the calculation of the lateral displacement cylinder:

In the search for information on the force that these actuators performed, it was found that they did not perform more than 50% of the total maximum force, knowing this, it was determined that the force exerted by this actuator was 20% of the total force.

	Displacement cylinder	
	ton	lbf
Force	0,6	1322,772

Speed

300	mm/s
0,3	m/s
11,811	in/s

System Pressure

P max (psi)	2600
P oper (psi)	2400
P min (psi)	2000

We proceed to calculate the cylinder that will execute the lateral mobility action of the forklift truck.

With equation 3 we determine the diameter of the cylinder.

Displacement Cylinder Bore

$$D=\sqrt{(force*4)/(\pi*P\_oper)}$$

0,837706596 (in)

Commercial diameter parker

1,5 (in)

PARKER maintains its mounts for double acting cylinders, allowing the cylinder to be fastened at four points to the chassis, providing safety in its operation (Figure 4 and 5).

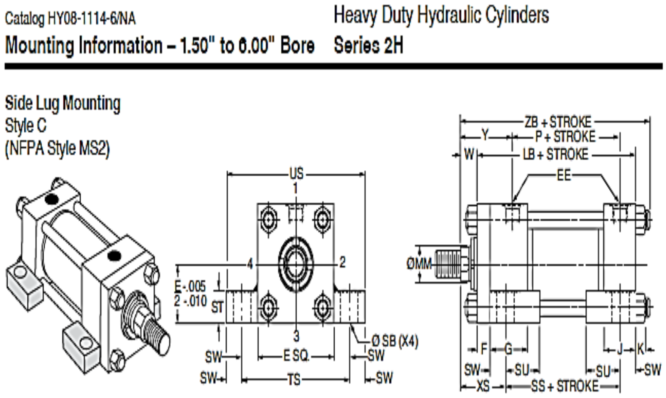

Fig. 3. Assembly selection. Source: [1]

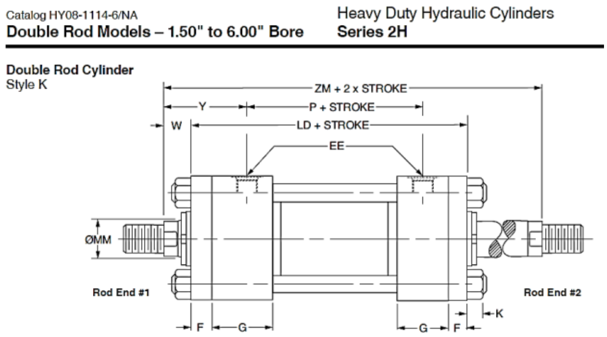

Fig. 4. Cylinder selection. Source: [1]

After determining the theoretical diameter, a cylinder is selected from the Parker catalog that meets the diameter conditions (Figure 6).

Style C – Dimensional and Mounting I

Bore Ø	Rod No.	MM Rod Ø	W	XS
1.50	1 (std.)	0.625	0.63	1.38
	2	1.000	1.00	1.75
2.00	1 (std.)	1.000	0.75	1.88
	2	1.375	1.00	2.13
2.50	1 (std.)	1.000	0.75	2.06
	2	1.750	1.25	2.56
	3	1.375	1.00	2.31

Fig. 5. Commercial diameter selection. Source: [1]

To verify that the selected cylinder complies with the required capacity, the critical diameter is calculated with formula 6 (Figure 7).

To perform the calculation, several parameters must be taken into account, such as:

The critical force

fuerza maxima critica	
3534,29174	lbf
1,57780881	ton

C=0.25 L=2LP =19.68 in

E=30*10^6 PSI

$$I = \frac{\pi * \emptyset_{\text{vástago}}{}^{4}}{64} \ (4)$$

$$Pandeo_{critico} = \frac{K * \pi^2 * E * I}{L_{Pandeo}{}^{2}} \ (5)$$

$$\emptyset_{Critico} = \left(\frac{64 * L_{pandeo} * carrera_{cilindro}{}^{2}}{\pi^3 * K * E}\right)^{1/4} \ (6)$$

Career

25 cm

9,84251969 in

c

0,25

modulus of elasticity

30000000 psi

stem diameter

0,65887574

With the calculated piston rod diameter, the selected cylinder and the commercial piston rod diameter are searched in the PARKER catalogue.

tilt cylinder diameter

D=√((force*4)/(π*P_oper))

0,592348015 (in)

Commercial diameter parker

1,5 (in)

Style C – Dimensional and Mounting I

Bore Ø	Rod No.	MM Rod Ø	W	XS
1.50	1 (std.)	0.625	0.63	1.38
	2	1.000	1.00	1.75
2.00	1 (std.)	1.000	0.75	1.88
	2	1.375	1.00	2.13
2.50	1 (std.)	1.000	0.75	2.06
	2	1.750	1.25	2.56
	3	1.375	1.00	2.31

Fig. 6. Stem diameter selection. Source: [1]

The cylinder was selected with a diameter of one inch in order to maintain a good safety factor in the operation of the system, avoiding failures.

Tilt cylinder

The following operating considerations were taken into account when calculating the lifting cylinder:

A free body diagram was made which aims to determine the force exerted on the tilt cylinder, for the static development, a plane was made with the load in vertical position in the most critical case which is that it occupies the entire area of the vertical forks, then trace its component of inclination at 10 °, from there the centroid is located idealizing that the weight is distributed throughout the truck, after this the weight component was resolved, resulting in the % of force exerted by the actuator.

