SKRIPSI – ME091329

PERENCANAAN LIFTING OPERATION SYSTEM (HYDRAULIC SYSTEM - SPUDCAN JETTING SYSTEM - LEG MECHANISM) PADA LIFTBOAT DENGAN STUDI KASUS L/B CAMERON CLASS 200

Firman Norma Akhmad
NRP 4212 105 002

Dosen Pembimbing
Ir. Hari Prastowo, M.Sc
Taufik Fajar Nugroho, ST. M.Sc

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Nama	: Firman Norma Akhmad
NRP	: 4212 105 002
Jurusan	: Marine Engineering
Pembimbing	: Ir. Hari Prastowo, M.Sc
	Taufik Fajar Nugroho, ST, M.Sc

Abstrak

Liftboat adalah kapal yang digunakan untuk layanan offshore, memiliki kemampuan mengangkat badan kapal, dan memiliki sistem propulsi, dilengkapi dengan crane untuk mendukung aktivitas offshore. Proses pengangkatan badan kapal membutuhkan suatu mekanisme. Sistem yang perlu disediakan diantaranya spudcan jetting system untuk memfasilitasi pengangkatan spudcan dari dasar laut. Sistem lain adalah sistem mekanis pada leg sehingga memungkinkan sistem hidrolik jack-up bekerja. Sistem hidrolik jack-up digunakan untuk mengangkat atau menurunkan leg atau badan kapal. Untuk mendesain sistem ini permasalahan diidentifikasi terlebih dahulu, dengan mencari literature yang relevan maka parameter ditentukan. Sistem hidrolik jack-up di rancang pada dua tekanan, yaitu 200 bar untuk mengangkat badan kapal (88.75% motor disp @200 rpm, 89.79% pump disp @750 rpm) dan 100 bar untuk mengangkat leg (28.79% motor disp @ 300rpm, 29.27% pump disp @1200 rpm). Charge pump dibutuhkan close loop circuit, internal pump 11.89 gpm x 2 units @3700 rpm dan external pump 44.2 gpm x 1 unit @2250 rpm. Spudcan jetting terdiri dari high pressure 80 bar pada upper ring dan low pressure 12 bar untuk lower ring.

Kata Kunci: Liftboat, Hydraulic Jack-up, Spudcan Jetting, Leg Mechanism.

DESIGN OF LIFTING OPERATION SYSTEM (HYDRAULIC SYSTEM – SPUD CAN JETTING SYSTEM – LEG MECHANISM) AT LIFTBOAT CASE STUDY L/B CAMERON CLASS.

Name	: Firman Norma Akhmad
NRP	: 4212 105 002
Department	: Marine Engineering
Supervisor	: Ir. Hari Prastowo, M.Sc
	Taufik Fajar Nugroho, ST, M.Sc

Abstract

Liftboat is an offshore service vessel, self elevating and self propelled vessel equipped with crane to support offshore activities.Lifting process on self-elevating jackup (liftboat)required a mechanism to facilitate operational system. There are some system which should be provided, among of them is spudcan jetting system which facilitate spudcan extraction from seabed. Other system is mechanism attached to the leg that allow hydraulic jack-up system workable. Hydraulic jack-up systemis utilized to rise up or lowering down the leg or the hull. To design these lifting operation systems, the problem were identified and defined first. By looking relevant literature review, the paramaters to design the system is determine.Hydraulic jacking system is set under two operating pressure, 200 bar for lifting the hull(88.75% motor disp @ 200 rpm, 89.79% pump disp with 750 rpm) and100 bar for lifting the leghull(28.79% motor disp with 300 rpm, 29.27% pump disp with 1200 rpm). Charge pump is required in close loop circuit, internal charge pump provide 11.89 gpm x 2 units at 3700 rpm and external pump provide 44.2 gpm x 1 unit at 2250 rpm. Spudcan jetting have two ring pipe, high pressure 80 bar on upper ring and low pressure 12 bar for lower ring

Keywords: Liftboat, Hydraulic Jack-up, Spud Can Jetting, Leg Mechanism.

VALIDATION SHEET

DESIGN OF LIFTING OPERATION SYSTEM (HYDRAULIC SYSTEM – SPUD CAN JETTING SYSTEM – LEG MECHANISM) AT LIFTBOAT CASE STUDY L/B CAMERON CLASS

UNDERGRADUATE THESIS (*SKRIPSI*)
Submitted for Bachelor Degree (S-1)
Marine Machinery System (MMS) Program
Marine Engineering Department
Faculty of Marine Technology
Sepuluh Nopember Institute of Technology

By :
Firman Norma Akhmad
NRP 4212 105 002

Approved by Supervisors :

1. Ir. Hari Prastowo, M.Sc ( )

2. Taufik Fajar Nugroho, ST. M.Sc ( )

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Firman Norma Akhmad
NRP 4212 105 002

Approved by Chairman of Marine Engineering Department :

Ir. AA. Masroeri, M.Eng, D.Eng ( )

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ACKNOWLEDGEMENTS

All praise is due to Allah,The creator of anything in the universe, there is no deity worthy of worshipped except Allah, who has the most beatiful name and the most perfect. We praise Him and thank Him. We repent to Him and seek for His forgiveness. Only by His permission we could finish what we are planning.

I would share this happiness after completion this undergraduate thesis especially to my parents who educate me, support me, and always make du'a for their children. To my wife, Dwi Purwaningtyastuti who has a lot of patience and support, and also to my children Syafiq and Aisyah who became one of my motivation to complete this study.

I would like to thank to Bapak Ir. Hari Prastowo M.Sc and Bapak Taufik Fajar Nugroho, ST, M.Sc for given me constant guidance and advice during the study. Thank to Bapak Sutopo Purwono F. ST, M.Eng, Ph.D the chief of Marine Machinery System Laboratory, thank to Bapak Dr. Eng. Trika Pitana ST, M.Sc as the coordinator of undergraduate thesis and Bapak Ir. AA. Masroeri, M.Eng, D.Eng. as a chairman of Marine Engineering Department.

Thanks to LJ 2012 friends and all my friends in Marine Engineering Department for being together for two years and I wish all of you can success in future.

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Related to Spudcan Jetting Calculation

Q : Flow rate (m^3/s)

D : Pipe diameter (m)

v : Velocity of fluid (m/s)

tm : Minimum required thickness (in)

t : Pressure design thickness (Psi)

c : Sum of mechanical allowances (in)

P : Internal design gage pressure (Psi)

S : Stress value for material (Psi)

E : Quality factor

Y : Coefficient value table 304.1.1 (ASME B 31.3)

d : Inside diameter of pipe (in)

Related to Hydraulic Calculation

w : weight (N) (kg.m/s2)

m : mass (kg)

g : gravity acceleration (m/s2)

τ : Torque (Nm))

ω : Angular velocity (rad/s)

v : velocity (m/s)

P : Power (W) (Nm/s)

t : time (s)

l : length or distance (m)

p : Pressure (N/m2)

d : displacement of motor hyd (m3)

Vg : Maximum Motor displacement (cm^3 / rev.)

Q : Flow rate (l/min)

ηv : Motor volumetric efficiency

n : rotation per minute (rpm)

Me: Hydraulic motor torque

pHD: High pressure (bar)

pND: Low pressure (bar)

ηmh: Motor mechanical-hydraulic Eff.

V : hydraulic reservoir capacity (cm3)

DP: change in pressure (psi)

BM: bulk modulus (psi)

Δt : time duration for pressurechange (s)

W : max. allowable working pressure, bar, kgf/cm2(psi)

BM: bulk modulus (psi)

t : minimum thickness of pipe, in mm (in.).

K : Coeficient (Table 1 - ABS Guide for Liftboat)

D : Actual external diameter of pipe, in mm (in.)

S : max allowable fiber stress, N/mm2 (kgf/mm2, psi)

M : Factor from 4-4-2/Table 1 ABS Guide for Liftboat

C : allowance for threading, grooving or mech. strength

CHAPTER I
INTRODUCTION

I.1 Background

The knowledge application gained form marine engineering department can be extensive not only in common merchant ships that learned in our course but also can be applied in other ships with common basic similiarity, but of course there are some difference in details. With basic similiarity in piping system design we can apply it into liftboat, an offshore service vessel. Liftboat is self elevating, self propelled vessel equiped with crane and with relatively large open deck space this can be utilized to carry any equipments and support of various offshore mineral exploration and production or offshore construction activities. A liftboat also has the capability of rapidly raising its hull clear of the water on its own legs so as to provide a stable platform from which maintenance and construction work may be conducted

Lifting process on self-elevating jackup (liftboat) at elevation mode and sail preparation mode required a mechanism that has to be design to facilitate this operational system. There are some system which should be provided, among of them is spudcan jetting system, this system is provide to facilitate spudcan extraction from seabed. Other system that should be provided is hydraulic system, hydraulic system is utilized to rise up or lowering down the leg or the hull body, so we must design the hydraulic jacking system which able is to handle the load weight of the hull body or leg.

We hope this theses will attract the students of marine engineering to develop their knowledge into larger maritime scope, especially in offshore system.

I.2 Problems Definition

To design lifting operation system of liftboat there are some issues that should be tackle, this thesis will cover these following issues :

a. How to design spud can jetting system to ease spud can extraction from seabed before sail preparation of Liftboat with L/B cameron Class 200 as a case study ?
b. How leg mechanism is working during the lifting of Liftboat with L/B cameron Class 200 as a case study ?
c. How to design hydraulic system for lifting operation of leg/hull body of Liftboat with L/B cameron Class 200 as a case study?

I.3 Scope of Research

To keep thesis focus at the problems which mentioned previously, here it is scope that will be done in this thesis :

a. Selection type of leg and mechanism system using on it and explanation how it work
b. Piping and Instrument diagram of spud can jetting system
c. Piping and Instrument diagram of hydraulic lifting system
d. The stability of lifboat is not investigated in this thesis
e. Ship strength is not investigated in this thesis

I.4 Objective

The objective of this thesis :

a. Selection type of leg and mechanism system of Liftboat with L/B cameron Class 200 as a case study
b. Design of spudcan jetting system that technically capable for leg lifting preparation of Liftboat with L/B cameron Class 200 as a case study
c. Design of hydraulic system that technically capable for hull/leg lifting of Liftboat with L/B cameron Class 200 as a case study
d. Piping and Instrument diagram of hydraulic lifting system and spud can jetting system

CHAPTER II
LITERATURE REVIEW

II.1 An Introduction of Liftboat

A liftboat, to diferentiate with a conventional jackup drilling rig is defined as a self-elevating, self-propelled vessel equipped with at least one crane and with open deck space that can be used for multiple purposes. Liftboats are the trucks of the offshore fleet. They carry any kind of equipment necessary to do whatever the job requires. Liftboat jobs include wireline, crane operations, pipe-laying, diver support platforms, work over and offshore coiled tubing operations, temporary housing for construction and service crews, wind turbine installation and servicing, and so on. The first liftboat was designed in 1955 by brothers Lynn and Orin Dean in Violet, Louisiana, USA.

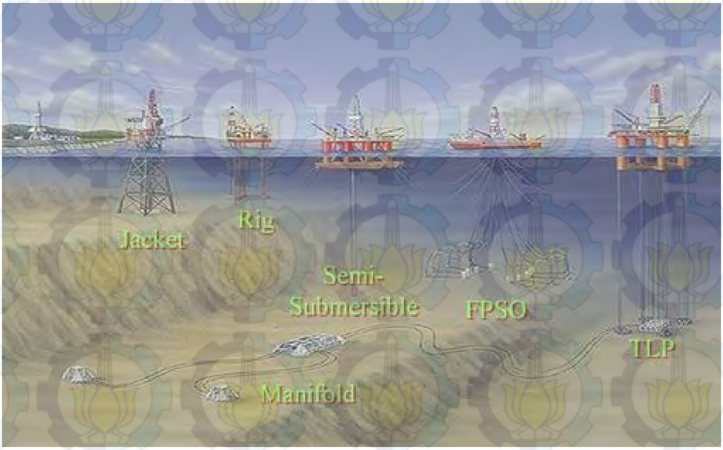


Figure 2.1 Conventional jackup rig comparassion with other offshore structure (*http://taflab.berkeley.edu/*)

A liftboat as a self-propelled, multi-purpose, self-elevating vessel some have referred to jack-up barges, lift barges, jack-boats and liftboat has close similiarity with the jack-up drilling rig. The jacking system for a liftboat is very different than the jacking system for a jack-up drilling rig. The two major differences center around speed and cycles. Speed of the liftboat jacking system is essential. While a typical jack-up drilling rig elevates at two feet per minute a liftboat could elevate at four to six feet per minute and lower the legs at 14-18 feet per minute. This gives the liftboat the ability to get on and off location significantly faster. *(Ronald E. Sanders, 2012)*

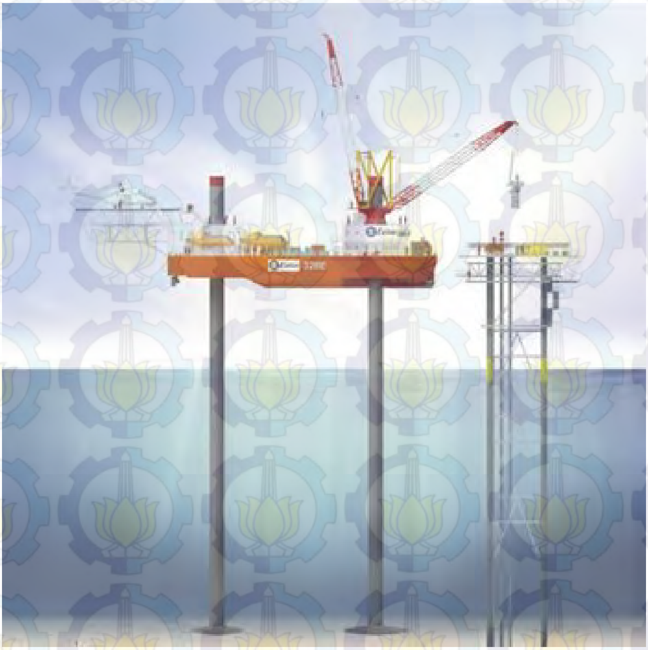


Figure 2.2 Liftboat Model
(*http://www.seatrade-global.com/*)

Advantages of Liftboats

Liftboats have historically proven to be a cost effective offshore service vessel. Major advantages include lower down time, no need for tug assists, stable work platform in the elevated position, negates the need for jack-up drilling rigs in a lot of cases, and negates the need for derrick barges in a lot of cases. They can be used for all types of offshore construction and maintenance to well intervention services like wire-line, coiled tubing, and nitrogen. They have even been used to build high-rise bridges *(Ronald E. Sanders, 2012)*

Liftboat in Different Operation

Figure 2.3 Liftboat at Windfarm Installation *(http://www.windenergynetwork.co.uk/wp-content/uploads/2011/08/Leviathan.jpg)*

Figure 2.4 Liftboat Perform Crane Operation
(*http://www.semcollc.com/200_ton_tandom_lift.jpg*)

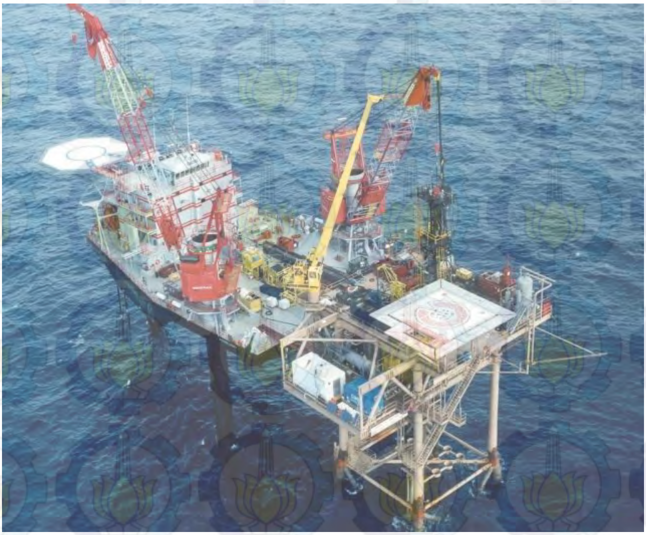


Figure 2.5 Liftboat Perform Maintenance at Fixed Platform
(*http://www.semcollc.com/legacy.jpg*)

Mode Operation of A Liftboat

There are basically four modes of a liftboat when transiting from one work site to another site. They are the sailing mode, preload mode, the elevated mode and sail preparation mode.

a. Sailing Mode

As a self propelled jack up, liftboat when moving to targeting site they could sail by their own propeller. It is not necessary to full retract the legs as long as they have enough clearance from the seabed to maintain their stability. This would increase stability of the jack-up and reduces the risk of wind overturning *(Ng Jun Jie, 2008)*

Figure 2.6 Liftboat on sailing mode
(http://theadvocate.com/)

b. Preload Mode

Liftboat units must load the soil that supports them to the full load expected to be exerted on the soil during the most severe condition and provide stable condition during working operation.

The jack-up unit has to be preloaded to simulate operating conditions. In this mode, the hull is jacked up slowly to a height no more than 5 feet above the sea level. By pumping in seawater from the surroundings to the onboard preload tanks, the hull carries extra weight apart from its own weight *(Ng Jun Jie, 2008)*

The possibility does exist that a soil failure or leg shift may occur during Preload Operations. To alleviate the potentially catastrophic results of such an occurrence, the hull is kept as close to the waterline as possible, without incurring wave impact. *(Bennet & KeppelFELS, 2005)*

Figure 2.7 Liftboat on preload mode
(http://farm4.staticflickr.com/3055/2429347089_0d888482a8_o.jpg)

c. Elevated Mode

Once preload operations are completed, the Unit may be jacked up to elevated mode. During these operations it is important to monitor the level of the hull, elevating system load and characteristics. Hydraulic lifting system is required when jack up the liftboat. Once the Unit reaches its operational air gap, the jacking system is stopped, the brakes set, and leg locking systems engaged (if installed). The Unit is now ready to begin operations *(Bennet & KeppelFELS, 2005)*

Figure 2.8 Liftboat on elevated mode
(http://offshoreliftboats.blogspot.com/)

d. Sail Preparation Mode

Once liftboat finishing their work in one site they need to move to another site. Before they can sail they need to lowering down the hull body until waterline draft level. Then they need a mechanism to ease spud can extraction from seabed, commonly the spud can are outfitted with an integrated water jetting system, this system is called spud can jetting system.

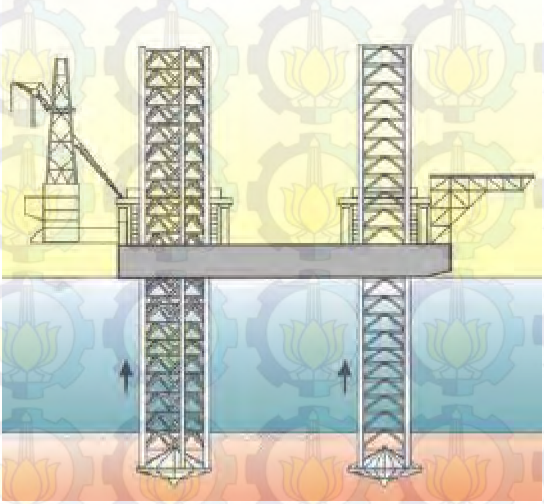


Figure 2.9 Spud can extraction from seabed
(Purwana, 2006)

II.2 Leg and Lifting Mechanism

Before go further into the detail of leg mechanism, we need to take a look of leg's type on liftboat.

a. Type of of liftboat based on amount of legs

The great majority of Jack Up Units in the world have no more than four legs, with three being the minimum required for stability.

- Three legs

  Units with 3 legs have the legs arranged in some triangular form. The main advantage of three-legged Units is that they completely eliminate the need to build extra leg(s). Furthermore, for a given hull size, they can carry more deck load in the afloat mode; and usually have a reduced number of elevating units (pinions, cylinders, etc), resulting in reduced power /maintenance requirements, and less weight.

Disadvantages of three-legged units include the fact that they require preload tankage and they have no leg redundancy. *(Bennet & KeppelFELS, 2005).*

Figure 2.10 Liftboat with three legs
(*http://www.montcooffshore.com/kayd.php*)

- Four legs
  Units with 4-legs usually have the legs arranged in some rectangular form. Four legged Units require little or no preload tanks on board. This is because four-legged Units can preload two legs at a time using the elevated weight as preload weight. This results in a savings of piping and equipment weights, and more usable space within the hull. Because of the fourth leg, these Units are stiffer in the elevated mode than a three-legged Unit. This apparent advantage may be offset by the fact that the additional leg adds wind, wave and current loads. In the afloat transit mode, the fourth leg is a disadvantage as its weight causes a direct

reduction in the afloat deck load when compared to an equivalent three-legged unit. *(Bennet & KeppelFELS, 2005).*

Figure 2.11 Liftboat with four legs
(http://offshore.laredogroup.org/petite.php)

b. Type of leg

All liftboat Units have legs. Their purpose is to provide elevation of the hull above the storm wave crest; withstand wave, current, and wind loads; and to transmit operational, environmental, and gravity loads between the hull and footings. There are two main leg types: cylindrical and trussed.

- Cylindrical legs

  Cylindrical legs are hollow steel tubes. They may or may not have internal stiffening, and may have rack teeth or holes in the shell to permit jacking of the hull up and down the legs. Cylindrical legs are currently found on Units operating in water depths less than 300 feet. The primary advantage of cylindrical legs is for Units that operate in shallow water as these Units are normally smaller and have less deck area. Cylindrical legs take up less deck area and are generally less complicated requiring less experience to construct than trussed legs. *(Bennet & KeppelFELS, 2005).*

Columnar legs and pads provide added buoyancy for the liftboat as the legs are lowered to the sea floor. This buoyancy in both the legs and pads (spud cans) helps reduce the bottom bearing pressure on the pads. In combination with the buoyancy, larger liftboat pads are designed to reduce bottom bearing pressure. This has the effect of less penetration for the typical liftboat *(Ronald E. Sanders, 2012)*

Figure 2.12 Liftboat with cylindrical legs
(*http://www.offshoreenergytoday.com/*)

- Trussed legs

The newer Units operating in water depths of 300 feet and greater all have trussed legs. The main reason for this is that cylindrical legs require more steel to provide the same resistance to environmental loads and provide the same elevated response as truss legged units. Trussed legs consist of chords and braces. In general, the braces provide the shear capacity of the leg while the chords provide the axial and flexural stiffness. One of the main benefits of the Trussed legs is that they allow for optimal steel utilization and result in lighter stiffer legs with reduced drag loads. *(Bennet & KeppelFELS, 2005).*

Figure 2.13 Liftboat with trussed legs *(http://www.offshore-industry.net/news/seajackkraken150309.htm)*

c. Type of lifting mechanism

All Jack Ups have mechanisms for lifting and lowering the hull. The most basic type of elevating system is the pin and hole system, which allows for hull positioning only at discrete leg positions. However, the majority of Jack Ups in use today are equipped with a Rack and Pinion system for continuous jacking operations. There are two types of power sources for Fixed Jacking Systems, electric and hydraulic.

Both systems have the ability to equalize chord loads within each leg. A hydraulic-powered jacking system achieves this by maintaining the same pressure to each elevating unit within a leg.

Care must be taken, however, to ensure that losses due to piping lengths, bends, etc., are either equalized for all pinions or such differences are insignificant in magnitude. For an electric powered jacking system, the speed/load characteristics of the electric induction motors cause jacking motor speed changes resulting from pinion loads, such that if jacking for a sufficiently long time, the loads on any one leg tend to equalize for all chords of that leg *(Bennet & KeppelFELS, 2005)*

There are two main basic lifting mechanism which employed in liftboat today : yoke and pin type and rack and pinion type.

- Yoke and Pin Type / Pin and Hole Type

  The functional principle is simple : The jacking system climbs up or down the platform's legs, much like a monkey. Two yokes surrounding each leg are fitted with sets of holding pins. The first such yoke snaps its pins in place in holes along the leg. The cylinders of this first yoke then lift the platform.

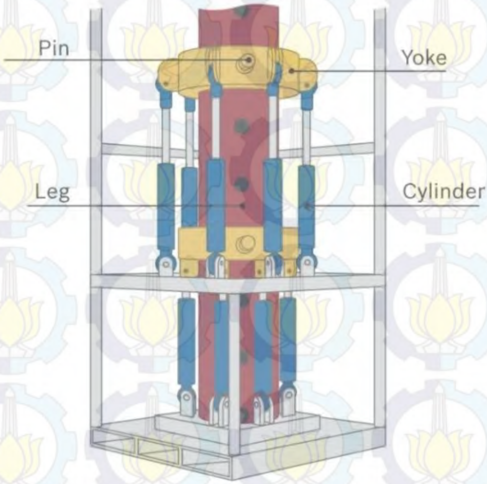


Figure 2.14 Liftboat leg attached with yoke and pin type /continuous jacking *(http://www.boschrexroth.com/)*

By the time the first yoke's cylinders have nearly reached the end of their strokes, the second yoke has moved into position to insert its set of pins in the mating holes. Those pins at the second yoke now take over the load. The pins at the first yoke retract and the cylinders raise the first yoke so that its pins can again assume the load in the next position.

This new continuous jacking system eliminates any interruptions during the lifting or lowering operation. It thus attains a constant jacking speed of one meter a minute or more. (*http://www.boschrexroth.com/*)

- Rack and Pinion Type

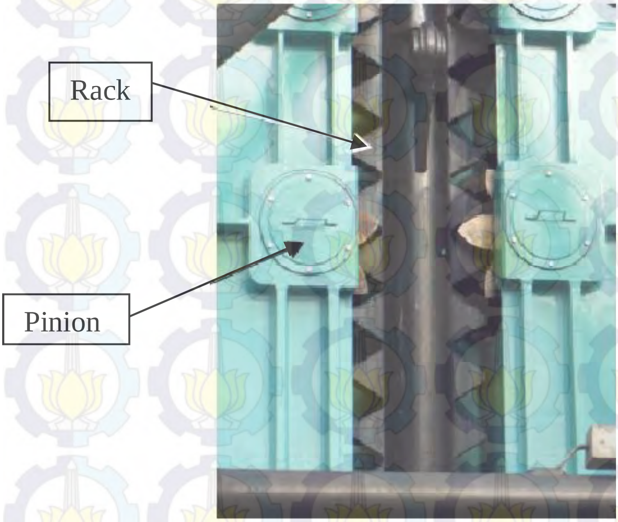


Figure 2.15 Liftboat leg attached with rake and pinion type
(Bennet & KeppelFELS, 2005)

This is probably the most used type of jacking system. Due to the harsh environment in the sea, the duration for a jacking system to be operated should first be took into consideration.

Rack and pinion jacking system is comprising of the jacking motors and the gear box, the lifting force is generated by the pinions in adjacent to rack, driving force is originated from the motors through the gear box, there are two kinds of power sources for rack and pinion system, electric and hydraulic. They are almost the same when it comes to the power source, one more advantage is that it is easy for it to regulate the speed. For an electric-driven system there must have electric motors plus Jacking MCC (Motor control center) and one frequency converter per legs. *(http://www.ship-oilrig.com)*

II.3 Spud Can Jetting System

a. Introduction

The legs and footings of a Jack up are steel structures that support the hull when the Unit is in the elevated mode and provide stability to resist lateral loads. Units with independent spud can footings have the same number of spud cans as there are legs. Spud cans are typically somewhat conical structures, with sloping tops and bottoms. The sloping top helps in sloughing off mud that may collect on top of the spud can in the event of deep penetration. The sloping bottom helps ensure that there will be some penetration, even in very stiff soils. Spud cans are normally designed to be free flooding when submerged, though they can be pumped dry for internal inspection.

There are many advantages of spud can footings. The biggest advantage is that they can be used on a great variety of seabeds. Units with spud cans have operated on seabeds of hard and soft soils, sloping bottoms (though they may be sensitive to large slopes on hard soils), and in areas where there are pipelines or other structures that must be avoided. In addition, spud cans do not require sensitive ballasting sequences or equipment and some rigs can retract the spud cans flush into the hull to permit easy dry transport of the Unit.

Units with spud cans exhibit larger bottom bearing pressure and result in increased soil penetrations when compared to mat

Units. Because of this high bearing pressure, spud cans leave impressions in areas with soft soils. If another Jack Up Unit later works in the same area, these old spud can impressions may induce horizontal forces on one or more legs if the spud cans tend to slide into the old spud can impressions. *(Bennet & KeppelFELS, 2005).*

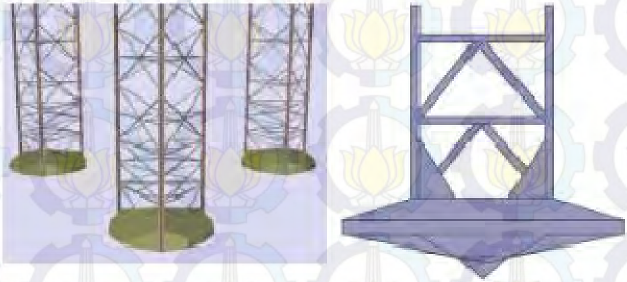


Figure 2.16 Typical profile of a spud can
(Bennet & KeppelFELS, 2005)

b. Spud Can Extraction

After an operation at a site, a jack-up rig may need to be relocated elsewhere. In this process of moving off-location, the jack-up rig is transformed back from the elevated mode to the floated one by firstly extracting the legs.

There are aided extraction methods for jack-up removal where a risk of leg retraction difficulty has been identified standard spud can extraction procedures may be supplemented with the following :

- Water jetting through spud can nozzles
- Cyclic loads
- Excavation of the soil present above the spud can

These methods may reduce breakout by either weakening the soil strength, reducing soil weight or shortening the shear plane, the methods can be applied individually, simultaneously or sequentially. The success of these methods is variable and much depends on the local ground conditions, the spudcan geometry, system efficiency and experience of the operators. *(Osborne et al., 2011)*

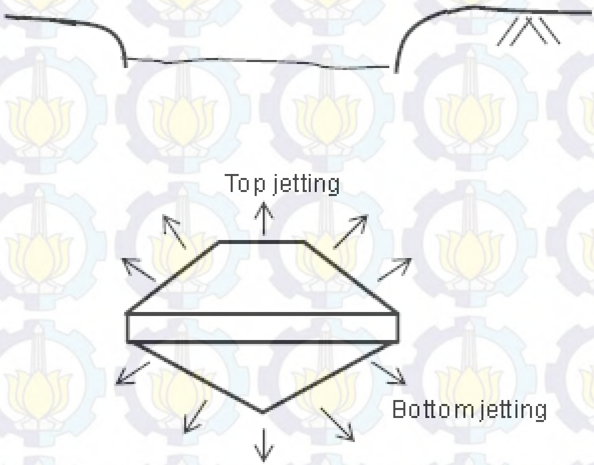


Figure 2.17 Water Jetting Method
(Osborne et al., 2011)

Application of pressurised fluid through the spudcan jetting nozzles (upper and / or lower surfaces).

