LNG REGASIFICATION PLANT DESIGN

A project report submitted in partial fulfillment of the requirements for the degree of

MASTER OF TECHNOLOGY

in

GAS ENGINEERING

(Academic Session: 2004-2006)

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CERTIFICATE

This is to certify that the Project Report on “LNG regasification plant design” submitted to University of Petroleum & Energy Studies, Dehradun by Mr. Vinod.R.G in partial fulfillment of the requirement for award of Degree of Master of technology in gas Engineering (Academic Session: 2004-06) is a bonafide work carried out by him under my supervision and guidance. This report has not been submitted anywhere else for any other degree or diploma.

Date: ....

....

Signature

(Dr.R.P.BADONI)

ACKNOWLEGEMENT

I would like to thank Dr. B.P.Pandey, Dean , Dr.D.N.Saraf , distinguished professor, college of engineering, Dr. R.P.Badoni M.tech program director and the other faculties who have provided me great help and given me inspiration in completing my final thesis.

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VINOD.R.G

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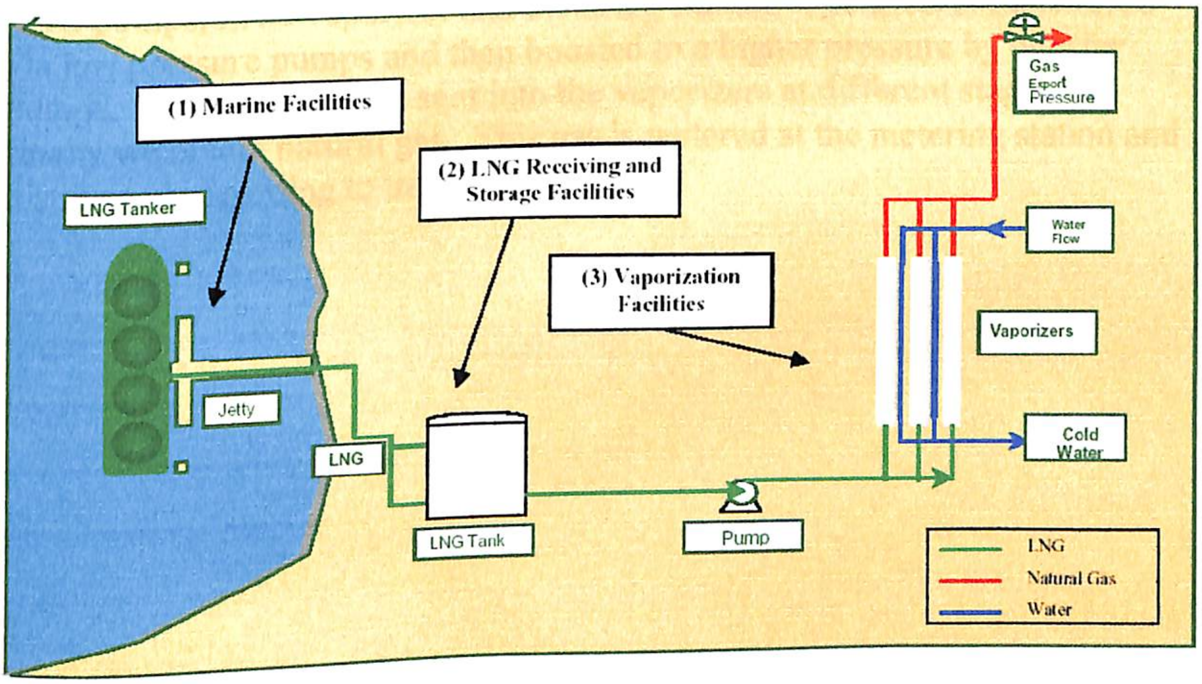
CHAPTER 1

Layout of LNG Regasification plant

Working of the Plant



LNG receiving area

The physical area of the receiving area is from the jetty to the end of the trestle structure. One LNG tanker may be berthed at the jetty at a time, and the liquid will be unloaded via three 12" unloading arms with the combination average capacity of 10000 cu m/hr.

Transfer to Tanks

From the jetty, there will be two 30" and 2.5 KM pipelines to transport the liquid to the storage tank. The pipeline will be located on the trestle. LNG will be stored in two refrigerated tanks of capacity 148000 cu m. at -160 degree C, with an operating pressure of .05 barg.

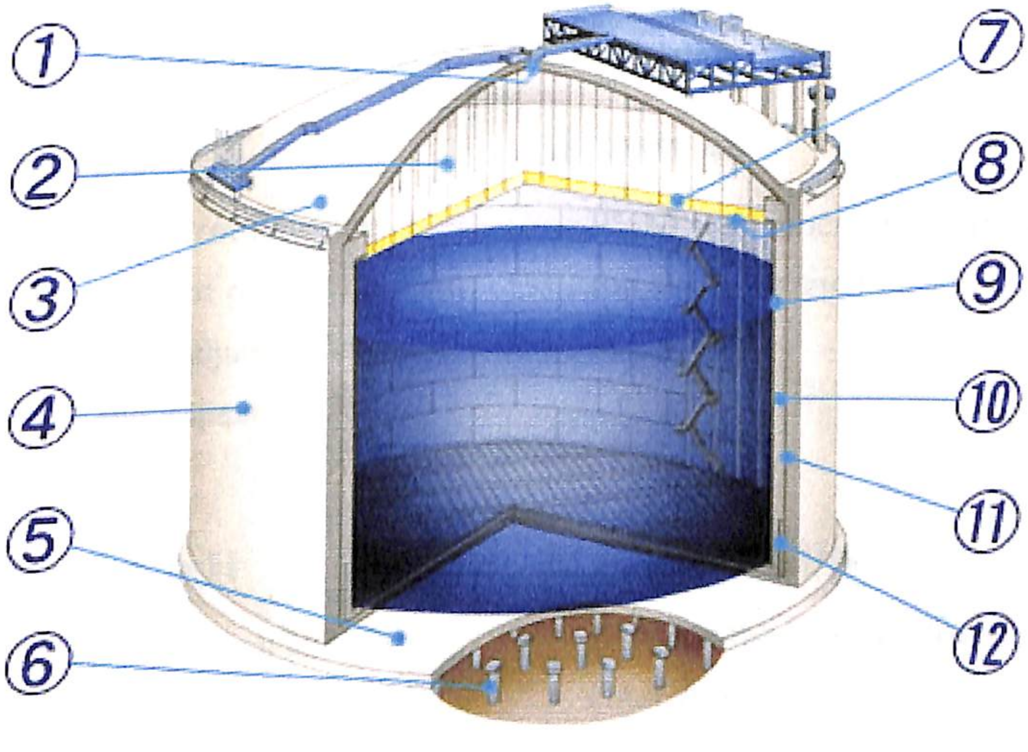
LNG vaporization and sent out

The send out facility comprises of the intank pumps, BOG condenser, HP LNG pumps, LNG vaporizer and metering station. The LNG is transferred via low pressure pumps and then boosted to a higher pressure by booster pumps. This LNG is then sent into the vaporizers at different stages and finally we obtain natural gas. This gas is metered at the metering station and distributed according to the requirements.

CHAPTER 2

Types of LNG storage tanks

FULL CONTAINMENT TANK



① Roof liner

A steel roof liner is installed on the inside face of the RC roof to keep natural gas vapor inside.

② Suspension rod

Stainless steel suspension rods hang the suspended deck from the roof structure.

③ RC roof

A concrete roof has the advantage of protection from heat from adjacent fires and from the impact of flying objects.

④ PC side wall

PC concrete wall is a LNG-tight design and monolithic connections are made between wall and roof, wall and bottom.

⑤ RC bottom slab

RC concrete bottom slab with piles is the foundation of the tank.

⑦ Roof insulation

Perlite powder or glass wool is used for roof insulation to maintain the inside temperature.

⑧ Suspended deck

Aluminum alloy suspended deck supports the roof insulation.

⑨ Inner shell

9%Ni steel inner tank is an open top structure and holds LNG in it.

⑩ Side insulation

Between the inner shell and outer side wall is filled with perlite powder to make side insulation.