To determine the angle of inclination was investigated in several brands finding that for this load capacity have these characteristics as evidenced in the following figures 8, 9 and 10.

Mecanismo de montacarga	
8. Inclinación del mástil adelante	6 grados
9. Inclinación del mástil atrás	10 grados

Fig. 7. Tilting mechanism. Source: [2]

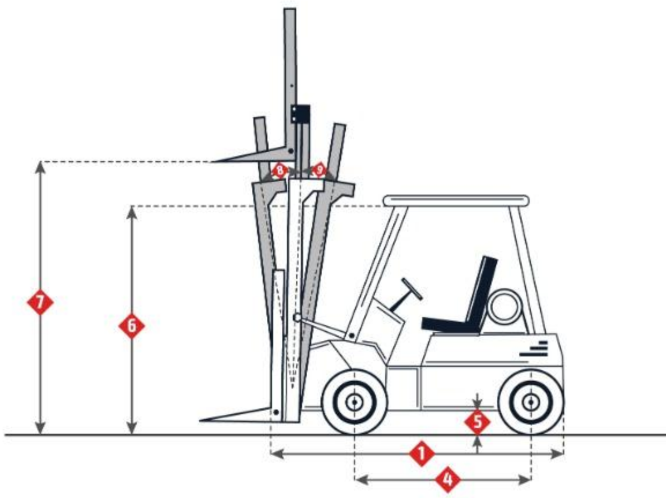

Fig. 8 Tilting mechanism. Source: [2]

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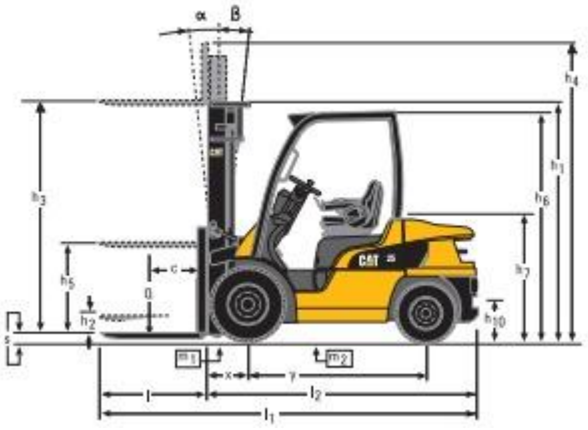


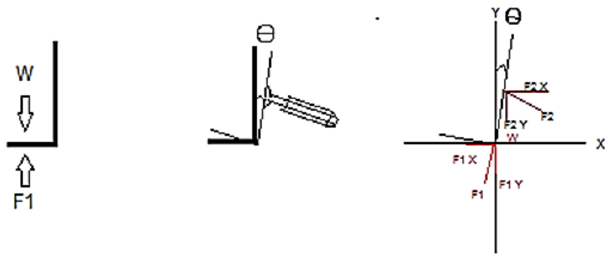
Fig. 9 Side view tilt mechanism. Source: [2]

Calculations to determine the force component that this actuator would develop.

W= Peso de la carga

F1=Fuerza del cilindro principal

F2=Fuerza del cilindro de inclinación



$w = 3\,Ton \qquad F1 = 3\,Ton \qquad \Theta = 10°$

$\sum Fy = 0; \quad f1y + F2y - w = 0$

$F2y = w - f1y = 3\,Ton - 3ton\cos 10° = 3ton - 2.954\,Ton$

$F2y = 0.455\,Ton$

$F2 = \frac{F2y}{\cos 80°} = \frac{0.0455}{\cos 80°} = 0.26\ Ton$

Approximately equal to 0.3 ton, which is equivalent to 10% of the system force.

Tilt cylinder

	ton	lbf
Force	0,3	661,386

System pressure

P max (psi)	2600
P oper (psi)	2400
P min (psi)	2000

Speed

300	mm/s
0,3	m/s
11,811	in/s

We proceed to calculate the cylinder that will execute the lateral mobility action of the forklift truck.

With equation 3 we determine the diameter of the cylinder.

The type of cylinder clamping is chosen (Figure 11, 12 and 13), PARKER has this type of assembly that allows the articulation of the cylinder with the forks. This system is placed in order not to overload the piece that is going to work with the tilting links.

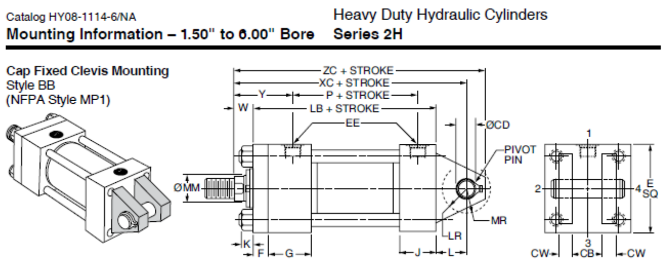


Fig. 10. Cylinder selection. Source: [1].

Style BB – Dimensional and Mounting Data

Bore Ø	Rod No.	MM Rod Ø	W	Y	Add Stroke	
					XC	ZC
1.50	1 (std.)	0.625	0.63	2.00	6.38	6.88
	2	1.000	1.00	2.38	6.75	7.25
2.00	1 (std.)	1.000	0.75	2.38	7.25	8.00
	2	1.375	1.00	2.63	7.50	8.25

Fig. 11. Selection of commercial diameter. Source: [1].