- Base jetting: To reduce the extraction resistance by altering the pore pressure component of the response.
- Top jetting: To loosen the soil present around and on top of the spudcan.

Early commencement of jetting i.e. when the hull is still at floating draught is recommended. Generally jetting at low pressure /high flow rate is recommended. However, high pressure /low flow rate may be applied briefly at the commencement of jetting to ensure the nozzles are unblocked. This is best achieved by a positive displacement pump (e.g. mud pump) where the volume of flow can be monitored and nozzle unblocking can be assured. *(Osborne et al., 2011)*

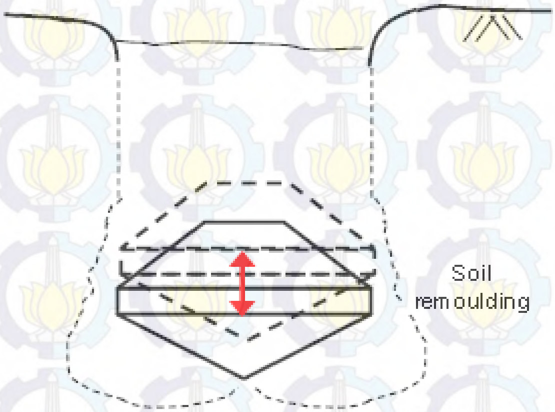


Figure 2.18 Cyclic Loading Method
(Osborne et al., 2011)

Cyclic loading using effect of small amplitude wave loading to remold the surrounding soil under cyclic loading conditions *(Osborne et al., 2011)*

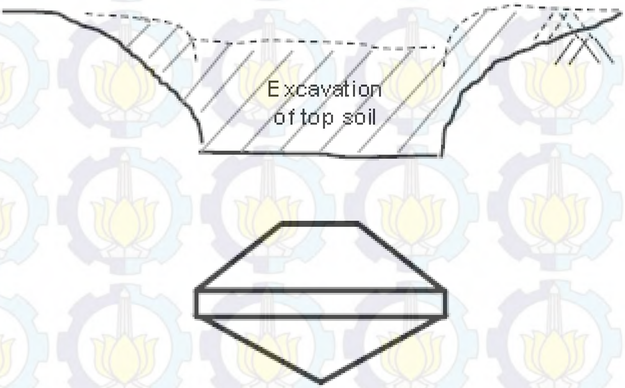


Figure 2.19 Excavation of Soil Method
(Osborne et al., 2011)

Excavation of soil method is mechanical removal of soil resting on the upper surface of the spudcan to reduce the weight of top soil and shorten the shear plane above the spudcan. However, in all but very deep penetrations or extended periods of jack-up operations this represents a comparatively minor overall contribution to the uplift resistance *(Osborne et al., 2011)*

Although there are published guidelines for the site-specific assessment of jack-up units (SNAME, 2008), the documents focus on the ability of the jack-up structure to withstand the design storm conditions while installed at the approved location. Guidance on the assessment of the ground conditions is given, together with methods for the prediction of foundation performance during installation. However, there is little guidance on the selection of engineering design parameters for the soils and the suggested bearing capacity prediction methods require highly experienced and competent geotechnical engineers for their successful application. Additionally there is inadequate guidance on the technical requirements for each site investigation which depend upon the ground conditions at each site under investigation *(Osborne, 2009)*

The extraction is completed by lowering the rig's hull into the sea in order to use its buoyancy to pull the spudcans, hopefully overcoming the soil resistance. However, depending on the level of embedment and the nature of the soil. Very few references exist in the literature to assess the reduction in soil resistance generated by water jetting *(Gaudin, C. et al., 2011)*

c. Spud Can Jetting

The leg jetting system is a water system with a series of intakes connections along the length of the leg and discharge nozzles on the spud can. Water is pumped from the rig down the leg to the nozzles that are used to break the adhesion of the spud can bottom to the soil. The leg jetting system may have the additional benefits of filling the cavity below the spud can with water so that the soil does not have to fill the cavity and it may provide some slight positive upward pressure to the bottom surface of the spud can. (*Bennet & KeppelFELS*, *2005*)

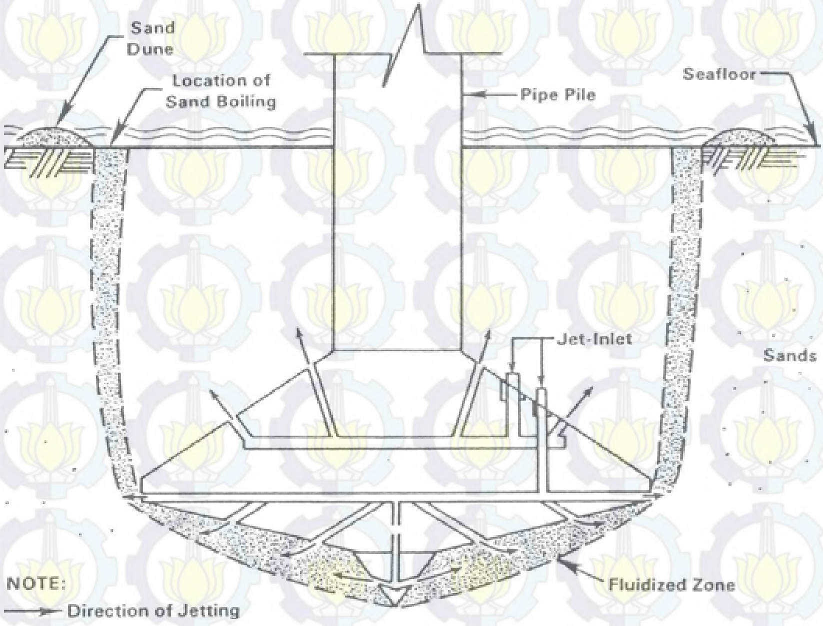


Figure 2.20 Schematic diagram of water jetting system
(Purwana, 2006)

Traditionally, the spudcans of most jack-up rigs are outfitted with an integrated water jetting system to assist in the penetration, if a certain penetration depth is required; or to ease extraction, if large uplift resistance is encountered. This system can provide highly-pressurized water in the range of 9~21 bar (KeppelFELS, pers. comm.) through an array of nozzles at both top and base of the spudcan. In the current understanding, the nozzles on the top serve to weaken the accumulated back-flow soil overlying the spudcan. At the base of the spudcan, the nozzles are mainly aimed at breaking adhesion or suction developed over the spudcan base and to fill cavity which is perceived to exist below the spudcan during pullout. *(Purwana, 2006)*

The legs are extracted by jacking down the hull into the water to generate buoyancy force typically at a rate of 0.45 m/min. To help ease the extraction process, the spudcan is traditionally equipped with water jetting system at the top and bottom sides. This water jetting essentially transfers pressurized water through an array of nozzles to break any resistance over the spudcan surface which is commonly perceived as soil adhesion of the spudcan bottom. *(Purwana, 2006)*

The jetting of water whilst the spud can is being pulled. Most modern mobile drilling rigs are equipped with a water jetting system integrated into the spud can to assist in their extraction. The water is supplied from pumps located on the hull through hoses down the jack-up legs. Typically these system have flow rates of around 60gal/min or 4 l/s. In deeply embedded clay material, where significant suction may developed at the spud can invert, the jetting aims to break the suction and reduce the reverse end bearing extraction resistance *(Randolph et al., 2011)*

d. Layout, Nummber and Diameter of Nozzle
   1) Each spud can is equipped with nozzle generally laid out in two rings
      - Below the underside of the spud can, the nozzle must be directed to facilitate destruction of the soil/spud can interface, without destroying the shear strength of the uderlying soil
      - On the upperside of the spud can, the nozzle must be directed to destroy the accumulated sediment
   2) There are normally 6 to 12 nozzle on each side of the spud can
      - Too few nozzles (3 to 4, for example, below each each spud can) locally limits the jetting effectiveness. This occurs when several nozzles are clogged

- Too many nozzle (more than 10 to 12 under each spud can) would be incompatible with the capacity of pumping units available on the platform

3) The diameter of the nozzles (of very hard steel) is generally 10 to 20 mm.
   The nozzle may be clogged by fine sediments. The installation of check valves on the jetting lines also incurs the risk of valve clogging by the sediments
4) The diameters of the water inlet pipe range from 40 to 50 mm (1.5" to 2").
5) The pumping facilities available on jackups, usable for jetting are :
   - At high pressure (8 to 10 Mpa) the drilling pump with a delivery of about 25 to 250 m^3/h
   - At low pressure (1 MPa), the fire fighting pumps with a delivery of about 100 m^3/h

(*Clarom, 1993*)

II.4 Hydraulic Lifting System

Power may be transmitted in several ways. For instance, the following are all possibilities :

- Electrics – by means of electrons at a potential
- Pneumatic – by means of a gas under a pressure
- Mechanics – by means of a mechanical structure under a load
- Hydraulic – by means of a liquid under a pressure

(Hunt et al., 1996)

Each of power transmission media has it own advantages and disadvantages. Here it is the list of major advantages and disadvantages of power transmissions.

<u>Electrical Type</u>

<u>Advantages :</u>

- Very flexible over considerable distance.

- Solenoid operation is quick and positive.
- Considerable variation of movement possible.
- Clean.
- Generally good control

Disadvantages :

- Separate power source required.
- Safety may be a problem.
- At non-rated speeds efficiency drops significantly.
- Forced air cooling required to be built in, which may cause additional costs in some environments.
- Power components tend to be larger than the equivalent hydraulic ones.
- Application dependent; prime mover maybe damage the stalling

Pneumatic Type

Advantages :

- Reasonably flexible over long lengths.
- No damage is stalling
- Clean and cheap and less hazardous medium
- Small very high speed motors are feasible
- Linear motion is extremely simple

Disadvantages :

- Lower power levels than hydraulic
- Fluid used (air) is compressible and hence the dynamic performance is restricted
- Static movement (for actuators , etc) is less precise but closed loop controlled actuators help.
- Efficiencies can be very low

Mechanical Type

Advantages :

- Simple to fit and change
- Direct application with no need of an interface or conversion device
- High efficiencies possible

Disadvantages :

- Lack flexibility
- Unable to operate over long distances
- Can be heavy

Hydraulic Type

Advantages :

- Compact for high powers.
- No damage in stalling; able to provide braking / controlled deceleration
- Actuator positioning is precise

Disadvantages :

- Complex, and sometimes difficult, pipe and hose fittings.
- Component life reduced by contaminants
- Leaks can be dangerous

(Hunt, T., et al, 1996)

Hydraulics is the science of forces and movements transmitted by means of liquids. It belongs alongside hydro-mechanics. A distinction is made between hydrostatics – dynamic effect through pressure times area – and hydrodynamics – dynamic effect through mass times acceleration.

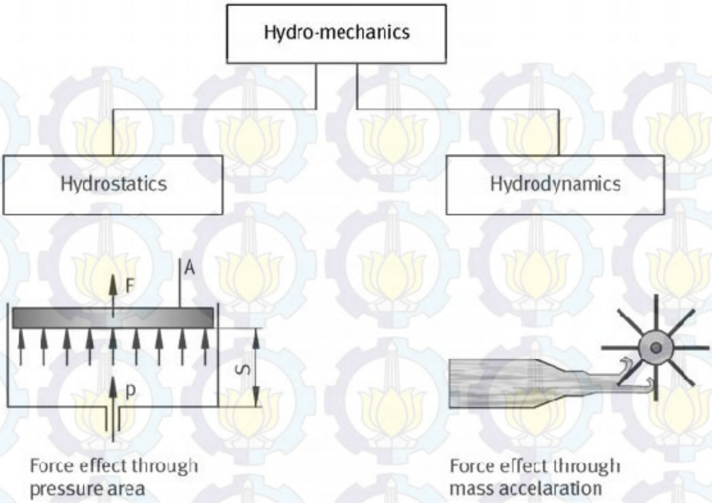


Figure 2.21 Hydromechanics classification
(Merkle, D. et al., 2003)

a. <u>Components of a hydraulic system :</u>

1. Power supply section

The power supply unit provides the necessary hydraulic power – by converting the mechanical power from the drive motor. The most important component in the power supply unit is <u>the hydraulic pump</u>. This draws in the hydraulic fluid from a <u>reservoir (tank)</u> and delivers it via a system of lines in the hydraulic installation against the opposing resistances. Pressure does not build up until the flowing liquids encounter a resistance.
<u>The oil filtration unit</u> is also often contained in the power supply section. Impurities can be introduced into a system as a result of mechanical wear, oil which is hot or cold, or external environmental influences. For this reason, filters are installed in the hydraulic circuit to remove dirt particles from the hydraulic fluid. Water and gases in the oil are also disruptive factors and special measures must be taken to remove them. <u>Heaters and coolers</u> are also installed for conditioning the hydraulic fluid. The

extent to which this is necessary depends on the requirements of the particular exercise for which the hydraulic system is being used. The reservoir itself also plays a part in conditioning the hydraulic fluid:

- Filtering and gas separation by built-in baffle plates,
- Cooling through its surface.

(Merkle, D. et al., 2003)

Selecting hydraulic pump

Factors to be considered in the selection of hydraulic pump :

a. Speed

The faster hydraulic pump is driven, the greater will be the displacement of the pump in gallons per minute up to the point of cavitation. A smaller pump driven at ahigh number of of revolution per minute may be able to deliver as much fluid in a given period of time as a large time driven at a slower number of revolution per minutes. *(Vosburgh, D., 1964)*

b. System Pressure

The desired system pressure should also be considered in selecting the hydraulic pump. Good design practice is to select a hydraulic pump that will have a maximum continnuous design pressure at least half again as much as the maximum system pressure calculated, or to say it different way, the maximum pressure required across the ports of fluid mmotor or to a hydraulic cylinder should not exceed two-thirds of the maximum rated pressure of the pump. This will provide some margin of safety to account pressure drop in the hydraulic system and take care some miscalculation by the engineer. *(Vosburgh, D., 1964)*

c. Variable or Fixed Displacement

A constant displacement pump is lower in price but there are many advantages to using the variable displacement pump. For instance, a variable displacement pump will be able to provide an efficient, variable speed, constant torque system when it is connected to a constant displacement hydraulic fluid motor. The displacement of a variable volume can be adjusted to match that demanded by the fluid motor.

When combined with a variable displacement fluid motor, a variable displacement pump will be able to provide a variable speed, constant torque drive up to a given speed, and then by changing the displacement of the fluid motor, a constant horsepower, variable speed drive could be obtainable. *(Vosburgh, D., 1964)*

d. Pump Types

Common types of hydraulic pumps to hydraulic machinery applications are :

- Gear pumps

The simplest gear-type pump uses a pair of mating gears rotating in an oval chamber to produce flow. As the gears rotate, the changing size of the chambers created by the meshing and unmeshing of the teeth provides the pumping action.

Another design uses an external rotating ring with internal gear teeth that mesh with an internal gear as it rotates. As the inner gear rotates, the tooth engagement creates chambers of diminishing size between the inlet and outlet positions to create flow. A more sophisticated variant of this principle is the gerotor pump, which has a non-concentric inner and outer rotor with different numbers of teeth. As the pair rotates, the changing volume of the space between the rotors creates the pumping action. Replacing the meshing teeth of the gerotor pump with low-friction rolling elements produces a gerotor pump.

All gear-type pumps have a fixed displacement. These pumps are relatively inexpensive compared to piston and vane type pumps with similar displacements, but tend to wear out more quickly and are not generally economically repairable. *(Erikcson, Rodney B., 2011)*

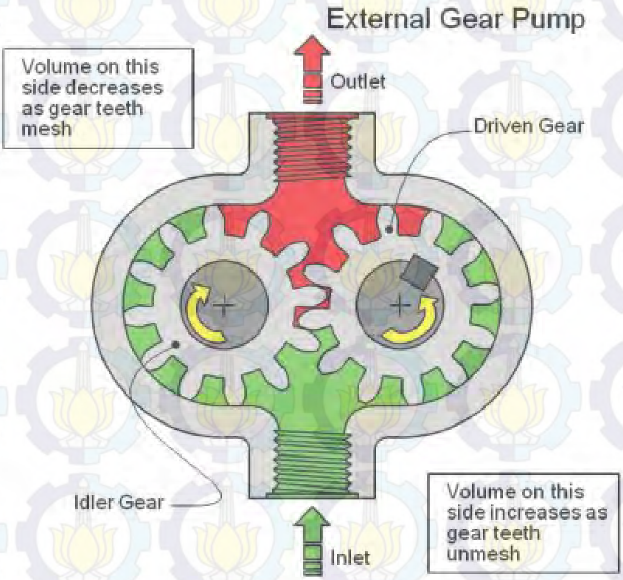


Figure 2.22 Gear Pump (External Type)
(Erikcson, Rodney B., 2011)

- Vane pumps

The most commonly encountered vane-type pump generates flow using a set of vanes, which are free to move radially within a slotted rotor that rotates in an elliptical chamber. A typical configuration uses an elliptical cam ring with the rotor spinning within in a cylindrical housing and a pair of side plates to form the pumping chambers. The changing volume of the cavity between adjacent vanes creates the pumping action as the rotor rotates.

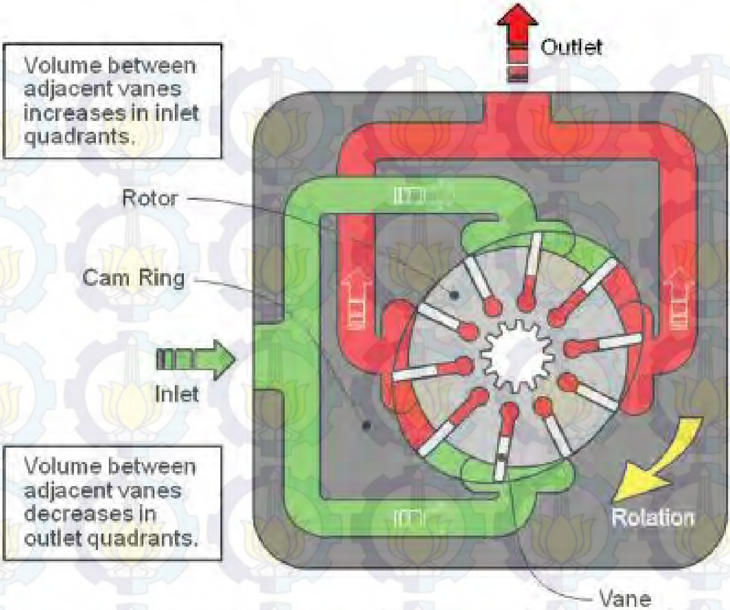


Figure 2.23 Vane Pump
(Erikcson, Rodney B., 2011)

It is possible to vary the displacement of a vane type pump, but this is not commonly done except for very specialized applications. The majority of the vane-type pumps used in industrial and mobile applications have a fixed displacement. Vane pumps can be hydraulically balanced, which greatly enhances efficiency. Some designs place the rotating group in a cartridge, which makes them very easy to repair. The entire rotating group is easily removed and replaced by simply removing the back cover, pulling out the old rotating cartridge and replacing it with a new one. Vane-type pumps are known for being very quiet in operation and producing very little vibration. *(Erikcson, Rodney B., 2011)*

- Piston pumps

Piston pumps can have the pistons arranged in a radial or axial fashion. Radial types tend to be specialized for applications requiring very high power, while axial piston pumps are available in a wide range of displacements and pressure capabilities that make them suitable for many mobile and industrial tasks. Axial-piston pumps consist of a set of pistons that are fitted within a cylinder block and driven by an angled swash plate powered by the input shaft. As the swash plate rotates, the pistons reciprocate in their respective cylinder block bores to provide the pumping action.

Axial-piston pumps are available with the input shaft and pistons arranged coaxially, or with the input shaft mounted at an angle to the piston bores. Bent axis pumps tend to be slightly more volumetrically efficient for technical reasons, but they also tend to be slightly larger for a given capacity and their shape can present packaging difficulties in some applications. A unique characteristic of a piston-type pump is that the displacement can be changed simply by changing the angle of the swash plate. Any displacement between zero and maximum is easily achieved with relatively simple actuators to change the swash plate angle. *(Erikcson, Rodney B., 2011)*

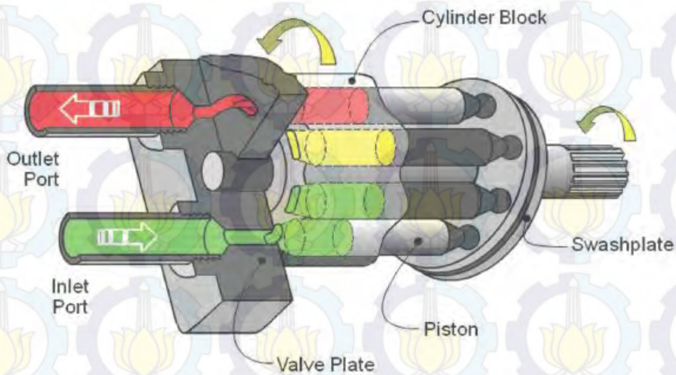


Figure 2.24 Piston Pump
(Erikcson, Rodney B., 2011)

Main difference of each type of pump

Piston-type pumps have a very good service life, provided contamination and heat are controlled. They also have the highest pressure ratings, and the significant advantage of variable displacement. This makes them the best choice for applications where high efficiency and high power density are important considerations. The ability to configure piston-type pumps with both pressure sensing and load sensing capabilities is an important advantage, particularly in mobile applications.

Vane-type pumps are widely used in constant flow/constant pressure industrial applications because they are quiet and easily repaired. They also have the unique attribute of allowing a “soft start” because vane-type pumps typically do not achieve full output at speeds below about 600 rpm. This characteristic can significantly reduce the starting current requirements of electric motors driving a vane-type pump which extends motor life.

Gear pumps are very common in constant flow/constant pressure applications on mobile equipment because of their low cost and dirt tolerance. They are also widely used as charge pumps to pressurize the inlets of piston and vane pumps because of their excellent inlet vacuum tolerance. *(Erikcson, Rodney B., 2011)*

e. Single direction or Bi-directional Pump

By controlling the volume of flow and its direction from a bi-directional pump, a hydraulic motor can be made to turn in either direction at infinitely variable speeds. A closed-loop circuit wastes very little energy. There is minimum shock when starting or changing direction because the pump starts from and passes through no-flow as it cycles. The hydraulic motor decelerates smoothly when pump flow goes to zero, slowing whatever load it drives

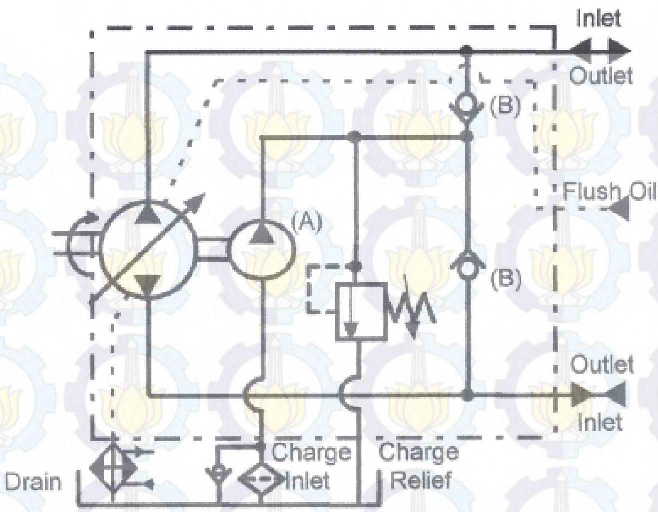


Figure 2.25 Bi-directional pump in closed-loop hydrostatic circuit (*http://hydraulicspneumatics.com/other-technologies/book-2-chapter-15-pumps*)

Axial- and radial-piston pumps can output fluid from either port while rotating in one direction. Closed-loop circuits take advantage of this feature of piston pumps. A closed-loop pump circuit sends fluid to an actuator while fluid from the same device comes back to the pump's inlet.

Normally, bi-directional pumps do not have a port piped to tank. Both ports hook directly to the cylinder or motor ports. Many bi-directional circuits operate hydraulic motors, because they accept and return nearly the same amount of fluid. The most common closed-loop circuit is the hydrostatic drive — often used on off-road equipment. (*http://hydraulicspneumatics.com/other-technologies/book-2-chapter-15-pumps*)

2. Hydraulic fluid

This is the working medium which transfers the prepared energy from the power supply unit to the drive section (cylinders or motors). Hydraulic fluids have a wide variety of characteristics. Therefore, they must be selected to suit the application in question. Requirements vary from problem to problem. Hydraulic fluids on a mineral oil base are frequently used; these are referred to as hydraulic oils. *(Merkle, D. et al., 2003)*

3. Valves

Valves are devices for controlling the energy flow. They can control and regulate the flow direction of the hydraulic fluid, the pressure, the flow rate and, consequently, the flow velocity. There are four valve types selected in accordance with the problem in question.

a. Directional control valves

These valves control the direction of flow of the hydraulic fluid and, thus, the direction of motion and the positioning of the working components. Directional control valves may be actuated manually, mechanically, electrically, pneumatically or hydraulically. They convert and amplify signals (manual, electric or pneumatic) forming an interface between the power control section and the signal control section.

Figure 2.26 Directional Control Valve
(Merkle, D. et al., 2003)

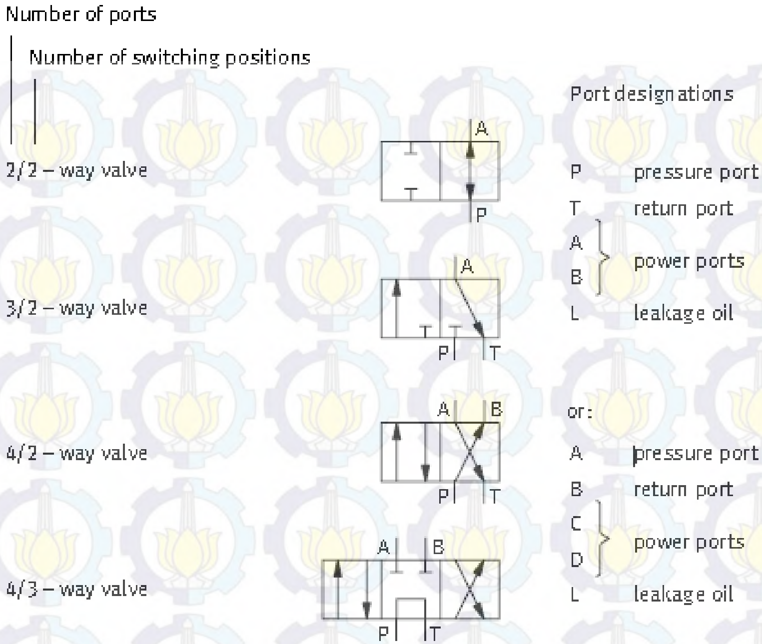


Figure 2.27 Directional Control Valve Symbol
(Merkle, D. et al., 2003)

Directional control valves are shown by means of several connected squares.

- The number of squares indicates the number of switching positions possible for a valve.
- Arrows within the squares indicate the flow direction.
- Lines indicate how the ports are interconnected in the various switching positions

(Merkle, D. et al., 2003)

b. Pressure valves
These have the job of influencing the pressure in a complete hydraulic system or in a part of the system. The method of operation of these valves is based on the fact that the effective

pressure from the system acts on a surface in the valve. The resultant force is balanced out by a counteracting spring.

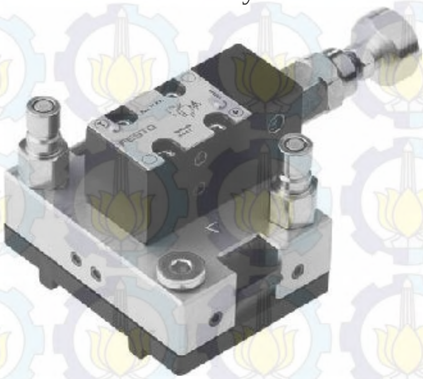
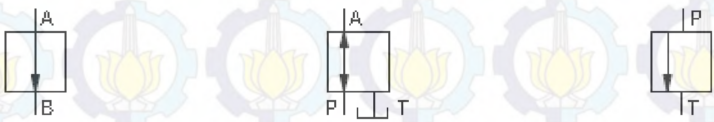


Figure 2.28 Pressure relief valve
(Merkle, D. et al., 2003)

Pressure valves are represented using squares. The flow direction is indicated by an arrow. The valve ports can be labeled P (pressure port) and T (tank connection) or A and B. The position of the valve within the square indicates whether the valve is normally open or normally closed.



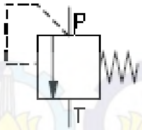
Open

Flow from P to A,
T closed

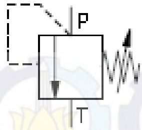
Closed

Figure 2.29 Pressure Relief Valve Symbol
(Festo Hydraulic Basic Level, 2003)

A further distinction is made between set and adjustable pressure valves. The latter are indicated by a diagonal arrow through the spring.



Set



Adjustable

Pressure valves are divided into pressure relief valves and pressure regulators :

- Pressure relief valves
  In the normally closed position the control pressure is detected at the input. This pressure acts on a valve via the control passage coming from the input on a piston surface which is held against the control pressure by a spring. If the force resulting from the pressure and the effective piston surface exceeds the spring force, the valve opens. In this way, it is possible to set the limiting pressure to a fixed value.

- Pressure regulators
  In the case of a normally open pressure regulator, the control pressure is detected at the output. This pressure is effective in the valve via the control passage on a piston surface and generates a force. This force works against a spring. The valve begins to close when the output pressure is greater than the spring force. This closing process causes a pressure drop from the input to the output of the valve (caused by the flow control). When the output pressure reaches a specified value, the valve closes completely. The specified maximum system pressure is set at the input of the valve, the reduced system pressure at the output. Thus, the pressure regulator can only be set to a smaller setting value than that set at the pressure relief valve.

(Merkle, D. et al., 2003)

c. Flow control valves

These interact with pressure valves to affect the flow rate. They make it possible to control or regulate the speed of motion of the power components. Where the flow rate is constant, division of flow must take place. This is generally effected through the interaction of the flow control valve with a pressure valve.

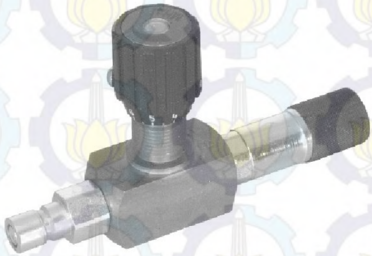


Figure 2.30 Flow control valve
(Merkle, D. et al., 2003)

In the case of flow control valves, a distinction is made between those affected by viscosity and those unaffected. Flow control valves unaffected by viscosity are termed orifices. Throttles constitute resistances in a hydraulic system. The 2-way flow control valve consists of two restrictors, one setting restrictor unaffected by viscosity (orifice) and one adjustable throttle.

