CHALMERS

Analysis of opportunities to implement Steam driven fans in new formaldehyde plants

Master's Thesis within the Innovative and Sustainable Chemical Engineering programme

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MASTER'S THESIS

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ABSTRACT

The purpose of this Master thesis (project) is to perform a conceptual design and an economic viability study of steam turbine system for production of power which can be used to partially or totally cover the power requirements of the recirculation blowers in a chemical process. In this project a process for production of 52500 ton/year of Formaldehyde 37 wt% with a consumption of 22300 ton/year of Methanol has been studied.

Based on a reactor using metal oxide catalyst, formaldehyde is produced by means of methanol catalytic oxidation. The formalin plant operates slightly above the atmospheric pressure. The speed of the recirculation blowers increases in order to overcome the pressure drop that occurs in reactor when the catalyst is getting old. Due to degradation of the catalyst the pressure drop increases therefore at some point the pressure drop reach a level where the catalyst must be replaced with new catalyst.

The oxidation is highly exothermic. In order to promote the high conversion of methanol the reaction is kept at given temperatures by means of oil cooling HTF. Oil cooling is achieved by generating steam from condensate. In the standard process, steam is produced at medium pressure.

Process heat integration analysis is carried out to find the maximum thermodynamic potential of power generation. The maximum steam superheating temperature and steam mass flow rate are estimated by means of simple Pinch Analysis calculations. By comparing the power production potential and the power consumption over an average year, different possible design solutions for turbo machinery arrangement (Single Shaft or Double shaft system) are considered and two main design options from vendors are discussed. The different designs are compared considering two main profitability criteria: payback time and net present value. In particular, the total investment cost for single shaft design is around 334’000 €. The revenues in terms of electricity savings is around 146’000 €/year. The payback time for this system is 3 years. The total investment cost for double shaft design is around 200’000 € and the revenues in terms of electricity savings is around 94’000 €/year. The payback time is also 3 years for this design. While the pay-back time does not help to identify the best system option, a discounted cash flow analysis shows that the net present value at the end of 10 year is higher for the single shaft system compared to the double shaft system case.

Contents

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Thank you all!

Notations

S	Entropy [J/kg K]
Q	Heat [J]
T	Temperature [C]
P	Pressure [bar]
h	Enthalpy [J/kg]
Cp	Specific heat capacity at constant pressure [J/kg K]
V	Relative speed [m/s]
m	Mass flow [kg/s]
n_s	Specific speed [m/s]
l	Length [m]
d_s	Specific diameter [m]
A	Area [m^2]
n	Speed [rpm]
ΔH	Heat of Vaporization [J/kg]
T	Time
HTF	Heat transfer fluid
ECS	Emission control system

Greek

η	Efficiency
η_{is}	Isentropic Efficiency
ω	Angular speed [rad/s]
ρ	Density [kg/m^3]

Suffixes

t	Turbine
c	Compressor

1 BACKGROUND

The present work focuses on potential improvements of the energy performance of the Formaldehyde process, a well consolidated process concept for the production of Formalin, a water solution of Formaldehyde, which is used as a key material for products such as coating, paint and plastics (Hagman & Walhelm, 2005).

Cash flows for industrial plants are normally dependent on the fluctuation of energy market prices due to fuel purchases, purchase or sales of electricity and district heating deliveries. Thus it is hard to evaluate the long term outcome of an economic investment in the plant's energy system. In the analysis of energy efficiency investments it is important to consider the uncertainties of the future energy market conditions. For plants located in countries in which prices of electricity are high, electricity production is often a robust investment (Svensson, 2008).

1.1 Introduction

Chemical processes are significant energy consumers mainly due to their large heating demands. As these demands are primarily met through the use of steam, they provide a good potential for efficient cogeneration of power through the use of steam turbines. Efficient cogeneration calls upon better designs of utility systems, as well as more efficient operation of the existing equipment.

Steam systems deliver the required heat, mechanical power, and electricity demands to typical chemical plants. Steam demands result from the heat required by process steam heaters as well as steam required as reactant by the reaction system. Several process devices require also electricity, such as electrical heaters or for motors. In order to avoid the costs of electrical drivers and their conversion losses, some of these process devices (e.g. compressors) can be coupled to steam turbines for direct exchange of mechanical power.

In the process under investigation in this thesis, steam is raised as a result of cooling of a high temperature exothermic reaction. There are different ways in which we can use steam internally in order to produce electricity with a steam turbine and use its power for running process equipments.

After talking with the engineers at the company initiating this work, it was decided to explore the option of using the power production from steam turbine to minimize the electricity required by the recirculation blowers.

Different configurations of steam driven recirculation blowers were investigated in this project. In particular, as steam can also be used for heating in nearby processes (i.e. sold), two types of steam turbines can be considered. If steam also is required for process heating, a back-pressure steam turbine can be used whereas if the steam is to be exclusively used for electricity production, a condensing steam turbine can be implemented (Cardu, 1993). A condensing steam turbine allows more power production and can be of interest if the largest part of the process power requirement has to be covered (e.g. in order to minimize the purchase of expensive grid electricity). However, for this configuration, a condenser is required after the turbine which can lead to high investment costs.

Here follows the brief description of the formalin process. Fresh air is pressurized, to overcome the pressure losses in the formalin process loop in the so called re-circulation blowers. These re-circulation blowers are the main energy consumers in the formalin process. Methanol is pressurized and the mixture of methanol-air is led into the formaldehyde reactor where a catalyst triggers the oxidation. Because that oxidation is exothermic, a large amount of heat is generated in this reactor. The reactor is cooled using pressurized oil as the cooling medium. The vaporized oil is then condensed in a condenser that produces medium pressure steam. After the formaldehyde reactor, the gas is sent to absorption tower where the formaldehyde is absorbed in water. After the absorption tower, it enters into the ECS (Emission Control System) to be cleaned from the rest of hydrocarbon, carbon monoxide, dimethyl ether and methanol it contains. This is done by oxidation in the ECS reactor. The exothermic reaction generates heat, and small amount of medium pressure steam is produced in the ECS steam generator. After the ECS these gases lead to the stack and out to the atmosphere. A more detailed description of the process is presented in the next chapter.