To verify that the selected cylinder complies with the required capacity, the critical diameter is calculated with the formula 6.

To perform the calculation, several parameters must be taken into account, such as:

The critical force= maximum pressure*trade diameter.

C=0.25 L=2LP =15.6 in

E=30*10^6 PSI

$$I = \frac{\pi * \emptyset_{vástago}{}^{4}}{64} \quad (4)$$

$$Pandeo_{critico} = \frac{K*\pi^{2}*E*I}{L_{Pandeo}{}^{2}} \quad (5)$$

$$\emptyset_{Critico} = \left(\frac{64*L_{pandeo}*carrera_{cilindro}{}^{2}}{\pi^{3}*K*E}\right)^{1/4} \quad (6)$$

Career

20 cm

7,87401575 in

c

0,25

modulus of elasticity

30000000 psi

stem diameter

0,589316377

Style BB – Dimensional and Mounting Data

Bore Ø	Rod No.	MM Rod Ø	W	Y	Add Stroke	
					XC	ZC
1.50	1 (std.)	0.625	0.63	2.00	6.38	6.88
	2	1.000	1.00	2.38	6.75	7.25
2.00	1 (std.)	1.000	0.75	2.38	7.25	8.00
	2	1.375	1.00	2.63	7.50	8.25

Fig. 13. Stem diameter selection. Source: [1].

The cylinder was selected with a diameter of one inch in order to maintain a good safety factor in the operation of the system, avoiding failures.

Calculation of the pump driving the three hydraulic cylinders

Since one of the characteristics is the speed of actuation of the cylinders is the sum of the flows required by each of them for their operation in order to determine the pump to supply that flow. Then we multiply the flow obtained by the volumetric efficiency which in this case is 0.94.

It should be noted that the system has the same operating speed.

Lift cylinder flow rate.

$$Q = V * A$$

Using equation 2, we calculate the area with the diameter of the commercial cylinder.

$$A = \frac{\pi * D^2}{4} \quad (2)$$

Diameter= 2 in

area

3,14159265 in^2

flow

37,1053508 in^3/s

9,63775346 gpm

36,4788776 lpm

Lateral displacement cylinder flow.

$$Q = V * A$$

Using equation 2, we calculate the area with the diameter of the commercial cylinder.

$$A = \frac{\pi * D^2}{4} \quad (2)$$

Diameter= 1.5 in

area

1,76714587 in^2

flow

20,8717598 in^3/s

5,42123632 gpm

20,5193686 lpm

Cylinder flow rate inclination.

$$Q = V * A$$

Using equation 2, we calculate the area with the diameter of the commercial cylinder.

$$A = \frac{\pi * D^2}{4} \quad (2)$$

Diameter= 1.5 in

area

1,76714587 in^2

flow

20,8717598 in^3/s

5,42123632 gpm

20,5193686 lpm

$$\Sigma QC1 + QC2 + QC3 = QTOT$$

Q1=9.36 gpm

Q2=5.41 gpm

Q3=5.41gpm

Qtot=20.18 gpm

With the above data it is possible to calculate the displacement of the pump.

$$Cilindrada = \frac{231\frac{in^3}{rev} * Q(gpm)}{N(rpm)} (7)$$

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N=1800 rpm

flow	
77,693	in3/s
20,18	gpm
76,38125964	lpm
VOLUMETRIC EFFICIENCY	
0,94	
73,03142	in3/s
18,9692	gpm
71,79838406	lpm
1800	rpm
pump displacement	
39,88799115	cm^3/rev

The theoretical pump displacement is 39.88cm^37rev.

In the Parker catalogue we select the pump with the displacement found above (Figure 14).

Tamaño bastidor PGP 517	0140	0160	0190	0230	0250	0280	0330	038	0440	0520	0700
Desplazamiento (cm³/rev)	14	16	19	23	25	28	33	3	44	52	70
Presión máx. cont. (bar)	250	250	250	250	250	250	250	25	220	200	160
Veloc. máx. trabajo (rpm)	3400	3400	3300	3300	3100	3100	3000	300	2800	2700	2400
Potencia absorbida (kW)	9,6	11,0	13,1	15,8	17,2	19,3	22,7	26,	27,0	28,6	31,2
Peso (kg)	7,92	8,00	8,12	8,29	8,37	8,50	8,70	8,9	9,16	9,49	10,24

Fig. 12. Selection of the hydraulic pump. Source: [3]

A 44 cm^3/rev pump was selected.

This pump has a maximum pressure of 3190.83 psi.

The required operating power is 27 kW.

Maximum working speed 2800 rpm

Fixed displacement gear pump

Purchase code: Pgp517

Valve calculation

To select the valve we refer to the Vickers catalogue of hydraulic valves, we proceed to search in the catalogue for the flow rate of each cylinder (Figure 15).

	gpm	in3/s	lpm
Q1 C.ELEV	9,36	36,036	35,4275813
Q2 C.DESP	5,41	20,8285	20,4768392
Q3 C.INCL	5,41	20,8285	20,4768392

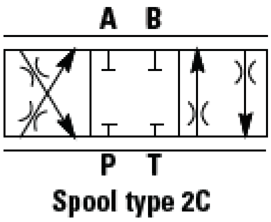

Fig. 13. Hydraulic diagram valve 4/. Source: [4]

For the three valves type 2c was chosen, which is a proportional valve with infinite positions, in the investigations it was decreed that the flow rate of the valve should be higher so that it does not affect the optimum performance (Figure 16, 17 and 18).

