Snøhvit LNG

Rotating equipment

Theory and main boosting
Outline

- The LNG process
- Rotating equipment at Melkøya
- Basic theory (short)
- Compressors - examples
  - Refrigeration cycle compressors – Precooling compressor 25-KA101
  - Refrigeration cycle compressors – Liquefaction compressor 25-KA102
  - Refrigeration cycle compressors – Subcooling compressor 25-KA103
- Pumps - example
  - LNG export pumps 25-PA101 A/B
Rotating equipment at Melkøya

The rotating equipment consist of:

- **Gas turbines**
  - for power generation by converting fuel gas energy to electric power – local grid connected to Melkøya plant for backup (40 MW)

- **Compressors**
  - for pressurising gases (compressible fluids) by supplying external power

- **Pumps**
  - for pressurising liquids (incompressible fluids) by supplying external power

- **Turbines**
  - for producing power from expanding liquids or gases by converting pressure energy to work
Rotating equipment at Melkøya

*Gas turbines and main compressors*

<table>
<thead>
<tr>
<th>Gas Turbogenerators and Compressors</th>
<th>Power each [MW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>81-XY-102/201/301/401/501 Gas Turbogenerators</td>
<td>45.83</td>
</tr>
<tr>
<td>25-KA-101 Precooling Cycle Compressor</td>
<td>65</td>
</tr>
<tr>
<td>25-KA-102 Liquefaction Cycle Compressor</td>
<td>32</td>
</tr>
<tr>
<td>25-KA-103 Subcooling Cycle Compressor</td>
<td>65</td>
</tr>
<tr>
<td>27-KA-101 N2/CH4 Compressor</td>
<td>16.4</td>
</tr>
<tr>
<td>24-KA-101 CO2 Compressor</td>
<td>12</td>
</tr>
<tr>
<td>20-KA-101 Stabilizer OVHD Compressor</td>
<td>8</td>
</tr>
<tr>
<td>23-KA-101 Regeneration Gas Blower</td>
<td>0.35</td>
</tr>
<tr>
<td>60-KA-101 A/B Air Separation Unit (ASU) Air Compressors</td>
<td>2.4</td>
</tr>
</tbody>
</table>
### Rotating equipment at Melkøya

**Main Pumps and Hydraulic Turbines**

<table>
<thead>
<tr>
<th>Pumps and Hydraulic Turbines (Power &gt;200 kW)</th>
<th>Power each [kW]</th>
</tr>
</thead>
<tbody>
<tr>
<td>22-PA-101 A/B Main MDEA Pumps</td>
<td>2900</td>
</tr>
<tr>
<td>22-PA-102 A/B MDEA Booster Pumps</td>
<td>410</td>
</tr>
<tr>
<td>24-PB-101 A/B CO2 Export Pumps</td>
<td>710</td>
</tr>
<tr>
<td>25-PA-101 A/B LNG Export Pumps</td>
<td>205</td>
</tr>
<tr>
<td>50-PA-101 A/B/C Hot Oil Pumps</td>
<td>800</td>
</tr>
<tr>
<td>54-PA-101 HC Disposal Pump</td>
<td>370</td>
</tr>
<tr>
<td>54-PA-102 HC Booster Pump</td>
<td>1620</td>
</tr>
<tr>
<td>55-PS-101 A/B/C/D/E Sea Water Cooling Pumps</td>
<td>2700</td>
</tr>
<tr>
<td>56-PA-101 A/B Tempered Cooling Water Pumps</td>
<td>710</td>
</tr>
<tr>
<td>58-PA-101 A/B Tempered Heating Water Pumps</td>
<td>230</td>
</tr>
<tr>
<td>71-PS-102/103 Fire Fighting Pumps</td>
<td>1480</td>
</tr>
<tr>
<td>22-CT-101 MDEA Hydraulic Turbine</td>
<td>1027</td>
</tr>
<tr>
<td>25-CT-101 Subcooling Cycle Liquid Expansion Turbine</td>
<td>852</td>
</tr>
<tr>
<td>25-CT-102 LNG Expansion Turbine</td>
<td>1430</td>
</tr>
</tbody>
</table>