The adjustable throttle gap is modified by changes in pressure. This adjustable throttle is also known as a pressure balance. These valves are depicted as a rectangle into which are drawn the symbol for the variable throttle and an arrow to represent the pressure balance. The diagonal arrow running through the rectangle indicates that the valve is adjustable. There is a special symbol to represent the 2-way flow control valve. *(Merkle, D. et al., 2003)*

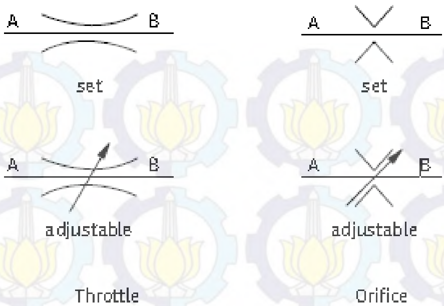


Figure 2.31 Flow control valve Symbol
(Merkle, D. et al., 2003)

d. Non-return valves

In the case of this type of valve, a distinction is made between ordinary non-return valves and piloted non-return valves. In the case of the piloted non-return valves, flow in the blocked direction can be released by a signal. The symbol for non-return valves is a ball which is pressed against a sealing seat. This seat is drawn as an open triangle in which the ball rests. The point of the triangle indicates the blocked direction and not the flow direction. Pilot controlled non-return valves are shown as a square into which the symbol for the non-return valve is drawn. The pilot control for the valve is indicated by a control connection shown in the form of a broken line. The pilot port is labelled with the letter X.

Figure 2.32 Non-return valve
(Merkle, D. et al., 2003)

Spring Loaded   Unloaded

Figure 2.33 Non-return valve Symbol
(Merkle, D. et al., 2003)

Shut-off valves are shown in circuit diagrams as two triangles facing one another. They are used to depressurise the systems manually or to relieve accumulators. In principle, wherever lines have to be opened or closed manually. *(Merkle, D. et al., 2003)*

e. Other Control valves

The control valve is one of the most expensive and sensitive parts of a hydraulic circuit.

- Sequence valves control the sequence of hydraulic circuits; to ensure that one hydraulic cylinder is fully extended before another starts its stroke, for example.
- Shuttle valves provide a logical or function.
- Pilot controlled Check valves are one-way valve that can be opened (for both directions) by a foreign pressure signal. For instance if the load should not be held by the check valve anymore. Often the foreign pressure comes from the other pipe that is connected to the motor or cylinder.
- Cartridge valves are in fact the inner part of a check valve; they are off the shelf components with a standardized envelope, making them easy to populate a proprietary valve block. They are available in many configurations; on/off, proportional, pressure relief, etc. They generally screw into a valve block and are

electrically controlled to provide logic and automated functions.

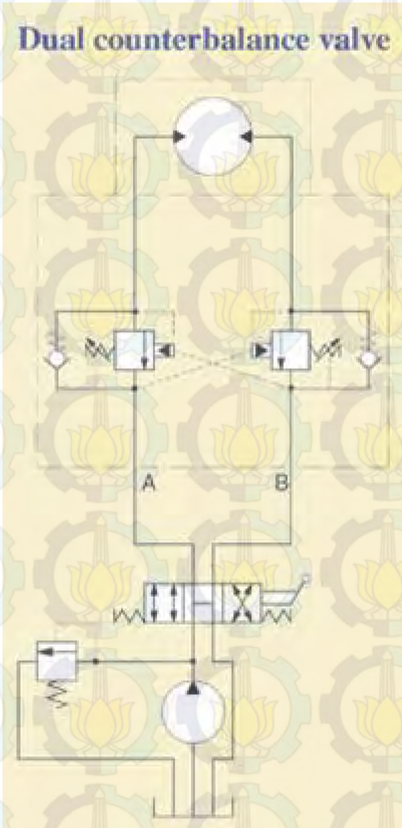


Figure 2.34 Typical motor circuit using a dual counterbalance valve, which may be used for vehicle propulsion, cab swing, or driving a winch.
(*http://hydraulicspneumatics.com/200/TechZone/ManifoldsHICs/Article/False/79464/TechZone-ManifoldsHICs*)

- Counterbalance valves are in fact a special type of pilot controlled check valve. Counterbalance valves prevent motors from drifting excessively due to control valve leakage. They can hold the load in the event of hose/tube failure, or limit overrun when a load is in a lowering or runaway mode — or the vehicle is going downhill. They provide a smooth, cushioned stop when the control valve is suddenly closed. Without a counterbalance valve there is no backpressure to hold a load on the motor or to prevent free rotation when the control valve shifts to the neutral position. Additionally, without a counterbalance valve, there is nothing to prevent motor rotation in the event of hydraulic line failure. A counterbalance keeps an actuator from running away even with variable flow rates.

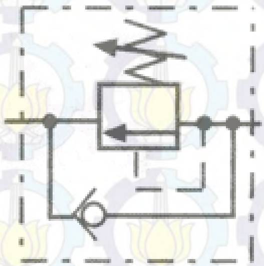


Figure 2.35 Internally piloted counterbalance valve
(http://hydraulicspneumatics.com/other-technologies/book-2-chapter-5-counterbalance-valve-circuits)

Figure 2.35 shows the symbol for an internally piloted counterbalance valve. Use an internally piloted counterbalance to hold a load back when the actuator does not need full power at the end of stroke. This type of counterbalance valve retards flow continuously, so it resists flow even after work contact stops the actuator. Note that it is necessary to adjust an internally piloted counterbalance valve every time the load changes. The following circuits show these characteristics and how to design around them.

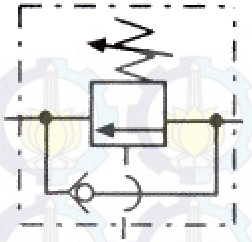


Figure 2.36 Externally piloted counterbalance valve (*http://hydraulicspneumatics.com/other-technologies/book-2-chapter-5-counterbalance-valve-circuits*)

Figure 2.36 shows the symbol for an externally piloted counterbalance valve. This valve's pilot supply is from a source other than the controlled load. An externally piloted counterbalance does not waste energy at the end of stroke and does not require adjustment for changing loads. However, an externally piloted counterbalance valve does waste a little more energy when moving the load to the work.

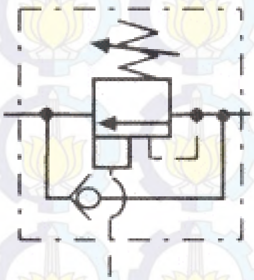


Figure 2.37 Internally/externally piloted counterbalance valve (*http://hydraulicspneumatics.com/other-technologies/book-2-chapter-5-counterbalance-valve-circuits*)

Figure 2.37 shows the symbol for an internally/externally piloted counterbalance valve. This valve has the best of both systems. As the load extends, internal pilot supply gives smooth control with little energy loss. After work contact, as system pressure builds, the external pilot fully opens the counterbalance to relieve all backpressure in the cylinder. *(http://hydraulicspneumatics.com/other-technologies/book-2-chapter-5-counterbalance-valve-circuits)*

- Hydraulic fuses are in-line safety devices designed to automatically seal off a hydraulic line if pressure becomes too low, or safely vent fluid if pressure becomes too high.
- Auxiliary valves in complex hydraulic systems may have auxiliary valve blocks to handle various duties unseen to the operator, such as accumulator charging, cooling fan operation, air conditioning power, etc. They are usually custom valves designed for the particular machine, and may consist of a metal block with ports and channels drilled. Cartridge valves are threaded into the ports and may be electrically controlled by switches or a microprocessor to route fluid power as needed. *(http://en.wikipedia.org/wiki/Hydraulic_machinery)*

4. Actuators

Actuators in hydraulic system can result in linear movement by hydraulic cylinder or in rotational movement by hydraulic motor.

a. Cylinders (linear actuators)

Cylinders are drive components which convert hydraulic power into mechanical power. They generate linear movements through the pressure on the surface of the movable piston.

The hydraulic cylinder converts hydraulic energy into mechanical energy. It generates linear movements. For this reason, it is also referred to as a “linear motor”. There are two basic types of hydraulic cylinder *(Merkle, D. et al., 2003)*

- Single-acting cylinders
  In single-acting cylinders, only the piston side is supplied with hydraulic fluid. Consequently, the cylinder is only able to carry out work in one direction

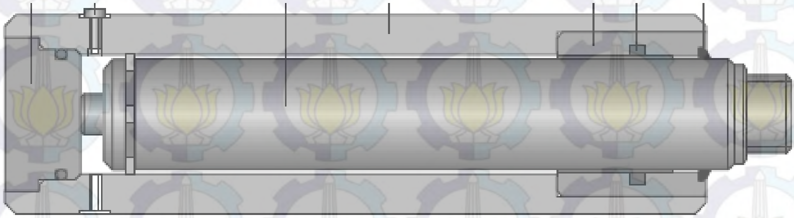


Figure 2.38 Single Acting Cylinder
(Merkle, D. et al., 2003)

- Double-acting cylinders
  In the case of double-acting cylinders, both piston surfaces can be pressurized. Therefore, it is possible to perform a working movement in both directions.

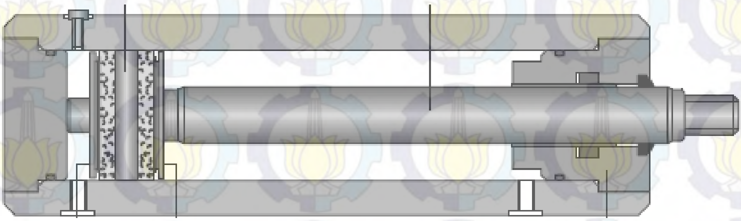


Figure 2.39 Double Acting Cylinder
(Merkle, D. et al., 2003)

b. Motors (rotary actuators)

Motor is the name usually given to a rotary hydraulic actuator. Motors very closely resemble pumps in construction. Instead of pushing on the fluid as the pump does, as output members in the hydraulic system, they are pushed by the fluid and develop torque and continuous rotating motion. Since both inlet and outlet ports may at times be pressurized, most hydraulic motors are externally drained.

Selecting hydraulic motors

Factors to be considered in the selection of hydraulic motors :

- Displacement is the amount of fluid which motor will accept in turning one revolution or in other words, the capacity of one chamber multiplied by the number of chambers the mechanism contains. Motor displacement is expressed in cubic inches per revolution (cu.in/rev) *(Sperry, R., 1970)*
- Torque is the force component of the motor's output. It is defined as a turning or twisting effort. Motion is not required to have torque, but motion will result if the torque is sufficient to overcome friction and resistance of the load. The torque is always present at the driveshaft, but is equal to the load multiplied by the radius, A given load will impose less torque on the shaft if the radius is decreased. However, the larger radius will move the load faster for a given shaft speed. Torque is usually expressed in pund inches. *(Sperry, R., 1970)*
- Pressure required in a hydraulic motor depends on the torque load and the displacement, A large displacement motor will develop a given torque with less pressure than a smaller unit. The size or torque rating of a motor usually is expressed in pound inches of torque per 100 psi of pressure. *(Sperry, R., 1970)*
- Fixed or Variable Displacement Motor

  Hydraulic motors can be either fixed- or variable-displacement,. Fixed-displacement motors drive a load at a constant speed while a constant input flow is provided. Variable-displacement motors can offer varying flow rates by changing the displacement. Fixed-displacement motors provide constant torque; variable-displacement designs provide variable torque and speed.

(*http://www.mobilehydraulictips.com/hydraulic-motors/*)
Variable displacement motors have variable torque and speed. With the input flow and operating pressure remaining constant, varying the displacement can vary the ratio between torque and speed to suit the load requirements. (*http://www.freestudy.co.uk/*)

- Type of motor hydraulic
  Three common types of hydraulic motors are used most often today with a variety of styles available among them, they are gear, vane and piston motors. Three different types of motors have different characteristics. Gear motors work best at medium pressures and flows, and are usually the lowest cost. Vane motors, on the other hand, offer medium pressure ratings and high flows, with a mid-range cost. At the most expensive end, piston motors offer the highest flow, pressure and efficiency ratings.
  (*http://www.mobilehydraulictips.com/hydraulic-motors/*)

b. Circuit in hydraulic system

1. Open Loop Circuit

In open loop circuit applications, the pump draws fluid from a reservoir and pushes this fluid into the hydraulic system. After passing through the control valve circuitry and the actuator, the fluid returns to the storage reservoir. Typically, the reservoir is sized so that it will hold a minimum of three times the volume that can be displaced by the pump in one minute. For example, a 30-gallon per minute pump would be mated with a 90-gallon reservoir. An open loop pump only pumps fluid in one direction. For this reason, an open loop pump is normally supplied with a large diameter low pressure inlet port and a smaller high pressure outlet port.

In open loop circuit design, the direction of the actuator's motion must therefore be accomplished by the use of directional control valves. Open loop hydraulic circuits are the most common in movable bridge applications. This is because most hydraulic movable bridges are actuated by hydraulic cylinders, which require large volumes of fluid as well as differential flow rates in and out of the hydraulic cylinders.

Another advantage of the open loop circuit is that several different actuator functions can be performed simultaneously from a single pump. A disadvantage of the open loop circuit design is its relatively large size and weight due to the large volume of oil required. *(Wisconsin Dept. of Transportation, 2011)*

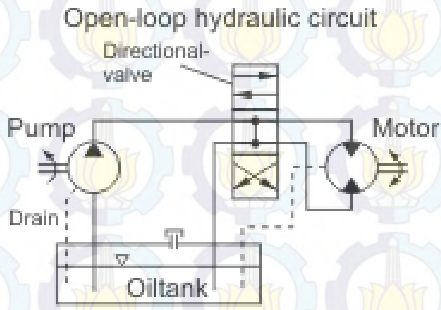


Figure 2.40 Open loop hydraulic circuit
(http://en.wikipedia.org/wiki/Hydraulic_drive_system)

2. Closed Loop Circuit

In a closed loop circuit design a single hydraulic pump is used to drive one or more hydraulic motors. The closed loop circuit is not viable for hydraulic cylinder applications because of the different fluid volume displacements during extension and retraction. The fluid that passes through the actuator is returned directly to the low pressure side of the pump. For proper operation, the pump must receive the same quantity of oil at its inlet as it is pumping from its outlet.

A charge pump is always used in a closed loop hydraulic circuit. The charge pump is usually a small fixed displacement pump (usually about 15% of the displacement of the main pump). The charge pump always works on the low pressure leg of the main loop pumping filtered fluid into the loop. The pressure in the low pressure leg is maintained at a value of usually between 100 to 300 psi by a relief valve. During operation, the main pump control can cause the pump's displacement to "go over center," which means that the main pump can pump high pressure oil from either of its two main ports. In other, words, it can cause a clockwise or counterclockwise flow of fluid through the main loop plumbing. This, in turn, will allow the actuator to operate in either direction of rotation. *(Wisconsin Dept. of Transportation, 2011)*

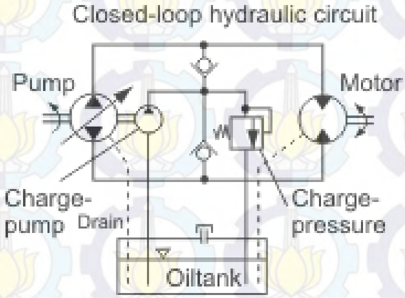


Figure 2.41 Closed loop hydraulic circuit
(http://en.wikipedia.org/wiki/Hydraulic_drive_system)

c. Hydrostatic Transmision

A hydrostatic transmission consist of a variable displacement pump and fixed or variable displacement motor, operating together in a closed circuit. In a closed circuit, fluid from the motor outlet flows directly to the pump inlet, without returning to the tank.

As well as being variable, the output of the transmission pump can be reversed, so that both the direction and speed of motor rotation are controlled from within the pump. This eliminates the need for directional and flow (speed) control valves in the circuit.

Because of the the pump and motor leak internally, which allows fluid to escape from the loop and drain back to the tank, a fixed displacement pump called a charge pump is used to ensure that the loop remains full or fluid during normal operation. The charge pump is normally installed on the back of the transmission pump and has an output of at least 20% of the transmission pump's output

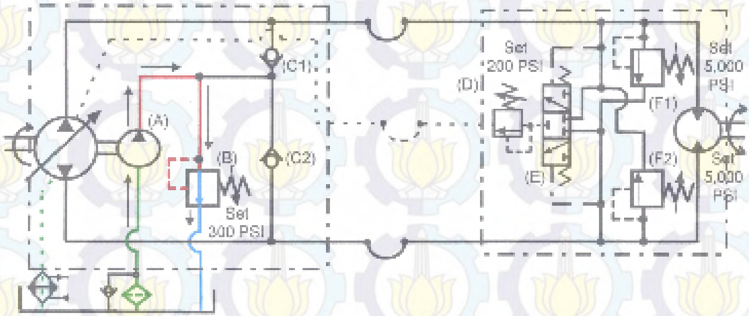


Figure 2.42 Typical hydrostatic transmission circuit (http://hydraulicspneumatics.com/other-technologies/book-2-chapter-15-pumps)

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In practice, the charge pump not only keeps the loop full of fluid, it pressurized the loop between 110 and 360 Psi, depending on the transmission manufacturer. A simple charge pressure circuit comprises the charge pump, a relief valve and two check valves, through which the charge pump can replenish the transmission loop. Once the loop is charged to the pressure ratingof the relief valve, through the case of the pumpor motor or both and back to tank. (www.insidersecretstohydraulic.com/newsletters/issue13.html)

1. Operating Principle

The operating principle of HSTs is simple: a pump, connected to the prime mover, generates flow to drive a hydraulic motor, which is connected to the load. If the displacement of the pump and motor are fixed, the HST simply acts as a gearbox to transmit power from the prime mover to the load. The overwhelming majority of HSTs, however, use a variable-displacement pump, motor, or both - so that speed, torque, or power can be regulated.

HSTs offer many important advantages over other forms of power transmission. Depending on its configuration, an HST:

- transmits high power in a compact size
- exhibits low inertia
- operates efficiently over a wide range of torque-to-speed ratios
- maintains controlled speed (even in reverse) regardless of load, within design limits
- maintains a preset speed accurately against driving or braking loads
- can transmit power from a single prime mover to multiple locations, even if position and orientation of the locations changes
- can remain stalled and undamaged under full load at low power loss
- does not creep at zero speed
- provides faster response than mechanical or electromechanical transmissions of comparable rating
- can provide dynamic braking.

2. Four functional types of HSTs (Hydrostatic Transmissions)

The configuration of an HST - whether it has a fixed- or variable displacement pump, motor, or both - determines its performance characteristics.

a. Fixed displacement pump – fixed displacement motor

The simplest form of hydrostatic transmission uses a fixed-displacement pump driving a fixed-displacement motor. Although this transmission is inexpensive, its applications are limited, primarily because alternative forms of power transmission are much more energy efficient. Because pump displacement is fixed, the pump must be sized to drive the motor at a fixed speed under full load. When full speed is not required, fluid from the pump outlet passes over the relief valve. This wastes energy in the form of heat.

b. Variable displacement pump – fixed displacement motor

Using a variable-displacement pump instead of one with a fixed displacement creates a constant torque transmission. Torque output is constant at any speed because torque depends only on fluid pressure and motor displacement. Increasing or decreasing pump displacement increases or decreases motor speed, respectively, while torque remains fairly constant. Power, therefore, increases with pump displacement.

c. Fixed displacement pump – variable displacement motor

Using a variable-displacement motor with a fixed-displacement pump produces a transmission that delivers constant power. If flow to the motor is constant, and motor displacement is varied to maintain the product of speed and torque constant, then power delivered is constant. Decreasing motor displacement increases motor speed but decreases torque, a combination that maintains constant power.

d. Variable displacement pump – variable displacement motor

The most versatile HST configuration teams a variable-displacement pump with a variable-displacement motor. Theoretically, this arrangement provides infinite ratios of torque and speed to power. With the motor at maximum displacement, varying pump output directly varies speed and power output while torque remains constant. Decreasing motor displacement at full pump displacement increases motor speed to its maximum; torque varies inversely with speed, and horsepower remains constant.

In Range 1, motor displacement is fixed at maximum; pump displacement is increased from zero to maximum. Torque remains constant as pump displacement increases, but power and speed increase.

Range 2 begins when the pump reaches maximum displacement, which is maintained while the motor's displacement decreases. Throughout this range, torque decreases as speed increases, but power remains constant. (Theoretically, motor speed could be increased infinitely, but from a practical standpoint, it is limited by dynamics.) *(http://hydraulicspneumatics.com/200/TechZone/HydraulicPumpsM/Article/False/6450/TechZone-HydraulicPumpsM)*

3. Loops in close circuit transmition

There are two primary fluid loops in a closed circuit transmission, each having a distinct design function and associated circuit elements. These are the main system power loop and the charge cooling loop. Fluid is transferred between the loops by leakage, and if so equipped, by the loop flushing shuttle. Fluid quality is controlled primarily by the charge/cooling loop, but the demand for high quality fluid is in the power loop. The loop flushing shuttle is the circuit element which is available to control this transfer of quality fluid.

The loop flushing shuttle removes some set volume of fluid from the power loop which must be made up by the charge pump. The fluid removed is presumably “dirty” and/or “hot” while the make-up fluid is cool and clean. This greatly improves the heat and contamination transfer out of the power loop, improving fluid quality and extending component life. While some hydrostatic systems can operate satisfactorily without a loop flushing shuttle, its addition can reduce circuit problems where the power loop generates heat or contaminants. *(Sauer Danfoss Application Manual, Transmission Circuit Recommendation,1997)*

4. Charge Pump

The charge pump is a critical component of the hydrostatic transmission. It is the heart of the hydrostatic transmission, for without charge flow and charge pressure, the transmission will cease to function. The primary function of the charge pump is to replenish fluid lost through leakage. In closed circuit hydrostatic systems, continual internal leakage of high pressure fluid is inherent in the design of the components used in such a system, and will generally increase as the displacements of the system’s pumps and motors increase. This “make up” fluid from the charge pump is added to the low pressure side of the closed circuit to keep the lines full of fluid and avoid cavitation at the pump. *(Sauer Danfoss Application Manual, Driveline Components, 1997)*

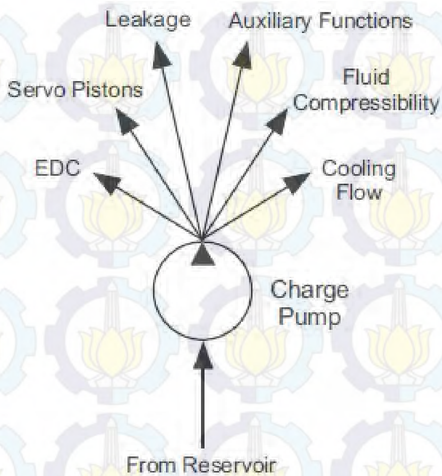


Figure 2.43 Charge pump functions
(Sauer Danfoss Application Manual, Driveline Components, 1997)

The charge pump functions associated with its volume flow rate are:

- replenish loop fluid lost through volumetric inefficiency of the pump and motor(s).
- replenish loop fluid lost through the loop flushing valve.
- provide make-up fluid in the loop for load-induced bulk modulus effects.
- provide flow to activate the servo control piston (on units with servo controls).
- provide constant flow to the Electric Displacement Control (EDC) for proper operation (on units with EDC's).
- provide a flow source for auxiliary circuit functions such as releasing parking brakes or shifting motor displacement.

The charge pump functions associated with its charge pressure are:

- maintain correct low loop pressure to ensure sufficient "hold down" forces on the rotating groups.
- provide sufficient pressure to activate the servo control system (on units with servo controls).
- control loop flushing flow by the differential pressure between the charge relief valve and the loop flushing relief valve.
- provide a pressure source for auxiliary circuit functions such as parking brake release.

(Sauer Danfoss Application Manual, Transmission Circuit Recommendation, 1997)

System conditions must be considered carefully in order to calculate the charge pump flow demand. Factors such as shaft speeds, system pressures, and system temperature all influence the leakages within the system.

An often overlooked, but extremely important aspect of the charge flow demand is the effect of fluid compressibility or the bulk modulus effect. The bulk modulus effect occurs when rapid system pressure spikes compress the fluid in the high pressure side of system. This results in an instantaneous reduction of volume of the return flow in the low pressure side of the system that must be made up by the charge pump in order to maintain a proper charge pressure.

Factors that influence the magnitude of the bulk modulus effect include: the length and size of the pressure conduits (which determine the volume of the fluid being compressed), the rise-rate of the pressure spike, the magnitude of the pressure spike, and the bulk modulus of the fluid and its sensitivity to pressure. *(Sinclair, B., IFPE, Paper 28.2)*

5. One Pump – Multi Motor Systems

One pump, multi-motor hydrostatic transmission systems can offer an advantage in installed cost and space over multi-pump, multi-motor systems. However, experience has shown that general rules of thumb for charge pump sizing, such as the “10% rule”, do not predict charge pump displacements that are capable of maintaining an adequate charge pressure during all vehicle operating conditions.

The charge pump is a critical component of the hydrostatic transmission, but also a large source of parasitic loss in a hydrostatic system. The primary function of the charge pump is to prevent cavitation in the pump by replenishing hydraulic fluid that is lost due to leakage. Additionally the charge pump must be capable of supplying fluid for the pump and motor controls and servo systems, heat dissipation via loop flushing flow at the motors, and the effects of fluid compressibility.

Many closed loop hydrostatic systems rely on system leakage to bring in fresh, cooled hydraulic fluid from the reservoir. This flow insures that the temperature of the fluid in the loop does not exceed the maximum temperature limits specified by the component manufacturers. However, in many systems, this is not adequate, especially when the system is operating at high speed and low system delta pressures. In these applications, there is a need to provide additional cooling flow into the loop to limit temperatures. *(Sinclair, B., IFPE, Paper 28.2)*

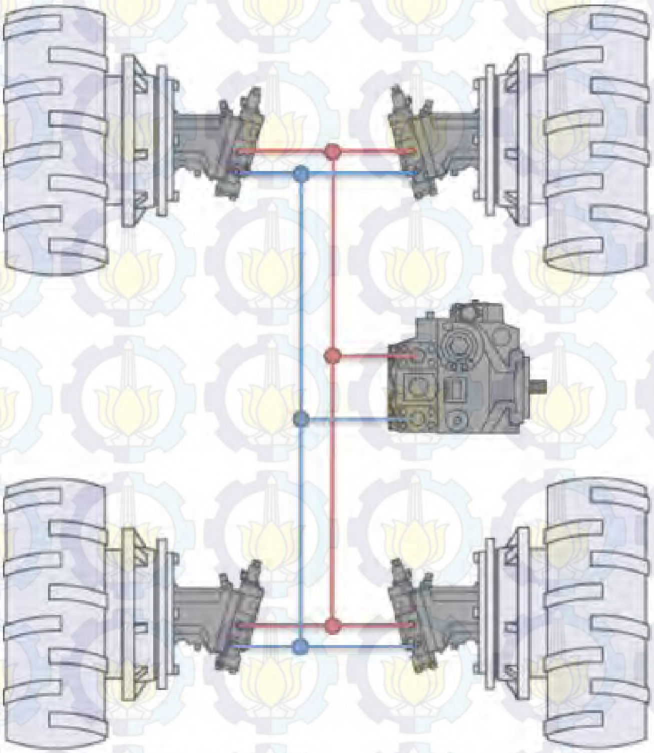


Figure 2.44 Sample of hydrostatic transmission one pump-multi motor *(Brent Sinclair -Danfoss Power Solutions presentation)*

Balancing loop flushing requirements and charge pump sizing is an important task for the system designer. If the loop flushing flow is undersized, the system will overheat. If the loop flushing flow is too high, the system will remain cool, but it will require a larger charge pump displacement and result in reduced system efficiency and higher operating costs. *(Sinclair, B., IFPE, Paper 28.2)*

d. <u>Hydraulic Brake</u>

Brakes are to be designed to engage automatically in the event of failure of power supply to the motor (fail-safe type). The brake holding capacity is to be at least equal to 120% of the maximum required brake torque associated with the maximum rated load applied to the climbing pinion from all loading conditions *(ABS Guide For Building and Classing Liftboat 2009, (Part.4, Chap.4, Sect. 13.3)*

Spring apply, hydraulic release (SAHR) brake circuits can provide service, emergency aand parking brake functions, requiring less hardware as compare to a conventional hydraulically actuted brake circuit. Spring applied hydrauuc release brake system are becoming increasingly important to off-highway equipment designers and engineers. A conventional hydraulic brake actuation system may required indeendent service, secondary, and parking brake circuits to provide braking in the event of any single failure in the service brake system. Spring apply hydraulic release brake system, on the other hand may consist of a single circuit, providing service, secondary and parking functions wit a common brake(s). The actuation circuit use to control this brake will require limited control hardware and plumbing. Consequently, this brake system will be easy to maintain, troubleshoot and is cost effective. As the name implies, the spring applied hydraulic release brake is mechanically actuated by springs and is dependant on hydraulic pressure to keep it released. Therefore, the brake actuation circuit, in a normal released mode, must maintain sufficient pressure to allow the spring applied hydraulic release brake to fully release. *(Middendorf, R.P. 1992)*

Wet brakes used in applications where the package is exposed to severe duty or to adverse environmental conditions such as marine winches or mining vehicles *(Mico, 2013)*

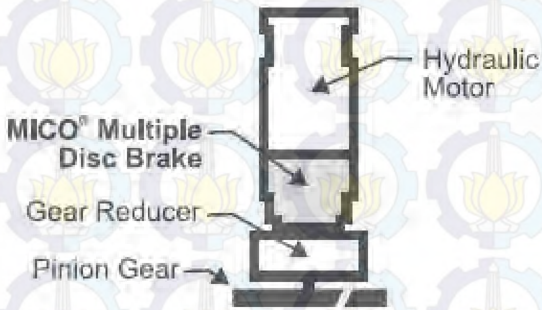


Figure 2.45 Typical brake application
(Mico, 2013)

e. <u>Planetary Gearbox</u>

Most jacking systems are run by hydraulic motors through a planetary gear system that turns a pinion gear that drives a rack attached to the leg. The jacking speed of the vessel's legs is low because of the large gear reduction necessary for the planetary drives to lift the huge load of the vessel and all its cargo out of the water. This kind of force is not necessary when lowering the legs. In fact, up to the point where the legs' displacement equals the legs' weight and it becomes buoyant the pinions, planetary drives and motors are acting more as brakes than drives. A trade can occur here and motors are acting more as brakes than drives. *(US 20100155682 A1 Patent)*

Planetary gear sets contain three major components or members. They are :

- The Ring Gear or Annulus which has internal teeth and wraps around the entire assembly.
- The Sun Gear which is the smallest gear and sits in the center of the assembly. The planetary pinions orbit around the sun gear, hence the name of the gear set.
- The Planetary Carrier which holds a set of Planetary Pinion Gears. The Pinion Gears interact with the Ring Gear and the Sun Gear at the same time.