Valve for lifting cylinder

Spool code	Spool symbol	Flow rating
For KBFDG5V-5 valves:		
2C70N45	2C	70 L/min (18.5 USgpm) "A" port flow 45 L/min (11.9 USgpm) "B" port flow
33C60N40	33C	60 L/min (17.2 USgpm) "A" port flow 40 L/min (10.6 USgpm) "B" port flow

Fig.144/3 valve selection. Source: [4]

Valve for displacement cylinder

Spool code	Spool symbol	Flow rating
For KBFDG5V-5 valves:		
2C70N45	2C	70 L/min (18.5 USgpm) "A" port flow 45 L/min (11.9 USgpm) "B" port flow
33C60N40	33C	60 L/min (17.2 USgpm) "A" port flow 40 L/min (10.6 USgpm) "B" port flow

Fig.154/3 valve selection. Source: [4]

Valve for tilt cylinder

Spool code	Spool symbol	Flow rating
For KBFDG5V-5 valves:		
2C70N45	2C	70 L/min (18.5 USgpm) "A" port flow 45 L/min (11.9 USgpm) "B" port flow
33C60N40	33C	60 L/min (17.2 USgpm) "A" port flow 40 L/min (10.6 USgpm) "B" port flow

Fig. 164/3 valve selection. Source: [4]

Diagram of the valve arranged by Vikers (Figure 19).

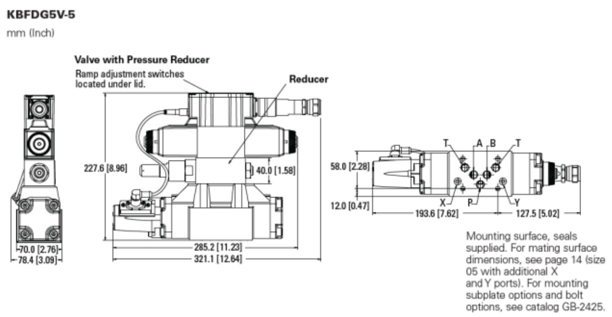

Fig. 17. Diagram of the 4/3 valve. Source: [4]

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Dimensions and installation of vikers valve (Figure 20).

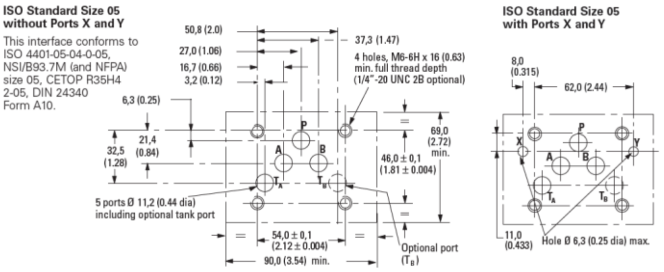

Fig. 18. Assembly of the 4/3 valve. Source: [4]

Valve flow rate depending on percentage or action of movement, infinite position valve (Figure 21).

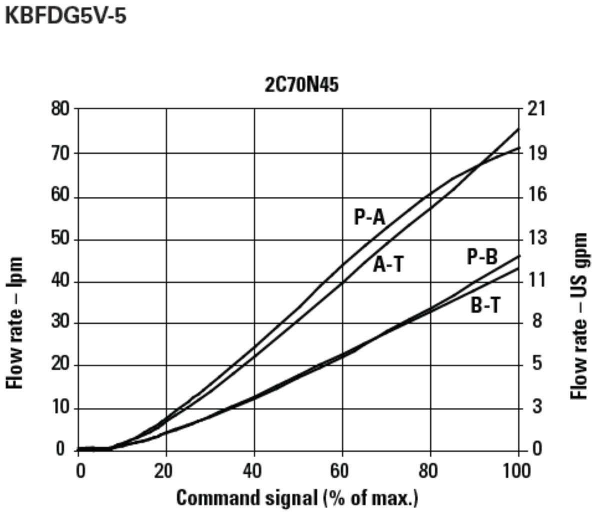

Fig. 19. 4/3 valve configuration. Source: [4]

Hydraulic motor selection

For the selection of the motor it is necessary to take into account several parameters, for this we make use of the equation number (8).

- $$\mathbf{V = W\, r\, \pi \ (8)}$$

The next step is to clear the angular velocity as shown below.

- $$\mathbf{W} = \frac{V}{r\,\pi}$$

To achieve congruence in the units it is necessary to make certain conversions as shown below.

- $$\mathbf{V} = 19\,\frac{Km}{h} * \frac{1000\,m}{1\,Km} * \frac{1\,h}{3600s} = 5,27\,\frac{m}{s}$$

- $$\mathbf{r} = 25.4\ \mathrm{cm}\,\frac{1\,m}{100\,cm} = 0,254\,m$$

Then we replace in the equation and determine the angular velocity.

- $$\mathbf{W} = \frac{5,24\,\frac{m}{s}}{0,254\,\pi} = 6,60\,\frac{rad}{s}$$

- $$6,60\,\frac{rad}{s} * \frac{1\,rev}{2\pi\,rad} * \frac{60\,s}{1\,min} = 63\,rpm$$

The following table shows the parameters necessary for the selection of the hydraulic motor, which were previously calculated (Table 1).