Since in the reference plant design there are only two heat exchangers (one between main reactor and water evaporation, one between ECS reactor and water preheating) it means that the theoretical electricity savings opportunities calculated here can be achieved only through major plant re-design which is beyond the scope of the present study.

There are different ways in which the steam turbine and the electrical motors can be coupled with the pressurization blower system. This is not only a question of investment and electricity savings related to the chemical process design point (nominal conditions) but some technological issues related to plant operations should also be considered. The plant start-up for instance requires installation of an electrical motor to be used to overcome the catalyst pressure drops, as steam starts to be available only after the exothermic catalytic reaction has started. Only at that point the circulation blowers can be switched from the electrical motor to be driven with a steam turbine.

With time, the catalyst inside the reactor degrades, so more pressure drop occurs for a given gas flow and more power is therefore required to overcome such pressure drops (Formox, 2006). In such case therefore, the contribution of the electrical motor can vary during the plant operation.

The plant behaviour during the year must be therefore taken into account in order to discern the best design among all the theoretical possible arrangements.

1.2 Objective of thesis

The objective of this master thesis is to investigate possible new design options that reduce the need to purchase the process electricity from the grid.

In particular, this work focuses on the investigation of steam expansion to recover the work production potential of the medium pressure steam raised by cooling the exothermic reactor which is today used only for heating purposes and often bled to much lower pressures. This involves understanding thermodynamic and technological limitations for maximum steam production, steam network parameters (maximum pressure and temperature, steam turbine backpressure), and the relative impact on process profitability of different arrangements of turbo-machinery.

The study is conducted as a conceptual design of a new re-circulation blower section of the chemical plant concept under investigation and not as a retrofit of already existing plant.

The profitability of the new steam driven re-circulation blower options is assessed by comparing associated cash flows with that of the electricity driven re-circulation blowers of the reference base case standard plant.

1.3 Thesis outline

The first chapter provides an introduction to the work carried out in this project by giving a background and also presenting the aims and objectives for the thesis project. The next chapter presents the detailed description of the plant including process description of the formaldehyde production, main thermal loads of the plant and major electricity consumption units.

Chapter 3 presents the characterization of the reactor pressure drop and of the blower power requirement.

Chapter 4 provides an overview of the maximum thermodynamic potential of power generation.

Chapter 5 discusses the conceptual design of steam driven re-circulation blowers for practical work recovery in the formaldehyde plant.

Chapter 6 gives the economic evaluations of different configurations of turbo machinery as indicated by vendors that were contacted during this study.

Results and Conclusions are presented in Chapter 7.

2 DETAILED DESCRIPTION OF THE STANDARD FORMALDEHYDE PRODUCTION PLANT (Reference Year 2012)

2.1 Production of formaldehyde

Formaldehyde is produced by partial catalytic oxidation of methanol in air. The reaction takes place on a metallic oxide catalyst, using a fixed tube bed vapour phase oxidation reactor, according to the following stoichiometry:

$$CH_3OH + 1/2\ 0_2 \rightarrow CH_20 + H_20 \qquad (2.1)$$

A low ratio of methanol to air is used to maintain the desired oxidation atmosphere. The methanol content in air is maintained between about 4 and 10 vol%. Such a high methanol content can be used because part of the gas from the absorber is recycled, which decreases the oxygen concentration sufficiently to avoid explosive mixtures (Formox, 2006).

As shown in the main process flow diagram in *Figure 2.1***,** the heat of reaction is removed from the main reactor by boiling a liquid, hereafter referred to as heat transfer fluid (HTF). The process gives a high yield of formaldehyde on a single passage, and also very high methanol conversion. The actual formaldehyde yield is in the range of 91-94 % of the theoretical. The remaining methanol is accounted by un-reacted methanol, carbon monoxide, dimethyl ether and a negligible amount of formic acid. The fresh air is supplied by a pressurization blower and, after mixing with recirculation gas, pushed through the process by two recirculation blowers. Methanol is supplied to the plant by a pump. The methanol is vaporized and the gas mixture is heated when the process gas-methanol mixture is passed through the methanol vaporizer.

The oxidation of the methanol takes place on the catalyst in the tubes of the main reactor. The tubes are loaded with the metallic oxide catalyst to a specific depth. The bottom and the top sections of the tubes are filled with small inert rings to improve heat transfer. The shell side of reactor is filled with HTF to remove part of the heat of reaction. The gas mixture is preheated by boiling the HTF in the top of the tubes. As the gas reaches the heated catalyst, the reaction starts and the temperature rise rapidly to a maximum. When the main part of the methanol has reacted, the temperature drops rapidly again and approaches the temperature of the boiling HTF when the gas leaves the reactor tubes. (Formox, 2006).

The circulation of HTF to the reactor shell and into HTF condenser is carried out by thermo siphon circulation. The vapours of HTF are condensed in a shell-and-tube heat exchanger. The condensation heat is recovered to produce medium pressure steam. The HTF condenser is therefore also operated as a steam boiler and it gives excess steam as compared to the steam production from ECS (Emission Control System) steam generator. During start-up, before the heat of reaction has achieved sufficient thermo siphon circulation, the HTF is circulated by a pump and heated by an electrical heater. During the conversion of methanol into formaldehyde, some side reactions occur, which are:

Side reactions:

$CH_3OH + 0_2 \rightarrow CO + 2\ H_20$ (2.2)

(Carbon monoxide)

$CH_3OH + 0_2 \rightarrow HCOOH + H_20$ (2.3)

(Formic Acid)

$2CH_3OH \rightarrow \quad CH_3OCH_3 + H_20$ (2.4)

(Dimethyl ether)

From the bottom of main reactor the reacted gases pass through the methanol vaporizer. These gases are cooled to a temperature to about 120°C in the methanol vaporizer, depending upon the flow of methanol. From the methanol vaporizer these gases flow into the Absorber. The reactive formaldehyde is absorbed in water, in the absorption tower. The gas is passing upward through water/formalin over different packing materials. These packing maximize the contact area between water and formaldehyde gas. So liquid is cooled in several steps and sometimes caustic soda is added from the top. All these measures will increase the efficiency of the absorption tower. The formalin solution is finally stored in a storage tank (Formox, 2006).

