UNIVERSITY OF NAIROBI

DEPARTMENT OF MECHANICAL AND MANUFACTURING ENGINEERING

FINAL YEAR PROJECT REPORT

[FME 561 &562]

<u>PROJECT TITLE:</u>

PUMPING SYSTEM DESIGN FOR PETROLEUM INDUSTRY IN LINE WITH VISION 2030 (FROM MOMBASA TO NAIROBI)

PROJECT CODE: GMN 01/2012

SUPERVISOR: ENGINEER G.M. NYORI

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In partial fulfillment of the Degree of Bachelor of Science in Mechanical Engineering

May 2012

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DECLARATION

Except where we have stated and acknowledged, we certify that the research work, findings, discussions and conclusion set out in this project is our original effort and has not been presented before to the best of our knowledge.

MORRIS META MIRITI - F18/1861/2007

Signature………………………………….

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Supervisor: Eng. G.M. NYORI

Signature……………………………………………………

Date of submission…………………………………………

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DEDICATION

We dedicate this project to our parents, siblings, relatives and friends for the moral and material support they have always accorded us during the entire time of this project

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ACKNOWLEDGEMENTS

First, we thank the Almighty God for guiding and giving us power, calm mind and grace that has brought us this far.

During the preparation of this project, we came across many people who have been instrumental in helping us complete this project. We would like to extend our sincere gratitude to them and truly appreciate them for their untiring support.

However, in an elite category that demands mentioning, we recognize the following persons:

Our special thanks goes to our project supervisor, Eng. G.M. Nyori- we are very grateful for the support, guidance, censure, concerned assistance, tolerance and comradeship that he gave us from the beginning of the project to its successful completion.

We would like to thank The Chairman's office (Dept. of Mechanical &Manufacturing Engineering) for the earnest help they granted us that paved way to different industries and institutions.

In a special way, we thank our esteemed lecturers Eng G.O. Nyangasi and Eng.J. Mwaka for broadmindedness and comradeship as we needed their assistance

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We also appreciate our friends and classmates, for their tireless support and encouragements throughout the project.

We are grateful to all our family members and friends who supported us all the time to the completion of this project.

To all, we say thank you and God bless you.

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LIST OF SYMBOLS

H- Head

H_f - friction head

G - Gravitational acceleration

K- Internal pipe roughness

N - Rotational speed

n_s - Specific speed

Q - Flow rate

Re - Reynolds number

η - Efficiency

ν - Kinematic viscosity

ABBREVIATIONS

AGO = Automobile Gas Oil

API = American Petroleum Institute

ASME = American Society of Mechanical Engineers

ASTM = American Society for Testing and Materials

DPK = Dual Purpose Kerosene

MAOP = Maximum Allowable Operating Pressure

MSP = Motor Spirit Premium

MSR = Motor Spirit Regular

SCADA = Supervisory Control and Data Acquisition

TAPS = Trans Alaska Pipeline System

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LIST OF FIGURES

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ABSTRACT

Increasingly, entire industry sectors and individual companies are re-orienting their operation strategies to align with the demands of rapid globalization. Sustainable mechanisms are becoming an integral component of these strategies. Small, medium and large companies are focusing more than ever on efficiency and production quality in order to address the growing challenges of this globalization.

The objective of this study was to design an efficient petroleum pumping system (from Mombasa to Nairobi) that meets the projected demand by 2030 and addresses various challenges such as leakages, spillage, pipe burst and degradation of the pipeline due to corrosion. In order to achieve this, extensive research was done so as to evaluate the impact of various design parameters. More ever, an insight to both local and international pipelines was undertaken in order to lay grounds for a sound design.

Owing to the delicate nature of petroleum products, an elaborate design analysis was carried out taking into consideration how various parameters affect the design. This was geared towards realizing an optimal design that averts possibilities of catastrophes.

Various challenges encountered during research work were captured in the discussion and several conclusions were dawn based on the analysis made. In addition, crucial recommendations were put forward in order to exhaustively address the various challenges encountered.

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CHAPTER ONE

1.0 Background Information

Oil and gas pipelines are among the biggest infrastructure projects in developing countries in recent years. A pipeline comprises of steel pipes (the line) that are almost always buried, pumping stations to propel the liquid, and terminals at both ends for the product being carried. Pipelines carry products between refineries, oil depots and port facilities. Their length can vary from a few kilometers to hundreds or thousands of kilometers. Often they are interconnected to form a network that can be very dense, forming a sort of spider web. These networks connect different product entry points with different exit points, thereby cost-effectively diversifying sources of supply and providing reliable delivery to meet specified deadlines.

Petroleum products are crucial to a country's economy. Many sectors of our economy rely on them for effective operation. These includes; transport, power generation, agriculture, industrial sector and others. Since large quantities of petroleum is in great demand and its not possible to use gravity in transportation of the product from Mombasa to Nairobi (Mombasa at sea level while Nairobi at 1630m above sea level), therefore, this welcomes the use of pipeline transport, though initial costs of installation is high as compared to transportation by the rail or trucks. Economic feasibility shows that pumping system design is more cost effective than any other means. Advantages of using pipeline are: It is a safe and reliable mode of transport system, it is an economical and dependable mode of transport particularly to the sensitive and strategic areas, and it provides a long term infrastructural option, the difficulties in handling large volume of products by rail or road from one loading point is reduced, minimum transit loss.

Before a given design is arrived at, the surveys are carried out, mapping the regions and setting a sketch on how the pumping system will appear and the Economic and Technical feasibility tests are carried out to determine the viability of the project design. In this project we are aimed at meeting the projected demand by vision 2030 as per the current rating of increment of 2% per annum.

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Problem statement

To provide an efficient, reliable, safe and cost effective means of transporting the required amount (current demand and projected demand by 2030) of liquid petroleum products from Mombasa to Nairobi.

Objectives

To design a pumping system for petroleum products that can;

1) Meet the projected demand by 2030.
2) Overcome pumping challenges e.g. leakage, spillage, pipe burst and pipeline corrosion.

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CHAPTER TWO

LITERATURE REVIEW

2.1 Pipeline Network.

Steel pipe is used in most pipelines transporting petroleum products. It is manufactured according to ISO specifications**.** Pipelines may be small or large, up to 1.22m in diameter. Nearly the entire mainline pipe is buried, but other pipeline components such as pump stations are above ground. Some lines are as short as 1km, while others may extend to 1,500km or more. Some are very simple, connecting a single source to a single destination, while others are very complex, having many sources, destinations, and interconnections.

2.2 Fuels Transported.

The products carried in liquid pipelines include a wide range of materials. Petroleum transmission lines transport petroleum products from the refinery to deport. Refined petroleum product, including gasoline, aviation fuels, kerosene, diesel fuel, heating oil, and various fuel oils, are sizable portions of the pipelines business, whether produced in domestic refineries or imported to coastal terminal

2.2.1 Product Qualities

Critical physical properties of the materials being transported dictate the design and operating parameters of the pipeline. Specific gravity, compressibility, temperature, viscosity, pour point, and vapor pressure of the material are the primary considerations. These design parameters are discussed in the following sections in terms of their influence on pipeline design.

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a) Specific Gravity/Density

The density of a liquid is its weight per unit volume. Density is usually denoted as kilograms of material per cubic metre. The specific gravity of a liquid is typically denoted as the density of a liquid divided by the density of water at a standard temperature (15.56^{o}C). By definition, the specific gravity of water is 1.00. Typical specific gravities for the distilled petroleum products gasoline, turbine fuel, and diesel fuel are 0.73, 0.81, and 0.84, respectively (PBS&J chemical properties for gasoline,diesel,and turbine fuel).

b) Compressibility

Petroleum products transported by pipeline are only slightly compressible. Thus, application of pressure has little effect on the material's density or the volume it occupies at a given temperature. Consequently, compressibility is of only minor importance in liquid product pipeline design. Liquids at a given temperature occupy the same volume regardless of pressure as long as the pressure being applied is always above the liquid's vapor pressure at that temperature.

c) Temperature

Pipeline capacity is affected by temperature both directly and indirectly. In general, as liquids are compressed, for example, as they pass through a pump, they will experience slight temperature increases. Most liquids will increase in volume as the temperature increases, provided the pressure remains constant. Thus, the operating temperature of a pipeline will affect its throughput capacity. Lowering temperatures can also affect throughput capacity, as well as overall system efficiency. In general, as the temperature of a liquid is lowered, its viscosity increases, creating more frictional drag along the inner pipe walls, requiring greater amounts of energy to be expended for a given throughput volume.

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d) Viscosity

From the perspective of the pipeline design engineer, viscosity is best understood as the material's resistance to flow. It is measured in centistokes. One centistokes (cSt) is equivalent to 1.0×10^{-6} square metres per second. Resistance to flow increases as the centistoke value (and viscosity) increases. Overcoming viscosity requires energy that must be accounted for in pump design, since the viscosity determines the total amount of energy the pump must provide to put, or keep, the liquid in motion at the desired flow rate. Viscosity affects not only pump selection, but also pump station spacing.

Typical viscosities for gasoline, turbine fuel, and diesel fuels are 0.64, 7.9, and 5 to 6 cSt, respectively. As the material's viscosity increases, so does its frictional drag against the inner walls of the pipe. To overcome this, drag-reducing agents are added to some materials. Such drag-reducing agents are large molecular weight (mostly synthetic) polymers that will not react with the commodity or interfere with its ultimate function. They are typically introduced at pump stations in very small concentrations and easily recovered once the commodity reaches it final destination. However, often, no efforts are made to separate and remove these agents.
Drag reduction can also be accomplished by mixing the viscous commodity with diluents. Common diluents include materials recovered from crude oil fractionation such as raw naphtha.

e) Pour Point

The pour point of a liquid is the temperature at which it ceases to pour. Once temperatures of materials fall below their respective pour points, conventional pipeline design and operation will no longer be effective; however, some options still exist for keeping the pipeline functional. These include:

a) Heating the materials and/or insulating the pipe to keep the materials above their pour point temperature until they reach their destination.

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b) Introduce lightweight hydrocarbons that are miscible with the material, thereby diluting the material and lowering both its effective viscosity and pour point temperature.

c) Introduce water that will preferentially move to the inner walls of the pipe, serving to reduce the effective coefficient of drag exhibited by the viscous petroleum product.

d) Mix water with the petroleum material to form an emulsion that will exhibit an effective lower viscosity and pour point temperature.

2.2.2 Vapor Pressure

The vapor pressure of a liquid represents the liquid's tendency to evaporate into its gaseous phase with temperature. Virtually all liquids exhibit a vapor pressure, which typically increases with temperature. Vapor pressures of petroleum liquids are determined using a standardized testing procedure and are represented as the Reid vapor pressure. Reid vapor pressures are critical to liquid petroleum pipeline design, since the pipeline must maintain pressures greater than the Reid vapor pressure of the material in order to keep the material in a liquid state. Pipelines handling such fuels must constantly monitor their vapor pressure and adjust operating conditions accordingly. Pipelines carrying liquids with high vapor pressures can be designed to operate under a variety of flow regimes.

Single-phase flow regimes intend for the entire amount of the material in the pipeline to be in the liquid state. Operators of single-phase liquid pipelines attempt to control pressure and flow to maintain a "full face" of liquids in the pipeline, minimizing the amount of volatilization that is allowed to occur. This maximizes system efficiency and also the longevity of system components. Failure to maintain a full face of liquids in a single-phase liquid pipeline can result in increased risks of fires and explosions. Single-phase liquid pipelines are the most common designs for petroleum liquids. However, pipelines can also be designed as two-phase systems in which both vapor and liquid phases of the material are expected to be present.