Table 1. Hydraulic motor selection parameters.

selección de la bomba	
Q1	9,36
Q2	5,41
Q3	5,41
Qtot	20,18
caudal	
77,693	in3/s
20,18	gpm
76,38125964	lpm
EFICIENCIA VOLUMETRICA	
0,94	
73,03142	in3/s
18,9692	gpm
71,79838406	lpm
1800	rpm
cilindrada de la bomba	
$Cilindrada = \frac{231\frac{in^3}{rev} * Q(gpm)}{N(rpm)}$	
39,88799115	cm^3/rev

The selection from the catalogue is shown below according to the theoretical data obtained (Figure 22 and 23).

Datos de desempeño del Motor

Serie	Desplazamiento	Máxima Presión de Salida Bar (PSI)	Máxima Velocidad de Giro RPM	Catálogo
Nichols™ LSHT Baja Velocidad, Alto par torsor				
110A	49 - 395 cm³/rev	170 (2500)	279 - 858	HY13-1590-004/US, EU
Torqmotors™				
Tamaño Pequeño	36 - 392 cm³/rev	140 (2030)	143 - 1141	HY13-1590-004/US, EU
Tamaño Mediano	41 - 392 cm³/rev	140 (2030)	191 - 1024	HY13-1590-004/US, EU
Tamaño Grande	81 - 1000 cm³/rev	241 (3500)	118 - 693	HY13-1590-004/US, EU

Fig. 20. Hydraulic motor selection. Source: [1].

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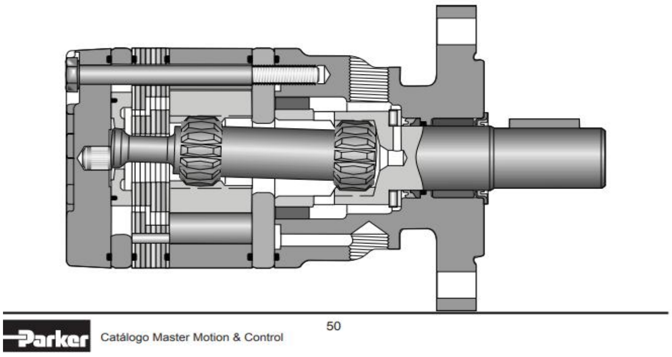


Fig. 21. Hydraulic motor selection. Source: [1].

In most of the companies to place orders it is necessary to order the products under code as shown in the following image (Figure 24).

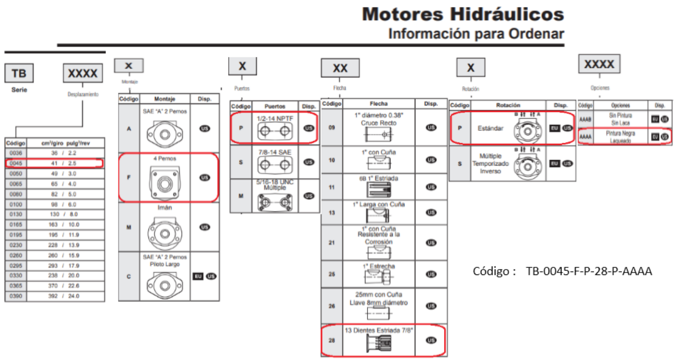


Fig. 22. Hydraulic motor selection. Source: [1]

Code: TB-0045-F-P-28-P-AAAA

Subsequently to the selection of the hydraulic motor we go to select the pipe in this case will be hose and its corresponding accessories.

For the selection of the hose it is necessary to take into account some parameters for this it is necessary to select the type of oil, in the following table shows the selected oil (Figure 25).

ANÁLISIS QUIMICO	NORMA	ISO 32	ISO 46	ISO 68	ISO 100	ISO 150
Color		Ámbar	Ámbar	Ámbar	Ámbar	Ámbar
Gravedad API		31.9	32	29.9	29	28.5
Viscosidad Cinemática a 40°C, (cSt),Min.	ASTM D445	28.8	41.4	61.2	90	135
Viscosidad Cinemática a 40°C, (cSt),Máx.	ASTM D445	35.2	50.6	74.8	110	165
Índice de Viscosidad, Min.	ASTM D 2270	90	90	90	90	85
Punto de Inflamación, °C (°F), Min.	ASTM D 92	175 (347)	185 (365)	195 (383)	205 (401)	205 (401)
Punto de Fluidez, °C (°F), Máx.	ASTM D 97	-18 (-0.4)	-15 (5)	-12 (10.4)	-12 (10.4)	-12 (10.4)
Número Total de Ácido TAN, (mg KOH/g)	ASTM D 664	0.5	0.5	0.5	0.5	0.5
Corrosión a la Lámina de Cobre, Máx.	ASTM D 130	1b	1b	1b	1b	1b
Determinación del Volumen de Espuma, (ml), Máx Secuencia I Secuencia II Secuencia III	ASTM D 892	150/0 75/0 150/0	150/0 75/0 150/0	150/0 75/0 150/0	150/0 75/0 150/0	150/0 75/0 150/0

Fig. 23ISO100 hydraulic oil. Source: [5]

The steps and data required for pipe selection are shown below.