**General:** Approximately 200 Pumps and Hydraulic Turbines.
Rotodynamic compressors – Technology selection map according to GE Nuovo Pignone (Vendor for Snøhvit)

<table>
<thead>
<tr>
<th>Precooling</th>
<th>Stage</th>
<th>Inlet flow (m³/h)</th>
<th>Discharge pressure (bara)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
<td>122000</td>
<td>7.6</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>138000</td>
<td>21.4</td>
</tr>
<tr>
<td>Liquefaction</td>
<td>1</td>
<td>125000</td>
<td>19.7</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>137000</td>
<td>18.8</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>25000</td>
<td>54.8</td>
</tr>
<tr>
<td>Subcooling</td>
<td>1</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>2</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Basic concepts – Pump classification chart

Head (m) is given by:

\[ \text{Head} = \frac{P_{\text{pol}}}{m \cdot g} \quad \text{(m)} \]

\( g \) is 9.81 m/s\(^2\)

Pumps classification:

• Head:
  • Low pressure
    Head < 20 m
  • Medium pressure
    20 m < Head < 60 m
  • High pressure
    H > 60 m

• Number of stages:
  • Single stage
  • Multi stage

• Flow direction:
  • Radial
  • Axial
  • Cone
Basic thermodynamics and working fluids

**Work:**
In a stationary process where work (i.e. energy) is either produced or supplied, the following ideal relationship applies (power):

\[ \int p dp = \int v dp \]  

This relationship is valid for all stationary work processes and the specific work is equal to the area to the left of the curve from 1 to 2 in the figure above.

To be able to calculate the compression work we need a connection between \( p \) and \( v \). This is given by the polytropic relation:

\[ p \cdot v^n = \text{constant} \]

where \( n \) is the polytropic coefficient and is assumed constant.

Inserting this relation produces the polytropic work (power):

\[ P_{pol} = \dot{m} \int_{p_1}^{p_2} v dp = \frac{n}{n-1} \cdot \dot{m} \cdot [p_2 \cdot v_2 - p_1 \cdot v_1] \]

This relationship is valid for all stationary work processes and the specific work is equal to the area to the left of the curve from 1 to 2 in the figure above.

A special case applies to liquids as these fluids can be regarded as incompressible, i.e. the specific volume is constant (\( n \) is \( \pm \infty \)):
Basic thermodynamics and working fluids

Work, alternative formulation:

The first law of thermodynamics can be used to obtain an alternative expression for the work required for boosting or expansion:

\[
\frac{dE}{dt} = \sum_{in} \dot{Q}_{in} - \sum_{out} \dot{W}_{out} + \sum_{in} \dot{m}_{in} (h + 1/2u^2 + gz)_{in} - \sum_{out} \dot{m}_{out} (h + 1/2u^2 + gz)_{out}
\] [W]

This reduces to the following assuming stationary process with negligible heat loss (adiabatic) and negligible contribution of potential and kinetic energy changes. The mass balance is given as well:

\[
\frac{dE}{dt} = \sum_{in} \dot{Q}_{in} - \sum_{out} \dot{W}_{out} + \sum_{in} \dot{m}_{in} (h + 1/2u^2 + gz)_{in} - \sum_{out} \dot{m}_{out} (h + 1/2u^2 + gz)_{out}
\] [W]

This gives:

\[
\dot{W} = \dot{m} \cdot (h_2 - h_1)
\] [W]

i.e. the p-h diagram can be used directly to obtain the work required by a compressor or delivered by an expander provided suction and discharge states (p, T) are given.

Here the following mass balance has been utilised (one inlet and one outlet):

\[
\dot{m} = \dot{m}_{in} = \dot{m}_{out} [kg / s]
\]
Basic thermodynamics and working fluids

Example: The liquefaction compressor at Melkøya have suction and discharge states as given in the table below. What is the power input to the compressor when the mass flow is 437093 kg/h? The polytropic coefficient $n$ is equal to 1.2345.