Typical vapor pressures for gasoline, turbine fuel, and diesel fuel are 0.1035, 0.0138, and 0.0138 Mpa, respectively

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2.2.3 Pumping sequence

This is the way product is pumped through the multi-product pipeline. The following sequence is adopted in a way that it minimizes product contamination:

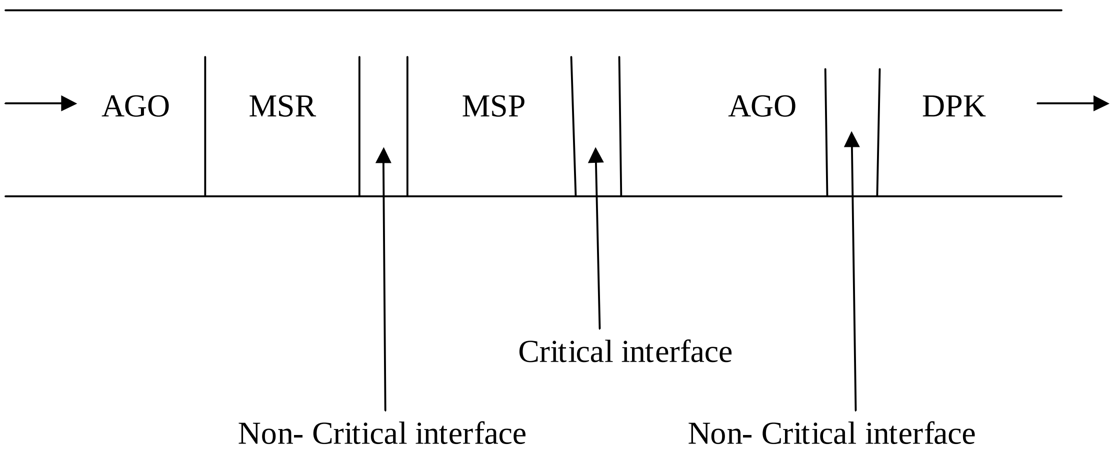

Fig. 2.1 Pumping sequence (source author 2012)

2.2.3.1 Interface monitoring

An interface is a region where two different grades of petroleum come into contact with each other and mixing takes place in a multi product pipeline. This is divided further into two:

a) **Critical interface**

This is a region where a spirits come into contact with distillates. E.g. MSP/AGO or MSR/AGO. Slopping is done in such interfaces

b) **Non- Critical Interface**

This is a region where spirits come into contact with each other or distillates with each other. In such interface, direct cut off is normally done.

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It is important to note that during pumping sequence DPK **MUST NOT** come in contact with MSP or MSR.

2.3 Pumps

A pump is a device used to move fluids such as liquids or slurries. A pump displaces a volume by physical or mechanical action. Pumps can be categorized according to their method of moving fluid i.e.

1) Direct lift
2) Displacement
3) Gravity

However, pumps mostly used in Pipeline transport are of the "Positive displacement" type or of the rotor dynamic type.

2.3.1 Positive Displacement Pumps

These can further be categorized into two;

a) Rotary type
b) Reciprocating type

a) Rotary Type:

Positive displacement rotary pumps are pumps that move fluid using the principles of rotation. The vacuum created by the rotation of the pump captures and draws in the liquid.

Rotary pumps are very efficient because they naturally remove air from the lines, eliminating the need to bleed the air from the lines manually. Positive displacement rotary pumps also have their weaknesses. Because of the nature of the pump, the clearance between the rotating pump and the outer edge must be very close, requiring that the pumps rotate at a slow, steady speed. If rotary pumps are operated at high speeds, the fluids will cause erosion, much as ocean waves polish stones or erode rock into sand.

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Rotary pumps that experience such erosion eventually show signs of enlarged clearances, which allow liquid to slip through and detract from the efficiency of the pump.

Positive displacement rotary pumps can be grouped into three main types:

i. Gear pumps are the simplest type of rotary pumps, consisting of two gears laid out side-by-side with their teeth enmeshed. The gears turn away from each other, creating a current that traps fluid between the teeth on the gears and the outer casing, eventually releasing the fluid on the discharge side of the pump as the teeth mesh and go around again.

ii. Screw pumps are a more complicated type of rotary pumps, featuring two screws with opposing thread, that is, one screw turns clockwise, and the other counterclockwise.

iii. Moving vane pumps are the third type of rotary pumps, consisting of a cylindrical rotor encased in a similarly shaped housing. As the rotor turns, the vanes trap fluid between the rotor and the casing, drawing the fluid through the pump.

b) Reciprocating-type pumps

Reciprocating pumps are those which cause the fluid to move using one or more oscillating pistons, plungers or membranes (diaphragms). Reciprocating-type pumps require a system of suction and discharge valves to ensure that the fluid moves in a positive direction. Pumps in this category range from having one cylinder; to in some cases four cylinders or more. Most reciprocating-type pumps are "duplex" (two) or "triplex" (three) cylinder.

Furthermore, they can be either "single acting" independent suction and discharge strokes or "double acting" suction and discharge in both directions.

The pumps can be powered by air, steam or through a belt drive from an engine or motor. Reciprocating pumps are typically used for pumping highly viscous fluids including concrete and heavy oils and special applications demanding low flow rates against high resistance (Rainbow,R.A, 2004).

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2.3.2 Rotor dynamic pumps:

This is a kinetic machine in which energy is continuously imparted to the pumped fluid by means of a rotating impeller, propeller or rotor, in contrast to the positive displacement pump in which fluid is moved by trapping a fixed amount of fluid and forcing the trapped volume into the pumps discharge. A good example of a rotor dynamic pump is the centrifugal pump.

a) Centrifugal pump;

A centrifugal pump is a rotor dynamic pump that uses a rotating impeller to increase the pressure and flow rate of a fluid. Centrifugal pumps are the most common type of pump used to move liquids through a piping system. The fluid enters the pump impeller along or near to the rotating axis and is accelerated by the impeller, flowing radially outward or axially into a diffuser or volute chamber, from where it exits into the downstream piping system. Centrifugal pumps are typically used for large discharge through smaller heads.

Centrifugal pumps are most often associated with the radial flow type. However, the term "centrifugal pump" can be used to describe all impeller type rotor dynamic pumps including the radial, axial and mixed flow variations.

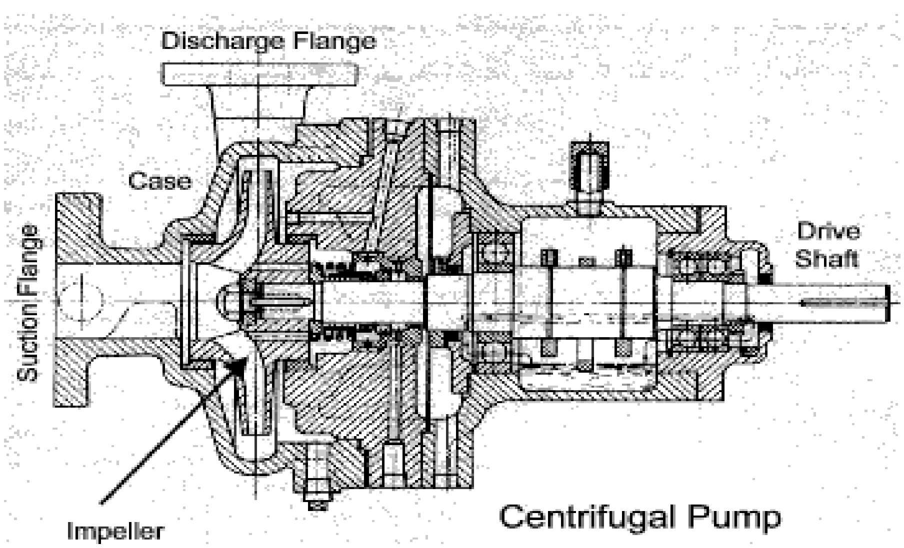

Fig.2.2: A Centrifugal Pump.(source engeneering tool box)

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2.3.4 Basic Pump Operating Characteristics

"**Head"** is a term commonly used with pumps. Head refers to the height of a vertical column of a fluid. Pressure and head are interchangeable concepts since they are related to each other i.e.

$$P = \rho g H$$

Where:

p = pressure

ρ = density of fluid

H = head of the fluid column

The total head of the pump is composed of several types of head that help define the pump's operating characteristics.

a) Total Dynamic Head

The total dynamic head of a pump is the sum of the total static head, the pressure head, the friction head, and the velocity head.

$$H = H_s + H_p + H_f + H_v$$

Where:

H_s = total static head

H_p = pressure head

H_f = friction head .

H_v = velocity head

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b) Total static head

The total static head is the total vertical distance the pump must lift the fluid. It is referenced to the fluid surface.

c) Friction Head

Friction head is the energy loss or pressure decrease due to friction when a fluid flows through the pipe networks. The velocity of the fluid has a significant effect on friction loss. Loss of head due friction occurs when a fluid flows through straight pipe sections, fittings, valves, around corners, and where pipe increases or decreases in size. Values for these losses can be calculated or obtained from friction loss tables. The friction head loss for a piping system is the sum of all the friction losses.

d) Velocity head

Velocity head is the energy of fluid due to its velocity. This is a very small amount of energy and is usually neglected in analysis.

c) Suction head

The suction head includes not only the vertical suction lift, but also the friction losses through the pipe, elbows, foot valves and other fittings on the suction side of the pump. There is an allowable limit to the suction head on a pump and the net positive suction head (NPSH) sets that limit. Operating a pump with a suction head greater than it was designed for, or under conditions with excessive vacuum at some point in the impeller, may cause cavitation. Cavitation is the implosion of bubbles of air and fluid vapor and makes a very distinct noise like gravel in the pump. The implosion of numerous bubbles will eat away at an impeller and it will eventually be filled with holes.

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2.3.5 Pump Arrangement

Pumps can be arranged in serial or parallel to provide an additional head or flow rate capacity.

a) Pumps in Serial

When two or more pumps are arranged in serial their resulting pump performance curve is obtained by adding their heads at the same flow rate as indicated in the figure below.

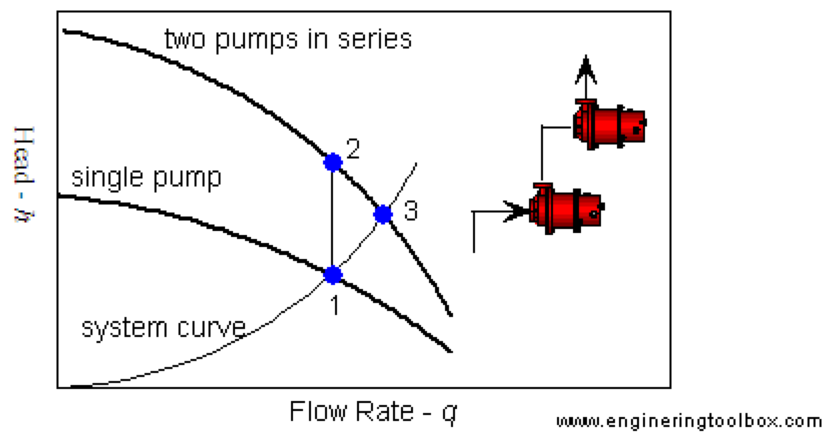

Fig.2.3 Pumps in Serial arrangement**.**

Centrifugal pumps in series are used to overcome larger system head loss than one pump can handle alone. For two identical pumps in series the head will be twice the head of a single pump at the same flow rate, as indicated in **point 2.** With a constant flow rate the combined head moves from 1 to 2. In practice the combined head and flow rate moves along the system curve to point 3.**Point 3** is where the system operates with both pumps running and. **Point 1** is where the system operates with one pump running.

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Series operation of single stage pumps is seldom encountered; more often multistage centrifugal pumps are used.

b) Pumps in Parallel

When two or more pumps are arranged in parallel their resulting performance curve is obtained by adding their flow rates at the same head as indicated in the figure below.

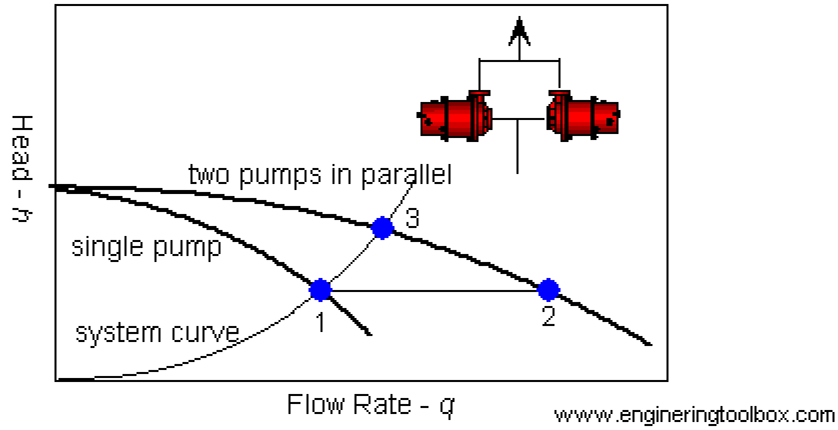

Fig.2.4 Pumps in Parallel

Centrifugal pumps in parallel are used to overcome larger volume flows than one pump can handle alone. For two identical pumps in parallel, and the head is kept constant, the flow rate doubles as indicated with **point 2** compared to a single pump. In practice the combined head and volume flow moves along the system curve as indicated from 1 to 3. **Point 3** is where the system operates with both pumps running. **Point 1** is where the system operates with one pump running. In practice, if one of the pumps in parallel or series stops, the operation point moves along the system resistance curve from point 3 to point 1, thus the head and flow rate are decreased.