- Kinematic viscosity of the oil.

$$V\ cm = 0.000969\ \frac{ft^2}{s}$$

- Fluid elasticity.

Ef =150000000 Pas

- Oil density.

$$d = 882\ \frac{kg}{m^2}$$

Pipe calculation

- Pipe diameter.

- $Dt = \dfrac{4*Q}{(1600*\pi*Vcm)*12}$ (9)

- $Dt = \dfrac{4*Q}{(1600*\pi*0.000969)*12}$

- $Dt = \dfrac{4*77{,}693\ \frac{in^3}{s}}{(1600*\pi*0.139536\ \frac{in^2}{s})} = 0.4431\ in$

After obtaining the data previously calculated we go to the catalog and select them as shown in the following images (Figure 26 to 29).

Fig. 24. Hose selection. Source: [6].

XXXXX-XX-XX Referencia	Manguera D.I.				Manguera D.E.	Índices de presión: Presión máx. de trabajo		Presión de. rotura mín.		min. radio de curvatura	Peso
	DN	Pulg.	Módulo	mm	mm	MPa	psi	MPa	psi	mm	kg/m
721TC-6	10	3/8	-6	9,5	20,0	28,0	4000	112,0	16000	65	0,59
721TC-8	12	1/2	-8	12,7	24,0	28,0	4000	112,0	16000	90	0,80
721TC-10	16	5/8	-10	15,9	27,0	28,0	4000	112,0	16000	100	1,10
721TC-12	20	3/4	-12	19,1	31,0	28,0	4000	112,0	16000	120	1,40
721TC-16	25	1	-16	25,4	38,0	28,0	4000	112,0	16000	150	1,99
721TC-20	32	1-1/4	-20	31,8	47,0	21,0	3000	84,0	12000	210	2,59
721TC-24	40	1-1/2	-24	38,1	53,0	17,5	2500	70,0	10000	250	2,99
721TC-32	50	2	-32	50,8	67,0	17,5	2500	70,0	10000	320	4,09

Fig. 25. Parker Hydraulic hose, terminals and equipment. Source: [6]

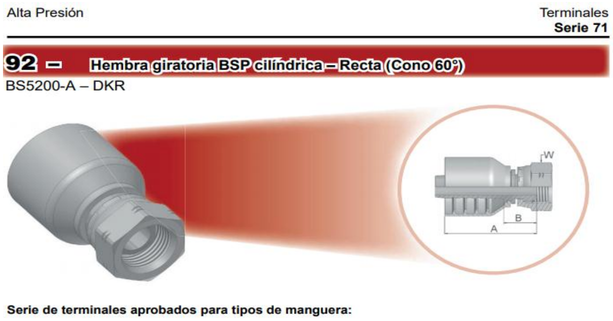
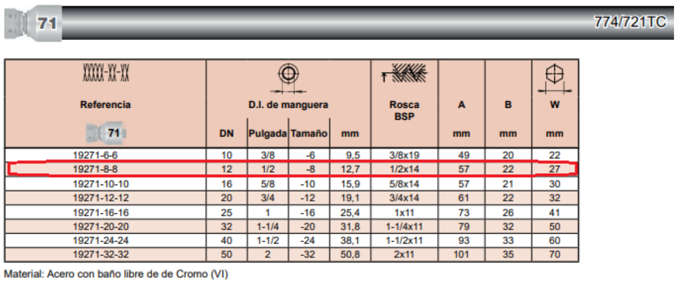


Fig. 26. Parker Hydraulic hose, terminals and equipment. Source: [6]



71 774/721TC

Referencia XXXXX-XX-XX 71	D.I. de manguera DN	Pulgada	Tamaño	mm	Rosca BSP	A mm	B mm	W mm
19271-6-6	10	3/8	-6	9,5	3/8x19	49	20	22
19271-8-8	12	1/2	-8	12,7	1/2x14	57	22	27
19271-10-10	16	5/8	-10	15,9	5/8x14	57	21	30
19271-12-12	20	3/4	-12	19,1	3/4x14	61	22	32
19271-16-16	25	1	-16	25,4	1x11	73	26	41
19271-20-20	32	1-1/4	-20	31,8	1-1/4x11	79	32	50
19271-24-24	40	1-1/2	-24	38,1	1-1/2x11	93	33	60
19271-32-32	50	2	-32	50,8	2x11	101	35	70

Material: Acero con baño libre de de Cromo (VI)

Fig. 27. Parker Hydraulic hose, terminals and equipment. Source: [6]

LOSSES AND WATER HAMMER

Elasticity of the pipe. With the previous data, we go to the supplier's catalogue and extract the following characteristics as shown in the image (6).

Et= 2*10□□ Pas

Pressure sling velocity (C). This is found with equation (10) and will be used to determine the overpressure in the system.

$$C = \frac{\sqrt{\frac{Ef}{d}}}{\sqrt{1+\frac{Ef*D}{Et*\delta}}} \quad (10)$$

$$C = \frac{\sqrt{\frac{150000000\ \text{Pas}}{882\ \frac{Kg}{m^3}}}}{\sqrt{1+\frac{150000000\ \text{Pas}*0.0221\ \text{m}}{2*1011\ \text{Pas}*0.00165\text{m}}}}$$

$$C = 412.3930\ \frac{m}{s}$$

Velocity in the pipeline.