⑪ Side liner

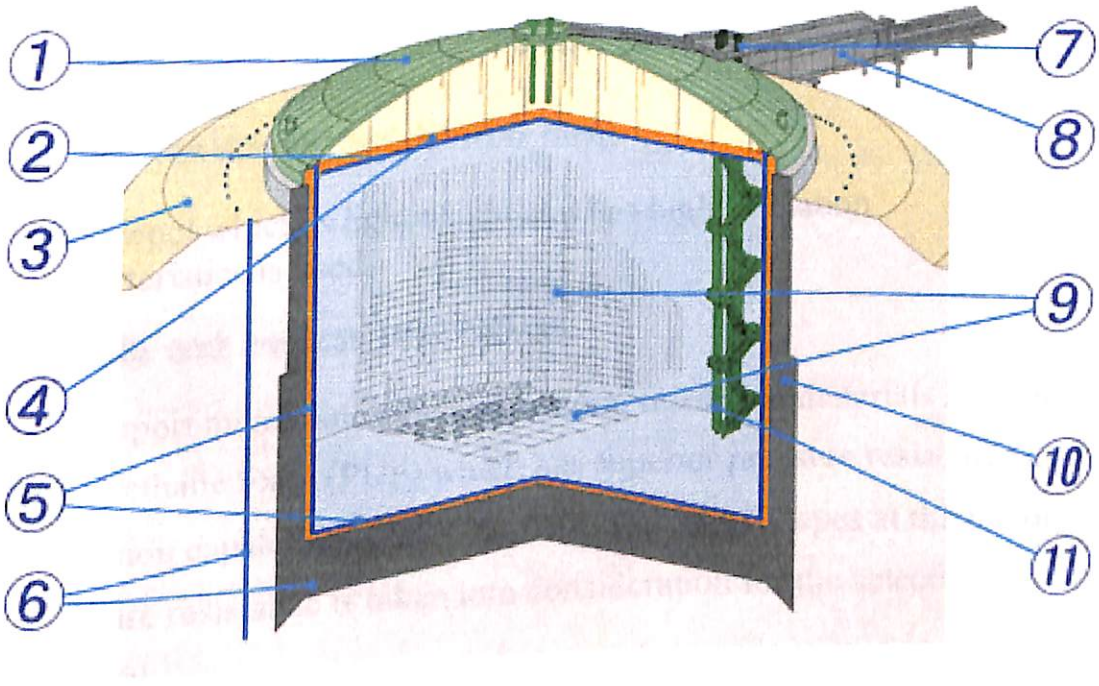
A steel side liner is installed on the inside face of the PC side wall to keep natural gas vapor inside.

⑥ RC piles

Piles support the tank structure.

⑫ Secondary barrier

In the very rare case of inner tank leakage, LNG will accumulate inside of the 9%Ni steel secondary barrier.



① Domed roof

The roof is provided to hold gas in the tank and accommodates all nozzles,openings and pump barrels. The roof structure is composed of rafters, rings and roof plates, and is supported on the top of the wall. In some cases the roof is covered by concrete to protect the steel surface.

② Suspension deck

The suspension deck of aluminum-alloy is hung from the roof structure and has the function to support the insulation material on the suspension deck. The level of the suspension deck is designed with consideration of sloshing wave height which may occur from earthquakes.

③ Berm

The berm is provided around the tank for increased safety.

④ Insulation on suspension deck

Glass wool which is lightweight and has high insulation characteristics is used.

⑤ Side and bottom insulation

To support membrane side and bottom, insulation materials are rigid polyurethane foam (PUF) which has superior pressure resistance and insulation capabilities and is fabricated in panel shapes at the factory. Pressure resistance is taken into consideration for the selection of the material.

⑥ Heater

Bottom and side heating pipes are provided to prevent frost heave.

⑦ Pump stage

The pump stage is provided for maintenance of submerged pumps.

⑧ Piping and sub-rack

Piping is to feed and discharge liquid and gas to/from tank, and is supported by sub-rack.

⑨ Membrane

The membrane is to seal in liquid and gas, and is made of stainless steel (SUS304) which has extensive usage for low temperature applications. MHI's membrane system is installed on the insulation panel and the pressure of liquid and gas is transferred to the tank via the insulation layer, insuring the operation is stable and safe. The MHI membrane is of a single corrugation type. Corrugations are formed by being bent in corrugation shape and arranged lengthwise and crosswise to absorb thermal contraction in any direction.

⑩ Side wall and base

The side wall and base are made of reinforced concrete to withstand forces such as gas and liquid pressure, soil pressure, water pressure, earthquakes and other loads. The structure and its construction method are determined by soil and other conditions.

⑪ Pump barrel framing

Discharge pump barrel and feed pipe have openings at the bottom of tank and are supported by the roof. Usually swing protection is installed in the bottom enabling it to slide upward and downward. An innner stair made of stainless steel is provided with the pump barrel for construction purpose.

CHAPTER 3

Selection of Rotating Equipments

Compressors are machines used to increase the pressure of the gas to provide some useful work. Depending on the pressure required and the discharge the compressors are selected. These machines are broadly classified into two broad categories, dynamic and positive displacement. The centrifugal compressor is a dynamic machine when compared to positive displacement. Centrifugal compressors do their work using inertia force applied to the gas by means of rotating blade impellers whereas positive displacement compressors trap the gas by the action of mechanical components and restricts the escape as compression takes place through direct volume reduction.

Reciprocating compressors

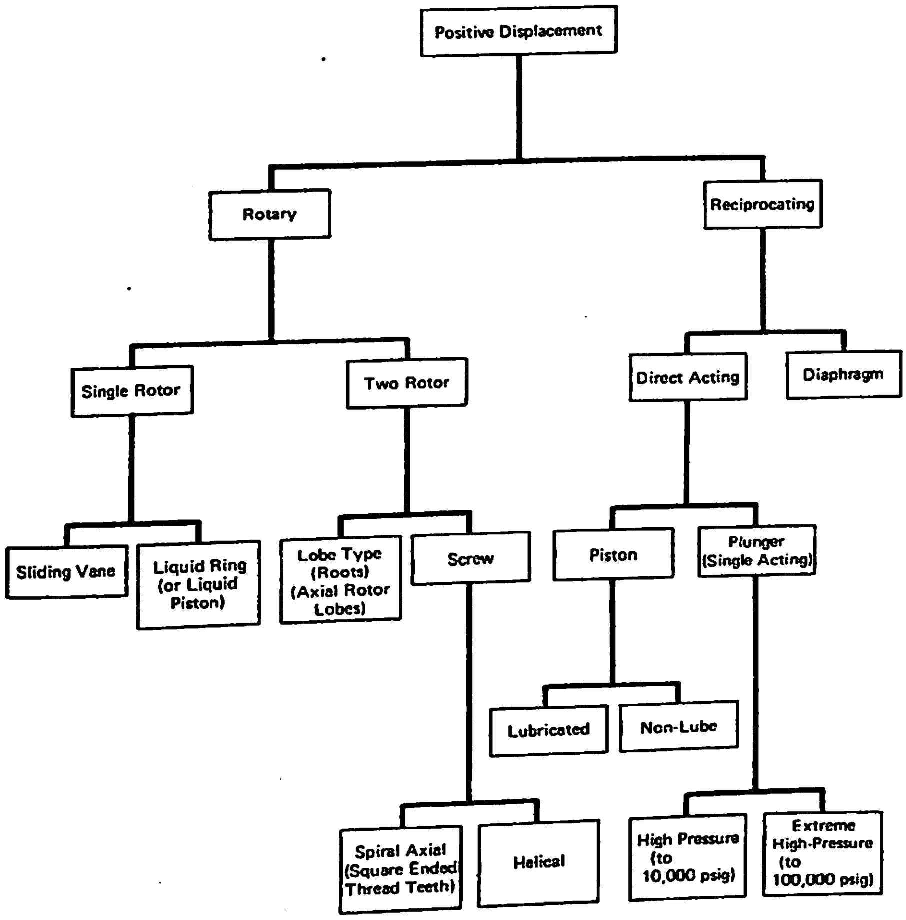


FIG. 2. Approximate ranges of application for reciprocating, centrifugal, and axial flow compressors.

These are available for almost all compressor applications. They are suitable for all pressures from vacuum to 100000 psig. They can handle volume from 5 to 7000 ACFM. Their overall efficiency vary from 80 % to 90 %.

Disadvantages

They are large and expensive.
They have high maintenance cost
High vibrations are produced during running because of moving parts and so strong foundation is to be laid.

Rotary Screw compressors

This compressor type is generally available for a pressure upto 250 psig and for a volume flow rate of 800 to 2000 ACFM. They are balanced machines and require light foundations. Because there is no rubbing action they do not contaminate the gases with lube oil.