2.3.6 Pump Power Requirements

The power added to fluid as it flows through a pump can be calculated using the following formula;

$$P_f = \rho g Q H$$

Where:

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ρ = density of fluid

g = gravitation acceleration

P_f = Power imparted on the fluid by the pump

Q = Flow Rate in cubic metres per second (m^3/s)

H = Total Dynamic Head (metres)

However, the actual power required to run a pump will be higher than this because pumps and drives are not 100% efficient. The power required at the pump shaft to pump a specific flow rate against a specific H is the brake power (P_b) which is calculated with the following formula:

$$P_b = \frac{P_f}{\eta_{pump} \ x \ \eta_{drive}}$$

Where:

P_b= Brake Power (continuous power rating of the power unit).

η_{pump} = Efficiency of the pump (usually read from the pump curve and having a value between 0 and 1).

η_{drive} = efficiency of the drive unit between the power source and the pump. For direct connection this value is 1, for right angle drives is 0.95 and for belt drives it can vary from 0.7 to 0.85.

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2.4 Valves

A valve is a device that regulates the flow of a fluid (gases, liquids, fluidized solids, or slurries) by opening, closing, or partially obstructing various passageways. Valves are technically pipe fittings, but are usually discussed as a separate category. In an open valve, fluid flows in a direction from higher pressure to lower pressure. Valves commonly used are:

1. Ball valves
2. Gate valves
3. Globe valves
4. Swing check valves
5. Twin seal valves

The above types of valves can further be categorized according to their functionality such as:

a) Non-return valves
b) Pressure control valves
c) Surge relieve valves
d) Thermal relieve valves

2.4.1 None Return Valves:

These types of valves are designed to restrict the back flow of product to one way only. They are mostly found on the inlets of the tanks.

a) Ball Valve:

Ball valve is a rotational motion valve that uses a ball-shaped disk to stop or start fluid flow. When the valve handle is turned to open the valve, the ball rotates to a point where the hole through the ball is in line with the valve body inlet and outlet. When the valve is shut, the ball is rotated so that the hole is perpendicular to the flow openings of the valve body and the flow is stopped. Most ball valve actuators are of the quick-acting type, which require a 90° turn of the valve handle to operate the valve. Other ball valve actuators are planetary gear-operated.

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This type of gearing allows the use of a relatively small hand wheel and operating force to operate a fairly large valve.

Some ball valves have been developed with a spherical surface coated plug that is off to one side in the open position and rotates into the flow passage until it blocks the flow path completely. Seating is accomplished by the eccentric movement of the plug. The valve requires no lubrication and can be used for throttling service.

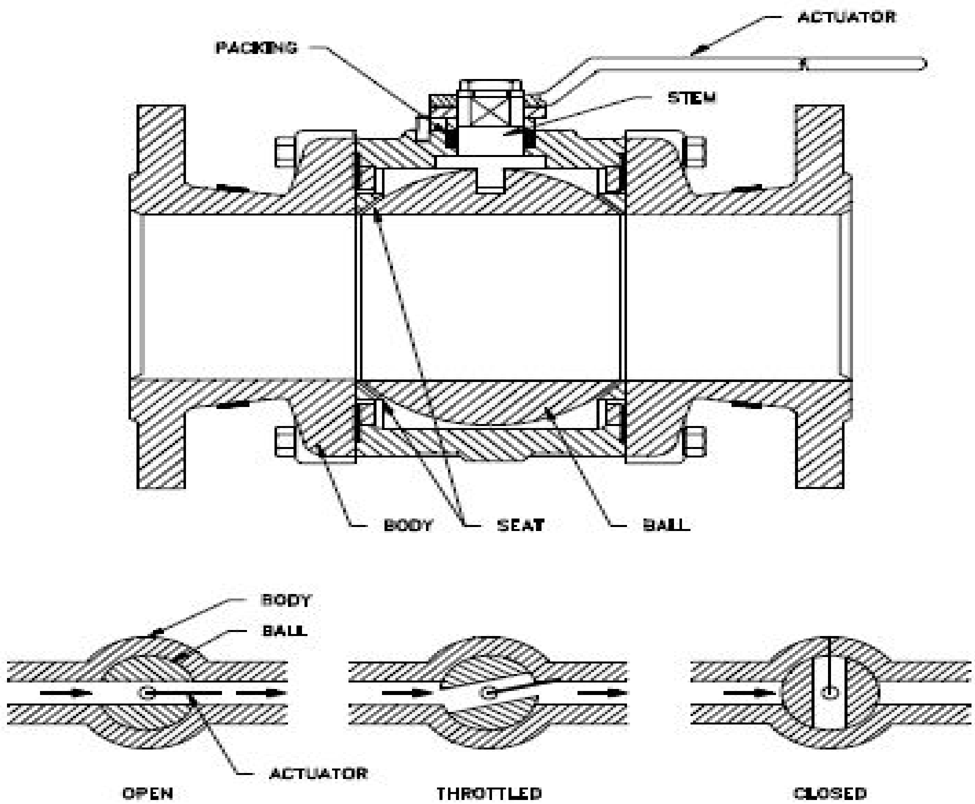

Fig. 2.5 Ball valve (www.source engeneering tool box)

The type of ball valve used for non-return purposes is usually a spring loaded ball that allows product to flow one way but the spring retracts when the fluid tries to flow back the other direction.

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b) Swing Check Valves

A swing check valve allows full, unobstructed flow and automatically closes as pressure decreases. These valves are fully closed when the flow reaches zero and prevent back flow. Turbulence and pressure drop within the valve are very low.

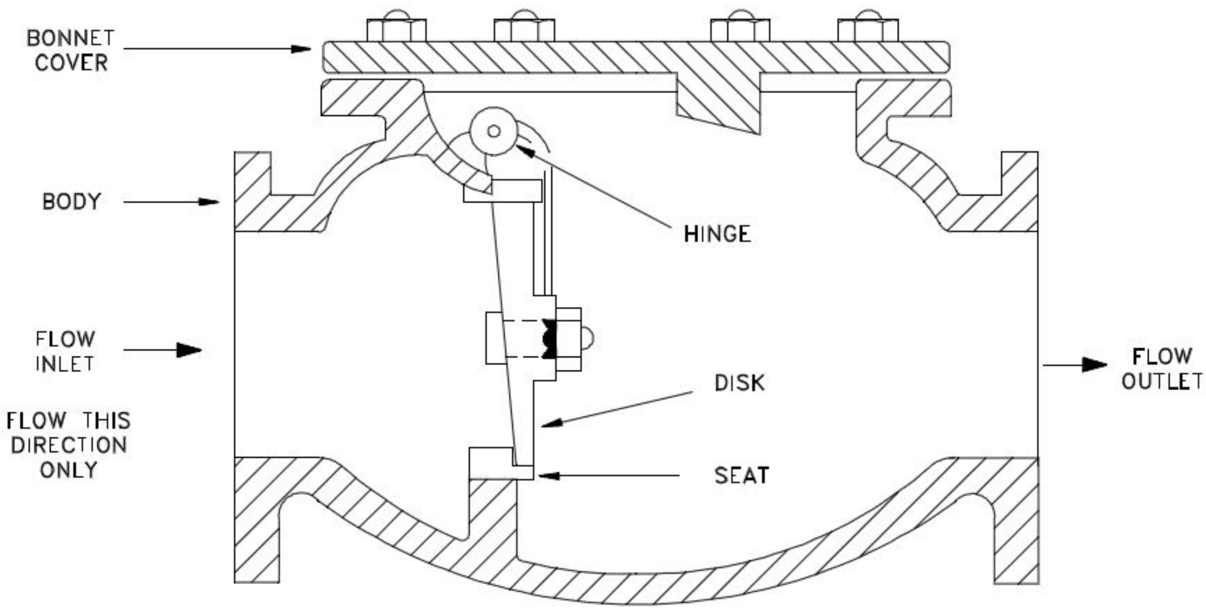

Fig.2.6 Swing Check Valves (www.source engeneering tool box)

A swing check valve is normally recommended for use in systems employing gate valves because of the low pressure drop across the valve. Swing check valves are available in either Y-pattern or straight body design. In either style, the disk and hinge are suspended from the body by means of a hinge pin. Seating is either metal-to metal or metal seat to composition disk. Composition disks are usually recommended for services where dirt or other particles may be present in the fluid, where noise is objectionable, or where positive shutoff is required.

Straight body swing check valves contain a disk that is hinged at the top. The disk seals against the seat, which is integral with the body.

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This type of check valve usually has replaceable seat rings. The seating surface is placed at a slight angle to permit easier opening at lower pressures, more positive sealing, and less shock when closing under higher pressures.

Swing check valves are usually installed in conjunction with gate valves because they provide relatively free flow. They are recommended for lines having low velocity flow and should not be used on lines with pulsating flow when the continual flapping or pounding would be destructive to the seating elements. This condition can be partially corrected by using an external lever and weight.

2.4.2 Pressure Control Valves

These types of valves are mostly gate valves that have and I&C mechanism. Their purpose is to balance the pressure between the main line and the bypass. Principle of operation is simple; when there is a pressure build up in the main line, the controller can relieve some of this pressure by opening the valve so as to redirect part of the flow to another part of the main line.

2.4.3 Surge Relief Valves.

These types of valves resemble oversized spool pieces and are placed on the main line. Their purpose is to relive some of the pressure if it happens to build up in the main line. Inside the valve is a rubber diaphragm which is pressurized on the inside surface of the valve by use of nitrogen set at a specific pressure. If this set pressure is exceeded, then the diaphragm will give in and the product will flow thought to the slope tanks. Service on this type of valve includes checking of the rubber diaphragm that it is not torn and the whole assembly is not passing product through.

2.4.4 Thermal Relief Valve

This type of valve is also known as an expansion valve/ safety valve. Relief and safety valves prevent equipment damage by relieving accidental over-pressurization of fluid systems. The main difference between a relief valve and a safety valve is the extent of opening at the set point pressure. A relief valve gradually opens as the inlet pressure increases above the set point. A relief valve opens only as necessary to relieve the over-pressure condition.

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A safety valve rapidly pops fully open as soon as the pressure setting is reached. A safety valve will stay fully open until the pressure drops below a reset pressure. The reset pressure is lower than the actuating pressure set point. The difference between the actuating pressure set point and the pressure at which the safety valve resets is called *blow down*. Blow down is expressed as a percentage of the actuating pressure set point.

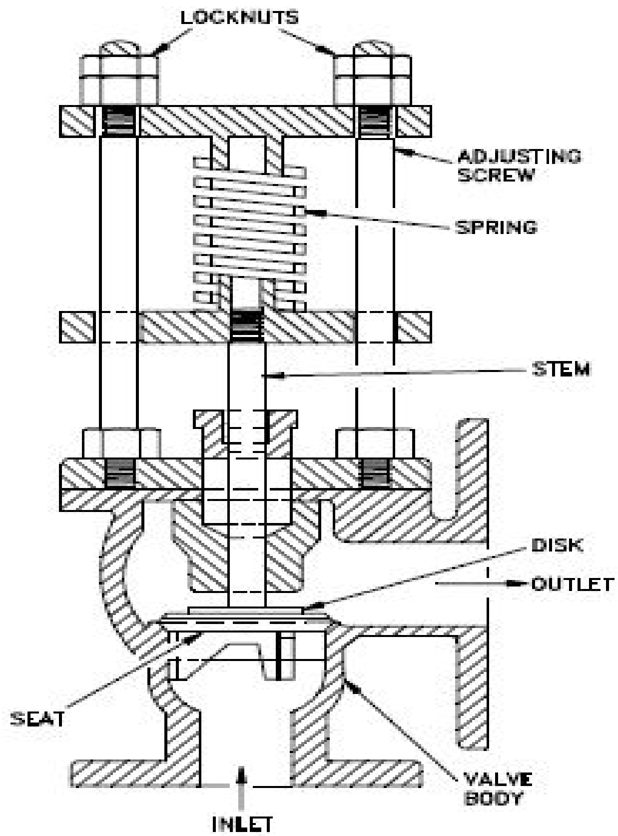

Fig. 2.7 Thermal Relief Valve(www.source engeneering tool box)

Relief valves are typically used for incompressible fluids such as liquid petroleum products. Safety valves are typically used for compressible fluids such as steam or other gases. Safety valves can often be distinguished by the presence of an external lever at the top of the valve body, which is used as an operational check.