This will be the velocity that the fluid will carry in the pipe, it will be used when determining the overpressure.

$$Vt = v * \frac{(diametro\ cilindro\ catalogo)^2}{(velocidad\ de\ onda\ de\ presion)^2}$$

$$Vt = 0.3\ \frac{m}{s} * \frac{(0.1778\ \text{m})^2}{((0.0127\ \text{m})^2}$$

$$Vt = 2.7\ \frac{m^2}{s}$$

Delta pressure.

Overpressure experienced by the system.

$$\Delta p = d * C * Vt \quad (11)$$

$$\Delta p = 882\ \frac{kg}{m^3} * 412.3930\ \frac{m}{s} * 6.4726\ \frac{m}{s}$$

$$\Delta p = 2354284.168\ \text{Pas}$$

$$\Delta p = 341.460\ \text{Psi}$$

LOSSES

To determine the losses it is necessary to calculate them one by one and finally add them up to obtain the total losses.

Friction losses

$$fr = \frac{64}{Re} \quad (12)$$

$$hf = fr * \frac{v_{tramo}{}^2}{2g} \quad (13)$$

$$hf = 0.032 * \frac{2.7\frac{m^2}{s}}{2g}$$

$$hf = 0.011\ \text{m}$$

Losses due to derivations

ht = k * Vt2/ 2g (14)

$$ht = 0.3 * \frac{2.7\frac{m^2}{s}}{2*9.81\frac{m}{s^2}}$$

ht = 0.11 m

ht totals = 0.11 *7

total ht = 0.77 m

Valve leakage

hval = k * Vt2/ 2g (15)

$$hval = 0.24 * \frac{2.7\frac{m^2}{s}}{2*9.81\frac{m}{s^2}}$$

hval = 0.089 m

hval = 0.089 * 3 = 0.267 m

By change of section.

$$hc = \left(\frac{1}{Cc} - 1\right)^2 * \frac{2.7\frac{m^2}{s}}{2g} \quad (16)$$

$$Cc = \frac{12.7\ mm}{19\ mm}$$

$$Cc = 0.66$$

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$$hc = \left(\frac{1}{0.66} - 1\right)^2 * \frac{2.7\frac{m^2}{s}}{2*9.81}$$

hc = 0.098

hc total = 0.098 * 12

hc = 1.18 m

Leakage of non-return valve.

$$han = k * \frac{v_{tramo}{}^2}{2g}$$

$$han = 2.5 * \frac{2.7^2}{2*9.81}$$

han = 0.92 m

Total losses.

htotal = hc + hval +hf +hf +han + ht

htotal = 1.18 + 0.267 m + 0.011 + 0.92 m + 0.77 m

htotal = 3.14 m

Oil hydraulic design in fluid sim

Taking into account all the research and development of the hydraulic system dimension of the truck, it was established that the most optimal design in fluid sim simulation is as follows (Figure 30).

For the design in the fluid sim hidraylic software we will have to have some main elements which would be the following ones listed by the amount of pieces needed from the mentioned reference

1-Double-acting cylinder with inlet and end-of-stroke damper
2- Double-acting cylinders
4-way manual lever valve 4/3-way with closing position
6- tanks
4-pressure relief valve
7-valve check

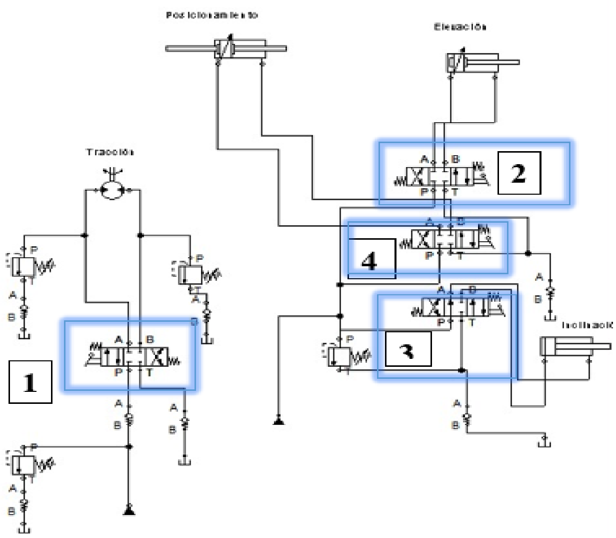

Fig. 28. Hydraulic design in fluid sim.

Hydraulic oil tank.

To select the hydraulic oil reservoir, the system flow rate is multiplied by 3 (Figure 31).

flow	
77,693	in3/s
20,18	gpm
76,38125964	lpm

76,38 lpm *3 = 231 litres

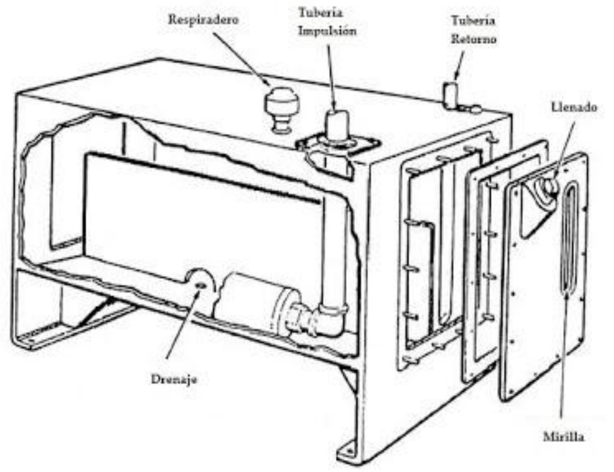

Fig. 29 Hydraulic tank. Source: [7]