Disadvantages

The machine is noisy
The range of operation of the machine is small.
The machines are designed for specific gas and compression ratio.

Sliding Vane compressor

These are available for a pressure upto 150 psig. They are suitable for volume range of 50 to 6000 ACFM. They do not require foundation as heavy as reciprocating machines. These machines are cheaper and require less maintanece.

Disadvantages

In pressure services these are limited to a compression ratio of 3.5 : 1 and to a differential pressure of 60 psig.
They can only be used to handle clean gases.
The efficiency will vary from about 60 to 75 %.

Centrifugal compressors

These are widely used in process plants. They are basically large volume machines. They are available for pressures of upto 5000 psig and can handle volume of gas from 1000 to 150000 ACFM. Because there are no rubbing faces there are no contamination.
Their efficiency range from 65 % to 75 %. The servicing factor is very high that only one compressor is required even in services requiring 3 or more years in continuous operation.

Axial Flow compressors

These are essentially for very high volume flow rate. These are more costlier than centrifugal compressors. Their efficiency is as much as 10 %

higher than centrifugal compressors. These machines have discharge pressure of 200 psig.

Limiting parameters

Horse Power

Reciprocating machines are available upto 15000 HP. Sliding vane compressors are available upto 400 HP per machine. Rotary screw compressors are available upto 6000 HP and centrifugal compressors are used on light molecular weight gases above 2000HP and on general hydrocarbon gases upto 500 HP.

Volume

A centrifugal machine will probably be uneconomical if the suction volume is below 2000 ACFM and the discharge volume is below 500 ACFM. Maximum volume is 150000 ACFM. A rotary screw compressor will probably be uneconomical if suction volume is less than 1000 ACFM or more than 20000 ACFM. Sliding vane compressors may be economic within the range of 50 to 3000 ACFM. Reciprocating compressors may be used for all capacities upto 7000 ACFM.

Pressure

Centrifugal Compressors can be made for discharge pressure of over 5000 psig. The maximum adiabatic head per casing is about 100000 ft. Rotary screw compressors have a maximum discharge pressure of 250 psig. The maximum compression ratio per casing is about 4.5 : 1. Sliding vane compressor have a maximum discharge pressure of 150 psig. The maximum compression ratio is about 3.5 and the maximum differential pressure is 60 psig.

Reciprocating compressors have a maximum discharge pressure of about 100000 psig. The compression ratio per cylinder depends on the value of ratio of specific heats and the value of inlet temperature.

Gas conditions

It is undesirable to have liquids or solids in the gas stream of any compressor. Centrifugal compressors can handle solids and liquids better than any other compressor.
Sliding vane compressor can handle small amount of liquid but the quantity should not be sufficient to dilute the lubricant. Axial compressors are not suitable for fouling gas services since the deposits will greatly reduce their efficiency. Reciprocating and sliding vane compressors are very susceptible to corrosion and wear.

Weather protection

All types of compressors listed earlier are suitable for operations in the open, without weather protection. Even a shelter such as is normally provided is not required for compressors.

Construction and working of positive displacement compressors

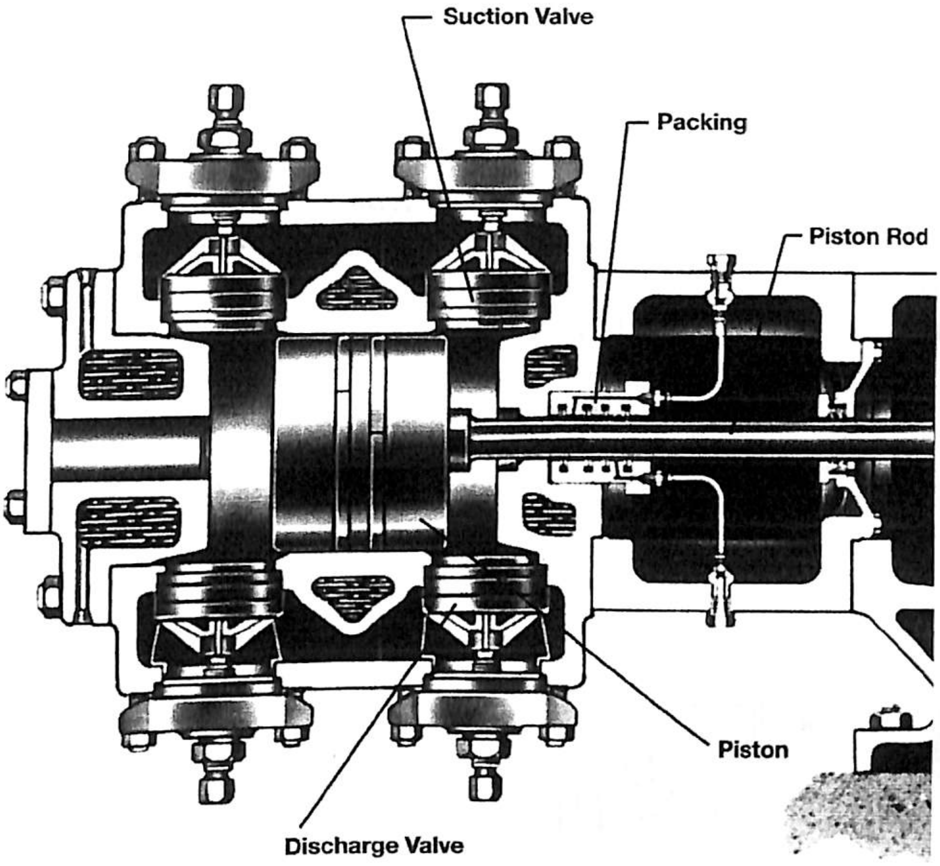


Figure 11-5. Typical double-acting compressor cylinder. *(Courtesy of Dresser-Rand Company.)*

Reciprocating compressors are the oldest and the most widely used type. Suction and discharge valves are spring loaded and work automatically from pressure differentials created between the cylinder and the piping by the moving piston.

The reciprocating compressor can be attained in both air cooled and water cooled models. Speed may range from 125 to 1000 rev/min. Piston speed range from 500 to 950 ft / sec, the majority being 700 to 850 ft/min. The nominal gas velocity is usually in the range of 4500 to 8000 ft/min and the operational discharge pressure ranges from vacuum to 50000psig.

Working of a reciprocating compressor

In position 1 the piston is moving away from the cylinder head and the suction valve is open, allowing the cylinder pressure to be equal to suction pressure and the gas to enter the cylinder. The discharge valve is closed at this point of time. At position 2 the piston has traveled full stroke within the cylinder and the cylinder is full of gas at suction pressure. The piston begins to move to the left, closing the suction valve. In moving from position 2 to position 3 the piston would move towards the cylinder head and the volume is reduced. This increases the pressure until the cylinder pressure is equal to the pressure in the cylinder and the discharge valve opens. The piston continues to move to the end of the stroke until it releases the compressed gas. As the piston reverses itself, the gas remaining within the cylinder expands until it equals suction pressure and the piston is again in Position 1.