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2.4.5 Gate Valves

A *gate* valve is a linear motion valve used to start or stop fluid flow; however, it does not regulate or throttle flow. The name gate is derived from the appearance of the disk in the flow stream. The disk of a gate valve is completely removed from the flow stream when the valve is fully open. This characteristic offers virtually no resistance to flow when the valve is open.

Hence, there is little pressure drop across an open gate valve. When the valve is fully closed, a disk-to-seal ring contact surface exists for 360°, and good sealing is provided. With the proper mating of a disk to the seal ring, very little or no leakage occurs across the disk when the gate valve is closed.

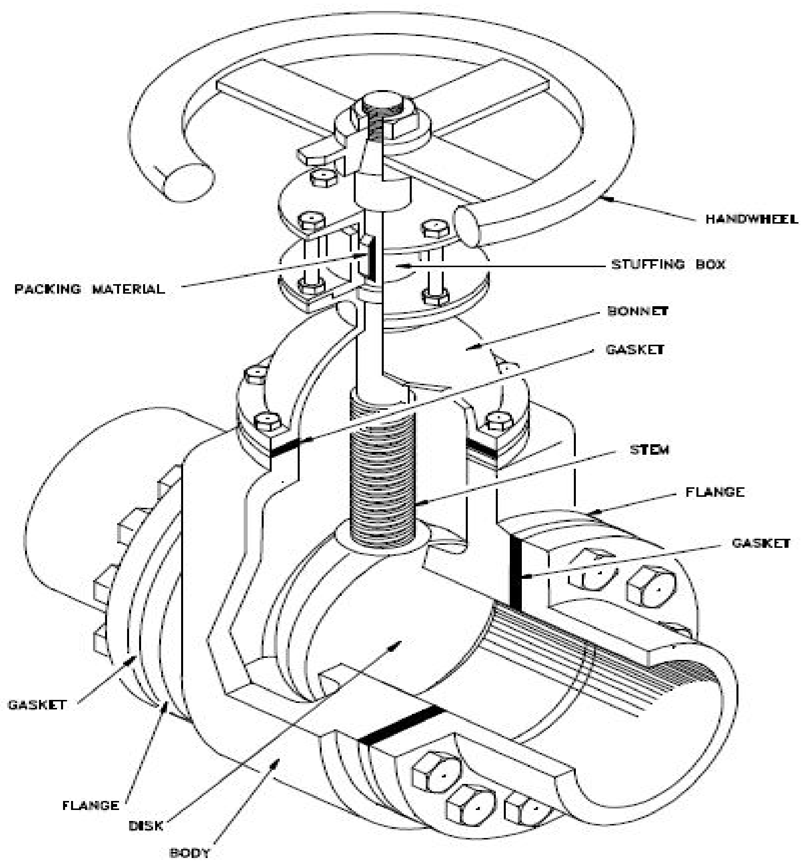

Fig. 2.8 Gate Valve

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2.4.6 Valve Components

a) Gaskets

The seals or packings used to prevent the escape of a gas or fluids from valves.

b) Bonnet

A bonnet acts as a cover on the valve body. It is commonly semi-permanently screwed into the valve body. During manufacture of the valve, the internal parts are put into the body and then the bonnet is attached to hold everything together inside.

c) **Ports**

Ports are passages that allow fluid to pass through the valve. Ports are obstructed by the valve member or disc to control flow. Valves most commonly have 2 ports, but may have as many as 20. The valve is almost always connected at its ports to pipes or other components. Connection methods include threading, compression fittings, glue, cement, flanges, or welding.

d) Disc

A disc or valve member is a movable obstruction inside the stationary body that adjustably restricts flow through the valve.

e) Seat

The seat is the interior surface of the body which contacts the disc to form a leak-tight seal. In discs that move linearly or swing on a hinge the disc comes into contact with the seat only when the valve is shut. In disks that rotate, the seat is always in contact with the disk, but the area of contact changes as the disc is turned. The seat always remains stationary relative to the body.

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f) **Stem**

The stem transmits motion from controlling device to the disc. The stem typically passes through the bonnet when present. In some cases, the stem and the disc can be combined in one piece, or the stem and the handle are combined in one piece.

2.5 Strainers.

A plant which operates 24 hours a day, seven days a week cannot afford any downtime. A blocked control valve would stop production leading to costly repairs, reduced profits and low level of customer service.

Benefits of strainers;

a) Safeguards plant, providing peace of mind
b) Reduces maintenance cost and downtime
c) Wide range of materials and end connections
d) Plant and control valve life is increased

The strainers remove debris from pipeline by directing the flow through a screen. There are different mesh options available which allows optimum selection for each specific application and type of fluid. All strainer screens have large screening and free area designed to remove as much dirt as possible without clogging up.

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CHAPTER THREE

RESEARCH AND DESIGN METHODOLOGY

3.0 Introduction

Petroleum pipelines are delicate infrastructure due to the nature of the products conveyed through them. Many pipelines in the world experience leakages, pipe burst and spillage in various points. These occurrences lead to severe fires due to flammability of petroleum products.

To design a pipeline, one must carefully examine all the parameters to avert any eventualities. In order to come up with an effective and efficient design, we first evaluated some petroleum pumping system design both local and international.

3.1 Kenya Pipeline Company.

Kenya Pipeline Company (KPC) was established on 6th September 1976 under the companies act (cap 486) and started commercial operation in 1978.The core business is to transport, store and distribute white petroleum oils in the country. The company is 100% owned by the government and complies with the provision of state corporation act (cap 446) of 1986. The company's operations are also governed by the relevant legislation like the finance act, the public procurement regulations among others. The main objective of setting up the company was to provide an efficient, reliable, safe and cost effective means of transporting petroleum products up country from Mombasa.

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3.1.1 Pumping Stations

Mombasa – Nairobi Pipeline has 8 Pumping Stations (PS) located at Mombasa (PS-1), Samburu (PS-2), Maungu (PS-3), Manyani (PS-4), Mutitu Andei (PS-5), Makindu (PS-6), Sultan Hamud (PS-7) and Konza (PS-8).

3.1.2 Current Installed Parameters

Flow Rate, Q_o = 880 M^3/hour

Pipe diameter, d_i = 0.3556m

Pipe length, L = 450 kms

3.1.3 Station Altitude

Pumping station 1 (PS 1) - Sea Level

Nairobi Terminal (PS 10) - 1630 Metres

3.1.4 Products Transported Include;

1) Motor Spirit Premium (MSP)
2) Motor Spirit Regular (MSR)
3) Automobile Gas Oil (AGO)-low sulphur distillate
4) Automobile Gas Oil (AGO)-high sulphur distillate
5) Illuminating Kerosene (IK)-Dual Purpose Kerosene (DPK) distillate
6) Aviation Turbine Oil (Jet A-1 fuel)-DPK distillate.

Source (http://www.kpc.go.ke)

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3.2 Alyeska Pipeline Service Company

Alyeska Pipeline Service Company was incorporated on August 14, 1970 to design, build, operate and maintain the pipeline, pump stations and Valdez Marine Terminal. Alyeska also helps assure safe tanker operations in Prince William Sound with its Ship Escort/Response Vessel System. The Trans Alaska Pipeline System (TAPS) transports crude oil from Alaska's North Slope, across 1287.5km of tundra, rugged mountains and rivers to Valdez, North America's northernmost ice-free port. Engineers designed the pipeline to endure and protect Alaska's harsh environment as it traverses three mountain ranges, three major earthquake faults and more than 500 rivers and streams. The pipeline corridor includes more than 550 crossing areas for caribou, moose and other wildlife.

3.2.1 Installed Parameters

a) Pressure

Design, maximum —8.136MPa

Operating, maximum — 8.136MPa

b) Pumping Stations

Original design had 12 pumping stations with 4 pumps each.

c) Leak detection systems

Leak detection systems detect and locate oil spills. These include; Pressure deviation, flow rate deviation, flow rate balance and line volume balance.

Source (http://www.alyeska-pipe.com)

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3.3 Design Methodology

3.3.1 Pipeline Design

A pipeline comprises of steel pipes (the line) that are almost always buried, pumping stations to propel the liquid, and terminals at both ends for the product being carried. Pipelines carry products between refineries, oil depots and port facilities. Their length can vary from a few kilometers to hundreds or thousands of kilometers. Often they are interconnected to form a network that can be very dense, forming a sort of spider web.

3.3.2 Factors Influencing Pipeline Design

The major steps in pipeline system design involve establishment of critical pipeline performance objectives and critical engineering design parameters such as:

a) Required throughput (volume per unit time for most petroleum products and metres per unit time for petrochemical feedstock);

b) Origin and destination points;

c) Product properties such as viscosity and specific gravity;

d) Topography of pipeline route;

e) Maximum allowable operating pressure (MAOP) and hydraulic calculations are done to determine;

i) Pipeline diameter, wall thickness, and required yield strengths;

ii) Number of, and distance between, pump stations; and

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iii) Pump station horsepower required.

3.3.3 Selection of Pipe Size

The dimensions of a pipeline, both the sizes and capacities of the various components, as well as the conditions under which the pipeline system operates dictate them system's capacity. Larger diameter pipes allow for higher mass flows of materials provided other components of the pipeline system, primarily pumps and pressure management devices, are properly sized and positioned. In general, the longer the segment of mainline pipes between pump stations, the greater the drop in line pressure. However, grade changes and the viscosity of the materials being transported can also have major influences on line pressures. Greater wall thicknesses are selected for high-pressure applications or when the pipe segment might be subjected to unusual external forces such as seismic activities and landslides. Also, in hard-to-reach places, such as beneath transportation routes and at river crossings or difficult-to-access environmentally sensitive areas, overbuilding in size or quality is sometimes chosen to accommodate future expansion requirements.

3.3.4 Determination of Pressure

Operating pressure of a pipeline is determined by the design flow rate vapor pressure of the liquid, the distance the material has to be transferred, and the size of line that carries the liquid. Pipe operating pressure and pump capabilities and cost typically drive decisions on line size, the number of pump stations, and the like. Grades notwithstanding, line pressure follows a saw tooth curve between pump stations. The maximum and minimum line pressure that can be tolerated, together with the physical properties of the materials noted earlier, dictate the spacing of the pump stations and the motive horsepower of the pumps.

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3.3.5 Movements at Pipe Bends

When unusual internal or external forces are applied to the pipe, it is most likely to respond to such forces by moving at the apex of side bends, sag bends, and over bends. Forces that can cause pipe movement can include a net outward force generated by internal pressure, thermal expansions or contractions of the pipe due to temperature extremes, hydraulic pressures from groundwater, or seismic activity in the vicinity of the pipe. Natural resistances to such forces include the axial stiffness of the pipe itself, the bearing and shear resistance of backfill and overburden materials, and the extent to which those materials were compacted during construction

3.3.6 Fluid Transients

Rapid changes in the flow rates of liquid or two-phase piping systems can cause pressure transients that generate pressure pulses and transient forces in the piping system. The magnitudes of these pressure pulses and force transients are often difficult to predict and quantify. As a result of water hammer, an unbalanced impulsive force called a “thrust” load is applied successively along each straight segment of a buried pipe. This causes a pressure imbalance between consecutive bends. Such hammering actions, if sufficiently strong and continued over long periods of time, can compromise the pipe’s integrity or introduce fatigue stress cracks.

3.3.7 Selection of Valves

Valves are installed at various locations along the mainline for various operational controls and to isolate segments of the pipeline for maintenance or replacement or to limit the amount of product in jeopardy of spilling in the event of a pipeline break. Typically, valves are installed at either side of sensitive or potentially problematic segments such as water body crossings. Such check valves can serve to quickly and efficiently isolate those segments of the pipeline in the event a problem should occur. Such isolation limits the scale of the adverse consequences that could occur in the event of a pipeline rupture in those segments. Check valves are placed at each significant change in grade to prevent backflow of product in the event of a failure of the upstream pumps.