From the top of the absorption tower the gas flow is split between ECS (Emission Control System) and recirculation blowers. To the ECS, the amount of gas is controlled by an oxygen control valve. This valve controls the proportions of recycle gas and fresh air in such a way that the oxygen concentration of the process gas is kept constant after the recirculation blowers. Approximately 1/3 of the gas after the control valve enters into the ECS system, where hydrocarbons, carbon monoxide, dimethyl ether, formaldehyde and methanol are converted into carbon dioxide and water through catalytic combustion. The oxidation of the gas is done in an ECS reactor over a bed of noble-metal catalyst.

The following oxidation reactions take place:

$CO + 1/2\ 0_2 \rightarrow CO_2$ (2.5)

$CH_3OCH_3 + 3\ 02 \rightarrow 2CO_2 + 3H_20$ (2.6)

$CH_20 + 0_2 \rightarrow CO_2 + H_20$ (2.7)

$CH_3OH + 3/2\ 0_2 \rightarrow CO_2 + 2H_20$ (2.8)

As the catalytic combustion is highly exothermic it generates heat which can be used for internal heat recovery and for steam production. After passing from ECS reactor and steam generator, the high temperature gas flow enters the other side of the ECS pre-heater. During start-up, the ECS reactor is heated by an electrical heater before the catalyst bed has ignited. Before the gas enters into the ECS reactor from ECS pre-heater, this ECS pre-heater preheats the ECS gas flow to the ECS reactors catalyst ignition temperature. After passing through the ECS pre-heater this gas is lead to the stack and out to the atmosphere (Formox, 2006).

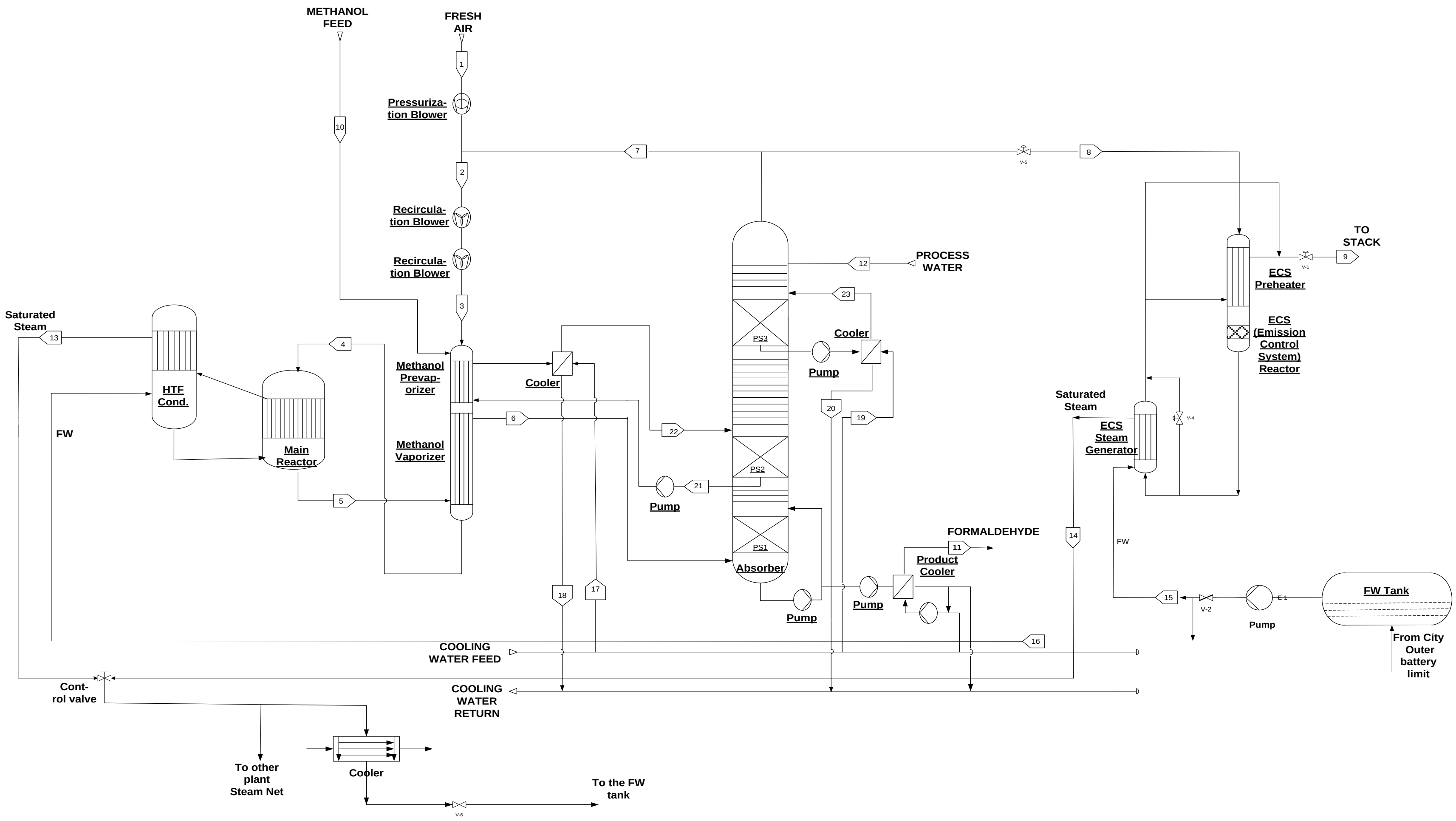

Figure 2.1 Process flow diagram of the current "standard" design (Formox, 2006).

