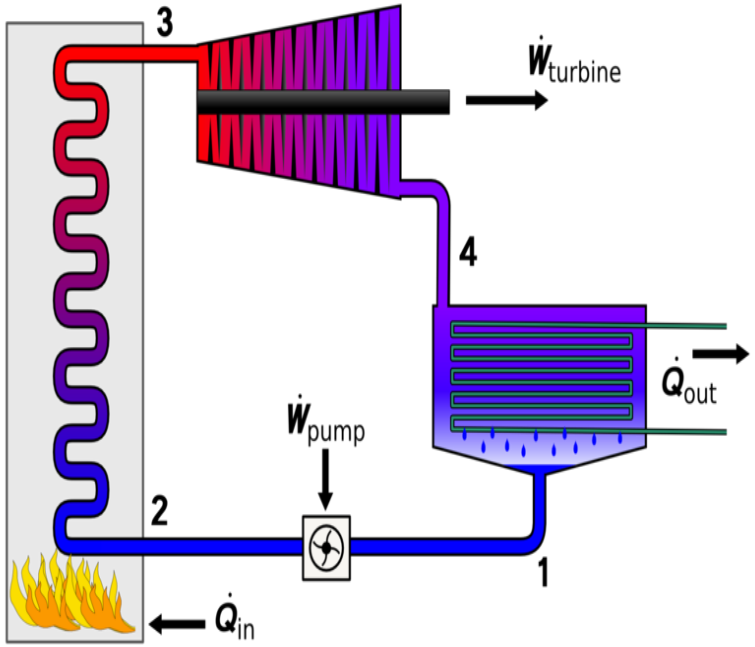
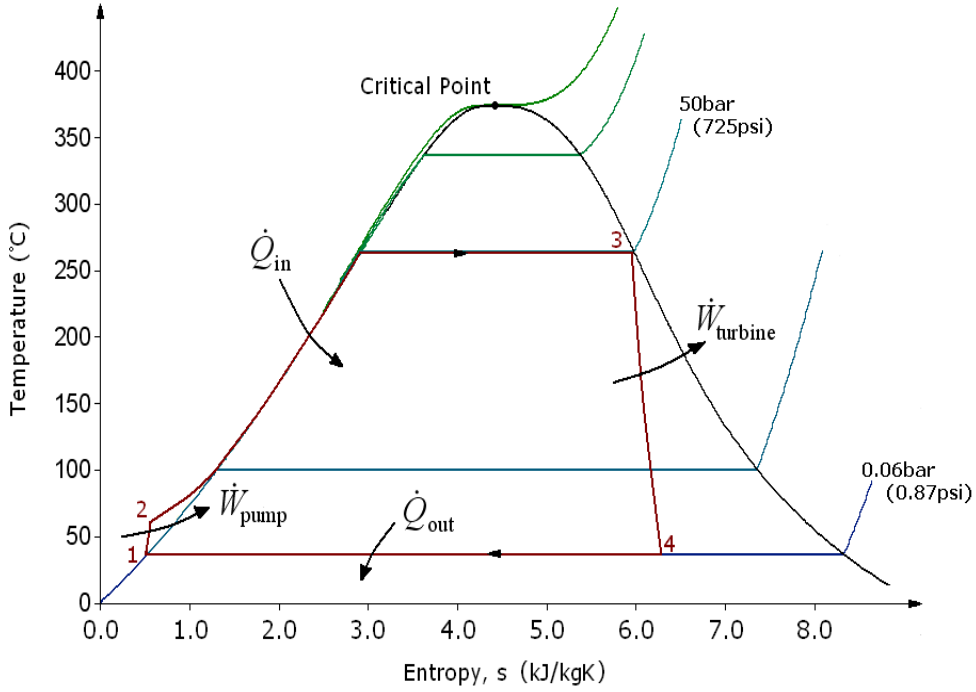
Upstream Process Engineering

Energy Efficiency – Part 2

Energy Recovery From Low Grade Heat – Rankine Cycle





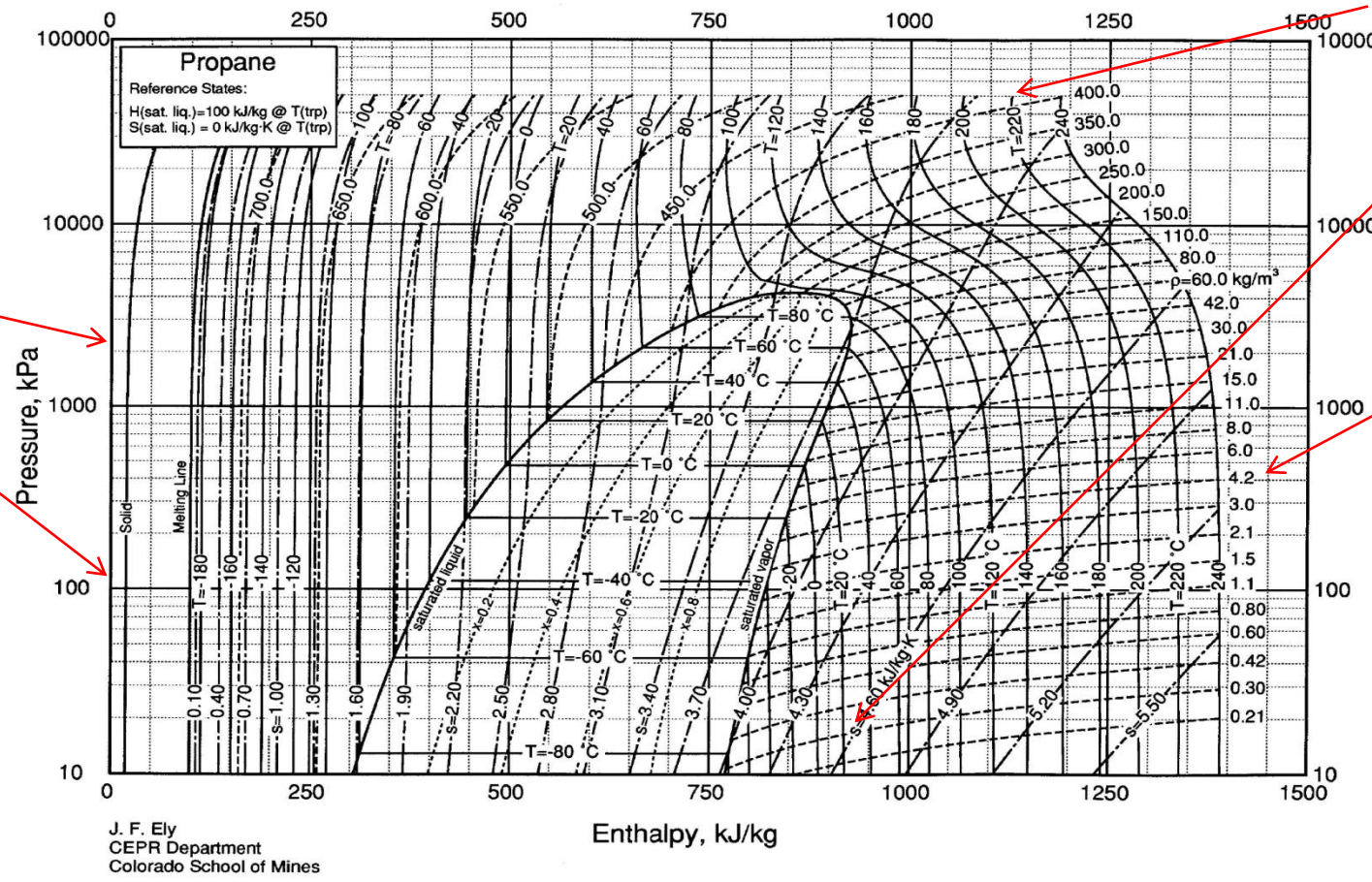
Heat is used to vapourise a liquid. The vapour is used to drive a turbine to extract work. The vapour is cooled to its liquid state then pumped to the vapouriser.

Thermodynamics/Phase Behaviour – Mollier Chart

Bubble Point, Saturation, Vapour Pressure at

60 DegC

-40 DegC

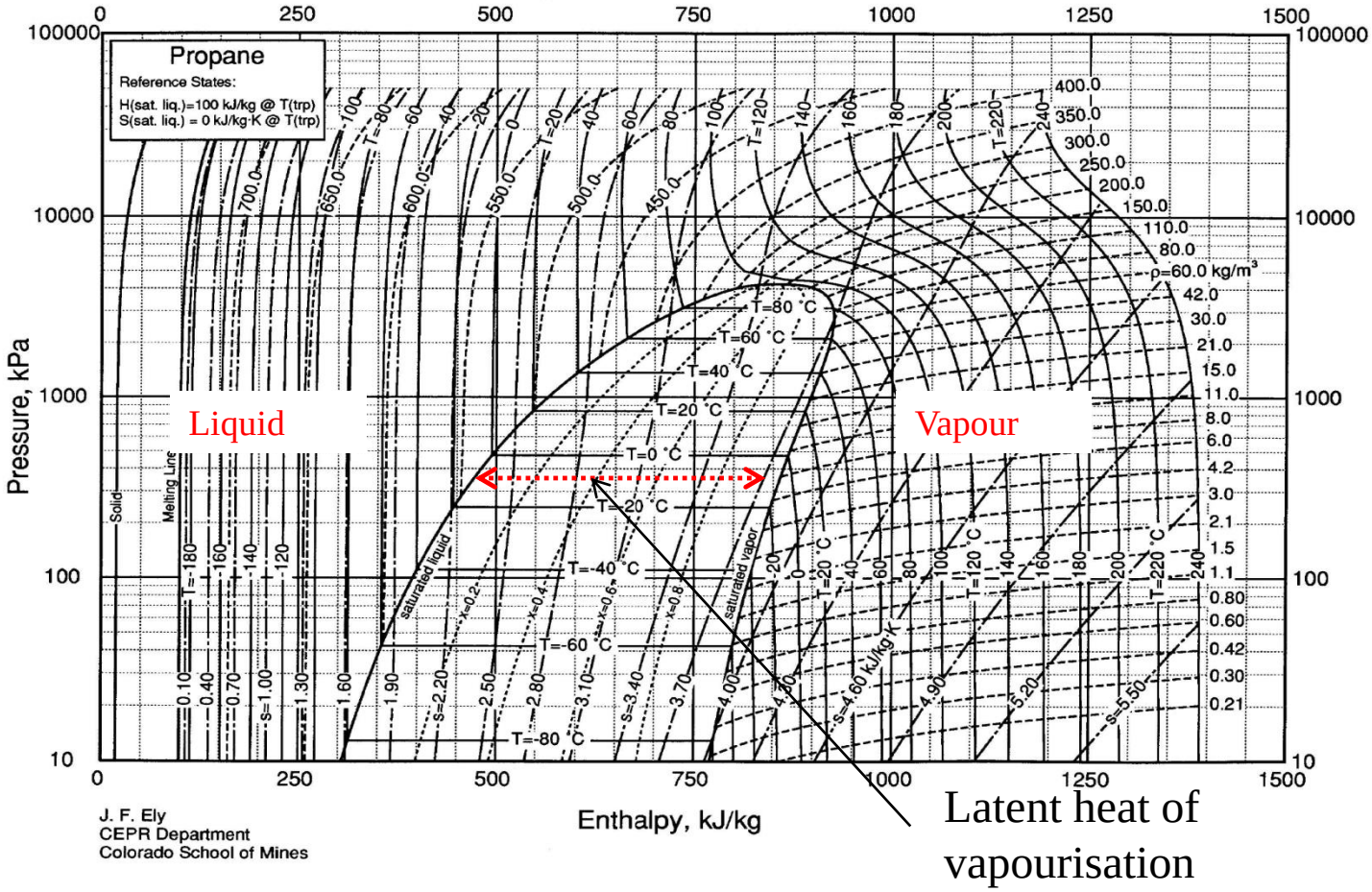


Isotherms

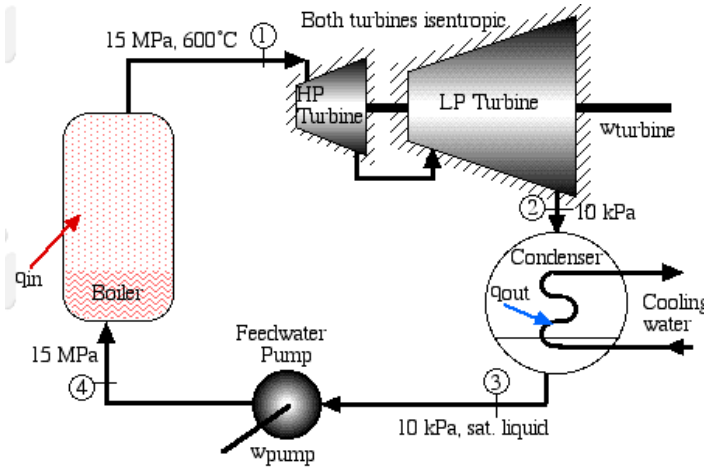
Constant Entropy

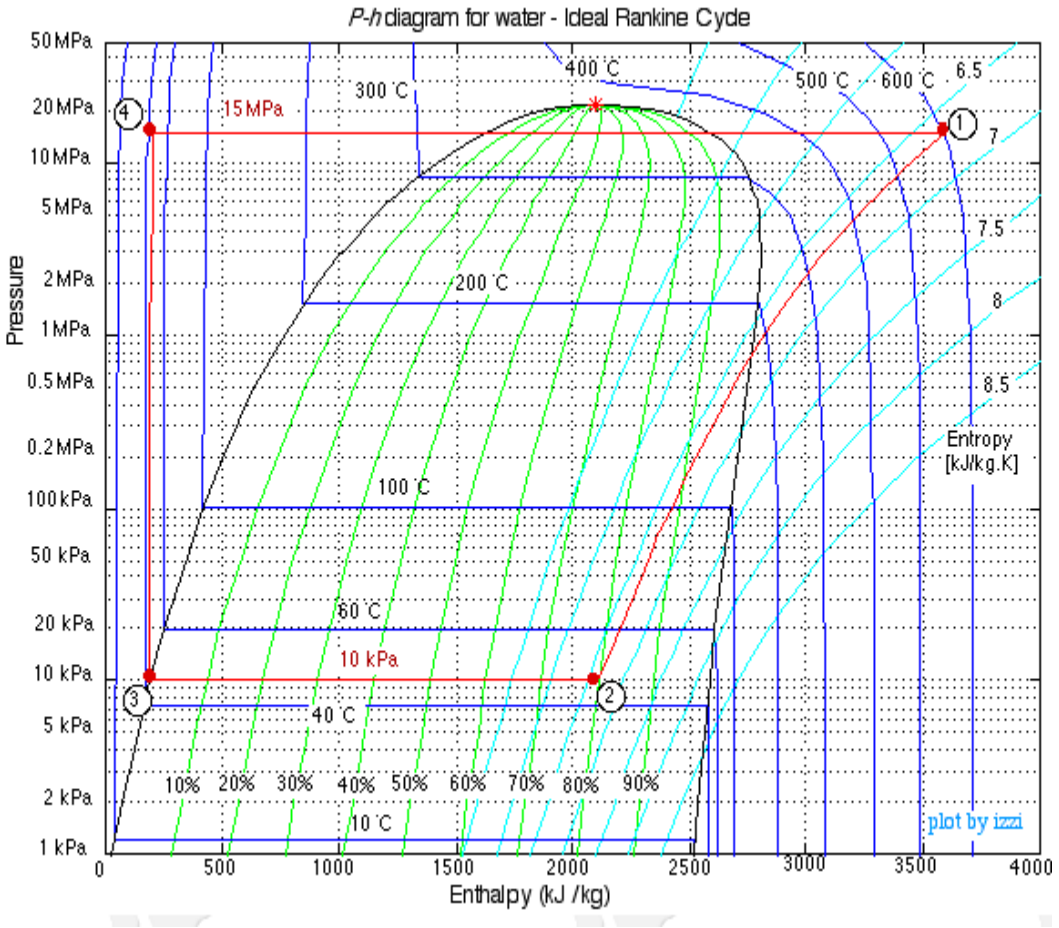
Density

Thermodynamics/Phase Behaviour – Mollier Chart



Rankine Cycle

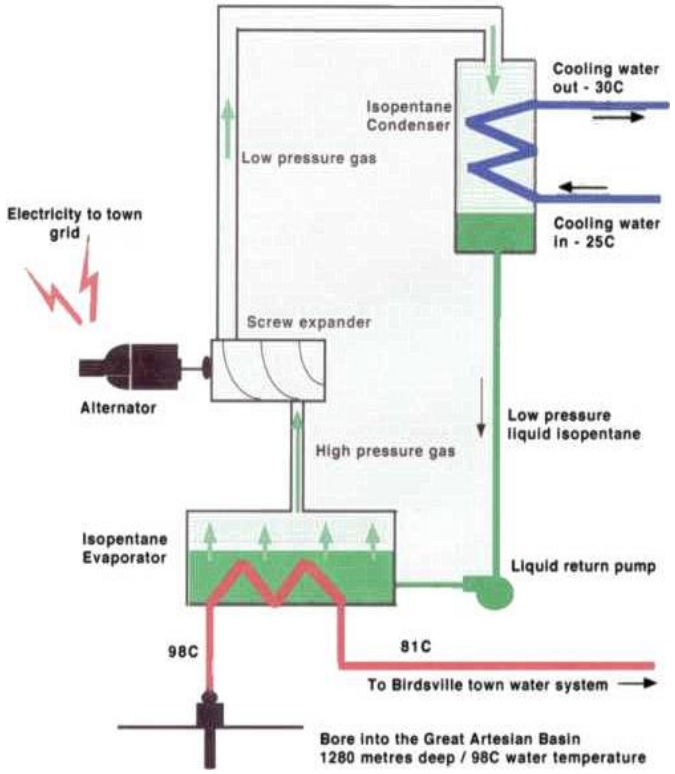




Organic Rankine Cycle

The Organic Rankine Cycle (ORC) uses a heat source to heat an organic motive fluid. The motive fluid is evaporated and is then expanded through a turbine, which generates power. The motive fluid is then condensed in a heat exchanger using a cooling medium (cooling water, seawater, air) from where it is pumped back to the evaporator. The overall net ORC cycle efficiency comparing extracted power to available thermal energy typically ranges from around 3% (with 90 ^{o}C source water) to 7% (120 ^{o}C source water) of the available thermal energy. Turbine efficiencies are around 80% and the electrical generator efficiency 90%.

This system is analogous to the steam/water Rankine cycle used in power stations to generate power; the main difference being that the motive fluid has to boil at a lower temperature than water. A schematic of a typical ORC is shown.



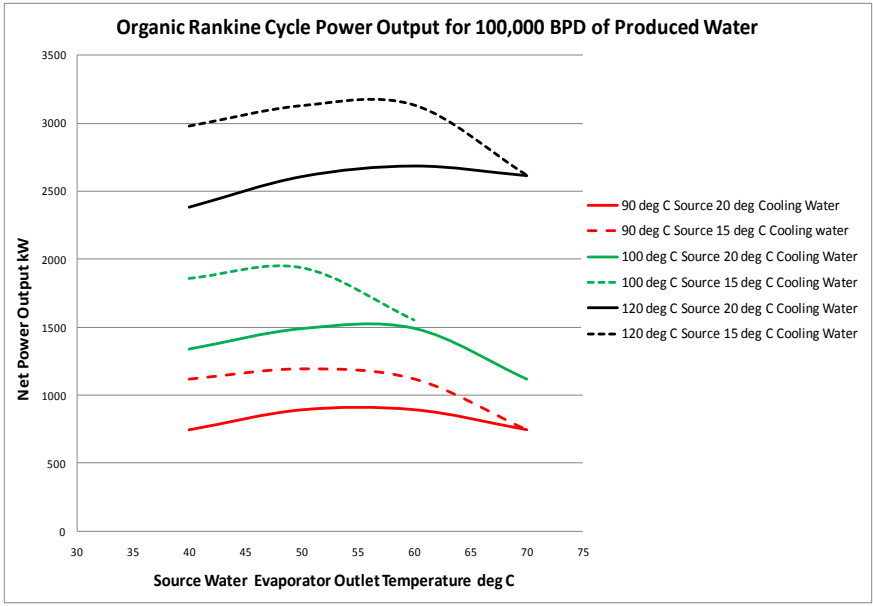
Organic Rankine Cycle Power Recovery

The power recovery in an ORC is dependent on a number of factors:

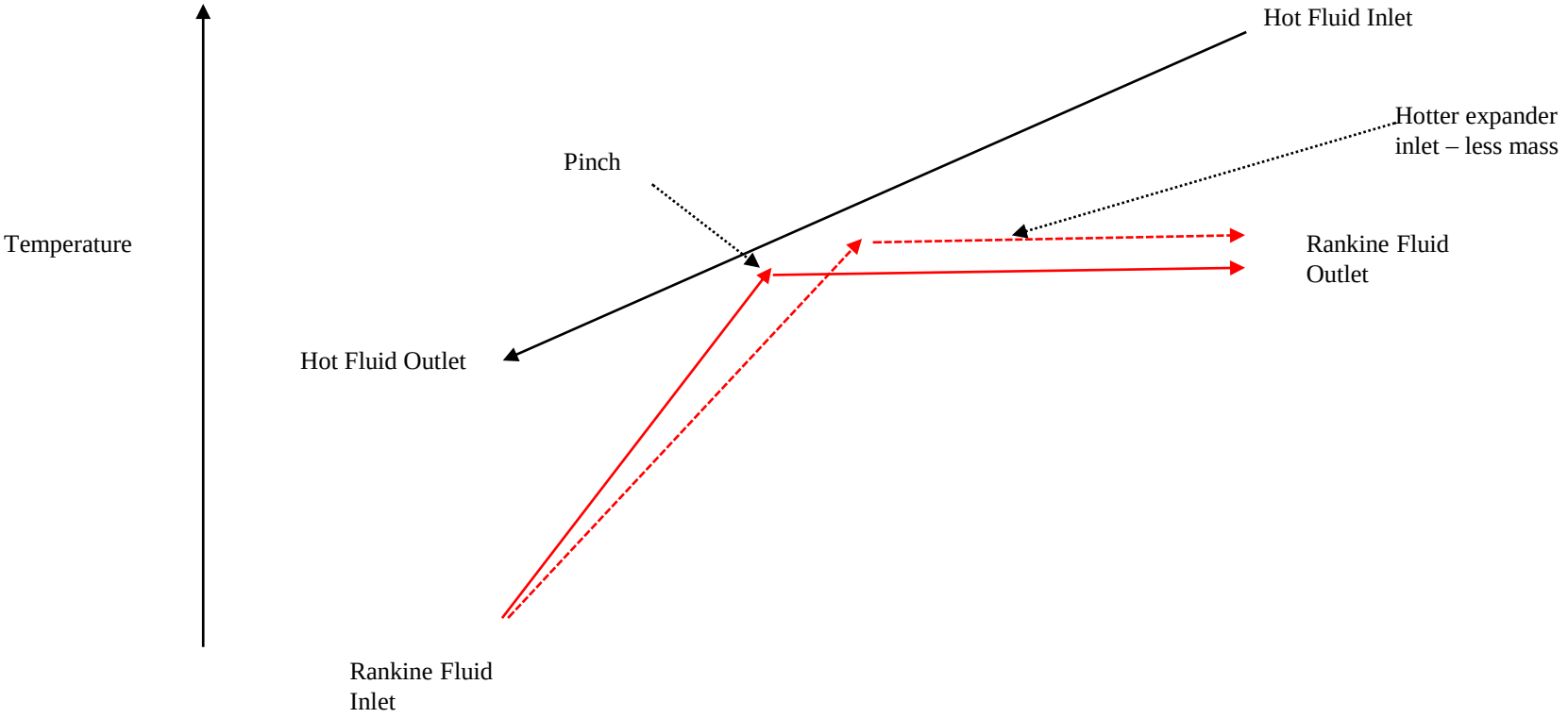
Motive fluid type
Temperature and flowrate of the source water
Temperature and flowrate of the cooling water
Outlet temperature of the source water from the evaporator

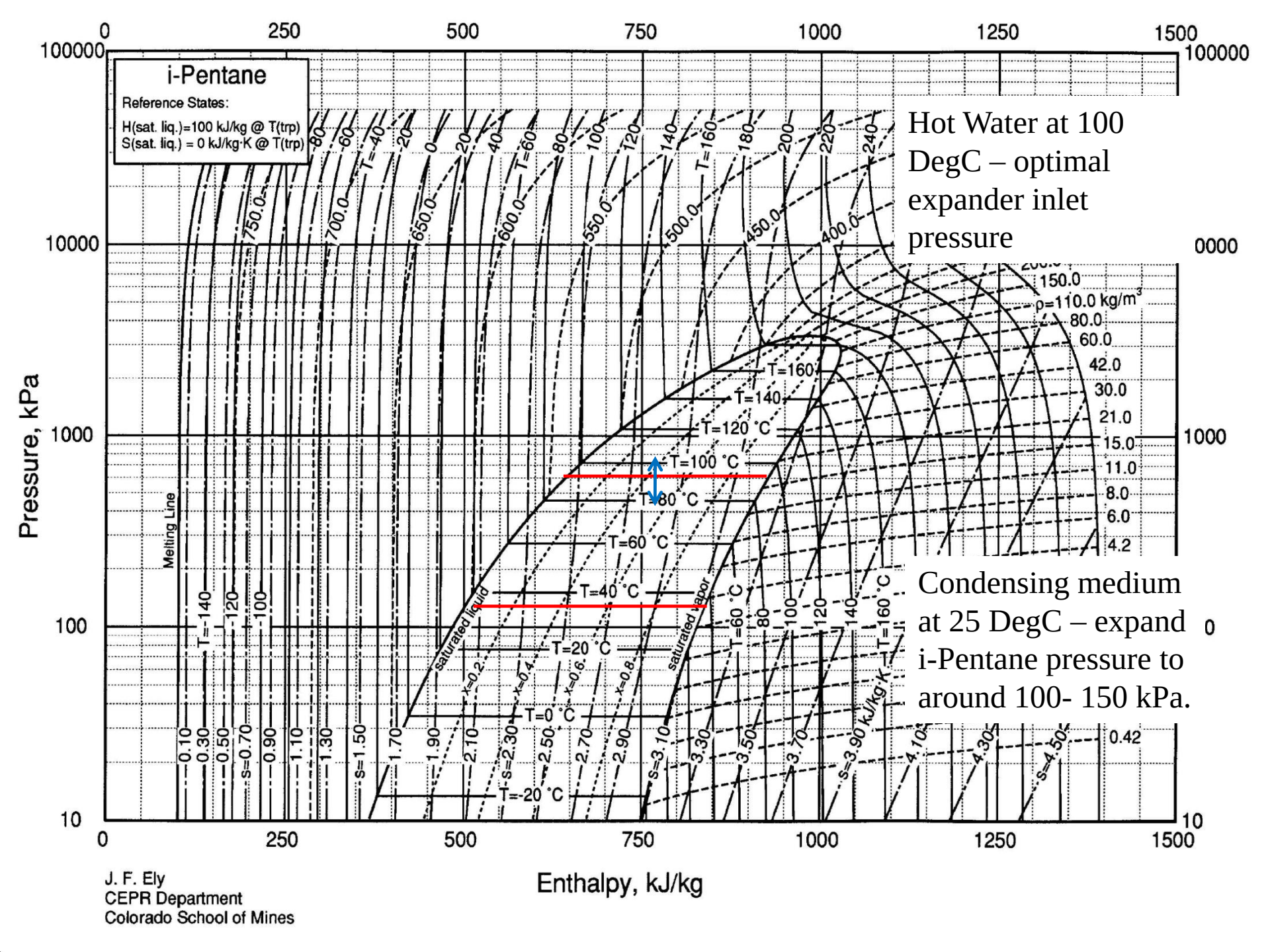
The influence of the above factors on power output is illustrated in the opposite figure which is for energy recovery from hot produced water.

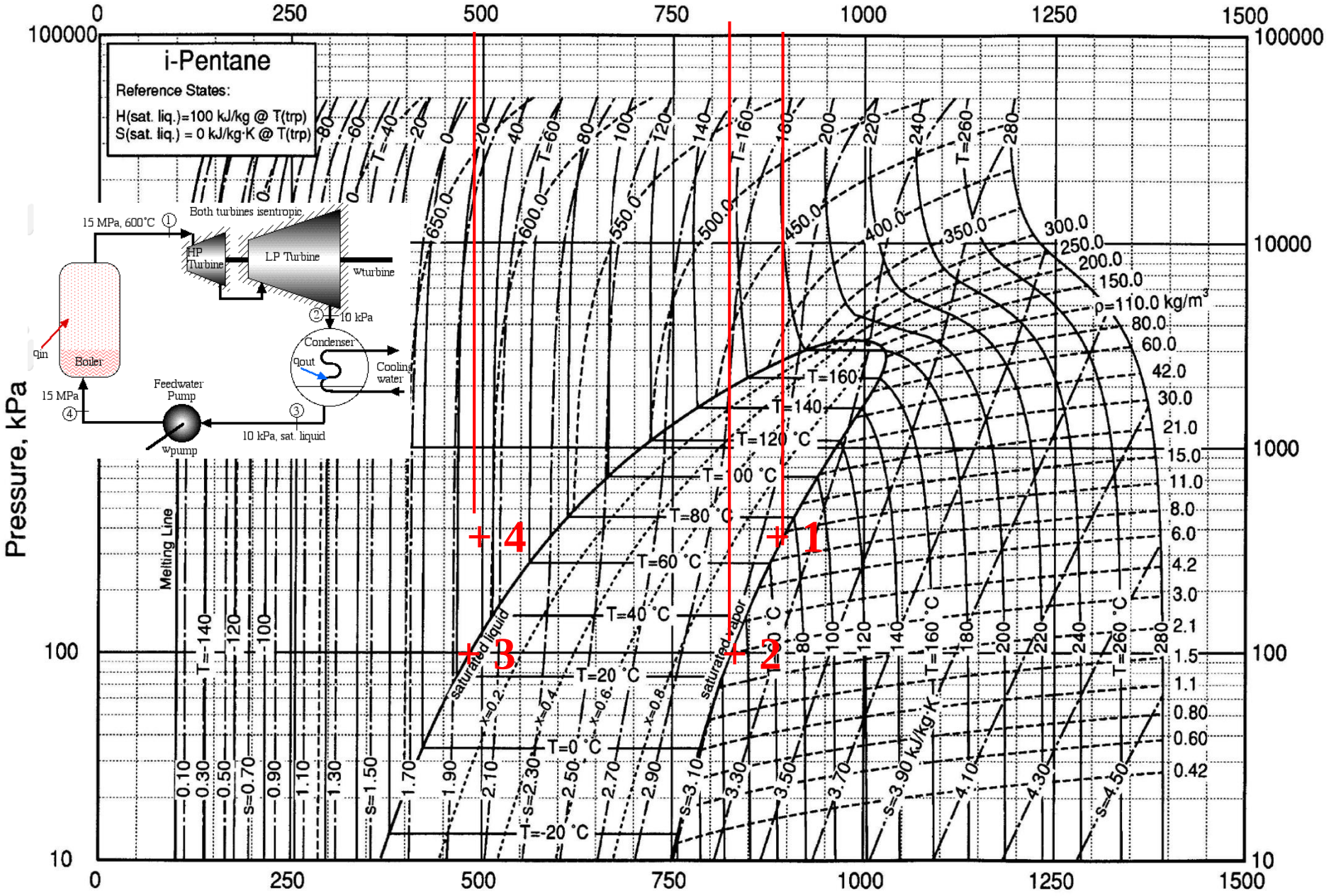
Expander/Turbine inlet pressure and temperature requires to be optimised. The more heat that is extracted from the water the lower the water outlet temperature. This will vapourise more ORC fluid but it will be at lower outlet pressure and temperature.



Organic Rankine Cycle Vapouriser/Evaporator




i-Pentane
Reference States:
H(sat. liq.)=100 kJ/kg @ T(trp)
S(sat. liq.) = 0 kJ/kg·K @ T(trp)
Hot Water at 100 DegC – optimal expander inlet pressure
Condensing medium at 25 DegC – expand i-Pentane pressure to around 100- 150 kPa.
Pressure, kPa
Enthalpy, kJ/kg
Melting Line
saturated liquid
saturated vapor
J. F. Ely
CEPR Department
Colorado School of Mines


i-Pentane
Reference States:
H(sat. liq.)=100 kJ/kg @ T(trp)
S(sat. liq.) = 0 kJ/kg·K @ T(trp)
Pressure, kPa
Enthalpy, kJ/kg
Melting Line
saturated liquid
saturated vapor
x=0.2
x=0.4
x=0.6
x=0.8
ρ=110.0 kg/m³
+1
+2
+3
+4
15 MPa, 600°C
Both turbines isentropic
HP Turbine
LP Turbine
Wturbine
10 kPa
Condenser
Cooling water
Boiler
Feedwater Pump
15 MPa
10 kPa, sat. liquid
Wpump
J. F. Ely
CEPR Department
Colorado School of Mines

Organic Rankine Cycle

Choice of Motive Fluid for Organic Rankine Cycle

The motive fluid for an ORC must have the following properties:

A favourable vapour pressure curve – it must be able to condense out at 25^0C and evaporate at a temperature of 60 - 100^0C. The difference in vapour pressures at these temperatures must be sufficient to drive a turbine.