Design in inventor

For this design the loader was characterized to have an approximate visualization of the vehicle, the ANSI, FEM and ISO standards were not taken into account for the passenger compartment, since the hydraulic positioning of the system was more evident, as well as the mechanisms that lift, move and tilt the load. "For the selection of materials must take into account the designer's criteria in terms of safety and cost factors, as a result an ergonomic, safe and reliable equipment was designed [8]. The analysis that was developed was based on the force versus unit strain curve of a material such as mild steel, but can be extended for various materials with errors within the acceptance range [9] (Figure 32 to 35).

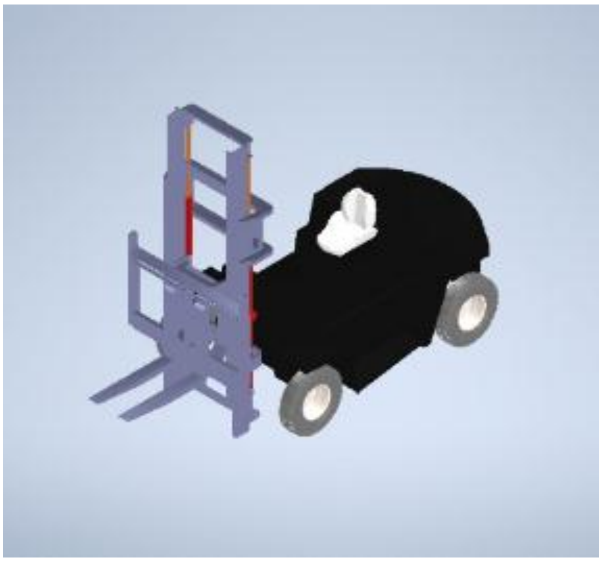

Fig. 30. CAD design.

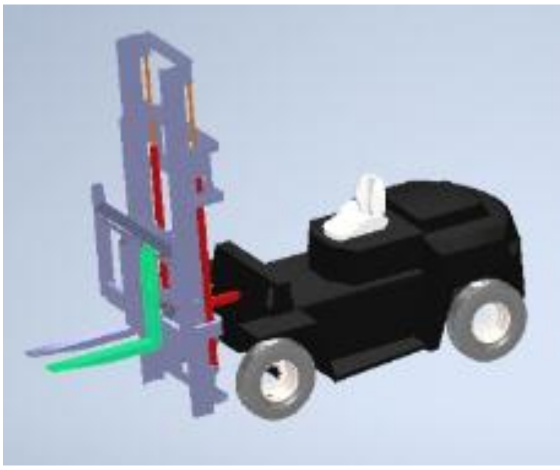

Fig. 31. CAD design.

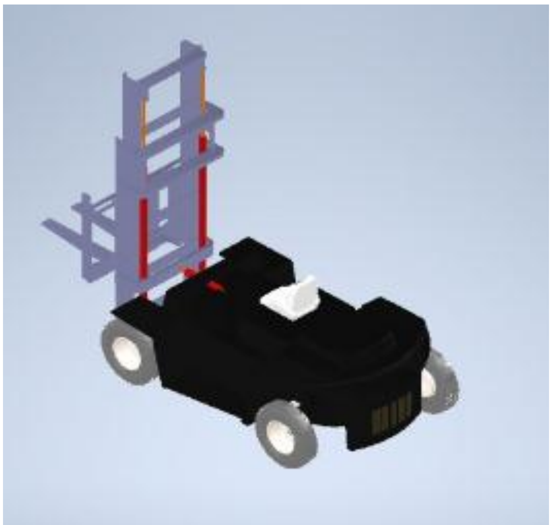

Fig. 32. CAD design.

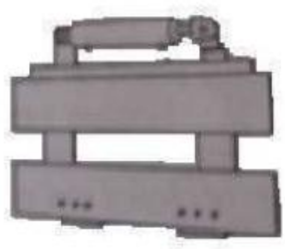

Fig. 33. CAD design guide. Source: [10]

COSTS

The final costs are shown below (Figure 36 through 40).

Forklift truck

Fig. 34. Forklift truck. Source: [11]

Forklift truck

Reference	Cost
FD30	$ 150.000.000

Cylinder 2H

Fig. 35. Hydraulic cylinders. Source: [12].

Reference	Cost
2H	$ 1.805.054

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Valve 2c

Fig. 36. 4/3 valve. Source: [12].

Reference	Cost
2C	$ 451.227

Displacement pump

Fig. 37. Hydraulic pump. Source: [12].

Reference	Cost
SAE Pallets	$ 33.000.000

Hose 721 TC

Fig. 38. Hydraulic oil transport hose. Source: [12].

Reference	Cost
721 TC	$ 2.000.000

III. CONCLUSION

For the development of this project it was necessary to investigate the operation of the forklifts as well as: trolley speed, speed in the cylinders, torque on the wheels, lifting height.
The selection of cylinders took into account the most commonly used types of clamping on forklift trucks, to ensure a machine with good performance

For the selection of components it was necessary to search and use catalogs of recognized brands in order to choose the most reliable components on the market.

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