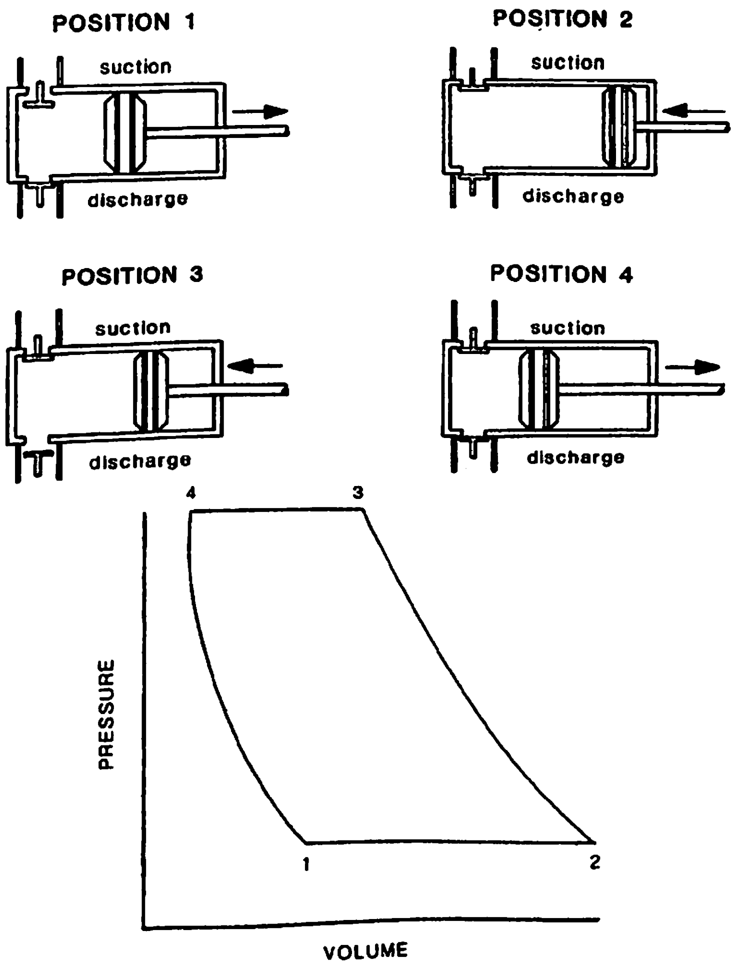


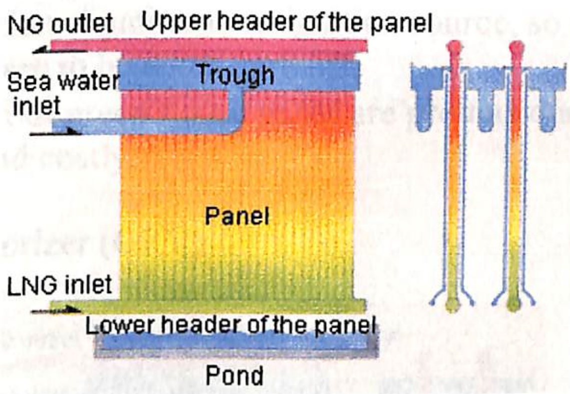
Figure 10-1. Reciprocating compressor action.

CHAPTER 4

LNG vaporizers selection

LNG import terminals have utilized submerged combustion vaporizer technology to regasify the LNG for a long time. The current trend in regasification is by using the heat source from the sea water in open rack vaporizer. Both these methods have encountered resistance during the permit application process due to environmental concern. For the SCV option NOx emissions are concern. For the ORV marine life impact is a concern. Air based vaporization avoids both environmental impact but often has problems of achieving desired gas send out temperature in the winter months.

Submerged Combustion Vaporizer



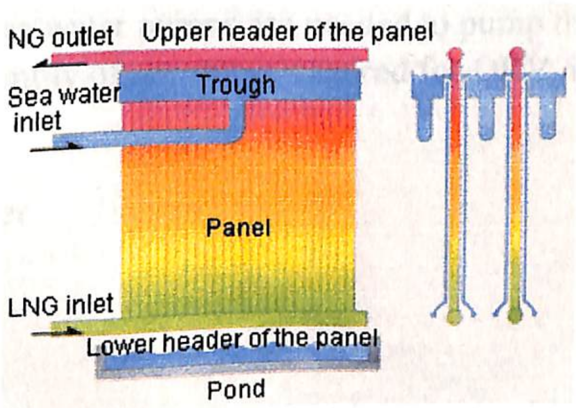
SCVs use a tube bundle submerged in a water bath to vaporize the LNG. The water temperature is maintained by burning the natural gas. Combustion products are bubbled through the distribution tube into the water bath, creating a two phase frothing action. Then two phase froth flows up through the tube bundle and the high velocity motion of gas water mixture efficiently scrubs the tube surface, minimizing ice build up. Heat is transferred from the water bath to the LNG fluid flowing inside the tube bundle. The tube bundle is a muti tube serpentine bundle mounted horizontally within the weir. Combustion products after disengaging from the gas/water, are normally discharged through a short stack. The stack temperature is about 80 deg F.

SCVs can offer extremely high thermal efficiency approaching 100 % due to the condensing combustion products water vapor in the water bath. Since the combustion products are bubbled directly into the water bath, almost all the available heat is transferred to the water. The tube bundle is always immersed in a high thermal capacity water bath. Thus these equipments provide rapid load fluctuations. Condensing the water vapor in the combustion products yield a net water production from the vaporizer water bath. The Ph of water has to be controlled as acids are produced during the operation. Low chlorine water is required to fill the bath to avoid chlorine stress cracking.

Some of the disadvantages of SCV

- It consumes approximately 1.3% of LNG throughput for fuel gas.
- A large blower is required to provide the combustion air at a pressure sufficient to force the combustion products through the water bath.
- The combustion chamber is an ignition source, so offshore applications are to be taken care off.
- More amount of green house gases are produced and reducing them is difficult and costly.

Open Rack Vaporizer (ORV)



It is a widely used and proven technology. Nearly two thirds of the vaporization is done through this technology. It uses the heat source from the sea water. The sea water flows over a series of panel coils to vaporize the LNG that is flowing counter current within the panel. The panels are multiple LNG tubes with integral fins. The ORV panels are coated with the zinc alloy to provide corrosion resistance against sea water. The sea water is cooled by exchanging heat with LNG. The maximum sea water temperature is limited to 15 deg F. With LNG approach temperature of 4 to 6 deg F. Sea water flows though the intake screens to remove the debris and marine life prior to being pumped to the ORV. Intake screens are designed and operated with inlet velocities set to minimize impingement and entrain marine organisms. The treated water is pumped through distribution troughs located on top of the ORV panels. The sea water flows downwards next to the vertical panel and is collected in a concrete basin beneath the ORV. The cooled sea water is gravity fed into the trench and routed to the sea water outfall. Dechlorination is expected to occur naturally as the sea water flows through the sea water handling system and mixes with the ambient sea water. These are considered extremely safe as they have no moving parts in contact with the flammable fuid.

Disadvantages of ORV

- Marine life entrained in the water column is a major concern.
- Sea water from the ORV is routed to the sea water outfall which could affect the local communities.
- To prevent bio fouling of sea water system, significant quantity of chlorine is added.
- Large sea water pumps are needed to pump the sea water.
- The quantity of sea water required for ORV is paramount.

Ambient air vaporizer

These use ambient air with the heat source to vaporize LNG. With these LNG is sent through several banks of extended surface exchangers. Heat is transferred to the LNG where it is vaporized and natural gas is discharged at a temperature approaching the ambient air temperature. The cooled air now more dense tends to travel down and to the bottom of the vaporizer. The size and performance of the unit is largly affected by the ambient air temperature and humidity. Air flow on the outside of the ambient air exchanger is controlled by two means, natural buoyancy of the cooled, dense air and if required installing forced draft air fans. The cooled ambient air can be directed away from the areas directly by operating personnel.

Because LNG is vaporized directly by air , condensed water forms a snow like frost on the vaporizer. This frost slowly reduces the performance and heat transfer. Accordingly the vaporizer must be regenerated by switching off the unit and built up ice crystals. Cooling moist ambient air condenses a significant amount of fresh water from the atmosphere. This water can be collected and used.

Disadvantages of AAV

- These cannot operate for peak shaving loads
- Ambient conditions impact the effectiveness of the vaporizer.

	SCV	ORV	Forced Draft	Natural Draft
Estimated capital cost factor	1.0	1.38	1.317	1.258
Annual Operating Cost	26.01	1.67	1.60	0.15
Life cycle cost	257.06	86.33	82.42	71.05
NOx	144.9	11	1.5	0.7
Chlorine emissions	NO	YES	NO	NO
NOx abatement compatibility	Somewhat	YES	YES	YES
Total Plot Area Sq feet	21000	19500	18100	24900
Weight	933	1750	1630	2535

Aerial coolers are often used to cool a hot fluid to near ambient temperature. They are mechanically simple and flexible and they eliminate the nuisance and cost of cold source.

In warm climate, the coolers may not be capable of providing as low a temperature as shell and tube exchanger. In these coolers the tube bundles are across either in the suction or in the discharge section of the blower. This type of heat exchanger are used to cool gases to ambient temperature.

When the tube bundle is on the discharge of the fan, the exchanger is referred to as Forced draft. When the tube bundle is on the suction side of the fan it is referred to as Induced draft. The below figure shows the diagram of an aerial cooler.