Ozone-friendly and non-detrimental to the environment.

The vapour pressure at 25^0C must not be so low that the system operates under extreme vacuum. This would not only lead to the risk of air ingress into the unit but also require large diameter pipes and equipment.

Flammability, volatility and toxicity are also important.

ORC fluids are:

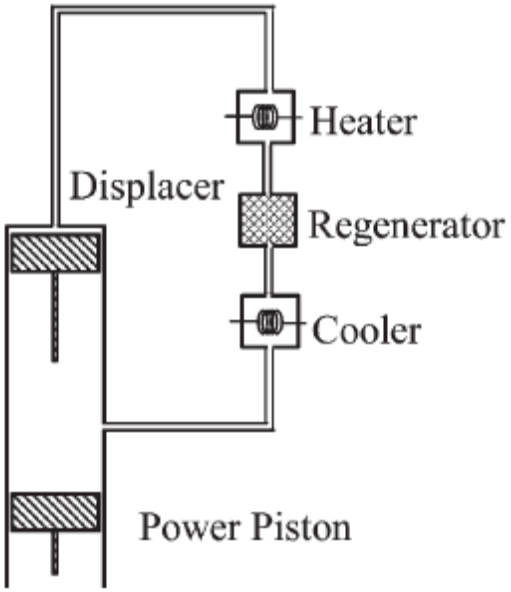
Refrigerant R134a – 1,1,1,2 tetrafluoroethane

Ammonia

Butane

Iso-pentane

Stirling Cycle



Achieving a high efficiency using the Stirling cycle is difficult in practice because it involves reversible heat transfer in all the components, including the regenerator. For small temperature differentials, such as geothermal water, a large heat exchange area is required. Stirling engines have generally been restricted to very hot sources from combustion. Until recently Stirling cycles have been mainly of only theoretical interest.

Stirling Cycle

WhisperGen a New Zealand company, has developed an "AC Micro Combined Heat and Power" Stirling cycle engine. These micro CHP units are gas fired boilers which generate power and hot water. These are now available in the UK through the major power suppliers and can be fitted as an alternative to domestic boilers. They claim efficiencies in excess of standard condensing boilers at around 95%.

Although not directly applicable to energy recovery from hot water, in the future these small scale Stirling Engines could be considered as an efficient generator to provide life support heating and power for control on for example on unmanned gas platforms. Emissions would be lower than from diesel generators or gas engines.



Larger scale Stirling Engines

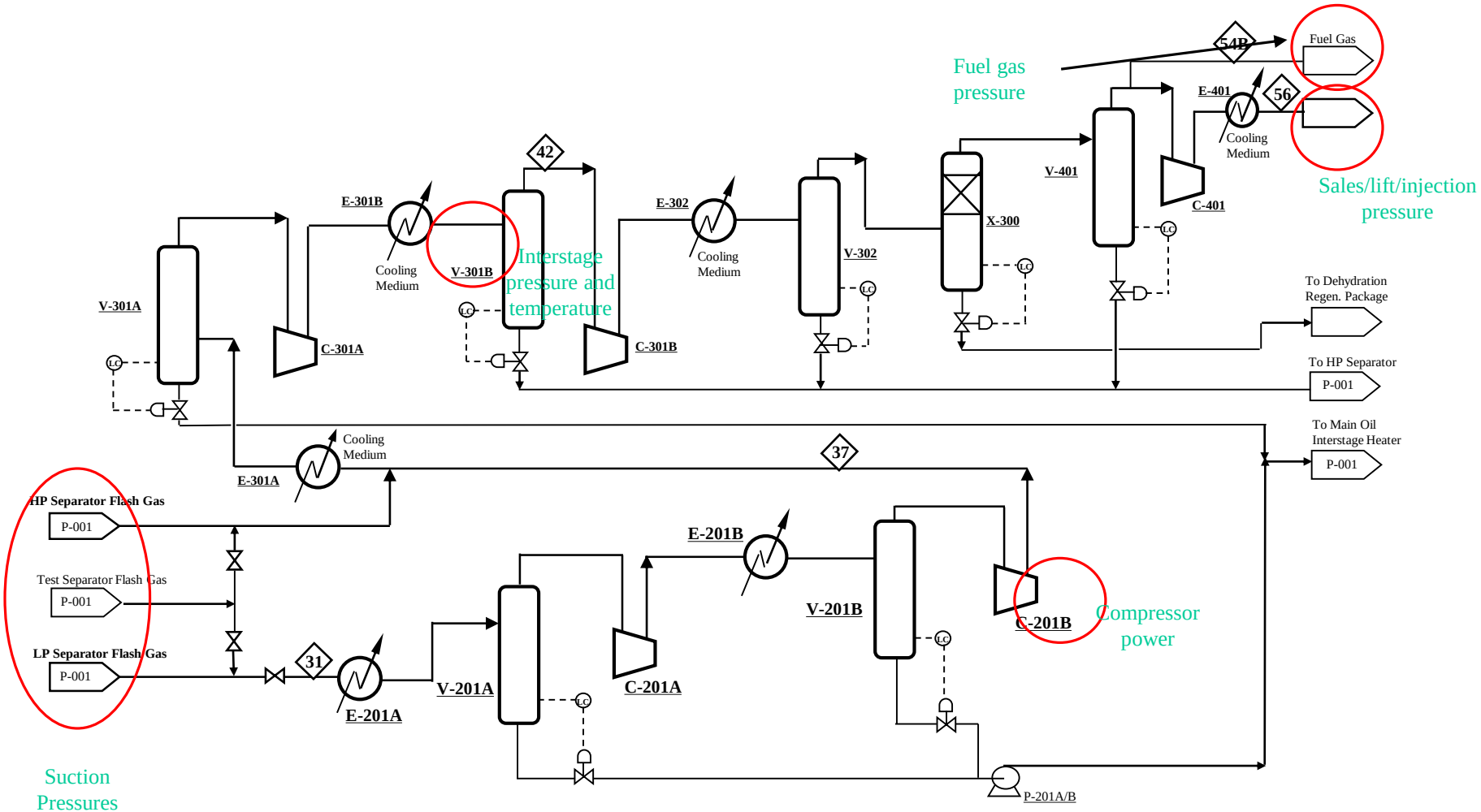
Deluge Inc a United States Company have developed a 16 cylinder CO_2 filled 250kW Stirling engine which operates on 4200 BPD of hot water at 80 ° C. Cooling water is required at 26 ° C at around 12000 BPD.

The temperature and flowrates are in an ideal range for produced water from oil and gas wells and seawater cooling source. Deluge Inc. claims an efficiency of up to 23%.

The Stirling machine was tested, funded by US government grants, at an onshore oil and gas test centre in Nevada 2005. The machine was used to pump oil wells using hot geothermal water. A commercial unit was subsequently supplied to Hawaii for installation in 2009.

A key issue is the complexity, size and weight of the Stirling Engine. A photograph of the unit is shown . The machine has a footprint of 73m2 for only 250kW of power and is likely to be heavy. The company say they hope to reduce the foot print to fit onto a standard lorry trailer in the future.

Gas Compression



Compression Thermodynamics Recap

- The basic thermodynamic equation for compression is: $$\Delta H = \int V dP = -W_{theor} \quad [1]$$
- For the isentropic compression process: $$P \cdot V^k = const = P_1 \cdot V_1^k \quad [2]$$
- Substitution of [2] into [1] yields: $$\Delta H = P_1^{1/k} \cdot V_1 \cdot \int_{P_1}^{P_2} P^{-1/k} \cdot dP \quad [3]$$
- or (solving): $$\Delta H = \frac{P_1^{1/k} \cdot V_1}{\left(\frac{k-1}{k}\right)} \cdot \left(P_2^{\frac{k-1}{k}} - P_2^{\frac{k-1}{k}} \right) \quad [4]$$
- Equation [4] is rewritten as: $$\Delta H = \frac{P_1 \cdot V_1}{\left(\frac{k-1}{k}\right)} \cdot \left(\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1 \right) \quad [5]$$
- Substitution of the ideal gas law into [5] yields: $$\Delta H = \frac{m \cdot Z_1 \cdot R \cdot T_1}{\left(\frac{k-1}{k}\right) \cdot MW} \cdot \left(\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1 \right) \quad [6]$$

ΔH	Enthalpy change (kJ/kg)
V	Gas volume (m^3)
P	Pressure (kPa)
W_{theor}	Theoretical work done (kJ/kg)
k	Ratio of specific heats (C_p/C_v)
m	Mass (kg)
Z	Compressibility factor (-)
R	Gas constant (=8.314 kJ/kmol.K)
T	Temperature (K)
MW	Molweight (kg/kmol)

Compression Thermodynamics

- Equation [6] (previous slide) is usually written as:

$$\Delta H_{isen} = \frac{Z_a \cdot R \cdot T_1}{\left(\frac{k-1}{k}\right) \cdot MW} \cdot \left(\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right) \qquad [7]$$

  which is the basic "head" equation for compressors, in which Z_a is the average compressibility ($(Z_1+Z_2/2)$) and ΔH_{isen} is the isentropic head (in kJ/kg). For the isentropic head in meters:

$$\Delta H_{isen}(meters) = \frac{1000 \cdot \Delta H_{isen}(kJ/kg)}{g\ (=9.81)}$$

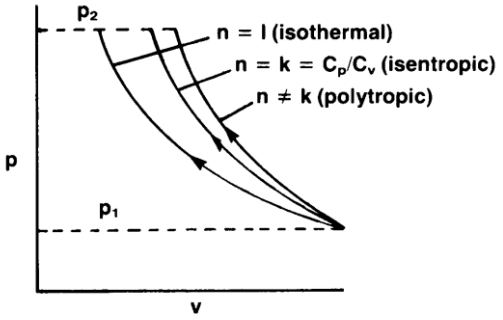
- In a steady state situation the entropy change in a system is written as: $\Delta S = Q/T_b + S_p$ [8]
  in which Q is the heat exchanged with surroundings, T is the absolute temperature of the system boundaries and S_p is the entropy production, reflecting the irreversibility of the process.
- An industrial machine will not be isentropic. The machine will have irreversibilites such as friction and noise. The ideal (isentropic) machine is corrected for inefficiencies as indicated below.

- Isentropic Process: $\Delta S = 0$, reversible ($S_p = 0$) and adiabatic process ($Q = 0$)
- Polytropic Process: $\Delta S \neq 0$, irreversible ($S_p \neq 0$), entropy is produced by internal friction in the system

$$\frac{n-1}{n} = \frac{k-1}{k \cdot E_{poly}}$$

n - Polytropic coefficient (n > k)

E - Polytropic efficiency



Compression Curves

Compression Thermodynamics

- Centrifugal Compressors
  - Head equation for practical applications:

$$\Delta h_{poly} = \frac{T_1 \cdot Z_a \cdot R}{\left(\frac{n-1}{n}\right) \cdot MW} \cdot \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

  - Temperature relationship:

$$T_2 = T_1 \cdot \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}}$$

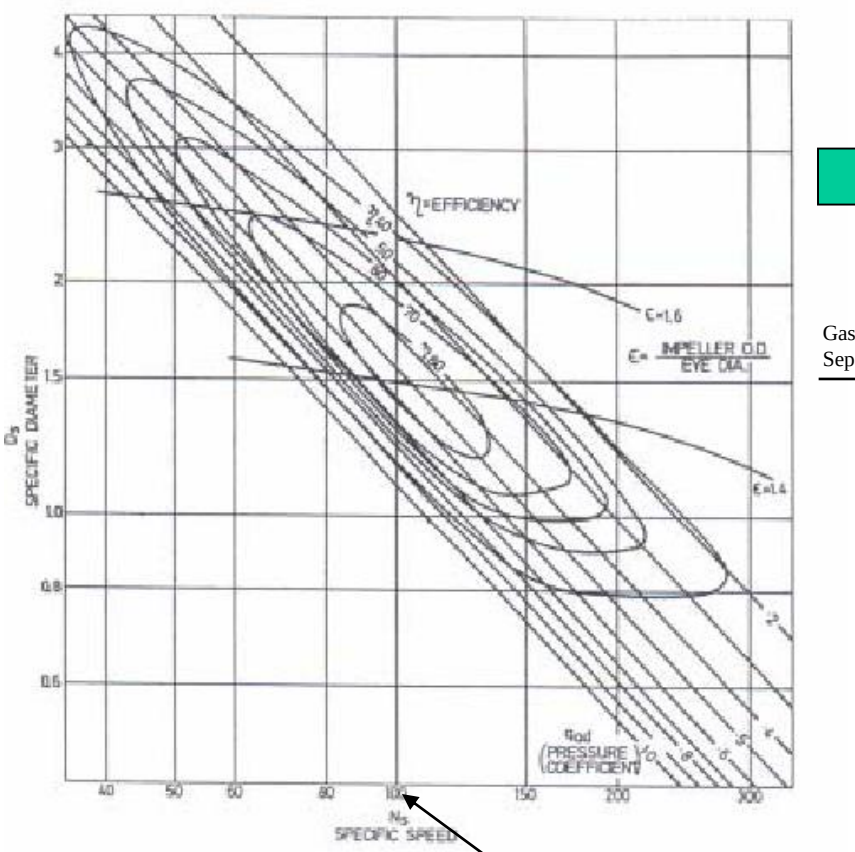
  - Power relationship

$$W = \frac{\phi_m \cdot \Delta h_{poly}}{E_{poly}}$$

Δh_{poly}	Polytropic head (kJ/kg)
T	Temperature (K)
Z_a	Average compressibility
R	Gas constant (8.314 kJ/kmol.K)
P	Pressure (Pa)
n	Polytropic coefficient
k	Ratio of specific heats (C_p/C_v)
MW	Molweight (kg/kmol)
W	Gas power (kW)
ϕ_m	Mass flow (kg/s)
E_{poly}	Polytropic efficiency - Function of volumetric inlet flow varying from
approx.	0.6 to 0.8 (compressor specific)
γ	Gas relative density (-)
1/2	Suction/Discharge

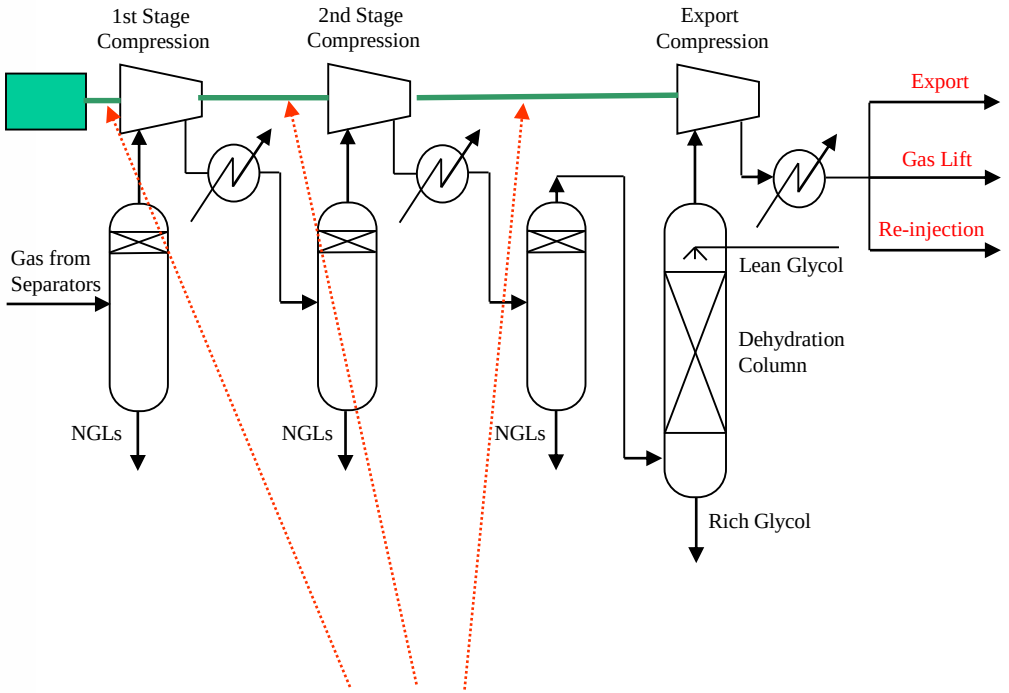
$$\frac{n-1}{n} = \frac{k-1}{k \cdot E_{poly}}$$

Centrifugal Compressor Speed Sensitivity Example



Optimum Specific Speed

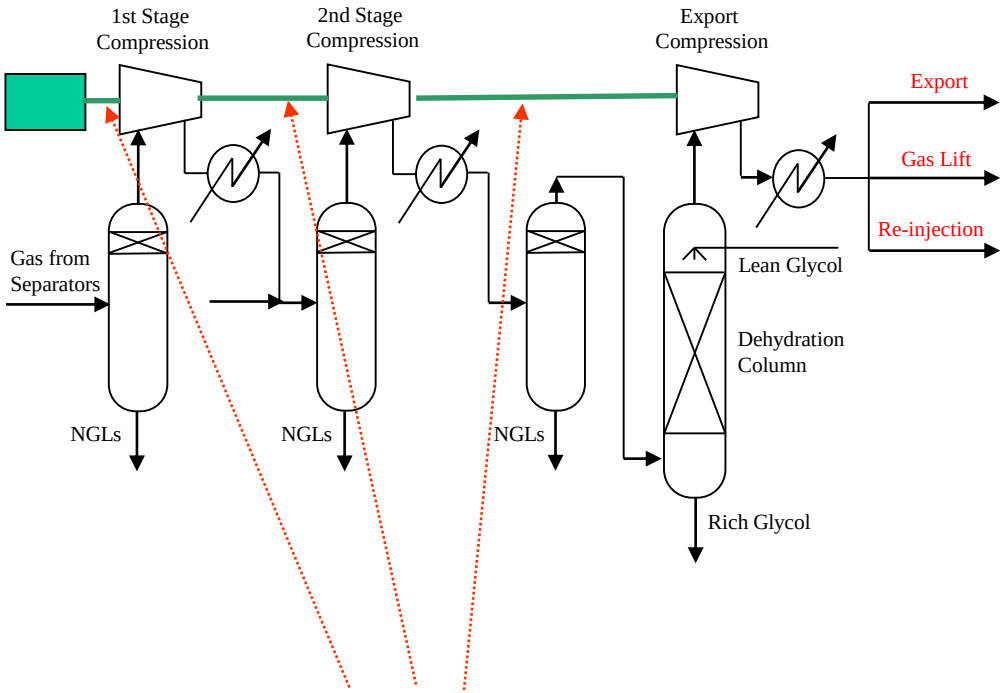
For peak efficiency



Single Shaft – what speed?

Centrifugal Compressor Train Design

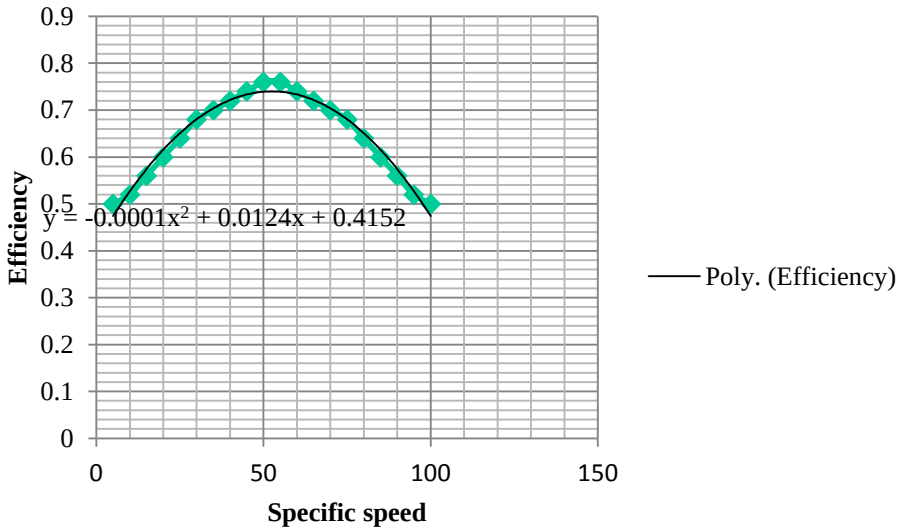
Compressor	1st Stage	2nd Stage	Export
Suction Volume Flow (m3/S)	8.95	1.95	0.65
Mol Weight	35	29.5	23.2
Z_a	0.955	0.858	0.807
k	1.096	1.155	1.223
Suction Pressure (bara)	3.79	17.8	54
Suction Density (kg/m3)	6.5	27.5	64.9
Mass rate (kg/s)	58.5	53.7	42.2
Suction Temperature (DegK)	258.9	270.5	291.3
Discharge Pressure (bara)	19	55	172.4
Compression Ratio	5.0	3.1	3.2



Single Shaft – what speed?

Efficiency Characteristics

For the compressors under evaluation the following efficiency characteristics are given.



Conduct a speed sensitivity analysis at 6000, 9000, 12000 rpm. Assume 3000m head per impeller stage and tip speed of 250 m/s.

1. Calculate specific speeds

$$N_s = \frac{N \cdot q^{0.5}}{H^{0.75}}$$

2. Calculate efficiencies
3. Calculate polytropic head
4. Calculate power
5. Identify optimum speed

Centrifugal Compressor Train Design Sensitivity study at three speeds – 6000, 9000 and 12000 rpm.

Head/impeller (m)	3000.00	3000.00	3000.00
Specific speed	44.28	20.67	11.93
Efficiency	0.77	0.63	0.55
k-1/k.E	0.11	0.21	0.33
Poly Head (KJ/kg)	103.97	83.25	119.10
Head (m)	10597.86	8485.97	12140.44
No of impellers	4	3	4
Power (kw)	7906.80	7122.99	9157.46
Total KW	24187.24		

Head/impeller (m)	3000.0	3000.0	3000.0
Specific speed	66.4	31.0	17.9
Efficiency	0.8	0.7	0.6
k-1/k.E	0.1	0.2	0.3
Poly Head (KJ/kg)	103.6	82.1	116.9
Head (m)	10560.8	8374.0	11912.4
No of impellers	4	3	4
Power (kw)	7588.3	6282.2	8151.2
Total KW	22021.6		

Head/impeller (m)	3000.0	3000.0	3000.0
Specific speed	88.6	41.3	23.9
Efficiency	0.7	0.8	0.7
k-1/k.E	0.1	0.2	0.3
Poly Head (KJ/kg)	104.5	81.5	115.3
Head (m)	10652.2	8308.4	11748.9
No of impellers	4	3	4
Power (kw)	8374.2	5793.4	7436.3
Total KW	21603.9		

Lowest Power at 12000rpm

Centrifugal Compressor Train Design Sensitivity study at three speeds – 6000, 9000 and 12000 rpm.

Head/impeller (m)	3000.00	3000.00	3000.00
Specific speed	44.28	20.67	11.93
Efficiency	0.77	0.63	0.55
k-1/k.E	0.11	0.21	0.33
Poly Head (KJ/kg)	103.97	83.25	119.10
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No of impellers	4	3	4
Power (kw)	8374.2	5793.4	7436.3
Total KW	21603.9		

Run 1st stage at 9000 rpm and 2nd and 3rd at 12000rpm – Power 20817 KW.

Centrifugal Compressor Efficiency

Significant improvements over time;

- Improved aero efficiency
- Reduction in parasitic losses due to better bearing and seal designs
- Improvements in manufacturing machine tool technology permitting complex geometries to be accurately machined

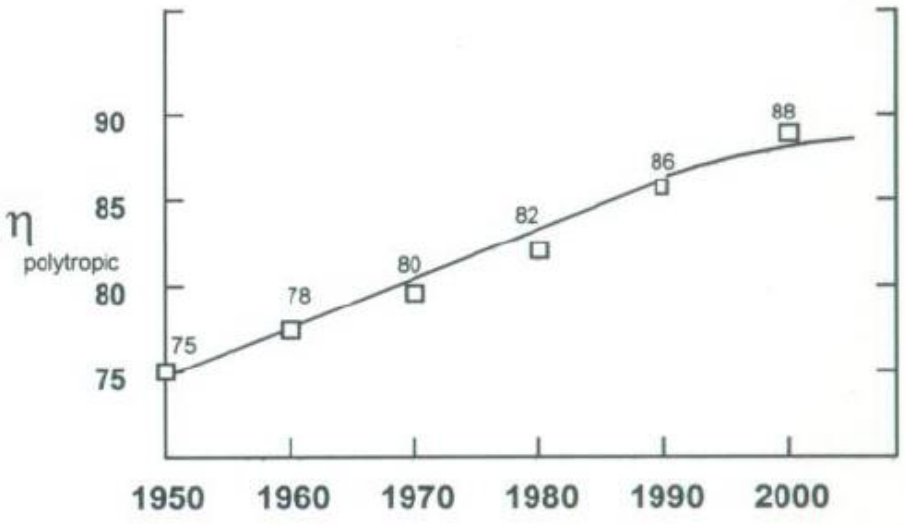


Figure 1. Flange to Flange Polytropic efficiency for large multistage centrifugal compressors.



Interstage Pressure and Temperatures

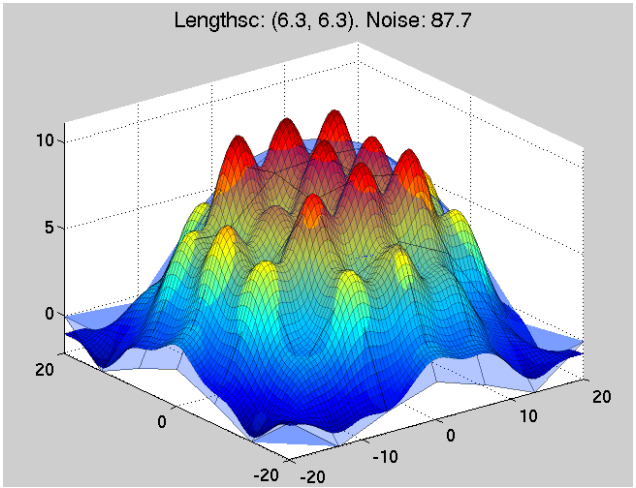
Set at the design stage to minimise CAPEX consistent with product recovery and sales specifications.

In operation there is usually some scope for minor manipulations.

Energy savings can be made by optimising interstage conditions.

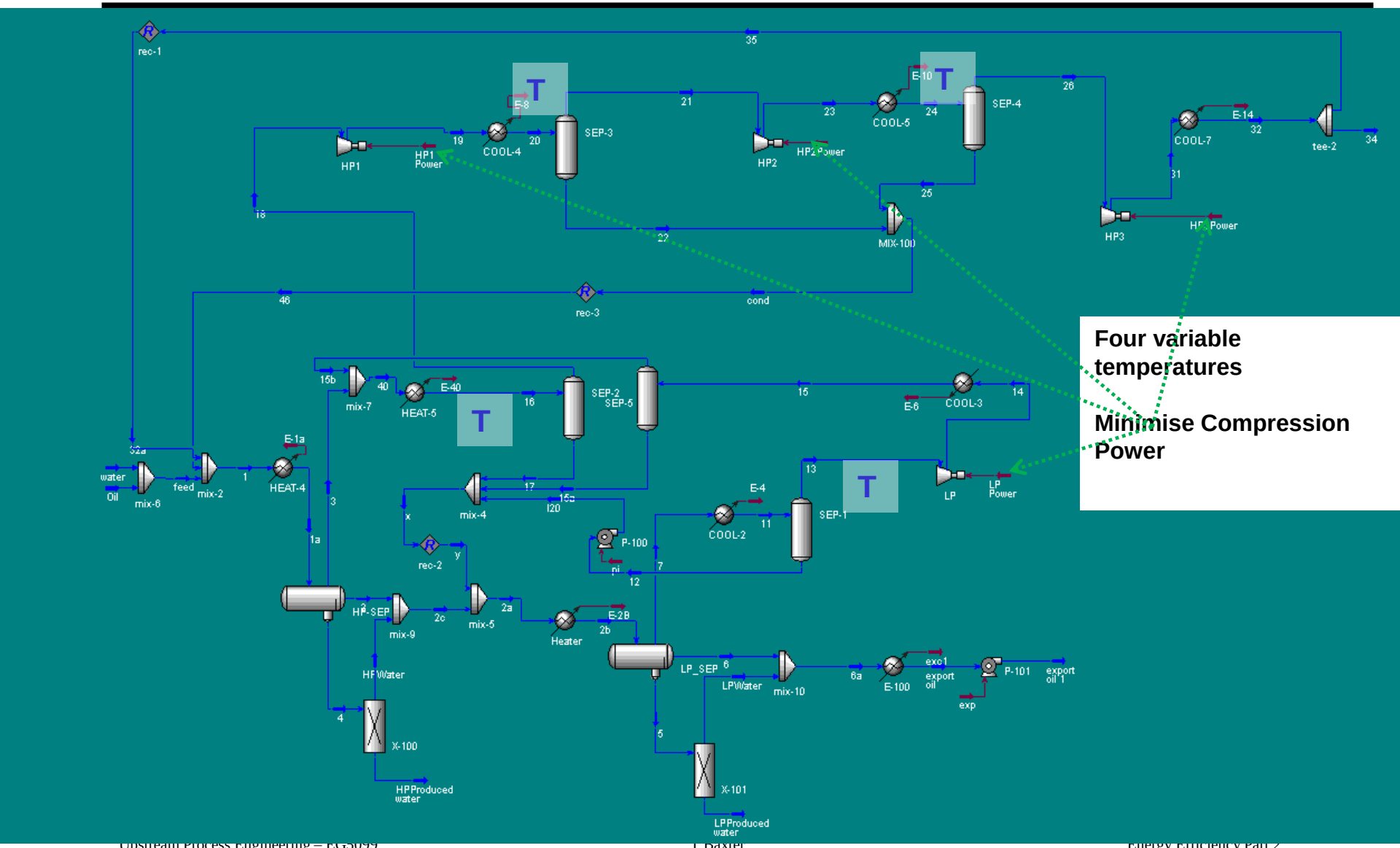
Optimisation Techniques

- SQP – Sequential Quadratic Programming
- BOX Method
- Fletcher Reeves
- Quasi Newton
- Evolutionary/Genetic Optimisation



A genetic algorithm is a search heuristic that mimics the process of natural selection. Genetic algorithms belong to the larger class of evolutionary algorithms (EA), which generate solutions to optimization problems using techniques inspired by natural evolution, such as inheritance, mutation, selection, and crossover.