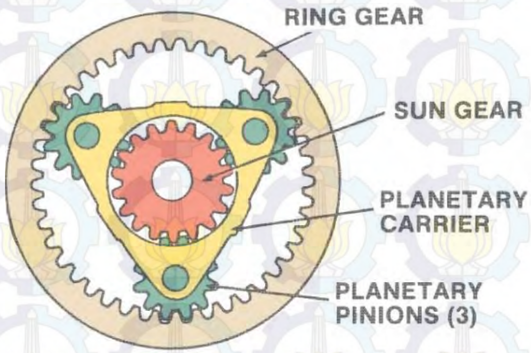


Figure 2.46 Planetary gear set components
(*https://wikis.engrade.com/planetarygearsetsoperati*)

Inside the automatic transmission, the planetary gear set provides the necessary forward and reverse gear ratios. Some transmissions use more than one planetary gear set. The layout of planetary gears is similar to the solar system, with planet pinion gears orbiting around a sun gear. The ring gear surrounds the entire gear set

Each member of the planetary gear set can spin (revolve) or be held at rest. Any one of the three members can be used as the input or driving member. At the same time, another member may be held at rest or stationary. Depending on which member is the input, which is the output, and which is held, either a torque increase (underdrive) or a speed increase (overdrive) condition will be produced. A reverse direction can also be produced if the planetary carrier is held stationary. The table below illustrates how several gear ratios can be achieved using a simple planetary gear set. *(https://wikis.engrade.com/planetarygearsetsoperati)*

No	Sun Gear	Carrier	Ring Gear	Speed	Torque	Direction
1.	Input	Output	Held	Max. Reduction	Increase	Same as input
2.	Held	Output	Input	Min. Reduction	Increase	Same as input
3.	Output	Input	Held	Max. Increase	Reduction	Same as input
4.	Held	Input	Output	Min. Increase	Reduction	Same as input
5.	Input	Held	Output	Reduction	Increase	Reverse of input
6.	Output	Held	Input	Increase	Reduction	Reverse of input
7.	When two members are held together, speed and direction are the same as input. Direct 1:1 drive occurs					
8.	When no member is held or locked together, output cannot occur. The result is a neutral condition					

Table 2.1. Laws of simple planetary gear operation *(https://wikis.engrade.com/planetarygearsetsoperati)*

A typical efficiency loss in a planetary gearbox arrangement is only 3% per stage. This type of efficiency ensures that a high proportion of the energy being input is transmitted through the gearbox, rather than being wasted on mechanical losses inside the gearbox. *(Lu, Eric. 2011)*

f. **<u>Hydraulic Jack-up on Liftboat</u>**

The jacking system for a liftboat is very different than the jacking system for a jack-up drilling rig. The two major differences center around speed and cycles.

Speed of the liftboat jacking system is essential. While a typical jack-up drilling rig elevates at 2 ft/min, a liftboat could elevate at 4-6 ft/min and lower the legs at 14-18 1ft/min. This gives the liftboat the ability to get on and off location significantly faster.

The jacking system for a liftboat encounters a very different operational cycle. It would not be uncommon for a liftboat to jack up and down in one year the same number of times that a jack-up drilling rig would encounter in its entire lifetime. So, the wear factors, redundancy, material grades and shock loads are different between jack-ups and liftboats. *(Ronald E. Sanders-Levingston Offshore, 2012)*

The hydraulic system incorporates a two speed liftboat jacking system that allows leg tagging at 10 ft/min., normal jacking at 4 ft/min. and preload jacking at 2 ft/min. Our liftboat jacking system also offers counterbalance or holding valves at each motor for pinion isolation and elevator like smoothness. *(http://hydraquip-csi.com/liftboat_systems.html)*

- **Hydraulic Fluid Reauirement on Liftboat**

According to Code Federal Regulation no. 46 – Shipping 58.30-10 Hydraulic fluid.

a) The requirements of this section are applicable to all fluid power transmission and control systems installed on vessels subject to inspection.
b) The fluid used in hydraulic power transmission systems shall have a flashpoint of not less than 200 °F. for pressures below 150 pounds per square inch and 315 °F. for pressures 150 pounds per square inch and above, as determined by ASTM D 92 (incorporated by reference, see § 58.03-1), Cleveland “Open Cup” test method.

c) The chemical and physical properties of the hydraulic fluid shall be suitable for use with any materials in the system or components thereof.
d) The hydraulic fluid shall be suitable for operation of the hydraulic system through the entire temperature range to which it may be subjected in service.
e) The recommendations of the system component manufacturers and ANSI B93.5 (incorporated by reference; see 46 CFR 58.03-1) shall be considered in the selection and use of hydraulic fluid.

[CGFR 68-82, 33 FR 18878, Dec. 18, 1968, as amended by CGFR 69-127, 35 FR 9980, June 17, 1970; USCG-1999-5151, 64 FR 67180, Dec. 1, 1999; USCG-2003-16630, 73 FR 65187, Oct. 31, 2008]

CHAPTER III
METHODOLOGY

III.1 Identify and Define the Problem

Identify the lifting operation process at liftboat, analyze supporting system for operation, and decide system to be design

III.2 Literature Review

The reference are required to support this thesis, the reference may come from these following resources :

a. Books
b. Journals
c. Thesis
d. Papers
e. Articles, etc

III.3 Design of the systems

On this thesis "Design of Lifting Operation System (Hydraulic System – Spud Can Jetting System – Leg Mechanism) at Liftboat Cameron Class 200" , design on this thesis it would cover :

a. Design of lifting mechanism on leg
   - Selection type of legs which will be used
   - Selection type of lifting mechanism which will be used
   - Explanation of lifting mechanism process
b. Design of spud can jetting system
   - Selection of Jetting Pump
   - Discharge arrangement at spud can
   - Design of Spud can jetting pressure
   - Selection of pipes, fittimgs and other suporting instruments

c. Design of hydraulic jack-up system
- Selection of Hydraulic Pump along with other instruments
- Design of hydraulic system pressure
- Selection of pipes/hoses, fittimgs and other suporting instruments

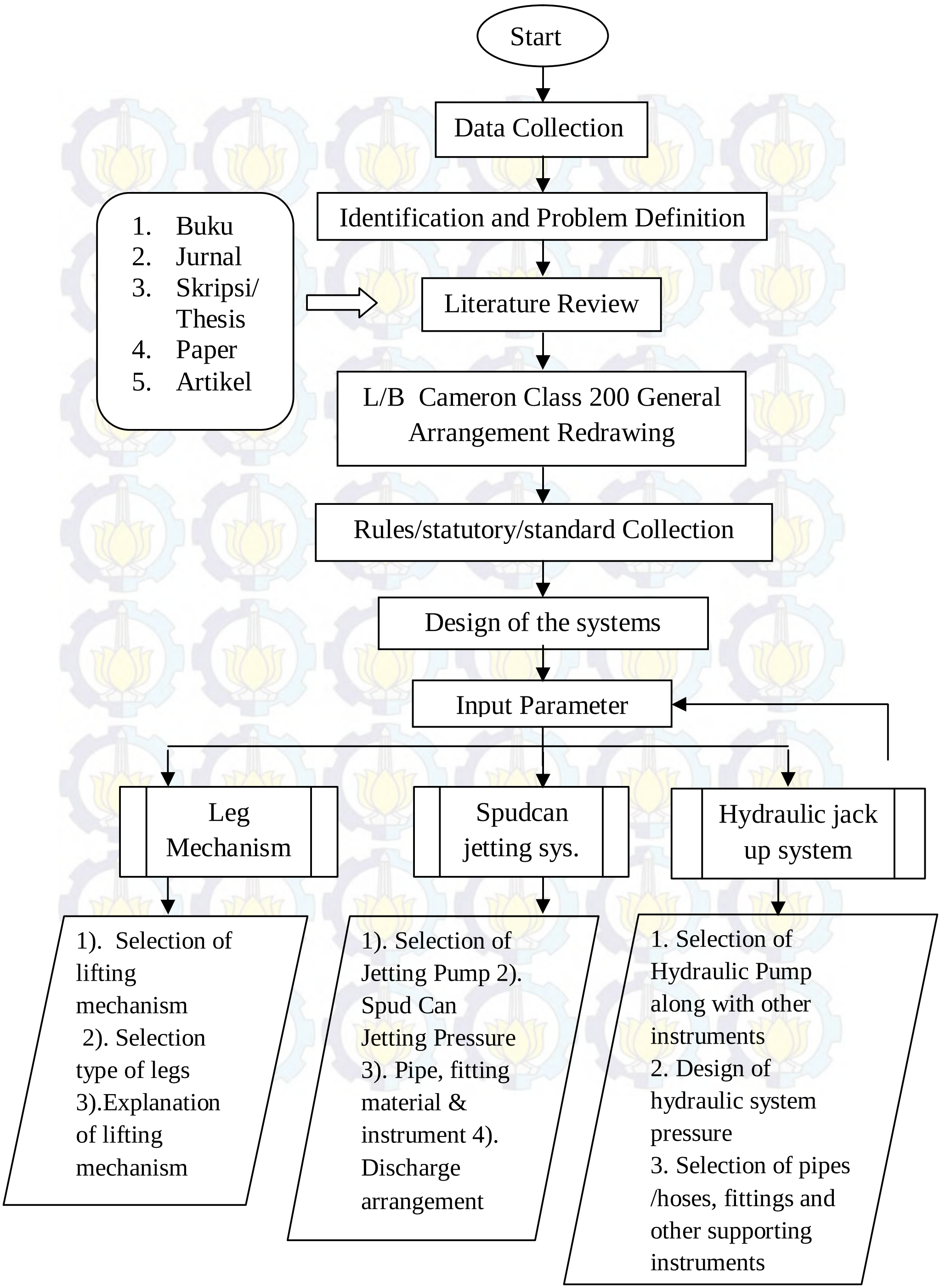
The classification that will be used is American Berau of Shipping (ABS) and also other regulations or codes related to liftboat design.

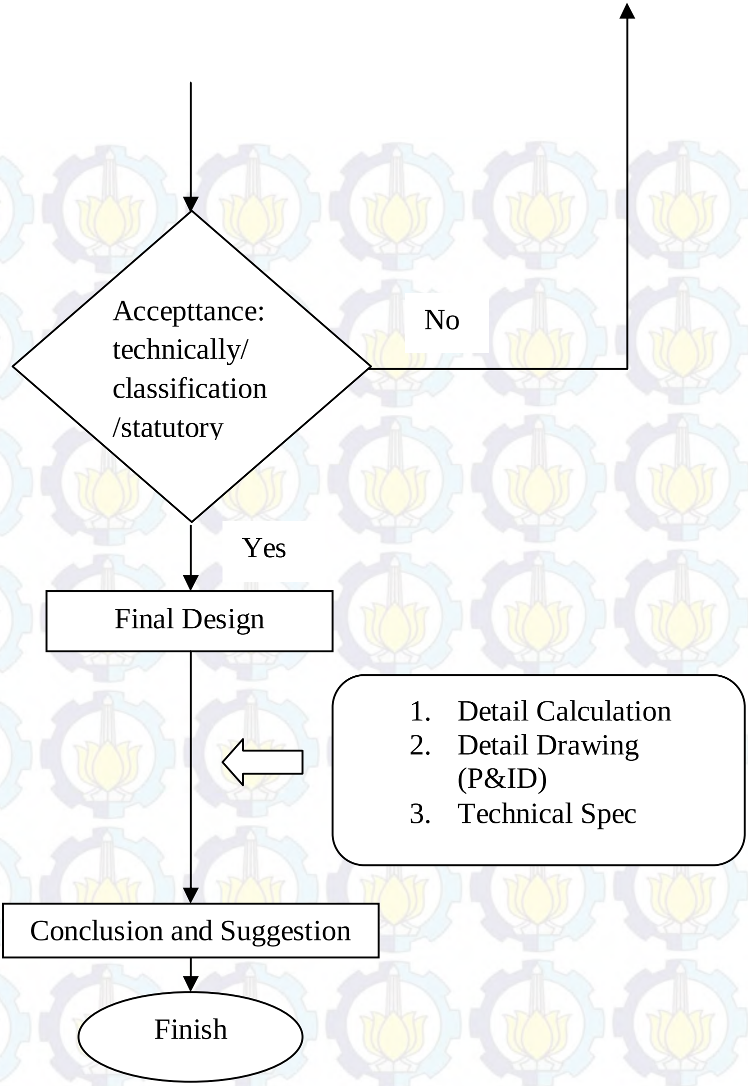
III.4 Verification

To check the design and the calculation is properly done, it is necessary to verify the design and the calculation whether it's already fulfill the requirement of classification or related codes.

III.5 Conclusions and Suggestions

At this step, conclusions will be taken and necessary suggestion will be provided after completion of thesis process.


Start
Data Collection
Identification and Problem Definition
1. Buku
2. Jurnal
3. Skripsi/ Thesis
4. Paper
5. Artikel
Literature Review
L/B Cameron Class 200 General Arrangement Redrawing
Rules/statutory/standard Collection
Design of the systems
Input Parameter
Leg Mechanism
Spudcan jetting sys.
Hydraulic jack up system
1). Selection of lifting mechanism 2). Selection type of legs 3).Explanation of lifting mechanism
1). Selection of Jetting Pump 2). Spud Can Jetting Pressure 3). Pipe, fitting material & instrument 4). Discharge arrangement
1. Selection of Hydraulic Pump along with other instruments
2. Design of hydraulic system pressure
3. Selection of pipes /hoses, fittings and other supporting instruments


Acceptance:
technically/
classification
/statutory
No
Yes
Final Design
1. Detail Calculation
2. Detail Drawing (P&ID)
3. Technical Spec
Conclusion and Suggestion
Finish

CHAPTER IV
ASSESMENT AND RESULTS

IV.1 Liftboat Vessel Data

Vessel Name : L/B Cameron (Class 200)
Website : http://offshoreliftboats.com/vessels

Figure 3.1 L/B Cameron (Class 200)

GENERAL DIMENSIONS

Length (Overall)	: "115'-0" = 35.05 m
Length (Barge Only)	: "111'-3" = 33.90 m
Beam (Overall)	: "74'-0" = 22.55 m
Beam (Barge Only)	: "66'-6" = 20.26 m
Depth (Barge Only)	: "10'-0" = 3.048 m
Open Deck Area	: 4500 Sq. Ft.= 418.06 m

HULL CHARACTERISTICS

Gross Tonnage : 190 Tons

Net Tonnage	: 129 Tons
Max. Deck Cargo (Estimated)	: 700,000 lbs.

PADS

Length : "32'-0" = 9.75 m
Width : "16'-0" = 4.87 m
Depth : "2'-0" = 0.61 m
Configuration : Raked

JACKING

Max. Working Water Depth : 163'-0" = 9.7536 m
Max. Sea Conditions (Jacking Up or Down) : 5'
Total Jacks : 24
Total Jacking Rating : 1183.5 S-Tons (Est.)

LEGS

Quantity	: 3	
Length	: 200'-0"	= 61 m
Size	: 66" O.D."	= 1.68 m
Wall Thicknesses	: "3/4" to 1"	= 25.4 mm

IV.2 Lifting Mechanism on Leg

In this sub-chapter, two major lifting mechanism on liftboat will be explained, include some variety on it.

1. Rack and Pinion Type
   a. Based on US Patent 4,655,640
      1. Components on this system :
         - Frame (no.14)
         - Upper Cross Member (no.28)
         - Upright Side Member (no.24, 26)
         - Rack (no.16)
         - Pinion (no.56, 58)
         - Apart Side Cheek (no.50, 54)
         - Means of Shaft (no.60)
         - Piston Cylinder (no.64, 66, 68, 70)

- Cross Member (no.84)
- Support Frame (no.18)
- Pivot Joint Establishing Pin (no.84, 86, 92, 94)
- Upper Wheel Support (no.34)
- Lower Wheel Support (no.32)
- Platform (no.10)
- Box Portion / Housing (no.98)
- Lock Element (no.100)
- Support Column / Leg (no.12)
- Corner Portion (no.22)
- Central Portion (no.36)
- Side Portion (no.44)
- Mounting Ear (no.72)
- Side Plate Portion (no.32)
- Lower End Member (no.20)

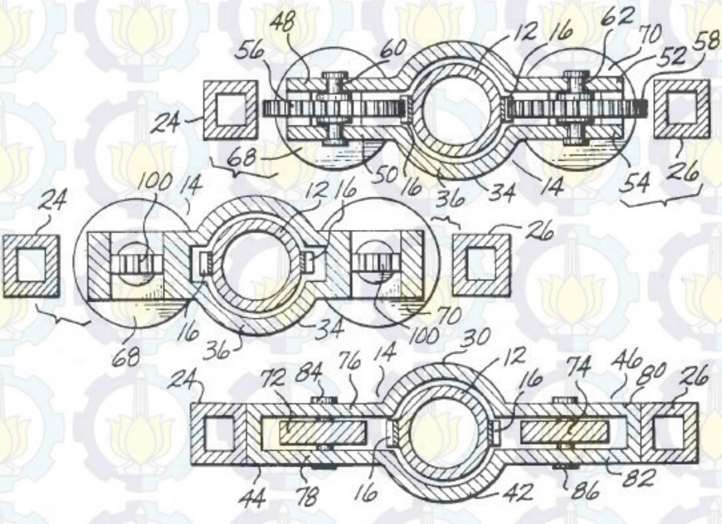


Figure 3.2 Section 4-4, Section 5-5, Sect. 6-6 (top to down)
(US Patent 4,655,640)

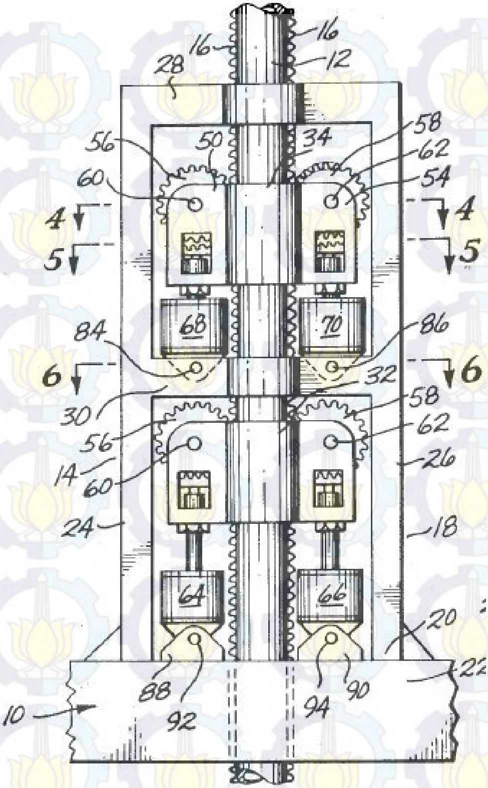


Figure.3.3 Elevational View of Jacking System
(US Patent 4,655,640)

2. The process how lifting mechanism work

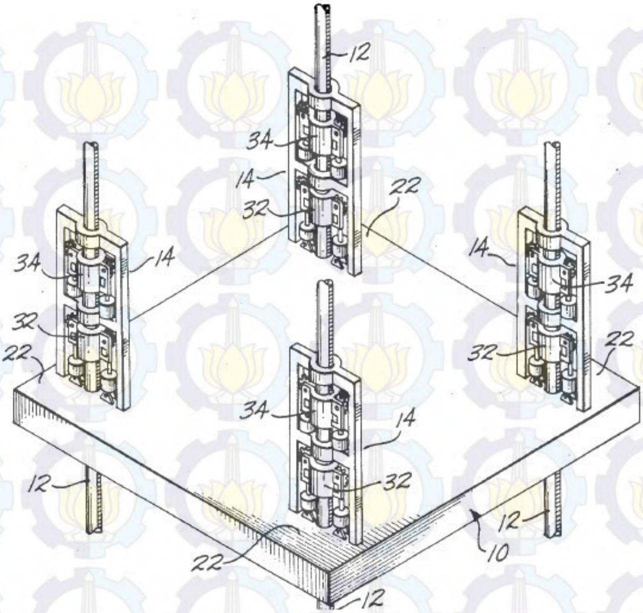


Figure 3.4 Isometric View of Jacking System in Platform
(US Patent 4,655,640)

At this type of rack and pinion, jacking system components most of them are attached to structure which called frame (14). Frame comprise of Upper cross member (28), Cross member (84), Upright side member (24,26), Lower end member (20) as the main structure parts of frame.

Wheel support (32,34) is placed on leg. Attached to this wheel support other components such as, piston cylinder (64, 66, 68, 70), pinion (56, 58), means of shaft (60, 62), apart side cheek (50, 54), piston rod (96) and pivot joint establishing pin (84, 86, 92, 94). There are a pair of rack attached to the Support Column/Leg (12) where opposite each other.

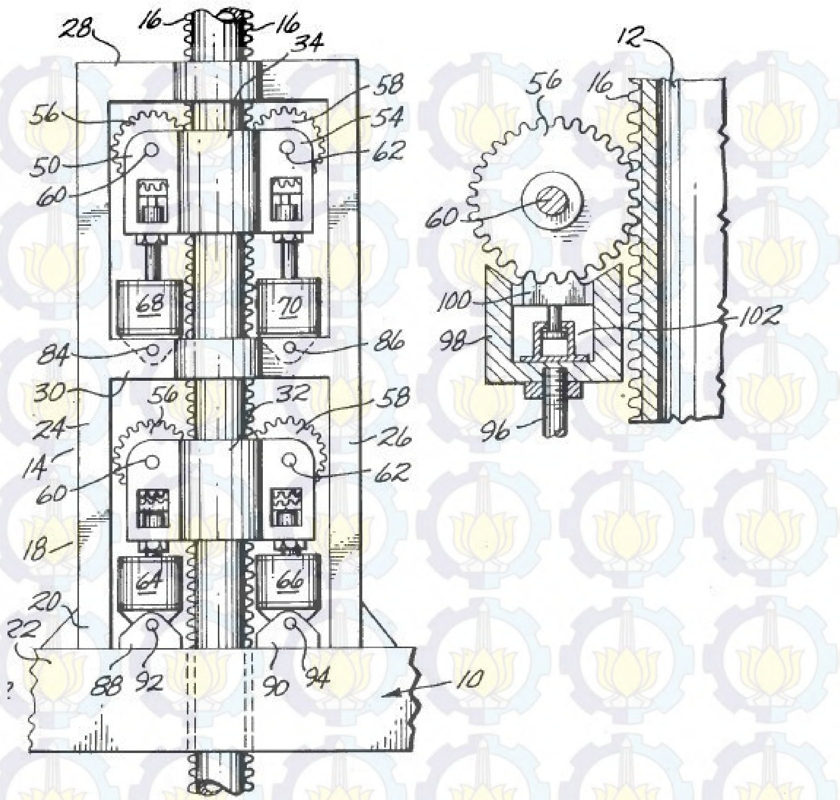


Figure 3.5 Elevational View of Jacking System Components
(US Patent 4,655,640)

When lifting operation is begin, In operation, one pair of hydraulic cylinder 64, 66 or 68, 70 are retracted and the other pair is extended. As shown in In Fig 3.5 (Elevational View of Jacking System) the piston cylinders 64, 66 are retracted and piston cylinder 68, 70 on extended position. If we want to make the platform (10) to be lift up, then the lock element (100) which associated with the lower pair of pinions (56, 58) are extended to lock the pinion (56, 58) in position relative to the column (12). On upper set of pinion (56, 58) it will left unlocked. Next, The cylinder (64, 66) are retracted while the upper piston (67, 70) is start to extended. When lower piston cylinder (64, 66) are retracted they will pull the platform (10) upwardly. The upper wheel support (34) is now being move upwardly by the lift force result by lower piston (64, 66). When lower piston (64, 66) is fully retracted now the upper piston (68, 70) on fully extended position. So to lift up the platform, the process can be repeated but now the upper piston (68, 70) will use to lift up the platform. Now lock element (100) for upper piston is moved into position of locking engagement with upper pinion (56, 58). The lock element in lower part need to be unlock. So, the upper piston (68, 70) are retracted, this will lift up the platform upwardly. The lower piston are extended, for repositioning purpose the lower wheel support (32).

The process for lowering down can be done as the process is reversible. To move downwardly the pinion (56,58) which connected to the retracted cylinder is locked, then the other part pinion is on unlocked position. By the extension of retracted of piston cylinder will make the platform move downwardly. When one pair of piston cylinder is at full extension, so the other cylinder which being fully retracted will take over the lowering process by doing the same process.

b. Based on US Patent 6,652,194 B2

1. Components on this system :
   - Tower (no.40)
   - Tubular Column (no.27)
   - Rack (no.32)
   - Piston Cylinder Unit (no.33)
   - Engagement / Disengagement Means (no.35)
   - Rack Engagement Member (no.34)
   - Pivot Attachment (no.33p)
   - Chord (no.26)

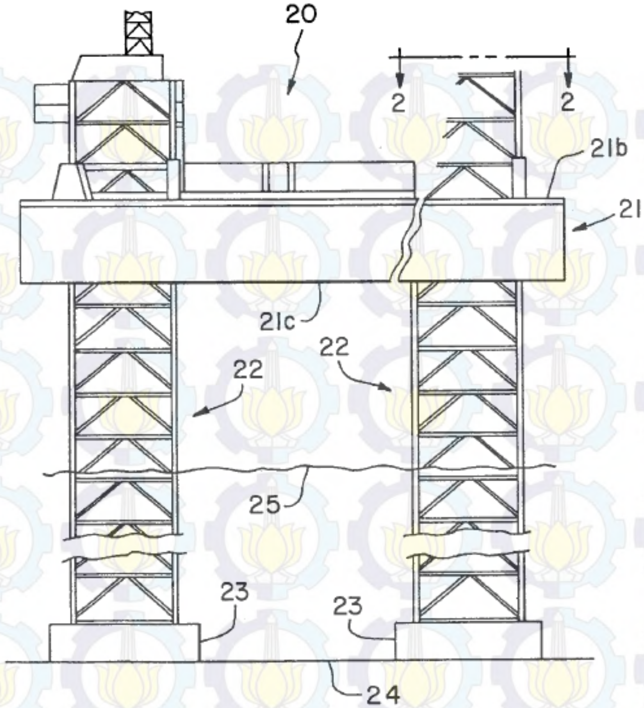


Figure 3.6 Elevational View of Jack-up Platform
(US Patent 6,652,194 B2)

In this figure, it is shown typical component of MODU (20). There are platform structure (21), upper deck (21b), bottom (21c), leg (22), leg footing (23). The environment shown are water level (25) and sea bed (24)

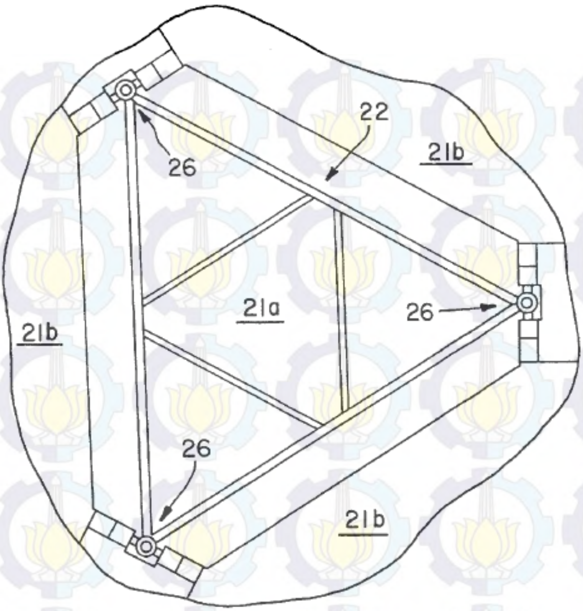


Figure 3.7 Plan View of Jack-up Platform at one leg
(US Patent 6,652,194 B2)

In this figure, jack up with thruss leg comprise of chord (26), leg (22), upper deck (21b), upper deck opening (21a).

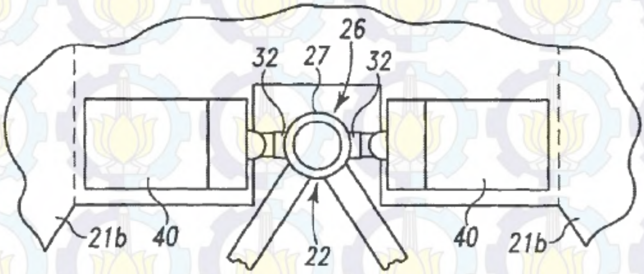


Figure 3.8 Plan View at jacking tower
(US Patent 6,652,194 B2)

On this plan view, it is shown tubular column (27), chord (26), toothed rack (32), tower (40), and upper deck (21b)

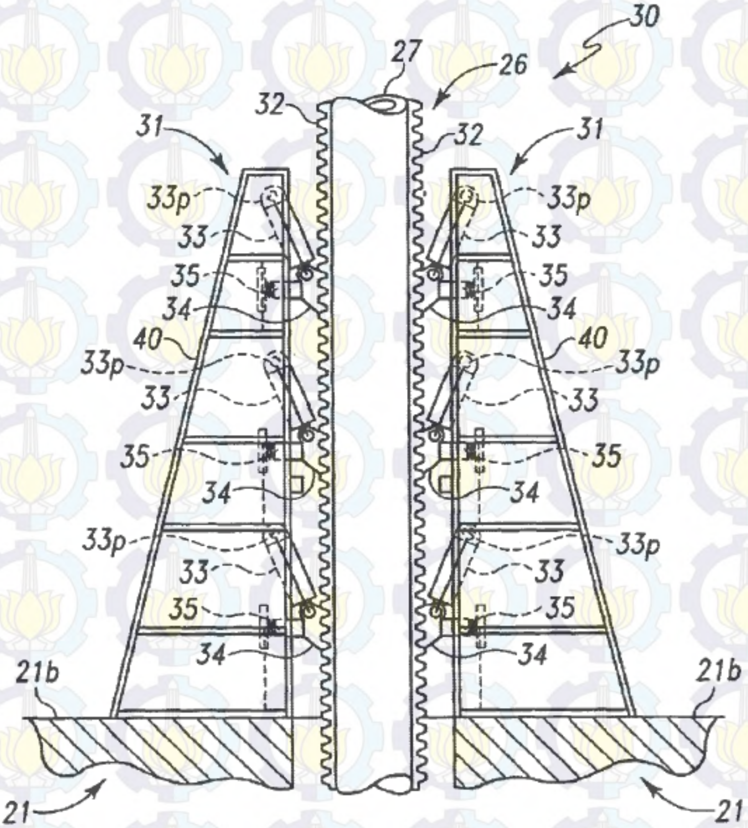


Figure 3.9 Elevational View of Jack-up Tower
(US Patent 6,652,194 B2)

In this figure, we can find tubular column (27), chord (26), toothed rack (32), set of tower (31), piston cylinder unit (33), pivot attachment (33p), rack engagement member (34), engagement/disengagement member (35) and tower (40).