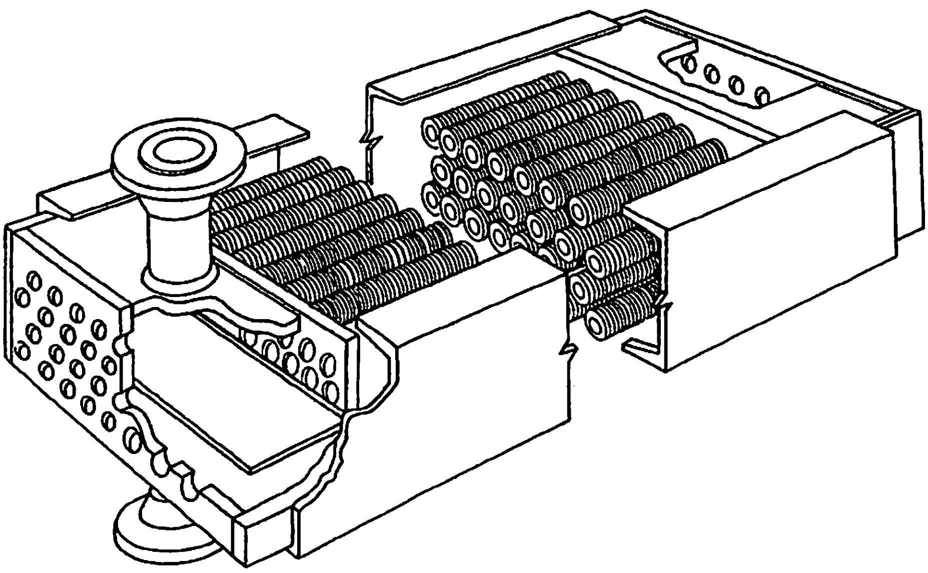


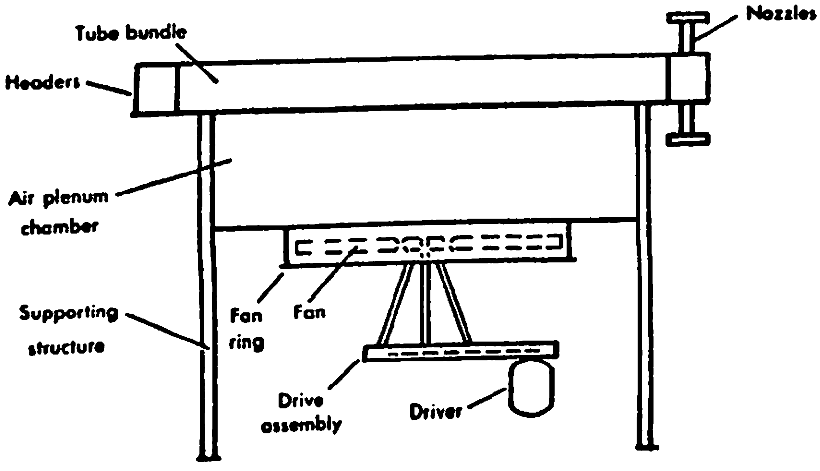
Figure 3-14. Aerial cooler.

The process fluid enters one of the nozzle on the fixed end and the pass partition plates force the flow through the tubes of the floating end. Here it crosses over the remainder of the tubes and flows back to the fixed end out though other nozzle. Air is blown across the finned section to cool the process fluid. Plugs are provided so that they can be plugged if they have leak or not in use. The tube bundle could also be mounted in a vertical plane, in which case air would be flown horizontally though the cooler.

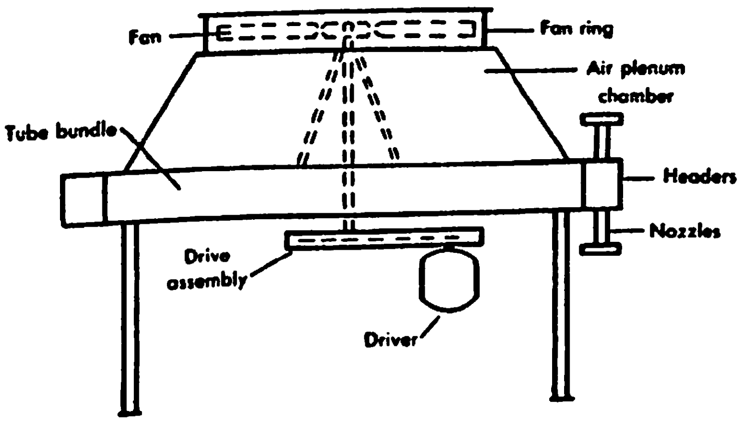
Forced air exchangers have tube length ranging from 6 to 50 ft. and tube diameter upto 2 inch. The tubes have fin on them since air is non fouling and it has low heat transfer efficiency. The fins increase efficiency by effectively adding surface area to the outside of the tubes. In a single aerial cooler there may be several fans with bundles of tubes.

Process outlet temperature can be controlled by louvers, variable fan speed, blade pitch or recirculation of process fluid. As the process flow rate and heat duty change some adjustments have to be made. Too cool a gas temperature would lead to the formation of gas hydrate and developing ice plugs in the cooler. Too cool the lube oil would reduce the viscosity of the oil leading to high pressure drop and inadequate lubrication.

Louvers are probably the most common type of temperature control device on air coolers. They may be automatically adjusted by sensing the fluid temperature. Blade pitch is probably the second most common and variable speed drive is third.



Forced draft



Induced draft

Figure 3-15. Side elevations of air coolers. (From Gas Processors Suppliers Association, *Engineering Data Book*, 9th Edition.)

CHAPTER 5

Calculations for the plant

Tank Calculation

Design Code : BS 7777

Tank dimensions

Inner Tank diameter	:	79.0 m
Design Liquid level inner	:	32.82 m
Hydrotest level	:	19.692 m
Product density	:	480 kg/m3
Operating Temperature	:	- 160 Deg C
Maximum design ambient temp.	:	45 Deg C
Corrosion allowance inner tank	:	0 mm
Inner tank material	:	9 % Ni steel
Youngs Modulus	:	20800 kg/mm2
Coeff of linear expansion	:	9.2 *10^(-6)

Thickness and height of each course of side wall

No. of Course	Nominal Thickness (mm)	Height (mm)
9	12	3730
8	12	3730
7	12	3730
6	12	3730
5	13.2	3730
4	15.6	3730
3	18.2	3730
2	20.8	3730
1	23.5	3730

Plate Thickness of inner tank shell

Minimum shell plate thickness

The minimum shell plate thickness is 12.0 mm according to BS 7777.

Maximum allowable design stress

The maximum allowable design stress is as follows

For operating	:	260 N/mm2
For testing	:	331.5 N/mm2

The maximum allowable stress under hydrostatic test shall be determined by the following expression

Min (UTS/2.35 , 0.2% PS/1.5 , 260 N/mm2)

And the maximum allowable stress under hydrostatic test shall be determined by the following expression

Min (0.2% PS*0.85 , 340 N/mm2)

In this case, the welding metal is weaker than the base metal and the yield strength and the ultimate tensile strength of welding metal are considered.

Their values at room temperatures are as follows

UTS = 660 N/mm2
0.2% PS = 390 N/mm2

Calculation Method

Under operating condition

$$T = D/20S \{98\ W\ (H-0.3) + P\} + c$$

Where,
T = Calculated minimum thickness (mm)
H = Height from the bottom of the course under consideration of the highest liquid level (m).
D = Tank inside diameter (m)
W = Maximum density of liquid under storage condition (g/m3)
S = Allowable design stress (N/mm2)
P = Design Pressure c = Corrosion allowance

Under testing conditions

Tt = D/20 * St {98 Wt (H-0.3) + Pt}

Where,

Tt = Calculated minimum thickness (mm)
H = Height from the bottom of the course considered
D = Tank inside diameter (m)
Wt = Maximum density of liquid under storage conditions (g/ m3)
St = Allowable design stress (N/ mm2)
Pt = Design pressure (mbar)

Calculated results

Course	Liquid level (m)	Calculated Thickness (mm)	Liquid level (mm)	Calculated Thickness (mm)	Design Thickness (mm)
9	2.980	2.0			11.8
8	6.710	4.6			11.8
7	10.44	7.3			11.8
6	14.17	10.0	1.042	0.9	11.8
5	17.9	12.6	4.772	5.3	13.0
4	21.63	15.3	8.502	9.6	15.4
3	25.36	18.0	12.232	14.0	18.0
2	29.09	20.6	15.962	18.3	20.6
1	32.82	23.3	19.692	22.7	23.3

Fabrication allowance of 0.2 mm is taken into consideration.