Energy Optimisation 1

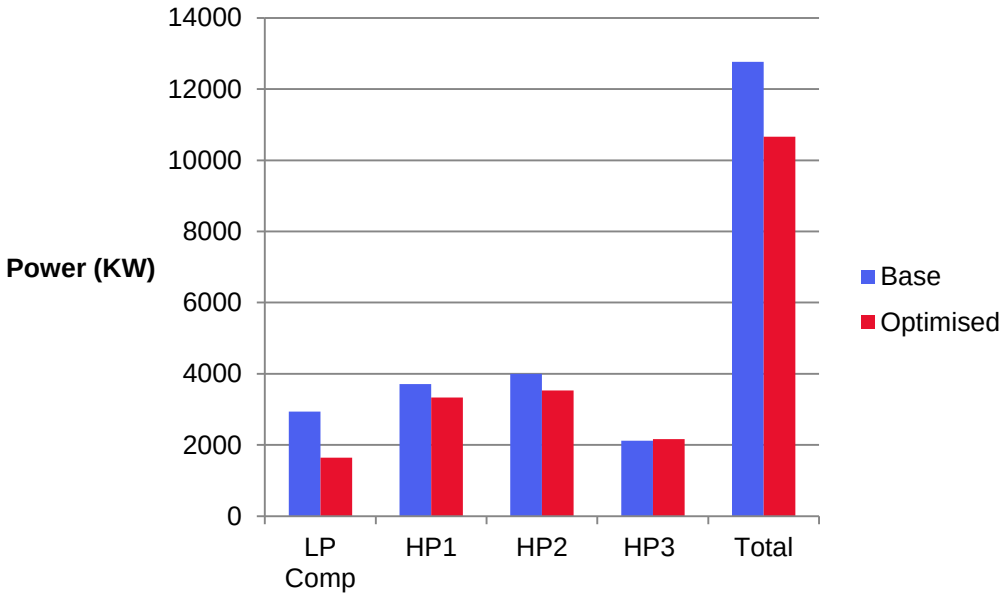


Four variable temperatures

Minimise Compression Power

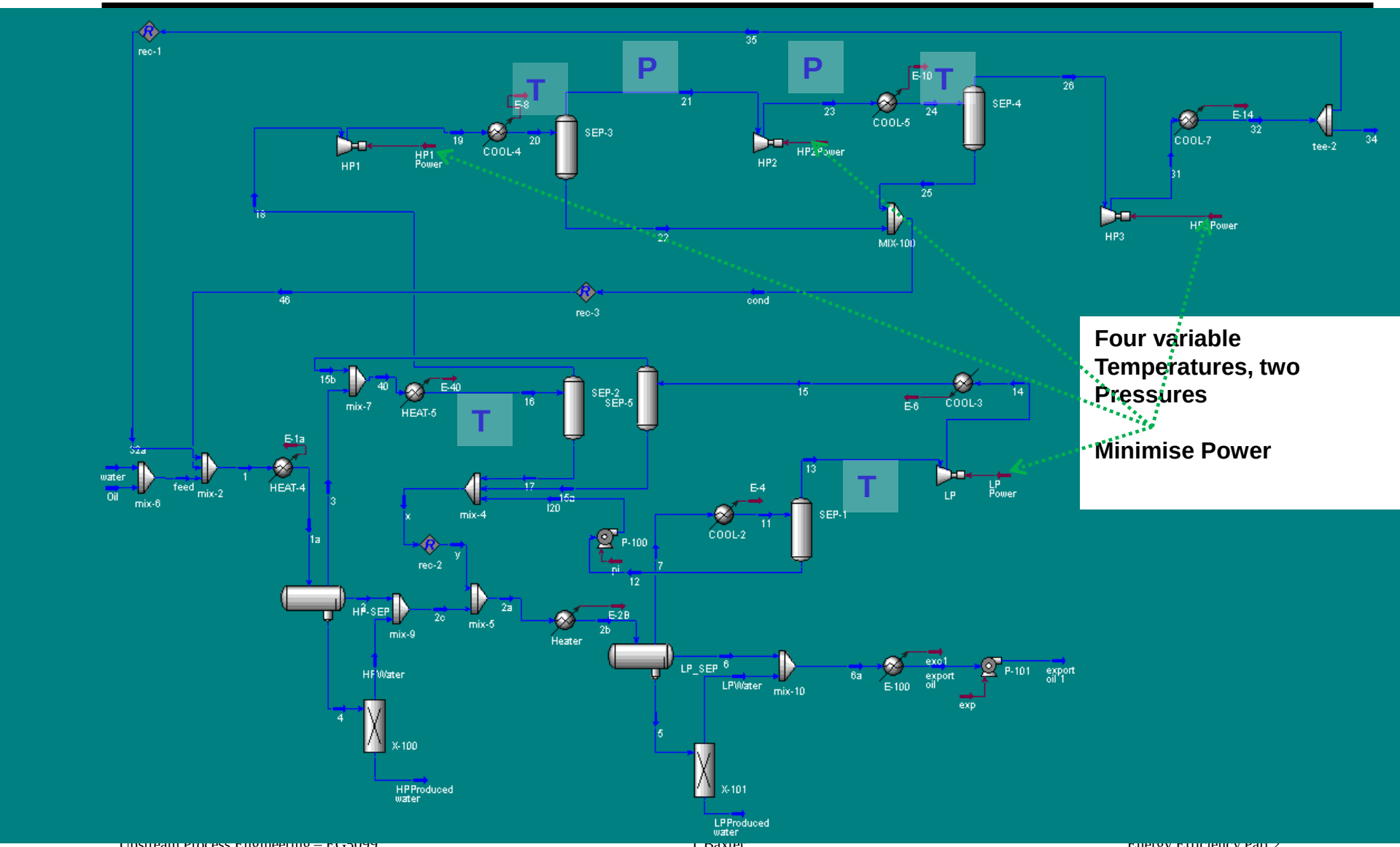
Energy Optimisation – Temperature Variables

Location	Unit	Optimised Value	Initial Value	Low bound	High bound
LP Comp Suction	Temp (° C)	25.7	30	25	35
HP1 Suction	Temp (° C)	29.0	30	25	35
HP 2 Suction	Temp (° C)	25.6	30	25	35
HP 3 Suction	Temp (° C)	33.3	30	25	35



15% Reduction in Power Requirements

Energy Optimisation 2

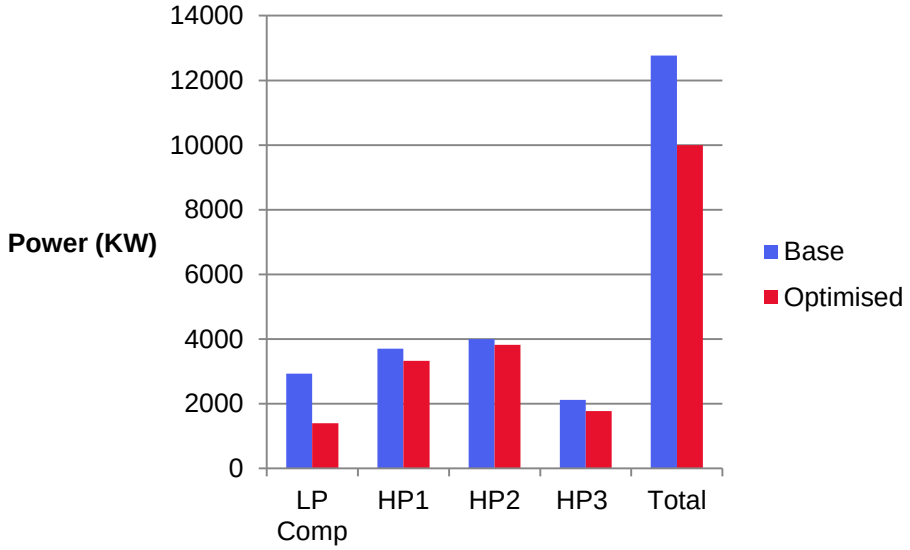


Four variable Temperatures, two Pressures

Minimise Power

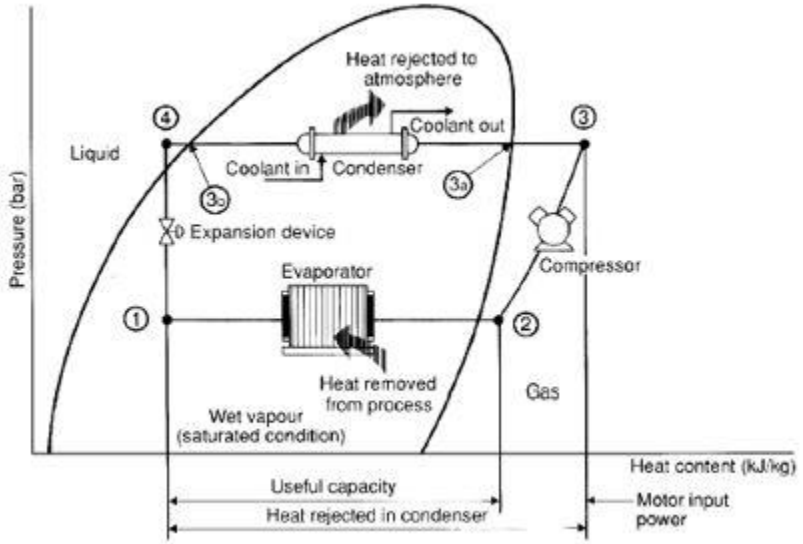
Energy Optimisation 2 – Temperature and Pressure Variables

Location	Unit	Optimised Value	Initial Value	Low bound	High bound
LP Comp Suction	Temp (° C)	25.9	30	25	35
HP1 Suction	Temp (° C)	28.5	30	25	35
HP 2 Suction	Temp (° C)	25.6	30	25	35
HP 3 Suction	Temp (° C)	34.3	30	25	35
HP 1 Discharge	Pressure (Barg)	25	24	22	28
HP 2 Discharge	Pressure (Barg)	84	74	60	90



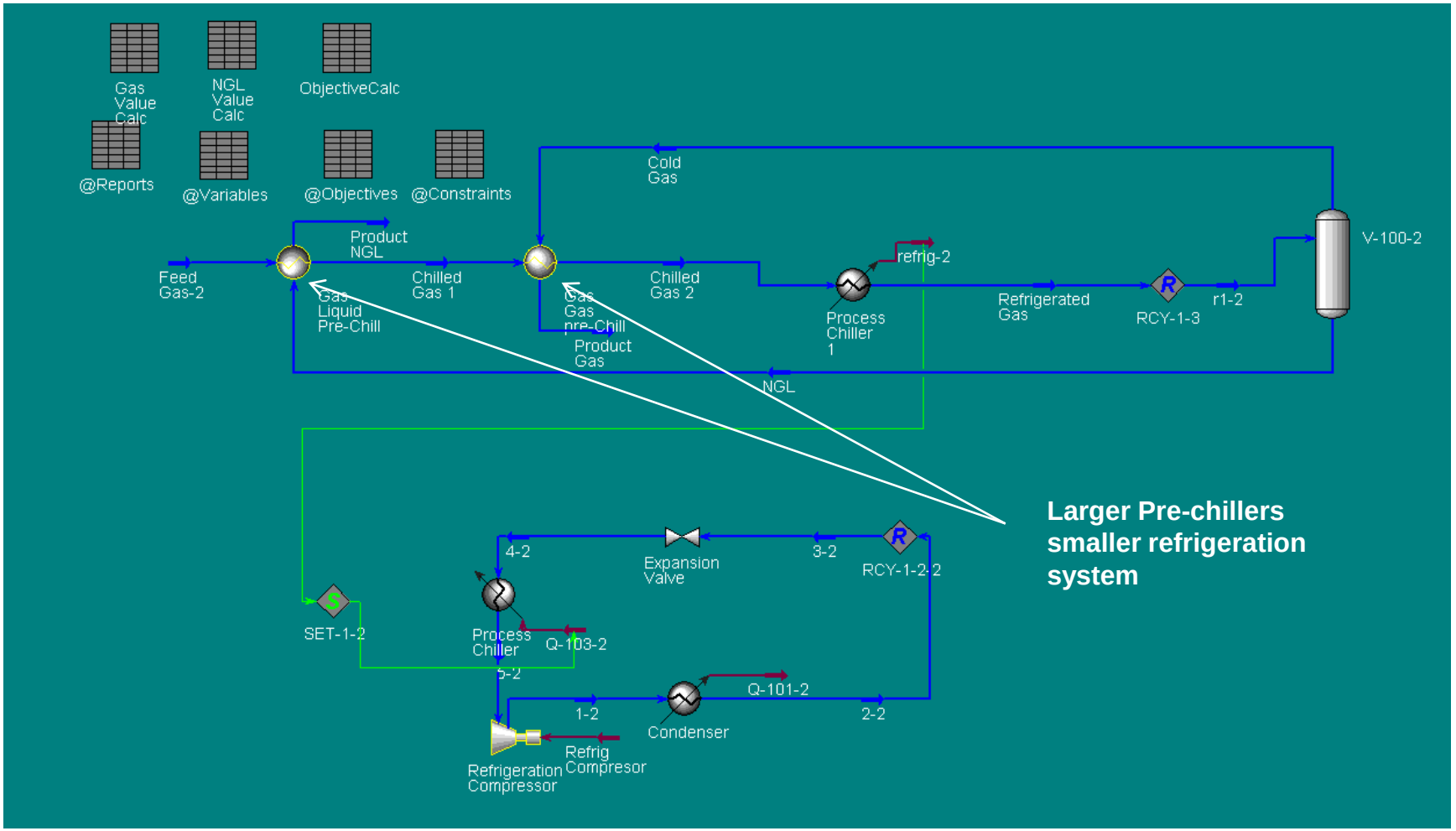
19% Reduction in Power Requirements

Refrigeration Optimisation





Refrigeration Optimisation 1



Refrigeration Optimisation 1

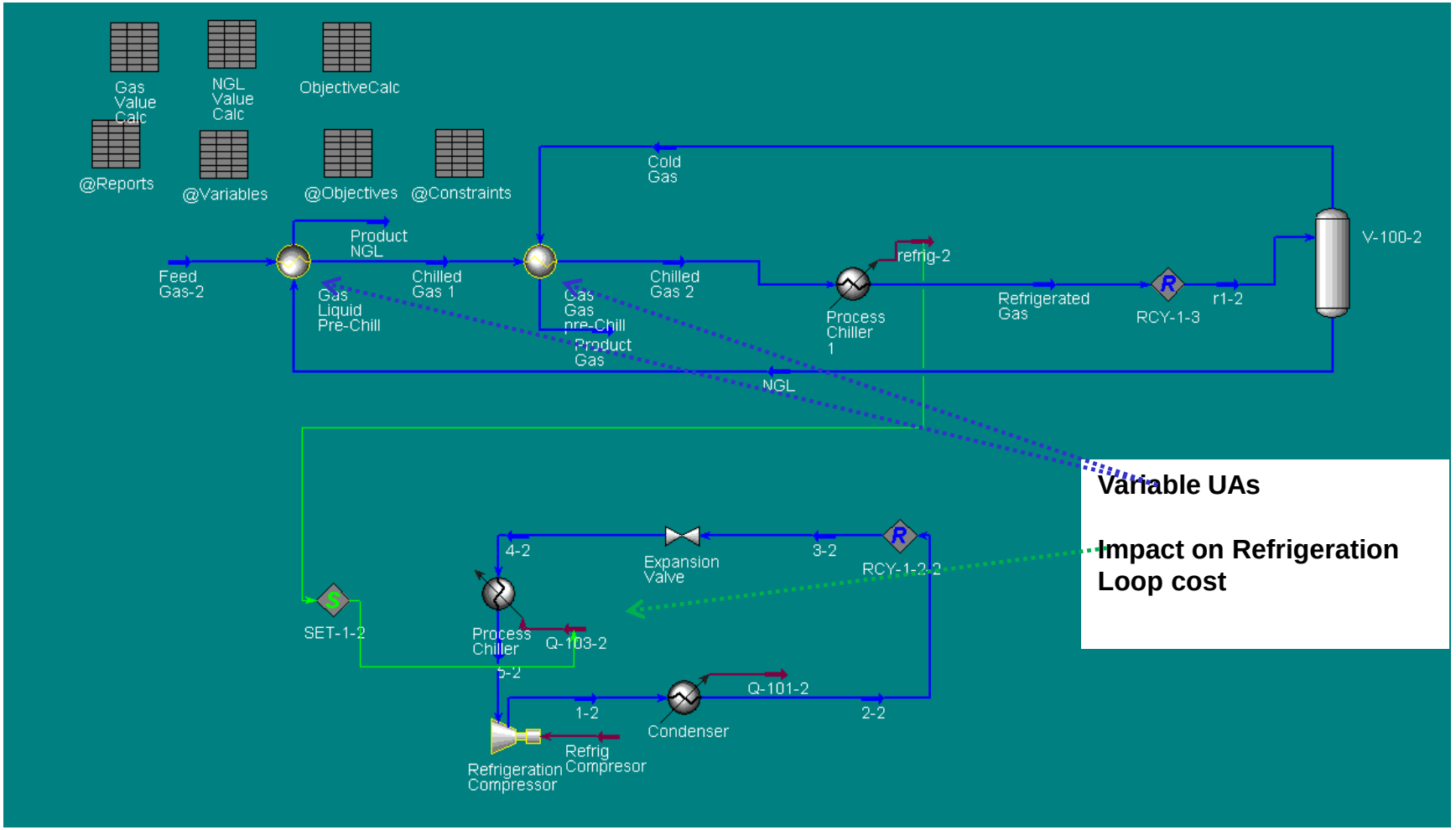
Objective – Minimise cost of Pre-Chillers and Refigeration Loop

2 Variable Optimisation – UA on both Pre-Chillers (overall heat transfer coefficient . Area)

Costs;

Gas-Liquid Prechiller
Gas-Gas Prechiller
Refrigeration Loop
Power (fuel) for refrigeration compressor
Cost of Carbon

Refrigeration Optimisation 1 – 2 variables



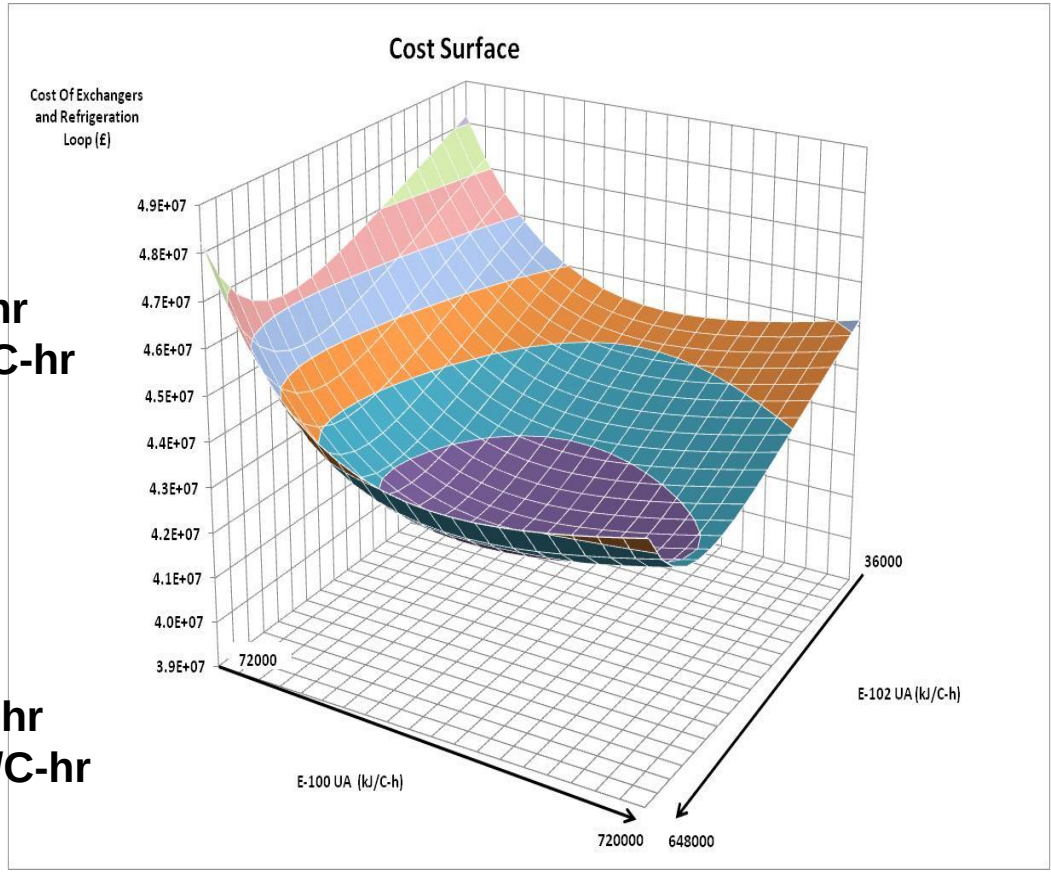
Refrigeration Optimisation 1 – 2 variables

Base case

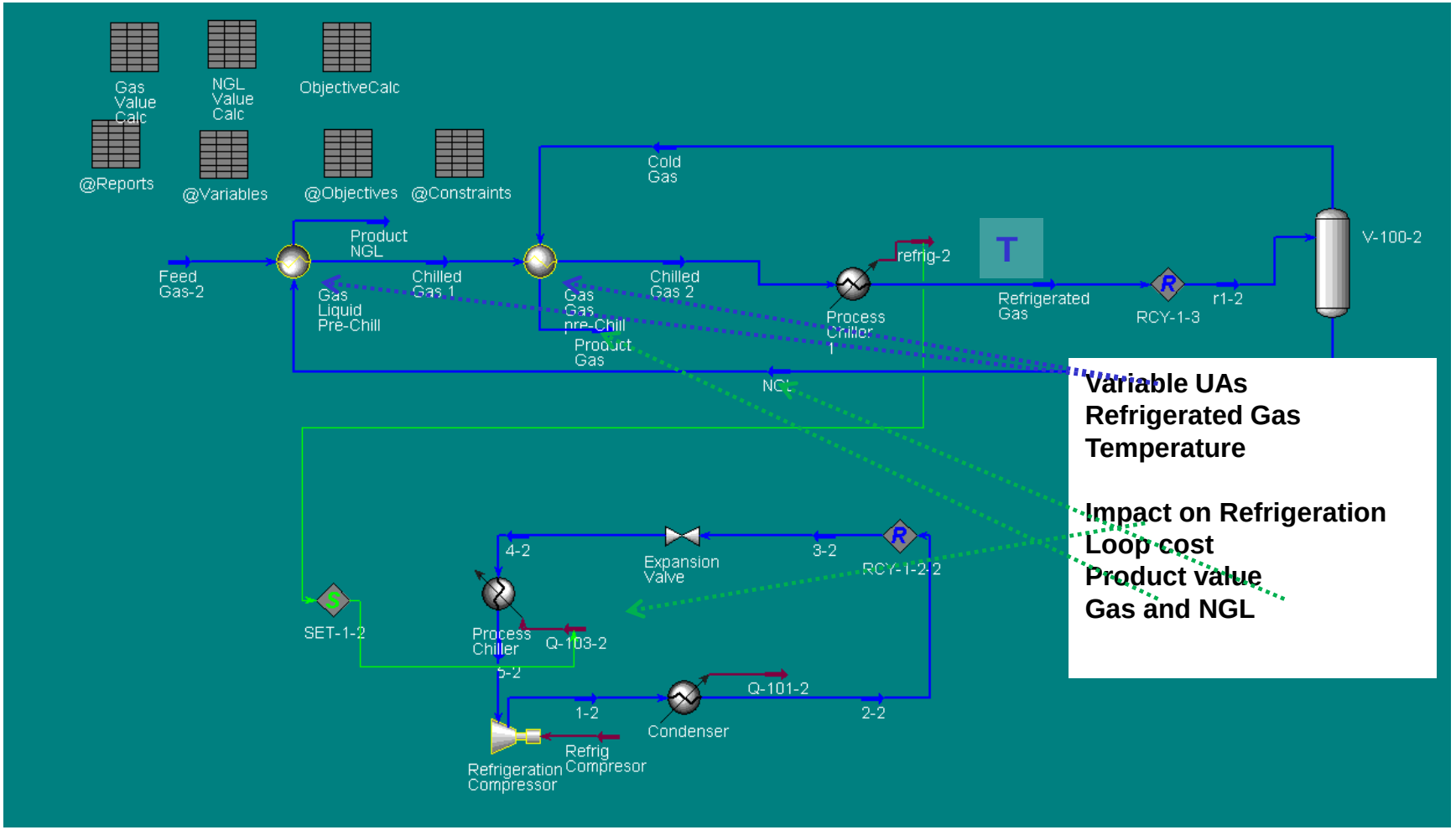
Gas-Gas Exchanger UA – 106000 kJ/C-hr
Gas-Liquid Exchanger UA – 106000 kJ/C-hr
Net Present Cost normalised to 100

Optimised Configuration

Gas-Gas Exchanger UA – 413000 kJ/C-hr
Gas-Liquid Exchanger UA – 616000 kJ/C-hr
Net Present Cost - 90



Refrigeration Optimisation 1 – 3 variables



Refrigeration Optimisation 2 – 3 variables

Objective – maximise NPV

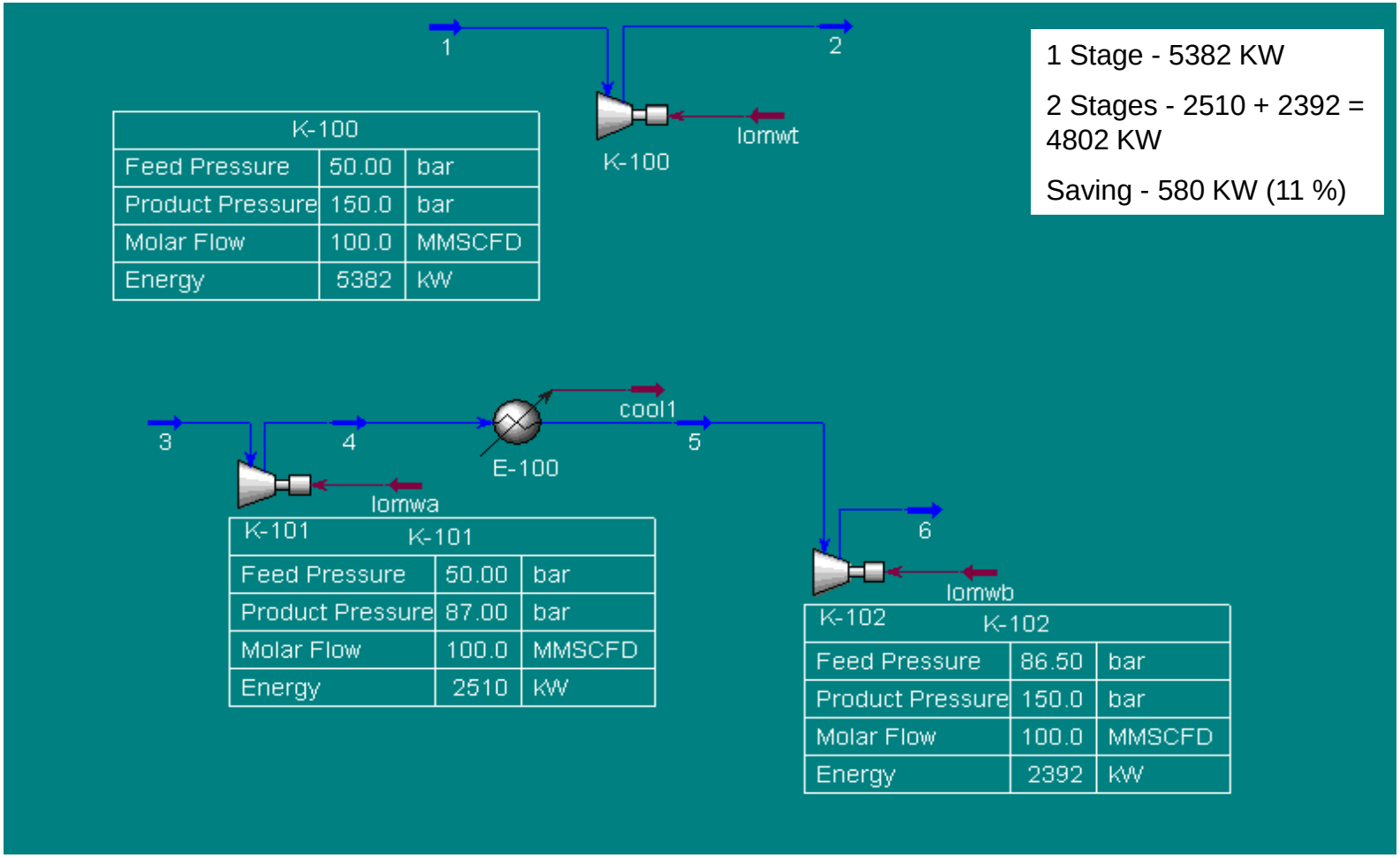
Base case

Gas-Gas Exchanger UA – 87 kJ/C-s
Gas-Liquid Exchanger UA – 87 kJ/C-s
Refrigerated gas temperature - -1° C
Net Present Value normalised to 100

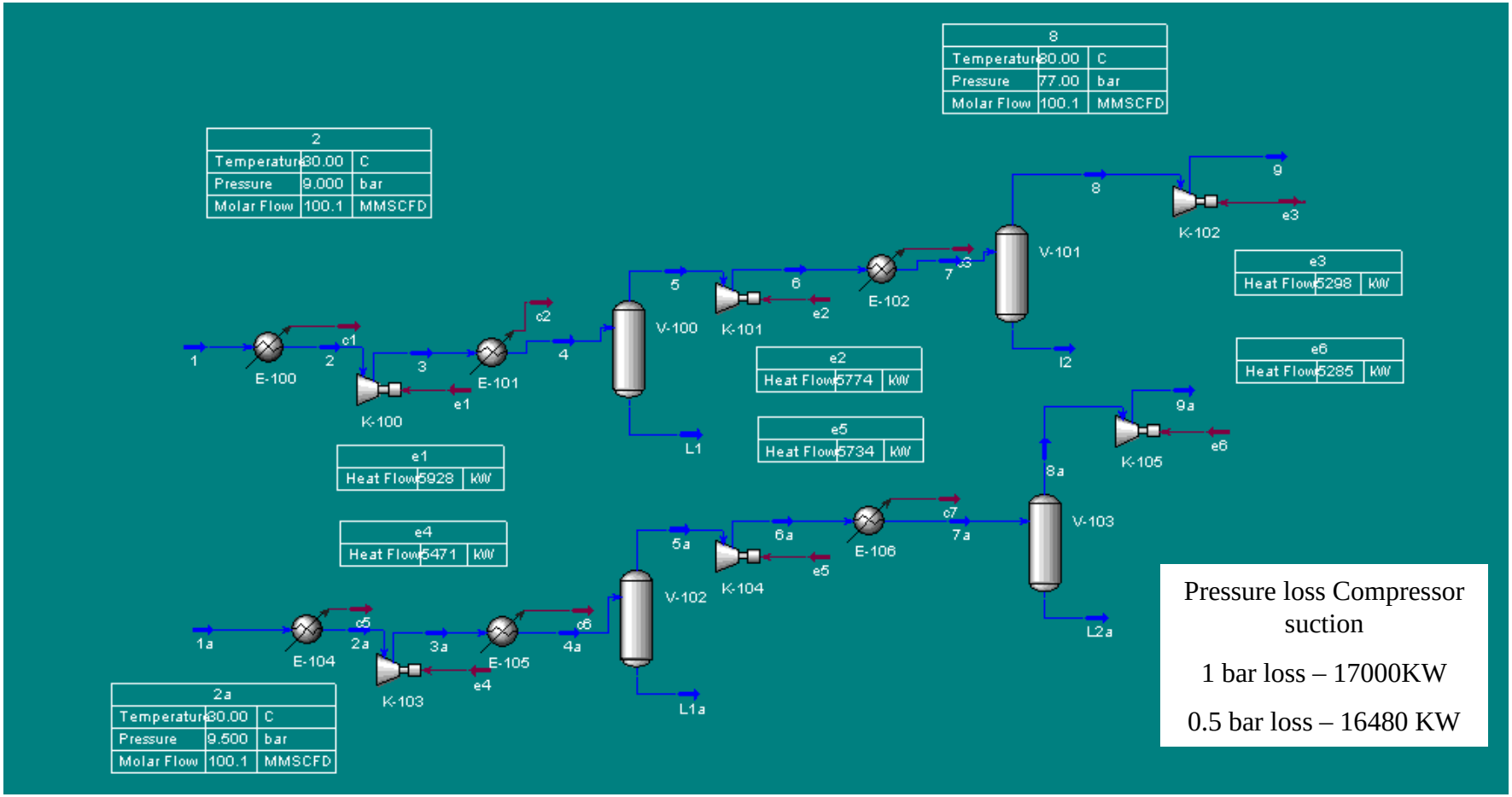
Optimised Configuration

Gas-Gas Exchanger UA – 34.55 kJ/C-s
Gas-Liquid Exchanger UA – 54.65 kJ/C-s
Refrigerated gas Temperature - plus 8° C
Net Present Value - 106

Compressor Interstage Cooling



System Pressure Losses



Pressure Losses

Significant pressure drop can occur through pipes, valves and exchangers.

At the design stage these are set to minimise CAPEX.

Use of larger pipe, low pressure drop heat exchangers and meters can deliver energy savings.

In operation unnecessary losses across control valves and manual valves can contribute to poor energy management.

Compression Power Example

30kg/s of hydrocarbon gas (MW = 22.0 kg/kmol) is compressed from 4 bara to 15 bara. Calculate the required compressor power and discharge temperature. The polytropic efficiency of the compressor is 75%. Z_a = 0.98. T_1 = 30°C

Specific gravity: 22.0 / 28.96 = 0.76

Ratio of specific heats (estimated):

k = 1.3 - 0.31* (0.76 - 0.55) = 1.23

Polytropic coefficient:

$$\frac{n-1}{n} \approx \frac{1.23-1}{1.23*0.75} = 0.25 \rightarrow n = 1.33$$

$$\Delta h_{poly} \approx \frac{303*0.98*8.314}{\left(\frac{1.33-1}{1.33}\right)*22.0} * \left[\left(\frac{15}{4}\right)^{\frac{1.33-1}{1.33}} - 1\right] = 175.5\ kJ/kg$$

or ΔH_{poly} = 175.5 * 1000 / 9.81 = 17890 m

$$W = \frac{30*175.5}{0.75} = 7000\ kW$$

The compressor is powered by a gas turbine with a thermal efficiency of 35%. The fuel gas to the turbine is methane. Calculate the CO2 emissions per year for this machine. Calorific value of methane is 33.9 MJ/sm3.