<table>
<thead>
<tr>
<th>Suction</th>
<th>Discharge</th>
</tr>
</thead>
<tbody>
<tr>
<td>$p$ [bar]</td>
<td>2.029</td>
</tr>
<tr>
<td>$T$ [$^\circ$C]</td>
<td>-57</td>
</tr>
<tr>
<td>$v$ [m$^3$/kg]</td>
<td>0.2842</td>
</tr>
<tr>
<td>$Z$ [-]</td>
<td>0.9611</td>
</tr>
<tr>
<td>$h$ [kJ/kg]</td>
<td>-91.9372</td>
</tr>
</tbody>
</table>

The compression process is illustrated in the p-H diagram below:
Basic thermodynamics and working fluids

Work continued:

The work required for this compressor can now be evaluated:

Alternative 1: Using:

$$P_{pol} = \frac{n}{n-1} \cdot \dot{m} \cdot [p_2 \cdot v_2 - p_1 \cdot v_1]$$

We get:

$$P_{pol} = \frac{1.2345}{1.2345 - 1} \cdot \frac{437093}{3600} \cdot [19.85 \cdot 0.04482 - 2.029 \cdot 0.2842] \cdot 10^5 = 20 \cdot 10^6 \text{ W} = 20 \text{ MW}$$

Alternative 2: Using:

$$P = \dot{m} \cdot [h_2 - h_1]$$

We get:

$$P = \frac{437093}{3600} \cdot [114.6 - (-91.94)] = 25.08 \cdot 10^6 \text{ W} = 25.08 \text{ MW}$$

Why is the results using calculation alternative 1 and 2 different from using calculation alternative 3?

• Alternative 2 gives the correct work to be applied because the efficiency of the compression process is included in the enthalpy.
• Alternative 1 are based on model assuming the compression to be ideal, i.e. we have to correct the values using the efficiency of the compression process.
• Efficiency definitions will now be treated
Basic concepts – **Loss mechanisms and efficiencies**

Loss mechanisms within the machine can be divided in:
- **Hydraulic losses** – friction between flowing fluid and walls, acceleration / retardation and deflection
- **Leakage losses** – due to clearances between rotating and static parts of the machine

These losses constitute the inner losses in the machine.

The ideal work $P_{pol}$ must be corrected for the inner losses. This is obtained by introducing the polytropic efficiency $\eta_p$:

$$P = \frac{P_{pol}}{\eta_p} \quad \text{Compressor / pump}$$

$$P = \eta_p \cdot P_{pol} \quad \text{Turbine}$$

In the examples given earlier the work obtained by correcting for the polytropic efficiency is 25 MW when the given value of $\eta_p$ is introduced (0.81).

To arrive at the shaft power required or produced, the following losses also has to be included:
- **Mechanical losses in the compressor, pump and expander** ($\eta_m$)
- **Gear losses** ($\eta_g$)

$$P_{shaft} = \frac{P}{\eta_m \cdot \eta_g} \quad \text{Compressor / pump}$$

$$P_{shaft} = \eta_m \cdot \eta_g \cdot P \quad \text{Turbine}$$
Compressors and pumps need a drive for supplying the power needed for compression. There are basically three different possible drives:

- **Electric motor (with frequency converter)** – this drive is often connected to the grid to have backup in case any local power generation is shut down.
- **Gas turbine** – utilises local supply of fuel gas and can also, applying proper design, produce steam for process heat or expansion
- **Steam turbine** – provided steam is available

All these drives have variable speed possibilities enabling fast and efficient adjustment of speed to absorb load variations

**Gear:**

- A gear is needed provided the speed of the drive and the compressor is not the same
- This is the normal situation for compressors, the drive has a lower speed than the compressor unless the compressors are big.
- The gear needs lube oil for bearings and / or seals.

**Coupling:**

- A coupling is needed between the drive and the compressor / pump as they are not usually on the same shaft.
- If a gear is present, a low speed coupling is needed between the drive and the gear and a high speed coupling is needed between the gear and the compressor / pump.
- The coupling is to transmit torque, compensate for misalignment and absorb axial displacement
- Couplings can be dry or oil lubricated
Compressor system considerations – **Scrubber and cooler**

A scrubber is utilised upstream of compressors to:
- Remove liquid droplets from the gas (compressor designed for dry gas)
- Protect the compressor from liquid slugs entering the compressor

The scrubber inlet, internals and outlet are elements that are designed to facilitate separation between liquid and gas and differ between scrubber designs.