2. The process how lifting mechanism work

There are three pairs of piston cylinder units (33a, 33b, 33c) each leg. To understand how the lifting mechanism is resulted by the piston cylinder unit we have to see figures below which illustrate phased operation of two sets of three hydraulically driven piston cylinder units to effect continuous linear motion.

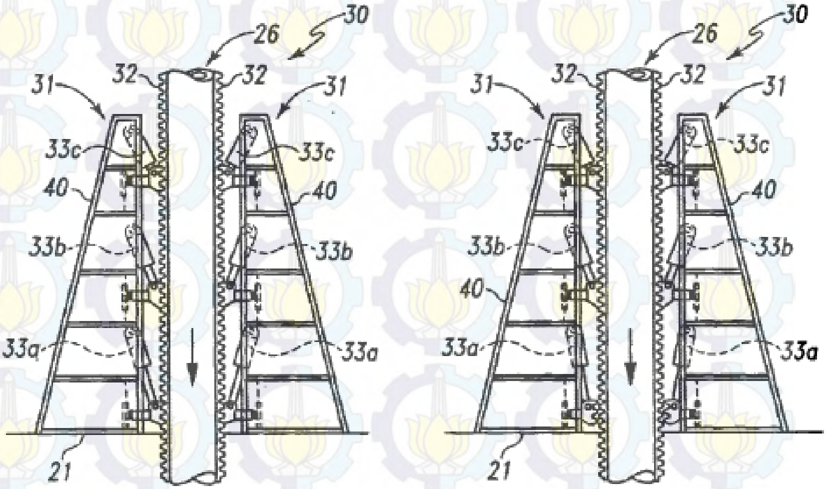


Figure 3.10 Phased Operation of Two Sets of Three Hydraulically Driven Piston Cylinder Unit (1 and 2 – left to right)
(US Patent 6,652,194 B2)

To provide continuous linear motion, the piston/cylinder units (33a, 33b, 33c) of each set (31) and the engagement and disengagement of their toothed rack engagement means (34) are phased so it their operation will be displaced in time. So when two of the piston/cylinder units have their rack engagement members (34) engaged with toothed rack (32) of a leg chord (26), their piston being extended it will drive the leg chord (26), in other way the third piston/cylinder unit will disengaged their rack engagement member (34) from toothed rack (32). The third piston being retracted to reposition its rack engagement member (34) for reengagement.

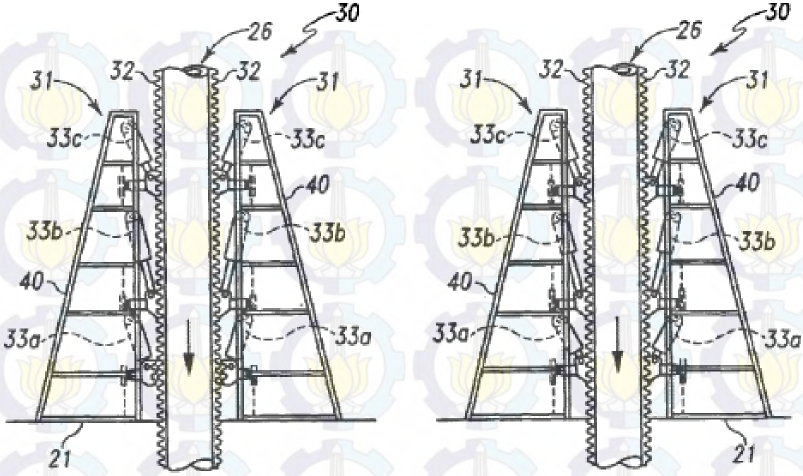


Figure 3.11 Phased Operation of Two Sets of Three Hydraulically Driven Piston Cylinder Unit (3 and 4 – left to right)
(US Patent 6,652,194 B2)

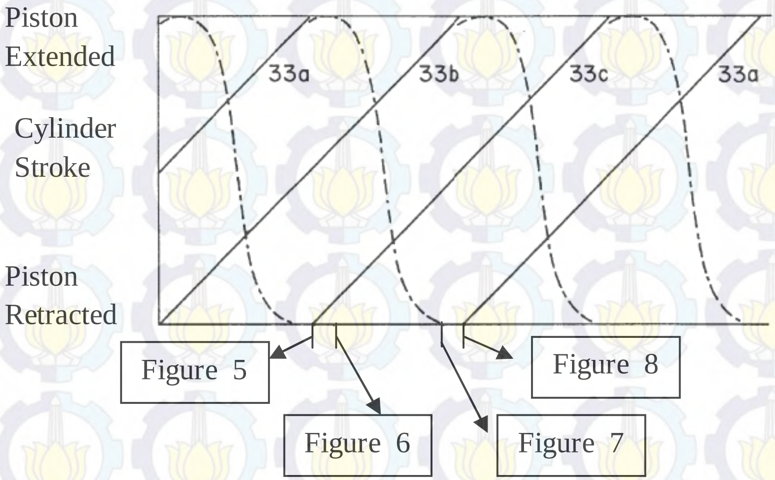


Figure 3.12 Phase Diagram of The Operation of The Piston
(US Patent 6,652,194 B2)

As illustrated in figure phase diagram, by the notation of figure phase operation no.1, piston/cylinder units 33a are in fully extended, 33b are in mid-stroke, 33c are in fully retracted which have just been engaged with toothed racks (32). In the next sequence as shown in figure phase operation no.2, the rack engagement members (34) of piston/cylinder units 33a have been disengaged from the toothed rack (32) while piston/cylinder unit 33b and 33c continue to drive toothed rack (32) and leg chord (26). At the time as illustrated in figure phase operation no.3, rack engagement member (34) for piston/cylinder units 33a is ready for reengagement and it piston have been retracted, the piston /cylinder units 33b are fully extended and piston/cylinder units 33c have been operated until the piston are in mid-stroke. Now as illustrated in figure phase operation no.4, the rack engagement members 34 of piston/cylinder units 33a are reengaged with toothed rack (32) as the pistons of piston/cylinder units 33b approach full extension and as the pistons of piston/cylinder units 33c are in mid-stroke. So as illustrated in figure phase diagram, phase of operation is creating continuous linear motion for jacking operation.

c. Based on GustoMSC Rack and Pinion jacking systems
   1. Components on this system :
   - Teeth pinion
   - Motor
   - Planetary gearbox
   - Tubular Column / Leg
   - Rack

Figure 3.13 Fixed Jacking System but with Electro Motor
(www.gustomsc.com)

2. The process how lifting mechanism work (hydraulic)

There are two types of power sources for Fixed Jacking Systems, electric and hydraulic. Both systems have the ability to equalize chord loads within each leg. *(Bennet & KeppelFELS, 2005)*

This type of jacking system is simple, hydraulic system which have hydraulic motor as actuator that will create rotary motion, then control of rotation will be adjust by planetary gear box, planetary gear box connected to pinion in mechanical connection. Rotary motion in pinion will be change to linier motion by rack teeth attached to the leg.

2. Pin and Hole Type
   a. Based on US Patent 8,425,155 B2
      1. Components on this system :
         - Upper Yoke (no.6)
         - Lower Yoke (no.7)
         - Locking Pin (no.9, 9')
         - Hydraulic Cylinders (no.8)
         - Leg Holes (no.11A-11J)
         - Cylinders for Locking Pin (no.10)
         - Jack House (no.5)
         - Leg (3)

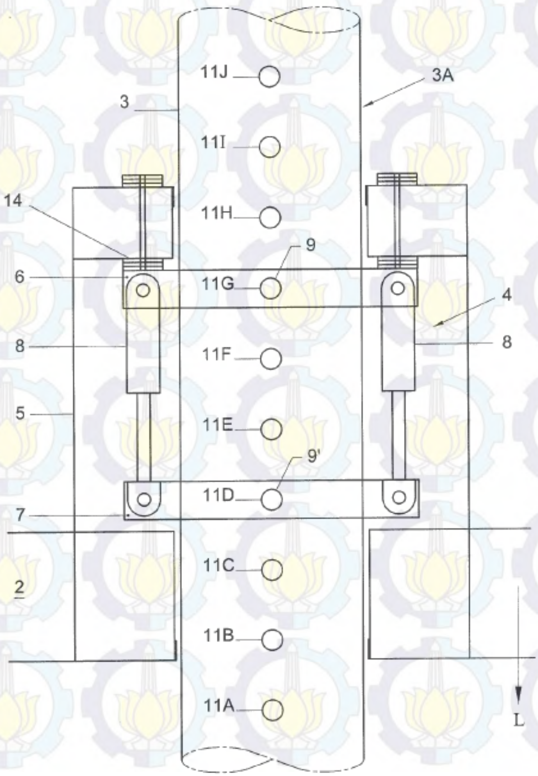


Figure 3.14 Partly Worked Open Side View of Jacking System

As illustrated in open side view, the locking pin (9’) located at lower yoke is in engaging position which moved by cylinder for locking pin (10) – as seen in cross sectional top view - , while the locking pin (9) at upper yoke now in in disengaging position by cylinder (10).

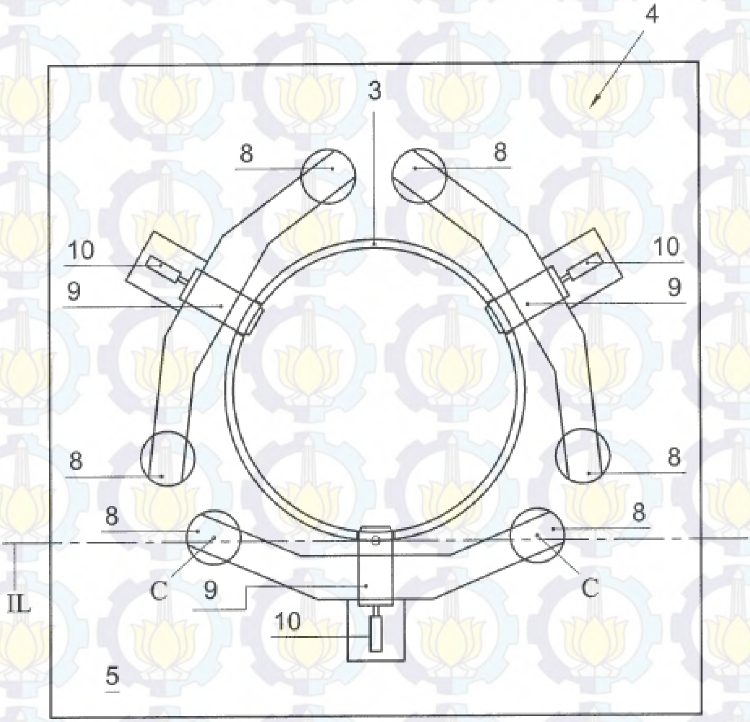


Figure 3.15 Jack House with Jack System Cross Sectional Top View

So when the load is take by lower yoke which it locking pin is in engage position, the cylinder (8) will push the leg (3) downward until the end of the stroke of the cylinder (8), now the upper yoke lifted up and in ready position to engange the locking pin with the leg holes (11). Now when the the cylinder (8) is fully extend, the cylinder for locking pin (10) will placed locking pin (9) in engaged position then the load will be take over by the uper yoke (6) while locking pin (9’) will disengaged. Next, the cylinder (8) will do the return stroke, during the return stroke the platform will not moving.

Selected leg mechanism for the design is rack and pinion type, because it is provide smooth continuous jacking motion, and popular system which interested to learned. Selected type of leg is cylindrical type which is suitable for liftboat less than 300 feet working depth and require less deck space for it installation

IV.3 Spud Can Jetting System

A. Flow Rate and Pressure of Spud Can Jetting System

Bearing capacity prediction methods require highly experienced and competent geotechnical engineers which require each site investigation which depend upon the ground conditions at each site under investigation *(Osborne, J.J. 2009)*. To determine value of flow rate and pressure, actual typical flow rate and pressure is taken from reference.

1. High Pressure Jetting System

- Flow rate = 25 m^3/h
- Pressure = 80 bar

(Clarom, 1993)

2. Low Pressure Jetting System

- Flow rate = 180 m^3/h
- Pressure = 12 bar

Typical low pressure, refer to :

a. Seajacks Hydra - Specification Sheet
b. Seajacks Kracken - Specification Sheet
c. Seajacks Leviathan - Specification Sheet
d. Seajacks Zarathan - Specification Sheet

B. Pipe Material and Pipe Schedule

According to ABS Rules Under 90 meters (Part. 2, Chap. 3, Sect. 12, 3) which is explain grade material of pipe and their relation to ASTM pipe.

According to explanation given in ABS Rules Under 90 meters (Part. 2, Chap. 3, Sect. 12, 3), the pipe under grades 1, 2 and 3 cover seamless and welded steel pipe. These grade is a nominal (average) wall thickness suitable for welding and suitable for forming operations involving coiling, bending and flanging.

Selected material : ASTM A 53 (Seamsless Carbon Steel)

a. High Pressure Jetting System

- Pipe diameter

Flow rate = 25 m^3/h = 0,0069 m^3/s

$$Q = v.A$$

$$Q = v \times \frac{\pi \times D^2}{4}$$

$$Q \times 4 = v \times \pi \times D^2$$

$$D^2 = \frac{Q \times 4}{v \times \pi}$$

$$D = \sqrt{\frac{(Q \times 4)}{\pi \times v}}$$

$= 0{,}07$ m

$= 66{,}5$ mm

Where,

Q = Flow Rate (m^3/s) = 0,0069

v = Velocity of fluid (m/s) = 2

A = Pipe cross section (m^2)

D = Pipe diameter (m)

π = 3,14

Based on calculation above 2 1/2 " diameter pipe is selected.

- Pipe Thickness

The required thickness of straight pipe calculated by following equation

$tm = t + c$ *(ASME B 31.3 Chap II Part 2, 304.1)*

Where,

tm = minimum required thickness, including mechanical, corrosion, and erosion allowances (in)

t = pressure design thickness (psi)

c = the sum of the mechanical allowances plus corrosion and erosion allowances (in)

= 0,1 *(Appendix H ASME B31.3-2002)*

Straight Pipe Under Internal Pressure

For t < D/6, the internal pressure design thickness for straight pipe shall be not less than that calculated by following equation.

$$t = \frac{PD}{2(SE+PY)}$$

Where,

t = pressure design thickness (psi)

P = internal design gage pressure

D = outside diameter of pipe

S = stress value for material from Table A-1 (ASME B 31.3)

E = quality factor from Table A-1A or A-1B (ASME B 31.3)

Y = coefficient from Table 304.1.1 (ASME B 31.3)

- P = 80 bar = 1160 psi
- D = 2,875 in
- S = 16.000 psi
- E = 1 (seamsless pipe)
- Y = 0,5

$$Y = \frac{d + 2c}{D + d + 2c}$$

Where,

D = outside diameter of pipe (in)

d = inside diameter of pipe (in)

(design : SCH 80)

= 2,88 - 0,22 = 2,66

c = mechanical allowance (in)

= 0,1 *(Appendix H ASME B31.3-2002)*

$$t = \frac{PD}{2(SE+PY)}$$

$$t = \frac{1160 \times 2.875}{2\ (16000 \times 1 + (1160 \times 0.5)}$$

= 0,1 in

So required thickness of straight pipe will be :

tm = t + c

= 0,1 + 0,1

= 0,2 in *(SCH 80 pipe)*

So pipe 2 1/2" SCH. 80 can be used

b. Low Pressure Jetting System

- Pipe diameter

Flow rate = 180 m^3/h = 0,0500 m^3/s

$$Q = v.A$$

$$Q = v \times \frac{\pi \times D^2}{4}$$

$$Q \times 4 = v \times \pi \times D^2$$

$$D^2 = \frac{Q \times 4}{v \times \pi}$$

$$D = \sqrt{\frac{(Q \times 4)}{\pi \times v}}$$

= 0,15 m

= 146 mm

Where,

Q = Flow Rate (m^3/s = 0,0500

v = Velocity of fluid (m/s) = 3

A = Pipe cross section (m^2)

D = Pipe diameter (m)

π = 3,14

Based on calculation above 6 " diameter pipe is selected.

- Pipe Thickness

The required thickness of straight pipe calculated by following equation

$tm = t + c$ *(ASME B 31.3 Chap II Part 2, 304.1)*

Where,

tm = minimum required thickness, including mechanical, corrosion, and erosion allowances (in)

t = pressure design thickness (psi)

c = the sum of the mechanical allowances plus corrosion and erosion allowances (in)

= 0,1 *(Appendix H ASME B31.3-2002)*

Straight Pipe Under Internal Pressure

For t < D/6, the internal pressure design thickness for straight pipe shall be not less than that calculated by following equation.

$$t = \frac{PD}{2(SE+PY)}$$

Where,

t = pressure design thickness (psi)

P = internal design gage pressure

D = outside diameter of pipe

S = stress value for material from Table A-1 (ASME B 31.3)

E = quality factor from Table A-1A or A-1B (ASME B 31.3)

Y = coefficient from Table 304.1.1 (ASME B 31.3)

- P = 12 bar = 174 psi
- D = 6,625 in
- S = 16.000 psi
- E = 1 (seamsless pipe)
- Y = 0,4

$$Y = \frac{d + 2c}{D + d + 2c}$$

Where,

D = outside diameter of pipe (in)

d = inside diameter of pipe (in)

(design : SCH 40)

= 6,63 - 0,28 = 6,35

c = mechanical allowance (in)

= 0,1 *(Appendix H ASME B31.3-2002)*

$$t = \frac{174 \times 6.625}{2\ (16000 \times 1 + (174 \times 0.4)}$$

= 0,04 in

So required thickness of straight pipe will be :

tm = t + c

= 0,04 + 0,1

= 0,14 in

So pipe 6" SCH 40 can be used.

C. Pump Selection

a. High Pressure Jetting System

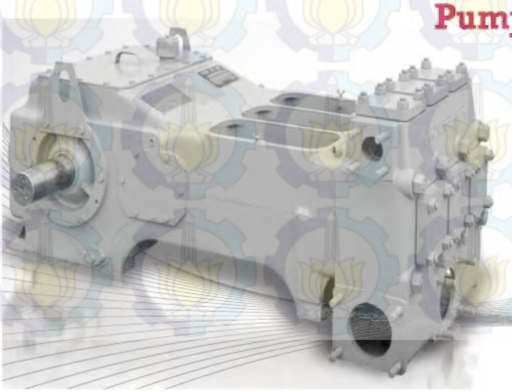
- Flow rate = 25 m3/h\
- Pressure = 80 bar

(Clarom, 1993)

PISTON/LINER SIZE		DISPLACEMENT PER REVOLUTION		DISPLACEMENT @ PUMP RPM 100			
in.	mm.	GAL	Liter	GPM	LPM	PSI	kg/sq. cm
4	102	.816	3.089	82	309	2944	206
4.5	114	1.032	3.907	103	391	2326	163
5	127	1.274	4.824	127	482	1884	132
5.5	140	1.542	5.837	154	584	1557	109
Input Power:			BHP	156			
			kW	116			

Gardner Denver

Pumping Perfected.™

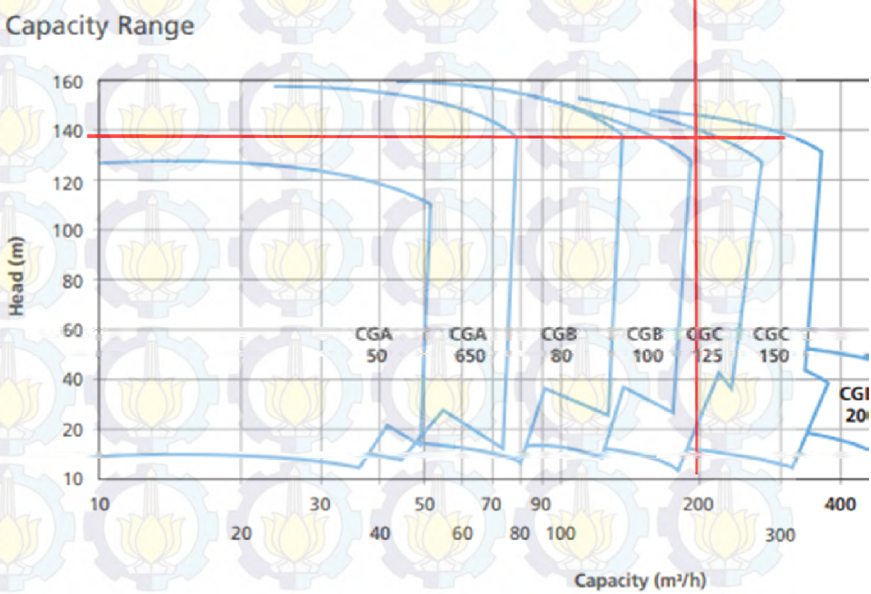


Pump Specification :

1. Maker : Gardner Denver
2. Model : THE Piston 55
3. Pressure : up to 1884 psi (129 bar)
4. Capacity : up to 127 gpm (28.8 m^3/h)
5. Type : Positive Displacement Pump (Piston)

b. Low Pressure Jetting System

- Flow rate = 180 m^3/h
- Pressure = 12 bar



Pump Specification :

1. Maker : Hamworthy
2. Model : CGC 125
3. Head : 140 m (max)
4. Capacity : 200 m^3/h (max)
5. Type : Centrifugal Pump

IV.4 Hydraulic Jack-Up System

Hydraulic jacking system is used to raise up or lowering down the liftboat platform or the leg.

Hydraulic Jack-Up Calculation

1. Jacking Data For Liftboat

- Max. Working Water Depth = 163'-0" = 9.75 m
- Total Jacks = 24 pcs
- Total Jacking Rating = 1183.5 S-Tons.

Where, 1 S-Tons = 0.9071 Ton (metric)

Total Jacking Rating = 1073.5529 Ton (metric)

- Total Holding Rating = 1578.1 S-Tons.

Where, 1 S-Tons = 0.9071 Ton (metric)

Total Holding Rating = 1431.4945 Ton (metric)

2. Jacking Speed Design

a) Setting on Location:

Lowering Legs	:	8	fpm	=	0.04	m/s
Raising Hull	:	4	fpm	=	0.02	m/s

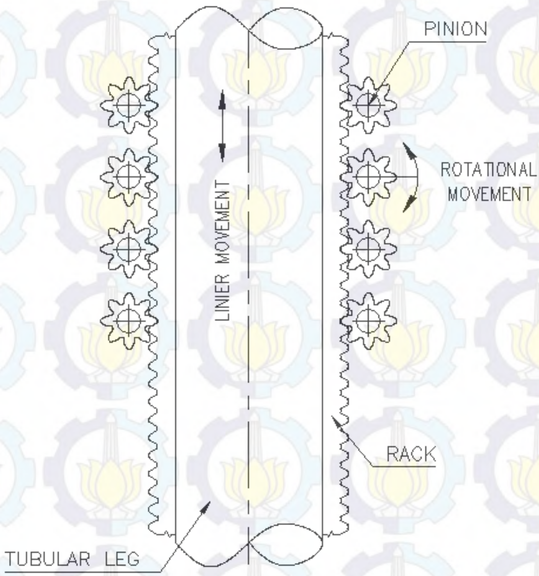
b) Departing Location:

Lowering Hull	:	4	fpm	=	0.02	m/s
Raising Legs	:	8	fpm	=	0.04	m/s

Typical jacking speed of liftboat, refer to :

a. Levingston 260E – Specification sheet
b. Levingston 320E – Specification sheet

3. Calculation For Raising / Lowering Liftboat Hull



a) Jacking Rating

According to ship particular total jacking rating = 1183.5 S-Ton
= 1073.5529 mTon
= 1,073,553 kg

- Liftboat leg = 3 units
- Jacks per leg = 8 units
- Total Jacks = 8 units x 3 leg = 24 units
- Jacking rating per leg = 1073.5529 / 3 leg
  = 357.85095 Ton
- Jacking rating per jacks = 44.73 Ton
  = 44731.369 kg

b) Force required for each jack

Force to handled by each jacks calculated as follow :

$w = m \times g$

$= 44731.369 \times 10$

$= 447{,}313.7$ N

$= 447.31369$ kN

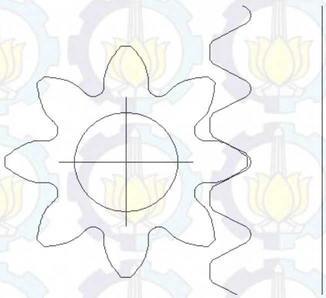
Where,

w = weight (N) (kg.m/s^2)

m = mass (kg)

g = gravity acceleration (m/s^2)

c) Torque required for each jack



Where,

τ = Torque (Nm)

F = Force (N)

= 447313.69 N

r = radius (m)

= 0.25 m

$\tau = F \times r$

$= 447313.69 \times 0.25$

$= 111828.42$ Nm

d) Angular speed required

$v = \omega \times r$

$$\omega = \frac{v}{r}$$

$$= \frac{0.02}{0.250}$$

$= 0.081$ rad/s

Where,

ω = Angular velocity (rad/s)

v = velocity (m/s)

= 0.02 m/s (elevating speed)

r = radius of pinion (m)

= 0.25 m

e) Power required

$P = \tau \times \omega$

$= 111828.42 \times 0.081$

$= 9089.41$ W

$= 9.089$ kW

Where,

P = Power (W) (Nm/s)

τ = Torque (Nm)

ω = Angular velocity (rad/s)

For purpose checking, power required can calculated with other formula :

$$P = \frac{W}{t}$$

Where,

P = Power (W) (Nm/s)

W = Work (Energy) (Nm) (Joule)

t = time (s)

$\tau = F \times l$

Where,

τ = Torque (Nm)

F = Force (N)

l = length or distance (m)

So,

$$P = \frac{F \times l}{t}$$

Where,

l/t = m/s = velocity (v)

$P = F \times v$

$= 447313.69 \times 0.02$

$= 9089.41$ W

$= 9.089$ kW

Where,

P = Power (W) (Nm/s)

F = Force (N) (kg.m/s^2)

= 447313.69 N

v = velocity (m/s)

Both calculation showing the same power

4. Calculation For Raising / Lowering Liftboat Leg

a) Jacking Rating

- Calculation the weight of each leg

Length of leg	=	200'-0"	=	61 m
Size of leg	=	66" O.D."	=	1.68 m
		ID	=	1.65 m
Wall thicknesses	=	"3/4" to 1"	=	0.0254 m

According to ASME B 36.10 weight per meter : 1065.82 kg

So for 200' pipe length : 64972.387 kg

= 64.97 ton

Additional 20% allowance given considering structure attached to it.

= 77.97 ton

Spud can weight = 9.10 ton

Total weight for each leg = 87.07 ton

- Jacking rating for each jack

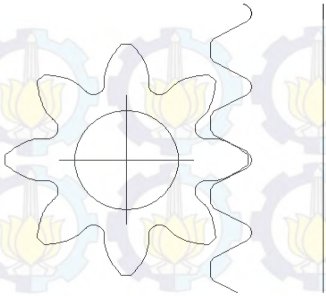
With 8 motors installed at each leg, jacking rating for each motor will be = $\frac{87071.75}{8}$ = 10883.97 kg

b) Force required for each jack

Force to handled by each jacks calculated as follow :

w = m x g	Where,
= 10883.969 x 10	w = weight (N) (kg.m/s^2)
= 108839.7 N	m = mass (kg)
= 108.8397 kN	g = gravity acceleration (m/s^2)

c) Torque required for each jack



Where,

τ = Torque (Nm)

F = Force (N)

= 108839.69 N

r = radius (m)

= 0.250 m

$\tau = F \times r$

$= 108839.69 \times 0.25$

$= 27209.922$ Nm

d) Angular speed required

$v = \omega \times r$

$$\omega = \frac{v}{r}$$

$$= \frac{0.04}{0.250}$$

$= 0.163$ rad/s

$= 1.553$ rpm

(rpm at pinion)

Where,

ω = Angular velocity (rad/s)

v = velocity (m/s)

= 0.04 m/s

r = radius of pinion (m)

= 0.25 m

$$1 \text{ rad/s} = \frac{60}{2\pi} \text{ rpm} = 9.55 \text{ rpm}$$

e) Power required

$P = \tau \times \omega$

$= 27209.922 \times 0.163$

$= 4423.24$ W

$= 4.423$ kW

Where,

P = Power (W) (Nm/s)

τ = Torque (Nm)

ω = Angular velocity (rad/s)

5. Calculation of Hydraulic Pump and Motor

1. Based On Raising Calculation

a) Hydraulic Motor

- Displacement of each hydraulic motor which is the pressure set at 200 bar to lift up the hull calculated as follow :

$\tau = p \times d$

$d = \dfrac{\tau}{p}$

$= \dfrac{124{,}253.80}{20{,}000{,}000}$

$= 0.0062127 \quad m^3$

$= 6212.6901 \quad cm^3$

Where,

τ = Torque (Nm)
= 111828.42 Nm (at pinion)
= 124253.8 Nm (with gear eff.)

p = Pressure (N/m^2)
= 200 bar = 20,000,000 N/m^2

d = displacement of motor hyd (m^3)

- Rotational speed of hydraulic motors

Motor displacement of motor to be selected :

Vg = Maximum Motor displacement (cm^3 / rev.)
= 35 cm^3 / rev.

(Sauer Danfoss Motor Series 40 M35 MV)

So the amount of displacement fulfilled by approx 90% of motor displ.

$= \dfrac{6212.6901 \ cm^3}{31.5 \ cm^3/\text{rev.}}$

= 197.23 times of hydraulic motor revolution

then hydraulic motor revolution set 200 rpm, motor displacement will be :

$= \dfrac{6212.6901}{200} = 31.1 \ cm^3/\text{rev} = 88.8 \ \%$ of max motor displ.

- Flow rate required by each hydraulic motor (8 units)

$$Q = \frac{Vg \times n}{1000 \times \eta_v} = \frac{6212.69}{900} = 6.90 \quad l / min$$

Where,

Q = Flow rate (l/min)

η_v = Motor volumetric efficiency = 0.9

Vg = Motor displacement (cm³ / rev) = 31.1 cm³ / rev

n = rotation per minute (rpm) = 200 rpm

- Torque output by each hydraulic motor

$$Me = \frac{Vg \times \Delta p \times \eta_{mh}}{20\pi} = \frac{5326.61}{62.8} = 84.82 \quad Nm$$

Where,

Vg = Motor displacement cm³ / rev. = 31.1 cm³ / rev.