Stifferners of inner tank

The pressure difference between inside and outside of inner shell is negligibly small. The annular space is filled with pearlite insulation, so the external pressure exerted by pearlite will be acting on the inner tank shell. The stifferners of the inner tank are set to prevent from the external buckling of the inner shell.

External Pressure

In order to limit the external pressure build up by the pearlite powder, the resilient glass wool blanket is installed and securely held against the outer surface of the inner tank shell.

1. At filling pearlite powder

When the pearlite powder will be filled in the annular space, the resilient glass wool blanket will be pressed by the pearlite powder pressure.

The pearlite powder pressure is calculated by the following expression

Pp = TW/2f (1 – e ^ (-2kfh/T)) = 56.3 Kgf/m2

Where,

Pp = Pearlite powder pressure
T = Thickness of the insulation space = 1.0 m
W = Gravity of pearlite powder = 65 Kgf/m2
F = Friction coeff between insulation and shell plate = 0.577
K = Ratio between horizontal pressure and vertical pressure = 0.33
H = Height of pearlite insulation = 37.08 m

The resilient glass wool blanket is pressed to 75 % thickness of initial thickness. The reduction is glass wool blanket thickness is 50 mm by above pressure.

2. At cool down

When the inner tank is cooled down, the inner tank shell will shrink. But the space made by the inner tank shrinkage will be filled by the pearlite powder and the powder pressure will not change.

3. At warm up

When the inner tank will be warmed up, the inner tank shell should be absorbed by the inner tank warm up. The thermal displacement of inner tank, dt, shell can be calculated by following expression

Dt = A * T * R == 77.4 mm

A = Coeff of linear expandsion of inner tank. (9.2 * 10^-6)
T = Temp difference (=213 = 45 + 168)
R = Radius of inner tank (39.5 m)

So, the total shrinkage of glass wool is as follows

127.4 = 77.4 + 50

Thus the compressed ratio is

64 % = 127.4 / 200

Location of stiffener

The following factor of safety is considered in designing the stiffeners.

For operating = 2.0
For warming up = 1.0

L <= D (t/D)^(0.5) { 0.45 + 2.6 E/P (t/D)^2}

Where,

L = Pitch of the stiffener cm
D = Inner tank diameter = 7.9 m
T = Average thickness of the shell plate cm
E = Youngs Modulus
P = External pressure

Calculation results of stiffener (Operating conditions)

Number	Average thickness	External pressure	Effective range of reinforcement		Pitch of stiffeners
Top girder	1.2	0.0113	1123.8	>	396
4	1.2	0.0113	1123.8	>	453
3	1.2	0.0113	1123.8	>	453
2	1.3	0.0113	1372.9	>	551
1	1.96	0.0113	3732.6	>	1459

0.0113 == 0.00563 * 2.0

Calculation results of stiffener (Warm up conditions)

Number	Average thickness	External pressure	Effective range of reinforcement		Pitch of stiffeners
Top girder	1.2	0.0236	559.0	>	396.0
4	1.2	0.0236	559.0	>	453.0
3	1.2	0.0236	559.0	>	453.0
2	1.3	0.0236	678.8	>	551.0
1	1.96	0.0236	1809.8	>	1459.0

Size of the stiffeners

The size of the stiffeners shall be determined in accordance with JGA guidelines. The required moment of inertia of stiffeners I, which is clued the effective sectional area of the shell plate can be calculated by the following expression.

I >= PD3 * LC / 8E (N2 -1)

Where,

I = moment of inertia of the stiffener
P = External pressure = 0.0236 kgf/cm2
D = inner tank diameter = 7.9 m
L = Covered range of each stiffener
E = Youngs modulus
C = Safety factor = 3
N = Number of buckling wave

N = [7.06 / (H/D)^2 *(Tm /D)]^(0.25)

H = Inner tank shell height = 3.357 m
Tm = Average thickness of the shell plate = 1.547 cm

Calculation results of size of stiffener (Warm up condition)

Number	Lower side pitch	Upper side pitch	Covered range	Required moment	Shell plate thickness	Width of stiffener	Moment of inertia	
4	453	396.0	424.5	1996	1.2	22.0	3354	OK
3	453	453.0	453.0	2130	1.2	22.0	3354	OK
2	551	453.0	502.0	2360	1.2	22.0	3354	OK
1	1459	551.0	1005.0	4725	1.56	27.0	6682	OK

Bottom plate of inner tank

The thickness of the bottom plate and the annular plate shall be as follows

Bottom plate thickness : 6.0 mm
Annular plate thickness : 14.0 mm

The plate thicknesses are according to para 7.2.3.1 of BS 7777 part 2.

Boil off Calculation

Properties of insulation materials

Material	Thickness (m)	Thermal Conductivity (kcal/hr m C)
Dry sand	Tb1 = 0.4	Kb1 = 0.5
Foam Glass	Tb2 = 0.408	Kb2=0.039
Base glass	Tb3 = 1.0	Kb3=2.0
Concrete ring beam	Tb11=0.3	Kb11=2.3
Leveling pearlite concrete	Tb21=0.508	Kb21=0.18
Base slab	Tb311=1.0	Kb31=2.0
Concrete wall	Ts1=0.8	Ks1=2.0
Perlite	Ts2=0.8	Ks2=0.0378
Fibre glass	Ts3=0.2	Ks3=0.033
Roof Concrete	Tr1=0.4	Tr1=2.0
Fibre glass	Tr2=1.0	Tr2=0.033

Heat influx calculation

$$Q = Qb + Qs + Qr = Qbi + Qbii + Qs + Qr$$

Where,

Q = Total heat flux
Qb = Heat flux through the botoom part
Qbi = Heat flux through the center part of the bottom

Qbii = Heat flux through the ring part of the bottom
Qs = Heat flux through the side wall part
Qr = Heat flux through the roof part

At day time

(a) Heat flux through bottom Qb

(i) Centre part

Qbi = Ubi * Abi * (T1 – T2)

= 0.0851 * 4347.5 *213.0

= 78804 Kcal/hr

Where,

Ubi = Average heat transfer coefficient

= (Tb1/Kb1 + Tb2/Kb2 + Tb3/Kb3)^(-1)

Abi = 4347.50

T1 – T2 = 45.0 - (-168.0) = 213 deg C

Ring Part

Qbii = Ubii * Abii * (T1 – T2)

= 0.288 * 805.54 * 213.0

= 49415 Kcal/hr

Ubii = (Tb11/Kb11 + Tb21/Kb21 + Tb31/Kb31)^(-1)

= 0.288 Kcal /hr.m2 C

Abii = 805.54 sq m

(b) Heat flux through side wall Qs

Qs = Us * As * Tdiff

= 0.0362 * 8884.04 * 229.1

= 73680 Kcal/ hr

Where,

Us = (Ts1/Ks1 + Ts2/Ks2 + Ts3/Ks3)^(-1)
= 0.0362 Kcal/hr m^2 C

Tdiff = 97702 - (-168) +45.0 - (-168)/2
= 229.1 deg C

(c) Heat flux through roof

Qr = q1 * Ar
= 7.6 *4901.67
= 37435 Kcal/ hr

Where

Ar = 4901.67 sq m

Assuming the spherical dome to be horizontal plane heat input q1 *Ar through the roof is determined through the heat balance q1=q2=q3

Q1 = Ur1 * Tdiff1

= 5 * 1.53

= 7.6 Kcal/hr m2

Ur1 : Average heat transfer coeff

Ur1 = (tr1/Kr1)^(-1)
= 5 Kcal/hr m2 deg C

Tdiffl = Tr – Ti
= 82.2 – 80.67
= 1.53 Deg C

Q2 = Heat transfer between inside of the roof and upper surface pf suspended deck insulation by heat radiation

Q2 = 4.88

Q3 = Heat transfer through the suspended deck insulation by thermal conduction
Q3 = Ur2 * Tdiff r2

= 0.033 * 231.09
= 7.6 Kcal/ hr sq m

Ur2 = (Tr2/ Kr2)^(-1)
= 0.033 Kcal/hr sq m Deg C

T diff r2 = Tu – Td
= 63.09 -(-168)
= 231.09 Deg C

Tr : Temperature at outer surface of roof (82.8 deg C)
Ti : Temperature inside the roof
Tu : Temperature at the upper surface of the suspended insulated deck
Td : Temperature at lower surface of suspended insulation deck
Ar : Area of spherical dome
Ad : Area of suspended deck insulation