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Bypasses may need to be installed around mainline valves or damaged pipeline segments to facilitate maintenance, repair, or replacement without shutting down operation of the pipeline. Bypasses typically consist of the requisite length of substitute pipe, each end of which is attached to a manually operated valve and two "hot taps" (devices that cut into the pipeline and divert the flow from the mainline pipe segment to be isolated to the bypass pipe). Once the bypass is positioned, the bypass valves are opened and the hot taps are operated to tap into the existing pipe. Such bypasses are typically removed (and the hot taps repaired) once the task is completed.

3.3.8 Valve Spacing and Rapid Shutdown

The valves spacing of and other devices capable of isolating any given segment of a pipeline are driven by two principal concerns:

a) Maintaining the design operating conditions of the pipeline with respect to throughput and flexibility and

b) Facilitate maintenance or repairs without undue disruption to pipeline operation and rapid shutdown of pipeline operations during upset or abnormal conditions.

Valve spacing and placement along the mainline are often selected with the intention of limiting the maximum amount of material in jeopardy of release during upset conditions or to isolate areas of critical environmental concern to the greatest extent

3.4 Pump Selection

3.4.1 Pump Designs

Pumps of various designs are used in petroleum product pipelines. Selection of pump design is based on desired efficiency as well as the physical properties of the materials being moved, especially viscosity and specific gravity.

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The pump's head pressure, or the pressure differential it can attain, is critical for selecting pumps that are capable of moving fluids over elevation changes. Two fundamental pump designs are in common use are centrifugal pumps and positive displacement pumps.

Centrifugal pumps are preferred for moving large volumes of material at moderate pressure, while positive displacement pumps are selected for moving small volumes of material at higher line pressures. Centrifugal pumps consist of two main components: the impeller and the volute. The impeller, the only rotating component of the pump, converts the energy it receives from the force that causes its rotation into kinetic energy in the fluid being pumped, while the volute converts the kinetic energy of the fluid into pressure.

Positive displacement pumps can be of various designs; however, two designs predominate in pipeline applications i.e. reciprocating and rotating pumps. Rotating pumps are often the pump design of choice for viscous fluids such as petroleum fuels. Unlike a centrifugal pump where power demands rise sharply with increasing fluid viscosity, the performance of rotating pumps is generally unaffected by variations in either fluid viscosity or line pressure.

3.4.2 Determining Location of Pumps and Pumping Stations

Desired material throughput values as well as circumstantial factors along the pipeline route are considered in designing and locating pump stations. Desired operating pressures and grade changes dictate individual pump sizes and acceptable pressure drops (i.e., the minimum line pressure that can be tolerated) along the mainline; grade changes also dictate the placements of the pump stations. Pump stations are often fully automated, but can also be designed to be manned and to include ancillary functions such as serving as pig launching or recovery facilities or serving as the base from which inspections of mainline pipe are conducted. Because there are multitudes of ways in which the desired operating conditions can be obtained and sustained, the outfitting and location of pump stations are also often influenced by economics, typically representing a compromise between few large-capacity pump stations and a greater number of smaller-capacity stations.

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The overall length of the pipeline (to its terminal destination) and the flexibility needed to add or remove materials along the course of the pipeline also dictate pump station placement. At a minimum, pump stations include pumps (components that actually contact the fluids in the pipeline and provide kinetic energy) and prime movers (power sources that provide power, typically some form of mechanical energy) to the pumps.

To facilitate maintenance and to prevent disruptions of pipeline operation as a result of equipment failure, most pump stations use several pumps arranged in parallel fashion.

Typically, all but one of the pumps is capable of producing the desired operating pressures and throughputs, so some pump is constantly off-line and in standby. Pump stations also represent locations where ownership or custody of the material is transferred. For the sake of accountability, such pump stations are also equipped with flow monitoring devices. Pump stations typically also have collocated facilities that support pipeline operation or facilitate shutdowns or maintenance on pipeline segments. Thus, breakout tanks for temporary storage of materials or for use in managing line pressures and controlling product surges are also present at pump stations. Finally, pump stations are, in some instances, collocated with terminal or breakout tankage facilities.

Although certain pump designs are preferred for certain applications, all pumps require regular maintenance and are subject to failure from a variety of factors. Pump maintenance, therefore, is critical to continued safe performance of pipeline systems

3.4.3 Driver Selection

The component that actually provides power to the pump is referred to as the prime mover. A wide variety of prime movers are in use, including electric motors, gas turbines, and diesel internal combustion engines. In recent years, most long-distance transmission pipelines have begun using electric motors or gas turbines. Virtually any prime-mover pump design combination is possible, with decisions resting primarily on the physical properties of the fluids being pumped, the desired throughputs, operating pressures, and transport speeds for the pipeline and for logistical needs such as meeting operating parameters, availability of power or fuel for the prime mover, and compatibility with SCADA systems in use and the sensors they rely on.

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Initial costs and maintenance demands can also influence selection. In terms of initial costs, electric motors are far less expensive than any other option. When maintenance costs are considered, times between major overhauls of prime movers vary, with electric motors and industrial turbines expected to require the fewest overhauls over time.

3.4.4 Pigging Devices and Pig Launching/Receiving Facilities

Pipeline pigs come in a wide variety of sizes and designs. Pigs are inserted into the pipeline while it is operational and are carried along by the fluid being pumped. Because they are solid devices constructed of various materials including metal, plastics, and rubber derivatives, pigs must be removed before reaching the next pump. Typically, pig traps, launchers, and recovery facilities are collocated with pump stations. Pigs are designed to perform a wide array of functions. Their basic purpose is threefold:

a) Provide a way to clean debris and scale from the inside of the pipe,
b) Inspect or monitor the condition of the pipe,
c) Act as a plug or seal to separate products in multi-product commercial pipelines or to isolate a segment for repair without depressurizing the remainder of the pipeline.

Pigs designed to clean the pipe can use mechanical means (often called scraper pigs) or chemicals. Pigs that monitor the condition of the pipe are categorized as in-line inspection tools. Monitoring pigs, also sometimes called “instrument pigs” or “smart pigs,” can perform a wide variety of functions. Geometry pigs check for deformation of the pipe (which can greatly influence throughput efficiencies, but can also be an early indicator of significant problems that could compromise pipeline integrity).

Pipeline curvature, temperature and pressure profiles, bend measurements, corrosion detection, crack detection, leak detection, and product sampling represent some of the other major functions performed by smart pigs. Magnetic flux leakage and ultrasonic technologies are employed for some of these inspections.

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Another type of pig recently developed is the gel pig. As the name implies, gel pigs consist of a series of gelled liquids that are introduced for a variety of purposes, including serving as a separator between products in a multi-product pipeline, collecting debris (especially after initial construction or repairs that involved opening the pipeline and dewatering the pipeline

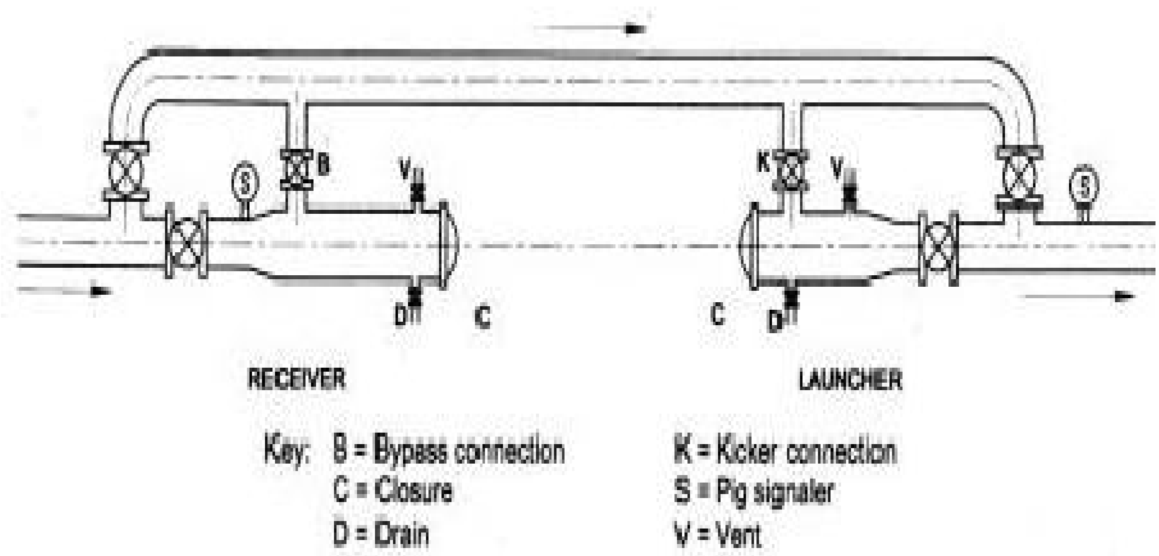

Fig. 3.1: Typical configuration of a pig launching/recovery facility.(source pipeline intergraty summary)

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Fig. 3.2 provides examples of the various types of pigs in use today.(source pipeline intergraty summary)

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3.5 Safety

Safety in pipeline design and construction is achieved by the proper design and application of the appropriate codes and system hardware components. Design codes as set forth in Pipeline Safety regulations provide appropriate safety factors and quality control issues during construction. Metering stations and SCADA systems provide continuous monitoring oversight of pipeline operations.

3.5.1 Measurement and Flow Control

a) Supervisory Control and Data Acquisition (SCADA)

A typical SCADA system collects data from, and supervises control of, third-party programmable logic controllers at each of the pipeline's pumping stations, mainline valves, and other areas where monitoring of critical conditions takes place. Along the entire length of the pipeline, block valves are remotely monitored and controlled using advanced real-time SCADA processors designed to support complex remote applications. The communications for the system is typically over the Ethernet and fiber optic lines as the backbone, backed up by public switched telephone networks. The output signals include;

i. *Pipeline mimic/displays*. The complete pipeline can be mimicked to provide the operator with instantaneous visual feedback on the status of any portion of the pipeline, including pumps, valves, tanks, etc. These visual schematics include overviews of the entire pipeline system or systems and drill-down screens that take the viewer to an individual location or piece of equipment.

ii. *Pump, compressor, and other equipment status*. Equipment operation can be displayed with status (on/off) and other critical parameters associated with a piece of equipment such as flow, discharge pressure, vibration, case temperature, etc.

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iii. *Valve status*. Valve information can be displayed with valve positions (open/throttle/closed) depicted.

iv. *Alarms and alerts*. Alarms and other operational indications are immediately available for operator response where complete system status is known and, in many cases, can be displayed. These can alert the controller to an unusual or abnormal operating situation or remind the controller about upcoming operating changes that need to be initiated. Often, system configurations allow the operator to intervene to validate the alarm or to take the necessary corrective actions.

3.5.2 Chemical additive

Drag-reducing agents or other chemicals to improve the flow characteristics of the pipeline are always present and something to consider in any analysis of pipeline systems. Furthermore, dependent on the product, there may be other chemicals that affect the properties of the fluids in transmission systems, such as the static reducer added to diesels. For high-viscosity crude oils (i.e., low API gravity numbers), a diluent is often added to enhance the pumpability of the petroleum products and reduce the frictional drag on the inside pipeline walls, thereby reducing the amount of energy needed to pump the crude. Such diluents are typically low-viscosity petroleum refining fractions such as raw naphtha.

3.5.3 Remedial Action for Corrosion Deficiencies

Companies must initiate remedial action as necessary to correct deficiencies observed during corrosion monitoring. Industry standards for allowable advancements of internal and external corrosion establish the action levels for such remedial activities. Remedial action may involve replacing the sacrificial anodes of a corrosion control system, a relatively minor construction activity compared to wholesale replacement of compromised pipe involving long-term shutdown of operations and major construction efforts equivalent to initial installation. To limit operational downtime, temporary bypass segments isolating the damaged segments may also be installed.

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a) Cathodic Corrosion Protection

Underground corrosion of steel pipelines can result from the flow of electrical current between areas of different electric potential.