Fuel energy required = 7000/0.35 = 20000kW

Methane required = 20/33.9 = 0.59 sm3/s

1 sm3 of methane = 1/23.96 kmol = 0.0417 kmol

Methane required = 0.59x0.0417 = 0.0246 kmol/s

If gas turbine is burning air at stoichiometric rate;

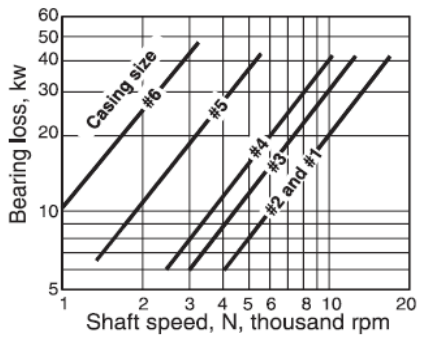
CO2 produced = 0.0246 kmol/s

= 34,100 tonnes/year

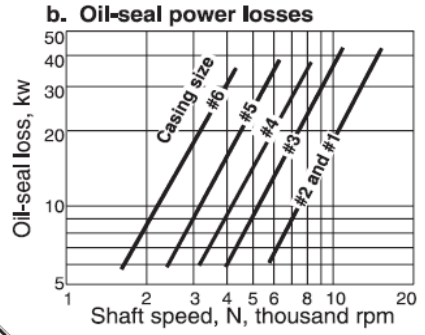
Centrifugal Compressor Energy Losses – Seals and bearings

Casing Size	Max Flow (inlet m^3/h)	Nominal Speed (rpm)
1	12 700	10 500
2	33 900	8 200
3	56 000	6 400
4	93 400	4 900
5	195 000	3 600
6	255 000	2 800

a. Bearing power losses



b. Oil-seal power losses



Courtesy Chemical Engineering Magazine

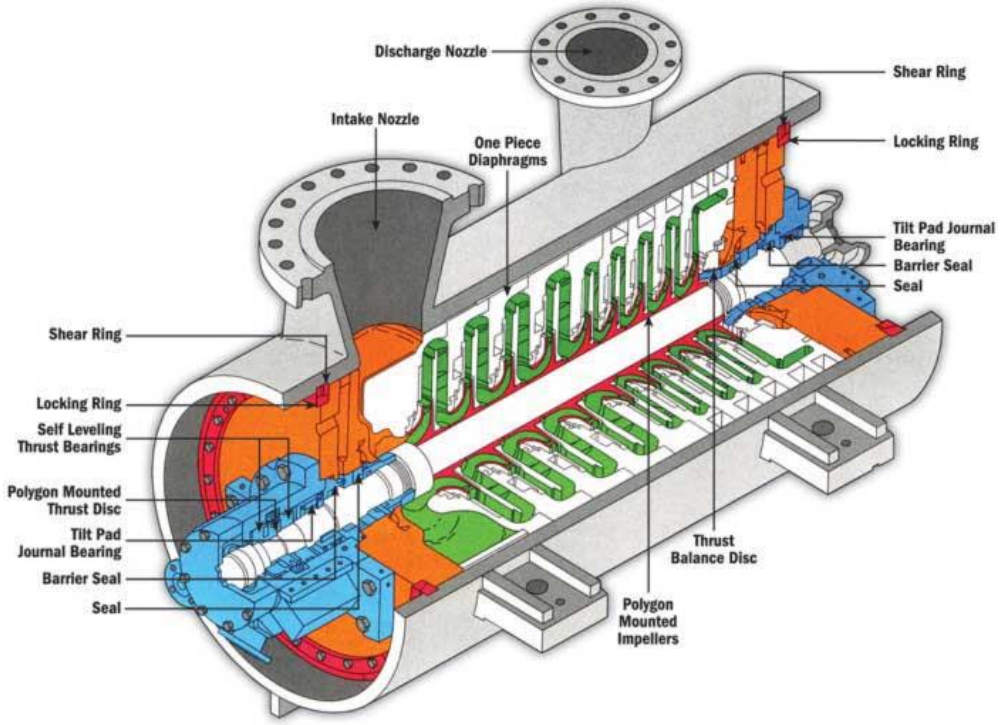
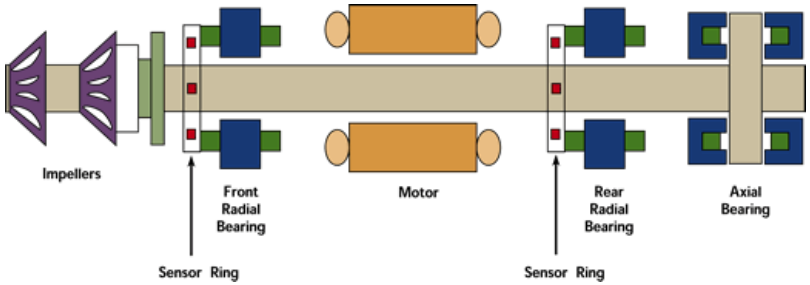
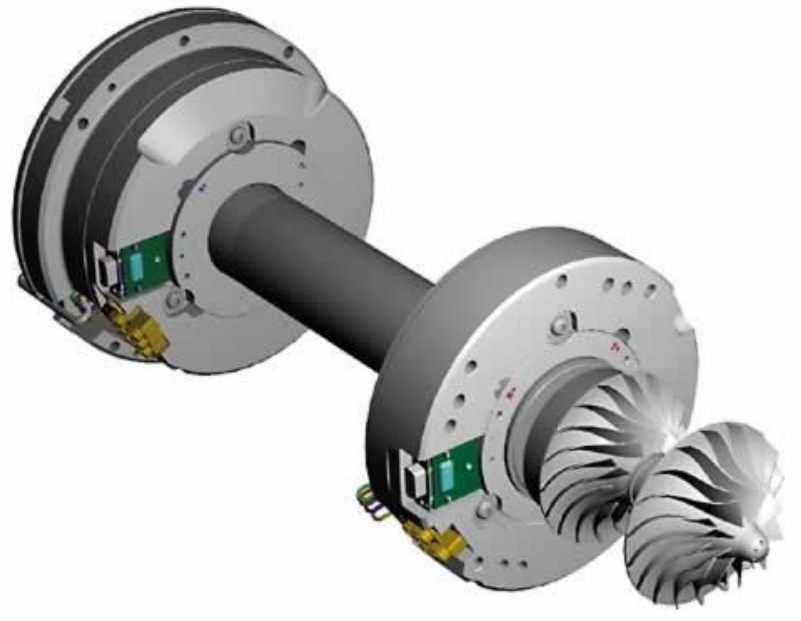


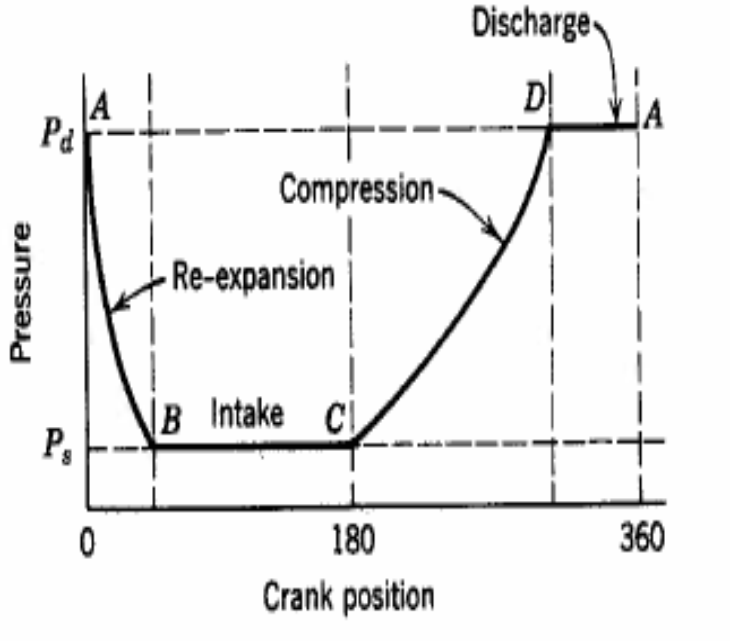
Figure 1. Major Components of Multistage Barrel-type Centrifugal Compressors (Dresser-Rand Co., Olean, NY)

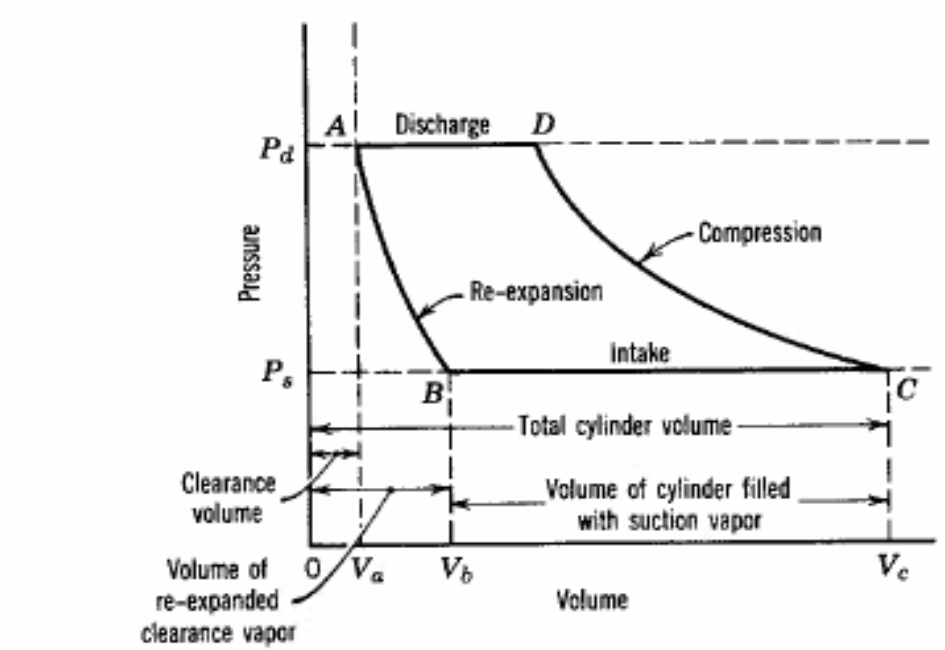
Magnetic Bearings





Reciprocating Compressors





Reciprocating Compressors

UNIVERSITY OF ABERDEEN

Volumetric efficiency

The actual volume of gas transferred from the suction line is the actual displacement of the cylinder. The ratio of this actual displacement to piston displacement is known as the total volumetric efficiency of the compressor

η_v= Volumetric efficiency (%)

V_{act} = actual volume of suction gas compressed per unit time

V_p = compressor piston displacement

$$\eta_V = \frac{V_{act}}{V_p} \text{ x } 100$$

Reciprocating Compressors

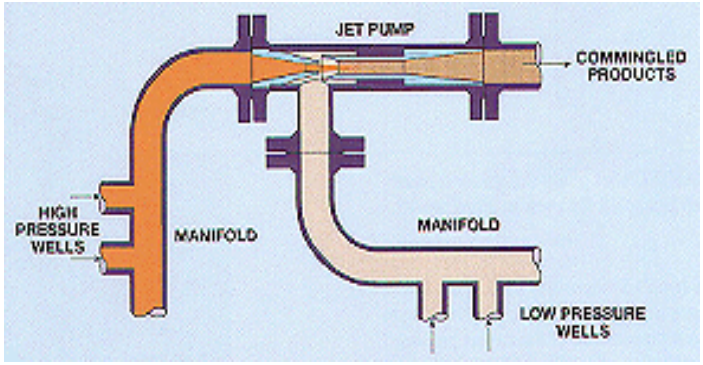
Effects of clearance

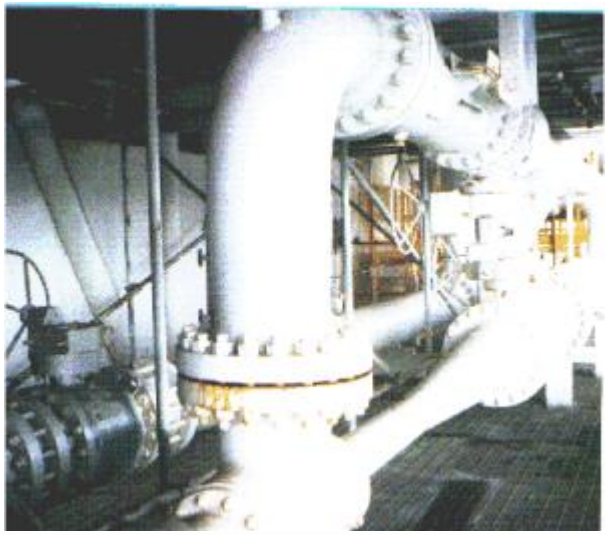
Wiredrawing

Cylinder heating

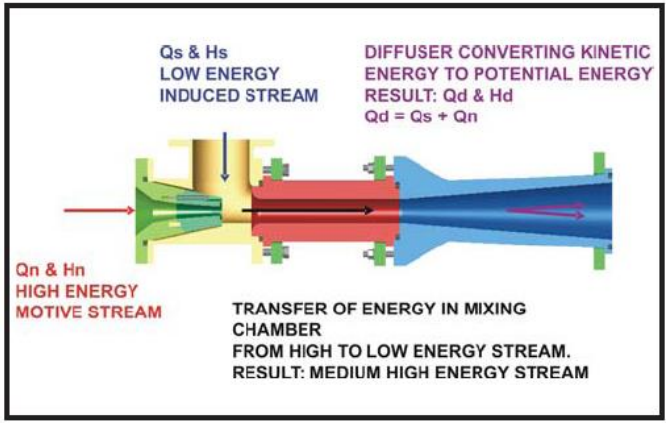
Cylinder and Valve leakage

HP Gas as Motive Energy for Low Pressure gas Wells

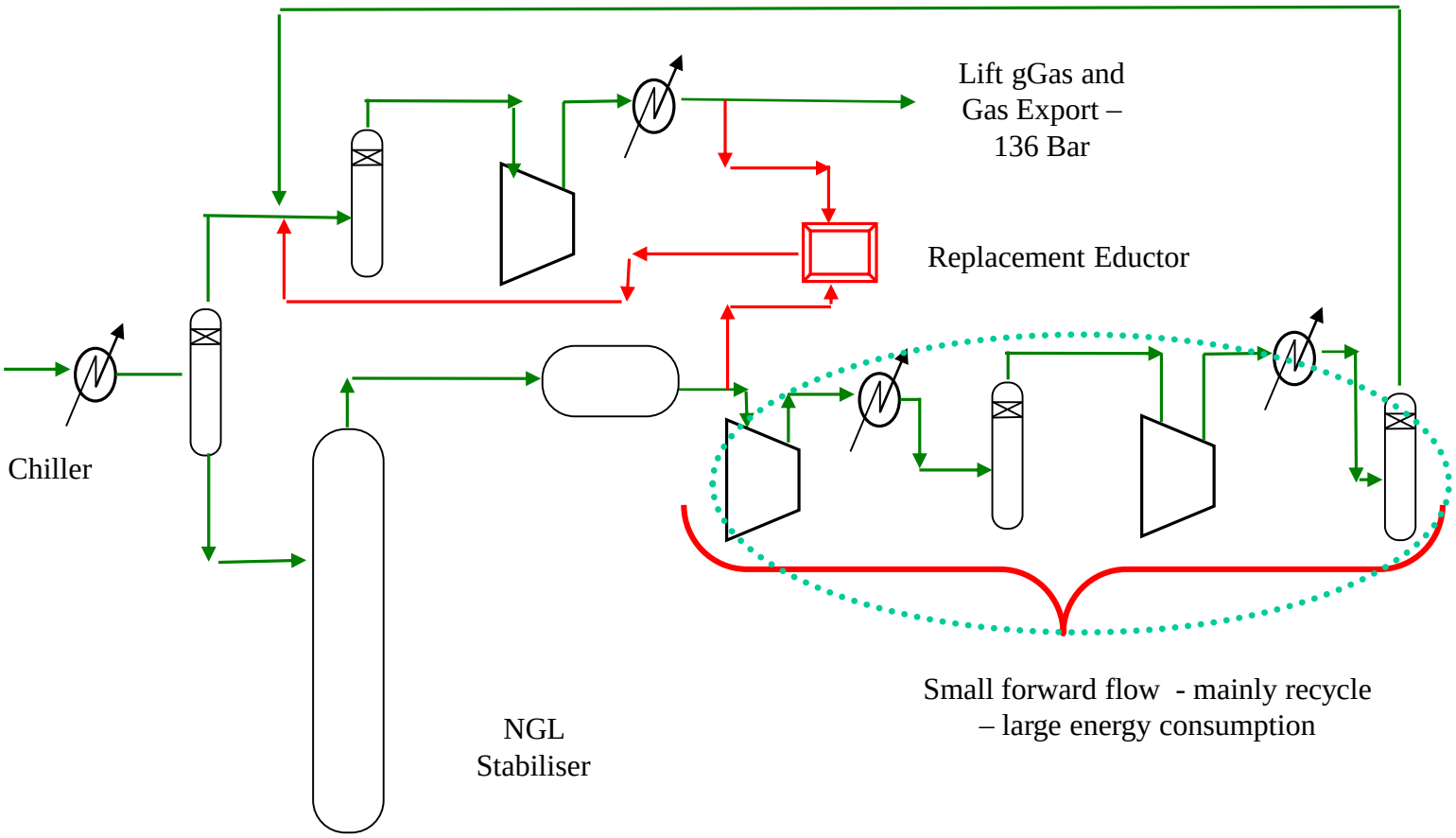




WELLCOM (BHRA) utilises energy from high pressure wells to drive products from low pressure wells. It achieves higher production rates, increased recovery and revenue at a low cost.
Traditionally, products of different wells are manifolded by choking the high pressure wells or by restricting the low pressure wells to match production pressures. This practice restricts total production and revenue.



HP Gas as Motive Energy

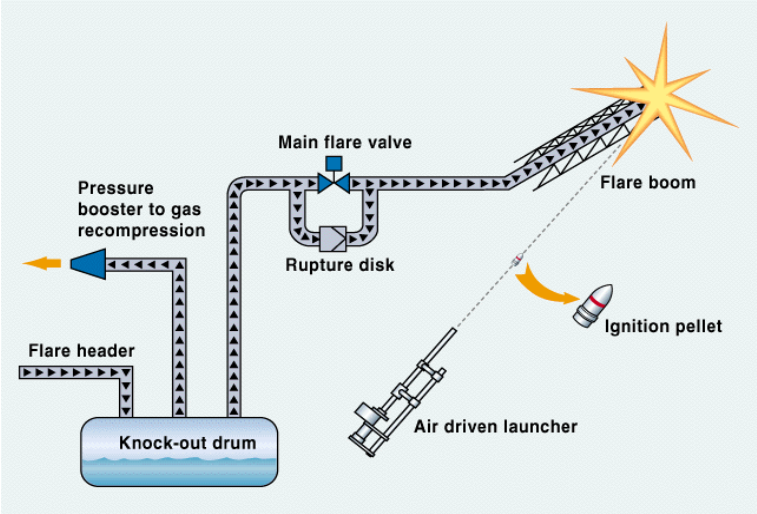


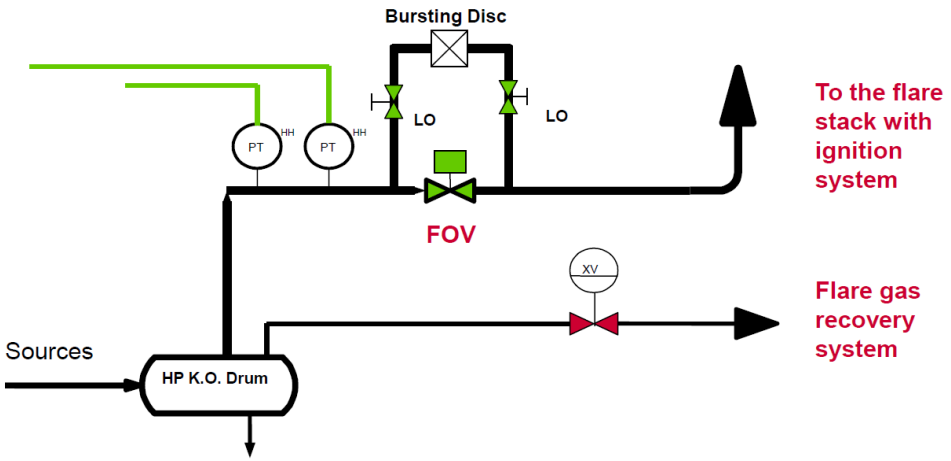


Flare Gas Recovery

Background flaring occurs on most oil installations - degasser vent, glycol reboiler vent, passing valves.

Flare Gas Recovery becoming custom and practice in Norway.

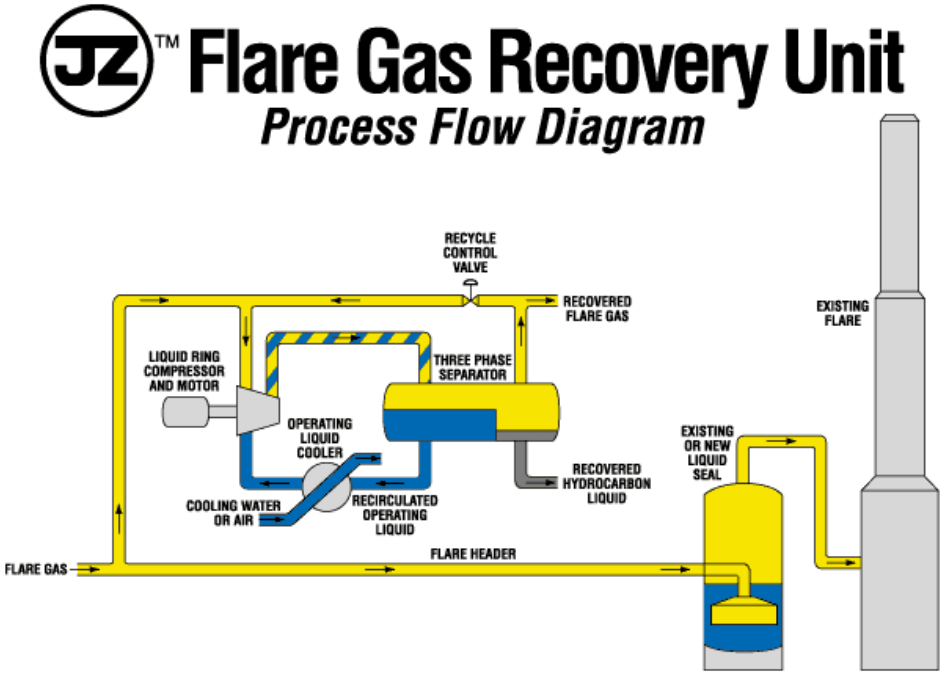




Gas is recovered from the flare knock-out vessel and returned to the compression train which reduces emissions and increases revenue from the gas.

The flare ignition system uses an air driven pellet which strikes against a metal plate at the flare tip to produce a spark.

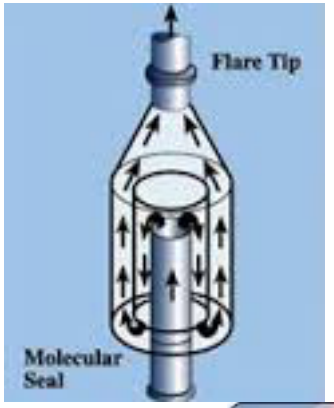
Flare Gas Recovery



Purge Gas

Typically a flare drum operates at a low pressure close to atmospheric pressure. This pressure is the built-up back pressure due to the continuous flared or purged gas flow in the vent stack. Pressure at the stack tip is atmospheric and back-pressure in the knock out drum is atmospheric pressure plus the frictional pressure drop from continuous venting/purging of the gas.

If the flow of gas to flare stack stops or is very low there is a possibility of air ingress into the flare stack and into the vent KO drum, vent network. This can result in an explosive mixture of air and hydrocarbons in the vent/flare network, which can be catastrophic. To prevent this a continuous flow of purge gas is used to prevent – purge gas is often process gas, hence there is continual combustion of gases.



Molecular Seal

A Molecular Seal can significantly decrease the amount of purge gas required for safe operation. The Molecular Seal design is based on the varying molecular weight between the air and the flared gas. Minimum purge-gas flow rates are allowed to pass through the Molecular Seal.

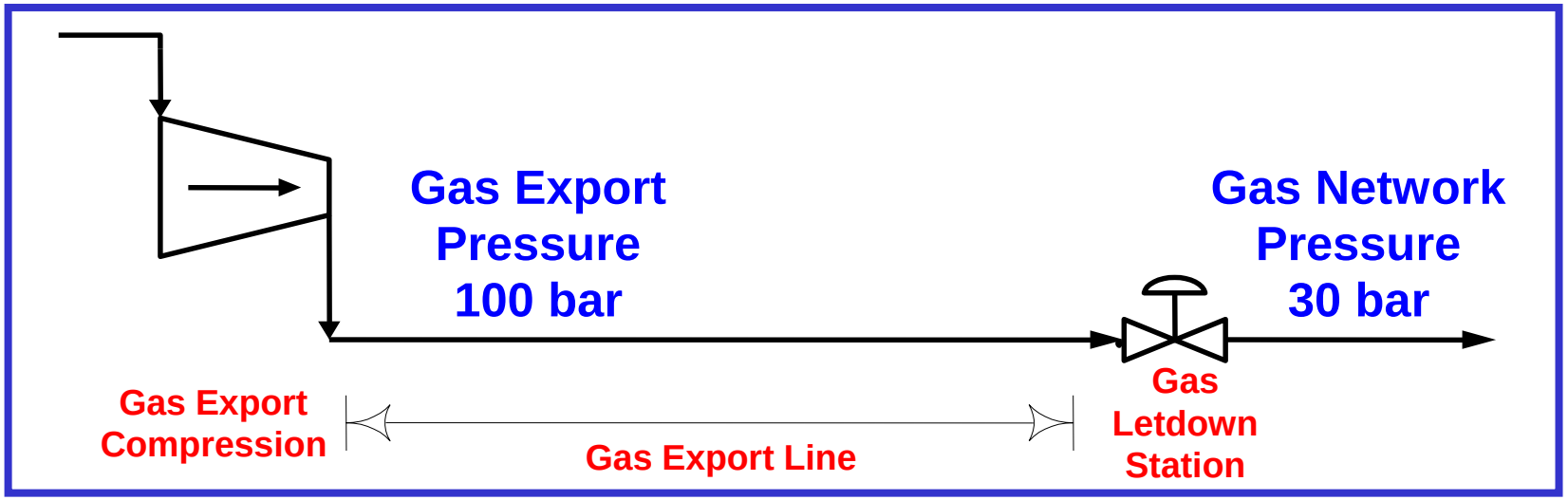


Pilot Gas

For most flare systems a continuous flow of process gas supplies the ignition system at the flare tip. This becomes a continuous combustion source.



Gas Export Line Pack



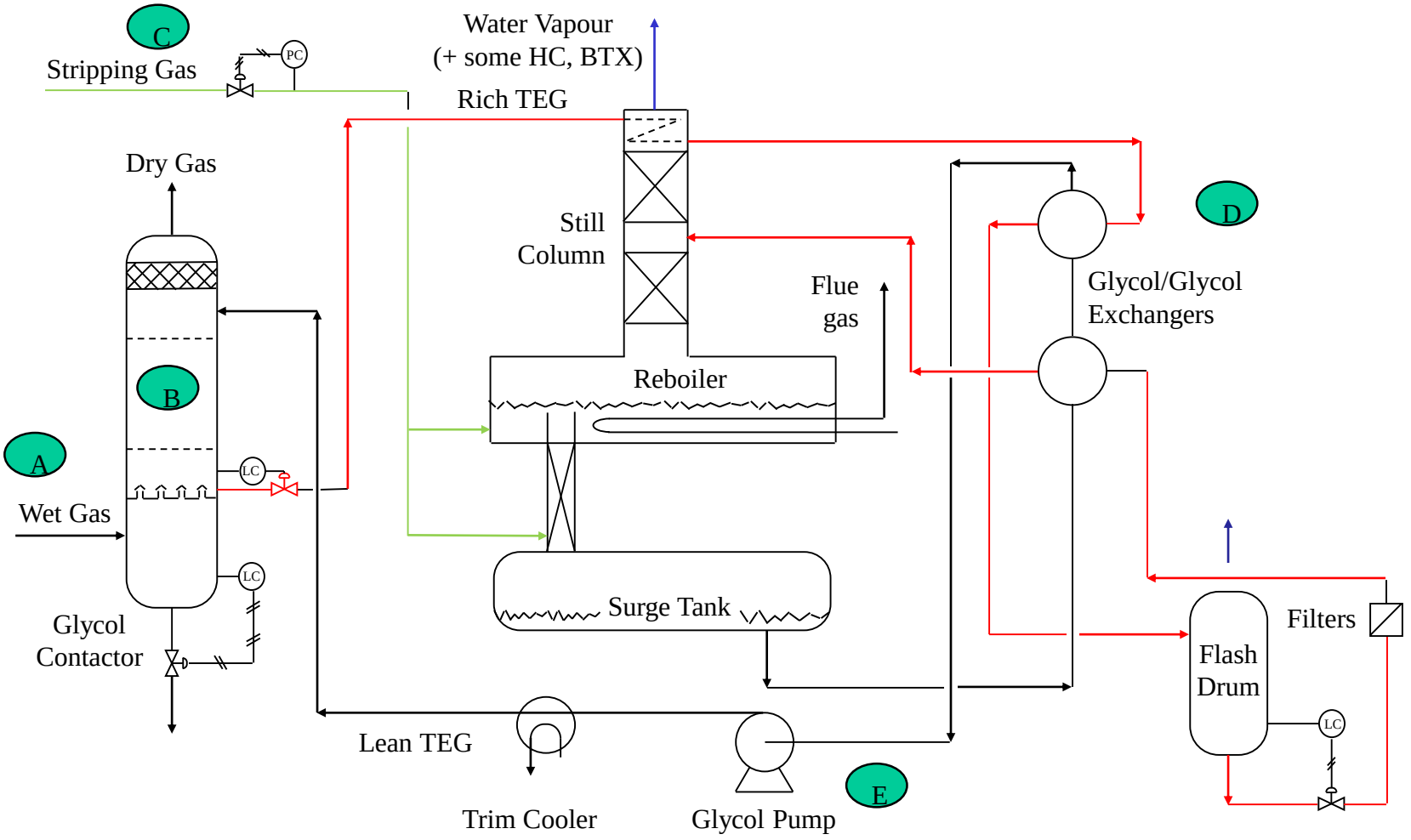
Export gas line pack – use pipeline as storage.

Provides compressor failure contingency.

Line pack to 100 bar.