(d) Total heat flux : (Q)

Q = Qbi + Qbii + Qs + Qr

= 78804 + 49415 + 73680 + 37435

= 39334 Kcal/hr

(e) Boil of rate at day time

Rate = 240Qt/(VpL)
= 2400 * 239334 * /(160873 * 425 * 122)
= 0.0689

Where ,

V : Tank content volume
p : Liquid density
L : Latent heat

At night time

(a) Heat influx through bottom :(Qb)

Same as day time

Therefore

(i) Center part Qbi = 78804 Kcal/ hr
(ii) Ring part Qbii = 49415 Kcal /hr

(b) Heat flux through side wall (Qs)

Qs = Ws * Ts * Tdiff s1
= 0.0362 * 8884.04 * 213
= 68504 Kcal/hr

Tdiff S1 = 45 -(-168)
= 213 Deg C

(c) Heat flux through the roof Qr :

Qr = q1 * Ar
= 6.303 * 4901.67
= 30896 Kcal/hr

(d) Total heat influx

Q = Qbi + Qbii + Qs +Qr
= 78804 + 49415 + 68502 + 30896
= 227617 Kcal/hr

(e) Boil off rate at night time

Rate = 2400Qt/(VpL)
= 2400 * 22761 / (160873 * 425 * 122)
= 0.0655

Boil of rate all day long

At day time = 0.0689 (wt %/day)
At night time = 0.0655 (wt%/day)

Assuming the day time and night time would go on for 12 hr each, boil off rate is equal to 0.0672 < 0.08 (wt %/day)

Boil off gas produced = 138000*0.64*600 *0.0672 / (3.8*24*100)
= 360 Kg / hr
= 0.5 mmscfd

Assuming two tanks implies total BOG generated is equal to 1 mmscfd.

Since the amount is less and pressure ratio is more we go for reciprocating compressor.

LIFTER COMPRESSOR SELECTION

Suction Pressure P1 = 35.73 sia
Discharge Pressure P2 = 150.73 psia
Volume to be handled V = 3200 Sm^3/hr = 1 MMscfd
Inlet Gas deviation factor = 0.99
Exit Gas deviation factor = 0.96
Specific Gravity = 0.65
K = 1.26

Standard Conditions

Temperature = 60 °F
Pressure = 14.73 Psia

BHP Calcuations

Suction Conditions

Pressure = 35.73 psia
Temperature = 560 °R

1. BHP = P1V1/229 *K/(K-1) * [(P2/P1)^((K-1)/K) - 1]

V1 = 1920 * 14.73 * 560 /324.73 /520

= 360 cfm.

BHP = 35.73* 360/229*1.26/0.26*[(150.73/35.73)^(0.26/1.26) - 1]

= 95

Accessories = 30 HP

Total HP required = 95 +30 = 125 HP

20 % Extra = 150 BHP

2. P2/P1 = 150.73/35.73 = 4.22

(BHP/MMSCFD)I Stage = 98 [At K=1.26 Compression Ratio = 2.57]

BHP = V*Pb*T1/14.4/Tb * (BHP/MMSCFD)

= 1*14.73*540/14.4/520*98

= 104 BHP

Accessories = 30 BHP

Total = 134 BHP

20% Extra = 160 BHP

3.
BHP= 0.0857*((Z1+Z2)/2)^(1/1.26)* Z1^(0.26/1.26)*[Qg*Ts/E]
*[K/(K-1)][(P2/P1)^((K-1)/K) −1]

BHP=0.0857*((0.98+0.92)/2)^(1/1.26)*(0.98)^(0.26/1.26)*[1*540/0.75]*

[1.26/0.26]* [4.22^(0.26/1.26) -1]

= 99.3 BHP

Accessories = 30 BHP
Total = 129.3 BHP

4. BHP = 22*F*n*R*Qg

= 22*1*1*4.22*1

= 93 BHP

Accessories = 30 BHP

Total BHP required = 123 BHP

20% Extra = 147 BHP.

Determination of Volumetric Efficiency

I Stage

Volumetric Efficiency = 96-R-C*[R^(1/K)*Z1/Z2 - 1]

Assume clearance C = 10%

Compression Ratio R = 4.22

Volumetric Efficiency = 96-4.22-0.1*[2.57^(1/1.26) –1]

= 91.57%

Determination of Piston Speed

Speed N = 960 rpm

Bore = 8.5 inch

Stroke L = 5 inch

$$L = 2 * r$$

Thus Crank Radius $r = 2.5$ inch

1 revolution $= 2 * 3.14 * r$

$= 2 * 3.14 * 2.5$

$= 15.7$ inch

960 rpm $= 15072$ inch/min

$= 6.3$ m/sec

Determination of Piston Displacement (PD)

I Stage

Double acting cylinder
N = 960 rpm
Stroke (S) = 5 inch
Bore (dc) = 8.5 inch
Diameter of rod (dr) = 2.5 inch

$$PD = [2*dc^2 - dr^2]*S*N / 2200$$

$$= [2*8.5^2 - 2.5^2]*5*960 / 2200$$

$$= 301.63 \text{ cfm}$$

Rod Load Calculations

I Stage

Double acting cylinder

Area of piston (a_p) $= 3.14 * 8.5^2 / 4$ $= 86.72$ sq. inch

Area of rod (a_r) $= 3.14 * 2.5^2 / 4$ $= 4.91$ sq. inch

Compressive Load (RLc)

$$RLc = a_p (P_{inter} - P_1) + a_r * P_1$$

P1 = 224.73 psia
Pinter = 577.56 psia
P2 = 1414.73 psia

$$RLc = 86.72*(150.73 - 35.73) + 4.91*35.73$$

$$= 27821.4 \text{ lb} < 50000 \text{ lb}$$

Tensile Load (RLt)

$$RLt = a_p (P_1 - P_{inter}) + a_r * P_{inter}$$

$$= 86.72*(35.73 - 150.73) + 4.91 * 150.73$$

$$= 26905.92 \text{ lb} < 50000 \text{ lb}$$

Driver Selection (Per Year Basis)

Engine

Operating Cost

Fuel Price

Power Generated = 410 KW
Efficiency of engine = 50 %
Thus Fuel Power 410 / 0.5 = 820 KW

This is equal to 820 KJ/sec = 196 Kcal/sec

1 cu meter = 8000 Kcal/sec
Thus the flow rate has to be 0.025 cu meter/sec to produce 820 KW
Thus the flow rate for the day = 2116.8 cu meter / day

1 Mscm = Rs 1700

Thus 2116.8 scm = Rs 3600

Fuel Price = 3600 * 300 = Rs 1080000

Lube Oil Price = Rs 8 * 80 * 300 = Rs 192000

Man Power Cost = 1*365*1000 = Rs 365000

Schedule Maintenance Down time cost

This goes for 5 days and 8 hours per day, this implies two full days.

Thus total gas = 5300 * 24 = 0.13 MMscmd

Thus total cost = 0.26*1000*1700 = Rs 442000

Maintenance Cost

Scheduled Maintenance

Lube Oil = 400 liters

5 times a year, this implies 400*5 = 2000 liters

Total cost of lube oil = 2000* 80 = Rs 1.6 lakh

One filter costs Rs 2000 thus 14 filters cost Rs 28000

This maintenance is done for 5 times in an year

Thus the total cost of scheduled maintenance is Rs 5*28000 = Rs 1.4 lakh

Number of people working on the machine at a time = 5

Wage paid to per person per day = Rs 1000

Thus total amount of money paid as wages = Rs. 25000

Thus total cost = 160000 + 140000 + 25000 = Rs 325000

Spares Cost = Rs 400000

Overhauling Cost (For 30 days and once in 3 years)

= 1/3 (spares + manhours)

Spares Cost = Rs 20 lakh
Man hours = Rs 30 *1000*5 = Rs 1.5 Lakh

Overhauling Cost = 1/3 (20 +1.5) = 7 Lakh

Down time cost = 130 * 1700 * 30 = Rs 66 lakh

For one year = Rs 22 lakh

Total Operating cost = Rs (22 + 4.42 + 3.65 + 10.8) = Rs 42 lakh

Total Maintanece Cost = Rs (3.25 + 4 + 7) = Rs 14.25 lakh

Calculation of Net Present Value considering 5 % escalation every year

(IN LAKH)

No of years	Capital Cost	Depreciation 10 %	Operating Cost	Maintanece Cost	Tax33.5% (D+O+M)	Outflow O+M-T	NPV
0	150	0	–	–	–	–	–
1		15	42	14.25	23.9	32.35	29.4
2		15	44.4	14.96	24.31	34.35	28.3
3		15	46.3	15.71	25.8	36.21	27.2
4		15	48.62	16.5	26.84	38.28	26.2
5		15	51	17.32	27.9	40.42	25.1
6		15	53.6	18.18	29.1	42.7	24.1
7		15	56.6	19.1	30.3	45.1	23.1
8		15	59.1	20	31.52	47.58	22.2
9		15	62.1	21	32.86	50.24	21.3
10		15	65	22.1	34.2	52.4	20.4

TOTAL

247.3

Motor

Considering cost of electricity = Rs 5 /Kwh
Power of the motor = 410 KW

Operating cost

Electricity cost = Rs 410 * 24 * 5 * 365

= Rs 180 lakh

Cost of labour

Considering one person = Rs 1000 * 365

= Rs 365000

Total cost = 180 + 4 = Rs 184 lakh

Maintanece Cost

There is no maintanece cost attached with the motor.