The area of higher potential (the anode) will be corroded, and the area of lower potential (the cathode) will not be subject to corrosion. In the case of a buried pipeline, the soil can act as an electrolyte, facilitating the transfer of electrons from a metallic object such as a buried pipe (the anode in this electrochemical engine) and the ground. Areas of different electrical potential exist throughout the trace of a pipeline, with the magnitudes of such electrical potential differences depending on soil types and myriad other local conditions.

In a cathodic protection system, anodes made of materials that are more conducive to electrical current delivery than the steel of the pipeline are electrically bonded to the pipe and installed along the pipeline route, resulting in the subsequent flow of electrical current from the anodes to the ground. As electrons flow from these anodes, the anodes deteriorate, earning them the common name of “sacrificial anodes” since they are being sacrificed and allowed to corrode instead of the pipe. The pipeline becomes the cathode of the system, and its corrosion is prevented as long as some anode material remains. Keifner and Vieth (1990) established that the magnitude of the corrosion currents for a given potential difference between two electrodes (Cathode and anode) depends on several factors:

i. *Soil resistivity*. This is determined by temperature, moisture content, and the concentration of ionized salts present. Generally, corrosion is high in low resistivity soils and can be low in very high resistivity soils.

ii. *Chemical constituents of the soil*. Corrosion can be low in very high resistivity soils.

iii. *Separation between anode and cathode*. Corrosion is more likely to occur when the anode and cathode are close together.

iv. *Anode and cathode polarization*. Protective films formed at the anode and cathode affect corrosion intensity.

v. *Relative surface areas of cathode and anode*. For a given magnitude of corrosion current, the depth of corrosion on the anode will be inversely proportional to anode area.

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Since power requirements for cathodic protection systems are relatively low, the application fits the capabilities of solar energy systems.

b) **Corrosion Inhibitor**

This is a chemical compound that is normally added in the product line with the product so that it can help in reducing the corrosion rate in the pipeline. A wide variety of chemicals can be used for this purpose although many are toxic, and must be used with care to protect the environment.

The corrosion inhibitor is added to the product line and flows with the product providing protection to the pipe. Commonly the corrosion inhibitor forms a thin film which prevents reactions between compounds in the product and in the pipe. The corrosion inhibitor is normally blended into the fluid continuously, or added periodically to maintain a protective film.

3.5.4 Leak Detection

Methods used to detect product leaks along a pipeline can be divided into two categories, externally based (direct) or internally based (inferential). Externally based methods detect leaking product outside the pipeline, as well as technologies like hydrocarbon sensing via fiber optic or dielectric cables. Internally based methods, also known as computational pipeline monitoring, use instruments to monitor internal pipeline parameters (i.e., pressure, flow, temperature, etc.), which are inputs for inferring a product release by manual or electronic computation.

The method of leak detection selected for a pipeline depends on a variety of factors including pipeline characteristics, product characteristics, instrumentation and communications capabilities, and economics. Pipeline systems vary widely in their physical characteristics and operational functions, and no one external or internal method is universally applicable or possesses all of the features and functionality required for perfect leak detection performance.

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3.5.6 Risk of Natural Hazards and Human Threats

Natural and man-made hazards, such as shaking from earthquakes, flooding, and even human chemical, biological, or nuclear attacks can cause harm to pipeline components, such as pump stations, pipelines segments, and storage tanks, etc.

3.5.7 Scour at Stream Crossings and Suspended Rock Crossings

Pipelines buried beneath or adjacent to rivers can be compromised over time by the erosive force of the moving water. Scouring can occur that would displace the cover materials and expose the pipe, subjecting it to additional lateral forces and possibly even causing sufficient displacement to break the pipe. High velocities of water in rocky areas or watercourses with steep banks have the highest scouring potential. Areas prone to flooding can also experience excessive water flow velocities during those periods that can also result in scouring action. The typical response to traversing rivers or drainage ways with high scouring potential is to bury the pipe at greater depths or to suspend the pipe above these areas. In addition, major river crossings are required to be inspected every 5 years for indications of scour and/or exposed pipe.

3.5.8 Fire Hazards

The proximity of pipelines carrying volatile materials raises the potential of a fire in one transmission system causing heating, stress, and/or rupture in another one.

The normal distance between buried pipelines will probably prevent significant transfer of heat underground. However, care should be taken at locations where the pipelines exit the ground, such as at pump and compressor stations and road/river crossings, to ensure adequate separation of lines and facilities or additional insulation and fire protection.

In the event of a fire in a petroleum pipeline, overhead electricity transmission lines could be damaged if adequate distance is not maintained.

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CHAPTER FOUR

DESIGN CRITERIA

4.1 Design Analysis

4.1.1 Demand

The projected demand by 2030 can be estimated by applying an annual increment of 2 **%** to the current flow rate.

Thus the estimated demand by 2030, Q is calculated as follows;

$$Q = Q_O (1+R)^n$$

Where:

Q_O = current flow rate

R = rate of increase per year in percentage

n = number of years

Q = projected demand by 2030.

$$Q = 880 (1+0.02)^{18}$$

$$Q = 1256.86 \text{ m}^3/\text{hr}$$

Projected demand in m^3/s is;

$$Q = \frac{1256.86}{3600} \text{ m}^3/\text{s}$$

$$Q = 3.49 \text{X} 10^{-1} \text{ m}^3/\text{s}$$

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4.1.2 Preliminary Pipe Size Selection

The main factors influencing pipe size selection are

1 flow rate

2 stress in the pipes

4.1.3 Pipe Diameter Estimation Considering the Projected Flow Rate

$$Q = VA$$

$$Q = \frac{V\pi {d_i}^2}{4}$$

$$d_i = \sqrt{\frac{4Q}{\pi V}}$$

Where:

V = flow velocity

A = cross sectional area of the pipe

d_i = internal diameter of the pipe

Applying a guideline velocity of 2m/s

$$d_i = \sqrt{\frac{4x0.349}{2\pi}}$$

$$d_i = 0.4714m$$

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4.1.4 Available Pipe Material and Standard

ISO Standard **ISO 3183:2007(E)**

Steel Grade L245 or A

Minimum test pressure, P_t = 5.4 MP

Yield strength of pipe material σ_y = 210Mpa

Internal roughness of the pipes, k = 0.1

Outside diameter, (d_o) = 508 mm

Specified wall thickness, (t) = 9.5mm

4.1.5 Calculation of internal pipe diameter (d_i)

$$d_i = d_o - 2t$$

$$d_i = (508 - 2x9.5) \text{ mm}$$

$$d_i = 489 \text{ mm}$$

4.1.6 Calculation of Design Velocity, V_d

$$V_d = \frac{Q}{A}$$

$$V_d = \frac{4Q}{\pi d_i^2}$$

$$V_d = \frac{4 \ x \ 0.349}{\pi x 0.489^2}$$

$$V_d = 1.8583 \text{m/s}$$

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4.2 Case 1

4.2.1 Calculation of Reynolds Number For Gasoline, *Re*

$$Re = \frac{V_d \ x \, d_i}{\nu}$$

$$Re = \frac{1.8583x0.489}{0.64x10^{-6}}$$

$$Re = 1.42\text{x}10^6$$

Relative roughness, k/d_i = $\frac{0.1}{489}$ = 0.0002

Coefficient of friction, λ = f (*Re*, k/d_i)

From the moody diagram in *appendix A*

$$\lambda = 1.45 \ x \ 10^{-2}$$

Head due to gasoline friction (Darcy equation)

$$H_f = \lambda \frac{LV^2}{2gd_i}$$

$$H_f = \frac{1.45 \ x \ 10^{-2} x \ 450 \ x 10^3 \ x \ 1.8583^2}{2 \ x \ 9.81 \ x \ 0.489}$$

$$H_f = 2.349 \ x \ 10^3$$

Neglecting minor losses in the pipeline system, the total dynamic head due to gasoline is given by:

$$\text{H} = H_s \ + \ H_f$$

H = 1630 + 2349 = 3979m

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Pressure to be generated by the pumping systems to overcome this head is;

$$P = \rho gH$$

Where,

p = total pressure to be generated by the pumping system

ρ = density of Gasoline

H =total dynamic head to be overcome by pumps

P = 730 x 9.81 x 3979 = 28.5Mpa

4.3 Case 2

4.3.1 Calculation of Reynolds Number For Turbine Fuel, *Re*

$$Re = \frac{V_{d\ x\ d_i}}{\nu}$$

$$Re = \frac{1.8583x0.489}{7.9x10^{-6}} = 1.15\text{x}10^5$$

$$Re = 1.15\text{x}10^5$$

Relative roughness, $\text{k}/d_i = \frac{0.1}{489} = 0.0002$

Coefficient of friction, $\lambda = \text{f}\ (Re,\ \text{k}/d_i)$

From the moody diagram in *appendix A*

$$\lambda = 1.85x10^{-2}$$

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Head due to Turbine Fuel friction (Darcy equation)

$$H_f = \lambda \frac{LV^2}{2gd_i}$$

$$H_f = \frac{1.85x10^{-2} x\ 450\ x10^3\ x\ 1.8583^2}{2\ x\ 9.81\ x\ 0.489}$$

$$H_f = 2.996\ x\ 10^3$$

Neglecting minor losses in the pipeline system, the total dynamic head due to diesel is given by:

$$H = H_s\ +\ H_f$$

$$H = 1630 + 2996 = 4626\text{m}$$

Pressure to be generated by the pumping systems to overcome this head is;

$$P = \rho g H$$

Where,

p = total pressure to be generated by the pumping system

ρ = density of Turbine Fuel

H =total dynamic head to be overcome by pumps

P = 810 x 9.81 x 4626 = 36.76Mpa

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4.4 Case 3

4.4.1 Calculation of Reynolds Number For Diesel, *Re*

$$Re = \frac{V_{d\,x\,d_i}}{\nu}$$

$$Re = \frac{1.8583x0.489}{6.0\ x\ 10^{-6}} = 1.515 \text{ x } 10^5$$

$$Re = 1.515 \text{ x } 10^5$$

Relative roughness, $k/d_i = \frac{0.1}{489} = 0.0002$

Coefficient of friction, λ = f (Re, k/d_i)

From the moody diagram in *appendix A*

$$\lambda = 1.8x10^{-2}$$

Head due to diesel friction (Darcy equation)

$$H_f = \lambda\frac{LV^2}{2gd_i}$$

$$H_f = \frac{1.8x10^{-2}x\ 450\ x10^3\ x\ 1.8583^2}{2\ x\ 9.81\ x\ 0.489}$$

$$H_f = 2.915\ x\ 10^3$$

Neglecting minor losses in the pipeline system, total dynamic head due to diesel is given by:

$$H = H_s + H_f$$

$$H = 1630 + 2915 = 4545\text{m}$$

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Pressure to be generated by the pumping systems to overcome this head is;

$$P = \rho g H$$

Where;

p = total pressure to be generated by the pumping system

ρ = density of Diesel

H =total dynamic head to be overcome by pumps

P = 840 x 9.81 x 4545 = 37.45Mpa

4.5 Governing Design

Total pressure due to gasoline (Case 1) forms the lower limit of design while that due to diesel (case 3) form the upper limit.