Δp = pHD - pND (bar)

pHD = High pressure (bar) = 200 bar

pND = Low pressure (bar) = 19.5 bar

η_{mh} = Motor mechanical-hydraulic Eff. = 0.95

- Flow rate required to be supplied by each hydraulic pump (2 units)

Q = Q motor x 4 units

= 6.90 x 4

= 27.61 l / min

= 27611.96 cm³ / min

b) Hydraulic Pump

- Rotational speed of hydraulic pumps

  Motor displacement of motor to be selected :

  Vg = Maximum pump displacement (cm^3 / rev.)

  = 45.9 cm^3 / rev. *(Sauer Danfoss Pump Series 40 M46)*

Pumps is arranged can provide required flow rate supply at approx 30% of pump max displacement :

$$= \frac{13437.00 \ cm^3}{13.77 \ cm^3/rev.}$$

= 975.82 times of hydraulic motor revolution

then hydraulic motor revolution set 1000 rpm, pump displacement will be :

$$= \frac{13437.00}{1000} = 13.4 \ cm^3/rev = 29.3 \ \% \text{ of max pump displ.}$$

- Flow rate produce by hydraulic pumps

$$Q = \frac{Vg \times n \times \eta_v}{1000}$$

$$= \frac{12093.30}{1300}$$

$$= 9.30 \ l/min$$

Where,

Q = Flow rate (l/min)

η_v = Motor volumetric efficiency

= 0.9

Vg = Displacement per rev (cm^3 / rev)

= 13.4 cm^3 / rev

n = rotation per minute (rpm)

= 1000 rpm

Due to volumetric efficiency rotation of pump to be increase at 1200 rpm for sufficient supply to hydraulic motor

$$Q = \frac{Vg \times n \times \eta_v}{1000}$$

$$= \frac{14511.96}{1000}$$

$$= 14.51 \text{ l / min}$$

Where,

Q = Flow rate (l/min)

η_v = Motor volumetric efficiency

= 0.9

Vg = Displacement per rev (cm³ / rev)

= 13.4 cm³ / rev

n = rotation per minute (rpm)

= 1200 rpm

c) Hydraulic Motor and Pump Selection

Hydraulic motor

No	Jacking Condition	Vg (cm^3/rev)	Motor Displ. (%)	rpm	Pressure (bar)	Motor Torque (Nm)
1.	Raising hul	31.06	88.75	200	200	84.82
2.	Raising leg	10.08	28.79	300	100	27.52

Table 4.1. Hydraulic Motor Performance When Raising Hull / Leg

<u>Selected hydraulic motor</u>

1. Maker : Danfoss series 40 motor, M35 MV
2. Product type : In-line, axial piston, variable, positive displ
3. Rotation : Clockwise (CW) & counterclockwise (CCW)
4. Control Option : Hydraulic 2 position
5. Displacement : 35 cm³ / rev.

6. System pressure : Rated pressure 210 bar
Max. pressure 345 bar

Hydraulic pump

No	Jacking Condition	Q req 4 units motor (l/min)	Vg (cm^3/rev)	rpm	Pump Displ. (%)	Q supply 4 units motor (l/min)
1.	Raising hul	27.61	41.21	750	89.79	27.82
2.	Raising leg	13.44	13.44	1200	29.27	14.51

Table 4.2. Hydraulic Pump Performance When Raising Hull / Leg

<u>Selected hydraulic pump</u>

1. Maker : Danfoss series 40 pump, M46 PV
2. Product type : In-line, axial piston, variable, positive displ
3. Rotation : Clockwise (CW) & counterclockwise (CCW)
4. Displacement : 45.9 cm^3 / rev.
5. System pressure: : Rated pressure 345 bar
   Max. pressure 385 bar

- Hydraulic reservoir capacity

V = (3-5) x Q pump

(additional 15 % must be provided to balance out fluctuations in level)

(FESTO Hydraulic Basic Level Textbook, page 108)

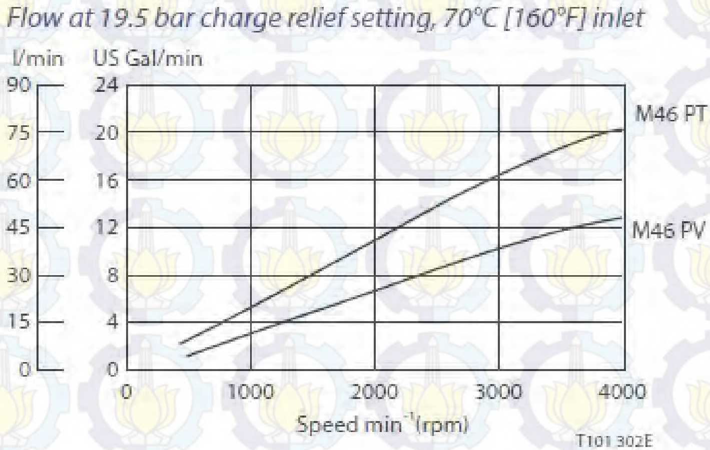
V = 4 x Q pumps

= 4 x 45.9 x 2 units

= 367 cm^3

6. Calculation of Hydraulic Charge Pump

Based on selected hydraulic pump, Sauer danfoss M46 PV, available internal charge pump shown in graphic below



Graph. Available internal charge pump output flow for M46 PV

(Sauer Danfoss M46 Piston Pumps Technical Information, page 37)

The maximum rpm of charge pump run in normal operation is avoided. Charge pump is set to work in normal operation at approximate 3700 rpm which produce = 45 l/m. Because the system is one pump-multi motor system, the general rules of tumb that for charge pump sizing, such as the "10% rule" need to conform with the actual operating condition.

- Operational Condition : Raising / Lowering Hull
- Configuration : (1 pumps for 4 motors) x 2
- System Pressure : 200 bar = 2901 psi

a) Leakage Requirement

- **Pump**

Series	:	Series 40	Pump Disp :	41.21	cm^3 / rev
Frame Size	:	M 46 PV	No of pumps :	2	units
Speed	:	750 RPM			
Vol Eff.	:	90 %			
Leakage	:	13.4 gpm			

$$\text{Pump leakage} = \frac{\text{Pump Disp x Pump RPM}}{231} \times \left(1 - \frac{\text{Pump Eff.}}{100}\right)$$

$$= \frac{41.21 \times 750}{231} \times \left(1 - \frac{90}{100}\right)$$

$$= 13.4 \text{ gpm}$$

(Sauer Danfoss Application Manual, Driveline Components, page 30)

For multiple pumps (2 pumps) leakage = 13.4 x 2

= 26.8 gpm

- **Motor**

Series	:	Series 40	Pump Disp :	27.61	l/min
Frame Size	:	M35 MV	No of motors :	4	units
Speed	:	200 rpm			
Vol Eff.	:	90 %			
Leakage	:	29.5 gpm			

$$\text{Pump flow} = \frac{\text{Pump Disp x Pump RPM}}{231} \times \frac{\text{Pump Eff.}}{100}$$

$$= \frac{27.61 \times 750}{231} \times \frac{90}{100}$$

$$= 80.7 \text{ gpm}$$

(Sauer Danfoss Application Manual, Driveline Components, page 30)

$$\text{Motor leakage} = \frac{\text{Pump Flow}}{\text{Number of motors}} \times 1 - \frac{\text{Motor Eff.}}{100}$$

$$= \frac{80.68}{4} \times 1 - \frac{90}{100}$$

$$= 2.02 \text{ gpm for each motor}$$

(Sauer Danfoss Application Manual, Driveline Components, page 30)

For total motor (8 motors) leakage = 2.02 x 8

= 16.1 gpm

- **Total leakage** = Leakage of pump + leakage of motors

  = 13.4 + 16.1

  = 29.5 gpm

b) Loop Flushing Requirement

Q Loop Flushing flow = 3 gpm x 8 unit motors

= 24 gpm

The amount of loop flushing will normally vary between 2-4 gpm depending on the charge pump displacement, input speed, and relative settings between the pump and motor charge relief valves
(Sauer Danfoss, Driveline Components, page 31)

c) Fluid Compressibility

Magnitude of pressure spike	:	2617.9	psi
Time duration	:	0.1	sec.
Bulk modulus	:	1,034	psi
Hose length	:	4.9	feet
Hose I.D.	:	0.88	inches
Hose Volume	:	35.9	in^3
Charge flow required	:	9.09	gpm

Typical values for the hydraulic fluid bulk modulus range from 690 to 1380 MPa. *(Charge Pump and Loop Flush Sizing for Closed Loop, One Pump, Multi-Motor Systems, page 4)*

Typical values for fluid bulk modulus adjusted for entrapped air are:

• small level	200,000	psi	=	1,379	Mpa
• moderate level	150,000	psi	=	1,034	Mpa
• 2% air	100,000	psi	=	689	Mpa

(Sauer Danfoss Application Manual, Fluids and filtration, page 19)

- hydraulic hose size

Flow rate	=	55.22	l / min	
	=	14.59	gpm	1 liter = 0.26 gallor

$$D^2 = \frac{Q \times 0.4081}{v}$$

$$= \frac{14.6 \quad x \quad 0.41}{9}$$

$$= 0.66$$

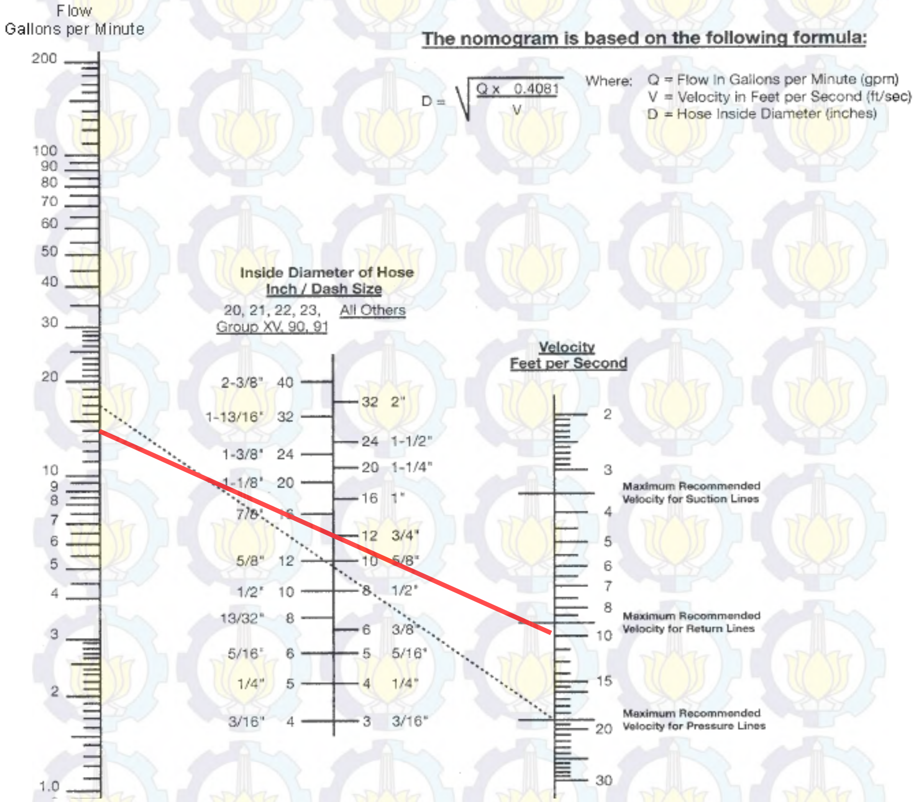
$$D = 0.81 \text{ in}$$

where,

Q = Flow in gallon per minute (gpm)

V = Velocity in feet per second (fps)

D = Hose inside diameter (inches)



(Parker Technical, Section E, page E-3)

Selected hydraulic hose size = 0.88 inch, dash no. 16

- Hose volume

$$V = (9{,}42) \times (I.D)^2 \times (Length)$$
$$= 35.9 \quad in^3$$

(Sauer Danfoss Application Manual, Driveline Components, page 31)

- Bulk modulus flow

$$Q = \frac{\Delta P \times V}{BM \times \Delta t} \times 0.26$$
$$= \frac{2617.9 \times 35.9}{1{,}034 \times 0.1}$$
$$= 9.09 \text{ gpm}$$

Where,

Q = additional charge flow required (gpm)

ΔP = change in pressure (psi)

BM = bulk modulus (psi)

Δt = time duration for pressure change (sec)

(Sauer Danfoss Application Manual, Driveline Components, page 30)

d) Hydraulic Brake Requirement

- Hydraulic release brake = 5 gpm

(Mico Brake Spting Apply Hydraulic Release page 30)

<u>Total charge flow requirements :</u>

1. Leakage Requirement	=	29.5	gpm
2. Loop Flushing Requiremen	=	24	gpm
3. Fluid Compressibility	=	9.09	gpm
4. Auxiliary Function	=	5	gpm
Total	=	67.6	gpm

e) Charge Pump Selection

- Available internal charge pump = 45.00 l / min
  from M46 hydrostatic pump = 11.89 gpm
- Total available internal charge = 11.89 x 2 units
  pump = 23.78 gpm

Based on data above, available internal charge pump which is optional for Danfoss M46 PV is not enough to cover charge flow requirement. So wxternal charge pump will be installed.

SAUER DANFOSS

Group 3 Gear Pumps
Technical Information
General Information

TECHNICAL DATA

Specifications for the SNP3 and SEP3 gear pumps

	Unit	22	26	44	48	55	63	75	90
		Frame size							
Displacement	cm^3/rev [in^3/rev]	22.1 [1.35]	26.2 [1.60]	44.1 [2.69]	48.3 [2.93]	55.1 [3.36]	63.4 [3.87]	74.4 [4.54]	88.2 [5.38]
SNP3									
Peak pressure	bar [psi]	270 [3910]	270 [3910]	270 [3910]	250 [3625]	250 [3625]	230 [3350]	200 [2910]	170 [2465]
Rated pressure	bar [psi]	250 [3625]	250 [3625]	250 [3625]	230 [3350]	230 [3350]	210 [3045]	180 [2610]	150 [2175]
Minimum speed	min^{-1} (rpm)	800	800	800	800	800	600	600	600
Maximum speed	min^{-1} (rpm)	3000	3000	3000	3000	2500	2500	2500	2500
Weight	kg [lb]	6.8 [15.0]	6.8 [15.0]	7.5 [16.5]	7.6 [16.8]	7.8 [17.3]	8.1 [17.9]	8.5 [18.7]	8.9 [19.6]
Moment of inertia of rotating components	$\times 10^{-6}$ kg·m^2 [$\times 10^{-6}$ lb·ft^2]	198 [4698]	216 [5126]	294.2 [6891]	312.2 [7408]	342.3 [8123]	378.3 [8977]	426.4 [10118]	486.5 [11545]
Theoretical flow at maximum speed	l/min [US gal/min]	66.3 [17.5]	78.6 [20.8]	132.3 [35.0]	144.9 [38.0]	137.8 [36.2]	157.5 [41.5]	186 [49.1]	220.5 [58.3]

By running SNP3 Gear pump frame size 75 at 2250 rpm with displacement displacement capacity 74.4 cm^3/rev it will produce :

$Q = Vg \times n$

$= 74.4 \times 2250$

$= 167400\ cm^3$

$= 44.22\ gpm$

Where,

Q = additional charge flow required (gpm)

Vg = Displacement per rev (cm^3 / rev)

= 74.4 cm^3 / rev

n = rotation per minute (rpm)

= 2250 rpm (max. rpm 2500)

So total charge pumps supply available :

$Q = Q\ internal + Q\ external$

$= 23.78 + 44.22\ gpm$

$= 68.00\ gpm$

This flow rate sufficient for charge pump requirement.

<u>Selected charge pump</u>

1.	Maker	:	Danfoss Gear Pump Group 3, SNP3 75	
2.	Product type	:	Gear pump, positive displ	
3.	Displacement	:	74.4 cm^3 / rev.	
4.	RPM	:	Min. Speed	600
			Max. Speed	2500
5.	System pressure:	:	Rated pressure	180 bar
			Peak pressure	200 bar

7. Calculation of Planetary Gearbox

a) Holding Rating

From the ship particulars it is known that :

- Total Holding Rating = 1578.1 S-Tons.

  = 1431.4945 Ton (metric)

  Where,

  1 S-Tons = 0.9071 Ton (metric)

- Liftboat leg = 3 units
- Jacks per leg = 8 units
- Total Jacks = 8 units x 3 leg = 24 units
- Holding rating per leg = 1431.4945 / 3 leg

  = 477.16484 Ton

- Jacking rating per jacks = 59.65 Ton

  = 59645.605 kg

b) Holding Force

Force to handled by each jacks calculated as follow :

w = m x g

= 59645.605 x 10

= 596,456.0 N

= 596.45605 kN

Where,

w = weight (N) (kg.m/s^2)

m = mass (kg)

g = gravity acceleration (m/s^2)

c) Torque Required For Holding

τ = Torque (Nm)

F = Force (N)

= 596456.05 N

r = radius (m)

= 0.25 m

$\tau = F \times r$

$= 596456.05 \times 0.25$

$= 149114.01$ Nm

Jack-up drive requirement :

- For jacking

Torque	=	124253.8	Nm	=	1099646.1	in-lbs
Jacking rate	=	44.73	Ton	=	49.31	S-Ton

- For holding

Torque	=	149114.01	Nm	=	1319659	in-lbs
Holding rate	=	59.65	Ton	=	65.75	S-Ton

d) Selection of Jacking Drive

Selected jackup drive :

1. Maker Oerlikon Fairfield
2. Model S130 Jacking Drive
3. Specification
   - Jacking

     Max. Torque = 1,300,000.00 in-lbs

     Max. Jack rate = 90.00 S-Ton
   - Holding

     Max. Torque = 2,330,000.00 in-lbs

     Max. Holding = 158.00 S-Ton

No	Condition	Motor (rpm)	Pinion (rpm)	Ratio
1.	Raising/Lowering hull	200	0.777	1 : 258
2.	Raising/Lowering leg	300	1.553	1 : 193

Table 4.3. Jacking Drive Performance When Raising Hull / Leg

8. Calculation of Hydraulic Brake

Brakes are to be designed to engage automatically in the event of failure of power supply to the motor (fail-safe type). The brake holding capacity is to be at least equal to 120% of the maximum required brake torque associated with the maximum rated load applied to the climbing pinion from all loading conditions (ABS Guide For Building and Classing Liftboat 2009, Part.4, Chap.4, Sect. 13.3)

Based on selected jacking drive, capability of jacking drive S130 at holding condition = 158.00 S-Ton

= 143,335 kg

w = m x g

= 143,335 x 10

= 1,433,352 N

= 1433.3519 kN

Where,

w = weight (N) (kg.m/s^2)

m = mass (kg)

g = gravity acceleration (m/s^2)

τ = F x r

= 14333519 x 0.25

= 3583379.7 Nm

Where,

τ = Torque (Nm)

F = Force (N)

= 14333519 N

r = radius (m)

= 0.25 m

Gear ratio = 1 : 258

Gear ratio = 1 : 258

- Torque for brake :

  τ = 13889.069 / 24 brake units

  = 579 Nm

- Brake capacity = 120% x Max. brake torque

  requirement = 120% x 578.71 Nm

  = 694.45 Nm

TORQUE

Code	Torque Rating N·m	Torque Rating (lb·in)	Initial Release Pressure bar	Initial Release Pressure (PSI)	Full Release Pressure bar	Full Release Pressure (PSI)
98	1107	(9800)	18.6	(270)	25.5	(370)
80	904	(8000)	15.2	(220)	20.7	(300)
70	791	(7000)	13.8	(200)	19.3	(280)
57	644	(5700)	12.4	(180)	19.3	(280)
55	622	(5500)	11.0	(160	15.2	(220)
45	508	(4500)	8.3	(120)	11.7	(170)
36	407	(3600)	6.9	(100)	9.6	(140)
13	1469	(13,000)	24.1	(350)	32.8	(475)

9. Hydraulic Fluid Selection

According to Code of Federal Regulation (CFR) 46, 58.30-10 - regarding hydraulic fluid, The fluid used in hydraulic power transmission systems shall have a flashpoint of not less than 200 °F for pressures below 150 pounds per square inch and 315°F for pressures 150 pounds per square inch and above.

The chemical and physical properties of the hydraulic fluid shall be suitable for use with any materials in the system or components thereof. The hydraulic fluid shall be suitable for operation of the hydraulic system through the entire temperature range to which it may be subjected in service. According to ABS Guide For Building and Classing Liftboat 2009, (Part. 4, Chap. 5, Sect. 1, 1.3), temperatur of hydraulic fluid is over 204°C (400°F)

Service	Pressure bar (kgf/cm2, psi)	Temperature °C (°F)
Vapor and Gas	over 10.3 (10.5, 150)	over 343 (650)
Water	over 15.5 (15.8, 225)	over 177 (350)
Lubricating Oil	over 15.5 (15.8, 225)	over 204 (400)
Fuel Oil	over 10.3 (10.5, 150)	over 66 (150)
Hydraulic Fluid	over 15.5 (15.8, 225)	over 204 (400)

Table 4.4. ABS Liftboat 2009 Guidance on hydraulic fluid

Based on requirements mentioned above, the selected hydraulic fluid using in this system is : **Chevron Hydraulic Oils AW 32**

10. Hydraulic Pipe and Material

According to ABS Guide For Building and Classing Liftboat 2009, (Part. 4, Chap. 5, Sect. 2, 5.9) Allowable Stress Values (S) for Steel Piping is depend on material grade using in their system. To understand material pipe grade can be used in hydraulic system, we have to refer to ABS Rules Under 90 meters (Part. 2, Chap. 3, Sect. 12, 3)

- Grades 1, 2 and 3

Grades 1, 2 and 3 cover seamless and welded steel pipe. Pipe ordered under these grades is of a nominal (average) wall thickness suitable for welding and suitable for forming operations involving coiling, bending and flanging, subject to the following limitations: Grade 1 furnace-butt-welded pipe is not intended for flanging; when seamless or electric-resistance-welded pipe is required for close-coiling or cold-bending, Grade 2 should be specified; this provision is not intended to prohibit the cold-bending of Grade 3 pipe. When pipe is required for close-coiling, this is to be specified on the order. Electric-resistance-welded Grades 2 and 3 may be furnished either non-expanded or cold-expanded, at the option of the manufacturer. When pipe is cold expanded, the amount of expansion is not to exceed 1.5% of the outside diameter pipe size.

- Grades 4 and 5

Grades 4 and 5 cover seamless carbon-steel pipe for high-temperature service. Pipe ordered to these grades is of a nominal (average) wall thickness and is to be suitable for bending, flanging and similar forming operations. Grade 4 rather than Grade 5 pipe should be used for close-coiling, cold-bending or forge-welding; this provision is not intended to prohibit the cold-bending of Grade 5 pipe.

- Grade 6

Grade 6 covers seamless carbon-molybdenum alloy-steel pipe for high-temperature service. Pipe ordered to this grade is of a nominal (average) wall thickness and is to be suitable for bending, flanging (vanstoning) and similar forming operations, and for fusion-welding.

- Grades 7, 11, 12, 13 and 14 (1998)

Grades 7, 11, 12, 13 and 14 cover seamless chromium-molybdenum alloy-steel pipe for high-temperature service. Pipe ordered to these grades is of a nominal (average) wall thickness and is to be suitable for bending, flanging (vanstoning) and similar forming operations, and for fusion-welding.

- Grades 8 and 9

Grades 8 and 9 cover electric-resistance-welded steel pipe 762 mm (30 in.) and under in diameter. Pipe ordered to these grades is of a nominal (average) wall thickness and is intended for conveying liquid, gas or vapor. Only Grade 8 is adapted for flanging and bending; this provision is not intended to prohibit the cold-bending of Grade 9 pipe. The pipe may be furnished either cold-expanded or non-expanded.

ABS Grade	ASTM Designation
1	A53, Grade A, Furnace-welded
2	A53, Grade A Seamless or Electric-resistance-welded
3	A53, Grade B Seamless or Electric-resistance-welded
4	A106, Grade A
5	A106, Grade B
6	A335, Grade P1
7	A335, Grade P2
8	A135, Grade A
9	A135, Grade B
11	A335, Grade P11
12	A335, Grade P12
13	A335, Grade P22
14	A335, Grade P5

Table 4.5. ASTM Designation

By the description above, the selected pipe grade to use in hydraulic system is grade no. 5. From ABS Rules Under 90 meters Part. 2, Chap. 3, Sect. 12, 3.11, ABS grade no.5 is refer to ASTM A 106 B, and preferable pipe based on it manufacture process is seamless pipe.

Maximum Allowable Working Pressure and Minimum Thickness According to ABS Guide For Building and Classing Liftboat 2009, (Part. 4, Chap. 5, Sect. 2, 5.9)

$$W = \frac{KS\,(t-C)}{D-M\,(t-C)} \qquad t = \frac{WD}{KS + MW} + C$$

Where,

W = maximum allowable working pressure, in bar, kgf/cm2 (psi)

t = minimum thickness of pipe, in mm (in.).

K = 20 (Table 1)

D = actual external diameter of pipe, in mm (in.)

S = max allowable fiber stress, N/mm2 (kgf/mm2, psi) From 4-4-2/Table 1

M = 0.8 factor from 4-4-2/Table 1

C = 0.02 allowance for threading, grooving or mechanical strength

= 0.00 mm (0.000 in.) for plain-end steel or wrought-iron pipe or tubing up to 115 mm O.D. (4 in. NPS) used for hydraulic piping systems

$$t = \frac{WD}{KS + MW} + C$$

$$= \frac{300 \times 168.3}{20 \times 103.5 + 0.8 \times 300}$$

$$= \frac{50490}{2310}$$

$$= 21.86 \text{ mm}$$

Where,

S = 103.5 N/mm^2 (ASTM A 106 B)

K = 20

D = 168.3 mm

M = 0.8

C = 0

W = 300 bar

Pipe dimension according to ANSI B36.10

No	Description	SCH	OD (mm)	Thk (mm)	ID (mm)
1	6" Pipe ASTM A 106 B	XXS	168.3	21.95	146.35

CHAPTER V
CONCLUSIONS AND SUGGESTIONS

1. Conclusion

a. Rack and pinion is used as leg mechanism at L/B Cameron Class 200 design
b. Spudcan jetting system at L/B Cameron Class 200 design at two ring :
   - High pressure : 80 bar @ 25 m3/h
   - Low pressure : 12 bar @ 180 m3/h
c. Hydraulic jacking system at L/B Cameron Class 200 design at two operating pressure :
   - Lifting hull: 200 bar @ 88.75% motor disp. and 89.79% pump disp.
   - Lifting leg : 100 bar @ 28.79% motor disp. And 29.27 % pump disp.
   - With configuration each leg : 2 pumps and 8 motors
d. Main Component P&ID of hydraulic jacking system are hydraulic pump, hydraulic motor, charge pump and hydraulic brake. Main component P&ID of spudcan jetting are high pressure pump and low pressure pump.

2. Suggestion

a. The research with other type of hydraulic system is possible
b. For those who interest in gear system, the variety in rack and pinion system make the possibility to be used as a research
c. Research in geotechnical engineering especially in offshore which has relation to spud can jetting system is only a few, so to make deeper research in this field consultation to the expert is required.

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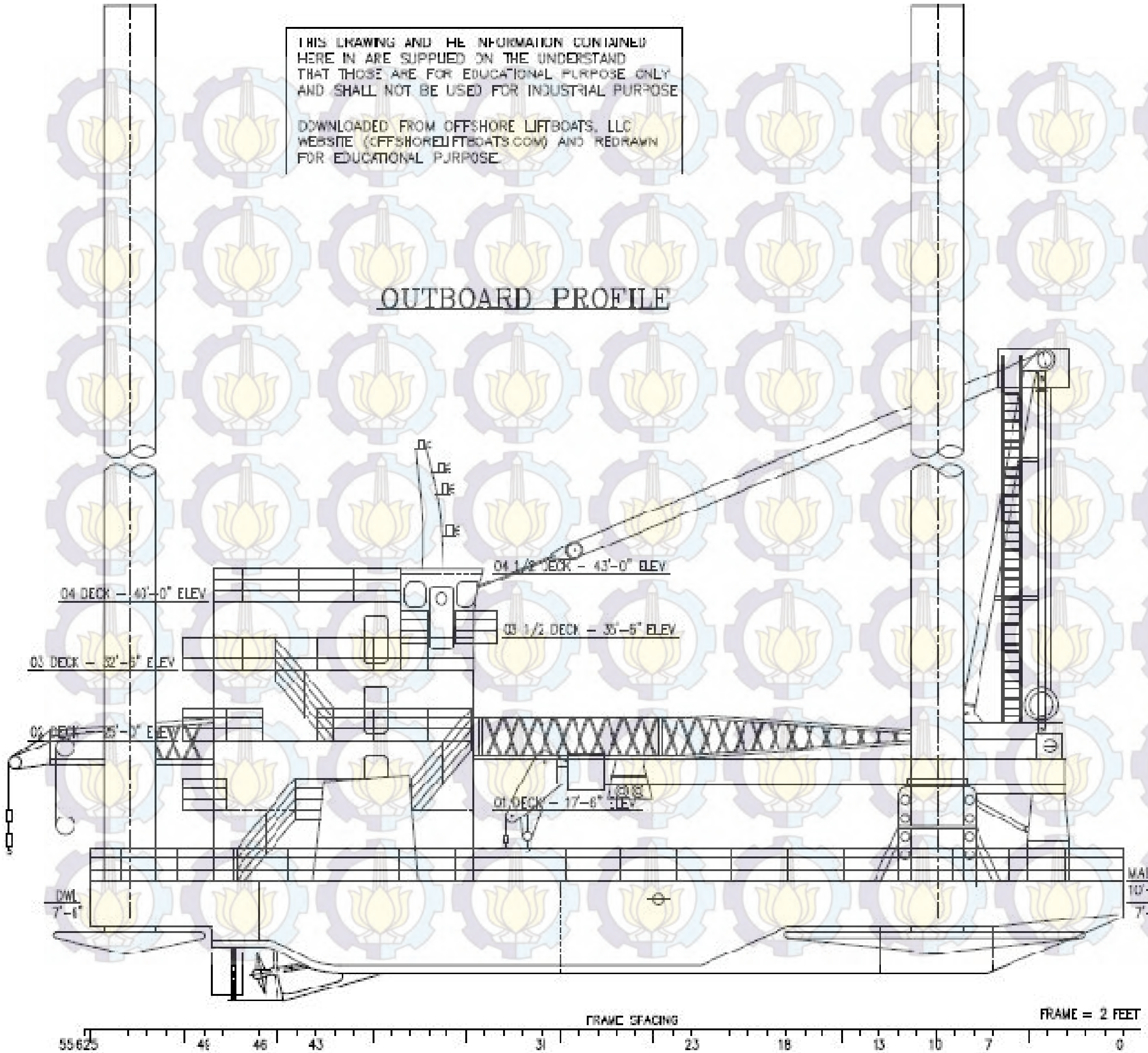
AUTHOR BIOGRAPHY

The author was born in Jember, East Java at 1984. Previous study at Senior High School (SMUN) 1 Jember graduated at 2002 and Surabaya Ship Building Polytechnic at Ship Design and Construction Program, graduated at 2006. Worked in a shipyard in Batam at 2007 as Piping Engineer and 2008 until now working as Technical Consultant for a Marine Consultant Company based in Singapore. Since 2012 continuing the study at Marine Engineering Department, Faculty of Marine Technology. Sepuluh Nopember Institute of Technology. Undergraduate thesis at Marine Machinery System with tittle *"Design of Lifting Operation System (Hydraulic System – Spudcan Jetting System – Leg Mechanism) at Liftboat with L/B Cameron Class 200 as a Case Study"*.