2.2 Definition of the reference plant operation

The current "reference" formaldehyde plant evaluated allows for a maximum capacity of 52500 ton/year of Formaldehyde with a consumption of 22300 ton/year of Methanol. For evaluation the plant is assumed to operate for 350 days/ year at 100% capacity. The production capacity of the plant load at 100% is 150 ton/day. Due to degradation of the catalyst the pressure drop increases inside the reactor therefore at some point the catalyst must be replaced with a new catalyst.

2.3 Main thermal loads of the plant and thermal utility requirements

The main thermal loads of the plants are the Main reactor and the ECS reactor as shown in *Table 2.1*. Heat is generated in the main reactor due to the exothermic reaction. The HTF pump is used to circulate the HTF into the reactor and the HTF condensers, at the start-up the HTF heater is used to heat the HTF to the correct start temperature in the reactors. The HTF is used on the shell side of reactor in order to recover heat which causes the HTF to boil in the reactor. The HTF vapours leave the reactor shell and move into the HTF condenser. In the HTF condenser the HTF vapours condenses. The condensed HTF is collected in the separator. The level in the separator is indicated by the level indicator. The steam is generated (by flowing water counter-currently) in the HTF condenser as the feed water evaporates due to heat of condensation of the HTF.

For start-up purposes (only 6 hours), the ECS reactor is equipped with an electric heater. The purpose of the electric heater is to ignite the bed, after which there is no more need of electric heater since the process is self-sufficient due to the exothermic reactions. The exhaust gas from the absorber passes into the catalytic chamber where it is oxidized over a bed of noble metal catalyst. The temperature increases due to the exothermic reaction. The hot gases leaving the catalyst bed then passes through the ECS steam generator which is also cooled by feed water. Only a minor part of the total process steam is generated in the ECS steam generator by cooling the hot stack gas leaving the catalyst bed (Formox, 2006).

Table 2.1 Summary about thermal loads of the plants and steam available for export in the standard plant

Main thermal loads of the plant	Q [kW]	Steam available for export in the standard plant [kg/hr]
Main reactor	2268	3390
ECS reactor	795	618

2.4 Main electricity consumption units

The major electricity consumption units in the formaldehyde plants are described below (Formox, 2006).

2.4.1 Pressurization Blowers

The pressurization blower is of Roots type. To maintain a constant pressure in the system, a pressure controller controls the inlet air by adjusting the blower speed with a frequency convertor.

2.4.2 Recirculation Blowers

The recirculation blowers, which supply the air for the oxidation process, are of centrifugal type. They are connected in series. The flow controller controls the process gas flow by adjusting the blower speed via frequency convertor. Immediately after the blowers, a sample of process gas is continuously withdrawn and passed through two oxygen analyzers. The measured process gas flow is used for calculating the methanol volume % in the process gas.

2.4.3 Electric heater with ECS reactor

For start-up purposes, the ECS reactor is equipped with an electric heater. The heater is switched on and off from the distributed control system (DCS).

2.4.4 Electric heater with HTF system

The HTF heater is used to heat the HTF to the correct start temperature in the reactors. It is operated only at start-up to regulate the reactor temperature.

2.4.5 Pumps

Pumps are used to move fluids into the different units like HTF condenser, absorber, and a product cooler.

2.4.6 Total electricity consumption of the plant

The total yearly average electricity consumption by (pressurization, recirculation blowers, pumps and heaters) is 3’400 MWh. The electricity consumed by the recirculation blowers is 2’180 MWh.

Table 2.2 Summary related to electricity consumption of the plants

Main electricity consumption units of the plant	Electricity consumption [MWh]
Re-circulation blowers	2’180
Pressurization blower	750
Pumps	350
Electric heaters	100

3 CHARACTERIZATION OF THE REACTOR PRESSURE DROPS AND OF THE BLOWER POWER REQUIREMENT

This chapter focuses on the power requirements of the recirculation blowers, the process units which have been identified as the potential location for new energy savings measures.

The power requirement of the recirculation blowers gets higher during the process operation due to degradation of the catalyst.

No clear data about pressure drops in the reactor are available except for the minimum and maximum values.

In this chapter the method for estimating how the pressure drops vary in the reactor when the catalyst is getting old is described and some conclusions are drawn on the consequent blower power requirement profile.

3.1 Pressure drop profile in the reactor

In order to estimate the pressure drop profile in the reactor, the length of normal cycle is 240 days/year. When the catalyst is fresh the pressure drop over the process is 0.47 bar, and when the catalyst is getting old the pressure drop over the process is 0.82 bar. When the catalyst is completely degraded then we need to replace the old catalyst with a fresh catalyst in order to increase the catalytic oxidation reaction in the main reactor.

By considering constant production of 150 ton/day a polynomial model of 2^{nd} degree was selected to estimate how pressure drop varies with time in the main reactor over the typical load (240 days/year).

The polynomial model of the pressure drop profile is then used to estimate the power profiles of the blowers and eventually to get the total electricity consumption over the typical load.

The general expression for the 2^{nd} degree polynomial is

$$y = \mathrm{ax}^2 + \mathrm{bx} + \mathrm{c} \tag{3.1}$$

Where "y" is the Δp in bar, "x" is the time in days and "a, b, c" are the constants. We have a boundary limit for pressure drop in the process,
At x=0, meaning when t=0 days, then y=c=Δp=0.47 from the above eq. (3.1)

x= 240, means when t= 240 days, then y=Δp=0.82

$$y=0.82=a.(240)^2+b.(240)+0.47 \tag{3.2}$$

The derivative of eqt (3.1) is,

$$\frac{dy}{dx} = 2ax + b \tag{3.3}$$

The derivative of equation (3.1) is 0 at y=0, because the slope of line is tangent, so b=0 then after simplification of eq. (3.2), we get Δp as a function of time.