A cooler is needed to remove heat from the gas after compression due to:
- **material limitations and / or**
- **process requirements**

For instance, if the gas after the compressor in the example shown earlier are to be cooled to 0°C, 20 MW of heat would have to be removed.
Compressor system considerations – Anti surge system

Surge: If flow is reduced at constant speed, backflow will eventually be experienced because the compressor suddenly loses its ability to produce the pressure existing in the discharge piping. Surge is an unstable flow situation than can cause severe damage to the compressor.

Surge is avoided by having a:
• Anti-surge system and, if needed, a
• Trip protection system

In addition to:
• Piping and
• Control system using flow, pressure, temperature and speed information

The Anti-surge system includes:
• Regulating valve (opening time on trip < 1s)

The trip protection system includes:
• On / off valve – fast acting
  • Cold gas bypass (CGB) – Kårstø (KUP): 0.26 s between trip initiation and 100% open valve
  • Hot gas bypass (HGB) – Kollsnes: 0.4 s between trip initiation and 80% open valve
Pump system considerations – Cavitation

Cavitation applies to the suction side of the pump and restricts the height difference between the pump inlet and the liquid level. Increased elevation decreases the suction pressure:
• At a certain elevation this pressure will become lower than the vapour pressure of the liquid
• This will lead to evaporation and inherent vapour bubble formation
• The vapour bubbles follow the liquid stream
• The pressure increases in the flowing direction and the bubbles collapse when the liquid pressure is higher than the vapour pressure

The effect of this cavitation is erosion due to the impact caused by the collapsing bubbles close to the wall. The impact of these shocks will exceed the ultimate strength of the wall material and this causes wear.

Factors increasing the cavitation risk are:
• Increasing liquid temperature
• Increased pressure losses in the suction piping

This imposes restrictions on the suction height and will, at high temperature or high pressure loss, cause the suction height to be negative.

A parameter specified for pumps considering cavitation is the Net Positive Suction Head – NPSH which is the net positive head above the vapour pressure. This parameter depends on the pump design and will increase with volume flow.
Rotating machinery design considerations - **Seals**

Seals are placed between rotating (moving) and stationary parts. The purpose of the seal is to minimise the leakage between high and low pressure areas.

Typical seal configurations are:

- **Labyrinth seals** – utilised where some leakage is acceptable / low pressure
  - Interstage seals in rotodynamic machines
- **Oil film seals** – utilised where no gas leakage is tolerated and the speed is high
  - Small oil leakage to process gas and surroundings. Dominating seal type in compressors before 1990
  - Either separate oil system or common oil system with bearings
  - Oil must be degassed and cleaned (viscosity, flame point, corrosion)
- **Mechanical seal** – oil leakage to process fluid must be kept at a minimum
  - Carbon based rings used as seal rings – limited lifetime due to wear
- **Dry mechanical seals (dry gas seals)** – no oil, low leakage, no wear
  - Seal elements “floating” on gas film between spring loaded stationary ring and rotating ring with microscopic grooves. The gas film is created by these tiny grooves.
  - Radial leakage flow between stationary ring and rotating ring (controlled) to obtain cooling
  - Axial distance between rings 2 – 10 μm
  - Very small leakage compared to labyrinths
  - Leakage flow is measured and vented or recompressed
  - Rotating ring is ceramic and stationary ring might be carbon
Rotating machinery design considerations – Illustration of seals

Possible seal arrangement in compressor:

- Between rotor / impellers and stationary parts labyrinth seals are used. See figure for one possible design:
- Between surroundings and rotor, dry gas seal is used. This seal is also separating the bearings from the process fluid.

Interstage labyrinths:
Rotating machinery design considerations - Bearings / axial load balancing piston

The axial and radial loads experienced in rotating machinery are dealt with by using axial and radial bearings. The loads are due to:

• The weight of the rotor and possible load due to vibrations caused by unbalance – taken by the radial bearings, unless it is a vertical machine. As an example; a rotor running at 5000 rpm having a centre of gravity 0.1 mm away from the centreline will be exposed to a radial load that is close to three times the load due to the weight.

• The pressure difference across the machine which causes an axial force directed towards the low pressure side of the machine – taken by the axial bearing. However, usually this bearing is not capable of absorbing all the axial thrust alone (only small or low pressure machines).