Commercially attractive but energy inefficient

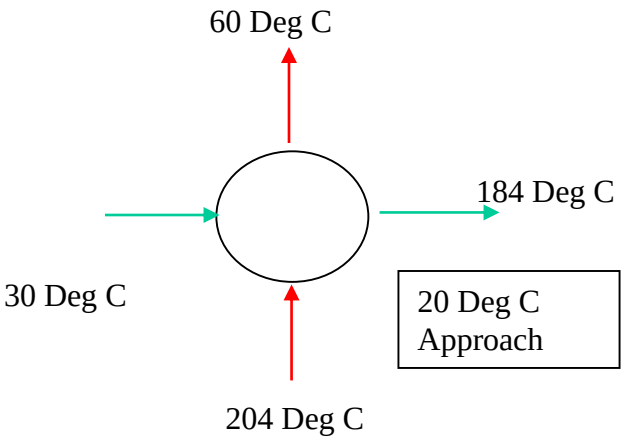
TEG Dehydration/Regeneration



TEG Dehydration/Regeneration

A – The colder the gas the lower the water content – lower reboiler duty
B – Pressure drop across bed – affects compression power
C – Stripping gas sent to flare
D – Heat exchanger temperature approach
E – Pump power – use HP rich glycol to power pump via a turbine

Lean/Rich Exchanger Temperature Approach



Exchanger OHTC – 600 W/M^2 DegC

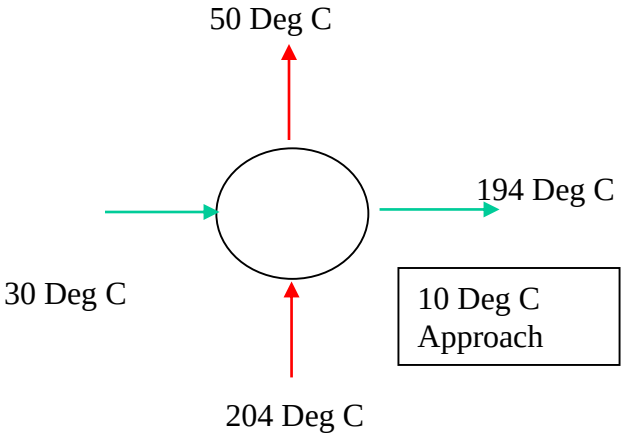
LMTD – 25 DegC

Rich Glycol Recirculation Rate -2 tonnes/hr

Heat Load – 241 kW

Exchanger Surface – 16 m^2

Exchanger Cost - £10,000



Exchanger OHTC – 600 W/M^2 DegC

LMTD – 14.4 DegC

Rich Glycol Recirculation Rate -2 tonnes/hr

Heat Load – 256 kW

Exchanger Surface – 30m^2

Exchanger Cost - £13,000

Reboiler saving – 15KW for £3000

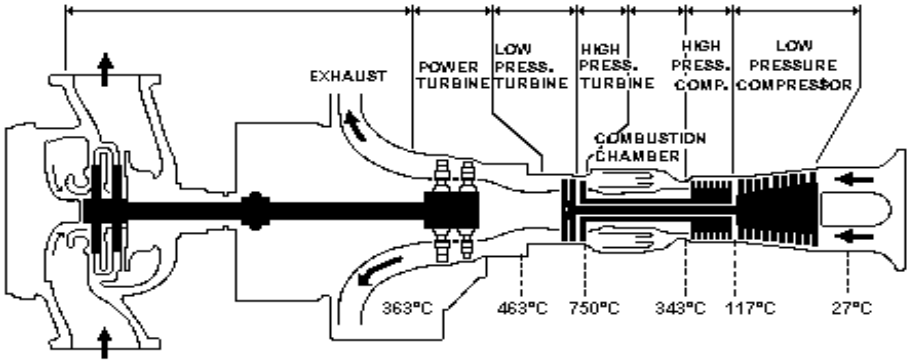
15KkW – 65 tonnes/year CO2

46 £/tonne of CO2 saved

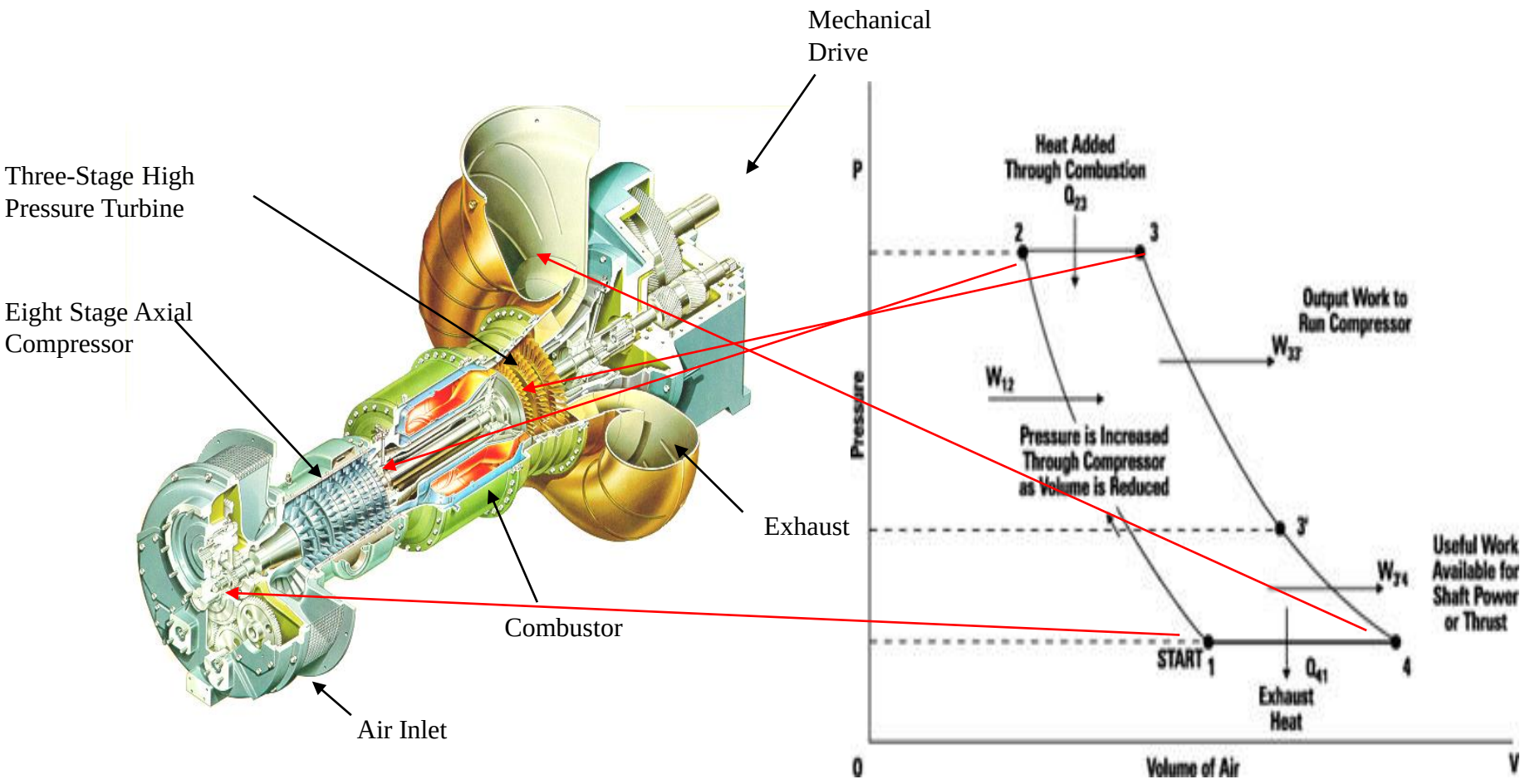
Gas Turbine Principles

The operating principles of a gas turbine are as follows:

- Air is drawn through inlet filters and silencers and adiabatically compressed in stages of axial compression, the air then passes to the combustion chamber
- Some of the air is fed directly to the fuel burners, the majority being used to cool the outer surfaces of the combustion chamber
- At higher combustion temperatures, the efficiency of the turbine increases however the the turbine blades would have a reduced operating life, therefore an economic compromise is required
- The hot gases then expand adiabatically through the power turbine section which drives the alternator/rotating plant.



Gas Turbines

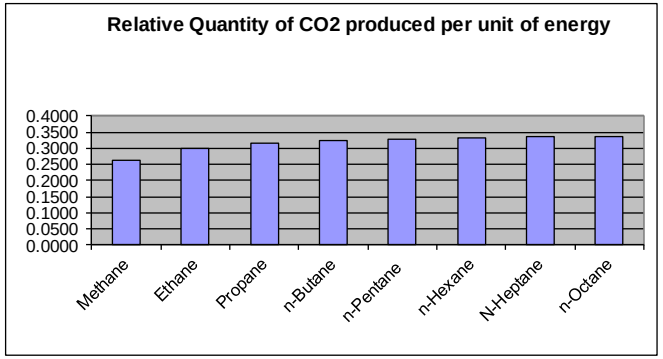


Combustion Chemistry

Component	Molecular Weight	Gross Calorific Value BTU/SCF	Lbmols required to produce 1 therm (100000BTU)	Lbs required to produce 1 therm (100000BTU)	Lbmols CO2 produced/lbmol of fuel gas	Lbmols CO2/Therm
Methane	16	1010	0.26	4.18	1.00	0.2612
Ethane	30	1768	0.15	4.48	2.00	0.2985
Propane	44	2517	0.10	4.61	3.00	0.3145
n-Butane	58	3262	0.08	4.69	4.00	0.3235
n-Pentane	72	4009	0.07	4.74	5.00	0.3291
n-Hexane	86	4756	0.06	4.77	6.00	0.3329
N-Heptane	100	5502	0.05	4.80	7.00	0.3357
n-Octane	114	6250	0.04	4.81	8.00	0.3377

$$CH_4 + 2O_2 \longrightarrow CO_2 + 2H_2O$$

$$C_8H_{18} + 25/2O_2 \longrightarrow 8CO_2 + 9H_2O$$



CO_2 Emissions from Hydrocarbons

Product	Average Carbon No	CO_2 kg per kg	CO_2 kg Per kWh	Other unit
Methane	1	2,75	0,178	1,92 kg CO_2 / scm
Gasoline	8	3,09	0,241	2,28 kg CO_2 / liter
Diesel	12	3,11	0,249	2,68 kg CO_2 / liter
Fuel Oil	25	3,12	0,268	3,97 kg CO_2 / liter
Coal	1	3,67	0,325	

Combustion Temperature

When natural gas is burned with air under stoichiometric conditions the resulting temperature is around 1940ºC (3500ºF) depending on the temperature of the combustion air. It is necessary to utilize excess of air in the combustion step. Excess air acts as a thermal diluent reducing the temperature of the combustion products, this temperature is set by the material limitations used in the turbine parts exposed to the hot gas .

From a cycle efficiency and engine specific power output (kW per kg/s of suction air flow) standpoint, it is important to minimize the amount of cooling air as well as the excess combustion air.

Air is also used for turbine blade cooling. This creates a large parasitic load on the cycle, since compression of the air requires mechanical energy, thus reducing the net power produced from the system, as well as reducing the overall efficiency of the system.

Gas Turbines

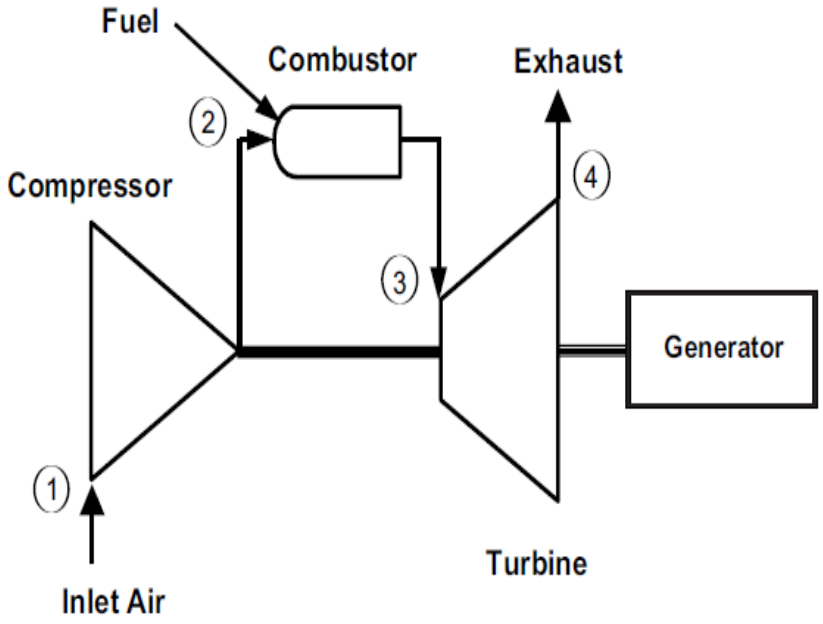


Figure 2. Simple-cycle, single-shaft gas turbine

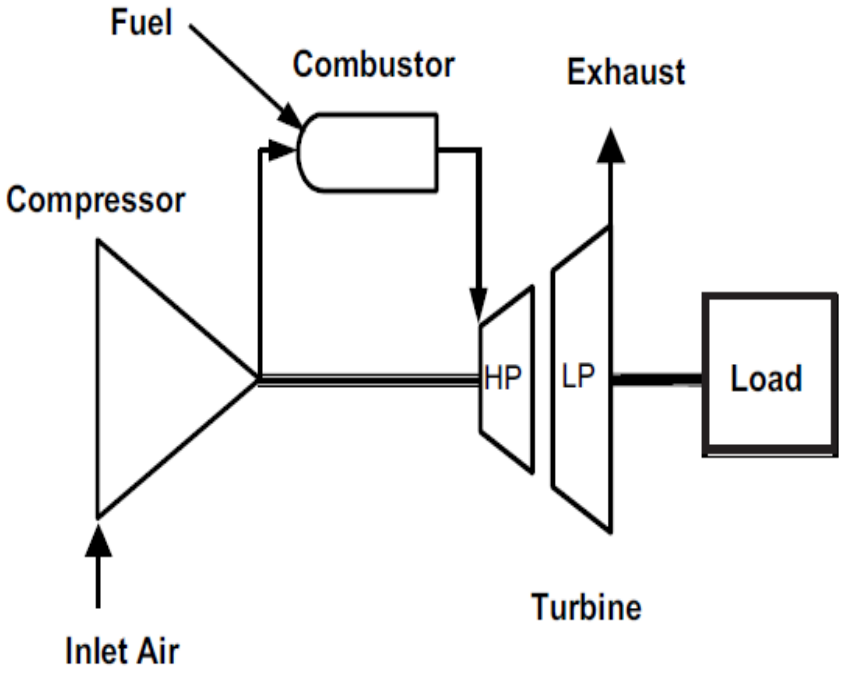


Figure 3. Simple-cycle, two-shaft gas turbine

Gas Turbines

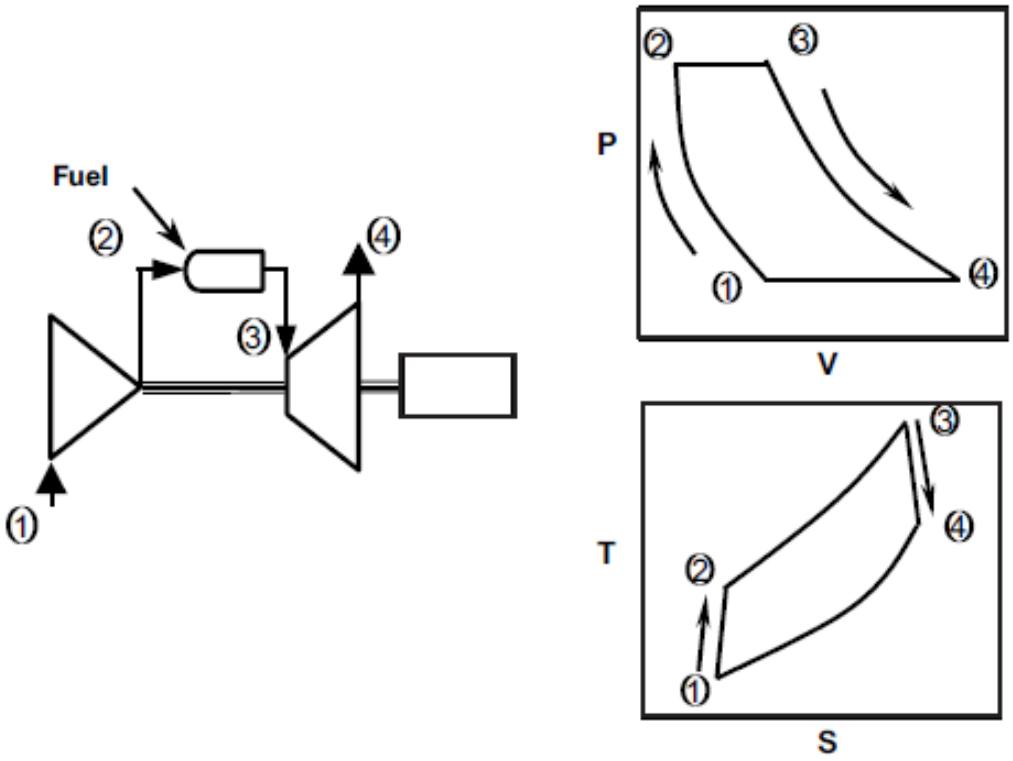


Figure 4. Brayton cycle

Cooling

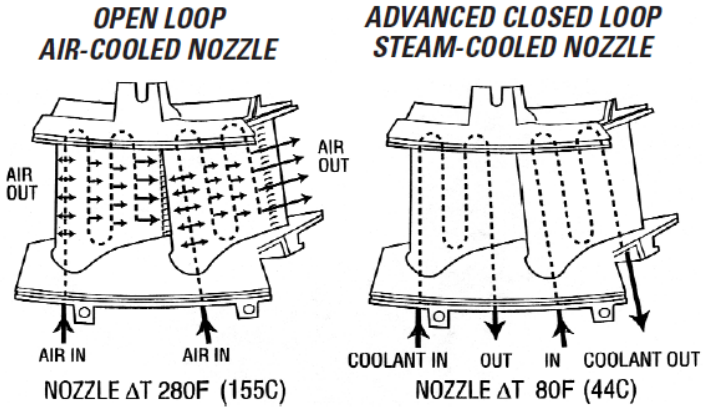


Figure 5. Comparison of air-cooled vs. steam-cooled first stage nozzle

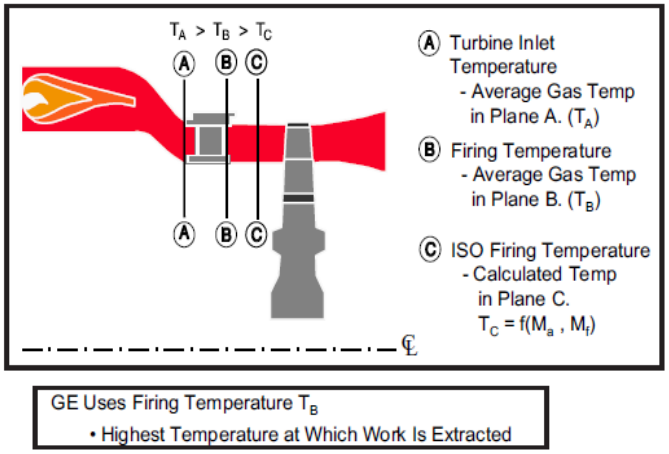


Figure 6. Definition of firing temperature

Gas Turbine Thermodynamics

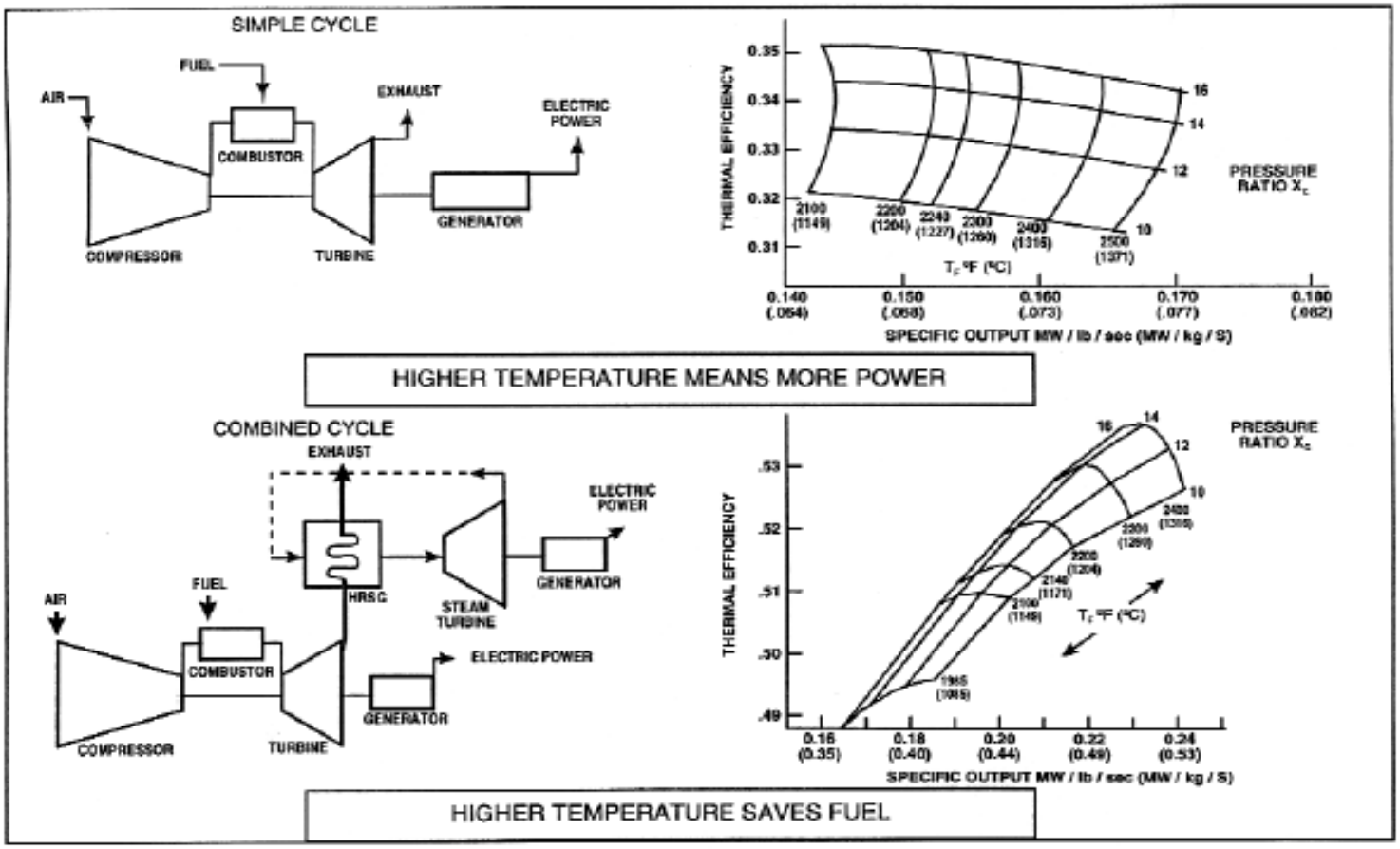


Figure 7. Gas turbine thermodynamics

Combined Cycle Gas Turbines (CCGT)

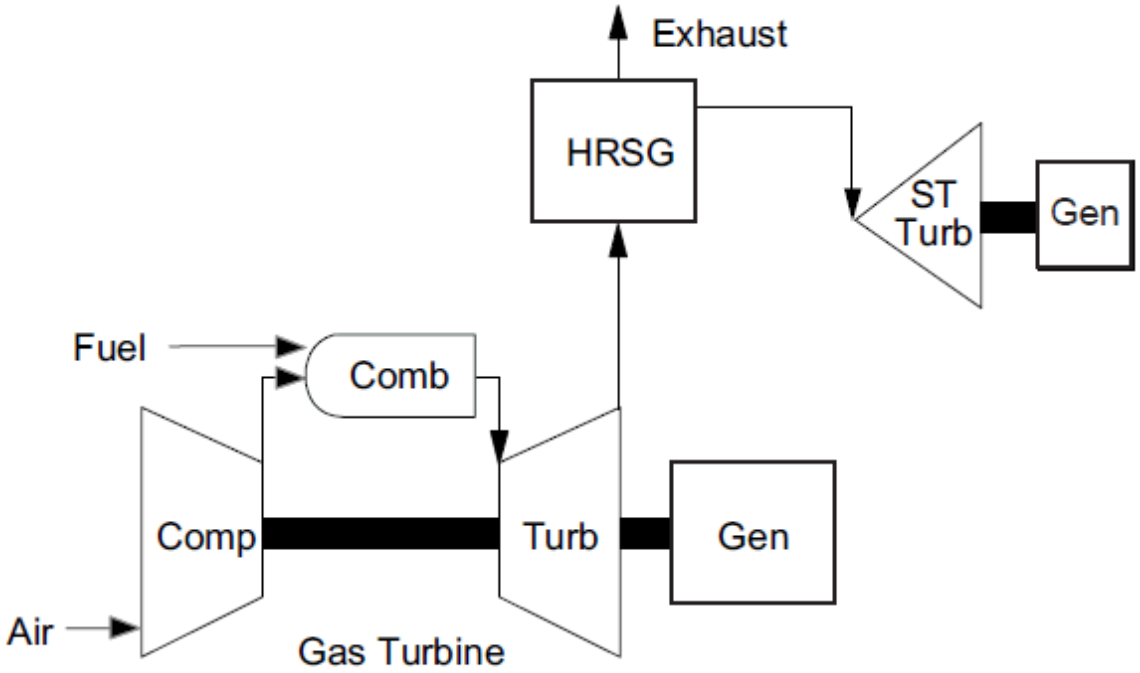


Figure 8. Combined cycle

Comparison of Options

	Single Cycle	CCGTCycle
No. HRSG req'd:	0	3 + 1
No. GTG req'd:	4 + 1	3 + 1
No. STG req'd:	0	1x100%
Heat Input:	136 MW	102 MW
Total Energy Output:	45 MW	50 MW
Efficiency:	33%	48.5%
Fuel Usage (mmscfd)	12.9	8.6
CAPEX (£ Million)	20	34

Additional Sales Gas – 4.3 mmscfd – 43430 therms/day@10p/therm – £1.6 million pa

9 year pay back

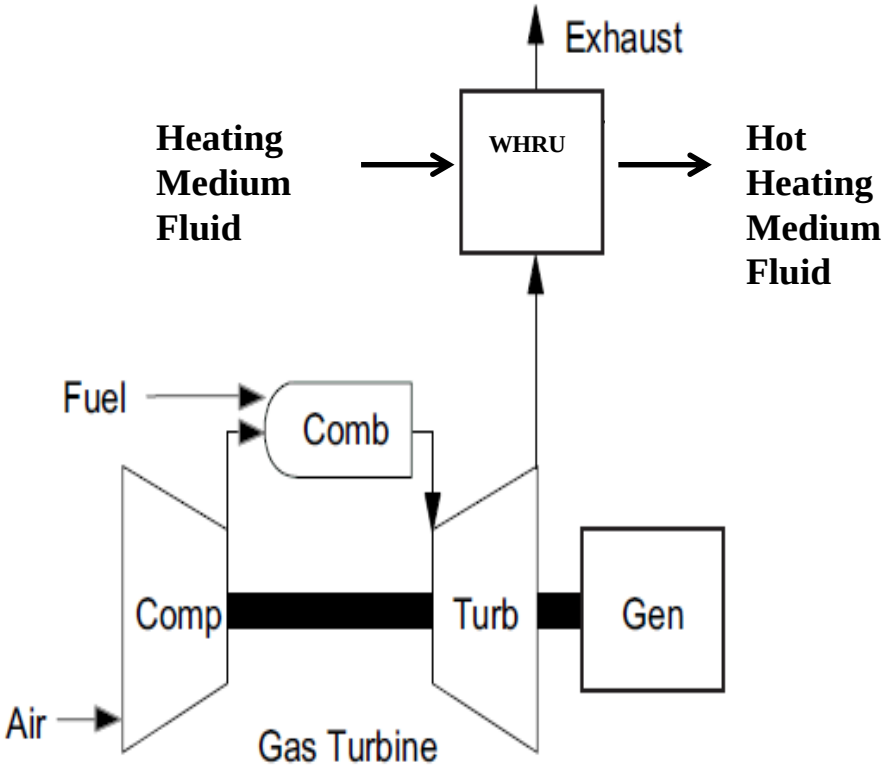
Oseberg and Eldfisk Offshore CCGT Alstom Units



There are no combined cycle gas turbines offshore in the UK. However, as a consequence of Norway's CO2 tax the incentive to deploy CCGT is much greater.



Combined Heat and Power



Hot heating medium fluid can be use as process heat e.g. interstage separator heating for emulsion breaking.

Combined Heat and Power (CHP)

A waste heat recovery unit involves recovering waste heat from the exhausts of power generation turbines or compressors by a closed loop heating mediumor steam generation system. This is essentially the fundamental unit operation for all CHP systems. Available heat is generally assumed to be 1.25 times the power rating.





Use waste heat from GT exhaust gases for process heating.

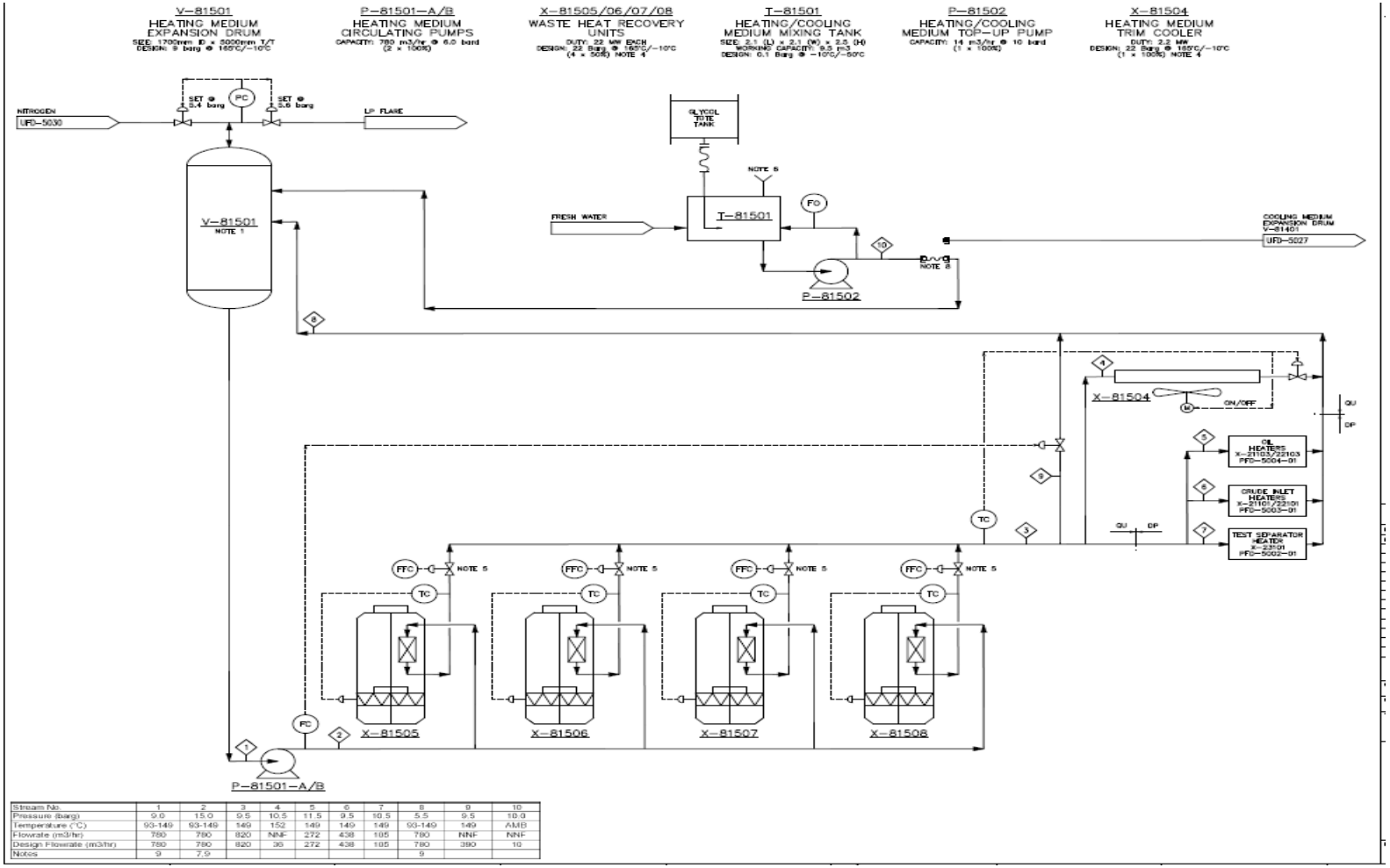
Heat Recovery Unit (HRU)

	3GENERAL INFORMATION											
1	APPLICABLE TO:		μ PROPOSAL		● PURCHASE		μ AS BUILT					
2	Item No.:	**EBN-6150, EBN-6160 Waste Heat Recovery Unit (WHRU) - Located on Production Deck**										
3	Site:	**BP Clair Platform/UK Continental Shelf, West of Shetland**										
4	Service:	**Turbine Generator Exhaust Gas Heat Recovery**										
5	WHRU Mfr.:			Turbine Mfr.:	Solar Industries			Turbine Model No.:	Titan 130 - 19501S			
6	**Note**:	μ Indicates Information to be completed by Purchaser						o By Manufacturer				
	PROCESS DESIGN CONDITIONS											
7	**Fluid Allocation**			**Process Side**				**Exhaust Side**				
8				**Specification**			**Actual**	**Specification**			**Actual**	
9	μ Service			Heat Medium				Gas Turbine Exhaust				
10	μ Heat Absorption	kW		**16 MW**				16 MW (100%)	9.6 MW (60%)			
11	μ Fluid			50% TEG / 50% water				Exhaust Gas (NG Fuel)				
12	μ Flow Rate	(kg/hr)		275,723				174,730	120,268			
13	Press. Drop: μ Allow.	o Calc.	o Max. (bar)	1.9	1.25	1.35		12" H_2O	6"	6"		
14	μ Avg. Flux Density Allowable	($kW/hr-m^2$)										
15	o Avg. Flux Density Calculated	($kW/hr-m^2$)										
16	μ Max. Flux Density, Bare	($kW/hr-m^2$)										
17	μ Velocity Limitation	(m/sec)										
18	o Max. Allow Inside Film Temp.	(ºC)										
19	μ Fouling Factors	(m^2-ºC/W)		0.00035								
	INLET CONDITIONS							**100%**	**60%**			
20	μ Temperature	(ºC)		93.1				494	557			
21	μ Pressure	(barg)		13.36				10" H_2O				
22	μ Liquid Flow	(kg/hr)		275,723				-	-			
23	μ Vapor Flow	(kg/hr)		–				174,730	120,268			
24	μ Liquid Density	(kg/m^3)		1,018				-	-			
25	μ Vapor Density	(kg/m^3)		–								
26	μ Vapor Molecular Weight			–				28.52				
27	μ Viscosity Liquid	(cP)		1.0				-	-			
28	μ Viscosity Vapor	(cP)		–								
29	μ Liquid Specific Heat	(kJ/kg-ºC)		3.60				-	-			
30	μ Vapor Specific Heat	(kJ/kg-ºC)		–								
31	μ Liquid Thermal Conductivity	(W/m-ºC)		0.39				-	-			
32	μ Vapor Thermal Conductivity	(W/m-ºC)		–								
	OUTLET CONDITIONS											
33	μ Temperature	(ºC)		150								
34	μ Pressure	(bar)										
35	μ Liquid Flow	(kg/hr)		275,723				-	-			
36	μ Vapor Flow	(kg/hr)		–				174,730	120,268			
37	μ Liquid Density	(kg/m^3)		960				-	-			
38	μ Vapor Density	(kg/m^3)		–								
39	μ Vapor Molecular Weight			–				28.52				
40	μ Viscosity Liquid	(cP)		0.6				-	-			
41	μ Viscosity Vapor	(cP)		–								
42	μ Liquid Specific Heat	(kJ/kg-ºC)		3.73				-	-			
43	μ Vapor Specific Heat	(kJ/kg-ºC)		–								
44	μ Liquid Thermal Conductivity	(W/m-ºC)		0.39				-	-			
45	μ Vapor Thermal Conductivity	(W/m-ºC)		–								

Heating medium is often a mixture of Tri-ethylene glycol and water.