0.82-0.47=a.57600 and a=6.076.E-6

$$\Delta P(t) = 6.076\mathrm{E} - 6.(\mathrm{t})^2 + 0.47 \tag{3.4}$$

Eqt (3.4) returns the pressure drop profile [bar] in the reactor over time [days]. The generated profile is shown in the figure below.

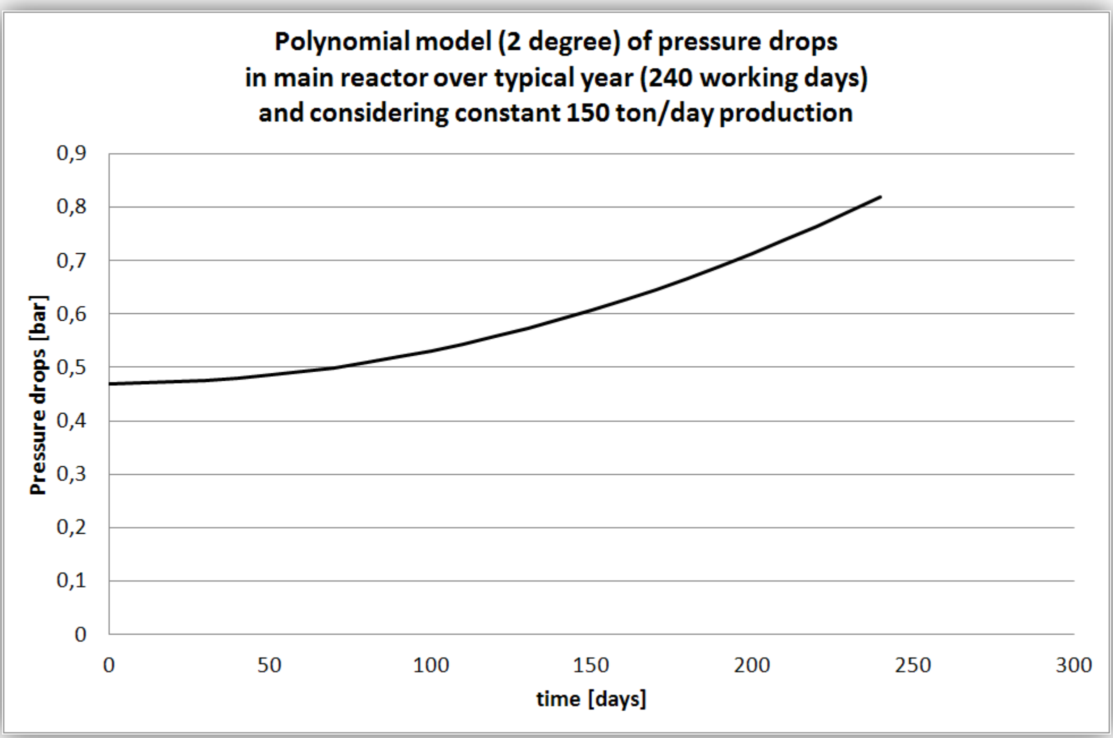

Figure 3.1 Pressure drop profile in the reactor

3.2 Estimation of the blower power profile and total yearly electricity consumption

In the standard design, the two recirculation blowers are connected in series (i.e. two stages, two different shafts) and they are driven by two electrical motors as shown in *Figure 3.2*.

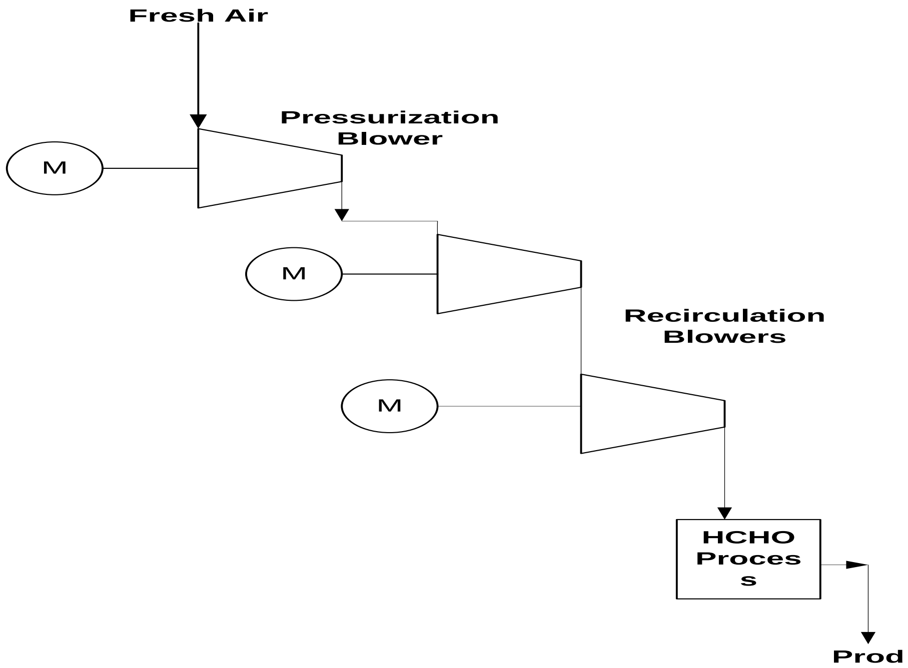

Figure 3.2 Combination of re-circulation blowers in the standard design

To estimate the blowers' power profiles and the total yearly electricity requirement according to the pressure drop profile above, some calculations are required to estimate the blower performance.

According to the given volumetric flow rate, available as a process data, the shaft power of the blower is calculated from the procedure described in Handbook of Modern Fan Technology (Fans, 1997). The other required data for estimation of blower power is the mass and energy balances. The procedure for calculating the blower power is described in Appendix.

Table 3.1 Electricity consumption in the recirculation blowers

Cases	100% (150 ton/day)
Pressurized + Fresh Catalyst (0,3-0,77 barg)	219 kW
Pressurized + Old Catalyst (0,3-1,12 barg)	351 kW

We use the same 2^{nd} degree polynomial model for estimation of outlet pressure *P2* after the 2^{nd} stage of recirculation blower.

$$P2(t) = 6.076\text{E} - 6.(\text{t})^2 + 0.47 \quad (3.5)$$

Eqt (3.5) returns the outlet pressure *P2* [bar] over time [days].