The axial balance piston / drum will take care of the axial thrust not absorbed in the axial bearing. Designed to take 70 to 80% of the axial force. Such a system could look like this:

The axial unbalance due to the pressure difference is illustrated by the arrows. B is the balance piston.
Rotating machinery design considerations - **Bearings**

Bearings can be of the following types:

- **Rolling** – load supported by rollers or balls
- **Hydrostatic** – load supported by high pressure fluid
- **Hydrodynamic** – load supported by lubricant film
- **Magnetic** – load supported by magnetic fields

Rolling bearings and hydrodynamic bearings dominates in large rotating machinery

Hydrodynamic bearings is used in the cooling compressors; the rubbing surface is a low friction metal (babbitt – tin or lead based alloys, copper-lead alloys, lead-bronze, etc) and it is lubricated by a film of oil. The capacity of hydrodynamic bearings are restricted by:

- **minimum oil film thickness** – low speed operation
- **babbitt temperature** – high speed operation
Rotating machinery design considerations – Lube oil system

A typical lube oil system consists of:
• pumps to obtain correct pressure (depending on bearing design)
• coolers to remove frictional heat (heater also to keep the viscosity at the right level at start up)
• filters to remove particles in the oil (this is very important to obtain high regularity and availability)
• degassing tank (if the system is common with a seal oil system – not shown in the figure)
Snøhvit – Refrigeration cycle compressors

• Precooling Cycle Compressor – Process flow diagram

Pressure (bara) | Temperature (°C) | Flow (10³ kg/h)
---|---|---
56 | 52 | 2.5
48 | 7.6 | -26.8
36 | 7.6 | 1.1
21 | 67.8 | 1164.7
21 | 31.5 | 20.3
44 | 48 | 10
40 | 40 | 40

AS valve 1. stage (surge control)
AS valve 2. stage (surge control)

Gas from:
• Cycle precooler in cold box (93.2%)
• Precooler in nitrogen remover cold box (1.7%)
• Refrigeration make up / fractionation system (5.1%)

Gas from:
• Cycle precooler in cold box (86.2%)
• Precooling cycle circulation drums (8.3%)
• Refrigeration make up / fractionation system (5.5%)

To flare

From make up header

Gas analyser connected to make up management system

Scrubber

After cooler 40 MW

Precooling condenser 17 MW

Liquid to precooling cycle refrigerant receiver

Sea water

Sea water

Variable speed drive

Variable speed drive

Variable speed drive

Variable speed drive
Snøhvit – Refrigeration cycle compressors

- 1 x Precooling Cycle Compressor – 25-KA-101
  - Vendor: GE Nuovo Pignone
  - Model: 3MCL1404 (Horizontally Split)
  - # of impellers: 4
  - Molecular Weight: 35.148 (Ethane/Propane)
  - Side stream (Suction/Side stream = 1/2)
  - Dry Gas Tandem Seals, Vendor: John Crane, Model 28 XP
    - Primary seal gas: Process gas
    - Secondary seal gas: \( N_2 \)
    - Separation gas: \( N_2 \)
  - Tilting pad bearings, Vendor: Kingsbury
    - Radial bearings (2) – Flooded lubrication
    - Thrust bearing – Directed lubrication
  - Each stage: AS control valve, opening time < 0.5s (ESD)
  - Hot gas bypass valve from 2. stage to 1. stage scrubber
  - Direct Drive: Electric VSDS/Siemens
  - Speed: 3600 RPM
  - Driver Rated Power: 65 MW
Snøhvit – Refrigeration cycle compressors size and weight

Precooling compressor 25-KA101

Weight compressor: 187 ton
Weight motor: 105 ton
Total weight skid: 334 ton

55 MW power consumption

Suction volute
Side-stream inlet
Discharge volute
Snøhvit – Refrigeration cycle compressors

- Liquefaction Cycle Compressor – Process flow diagram
Snøhvit – Refrigeration cycle compressors