Glycol prevents both freezing and boiling.

Typical HRU UFD



Stream No.	1	2	3	4	5	6	7	8	9	10
Pressure (barg)	9.0	15.0	9.5	10.5	11.5	9.5	10.5	5.5	9.5	10.0
Temperature (°C)	93-149	93-149	149	152	149	149	149	93-149	149	AMB
Flowrate (m3/hr)	780	780	820	NNF	272	438	105	780	NNF	NNF
Design Flowrate (m3/hr)	780	780	820	36	272	438	105	780	390	10
Notes	9	7,9						9		

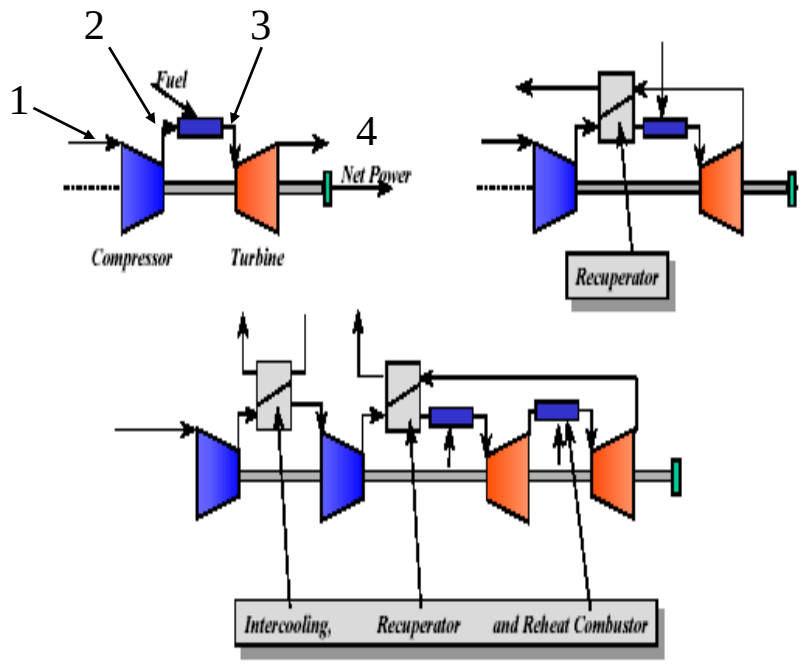
Recuperator – Intercooling- Reheat Thermodynamics

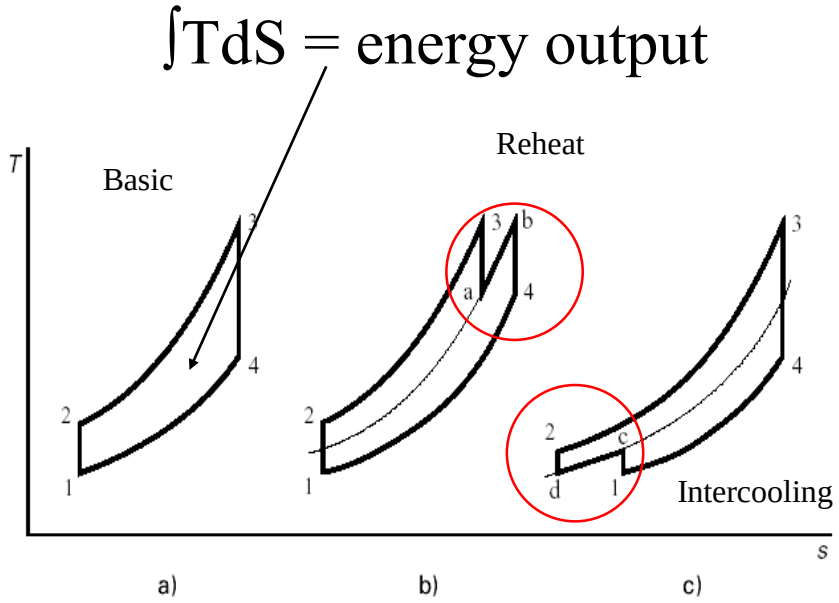
Regeneration involves the installation of a heat exchanger (**recuperator**) through which the turbine exhaust gases pass. The compressed air is then heated in the exhaust gas heat exchanger, before the flow enters the combustor.

Intercooling also involves the use of a heat exchanger. An intercooler is a heat exchanger that cools compressor gas during the compression process. For instance, if the compressor consists of a high and a low pressure unit, the intercooler could be mounted between them to cool the flow and decrease the work necessary for compression in the high pressure compressor. The cooling fluid could be atmospheric air or water (e.g., sea water in the case of a marine gas turbine). It can be shown that the output of a gas turbine is increased with a well-designed intercooler.

Reheating occurs in the turbine and is a way to increase turbine work without changing compressor work or melting the materials from which the turbine is constructed. If a gas turbine has a high pressure and a low pressure turbine at the back end of the machine, a reheater (usually another combustor) can be used to "reheat" the flow between the two turbines. This can increase efficiency by 1-3%. Reheat in a jet engine is accomplished by adding an afterburner at the turbine exhaust, thereby increasing thrust, at the expense of a greatly increased fuel consumption rate.

Recuperator – Intercooling-Reheat Thermodynamics





Factors Affecting Performance

Air Temperature
Site Elevation
Humidity
Inlet and Exhaust Losses

Factors Affecting Performance

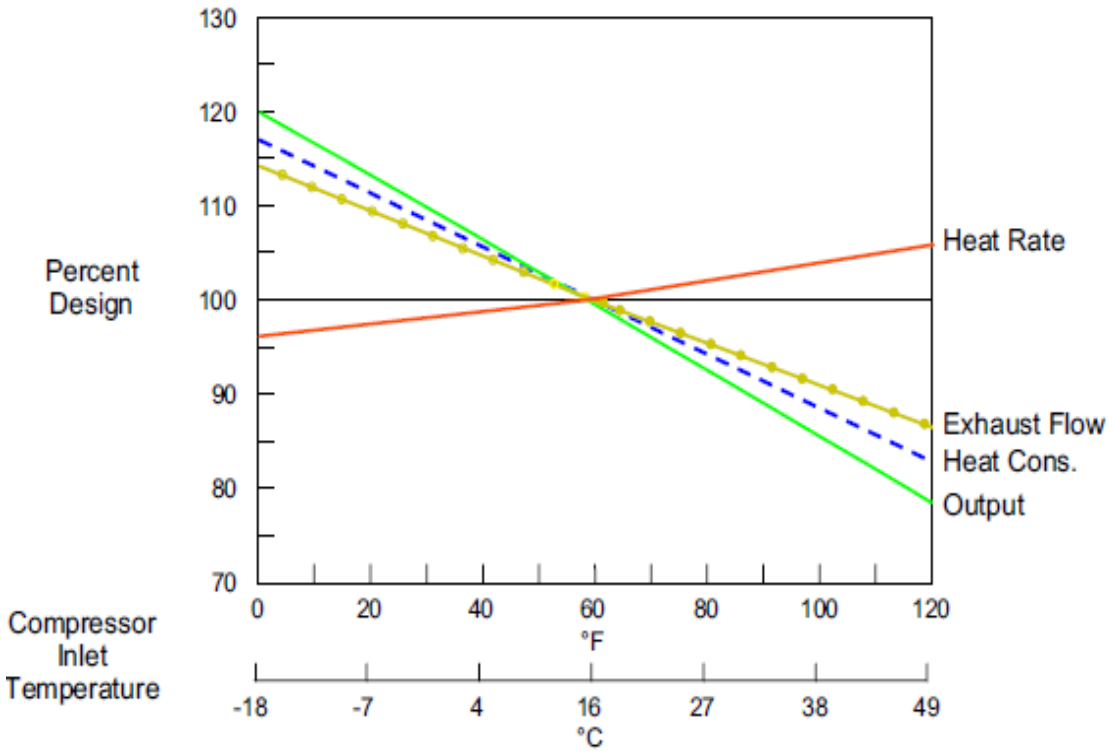


Figure 9. Effect of ambient temperature

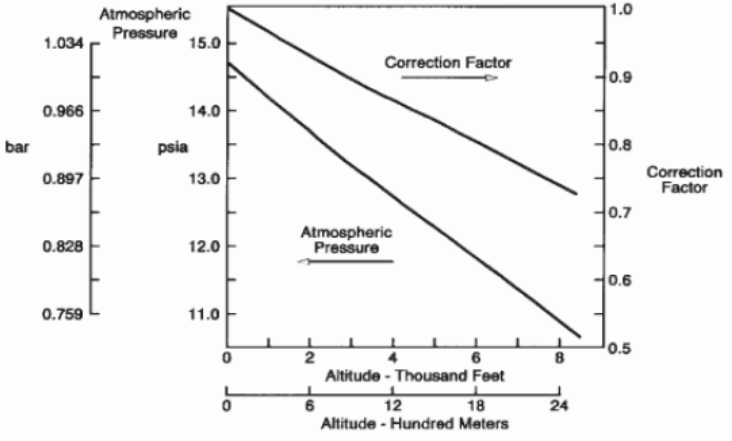


Figure 10. Altitude correction curve

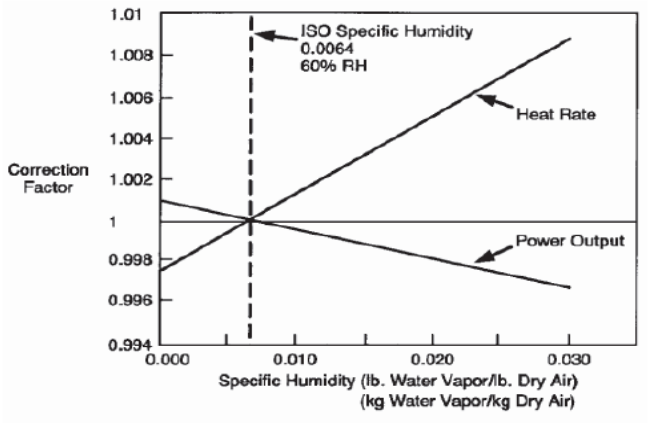


Figure 11. Humidity effect curve

Factors Affecting Performance

Fuels

Several side effects must be considered when burning this kind of lower heating value fuels:

• Increased turbine mass flow drives up compressor pressure ratio, which eventually encroaches on the compressor surge limit

• The higher turbine power may exceed fault torque limits. In many cases, a larger generator and other accessory
equipment may be needed

• High fuel volumes increase fuel piping and valve sizes (and costs).

• Lower calorific gases gases are frequently saturated with water prior to delivery to the turbine. This increases the combustion products heat transfer coefficients and raises the metal temperatures in the turbine section which may require lower operating firing temperature to preserve parts lives.

• As the calorific value drops, more air is required to burn the fuel. Machines with high firing temperatures may not be able to burn low calorific value gases

Combustion Emissions

Carbon MoNOxide emissions occur because of incomplete combustion of fuel carbon. CO emissions for distillate and other liquid fuels are generally higher than for natural gas.

Unburned Hydrocarbons are formed by the incomplete combustion of fuel. They are directly related to combustion efficiency.

Sulphur Oxide discharges are a direct result of the Sulphur in the fuel. The high temperature and oxygen content favour the formation of SO_2.

Particulate Emissions are influenced by the fuel properties and combustion conditions. Particulate emissions tend to be higher when burning liquid fuels.

NOx is usually the most significant pollutant. The quantity of NOx produced is influenced by the firing temperature, compressor outlet temperature, residence time within the combustor and the fuel burned. The trend towards higher efficiencies leads to higher compression ratios and firing temperature, hence higher NOx levels.

NOx reduction is tackled by:

1.Use of a heat sink – water quenching

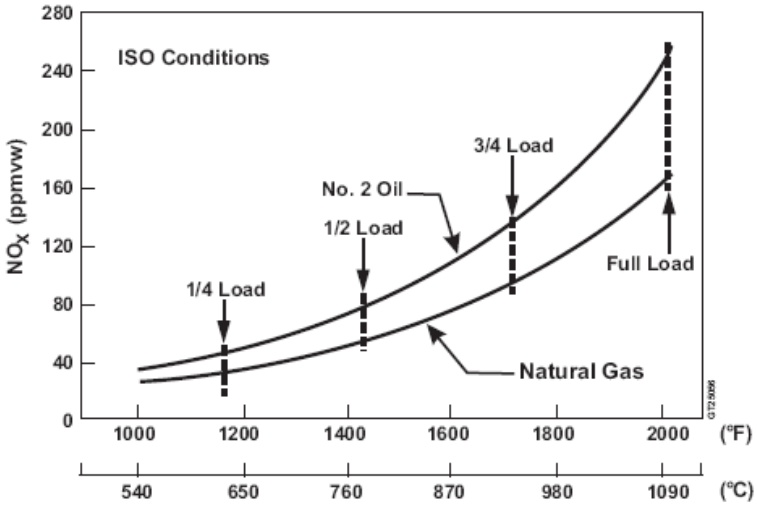
2. Modifications to fuel/air ratios and combustor design

3. Catalytic combustion

Gas Turbines - Typical Emissions

Higher combustion temperature better the efficiency.
Higher combustion temperature higher the NOx emissions.

Major Species	Typical Concentration (% Volume)	Source
Nitrogen (N_2)	66 - 72	Inlet Air
Oxygen (O_2)	12 - 18	Inlet Air
Carbon Dioxide (CO_2)	1 - 5	Oxidation of Fuel Carbon
Water Vapor (H_2O)	1 - 5	Oxidation of Fuel Hydrogen
Minor Species Pollutants	**Typical Concentration (PPMV)**	**Source**
Nitric Oxide (NO)	20 - 220	Oxidation of Atmosphere Nitrogen
Nitrogen Dioxide (NO_2)	2 - 20	Oxidation of Fuel-Bound Organic Nitrogen
Carbon Monoxide (CO)	5 - 330	Incomplete Oxidation of Fuel Carbon
Sulfur Dioxide (SO_2)	Trace - 100	Oxidation of Fuel-Bound Organic Sulfur
Sulfur Trioxide (SO_3)	Trace - 4	Oxidation of Fuel-Bound Organic Sulfur
Unburned Hydrocarbons (UHC)	5 - 300	Incomplete Oxidation of Fuel or Intermediates
Particulate Matter Smoke	Trace - 25	Inlet Ingestion, Fuel Ash, Hot-Gas-Path Attrition, Incomplete Oxidation of Fuel or Intermediates



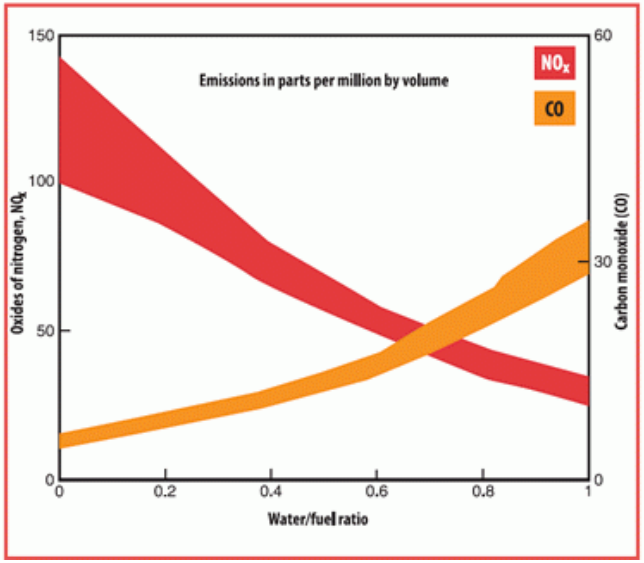
Gas Turbine Low NOx Emissions

Water or steam injection

Injecting water or steam into the turbine combustion chamber limits NOx formation by reducing the average combustion temperature. The typical rate of steam or water injection is 50-100% of the fuel input rate. Although this procedure has the additional advantage of slightly increasing turbine output, there are disadvantages, notably a minor reduction in CHP system efficiency and a possible increase in carbon moNOxide (CO) levels in the exhaust as water or steam levels are increased – the result of partial quenching of the flame.

To avoid turbine damage, the water or steam injected must have a high degree of purity. Even so, injection tends to reduce the life of some of the turbine components and has operating cost implications for the turbine and its associated systems. Gas and gas-oil firing systems can usually incorporate water or steam injection, and virtually all gas turbines can be fitted with this facility.

The equipment needed to treat and inject water or steam into a gas turbine increases the capital cost of a CHP scheme by around 2-3%. Operating costs typically rise by 1-2%.



Low-NOx burners

Low-NOx burners, sometimes known as dry low-NOx (DLN), have been developed for large gas turbines and are now available for most smaller turbines. The burners are designed to operate with a lower temperature flame to reduce NOx emissions. At present, some of these burners can burn only gaseous fuels in low-NOx mode and have no capability to burn gas-oil/diesel as a stand-by fuel. This limits their application on sites that use interruptible gas tariffs and, therefore, have a dual-fuel requirement.

Catalytic control

Current research is seeking to incorporate catalysts into turbine combustion systems but has yet to establish the feasibility, durability and cost-effectiveness of the catalytic control of NOx emissions in gas turbines.

Gas Turbine Data

MANUFACTURER	MODEL	TYPE	ISO RATING (MW)	EFFICIENCY (%)	DIMENSIONS L:W:H (m)	WEIGHT (tonnes)
Solar	Centaur 40	H	3.5	28.5	9.8:2.5:2.6	30
	Centaur 50	H	4.6	29.4	9.0:2.5:2.8	32
	Taurus 60	H	5.5	30.5	11:2.5:3.1	32
	Taurus 70	H	7.5	33	12:2.8:3.1	55
	Mars 90/100	H	9.4 - 10.7	31.5	15:2.8:3.8	70
	Titan 130	H	14	33.5	14:3.1:3.8	75
Alstom Power - (formerly Ruston and EGT)	TA1750	HDI	1.3	18	9.0:2.6:3.1	23
Lincoln Design and Manufacture	TA2500	HDI	1.9	20	9.5:2.4:2.8	25
	TB5000	HDI	4.0	25	9.7:2.4:2.4	34
	Typhoon	LWI	4.35 - 5.25	30.5	8.0:2.4:3.2	36
	Tornado	LWI	6.75	31.5	11.0:2.4:3.3	56
	Tempest	LWI	7.9	31.2	10.75:2.4:3.6	57
	Cyclone	LWI	12.9	35	13.5:2.7:3.9	75
Alstom Power - (formerly ABB Stal)	GT35	H	15 - 20	32	25.0:4.0:4.2	160
Finspong Design and Manufacture	GT10B	LWI	24.8	34	20.5:4.5:5.3	185
	GT10C	LWI	29.1	36	20.5:4.5:5.3	185
	GTX100	LWI	43	37	22.0:4.5:6.0	275
Rolls-Royce/Allison	Avon	A/D	14.58	27	-	165
	RB211	A/D	23.0 - 30.8	36	-	185
	501-K	A/D	3.95 - 5.27	29.5	-	27
	601-K	A/D	6.5 - 7.92	31	-	63
GE	LM1600	A/D	13.74	35	-	-
	LM2500	A/D	22-30	38	-	-
	LM6000	A/D	42	41.5	-	-
	MS5001E (Frame 5)†	HDI	26.3	28.5	21.5:3.28:4.0	288
	MS6001B (Frame 6)	HDI	42.1	32.5	22.6:3.28:3.8	276

- HDI = Heavy Duty Industrial
- A/D = Aeroderivative
- H = Hybrid (mixture of industrial and aeroderivative)
- LWI = Light Weight Industrial

† Frame 5 is a relatively old design which accounts for the lower power/weight ratio than the Frame 6

Gas Turbines - Efficiencies

Aero derivative more efficient than industrial

Larger machine better efficiency

Each machine has a best efficiency point

Life of field considerations

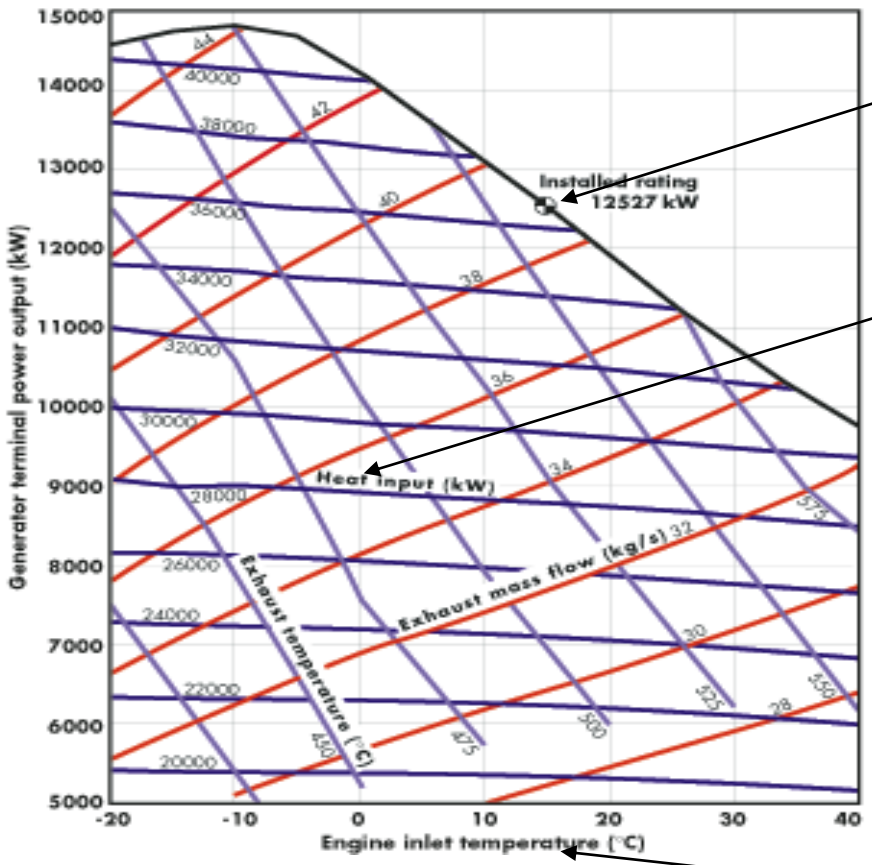
Ambient effects

Spinning reserve

Operating points

Performance degradation

Gas Turbine Power Rating



Full load rating at 15 DegC

Heat input – combustion energy in fuel.

Air temperature

Spinning Reserve

Compare one machine on full load with two Machines at 50% load

Point 1 (100% full load)

- Ambient 10° C
- Generator Power 13200 kW
- Heat Input 38000 kW
- Efficiency 13200/38000 x 100 = 34.7%

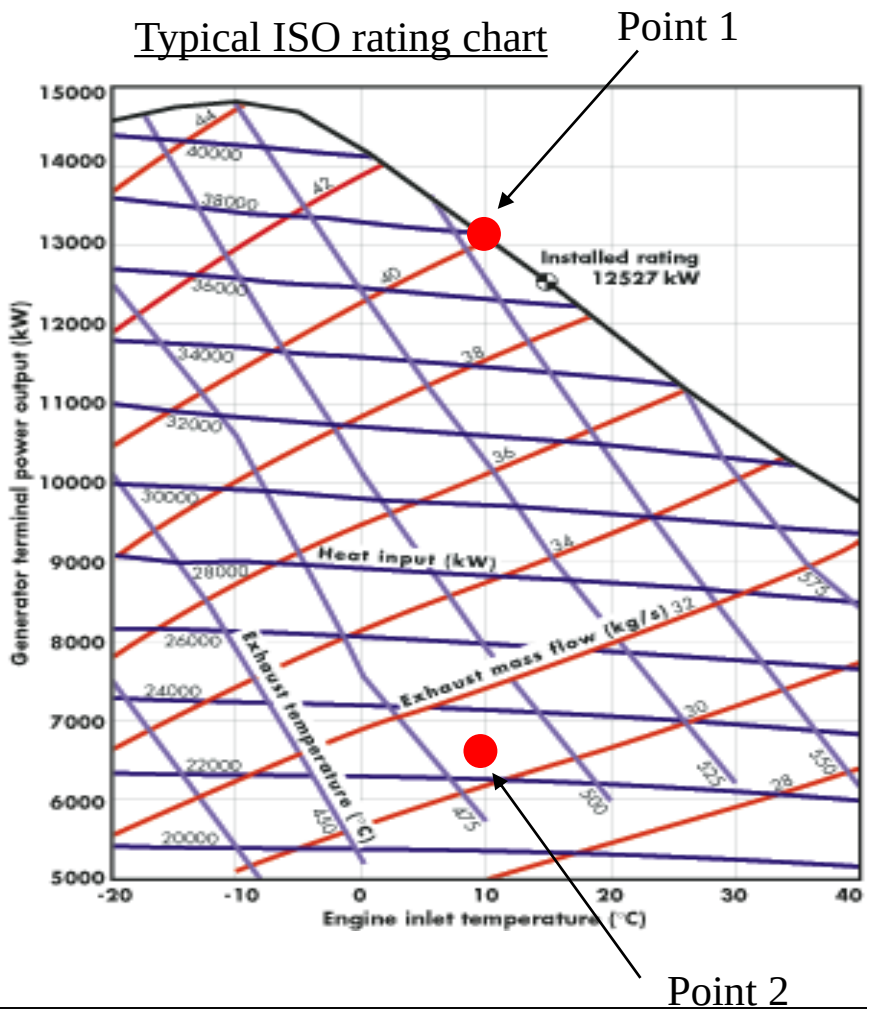
Point 2 (50% thermal load)

- Ambient 10° C
- Generator Power 6600 kW
- Heat Input 22750 kW
- Efficiency 6600/22750 x 100 = 29.0%

Result of 50% loading is;

- 20% increase in fuel mass flow for same total load.

Typical ISO rating chart



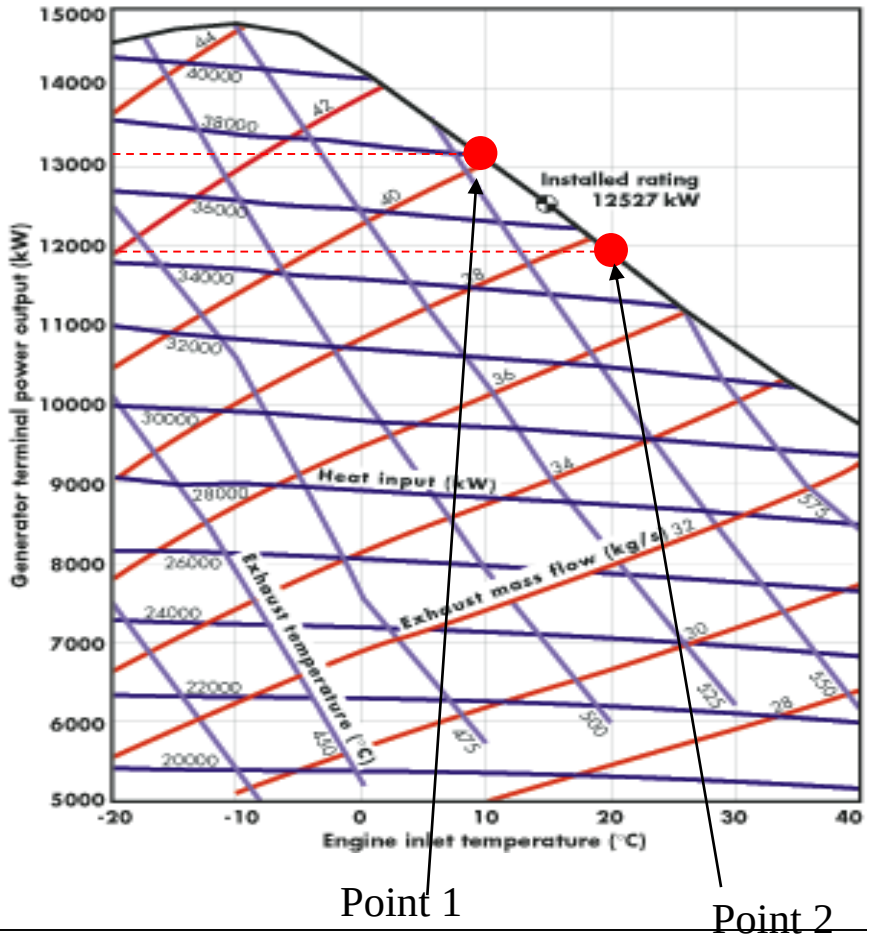
GT Characteristics -Ambient Effects

Efficiency Example (10˚ C Ambient temperature increase)

- Point 1- Full Load
  - Ambient 10˚ C
  - Generator Power 13200 kW
  - Heat Input 38000 kW
  - Thermal Efficiency = 13200/38000 = 0.356

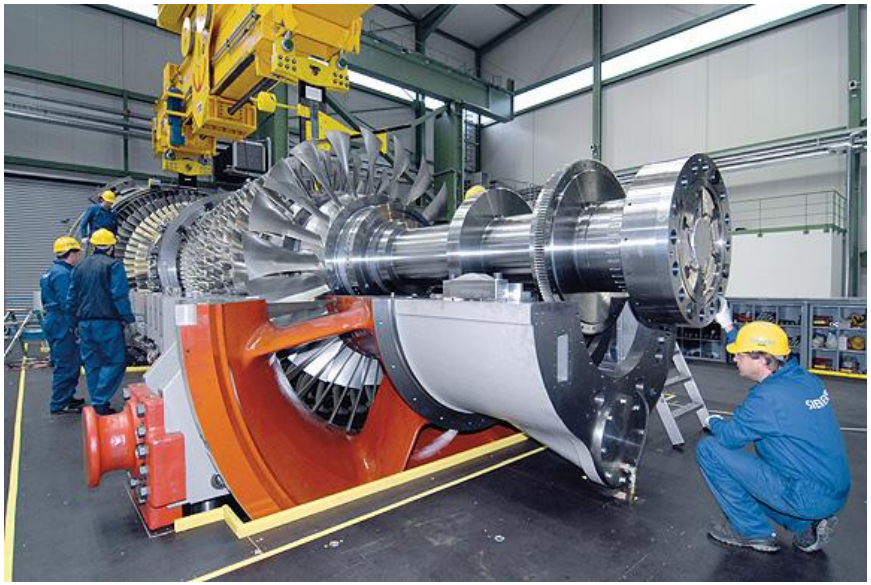
- Point 2
  - Ambient 20˚ C
  - Generator Power 11950 kW
  - Heat Input 35200 kW
  - Thermal Efficiency = 11950/35200 = 0.339

Typical ISO rating chart



Emerging Technology





Designed to work in a "combined cycle" power plant the *H System*™ will be the most efficient power generation system in the world. It will be the first gas turbine to top the 60 percent efficiency threshold -- the "four minute mile" of turbine technology. When the Energy Department began its advanced turbine development program in the early 1990s, the best turbines available had efficiencies of about 50 percent.