The procedure described in Appendix 9.1 from eqt. (9.1) to eqt. (9.19) is used to estimate the total fan shaft power requirement per load. The blower power profile over a typical load is shown in *Figure 3.3*.

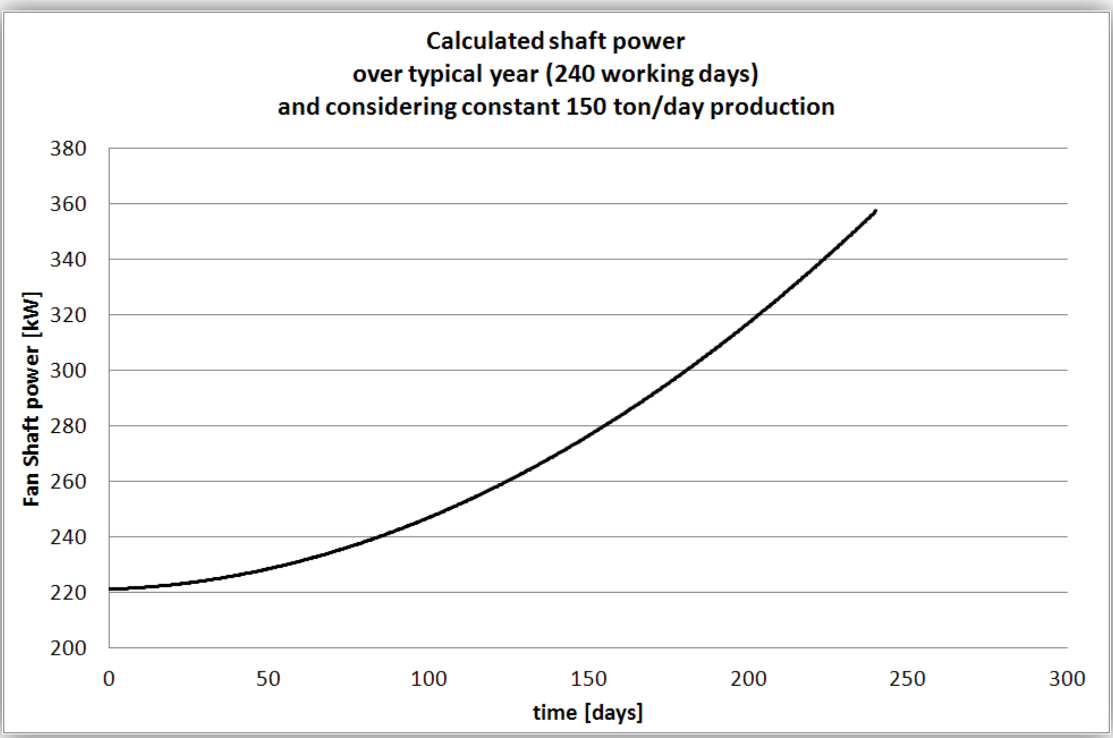

Figure 3.3 Variation of blower power profile with time

The power profile was estimated by assuming constant blower efficiency and by calculating the corresponding power by imposing a progressively higher outlet pressure as indicated by the pressure profile obtained above.

By integrating numerically the power profile in *Figure 3.3*, over time, the electricity consumption for one catalyst cycle (240 days) is 1’570’000 kWh is obtained for the recirculation blowers.

In order to calculate annual electricity consumption for recirculation blowers 350 days we multiply the electricity consumption for one catalyst cycle by a factor of (350 days/240 days) 1.46. The total electricity consumption for recirculation blowers is 2’290’000 kWh/year.

4 ESTIMATION OF POWER PRODUCTION POTENTIALS

4.1 From thermal loads to aggregate heat load/temperature profile

4.1.1 What is Pinch Analysis?

Pinch Analysis is used to identify the opportunities for improving the integration of processes, in order to decrease the amount of external heating and cooling demand in a chemical processes. The integration of processes can be achieved by increasing the shares of heating and cooling by internal heat exchange. The maximum potential for heat exchanging and the minimum heating and cooling demand can be determined by using Pinch Analysis. A stream which needs heating demand is known as cold stream and a stream which needs cooling demand is known as hot stream. The hot composite curve is formed by adding the heat loads for all the hot streams over a temperature range. Similarly, the cold composite curve is formed by adding the heat loads for all the cold streams over a temperature range. The red line represents the hot composite curve and blue line represents the cold composite curve as shown in *Figure 4.1*. For a chosen minimum allowable temperature difference (ΔT_{min}) in heat exchangers, the process Pinch point can be determined. The minimum temperature difference occurs at the Pinch point. The curve also shows how much heat can be recovered internally within the process (Q_{HX}). The minimum heating and cooling demand ($Q_{H, min}$ and $Q_{C, min}$) can be calculated from the composite curve. If there is maximum internal heat exchanged (Q_{HX}), than ($Q_{H, min}$ and $Q_{C, min}$) are the minimum external utility needs of the process. It means that we need to fulfill this ($Q_{H, min}$ and $Q_{C, min}$) externally from outside the processes. The composite curve temperature profiles can be plotted in terms of shifted or interval temperature, which can be obtained by adding $\Delta T_{min}/2$ to cold streams and subtracting $\Delta T_{min}/2$ from the hot streams. It means that if the hot and cold composite curves intersect, the temperature difference is equal to the chosen minimum temperature difference.