ATTACHMENTS

1. General Arrangement of L/B Cameron Class 200 *(redrawing)*
2. P&ID Hydraulic Jack-up System
3. P&ID Spudcan Jetting System
4. Hydraulic Motor Specification
5. Hydraulic Pump Specification
6. Hydraulic Charge Pump Specification
7. Hydraulic Jacking Drive Specification
8. Hydraulic Brake Specification
9. High Pressure Jetting Pump Specification
10. Low Pressure Jetting Pump Specification



GENERAL INFORMATION

HULL LENGTH	111'-3"
HULL WIDTH	65'-6"
HULL DEPTH	13'-0"
LENGTH OVERALL	115'-0"
WIDTH OVERALL	74'-0"
FOOTING LENGTH	32'-0"
FOOTING WIDTH	15'-0"
FOOTING DEPTH	2'-6"
LEG LENGTH OVERALL	200'-0'
LEG DIAMETER	68'
LEG THICKNESS	1' / 3/4"
LEG MATERIAL	API 5L-X65
JACKS PER LEG	QTY WEIGHT
JACKING RATING/JACK	65.75 S-TONS
HOLDING RATING/JACK	87.67 S-TONS
JACKING RATING/LEG	526.0 S-TONS
HOLDING RATING/LEG	701.3 S-TONS
TOTAL JACKS	QTY 24
TOTAL JACKING RATING	1578.0 S-TONS
TOTAL HOLDING RATING	2104.0 S-TONS
FUEL OIL CAPACITY	9730 GAL
POTABLE WATER CAPACITY	13194 GAL
JACKING HYDRAULIC OIL	1594 GAL
CRANE HYDRAULIC OIL	789 GA
ACCOMODATION CREW	7 MEN
ACCOMODATION VIP	2 MAN
ACCOMODATION FACS	34 MEN
ACCOMODATION TOTAL	43 MEN

FOR M/V KYLIE HALMAR H-153
FOR M/V CAMERON HALMAR H-154

REVISION

DYNAMIC MARINE, INC.
2483 LITTLE LANE-NE, WESSON, MS 39191
601-643-2411

MARINE INDUSTRIAL FABRICATION

OFFSHORE LIFTBOAT, LLC
CLASS 200 LIFTBOAT
OUTBOARD PROFILE

DATE : 10NOV08 | APPV :

MARINE ENGINEERING DEPARTMENT
FACULTY OF MARINE TECHNOLOGY
UNDERGRADUATE THESIS 2013/2014

DESIGN OF LIFTING OPERATION SYSTEM
(HYDRAULIC SYSTEM-SPUD CAN JETTING SYSTEM-LEG MECHANISM)

RE-DRAWN BY: FIRMAN NORMA AKHMAD 4212 105 002 SIGNED	LECTURER 1: IR. HARI PRASTOWO, M.Sc	SIGNED
	LECTURER 2: TAUFIK FAJAR, ST, M.Sc	SIGNED
DATE: 04-08-2014 / SCALE: 1:150 (A3)	DRAWING NO. 01- 42 12 002 -GA	REV: 0

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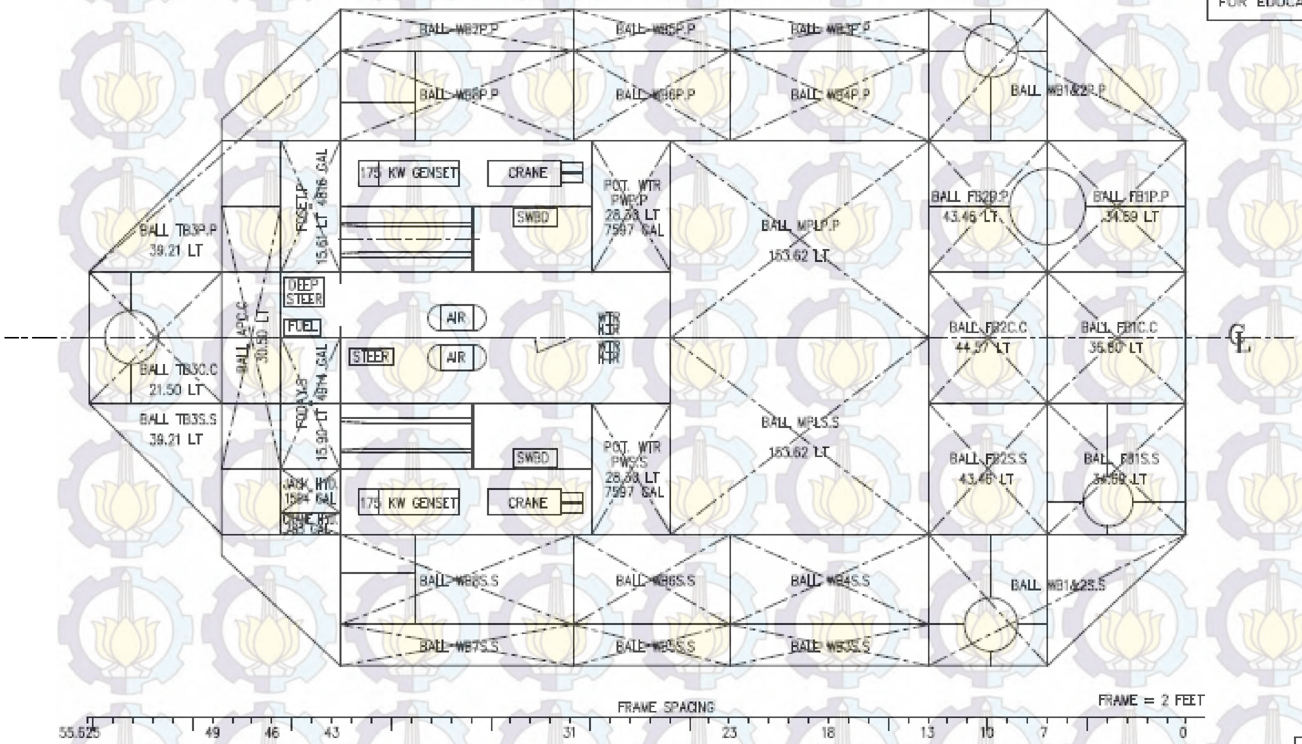

GENERAL ARRANGEMENT

FOR M/V KYLE HALIMAR H-153
FOR M/V CAMERON HALIMAR H-154

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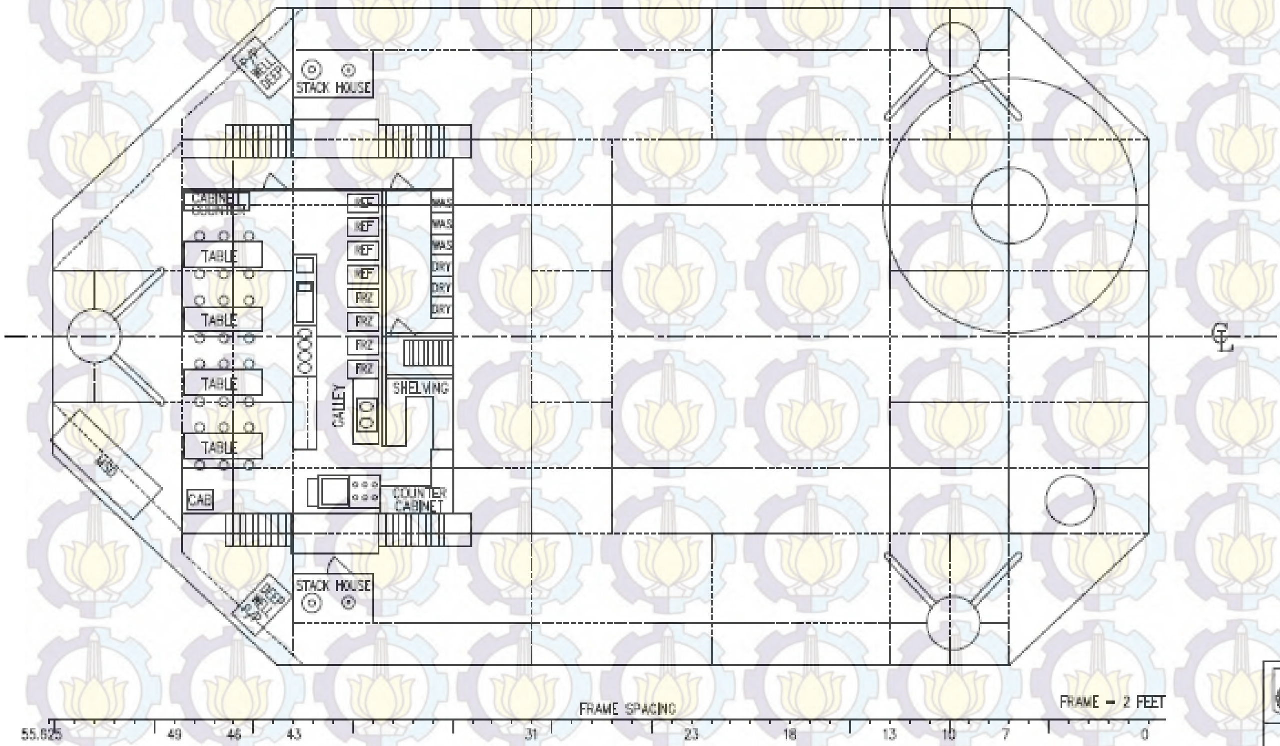

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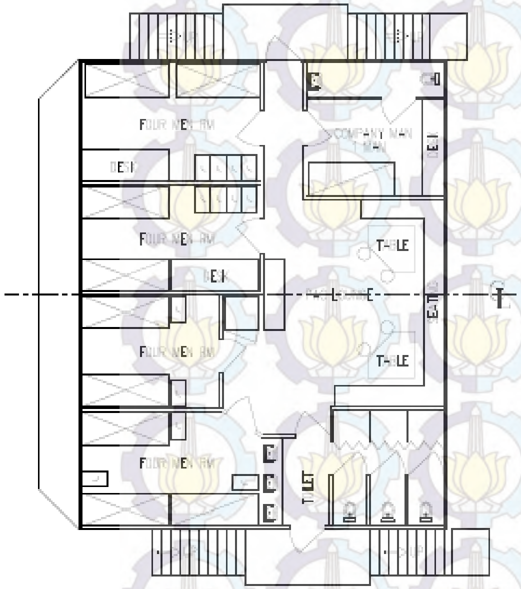
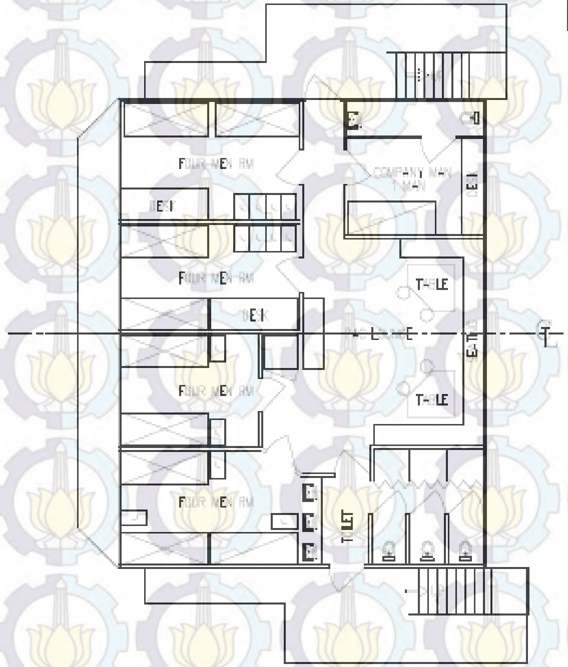


GENERAL ARRANGEMENT
01 LEVEL



GENERAL ARRANGEMENT
02 LEVEL

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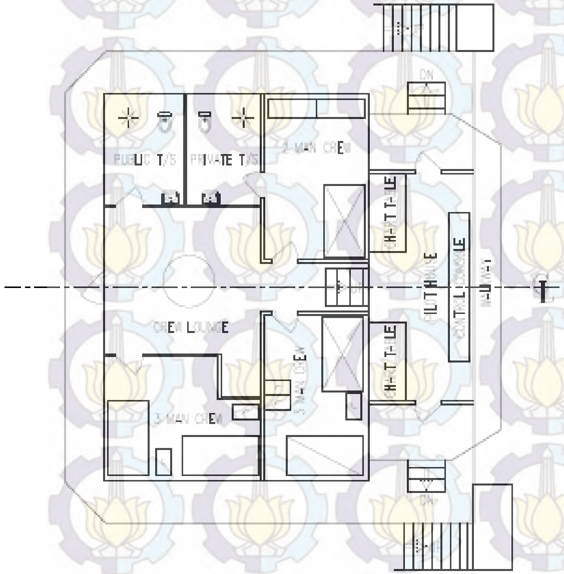
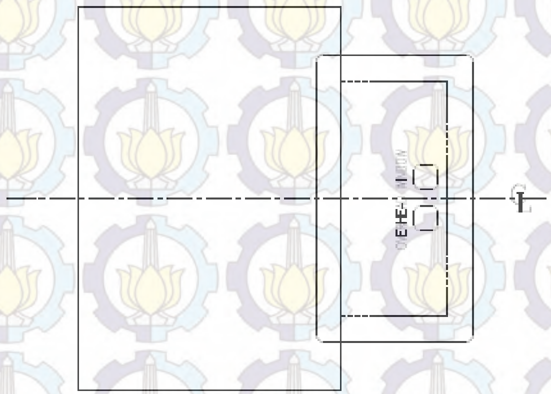


GENERAL ARRANGEMENT
03 LEVEL



GENERAL ARRANGEMENT
04 LEVEL

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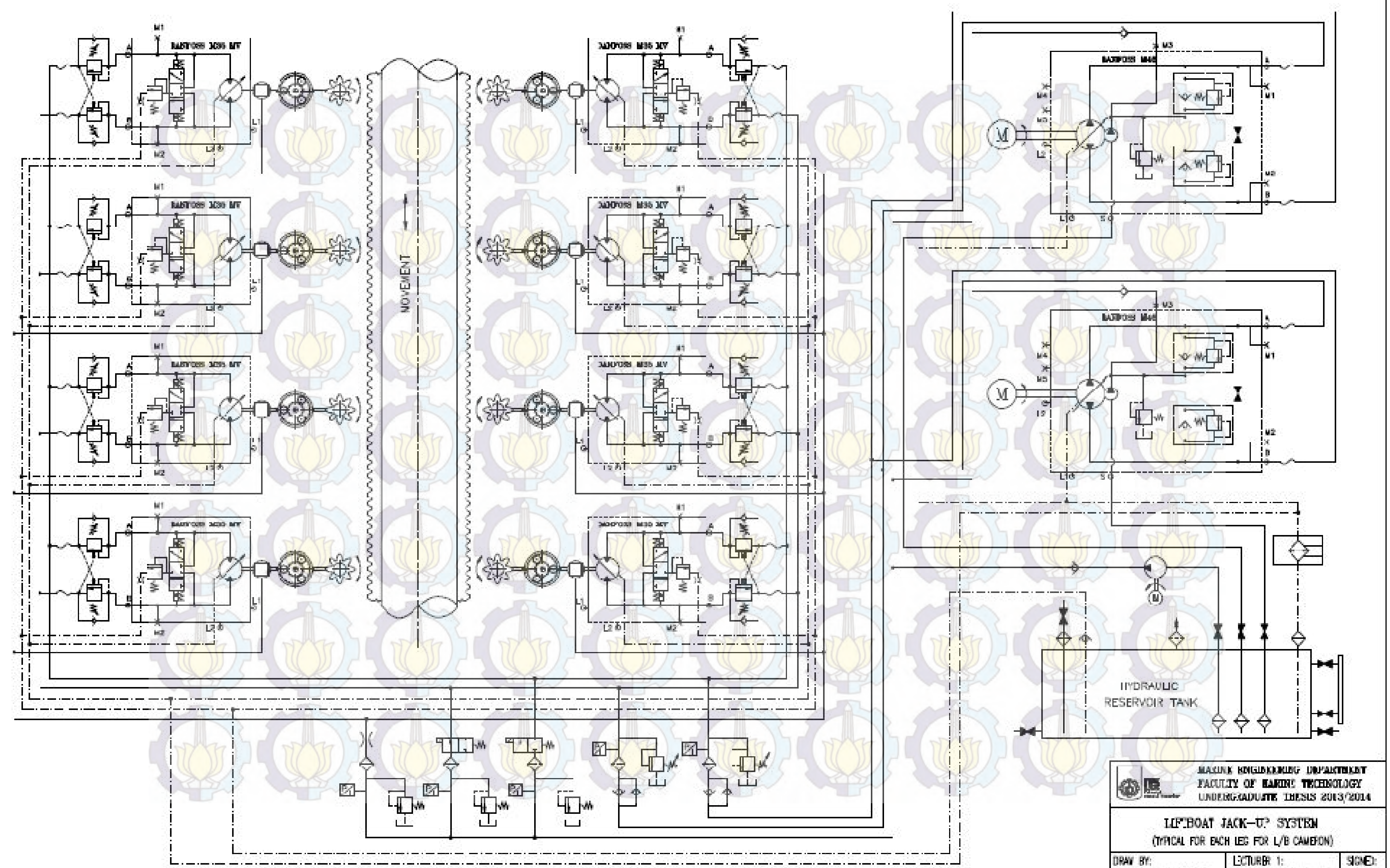

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LIFTBOAT JACK-UP SYSTEM
(TYPICAL FOR EACH LEG FOR L/B CAMERON)

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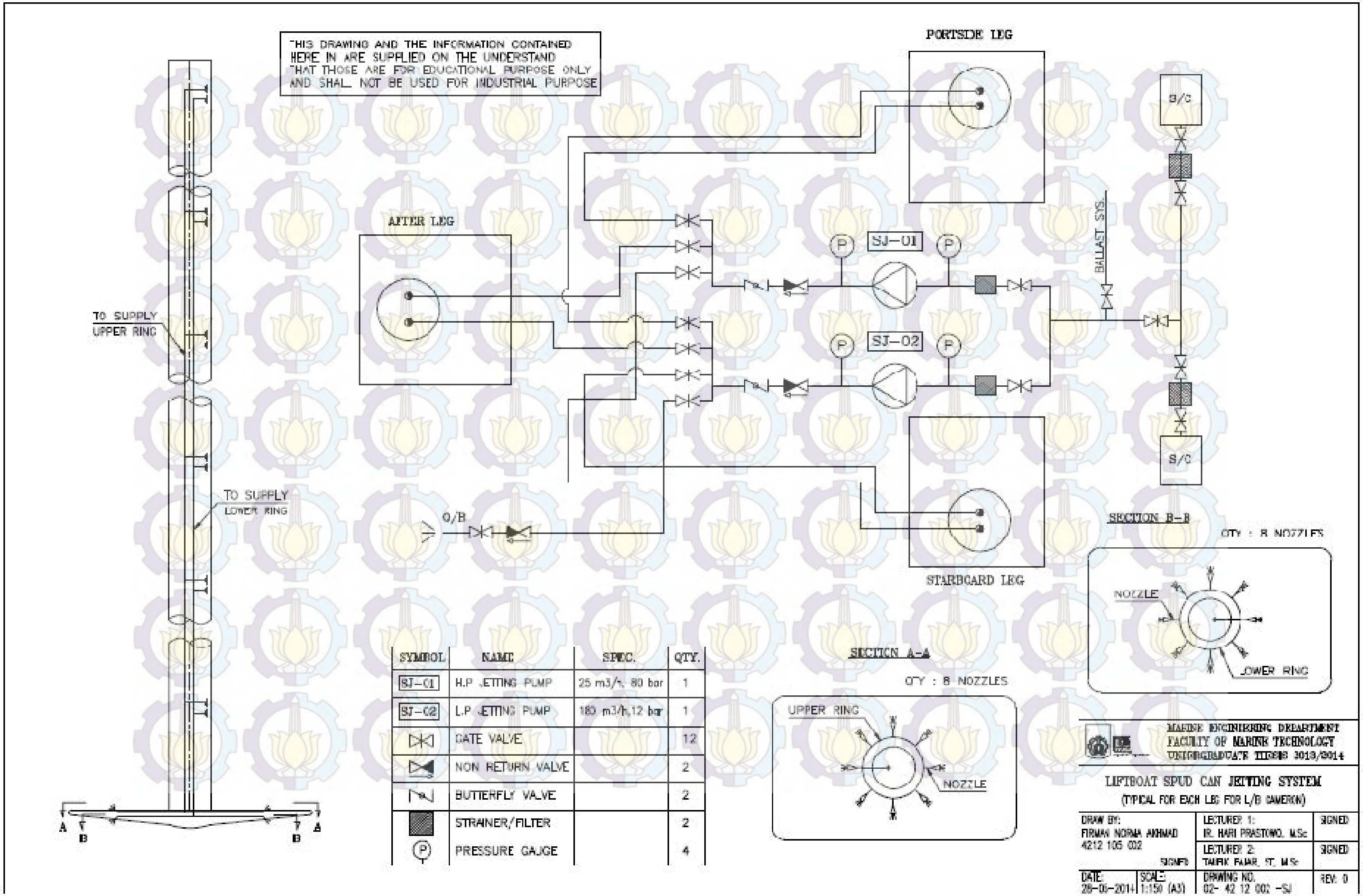

SYMBOL	NAME	SPEC.	QTY.
SJ-01	H.P JETTING PUMP	25 m3/h, 80 bar	1
SJ-02	L.P JETTING PUMP	180 m3/h,12 bar	1
	GATE VALVE		12
	NON RETURN VALVE		2
	BUTTERFLY VALVE		2
	STRAINER/FILTER		2
P	PRESSURE GAUGE		4

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LIFTBOAT SPUD CAN JETTING SYSTEM
(TYPICAL FOR EACH LEG FOR L/B CAMERON)

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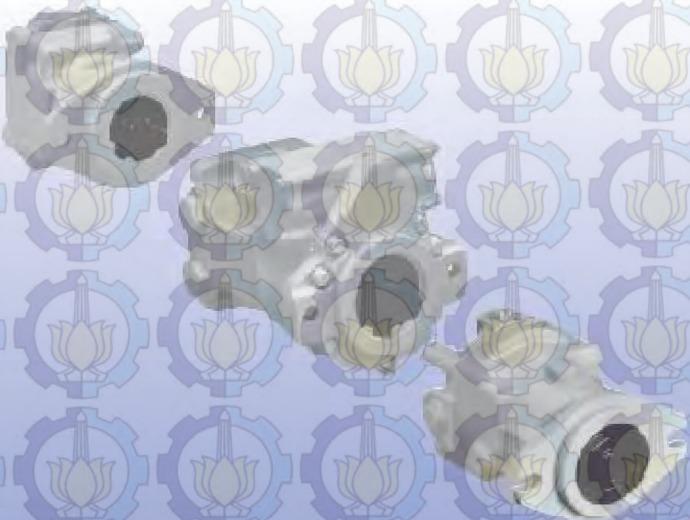
BACK

SAUER
DANFOSS

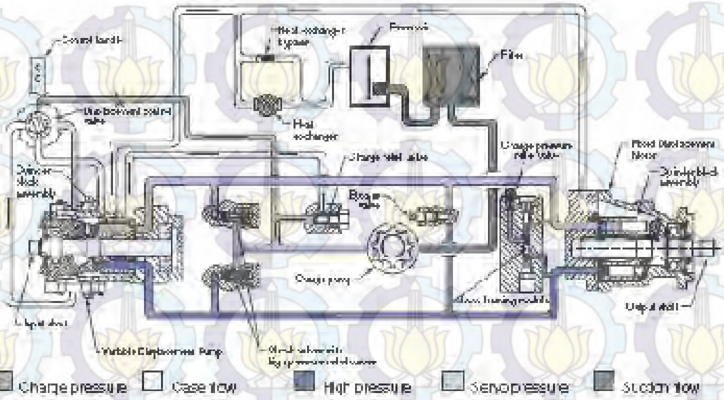
Series 40



Axial Piston Motors

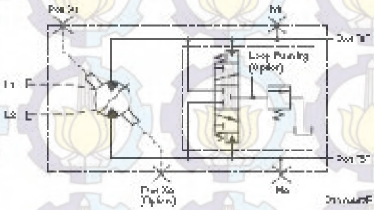
Technical Information

System Circuit Description



A Series 40-M35 fixed motor (right) is shown in a hydrostatic circuit with a Series 40-M46 variable pump. The white half of the circuit includes pump features. A suction filtration configuration is shown. Pressure regulation valves are included on the pump. A loop flushing module is included on the motor. Note the position of the reservoir and heat exchanger.

Motor Circuit Description



A Series 40 - M46 variable motor circuit schematic is shown above. The system ports "A" and "B" hook to the high pressure work lines. The motor receives pressurized fluid in its inlet port and discharges de-energized fluid through the outlet port. Either port can act as inlet or outlet; flow can be bidirectional. System port pressure can be gauged through ports M1 and M2. The motor has two case drains (L1 and L2). The motor may or may not include loop flushing. Loop flushing provides additional cooling and filtration capacity.

Technical Specification

General Specification

Specifications for Series 40 motors are listed on these two pages. For definitions of the various specifications, see the related pages in this publication. Not all hardware options are available for all configurations; consult the Series 40 Motor Model Code Supplement or Price Book for more information.

General Specifications	
Motor Type	In-line, axial piston, positive displacement motors.
Direction of Rotation	Bidirectional, see outline drawings for rotation vs flow direction information.
Installation Position	Discretionary, the housing must be filled with hydraulic fluid.
Filtration Configuration	Suction or charge pressure filtration
Other System Requirements	Independent braking system, circuit overpressure protection, suitable reservoir

Table 1E

Specific Data

Specific Data								
Frame Size		N25 MF	N35 MF	N44 MF	N46 MF	N35 MV	N44 MV	N46 MV
Motor Configuration		Fixed Motor				Variable Motor		
Displacement	cm³/rev	25	35	44	46	35	44	46
	in³/rev	1.50	2.14	2.65	2.80	2.14	2.65	2.80
Weight	kg	11	11	11	14	21	21	23
	lb	25	25	25	30	47	47	51
Mass moment of inertia of the internal rotating parts	kg·m²	0.0017	0.0029	0.0028	0.0046	0.0028	0.0028	0.0049
	lb·ft²	0.040	0.067	0.066	0.110	0.067	0.066	0.116
Two (2) bolt flange, size B (SAE J744)		O	O	O	O	O	O	O
Cartridge flange		-	-	-	-	-	-	O
Port connection SAE straight thread O-ring boss	axial	O	O	O	O	-	-	O
	side		O	O	O			O
	twin	O	O	O	O	O	O	O
Output shaft options	tapered	-	O	O	O	-	-	O
	straight key	-	O	O	O	-	-	-
	splined	O	O	O	O	O	O	O
Control options		-	-	-	-	DDC	DDC	HDC
Loop flushing		O	O	O	O	O	O	O
Displacement limiters		O	O	O	O	O	O	O
Speed sensors		O	O	O	O	-	-	O

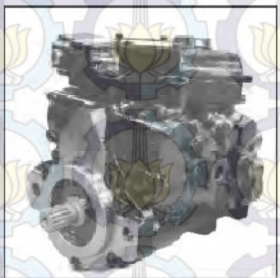
Table 2E

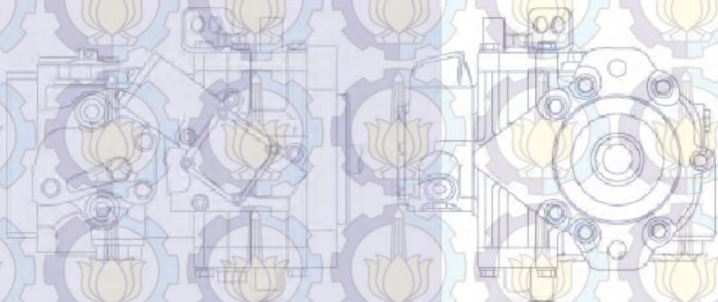
O = Option
- = not available

M46

Single and Tandem Axial Piston Pumps

Technical Information





Design Specifications

Product line	Series 40 Pumps
Pump type	In-line, axial piston, variable, positive displacement pumps
Direction rotation	Clockwise (CW) or counterclockwise (CCW) available
Installation position	Discretionary, the housing must be filled with hydraulic fluid
Filtration configuration	Suction or charge pressure filtration
Other system requirements	Independent braking system, suitable reservoir and heat exchanger

Technical Specifications

Model		Unit	M46 Single Pump	M46 Tandem Pump
Displacement		cm^3/rev [in^3/rev]	45.9 [2.80]	45.9 x 2 [2.80 x 2]
Shaft Speed	Minimum	min^{-1} (rpm)		
	Rated	min^{-1} (rpm)	4000	
	Maximum	min^{-1} (rpm)	4100	
System Pressure	Maximum working*	bar [psi]	345 [5000]	
	Maximum		385 [5585]	
	Minimum low loop		10 [145]	
Weight (MDC without aux pad)		kg [lb]	33 [73]	59 [131]
Mass moment of inertia of the rotating components		kg·m^2 [slug·ft^2]	0.0050 [0.0037]	0.0100 [0.0073]
Charge Pressure	Minimum	bar [psi]	6 [87]	
	Maximum		31 [450]	
Control Pressure	Minimum @ corner power	bar [psi]	21.5 [312]	
Case Pressure	Continuous	bar [psi]	1.7 [25]	
	Maximum (cold start)		5.2 [75]	
Inlet Pressure	Rated	bar absolute	0.8 [6]	
	Minimum	[inches of Mercury vacuum]	6 [9.2 Maximum]	

* Operation above maximum working pressure is permissible with Sauer-Danfoss application approval.