- **1 x Liquefaction Cycle Compressor – 25-KA-102**
  - Vendor: GE Nuovo Pignone
  - Model: MCL1406 (Horizontally Split)
  - # of impellers: 6
  - Molecular Weight: 29.88 (Methane/Ethane/Propane)
  - Dry Gas Tandem Seals, Vendor: John Crane Model 28 XP
    - Primary seal gas: Process gas
    - Secondary seal gas: N₂
    - Separation gas: N₂
  - Tilting pad bearings, Vendor: Kingsbury
    - Radial bearings (2) – Flooded lubrication
    - Thrust bearing – Directed lubrication
  - AS control valve, opening time
    < 0.5s (ESD)
  - Hot gas bypass valve from discharge to scrubber
  - Direct Drive: Electric VSDS/Siemens
  - Speed: 3600 RPM
  - Driver Rated Power: 32 MW
Snøhvit – Refrigeration cycle compressors size and weight

Liquefaction compressor 25-KA102

Weight compressor: 133 ton
Weight motor: 58 ton
Total weight skid: 229 ton

26 MW power consumption

Suction volute
Discharge volute
Snøhvit – Refrigeration cycle compressors size and weight

Liquefaction compressor 25-KA102
Snøhvit – Refrigeration cycle compressors

- Subcooling Cycle Compressor - process flow diagram

Variable speed drive

34 MW

1. stage

Inter-cooler 14 MW

Sea water

18.8

61

30”

18.3

9.9

30”

54.8

102.3

2. stage

After cooler 29 MW

Sea water

18.3

9.9

Inter-cooler 14 MW

123 MW

Hot gas bypass valve 2. stage

(surge protection)

Recycle valve (start up)

Gas to cycle precooler in cold box and subcooling cycle refrigerant receiver in cold box

Gas from:
- Cycle precooler in cold box (94.4%)
- Main cooler in nitrogen remover cold box (5.6%)

Pressure (bara)  ⬜️
Temperature (°C)  ⬜️
Flow (10³ kg/h)  ⬜️

C1, C2 and N2 from make up header

To flare

Gas analyzer connected to make up header

Hot gas bypass valve 1. stage (surge protection)

AS valve 2. stage (surge control)
Snøhvit – Refrigeration cycle compressors

- **1 x Subcooling Cycle Compressor – 25-KA-103**
  - Vendor: GE Nuovo Pignone
  - Two Casings
    - Stage 1: MCL1406 (Horizontally Split) - 6 impellers
    - Stage 2: BCL1007 (Vertically Split) - 7 impellers
  - Molecular Weight: 22.22 (Methane/Ethane/Nitrogen)
  - Dry Gas Tandem Seals, Vendor: John Crane, Model 28 XP
    - Primary seal gas: Process gas
    - Secondary seal gas: \( \text{N}_2 \)
    - Separation gas: \( \text{N}_2 \)
  - Tilting pad bearings, Vendor: Kingsbury
    - Radial bearings (2+2) – Flooded lubrication
    - Thrust bearing (1+1) – Directed lubrication
  - AS control valve 2. stage, opening time < 0.5s (ESD)
  - Hot gas bypass valve both stages
  - Direct Drive: Electric VSDS/Siemens
  - Speed: 3600 RPM
  - Driver Rated Power: 65 MW
Snøhvit – Refrigeration cycle compressors size and weight

Subcooling compressor 25-KA103

Weight compressors: 249 ton
Weight motor: 105 ton
Total weight skid: 424 ton

33 + 22 MW power consumption

Suction volute
Discharge volute
Snøhvit – LNG export pumps

- 2 x LNG export pump 25-PA-101A/B
  - Vendor: Ebara
  - Model: 16ECC-24
  - # of impellers: 1

- Molecular weight: 17.39 (LNG)
- Ball bearings (2)
- Speed: 1500 rpm
- Driver shaft power: 300 kW
Snøhvit – LNG export pumps

Design:
• Submerged pumps – motor and pump are submerged in liquid inside a pressure vessel
• Orientation is vertical with motor on top
• Radial pump units with axial inflow
• Helical inducer in inlet – allows very low NPSH
• Cooling of motor and lubrication of bearings by a small portion of the liquid being pumped
• Axial load (thrust) handled by a specially designed thrust equalizing mechanism as shown below:
Snøhvit – LNG export pumps size and weight

Total weight: 8600 kg

Fill / drain: 25 mm pr minute, at least 3 hours to fill vessel