Calculation of Net Present Value considering 5 % escalation every year

(IN LAKH)

No. of years	Capital Cost	Depreciation 10%	Operating Cost	Maintanece Cost	Tax 33.5%	Outflow	NPV
0	120	12	–	–	–	–	–
1		12	184	0	65.7	18.3	107.5
2		12	193.2	0	70.62	125	105.6
3		12	203	0	72	131	98.4
4		12	213	0	75.4	137.6	94
5		12	223	0	78.7	144.3	89.6
6		12	235	0	82.7	152.3	86
7		12	246	0	86.4	159.6	82
8		12	259	0	90.8	168.2	78.4
9		12	271	0	94.8	176.2	74.7
10		12	285	0	99.5	185.5	71.5

TOTAL

887.7

LNG pump selection

Q = 8.2 cu m/min
= 8.2 * 10^3 lit/min
= 51 bbl/min = 2216 gpm

Specific speed Ns = rpm*(gpm)^0.5/(H)^0.75

= 3600*(2216)^0.5/(726)^0.75

= 1250

Thus from the graph we have 6 vanes at 25 degree discharge angle.

Power P = pgQh/1000 = 480*9.81*0.14*240/1000 = 180 kw

D2 = 1840*1.075*(726)^0.5/3600 = 15"

Km2 = 0.125
Cm2 = Km2 * (2gH)^0.5
= 0.125 *(2*9.81*746)^0.5
= 15.1 ft/sec

Constant b = gpm*.0321/(Cm2*(D283.14 – Z *Su)
From graph no. of blades Z = 6
Su = 0.5"
Constant b2 = 2216*0.321/8.2*(15*3.14 – (6*0.5))
= 2"

Eye diameter

D1/D2 = 0.47
D1 = 15*0.47 = 7.05"

From tables shaft diameter = 3"

Eye area = (38.46)^2 – (7)^2 = 31 in square

Cm1 = gpm * 0.321/area = 2216*0.321/31 = 23ft/sec

Ut = dia * rpm /229 = 7*3600/229 = 110 ft/sec

From the figure NPSH required = 42 feet.

Thus specific speed Nss = 3600 *(2216)^0.5/(42)^0.75 = 10272

LNG pipeline

P1 = 6 barg
P2 = ?
L = 2.5 km
Q = 10000 cu m/ hr
D = 30"

Hf = fLV^2/(2gD)
= 0.032 * 2.5 * 1000 * (6.17)^2/(2 * 9.81 *0.76^1.33)
= 204

Applying Bernoulli's equation and assuming no elevation and velocity change.

7 *10 ^5/ (480*9.81) - P2 /(480* 9.81) = 204

P2 = 2.6 *10^5 pascals

LNG is generally loaded from the top.

Z = pgH
= 480 * 9.81 * 34
= 1.6 * 10^5 pascals

Thus pressure at the top is 1 barg

Heat Exchanger Design

Shell and tube heat exchanger design

Total Flow Q = 21 mmscmd
7 exchangers
Q = 3.0 mmscmd
Sp gr = .065
T1 = 200 deg R
T2 = 420 deg R

N = q / [U * A * LMTD * L]

Heat duty (q)

T1 = 200 deg R   P1 = 1473 psia
T2 = 420 deg R   P2 = 1400psia

Ppc = 667.8 psia   Tpc = 343.37 deg R

Ppr = 1473/667.8 = 2.2   Tpr = 200/343.37 = 0.58

Heat duty q = 41.7 * Tdiff * Cg * Q

Cg = 2.64 * (29 * Sp. Gr. * C + (Cp1 – Cp2)

C = 0.39 Btu/lb deg F

Cp1 – Cp2 = 40

Cg = 2.64 * (29 *0.65 *0.39 + 40)

= 125

Heat duty q = 41.7 * 220*125*3*36.1

= 124.2 *10^6 Btu/hr.

Gas   Water

T1 = -260 deg F   T1 = 203 deg F
T2 = -40 deg F   T2 = 77 deg F

LMTD = (436 -37)/ ln (463/37) = 426/2.53 = 163

Correction factor = 0.95

Thus corrected LMTD = 161

Tube size = 5/8" 16 BWG

U = 100 Btu/hr ft^2 deg F

Assume length = 60 feet

No. of tubes = 124.6 * 10^6/(100*0.2618*161*60)

= 490 tubes

Area = 124.6 *10^6/(6.2 *161*134) = 928 sq feet.

Water requirement calculation

Qw = q /(14.6*Tdiff)

= 124.6*10^6 /(14.6*126)

= 67459 bwpd

Qw per tube = 67459*2/1630 = 82 bwpd

<u>Air Exchanger design</u>

Q = 20 MMSCMD
= 722 MMSCFD
Sp gr = 0.65
T1 = -40 deg F
T2 = 60 deg F

Number of tubes (N)

N = q /(U*A*LMTD*L)

T1 = -40 deg F P1 = 1400 psia
T2 = 60 deg F P2 = 1350 psia

Ppc = 667.8 psia Tpc = 343.37 psia

Ppc =1400/667.8 = 2.2

Tpc = 420/343.37 = 1.4

Heat duty q = 41.7 * Tdiff * Cg * Q

Cg = 2.64 * (29 * Sp. Gr. * C + (Cp1 – Cp2)

C = 0.39 Btu/lb deg F

Cp1 – Cp2 = 40

Cg = 2.64 * (29 *0.65 *0.39 + 4)

= 30

Heat duty q = 41.7 * 100*30*722

= 90 *10^6 Btu/hr.

Gas	Water
T1 = -40 deg F	T1 = 95 deg F
T2 = 60 deg F	T2 = 70 deg F

LMTD = (135-10)/ ln (13.5) = 48

Correction factor = 0.95

Thus corrected LMTD = 46

Tube size = 5/8" 16 BWG

U = 70 Btu/hr ft^2 deg F

Assume length =100 feet

No. of tubes = 90 * 10^6/(70*0.2618*46*100)

= 1067 tubes

Plot Area = 90 * 10^6/(4*46*134)

= 15249.8 sq feet

Metering of gas

Diameter of the pipe = 24"
Diameter of the orifice = 8"
Differential water head dh = 60"
Upstream pressure Pf= 1000 psia
Flowing temperature Tf = 80 deg F
Base Temperature Tb= 60 deg F
Base Pressure Pb= 14.73 psisa
Discharge q = ?

Constant B = 8/24.065 = 0.3324

Average (hw*Pf)^0.5 = (60*1000)^0.5 = 245

Value of the fraction hw/Pf = 60/1000 = 0.06
Value of constant Fb = 14035.3 from tables
Constant b = 0.0134
Constant Fr = 1 + (0.0134/245) = 1.00005
Y1 = 0.993
Ttb = 540/520 = 1.038
Ftf = (540/510)^0.5 = 1.03
Fg = (1/0.65)^0.5 = 1.24
Z = 0.98
Fpr = (1/0.98)^0.5 = 1.01

K = 14035.3*1.00005*0.993*1.038*1.03*1.24*1.01
= 18662.5

Discharge q = K * (hw*Pf)^0.5

= 18662.5*245 = 110 MMSCFD = 3 MMSCMD

Total flow of gas = 20 MMSCMD

Thus we need 7 pipes of 24" diameter to measure the total flow.

CHAPTER 6

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