Operating Parameters

Fluid Viscosity	Minimum	mm²/s (cSt) [SUS]	7 [49]
	Continuous		12-16 [70-278]
	Maximum (cold start)		1600 [7 500]
	Intermittent (1)		6 [46]
Fluid Temperature	Minimum (intermittent cold start)	degrees C [degrees F]	- 40°C [- 40° F]
	Continuous		82.2°C [180°F]
	Maximum intermittent (1)		104.4°C [220°F]

(1) Intermittent equals a short period of time at less than one minute per incident and not exceeding two percent of duty cycle based on load life.

Options

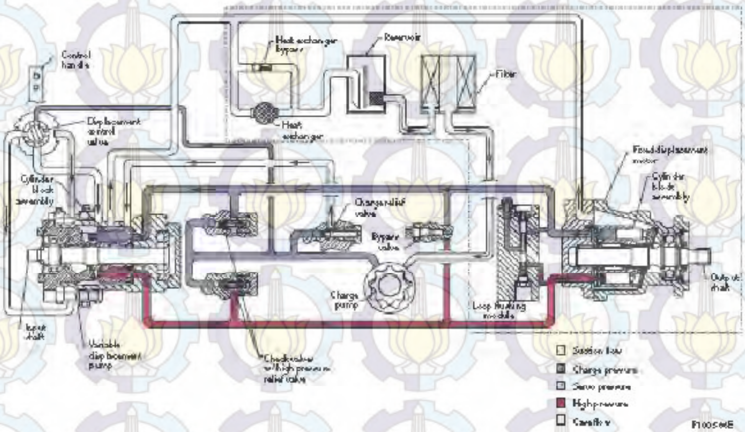
		Single	Tandem
Mounting Flange	SAE - B	X	X
Input Shaft	Ø 25.4 mm [1.000 in] straight keyed	X	X
	Ø 25.4 mm [1.000 in] 1:8 taper (SAE J501)	X	X
	13-tooth, 16/32 pitch (ANSI B92.1 1970 - Class 5)	X	-
	15-tooth, 16/32 pitch (ANSI B92.1 1970 - Class 5)	X	X
	19-tooth, 16/32 pitch (ANSI B92.1 1970 - Class 5)	-	X
Auxiliary Mounting Flange	9-tooth internal spline, 16/32 pitch (SAE A)	X	X
	11-tooth internal spline, 16/32 pitch (SAE A)	X	X
	13-tooth internal spline, 16/32 pitch (SAE B)	X	X
	15 - tooth internal spline, 16/32 pitch (SAE BB)	-	X
Main Port Configuration	1-5/16 - 12 SAE A straight thread O-ring Ports (SAE J514)	X	X

Fluid Specifications

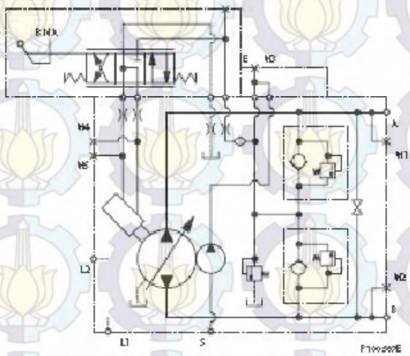
Ratings and data are based on operation with premium petroleum-based hydraulic fluids containing oxidation, rust, and foam inhibitors.

Parameter		Unit	Minimum	Continuous	Maximum
Viscosity		mm²/sec (cSt) [SUS]	7 [47]	12-60 [70-278]	1600 [7500]
Temperature		°C [°F]	-40 [-40]	82 [180]	104 [220]
Cleanliness			ISO 4406 Class 18/13 or better		
Filtration efficiency	suction filtration		$\beta_{35-44}=75$ ($\beta_{10}\geq1.5$)		
	charge filtration		$\beta_{15-20}=75$ ($\beta_{10}\geq10$)		

System Circuit Diagram

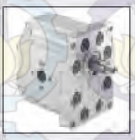
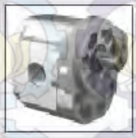
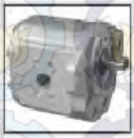
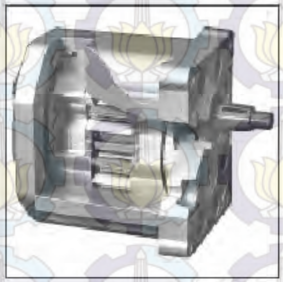


M46 Pump Schematic Diagram



Group 3
Gear Pumps

Technical Information

PUMP DESIGN

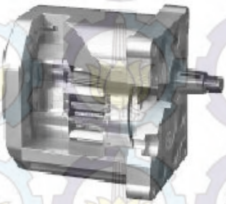
SEP3

The SEP3 gear pump is available in a limited displacement range from 22.0 to 44.1 cm^3/rev [from 1.34 to 2.69 in^3/rev]. Suitable for applications where the pressure is lower than 210 bar [3045 psi], the SEP3 range is released into SAE and European configurations. The overall length is reduced by 12 mm [0.47 in] in respect of the SNP3.

SNP3

The SNP3 is available in the full displacement range from 22.0 to 88.2 cm^3/rev [from 1.34 to 5.38 in^3/rev], and with higher pressure ratings than the SEP3. This is due to the pressure balance on each side of the gears obtained with pressure-balance plates made in antifriction alloy that contribute to high volumetric efficiency and maximum sealing as well.

SNP3 C001 (cutaway)



[illegible]

TECHNICAL DATA

Specifications for the SNP3 and SEP3 gear pumps

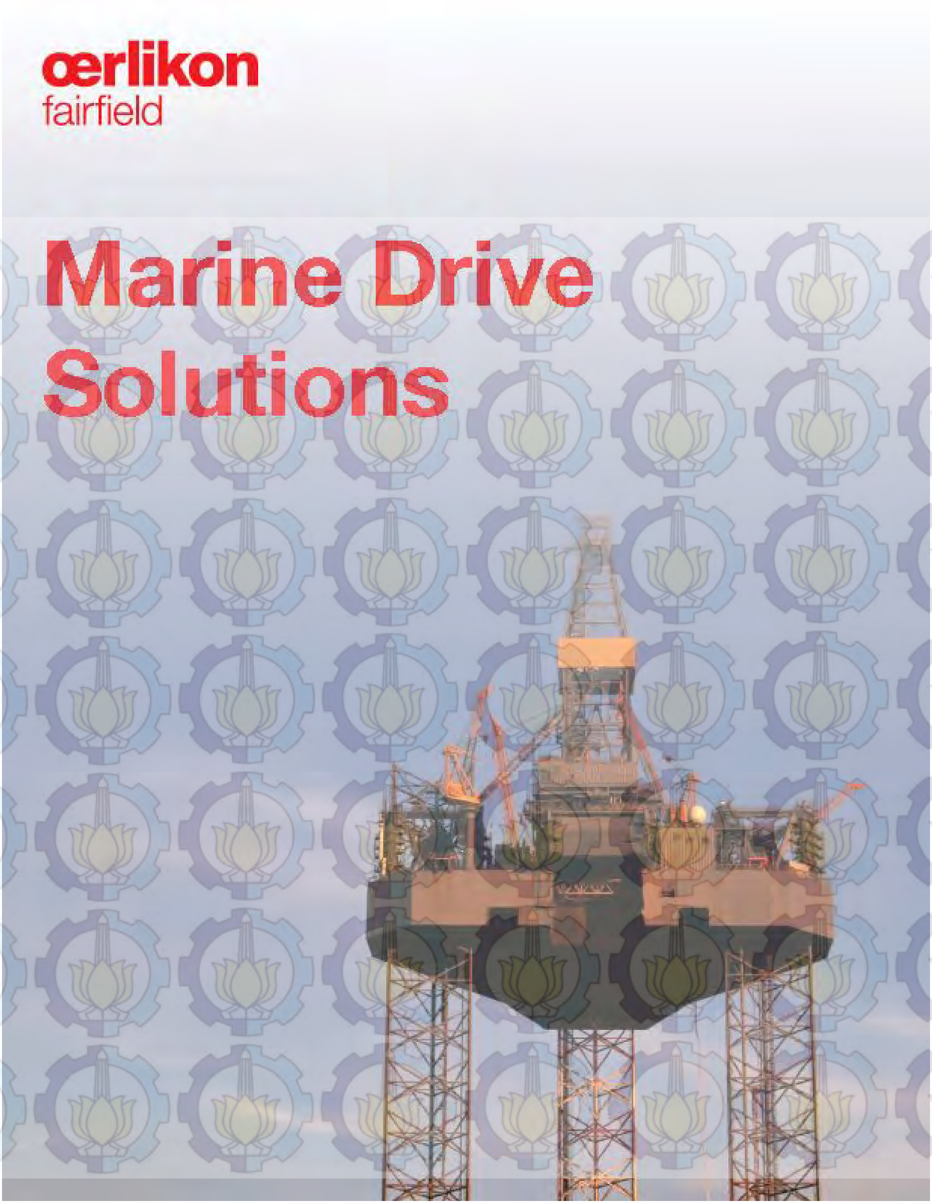
	Unit	Frame size 22	26	33	38	44	48	55	63	75	90
Displacement	cm^3/rev [in^3/rev]	22.1 [1.35]	26.2 [1.60]	33.1 [2.02]	37.9 [2.32]	44.1 [2.69]	48.3 [2.93]	55.1 [3.36]	63.4 [3.87]	74.4 [4.54]	88.2 [5.38]
SNP3											
Peak pressure	bar [psi]	270 [3910]	270 [3910]	270 [3910]	270 [3910]	270 [3910]	250 [3625]	250 [3625]	230 [3350]	200 [2910]	170 [2465]
Rated pressure	bar [psi]	250 [3625]	250 [3625]	250 [3625]	250 [3625]	250 [3625]	230 [3350]	230 [3350]	210 [3045]	180 [2610]	150 [2175]
Minimum speed	min^{-1} (rpm)	800	800	800	800	800	800	800	600	600	600
Maximum speed	min^{-1} (rpm)	3000	3000	3000	3000	3000	3000	2500	2500	2500	2500
Weight	kg [lb]	6.8 [15.0]	6.8 [15.0]	7.2 [15.8]	7.3 [16.1]	7.5 [16.5]	7.6 [16.8]	7.8 [17.2]	8.1 [17.9]	8.5 [18.7]	8.9 [19.6]
Moment of inertia of rotating components	$x\ 10^{-6}$ kg·m^2 [$x\ 10^{-6}$ lb·ft^2]	196 [4698]	216 [5126]	246 [5838]	267.2 [6340]	290.3 [6891]	312.2 [7408]	342.3 [8123]	378.3 [8977]	426.4 [10118]	486.5 [11545]
Theoretical flow at maximum speed	l/min [US gal/min]	66.3 [17.5]	78.6 [20.8]	99.3 [26.2]	113.7 [30.0]	132.3 [35.0]	144.9 [38.0]	137.8 [36.2]	157.5 [41.5]	186 [49.1]	220.5 [58.3]
SEP3 (01 and 02 configuration)											
Peak pressure	bar [psi]	230 [3350]	230 [3350]	230 [3350]	230 [3350]	200 [2910]					
Rated pressure	bar [psi]	210 [3045]	210 [3045]	210 [3045]	210 [3045]	180 [2610]					
Minimum speed	min^{-1} (rpm)	1000	1000	1000	1000	800					
Maximum speed	min^{-1} (rpm)	3000	3000	3000	2800	2600					
Weight	kg [lb]	5.7 [12.57]	5.8 [12.79]	6.1 [13.45]	6.2 [13.67]	6.4 [14.11]					
Moment of inertia of rotating components	$x\ 10^{-6}$ kg·m^2 [$x\ 10^{-6}$ lb·ft^2]	198 [4698]	216 [5126]	246 [5873]	294.2 [6981]	312.2 [7408]					
Theoretical flow at maximum speed	l/min [US gal/min]	66.3 [17.5	78.6 [20.8]	99.3 [26.2]	113.7 [30.0]	132.3 [35.0]					

Caution

The rated and peak pressure mentioned are for pumps with flanged ports only. When threaded ports are required a de-rated performance has to be considered. To verify the compliance of an high pressure application with a threaded ports pump apply to a Sauer-Danfoss representative.

oerlikon
fairfield

Marine Drive Solutions



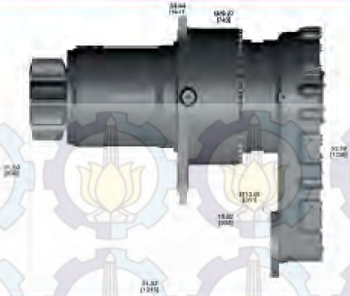
With more than 30 years experience in the industry, Fairfield has the engineering, design and application expertise to ensure your equipment is powered with precisely the right drive system—built to deliver the smooth, reliable performance that gives you a strong competitive edge.

Jack-Up Leg Drives

Look to Fairfield for a wide range of engineered drive solutions that serve a variety of offshore applications.

- Jack-up leg drives for workboats and offshore platforms
- Swing drives for cranes and boom rotation
- Winch and hoist drives
- Custom-designed drives for unique offshore applications
- Products certified by the American Bureau of Shipping (ABS) and Det Norske Veritas (DNV)

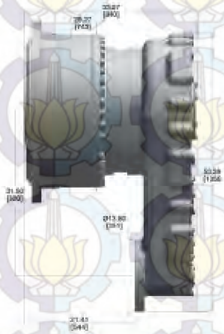
High-Torque Jack-Up Drives



S130 Jacking Drive

Max. Jacking (in.lb.)	Short Ton	KIPS	Max. Holding (in.lb.)	Short Ton	KIPS	Storm Holding (in.lb.)	Short Ton	KIPS	Ratio	Design Temp
1,300,000	90 ST	180 Kips	2,330,000	158 ST	316 Kips	3,100,000	210 ST	420 Kips	2028:1 Shown (Other Ratios Available)	-10°C

* Pinion load ratings assume pitch radius of 7.38'



S650 Jacking Drive

Max. Jacking (in.lb.)	Short Ton	KIPS	Max. Holding (in.lb.)	Short Ton	KIPS	Storm Holding (in.lb.)	Short Ton	KIPS	Ratio	Design Temp
6,000,000	256 ST	518 Kips	9,000,000	380 ST	760 Kips	12,000,000	507 ST	1,014 Kips	2,175:1 Shown (Other Ratios Available)	0°C

* Pinion load ratings assume pitch radius of 11.8'

Reference for all Jacking Systems:

S. Ton Rating = in.lb. / (pinion P.R.x2000)

Kips = (short ton x 2000) / 1000

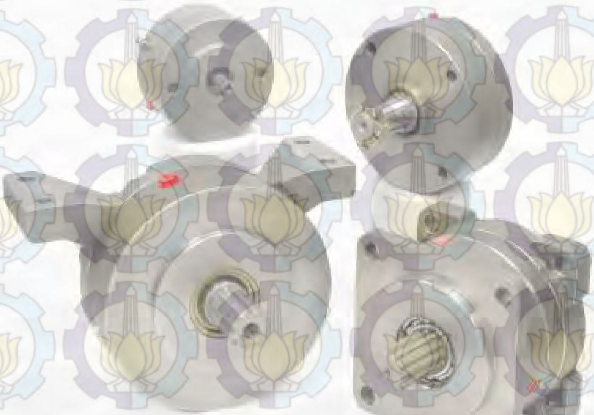
** Pinion spline interface to customer specifications

MICO®

Innovative Braking and Controls Worldwide

Multiple Disc Brakes

posi-torque winch brakes, pressure override brakes,
wheel mount brakes, and driveline brakes



Spring Apply Hydraulic Release Multiple Disc Brakes

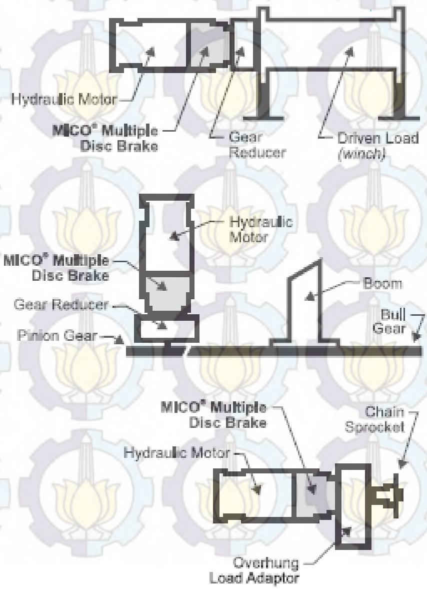
![MICO]

Multiple Disc Brakes
(spring apply, hydraulic release)

MICO engineers are innovators in the design of spring apply, hydraulic release multiple disc brakes, wheel brakes, closed-output motor brakes, posi-torque winch brakes and more. The engineers are committed to improving the product while reducing cost. Simple, straight forward designs result in rugged brake products. These products require less maintenance because they are designed with fewer moving parts. They are truly superior in reliability and performance.

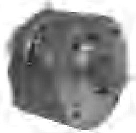
MICO® Multiple Disc Brakes are designed for use with heavy-duty machinery and off-highway vehicles in the construction, material handling, agriculture, mining, sanitation, utilities and timber industries. They are also used in a multitude of winching applications. Brakes of this type reduce maintenance and downtime by preventing contaminants, which cause brake lining wear, from entering the brake. They will provide consistent braking torque, positive hold, and long life in rugged environments.

Typical Multiple Disc Brake Applications



MICO®

C-Mount Pressure Override Brakes, Modular Design

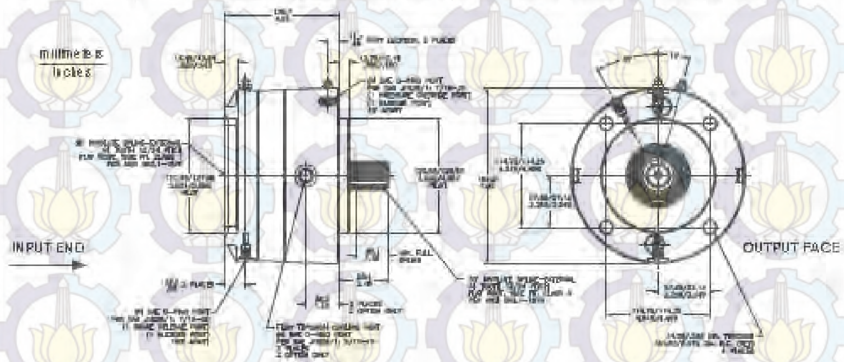


FEATURES

- Secondary system for service braking with fail-safe backup
- Standard SAE mounting flanges
- Service brake can be modulated with automotive type master cylinder or hydraulic valve
- Oil cooled option for added capacity
- Nitrile case seals
- Compact modular design

13-597-008
(SPC-141470-CZ)

For detailed information on other model numbers go to www.mico.com/service-literature/installation-drawing-search. If the drawing is not available, contact MICO for more information.



SPECIFICATIONS

FAIL-SAFE BRAKE

Torque range at 0 bar (0 PSI) back pressure	407 - 1469 N·m (3600 - 13,000 lb·in)
Release pressure range	9.7 - 25.5 bar (140 - 370 PSI)
Maximum continuous pressure	206.8 bar (3000 PSI)
Maximum speed	4000 RPM (See note below)
Volume of oil to release brake	16.4 cm³ (1.0 in³)
Fluid type	Mineral base hydraulic oil
Maximum operating temperature	132 °C (270 °F)
Approximate weight	20 kg (44 lb)
Optimal flow through cooling (wet design)	3.8 - 26.5 L/min (1 - 7 GPM)
Maximum case pressure	2.1 bar (30 PSI)
Sump cooling fluid volume (wet design) (horizontal)	118.3 mL (4 fl oz)
(vertical)	Contact MICO

SERVICE BRAKE

Maximum torque	(dry design) 1062 N·m (9400 lb·in) (wet design) 700.6 N·m (6200 lb·in)
Calculated torque	(dry design) T = 10.10 x (PSI - 70) (wet design) T = 6.66 x (PSI - 70)
Maximum operating pressure	69.0 bar (1000 PSI)
Maximum energy input (wet or dry design)	406,800 joule (300,000 ft·lb) (one stop, no damage)
Maximum energy input rate (dry design)	101,700 joule/s (75,000 ft·lb/s) (one stop, no damage)
(wet design)	203,400 joule/s (150,000 ft·lb/s) (one stop, no damage)
Piston volume	5.2 cm³ (0.32 in³)
Fluid type	Mineral base hydraulic oil

NOTE: Due to energy capacity limitations, maximum speed at time of service apply is dependent on product application.

CATALOG CODE (See NOTE on the top of page 6)

Not all of the brake combinations are possible due to certain design limitations.

NOTE: On oil-cooled models (Z option) actual torque is 67% of value shown on torque code chart.

3 P C - □ □ □ □ □ □ - □ □ □ □ □

3P - PRESSURE OVERRIDE

OUTPUT FACE

C - SAE C-Mount 4-Bolt

OUTPUT SPLINE / INPUT SPLINE

1400
1406
14/13
14/14
23/23

SAE Designation
00 = used with "R" only
06 = 25.4 mm (1.00 in) diameter 6B
13 = 13T 8/16
14 = 14T 12/24
23 = 23T 16/32

For other configurations, consult a MICO Applications Specialist.

NOTE: MICO recommends that all applications for pressure override brakes have a completed Data Sheet submitted to the MICO Application Department. Complete the online Application Data Sheet (80-500-010) at www.mico.com/products/brakes/multiple-disc-brakes.

OPTION

(Available separately or in combination)

S - Speed Sensor

Z - Oil Cooled - see note above

INPUT FACE

C - SAE C-Mount Standard
C2 - SAE C-Mount 2-Bolt
C24 - 2 Bolt and 4-Bolt C-Mount
D - SAE D-Mount
K4 - Eaton Standard 4000
M - 4-Bolt and SAE A-Mount 2-Bolt
R - Closed Face

See page 48 for Input Face Dimensions

TORQUE

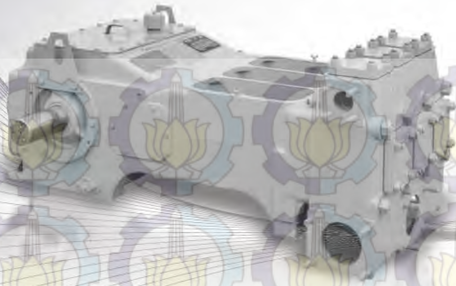
Code	Torque Rating N·m	(lb·in)	Initial Release Pressure bar	(PSI)	Full Release Pressure bar	(PSI)
98	1107	(9800)	18.6	(270)	25.5	(370)
80	904	(8000)	15.2	(220)	20.7	(300)
70	791	(7000)	13.8	(200)	19.3	(280)
57	644	(5700)	12.4	(180)	19.3	(280)
55	622	(5500)	11.0	(160)	15.2	(220)
45	508	(4500)	8.3	(120)	11.7	(170)
36	407	(3600)	6.9	(100)	9.6	(140)
13	1469	(13,000)	24.1	(350)	32.8	(475)

Other torques and/or release pressures are available upon request.

ASSIGNED NUMBERS

CATALOG CODE	MODEL NUMBER
3PC-140080-R	13-597-032
3PC-140645-M	13-597-034
3PC-140670-MZ	13-597-044
3PC-140680-M	13-597-026
3PC-141313-DZ	13-597-052
3PC-141398-DZ	13-597-046
3PC-141436-C	13-597-014
3PC-141436-C24	13-597-042
3PC-141436-M	13-597-040
3PC-141445-C24Z	13-597-030
3PC-141445-K4	13-597-072
3PC-141455-C	13-597-002
3PC-141455-CZ	13-597-016
3PC-141457-CZ	13-597-050
3PC-141470-C	13-597-004
3PC-141470-CZ	13-597-008
3PC-141470-C24Z	13-597-054
3PC-141480-C	13-597-018
3PC-141480-CZ	13-597-024
3PC-141498-C	13-597-010
3PC-141498-CZ	13-597-022
3PC-141498-C24Z	13-597-056
3PC-232380-CZ	13-597-070

THE Horizontal Drilling Duty Pump



SPECIFICATIONS

Maximum Input	*215 BHP (160 kw)*
Maximum RPM	*300 RPM*
Number of Plungers	*3*
Plunger Load	*37,000 lbs. (16,985 kg)*
Discharge Hydrostatic Test Pressure	*3,600 PSI (1,633 kg/cm2)*
Oil Capacity	*12 gal. (45 liters)*
Pump Weight	*3,990 lbs. (1,810 kg)*

PISTON/LINER SIZE		DISPLACEMENT PER REVOLUTION		DISPLACEMENT @ PUMP RPM											
				50				100				150			
in.	mm.	GAL	Liter	GPM	LPM	PSI	kg/sq. cm	GPM	LPM	PSI	kg/sq. cm	GPM	LPM	PSI	kg/sq. cm
4	102	.816	3.089	41	154	2944	206	82	309	2944	206	122	463	2718	191
4.5	114	1.032	3.907	52	195	2326	163	103	391	2326	163	155	588	2100	148
5	127	1.274	4.824	64	241	1884	132	127	482	1884	132	191	724	1700	120
5.5	140	1.542	5.837	77	292	1557	109	154	584	1557	109	231	876	1435	100
Input Power:			BHP	79				156				215			
			kW	59				116				160			

PISTON/LINER SIZE		DISPLACEMENT PER REVOLUTION		DISPLACEMENT @ PUMP RPM											
				200				250				300			
in.	mm.	GAL	Liter	GPM	LPM	PSI	kg/sq. cm	GPM	LPM	PSI	kg/sq. cm	GPM	LPM	PSI	kg/sq. cm
4	102	.816	3.089	163	618	2034	143	204	772	1625	114	245	927	1354	95
4.5	114	1.032	3.907	206	781	1600	112	258	977	1280	90	310	1173	1070	75
5	127	1.274	4.824	255	965	1300	91	319	1206	1040	73	383	1448	870	61
5.5	140	1.542	5.837	308	1167	1076	76	386	1460	860	60	462	1752	720	51
Input Power:			BHP	215				215				215			
			kW	160				160				160			

A lubrication system supplying oil at 50 psi, 12 gpm and 200 SUS (44cSt) minimum values is required. Oil cooling recommended and may be required to maintain these values.

For operation in the Yellow shaded area expendable part life decreases and excellent NPSH is required to prevent cavitation.

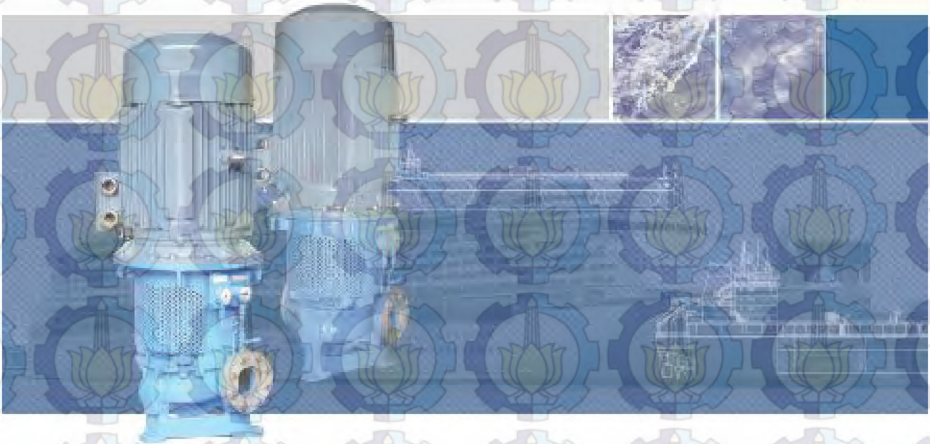
Performance figures based on 100% volumetric efficiency and 90% mechanical efficiency. These figures are for use in directional drilling duty cycle.

This equals full rod load 33% of the time, 80% rod load 50% of the time and 25% rod load 17% of the time, for a total of 1456 hours per year.

Gardner Denver

Pumping Perfected.

Hamworthy

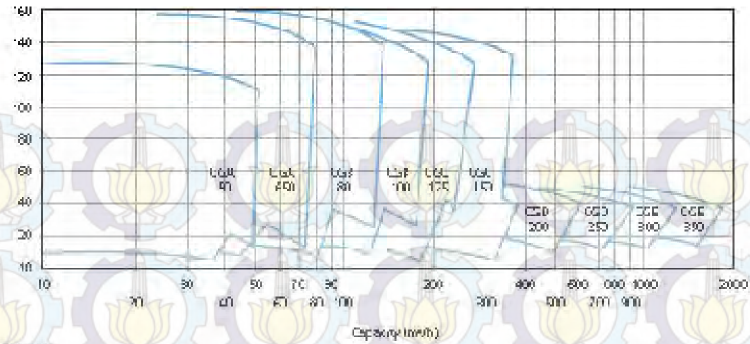


Engine Room Pumps

Model CG

Pump Systems

Capacity Range

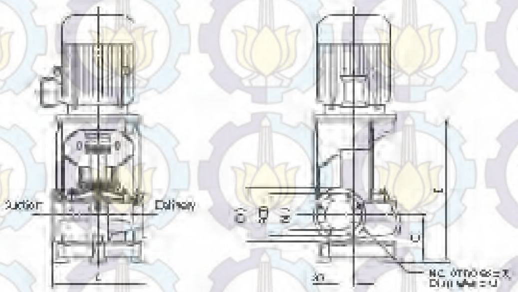


Materials

Duty	Sea Water	Fresh Water
Casing	Bronze	Cast Iron
Impeller	Ni. Al. Bronze	Ni. Al. Bronze
Shaft	Stainless Steel	Stainless Steel
Sealing	Mech. Seal	Mech. Seal
Bearing	Ball/Bush	Ball/Bush

Other materials are available on request.

Outline Dimensions



Pump size	Flange dimensions to JIS 5K	Pressure rating 10 bar									Dimensions (mm)		
	Delivery flange					Suction flange							
	ND	OD	PCD	d	X	ND	OD	PCD	d	X	A	C	B
CGA 50	50	165	125	18	4	65	185	145	18	4	420	165	576
CGA 65	65	185	145	18	4	80	200	160	18	8	420	195	592
CGB 80	80	200	160	18	8	100	220	180	18	8	450	218	643
CGB 100	100	220	180	18	8	125	250	210	18	8	450	224	657
CGC 125	125	250	210	18	8	150	285	240	22.5	8	500	247	744
CGC 150	150	285	240	22.5	8	200	340	295	22.5	8	560	280	758
CGD 200	200	340	295	22.5	8	250	395	350	22.5	12	630	352	970
CGD 250	250	395	350	22.5	12	300	445	400	22.5	12	710	352	980
CGE 300	300	445	400	22.5	12	350	505	460	22.5	16	800	430	1257
CGE 350	350	505	460	22.5	16	400	565	515	25.0	16	900	540	1327