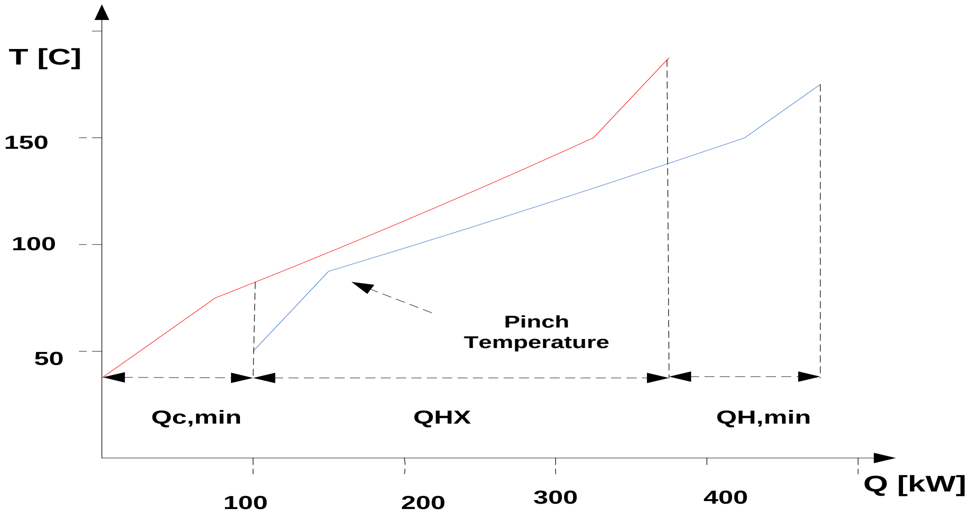

Figure 4.1 Composite Curves

Grand composite curve (GCC) is made in the similar manner, where all streams (both hot and cold) in the temperature range is added. For constructing GCC interval temperature are used. For the utility levels needed, GCC is used to conduct a graphical analysis and also how we can set the corresponding utility loads to get the minimum utility costs. The potential for producing steam or hot water for district heating or for installation of heat pumps can be determined from GCC. Only external heating is required above the Pinch and below the pinch only external cooling. When designing a heat exchanger network for obtaining minimum heating and cooling demand from outside the process, the design should start at the Pinch point. There are three "golden rules" when designing a network for maximum heat recovery shown in

Figure 4.2. There should be no cooling above the pinch, below the pinch there should be no heating, and no heat transferred across the pinch. Violations of any of these rules results in higher use of energy than needed for a maximum heat recovery network. (Kemp, 2007).

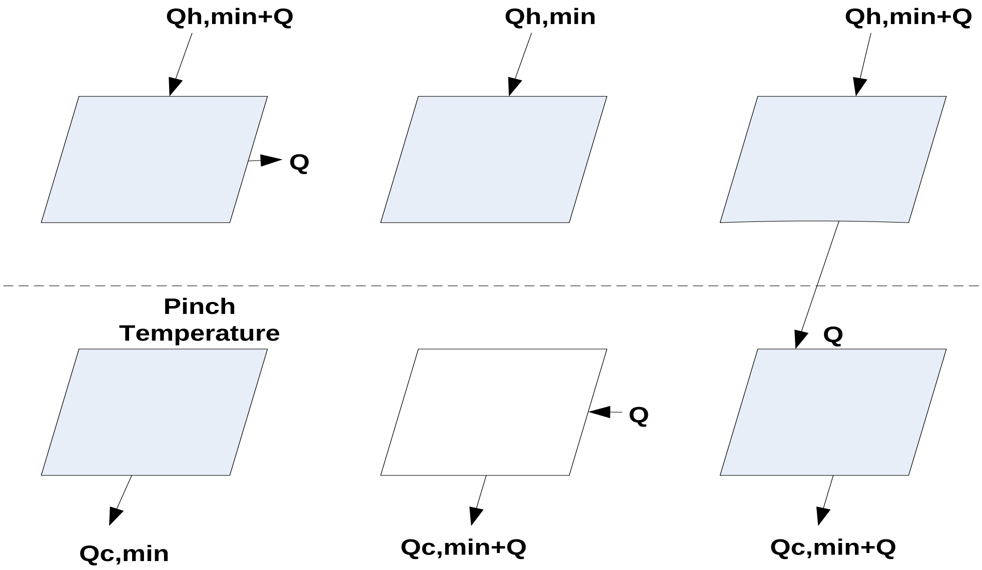

Figure 4.2 Illustration of violations of the three golden rules

4.1.2 Grand Composite Curve of the formaldehyde plant

The general problem that we are investigating is related to the maximum power that we can generate with a steam turbine only considering thermodynamic limits. For this purpose we collected all the main thermal streams from the process. Temperatures and heat loads of the main process thermal streams are summarized in Table 4.1.

Table 4.1 Thermal streams of the process

Type	Tstart [°C]	Ttarget [°C]	Q [KW]	ΔT/2 [K]
Main reactor	290	289	2268	10
ECS reactor	530	120	795	15
ECS Preheater	27	200	368	15

4.1.3 Operating parameters of a heat recovery steam turbine cycle for maximum power generation

Power generation can be achieved through a heat recovery steam cycle. The steam cycle is also referred as Rankine cycle, for which a typical (T,s) diagram is shown in *Figure 4.3*. This thermodynamic cycle is commonly used in many power stations, the only difference with respect to the case investigated in this thesis being the fact that the required thermal input is not provided by a process but by burning fuel.

The working fluid in a Rankine cycle follows a closed loop and it is reused constantly. In order to achieve higher cycle efficiency steam is normally superheated before being expanded in the steam turbine. The steam at the turbine outlet is then condensed and subsequently fed again into the steam boiler.

The conversion of the mechanical work into electricity is achieved with an electrical generator coupled with a shaft to the turbine.