The efficiency gains have been achieved because the turbine fires natural gas nearly 170 ^{o}C hotter than conventional turbines, reaching temperatures of 1430 ^{o}C. Advanced cooling techniques and new alloys were developed to handle the hotter temperatures. The turbine also employs the world's largest single crystal aerofoils, making the turbine blades much more resistant to high temperature cracking than the multi-directional crystal design currently used.

H System – Baglan Bay

Comparison of combined cycle plant performance characteristics				
	7FA	7H	9FA	9H
Firing temperature (°C)	1300	1430	1300	1430
Air flow (kg/s)	444	558	640	685
Compressor compression ratio	16	23	17	23
Specific work (MW/kg/s)	0.58	0.72	0.61	0.70
Combined cycle net output (MWe)	260	400	390	480
Net thermal efficiency (%)	56	60	56.7	60
NO_x (ppmvd at 15% O_2)	9	9	9	9





The world's largest gas turbine, the 9H is 12 meters long, 5 meters in diameter, weighs 370 tons and is rated to produce 480 megawatts of electricity in combined-cycle operation, enough power for approximately 600,000 homes. The 9H technology was introduced in 1995, and the first 9H machine completed extensive testing in Greenville prior to being shipped to South Wales.
The H System combines the gas turbine, a steam turbine and a heat recovery steam generator into a seamless, combined-cycle system with each component optimized for the highest level of performance. The use of advanced materials and an innovative steam cooling system enables the H System to operate at the higher firing temperatures (1430 degrees C class) required for increased thermal efficiency.
The H System burns natural gas, a much cleaner fuel than other fossil fuel options. This technology, in combination with the steam cooling and high efficiency, is capable of achieving very low emissions. Although the new Baglan Bay plant (South Wales) is far larger than the oil-fired plant currently operating at the site, it will produce far lower emissions, reducing carbon dioxide emissions by 66%, nitrous oxide by 88% and sulphur dioxide by 99%.

Load List

It is important to identify the utilisation voltage of each piece of equipment in order to properly design the electrical system.

An electrical load list is the best way to determine the overall system requirements in terms of the operating load.

The load list should include the power rating of each individual consumer and whether it is run continuously, intermittently or as standby for another consumer.

Once this has been accomplished, local loads may be grouped to be served from switchgear which in turn are served from transformers.

When generating power, the starting loads need to be considered in addition to the total steady state load. Starting loads of three to five times full load are often required.

Load List

Description	Power (kW)	
	Continuous	Intermittent
MOL pumps	1800	-
Crude export pumps	-	3200
Drilling	-	3160
Separation	382	-
Compression	5000	600
Gas dehydration	350	-
Water injection	2500	-
Seawater lift pumps	380	190
Seawater treatment	50	50
Fresh water	-	80
Electrochlorinator	250	-
Fuel gas	150	-
Cooling medium	160	-
Heating medium	150	-
Relief system	25	25
Closed drains	20	-
Instrument air	360	180
Fuel oil	50	-
Chemical injection	100	150
Miscellaneous	2000	1000
Total (MW)	13.727	8.635

Typical Electrical Load List

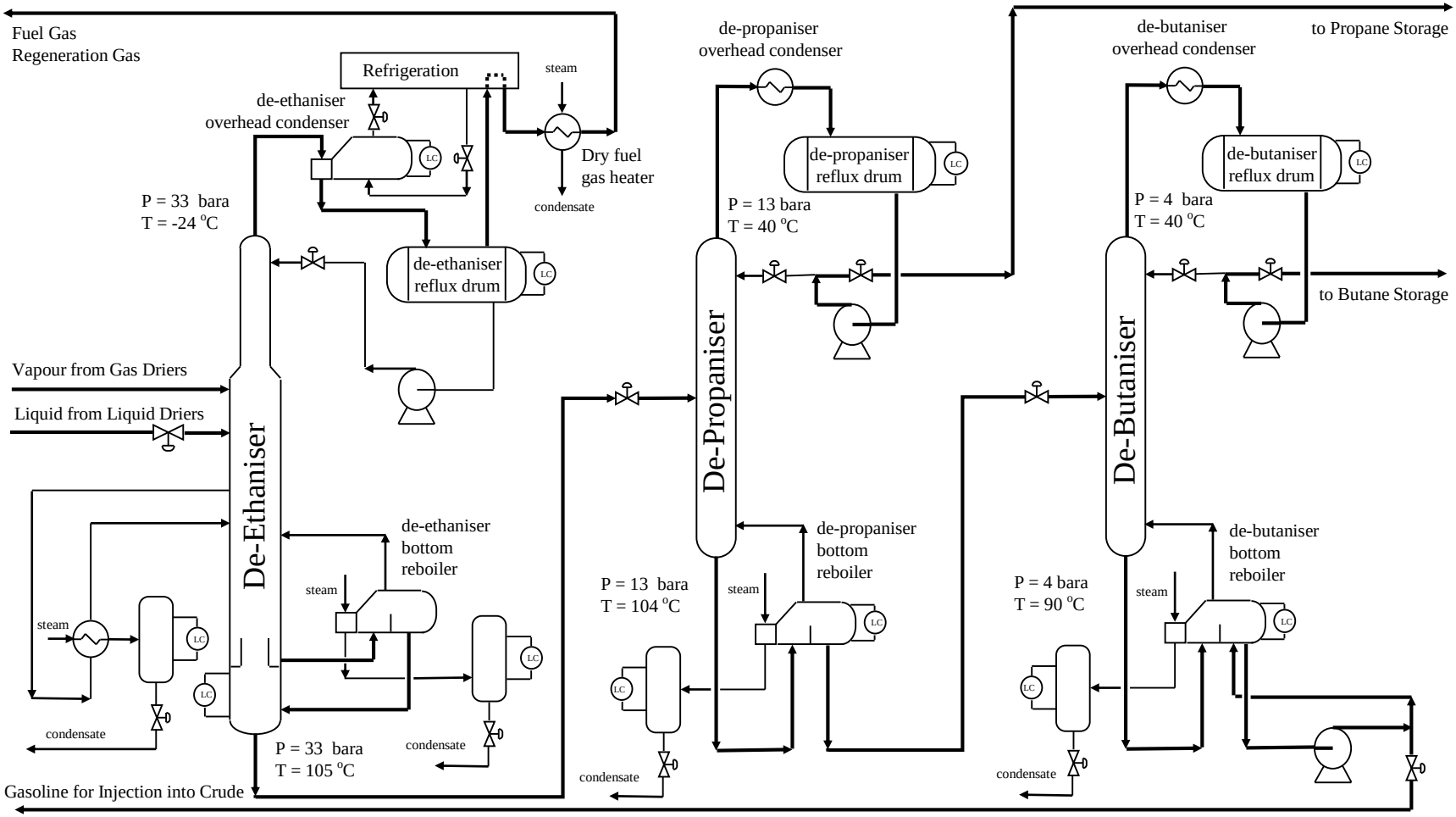
Fractionation

The fractionation process is commonly seen at an oil terminal and refinery facilities. Its purpose is to recover valuable light-end constituents of the gas mixture such that the required LPG specifications can be met.



Fractionation is the one of the most common separation techniques. It consumes significant amounts of energy, in terms of cooling and heating requirements and may contribute to as much as 50% of plant operating costs. Therefore the best way to reduce operating costs of existing (or new) units, is to improve their efficiency and operation via process optimisation and control. To achieve this improvement, a thorough understanding of fractionation principles and column design is essential.

Typical LPG Fractionation



Fractionation

Key Parameters – Operational Conditions

The type of cooling medium available determines the operating pressure range of the column

To maximise the relative volatility low pressure operation is favoured. However reducing the pressure may lead to a more expensive cooling medium

Increasing pressure may exceed the critical temperature of the bottom product. This results in the failure to achieve product specification

Cooling medium could be air, water or refrigerant. The choice of medium depends on the required temperature within the unit

Fractionation

Key Parameters – Reflux Ratio & Number of Stages

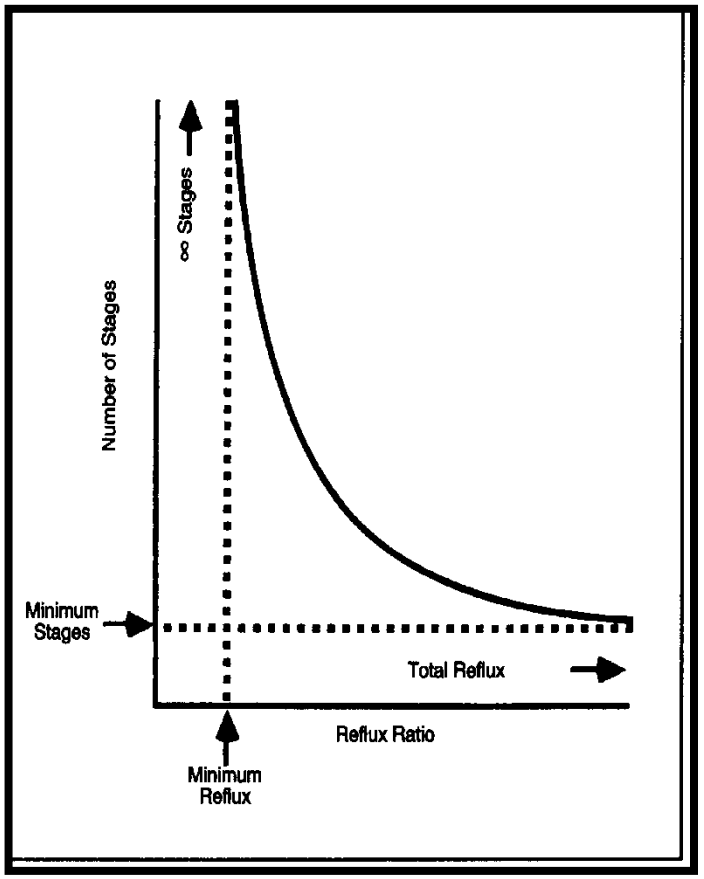
The reflux ratio and the number of stages are the primary parameters in the capital cost versus energy efficiency

Reflux ratio is defined as the ratio of molar rate of reflux liquid divided by the molar rate of net overhead product

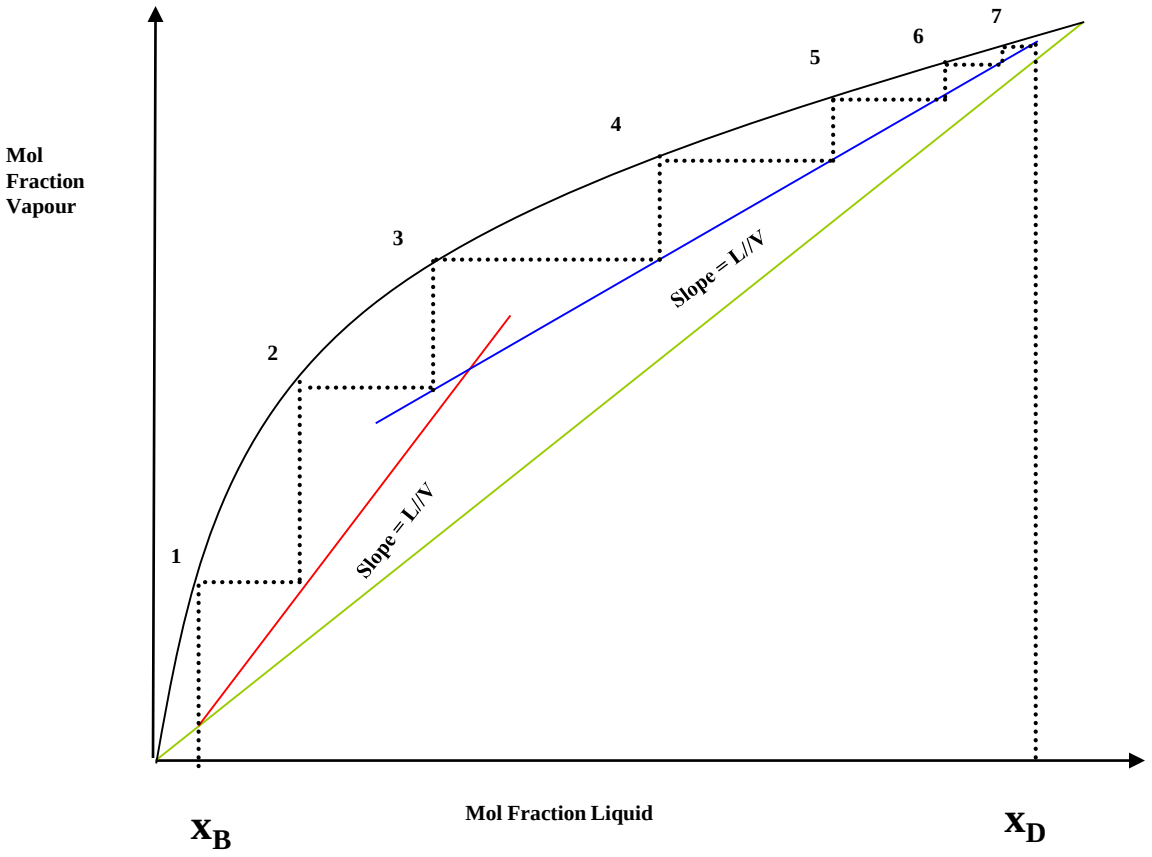
Reboiler duty is a direct function of the reflux ratio as the fractionator is designed to achieve an overall heat & mass balance

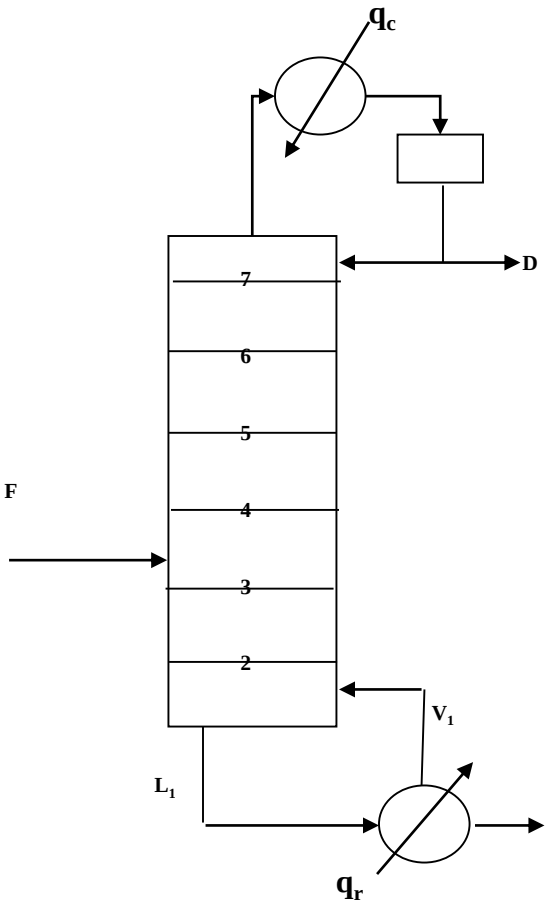
Fractionator column can only produce a desired separation between the limits of minimum reflux and minimum stages

At minimum reflux an infinite amount of stages is required. At maximum reflux a minimum amount of stages is required

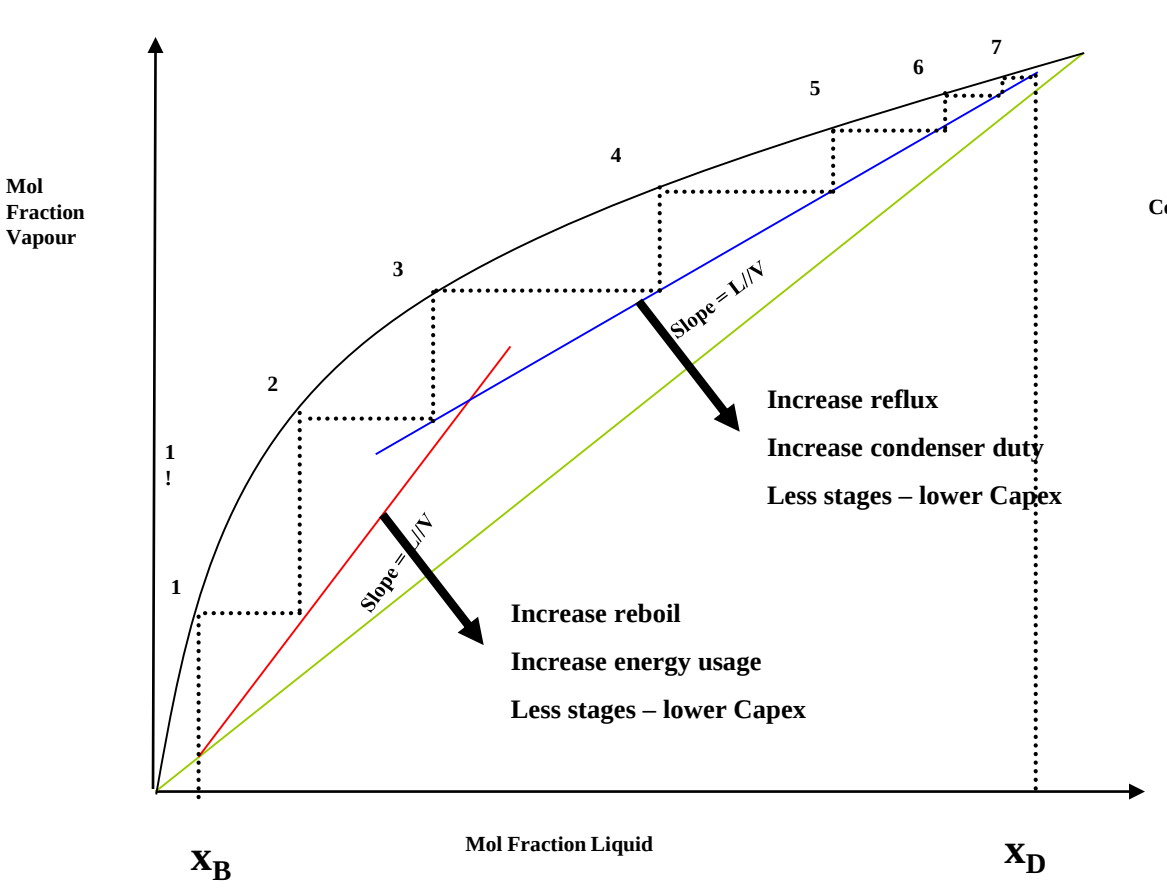


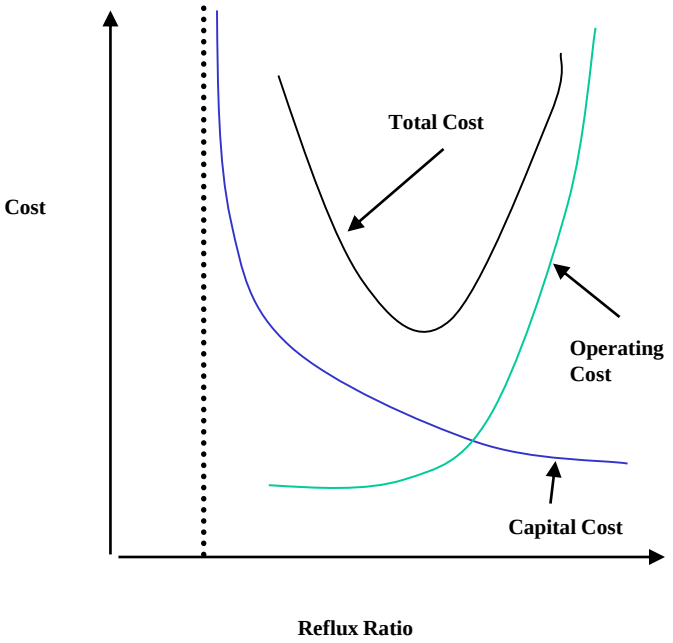
Fractionation





Fractionation





Fired Heaters

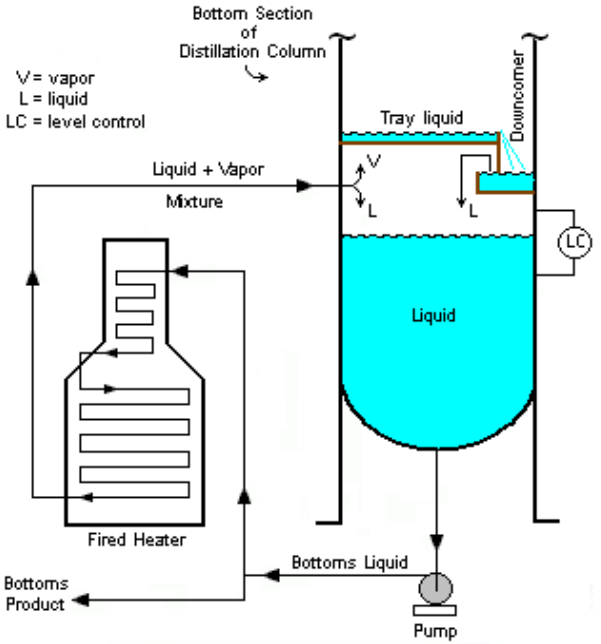
Fired heaters are used to transfer heat directly to the process fluid, they generally have a large duty, but ranging from a few 10s of kW to 60000 kW

There are two basic configurations, direct fired and indirectly fired

A fired heater consists of the following

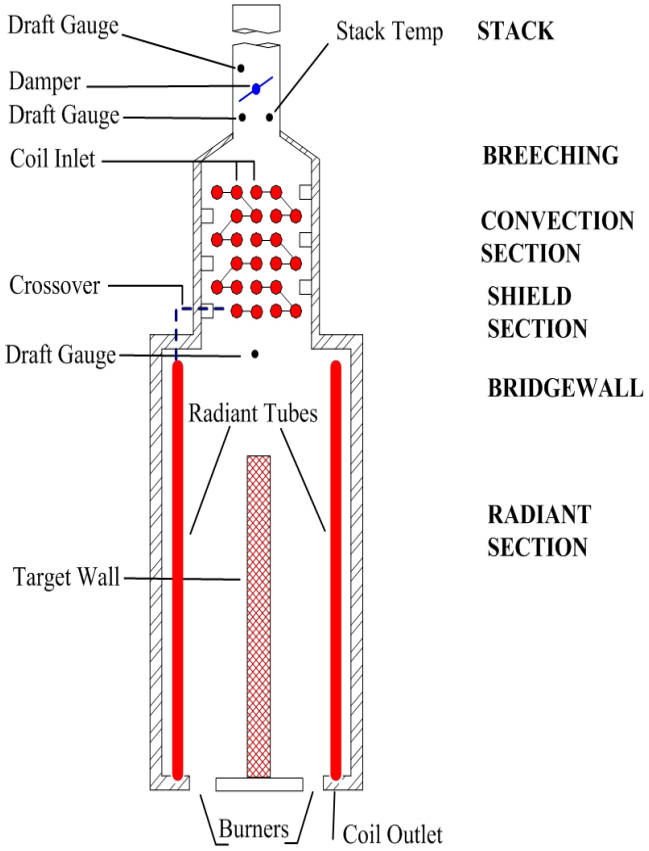
- A combustion chamber lined with refactory and burners
- Tubes located within the combustion chamber where heat is transferred to the process fluid by radiation
- Tubes located external to the combustion chamber in a convection zone which is also lined with refactory
- Stack for disposal of flare gas
- Air supply system by fan or induced draft
- Instruments and controls

Typical applications of fired heaters include boilers, onshore glycol and amine regenerators and heating medium heaters





Fired Heaters



Radiant Section: The radiant tubes, either horizontal or vertical, are located along the walls in the radiant section of the heater and receive radiant heat directly from the burners or target wall. The radiant zone with its refractory lining is the costliest part of the heater and most of the heat is transferred here. This is also called the firebox.

Convection Section: The feed charge enters the coil inlet in the convection section where it is preheated before transferring to the radiant tubes. The convection section removes heat from the flue gas to preheat the contents of the tubes and significantly reduces the temperature of the flue gas exiting the stack. Too much heat picked up in the convection section is a sign of too much draft. Tube temperature readings are generally taken in both convection and radiant sections.

Shield Section: ust below the convection section is the shield (or shocktube) section, containing rows of tubing which shield the convection tubes from the direct radiant heat. Several important measurements are normally made just below the shield section. The bridgewall or breakwall temperature is the temperature of the flue gas after the radiant heat is removed by the radiant tubes and before it hits the convection section. Measurement of the draft at this point is also very important since this determines how well the heater is set up. This is also the ideal place for flue gas oxygen and combustibles measurement.

Stack and Breeching: The transition from the convection section to the stack is called the breeching. By the time the flue gas exits the stack, most of the heat should be recovered. Measurement of stack emissions for compliance purposes is normally made here.

Fired Heaters – Efficiency Considerations

Efficiency: The thermal efficiency is simply E = Heat absorbed/Fuel calorific value

Excess Air: A higher combustion air rate is necessary than theoretically required for complete combustion of the fuel. This caused by variations in the distribution of air and fuel to the individual burners, together with imperfect mixing of air and fuel in the burner and the flame. However no more air excess air should be supplied to that actually required as any additional air must be heated up hence reducing efficiency. This is generally around 10% excess air.

Burners: Burners are classified according to the type of fuel combusted: gas, liquid or a combination. The air-fuel mixing efficiency is a function of the burner design. The better the mixing the better the efficiency. However this comes at a cost of more expensive burners.

Air Preheat: Efficiency can be improved by preheating the combustion air. This is accomplished by using the flue gas to heat the incoming air.

Fired Heaters – Data Sheet

CHARACTERISTICS

Absorbed heat (Maximum)	MW	14.38 (1)	Max allowable heat flux	kcal/h.m²	(3)

OPERATING CONDITIONS

Run case			Heater Design Case	
Absorbed Duty (Total)		MW	14.38	
			Inlet	Outlet
Fluid name			FEED (HC+H2+H2S)	
Fluid quantity, total		kg/h	329,205	
Vapor (Total)		kg/h	98,215	224,853
Liquid (Total)		kg/h	230,991	104,352
Steam		kg/h		
Free Water		kg/h		
Non condensable (H2) (mole fraction on total inlet fluid)				
Vaporized weight		%	29.8	68.3
Temperature		°C	308.1	339.9
Vapor (10)	Density	(PT) kg/m3	See enthalpy tables HC 82-F1601 Heater Design Case HC 82-F1601 - 37.3 BAR G HC 82-F1601 - 36.3 BAR G HC 82-F1601 - 35.3 BAR G HC 82-F1601 - 34.3 BAR G HC 82-F1601 - 33.3 BAR G	
	Viscosity	cP		
	Molecular weight	kg/kmol		
	Specific heat	kcal/kg.°C		
	Thermal cond.	kcal/h.m.°C		
Liquid (10)	Density	(PT) kg/m3		
		cP		
	Specific heat	kcal/kg.°C		
	Thermal cond.	kcal/h.m.°C		
Inlet pressure		bar g	37.3	
Pressure drop	(allow)	bar	4.0 (2)	

CONSTRUCTION

Design pressure	bar g	50.0 (6)
Tube material (process side)		SS 321 H / SS 347 H (14)
Corrosion allowance	mm	1.5
Fuel		FUEL GAS (7) (16)

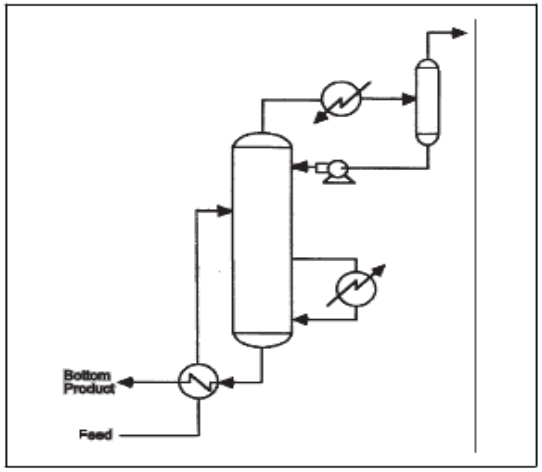


Fractionation

Feed/Product Exchangers

One of the simplest ways to reduce the reboiler fuel requirement is to preheat a liquid feed stream. This can be accomplished with a feed/product exchange as shown. In general this heat input will decrease the reboiler duty. However, since the feed is now partially vaporized, the overhead condenser duty will tend to increase . This increased condenser duty must be offset by reboiler duty. The net reboiler savings will be close to, but not equal to, the heat input to the feed. The net effect will depend on many system parameters; but feed/product exchange is generally an attractive heat conservation application.

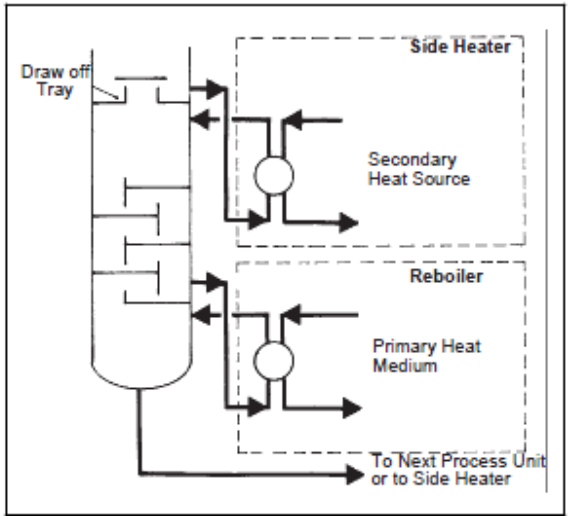
Example Feed/Product Exchanger



Side Heaters

Side heaters can be used to add heat to a tower several trays up from the reboiler. Because of the temperature gradient in the column, this heat is applied at a much lower temperature than the reboiler. The heat source for this side heater can be any stream which requires cooling and is at a high enough temperature level to be useful. Often, the bottom product is used to side-heat the column. In cryogenic plants, the feed gas often supplies the reboil heat.

Example Side Heater



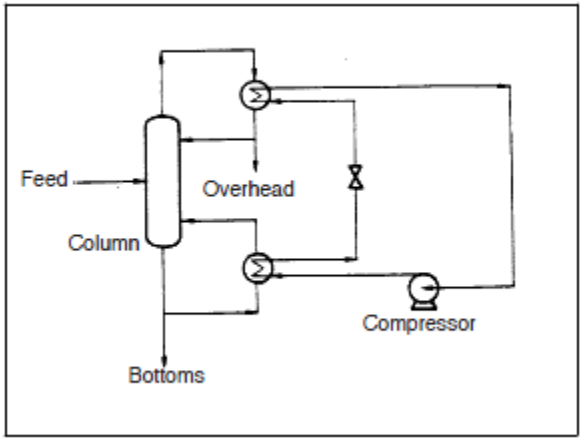
Fractionation

Heat Pumping

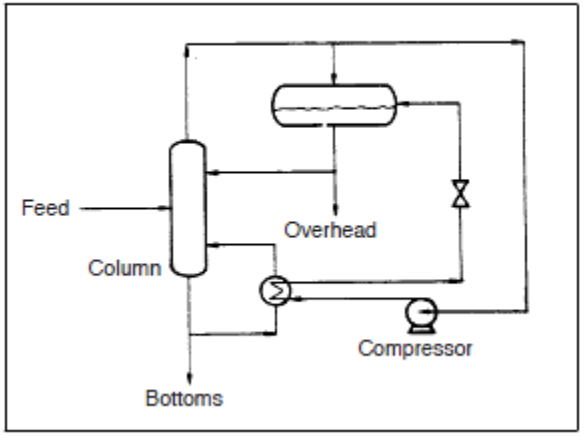
One technique for energy conservation in fractionation systems is the use of a heat pump. Heat pumping usually employs an external working fluid as shown . Compression is used to raise the temperature of the working fluid above that required for the reboiler. The fluid leaving the reboiler is then flashed and used to condense the reflux. The net result is that the heat absorbed in the condenser is used to reboil the column. The main operating cost then becomes the compressor rather than the normal heating and cooling utilities.

An alternative to the basic heat pump is to use the column overhead as the working fluid. This alternative, vapor compression, eliminates the overhead condenser . It is often difficult to find a working fluid to reboil and condense in a single fractionator. However, often plants have several fractionators with condensers and reboilers at a variety of temperatures. It may be possible to link a condenser and reboiler from separate columns.