For optimal design, the design should be based on the upper limit. Thus the total pressure to be overcome by the pumping system assuming that the entire pipeline is transporting diesel is;

P = 37.45Mpa

4.5.1 Design of Pumping Station

The total pressure to be generated by the pumping system is;

P = 37.45Mpa

Since the minimum test pressure (P_t) for the pipe standard selected is;

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P_t = 5.4Mpa

Number of pumping stations (N_p) is given by;

$$N_p = \frac{P}{P_t}$$

$$N_p = \frac{37.45}{5.4} \approx 6.935$$

Considering the delicate nature of the products conveyed through the pipeline, its appropriate to take 8 pumping stations.

$$N_p = 8$$

Designing pumping stations (PS) for constant static head static, static head to be overcome by each pump(H_{sp}) is calculated as;

$$H_{sp} = \frac{1630}{8} = 203.75\text{m}$$

PS I – PS 2

Static head, H_s= 203.75m

Distance = 65Km

$$H_f = \lambda \frac{LV^2}{2gd_i}$$

$$H_f = \frac{1.8x10^{-2} x\ 65\ x10^3\ x\ 1.8583^2}{2\ x\ 9.81\ x\ 0.489}$$

H_f =421.06m

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Total head to be overcome by pumping station 1 (PS 1)

$$H = H_s + H_f$$

$$H_1 = 203.75 + 421.06 = 624.81\text{m}$$

PS 2 – PS 3

Static head, H_s = 203.75m

Distance = 60Km

$$H_f = \lambda \frac{LV^2}{2gd_i}$$

$$H_f = \frac{1.8x10^{-2} x\ 60\ x10^3\ x\ 1.8583^2}{2\ x\ 9.81\ x\ 0.489}$$

$$H_f = 388.67m$$

Total head to be overcome by pumping station 2 (PS 2)

$$H = H_s + H_f$$

$$H_2 = 203.75 + 388.67 = 592.42\text{m}$$

PS 3 – PS 4

Static head, H_s= 203.75m

Distance = 58Km

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$$H_f = \lambda \frac{LV^2}{2gd_i}$$

$$H_f = \frac{1.8x10^{-2} x\ 58\ x10^3\ x\ 1.8583^2}{2\ x\ 9.81\ x\ 0.489}$$

$$H_f = 375.71m$$

Total head to be overcome by pumping station 3 (PS 3)

$$H = H_s + H_f$$

$$H_3 = 203.75 + 375.71 = 579.46\text{m}$$

PS 4 –PS 5

Static head, H_s= 203.75m

Distance = 56Km

$$H_f = \lambda \frac{LV^2}{2gd_i}$$

$$H_f = \frac{1.8x10^{-2} x\ 56\ x10^3\ x\ 1.8583^2}{2\ x\ 9.81\ x\ 0.489}$$

$$H_f = 362.76m$$

Total head to be overcome by pumping station 4 (PS 4)

$$H = H_s + H_f$$

$$H_4 = 203.75 + 388.67 = 566.51\text{m}$$

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PS 5 –PS 6

Static head, H_s = 203.75m

Distance = 59Km

$$H_f = \lambda \frac{LV^2}{2gd_i}$$

$$H_f = \frac{1.8x10^{-2} x\ 59\ x10^3\ x\ 1.8583^2}{2\ x\ 9.81\ x\ 0.489}$$

$$H_f = 382.19m$$

Total head to be overcome by pumping station 5 (PS 5)

$$H = H_s + H_f$$

H_5 = 203.75 + 382.19 = 585.94m

PS 6 –PS 7

Static head, H_s = 203.75m

Distance = 53Km

$$H_f = \lambda \frac{LV^2}{2gd_i}$$

$$H_f = \frac{1.8x10^{-2} x\ 53\ x10^3\ x\ 1.8583^2}{2\ x\ 9.81\ x\ 0.489}$$

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$H_f = 343.32m$

Total head to be overcome by pumping station 6 (PS 6)

$H = H_s + H_f$

$H_6 = 203.75 + 343.32 = 547.07$m

PS 8 –Deport

Static head, $H_s =$ 203.75m

Distance = 49Km

$$H_f = \lambda \frac{LV^2}{2gd_i}$$

$$H_f = \frac{1.8x10^{-2} x\ 49\ x10^3\ x\ 1.8583^2}{2\ x\ 9.81\ x\ 0.489}$$

$H_f = 317.41m$

Total head to be overcome by pumping station 1 (PS 1)

$H = H_s + H_f$

$H_8 =$ 203.75 + 317.41 = 521.16m

4.6 Pump Selection

Taking the pumping station with highest head (PS1) as the upper limit of our design;

H=624.81m

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Converting head of diesel above into head of water

$$p_d = \rho_d g H_d$$

$$p_w = \rho_w g H_w$$

But;

$$p_w = p_d$$

Therefore;

$$H_w = H_d \frac{\rho_d}{\rho_w}$$

$$H_w = 624.81 \; x \; \frac{0.84}{1}$$

$$H_w = 524.84\text{m}$$

Where:

H_d = Head due to diesel

H_w = equivalent head of water

p_d = pressure due to diesel column

p_w = pressure due to water column

ρ_d = desity of diesel

ρ_w = desity of water

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Using 4-stage centrifugal pump to overcome this head;

Head to be overcome by one stage (H_{stage}) assuming series connection is;

$$H_{stage} = \frac{524.84}{4}\text{ m}$$

$$H_{stage} = 131.21\text{m.}$$

Thus the specific speed (n_s) of the pump is calculated as;

$$n_s = \frac{N\sqrt{Q}}{H_p^{0.78}} = \text{f (d/D)}$$

Where:

N= rotational speed of the pump

Q= flow rate at peak efficiencies

H_p= total head generated at peak efficiency

d= inlet diameter of the pump impeller

Using a four stage centrifugal pump with a speed of 2900r.p.m as indicated in appendix D;

$$n_s = \frac{N\sqrt{Q}}{H_p^{0.78}}$$

$$n_s = \frac{2900\sqrt{0.349}}{131.21^{0.78}}$$

$$n_s = 38.18$$

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The power added to fluid as it flows through a pump can be calculated with the following formula;

$$P_f = \rho_w \text{g} \text{Q} H_w$$

$$P_f = 1000 \text{ x } 9.81 \text{ x } 0.349 \text{ x } 524.84$$

$$P_f = 1796.89\text{kw}$$

Using η_{drive}= 1.0 for direct connection as indicated in the theory and η_{pump}= 0.75 as indicated in appendix F;

$$P_b = \frac{P_f}{\eta_{pump} \ x \ \eta_{drive}}$$

$$P_b = \frac{1796.89}{1.0 \ x \ 0.75}$$

$$P_b = 2395.85kw$$

Where;

P_f = Power imparted on the fluid by the pump

P_b= Brake Power (continuous power rating of the power unit).

4.7 Stresses in Pipes

The stress in pipe due to internal fluid pressure is determined by Lame's equation. According to Lame's equation, tangential stresses at any radius r,

$$\sigma_t = \frac{P_p r_i^2}{r_o^2 - r_i^2} [1 + \frac{r_o^2}{r^2}] \ldots\ldots\ldots\ldots\ldots\ldots \text{(i)}$$

And radial stresses at any radius r,

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$$\sigma_r = \frac{P_p r_i^2}{r_o^2 - r_i^2}\left[1 - \frac{r_o^2}{r^2}\right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(ii)}$$

Where:

P_p = internal fluid pressure in the pipe

r_i =inner radius of the pipe

r_o =outer radius of the pipe

The tangential stress is maximum at the inner surface (when $r=r_i$) of the pipe and minimum at the outer surface (when $r=r_o$) of the pipe.

Substituting the values of $r=r_i$ and $r=r_o$ in equation (i), we find that the maximum tangential stress at the inner surface of the pipe,

$$\sigma_{t(max)} = \frac{P_p[\,r_o^2 + r_i^2]}{[r_o^2 - r_i^2]}$$

$$\sigma_{t(max)} = \frac{5.15\ X\ 10^6[0.508^2 + 0.489^2]}{[0.508^2 - 0.489^2]}$$

$$\sigma_{t(max)} = 135.17\text{Mpa}$$

And minimum tangential stress at the outer surface of the pipe,

$$\sigma_{t(min)} = \frac{2P_p r_i^2}{[r_o^2 - r_i^2]}$$

$$\sigma_{t(min)} = \frac{2\ X\ 5.15\ X\ 10^6\ X\ 0.489^2}{[0.508^2 - 0.489^2]}$$

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$$\sigma_{t(min)} = 130.02\text{Mpa}$$

The radial stress is maximum at the inner surface of the pipe and zero at the outer surface of the pipe. Substituting the values of $r=r_i$ and $r=r_o$ in equation (ii), we find that the maximum radial stress at the inner surface of the pipe,

$$\sigma_{r\,max} = -p \text{ (compressive)}$$

$$\sigma_{\mathrm{r\,max}} = -\ 5.15\text{Mpa}$$

And minimum radial stress at the outer surface of the pipe,

$$\sigma_{rmin} = 0$$

Based on the maximum stress in the pipe i.e.σ_t, a factor of safety is evaluated

$$s.f = \frac{\sigma_y}{\sigma_{t(\max)}}$$

$$s.f = \frac{210}{135.17}$$

$$s.f = 1.55$$

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CHAPTER FIVE

5.1 DISCUSSION

The projected demand by 2030 was estimated to be 1256.86 m^3/hr. Pipe size to meet the demand was then selected from the ISO Standard **ISO 3183:2007(E)** Steel Grade L245 or A with the following parameters; Minimum test pressure, P_t = 5.4 MPa, Internal roughness of the pipes, k = 0.1**,** Outside diameter, d_o = 508 mm**,** Specified wall thickness, *t* = 9.5mm. Based on these parameters, internal pipe diameter d_i was calculated and thereafter the friction head was calculated, taking into consideration the different nature of products conveyed i.e. gasoline, jet fuel kerosene and diesel. The parameters for diesel yielded the highest value of friction head (H_f = 2915 m), hence, the governing design was based on it. Static head from PS 1 located in Mombasa to PS 2 located in Nairobi is 1630m. Thus the total dynamic head, H to be overcome by the pumps was found to be 4545m. For optimal design, 8 pumping stations were chosen based on the pressure limits of the pipe standard. The maximum total head in metres of water to be overcome by each pump was found to be 524.84m. Since this was the maximum head in all the pumping stations, PS 1 was selected as the governing station.

Using discharge vs. head table for a pump of speed, N = 2900rpm, a four stage centrifugal pump was selected. The specific speed of the pump selected was found to be 38.18. The head to be overcome by each stage of the pump was 131.21m. The power to be imparted on the fluid in order to overcome the total dynamic head was 1796.89kw.Since there are losses in the pump, the driver should produce a total power of 2395.85kw.

Stresses due to internal fluid pressure were determined using Lames Equation. This was based on the upper limit (PS I). The maximum value of stresses was the circumferential stress, σ_t with the value given below.

$$\sigma_{t(max)} = 135.17\text{Mpa}$$

The yield value(σ_y) of the pipe material was 210Mpa; hence a factor of safety of 1.55 was employed.

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However we encountered difficulties in obtaining substantial information from Kenya Pipeline Company due to security reasons. in addition, most of the pumps in Kenyan market are low pressure pumps hence characteristic curves of high pressure pumps are not readily available.

5.2 CONCLUSION

An elaborate design that meets projected demand by 2030 was arrived at. The projected demand by 2030 demand ($Q = 1256.86\ m^3/hr$) was arrived at by applying an annual increment of 2% to the current demand. The design maximum operating pressure was 5.15MPa. This pressure was contrasted with the yield stress of the pipe material (210MPa) yielding a safety factor of 1.55. This averts various pumping challenges e.g. leakage, spillage, and pipe burst. Pipeline corrosion was curbed by application of cathodic protection, coating and use of anti-rust agents.

5.3 SUGGESTIONS AND RECOMMENDATIONS

i. All valves should be automated to ensure maximum safety due to automatic response.
ii. Modern methods for leak detection e.g. use of laser beams should be employed.

iii. Accurate pigs should be used to ascertain the pipe size to avoid introduction of eddies in the pipeline

iv. Booster pumps should be installed at various locations along pipeline to maintain the fluid pressure in the pipeline

v. We suggest that another group of students to undertake the same project and examine the following areas; route survey and feasible pump location, leak detection along the pipeline, design of pipe bends, automation of the valves and calibration of the tanks to ensure high safety measures.