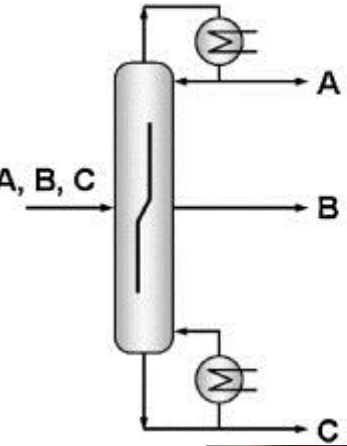
Heat Pumping

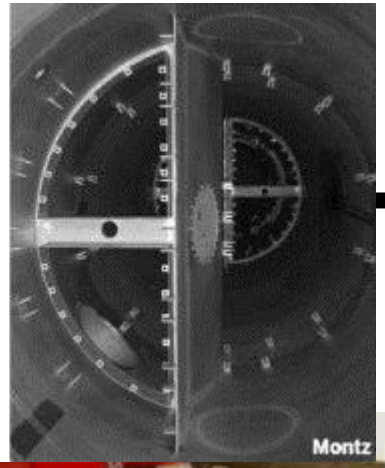


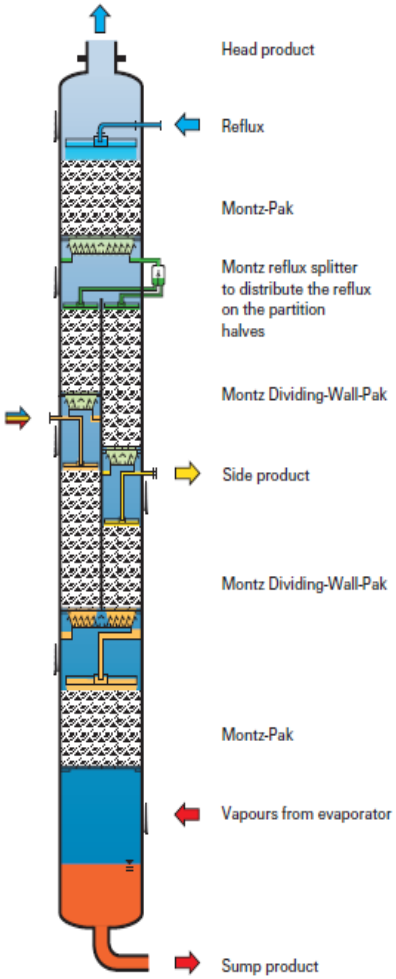
Vapor Recompression



Divided Wall



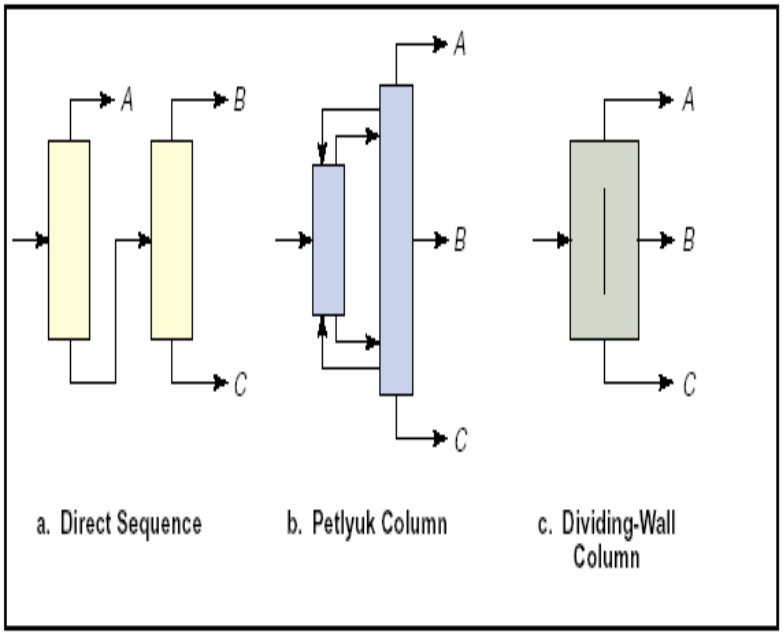


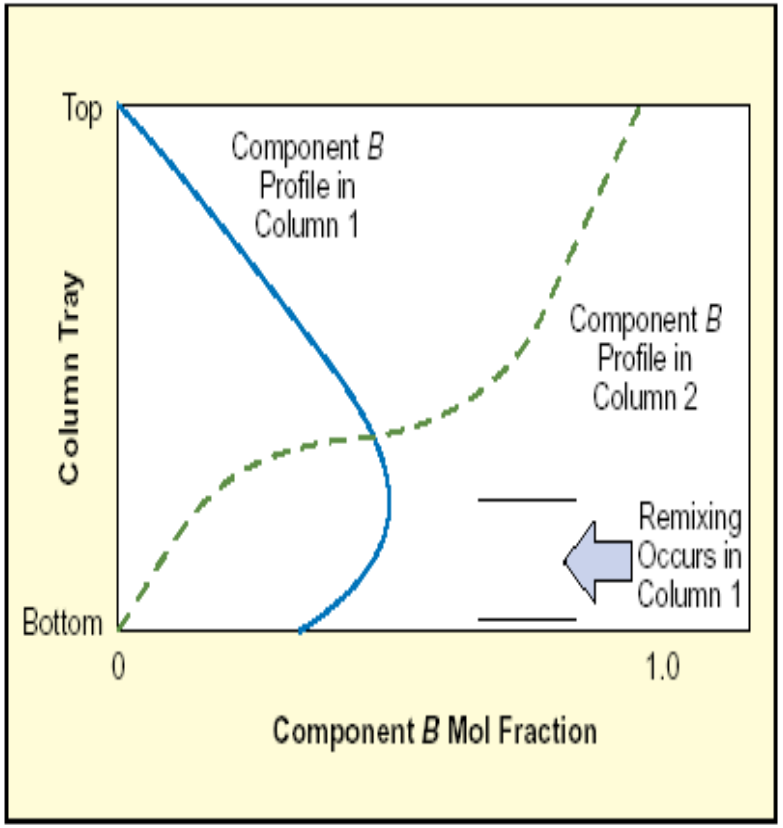






Divided Wall Column





Direct Sequence – concentration of component B builds to a maximum at a tray near to the bottom

Below this C continues to increase diluting B- remixing is thermally inefficient.

Key Learnings

- Organic Rankine Cycle
- Use of Mollier charts
- Compressor thermodynamics
- Factors affecting efficiency of reciprocating and centrifugal compressors
- Flare gas recovery
- Optimisation applications
- Energy efficiency associated with TEG dehydration
- Gas turbine thermodynamics – Brayton Cycle
- Simple, combined gas turbine cycles and combined heat and power
- Factors affecting gas turbine efficiency
- Use of gas turbine ISO rating charts
- Life of field electrical load list
- Fired heater principles
- Energy usage in fractionators

Fractionation
Fenske-Underwood-Gilliland Method

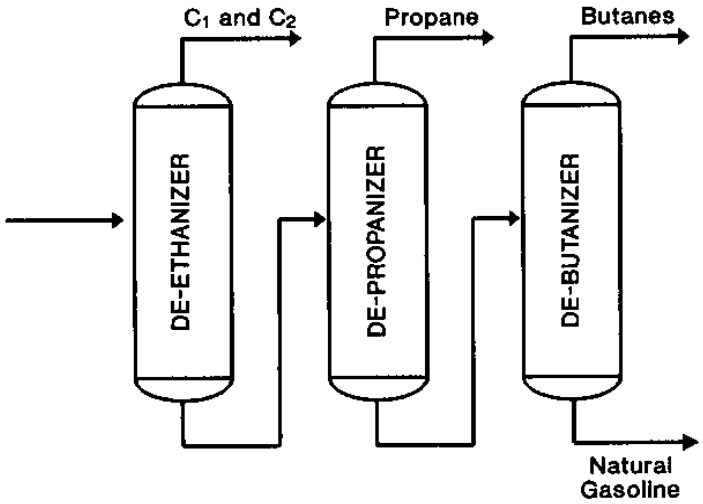


Figure 1: Series of Fractionator Towers

In hydrocarbon processing, it is often necessary to separate components in the raw feed stream into saleable products. This is can be achieved using process equipment known as fractionation columns. Each fractionation column can produce one refined component stream from a mixture of the others, therefore a number of sequential columns may be necessary depending on the number of components present in the initial feed (Figure 1).

Nomenclature

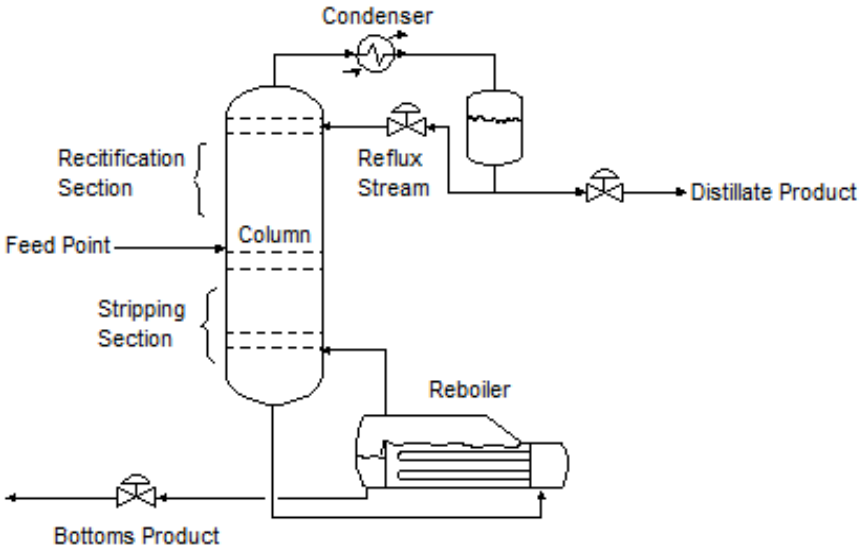


Figure 2: Elements of a Fractionator

A fractionation column consists of the following:

1.Column with trays: each tray is where vapour-liquid equilibration is allowed and where separation occurs.

2.Feed point: where the material to be separated enters, the position of this depends on the characteristics of the feed material.

3.Condenser: this condenses the vapour product from the column.

1. If the distillate product leaves as a saturated vapour it is a partial condenser (reflux is still condensed).
2. If the distillate product leaves as a saturated liquid it is a total condenser

4.Reflux Stream: this is condensed product which is sent back down the column to allow vapour-liquid contact at the trays, which in turn enhances separation.

5.Distillate Product: this is the top product of the column which can either be a saleable product or sent on for further processing.

6.Reboiler: this operates at the bottoms product bubble-point and sends vapourised components back up the column to allow for vapour-liquid equilibration and separation.

7.Bottoms Product: this is can be a saleable product or can be sent on for further processing.

The section of the column above the feed point is referred to as the rectification section, while the section below is known as the stripping section.

Fenske-Underwood-Gilliland Method

Rigorous mechanical design of fractionators is a complex task due to the number of variables involved and specialty knowledge required. Therefore this is usually completed by the equipment vendor, however, specification by the operating engineer is made through more empirical, short-cut calculations. One such method is the Fenske-Underwood-Gilliland method, which is presented in the following slides. The overall procedure is given below:

1. Specify feed stream characteristics: composition, flow rate, temperature, pressure.
2. Carry out a mass balance on the column based on feed composition and product specifications.
3. Calculate condenser temperature.
4. Calculate bubble-point pressure of the distillate product.
5. Assume no pressure difference across column, and calculate bubble-point temperature of the bottoms product.
6. Calculate minimum number of trays and minimum reflux rate for specified product purities.
7. Calculate actual number of trays and reflux rate.
8. Carry out column energy balance, and determine condenser and reboiler duties.
9. Size the column.

Tower Mass Balance

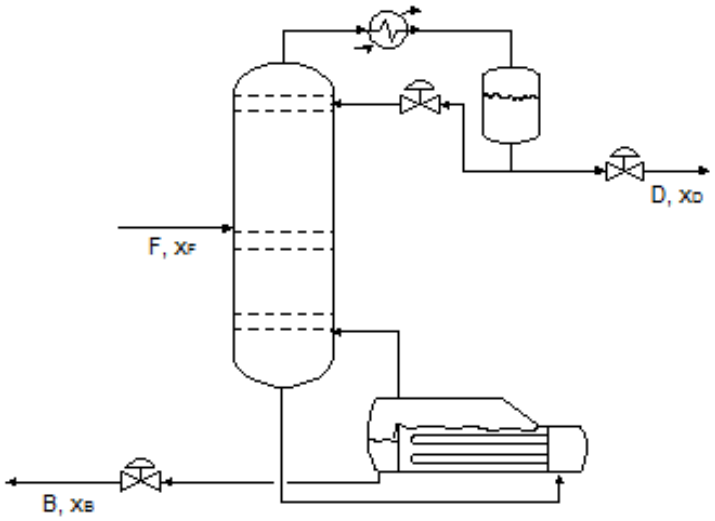


Figure 3: Mass Balance on Fractionator

Once the feed and product specifications are defined, a mass balance is completed on the fractionating column. From Figure 3, the following relationships can be derived (Equation 1 and 2):

$$\frac{D}{F} = \frac{x_F - x_B}{x_D - x_B} \qquad \text{Equation 1}$$

$$\frac{B}{F} = \frac{x_F - x_D}{x_B - x_D} \qquad \text{Equation 2}$$

As there are usually a large number of components, it is common practice to define the "heavy" and "light" keys for the system. The heavy key is the highest boiling component of the distillate stream, and the light key is the lowest boiling component present in the bottoms stream. An assumption is made that any components lighter than the heavy key is present in the bottoms and no components heavier than the light key are present in the distillate.

B	Molar Flow of Bottoms Product
D	Molar Flow of Distillate Product
F	Molar Flow of Feed
x_B	Component x Mole Fraction in Bottoms
x_D	Component x Mole Fraction in Distillate
x_F	Component x Mole Fraction in Feed

VLE Data

The next step is to determine the condenser operating temperature, the bubble/dew points of the bottoms and distillate streams, and the operating pressure of the column.

The condenser temperature is based on the cooling medium available (e.g. air, water) and an assumed temperature approach (e.g. 10° C) for the heat exchange equipment to be used.

Calculate the bubble/dew point of the distillate at condenser temperature

- For a liquid distillate product, the bubble point pressure is calculated at the condenser temperature from flash calculations.
- In the case of a vapour distillate product, the dew point pressure is calculated.

A margin of error for additional pressure drops is usually added to the calculated bubble/dew point pressure to give the operating pressure of the column.

Assuming no pressure drop across the column, calculate the bubble point of the bottoms stream at the operating pressure of the column.

Fenske Equation for Minimum Theoretical Plates

The Fenske equation (Equation 3) can then be used to determine the minimum number of theoretical plates required for separation.

- For a total condenser, it is applied between the top plate of the column and the reboiler.
- For partial condensers, it applies between the distillate product and reboiler.

$$S_m = \frac{\log\left[\left(\frac{x_{LK}}{x_{HK}}\right)_D \left(\frac{x_{HK}}{x_{LK}}\right)_B\right]}{\log \alpha_{avg}} \qquad \text{Equation 3}$$

x_{LK}	Mole Fraction Light Key
x_{HK}	Mole Fraction Heavy Key
D	Distillate
B	Bottoms
S_m	Minimum Number of Theoretical Stages
α_{avg}	Relative Volatility at Average Tower Conditions

All the parameters of the equation are known apart from the relative volatility, α_{avg} – this is defined as the ratio of the K-value of the light key divided by the K-value of the heavy key at the average temperature of the column and is a gauge of the ease of separation. The average temperature for a total condenser is calculated as the average of the top plate temperature and the reboiler temperature.

Stages vs. Reflux

In the design of a fractionating column, there is a trade-off between the number of stages and the reflux rate - This is shown graphically in Figure 5.

At minimum reflux rate, infinite theoretical stages are required for separation. At minimum stages, infinite reflux is required.

While these two points are not practical as operating points, it is useful to see the correlation between stages and reflux rate in designing the tower as there are infinite combinations which will be effective.

However, it is often desired to minimise the reflux rate to save on the heating and cooling costs of the reboiler and condenser – therefore a rule of thumb for reflux rate is around 1.25 × minimum.

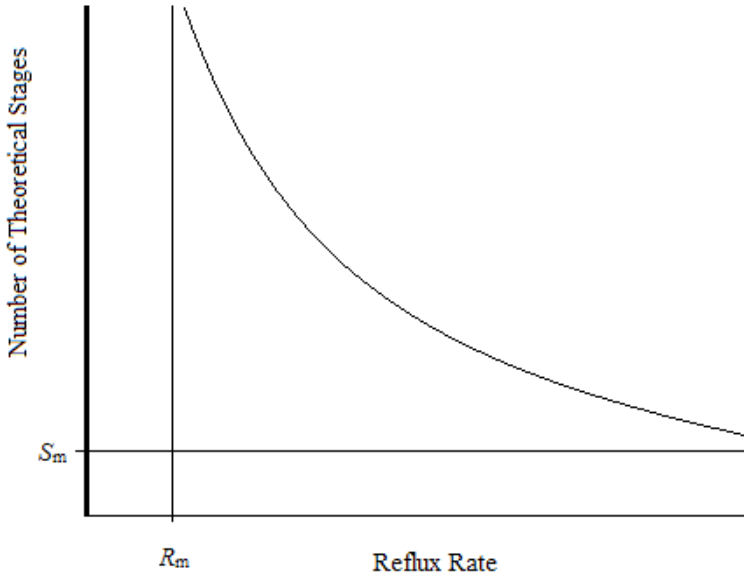


Figure 4: Theoretical Stages vs. Reflux

Underwood Method for Minimum Reflux

The next step is to calculate the minimum reflux rate required. There are two equations used in calculating the minimum reflux flow rate, Equation 4 and Equation 5:

$$\sum_{i=1}^{i=c} \frac{\alpha_i f_i}{\alpha_i - \varphi} = F(1-q) \qquad \text{Equation 4}$$

$$\sum_{i=1}^{i=c} \frac{\alpha_i d_i}{\alpha_i - \varphi} = L_m + D \qquad \text{Equation 5}$$

- α_i Component Relative Volatility w.r.t. Heavy Key (K_i/K_{HK})
- φ Constant
- d_i Component Mole Fraction in Distillate
- f_i Component Mole Fraction in Feed
- L_m Minimum Reflux Rate
- q Total Heat Required to Convert One Mole Feed to Saturated Vapour
  - For bubble-point feed, q=1
  - For dew-point feed, q=0
  - For 2-phase feed, 0<q<1

Solving Equation 4 for a value of the constant φ is an iterative process, which requires use of a solver such as goalseek in Microsoft Excel. Once φ is found it can be used to solve equation 5 for the reflux rate.

Gilliland Method for Theoretical Stages

The Gilliland correlation is then used to calculate the actual number of stages (and hence trays) required by the tower. This equation relates the number of theoretical stages to carry out separation with the actual reflux rate of any given tower. For tray-type towers, the equation takes the form given in Equation 6. The relation is shown graphically in Figure 5.

$$Y = 0.75(1 - X^{0.5668}) \qquad \text{Equation 6}$$

$$X = \frac{R - R_m}{R + 1}$$

$$Y = \frac{S - S_m}{S + 1}$$

R Actual Reflux Ratio

S Actual Number of Theoretical Stages

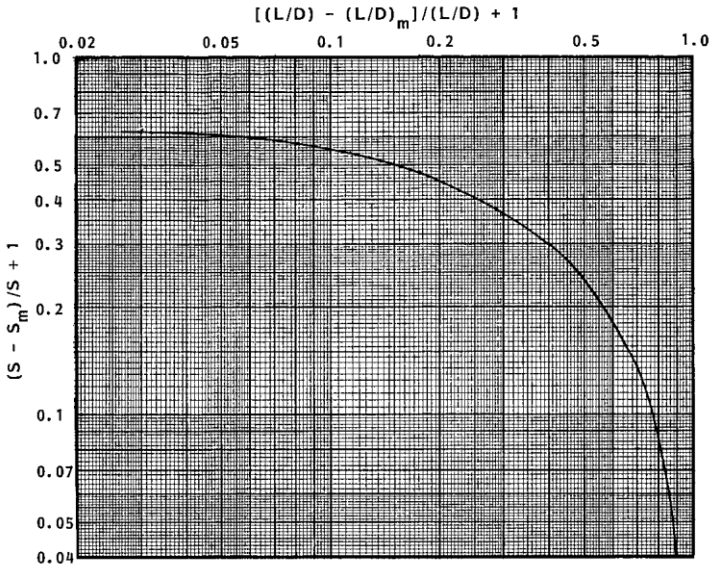


Figure 5: Gilliland Relation Chart

Assuming a value of the actual reflux rate of 1.25 times minimum, *X* can be calculated and hence Equation 6 can be solved for *Y*. Finally, the theoretical stages can be calculated from the value of *Y*.

Tray Efficiency

The actual number of trays required for the fractionation column is determined by converting the theoretical stages calculated by the Gilliland method into an actual tray value using a tray efficiency value. A commonly used relation for the tray efficiency for fractionators and absorbers is that of O'Connell, given in Figure 6.

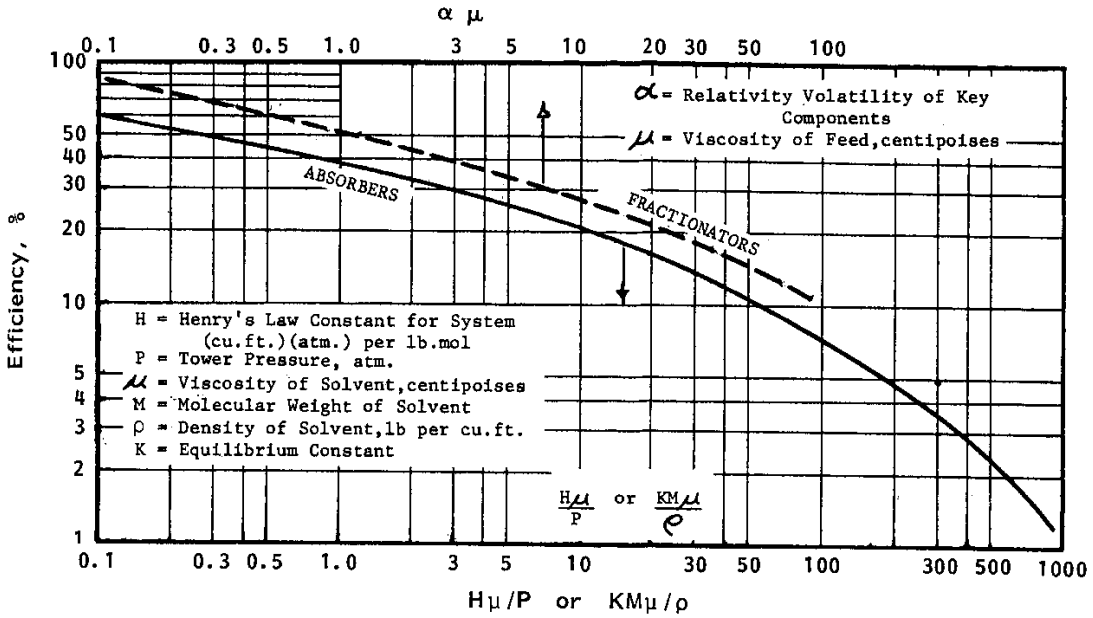


Figure 6: O'Connell Relation

The reboiler will usually count as the first stage of the tower, therefore:

$$T_{\mathrm{n}} = \frac{S-1}{\eta} \qquad \text{Equation 7}$$

T_n Number of Actual Trays

η Tray Efficiency

Energy Balance

Finally, an energy balance is carried out on the tower to determine the duties of the condenser and reboiler. From Figure 3, the overall energy balance can be written as (Equation 8):

$$Q_B + Q_C = h_D D + h_B B + h_F F \qquad \text{Equation 8}$$

In this balance there are two unknown quantities: the duties of the reboiler and condenser. The duty for a total condenser is as follows (Equation 9):

$$Q_C = V_1(h_D - h_1) \qquad \text{Equation 9}$$

For a partial condenser, it is as follows (Equation 10):

$$Q_C = \mathrm{D}(h_D - h_1) + R(h_R - h_1) \qquad \text{Equation 10}$$

V_1 is the vapour leaving the top tower of the column, this can be calculated as the reflux flow plus the distillate product flow. The (h_D – h_1) value represents the latent heat of the vapour exiting the column. For partial condensers the non-condensed distillate stream is also included in the balance.

The duty for the reboiler can then be calculated from the remaining values of the overall energy balance, all that is required is the enthalpy values of the feed, bottoms and distillate fluids. These can be calculated from specific heats.

Sample Calculation

To demonstrate use of the method, the following is a sample calculation. The problem statement is as follows:

A depropaniser with feed stream composition given in Table 1 is to be designed to recover 85% of the C2 fraction in the distillate product and 99% of the iC4 fraction in the bottoms product. Other design factors relevant to the design are given in Table 2. Specify the column parameters.

Table 1: Feed Composition

Component	Mole Fraction
C_2	0.4
C_3	10.5
iC_4	69.0
nC_4	20.1

Table 2: Other Design Factors

Ambient Air Temperature	30° C
Condenser Temperature Approach	5° C
Condenser Pressure Drop	300 kPa
Condenser Type	Total

Sample Calculation: Mass Balance

The specification is to recover 85% of C2 at the top and 99% of iC4 at the bottom. As there are more than two components in the system we need to define the "heavy" and "light" keys. The heavy key is the heaviest component by volatility in the distillate product, while the light key is the lightest component in the bottoms product.

In this case we take isobutane to be the heavy key and propane to be the light key. It is assumed that the other two components partition perfectly in the distillate and bottoms

Using Equation 1 or Equation 2, the column product flows can be determined - the mass balance is as follows based on 100 mol·hr-1 of feed (Table 3):

Table 3: Fractionator Mass Balance

Component	Feed	Distillate, D		Bottoms, B	
	Moles	Moles	%	Moles	%
C_2	0.4	0.4	4	0.0	0.0
C_3	10.5	8.9	89	1.6	1.8
iC_4	69.0	0.7	7	68.3	75.9
nC_4	20.1	0.0	0.0	20.1	22.3
Total	100	10	100	90	100

Sample Calculation: VLE Data

The condenser temperature is dependent on the cooling medium – in this case we have cooling air at a temperature of 30˚ C and temperature approach of 5˚ C. Therefore the distillate product will be at 35˚ C when it leaves the condenser. Assuming a total condenser also, flash calculations with corresponding K-values are used to calculate the bubble-point pressure of the distillate at this temperature. Here process modelling software has been used to obtain the VLE data:

Condenser Temperature: 35˚ C
Distillate Bubble Point Pressure: 13 bar
The column operating pressure is largely dictated by the condenser pressure, however the pressure drop of the condenser is also known (300 kPa) and therefore must be added.
Column Operating Pressure: 13.3 bar
Thus the vapour leaving the top of the column (top plate) must be at its dew-point temperature at column pressure.
Top Plate Temperature: 38˚ C.
Assuming no column pressure difference, the reboiler temperature will be that of the bottoms bubble-point.
Reboiler Temperature: 85˚ C
Optimum separation is achieved when the feed enters at bubble-point, therefore:
Feed Temperature: 74˚ C

Sample Calculation: Minimum Stages

All the parameters of the equation are known apart from the relative volatility, α_{avg} – this is defined as the ratio of the K-value of the light key divided by the K-value of the heavy key at the average temperature of the column. The average temperature for a total condenser is calculated as the average of the top plate temperature and the reboiler temperature.

Top Plate Temperature: 38° C
Reboiler Temperature: 85° C

Average Temperature: $\frac{38+85}{2} = 61.5°$ C

At average temperature and operating pressure of column:

$\mathbf{K_{C3}} = 1.66$

$\mathbf{K_{iC4}} = 0.88$

$\boldsymbol{\alpha_{avg}} = \frac{1.66}{0.88} = 1.89$

Therefore using Equation 3,

$$S_m = \frac{\log\left[\left(\frac{x_{LK}}{x_{HK}}\right)_D \left(\frac{x_{HK}}{x_{LK}}\right)_B\right]}{\log \alpha_{avg}}$$

$$S_m = \frac{\log\left[\left(\frac{8.9}{0.7}\right)_D \left(\frac{75.9}{1.8}\right)_B\right]}{\log 1.89}$$

$$S_m = 9.9$$

Minimum of 9.9 stages for separation.

Sample Calculation: Minimum Reflux

As we have previously assumed that the feed enters at bubble-point, we can take q=1 – therefore the RHS of Equation 4 becomes zero. The next step is to solve Equation 4 for a value of the constant φ.

$$\sum_{i=1}^{i=c} \frac{\alpha_i f_i}{\alpha_i - \varphi} = F(1-q)$$

The value of φ will always be between relative volatilities of the heavy and light keys, setting φ=1.6;

Table 4: First Iteration of Equation 4

Component	**f_i**	**K-value @ 61.5 & 13.3. bar**	**α_i**	**$\alpha_i f_i$**	**α_i - φ**	**$\frac{\alpha_i f_i}{\alpha_i - \varphi}$**
C_2	0.4	3.8	4.2	1.7	2.6	0.653
C_3	10.5	1.6	1.8	18.9	0.2	94.5
iC_4	69.0	0.9	1.0	69.0	-0.6	-115
nC_4	20.1	0.6	0.67	13.5	-0.93	-14.5
					Total	-34.3

From Table 4, this value of the constant did not result in zero on the RHS of Equation 4, therefore the calculation must be repeated, however from using goalseek, φ=1.626.

Sample Calculation: Reflux

Now Equation 5 must be used to calculate the minimum reflux using this value of the constant, see Table 5:

$$\sum_{i=1}^{i=c} \frac{\alpha_i d_i}{\alpha_i - \varphi} = L_{\mathrm{m}} + D$$

Table 5: Calculation of LHS of Equation 5

Component	d_i	α_i	$\alpha_i d_i$	$\alpha_i - \varphi$	$\frac{\alpha_i d_i}{\alpha_i - \varphi}$
C_2	0.4	4.2	1.68	2.57	0.65
C_3	8.9	1.8	16.0	0.17	94.1
iC_4	0.7	1.0	0.7	-0.626	-1.12
nC_4	0	0.67	0	-	-
				Total	93.63

Solving Equation 5 for L_{m}:

$$93.63 = L_{\mathrm{m}} + 10$$

$$L_{\mathrm{m}} = 83.63 \frac{\text{moles}}{100 \text{ moles feed}}$$

Converting this value to reflux per distillate flow:

$$L_{\mathrm{m}}/D = R_{\mathrm{m}} = 8.36$$

Sample Calculation: Number of Stages

We now need to calculate the actual number theoretical stages of stages required for the tower, this is determined by the Gilliland equation:

$$Y = 0.75(1 - X^{0.5668})$$

$$X = \frac{R - R_m}{R + 1} \qquad Y = \frac{S - S_m}{S + 1}$$

A typical value for the actual reflux rate is 1.25 times minimum:

$$R = 1.25R_{\mathrm{m}} = 10.45$$

Calculating X:

$$X = \frac{10.45 - 8.36}{10.45 + 1} = 0.182$$

Calculating Y:

$$Y = 0.75(1 - 0.182^{0.5668}) = 0.46$$

Knowing the minimum theoretical stages (S_{m}=9.9), we rearrange and solve for S:

$$0.46 = \frac{S - 9.9}{S + 1}$$

$$\boldsymbol{S = 19.2 \text{ actual stages}}$$

Sample Calculation: Actual Number of Trays

We now wish to convert this value to a number of tower trays in reality.

A typical value for tray efficiency in a fractionating column is 65%, additionally the reboiler will usually count as the first stage of the tower, therefore:

$$T_{\mathrm{n}} = \frac{S - 1}{\eta}$$

$$T_{\mathrm{n}} = \frac{19.2 - 1}{0.65}$$

$$\boldsymbol{T_{\mathrm{n}}} = 28 \text{ trays}$$

The tower will require 28 trays.

Sample Calculation: Energy Balance

Starting from the overall energy balance given in Equation 7:

$$Q_B + Q_C = h_D D + h_B B + h_F F$$

We realise there are two unknown quantities: the duties of the reboiler and condenser. The duty for a total condenser is (Equation 8):

$$Q_C = V_1(h_D - h_1)$$

The $(h_D - h_1)$ value represents the latent heat of the vapour exiting the column – this was found by a process simulation to be 1.53×10^4 kJ·kgmol^{-1}.

Therefore;

$$Q_C = (104.5 + 10)1.53 \times 10^4$$

$$\boldsymbol{Q_C = -487 \text{ kW per 100 kgmol·hr}^{-1} \text{ feed}}$$

The duty for the reboiler can then be calculated from the remaining values of the overall energy balance, using specific heats found from process simulation (Table 6):

Table 6: Specific Heats of Column Streams

Stream	Specific Heat (kJ·kgmol^{-1})
Feed	164.3
Bottoms	129.0
Distillate	85.7

Solving the energy balance for Q_B;

$$Q_B - 1.75 \times 10^6 = -10 \cdot 85.7 \cdot 35 - 90 \cdot 129.0 \cdot 85 + 100 \cdot 164.3 \cdot 74$$

$$\boldsymbol{Q_B = 541 \text{ kW per 100 kgmol·hr}^{-1} \text{ feed}}$$