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ISO 3183:2007(E)

APPENDIX A: Pipe Grades, Steel Grades

PSL	Delivery condition	Pipe grade/steel grade [a,b]
PSL 1	As-rolled, normalizing rolled, normalized or normalizing formed	L175 or A25
		L175P or A25P
		L210 or A
	As-rolled, normalizing rolled, thermomechanical rolled, thermomechanical formed, normalizing formed, normalized, normalized and tempered; or, if agreed, quenched and tempered for SMLS pipe only	L245 or B
	As-rolled, normalizing rolled, thermomechanical rolled, thermomechanical formed, normalizing formed, normalized, normalized and tempered or quenched and tempered	L290 or X42
		L320 or X46
		L360 or X52
		L390 or X56
		L415 or X60
		L450 or X65
		L485 or X70

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ISO 3183:2007(E)

APPENDIX B: TEST PRESSURE FOR THREADED PIPE (K=0.1)

Specified outside diameter D mm (in)	Specified wall thickness t mm (in)	Test pressure MPa (psi) minimum			
		Grade			
		L175 or A25	L175P or A25P	L210 or A	L245 or B
10,3 (0.405)	1,7 (0.068)	4,8 (700)	4,8 (700)	4,8 (700)	4,8 (700)
13,7 (0.540)	2,2 (0.088)	4,8 (700)	4,8 (700)	4,8 (700)	4,8 (700)
17,1 (0.675)	2,3 (0.091)	4,8 (700)	4,8 (700)	4,8 (700)	4,8 (700)
21,3 (0.840)	2,8 (0.109)	4,8 (700)	4,8 (700)	4,8 (700)	4,8 (700)
26,7 (1.050)	2,9 (0.113)	4,8 (700)	4,8 (700)	4,8 (700)	4,8 (700)
33,4 (1.315)	3,4 (0.133)	4,8 (700)	4,8 (700)	4,8 (700)	4,8 (700)
42,2 (1.660)	3,6 (0.140)	6,9 (1 000)	6,9 (1 000)	6,9 (1 000)	6,9 (1 000)
48,3 (1.900)	3,7 (0.145)	6,9 (1 000)	6,9 (1 000)	6,9 (1 000)	6,9 (1 000)
60,3 (2.375)	3,9 (0.154)	6,9 (1 000)	6,9 (1 000)	6,9 (1 000)	6,9 (1 000)
73,0 (2.875)	5,2 (0.203)	6,9 (1 000)	6,9 (1 000)	6,9 (1 000)	6,9 (1 000)
88,9 (3.500)	5,5 (0.216)	6,9 (1 000)	6,9 (1 000)	6,9 (1 000)	6,9 (1 000)
101,6 (4.000)	5,7 (0.226)	8,3 (1 200)	8,3 (1 200)	8,3 (1 200)	9,0 (1 300)
114,3 (4.500)	6,0 (0.237)	8,3 (1 200)	8,3 (1 200)	8,3 (1 200)	9,0 (1 300)
141,3 (5.563)	6,6 (0.258)	8,3 (1 200)	8,3 (1 200)	8,3 (1 200)	9,0 (1 300)
168,3 (6.625)	7,1 (0.280)	a	a	8,3 (1 200)	9,0 (1 300)
219,1 (8.625)	7,0 (0.277)	a	a	7,9 (1 160)	9,2 (1 350)
219,1 (8.625)	8,2 (0.258)	a	a	9,3 (1 340)	10,8 (1 570)
273,1 (10.750)	7,1 (0.280)	a	a	6,5 (930)	7,5 (1 090)
273,1 (10.750)	7,8 (0.307)	a	a	7,1 (1 030)	8,3 (1 200)
273,1 (10.750)	9,3 (0.365)	a	a	8,5 (1 220)	9,8 (1 430)
323,9 (12.750)	8,4 (0.330)	a	a	6,4 (930)	7,5 (1 090)
323,9 (12.750)	9,5 (0.375)	a	a	7,3 (1 060)	8,5 (1 240)
355,6 (14.000)	9,5 (0.375)	a	a	6,6 (960)	7,7 (1 130)
406,4 (16.000)	9,5 (0.375)	a	a	5,8 (840)	6,8 (980)
457 (18.000)	9,5 (0.375)	a	a	5,2 (750)	6,0 (880)
508 (20.000)	9,5 (0.375)	a	a	4,6 (680)	5,4 (790)

[a] Not applicable.

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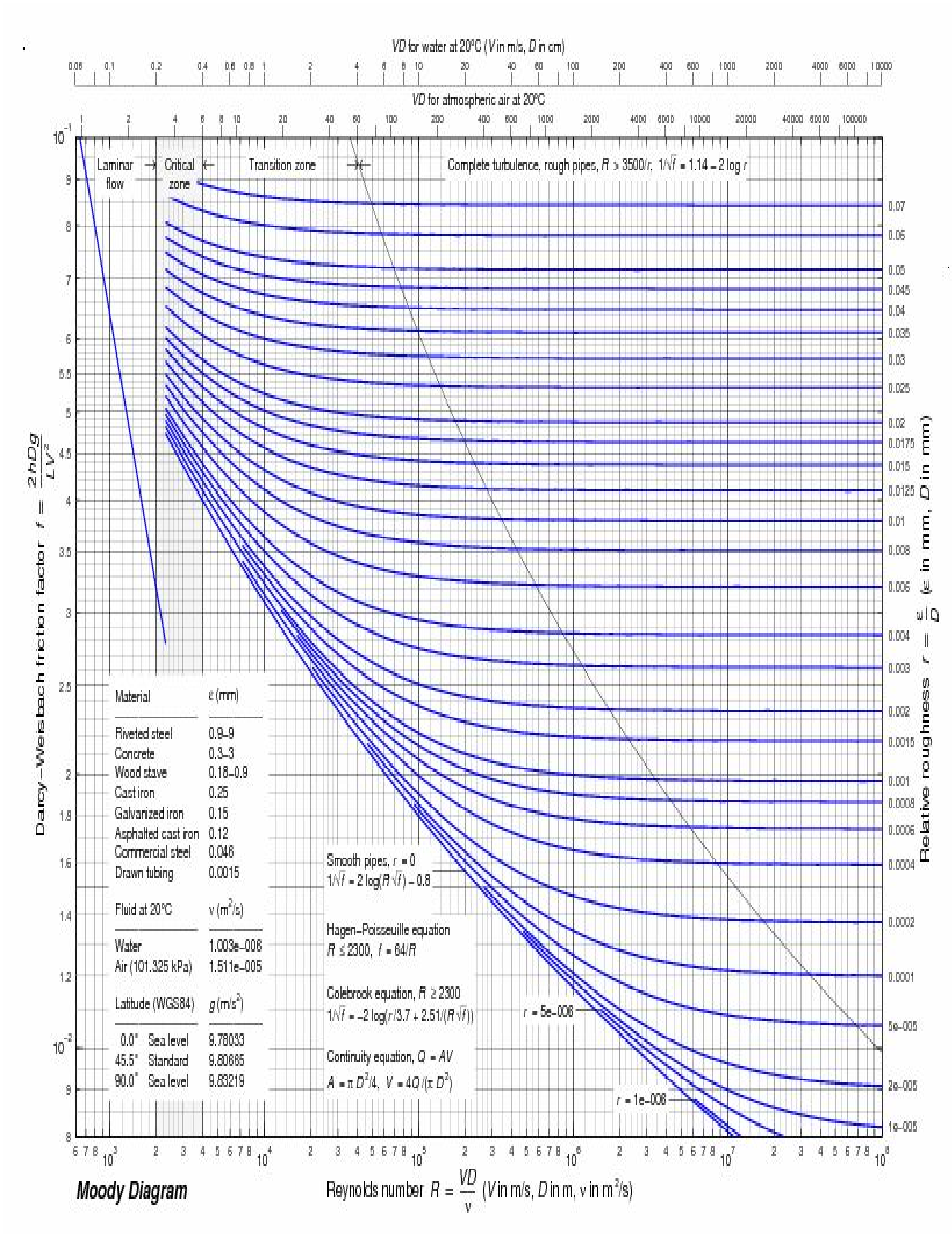
APPENDIX C: Moody diagram



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APPENDIX D: 1 Guide for Pump Selection with Operating Speed of 2900 R.P.M.

PERFORMANCE REQUIREMENTS TO BE MET SPECIFIED AS DISCHARGE FLOW DESIRED AND PRESSURE HEAD TO BE OVERCOME

SPECIFIED SPEED FOR VARIOUS DISCHARGE FLOWS AND PRESSURE HEADS IS USED AS AN INDICATOR OF PUMP TYPE FOR

ROTODYNAMIC MACHINES, MAXIMUM EFFICIENCY OCCURS IN SPECIFIED SPEED RANGE 29-58(METRIC UNITS)

SPECIFIED SPEED OF	<12.5	METRIC UNITS	MULTI-STAGE (RADIAL OR MIXED) FLOW
SPECIFIED SPEED OF	12.5-96	METRIC UNITS	RADIAL(CENTRIFUGAL) FLOW TYPE
SPECIFIED SPEED OF	81-175	METRIC UNITS	MIXED FLOW TYPE
SPECIFIED SPEED OF	> 175	METRIC UNITS	AXIAL FLOW TYPE

	DESIRED DISCHAGE FLOW IN M^3/S													
	0.0001	0.001	0.01	0.1	1	2	3	4	5	6	7	8	9	10
$M_W H$	MULT-STAGE TYPE OR POSITIVE DISPLACEMENT			CENTRIFUGAL TYPE PUMPS				MIXED FLOW TYPE PUMPS						
200	1	2	5	17	55	77	94	109	122	134	144	154	164	172
190	1	2	6	18	57	80	98	113	127	139	150	160	170	179
180	1	2	6	19	59	83	102	118	132	145	156	167	177	187
170	1	2	6	19	62	87	107	123	138	151	163	174	185	195
160	1	2	6	20	64	91	112	129	144	158	171	182	193	204
150	1	2	7	21	68	96	117	135	151	166	179	191	203	214
140	1	2	7	23	71	101	123	143	159	175	189	202	214	225
130	1	2	8	24	75	107	130	151	168	185	199	213	226	238
120	1	3	8	25	80	113	139	160	179	196	212	226	240	253
110	1	3	9	27	85	121	148	171	191	209	226	241	256	270
100	1	3	9	29	92	130	159	183	205	225	243	259	275	290
90	1	3	10	31	99	140	172	198	222	243	263	281	298	314
80	1	3	11	34	108	153	188	217	242	266	287	307	325	343
70	1	4	12	38	120	169	208	240	268	294	317	339	359	379
60	1	4	13	43	135	190	233	269	301	330	365	380	404	425
50	2	5	15	49	154	218	267	308	345	378	408	456	463	488
40	2	6	18	58	182	258	316	365	408	447	482	516	547	577
30	2	7	23	72	226	320	392	452	506	554	599	640	679	715
20	3	10	31	97	307	434	531	613	686	751	811	867	9200	970
10	5	16	52	163	516	729	893	1031	1153	1263	1364	1459	1547	1631
5	9	27	87	274	867	1227	1502	1735	1939	2124	2295	2453	2602	2743
1	29	92	290	917	2900	4101	5023	5800	6485	7104	7673	8202	8700	9171
	CENTRIFUGAL		AXIAL FLOW TYPE PUMPS INDICATED FOR PERFORMANCE IN THIS RANGE											

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ISO 3183:2007(E)

APPENDIX E: MINIMUM YIELD STRENGTH

Pipe grade	Specified outside diameter D mm (in)	Percentage of specified minimum yield strength for determination of S	
		Standard test pressure	Alternative test pressure
L175 or A25	$\leqslant$ 141,3 (5.563)	60 [a]	75 [a]
L175P or A25P	$\leqslant$ 141,3 (5.563)	60 [a]	75 [a]
L210 or A	any	60 [a]	75 [a]
L245 or B	any	60 [a]	75 [a]
L290 or X42 to L830 or X120	$\leqslant$ 141,3 (5.563)	60 [b]	75 [c]
	> 141,3 (5.563) to $\leqslant$ 219,1 (8.625)	75 [b]	75 [c]
	> 219,1 (8.625) to < 508 (20.000)	85 [b]	85 [c]
	$\geqslant$ 508 (20.000)	90 [b]	90 [c]

[a] For $D \leqslant$ 88,9 mm (3.500 in), it is not necessary that the test pressure exceed 17,0 MPa (2 470 psi); for $D >$ 88,9 mm (3.500 in), it is not necessary that the test pressure exceed 19,0 MPa (2 760 psi).

[b] It is not necessary that the test pressure exceed 20,5 MPa (2 970 psi).

[c] For $D \leqslant$ 406,4 mm (16.000 in), it is not necessary that the test pressure exceed 50,0 MPa (7 260 psi); for $D >$ 406,4 mm (16.000 in), it is not necessary that the test pressure exceed 25,0 MPa (3 630 psi).

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APPENDIX F: Pumping equipment efficiency and useful life- *Based on 2,000 hours per year of use. With proper maintenance and fewer hours of annual use, the useful life could be increased.

Type	**Attainable Efficiency Percent**	**Useful Life* (Yrs)**
Centrifugal-type pump	75-85	15
Right-angle pump drive (gear head)	95	15
Automotive engines - gasoline - LPG	20-26 20-26	9 14
Light industrial engine (diesel)	25-37	14
Electric motors	85-92	25

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