Efficiency of cycle $=W/Q_1$

$W=m.[(h4-h5)-(h1-h7)]$

$Q_1= m.(h4-h1)$

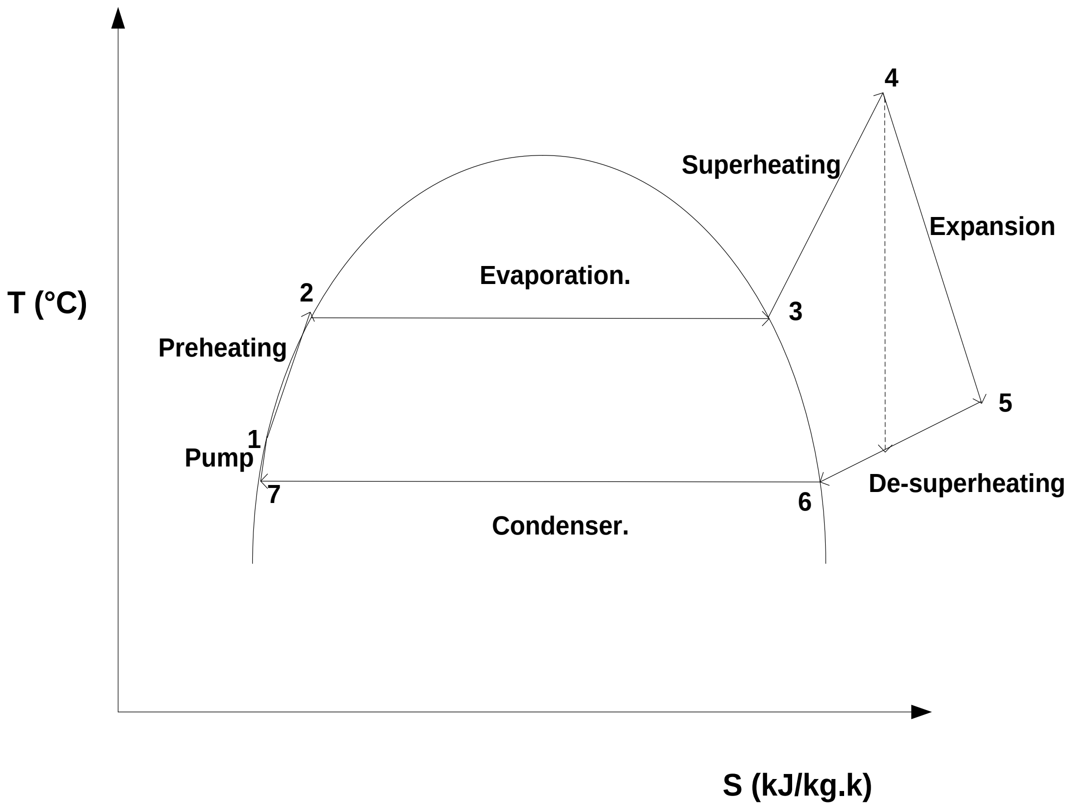

Figure 4.3 Rankine Cycle

In order to estimate the potential for power generation a foreground/background analysis is conducted. The foreground/background analysis is an approximate analysis. It is due to fact that the choice of ΔT_{min} affects the shape of the grand composite curve (GCC) which the analysis is based on. The foreground/background analysis should therefore be a guiding method to identifying integration possibilities between processes.

This starts with plotting the process grand composite curve (GCC). Once the GCC of the process is available as shown in *Figure 4.4,* the information about the heat availability at all temperature levels can be read directly from the diagram. The composite curve of the formaldehyde plant is shown in *Figure 4.6*.

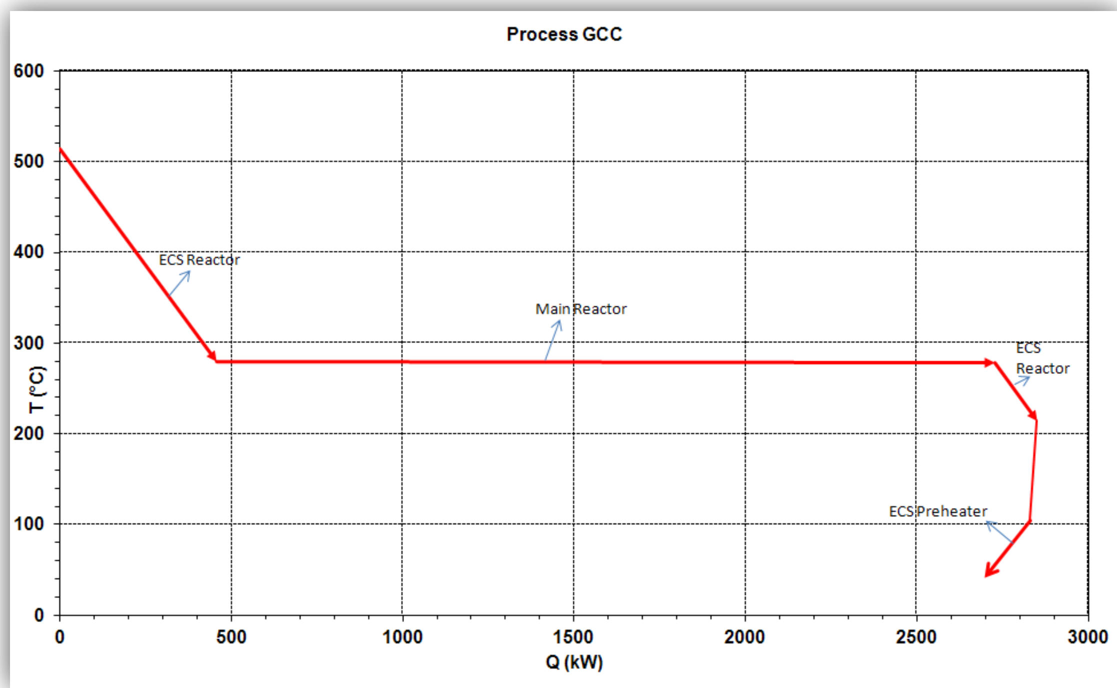

Figure 4.4 Grand Composite Curve (GCC) (formaldehyde plant)

The steam cycle GCC is shown in *Figure 4.5*. Thus the maximum power generation results from the activation of pinch points between the process GCC and steam cycle GCC. For this purpose a number of iterations are required so that the steam cycle flow rate is adjusted so as to maximize conversion of excess heat from the process to electric power.

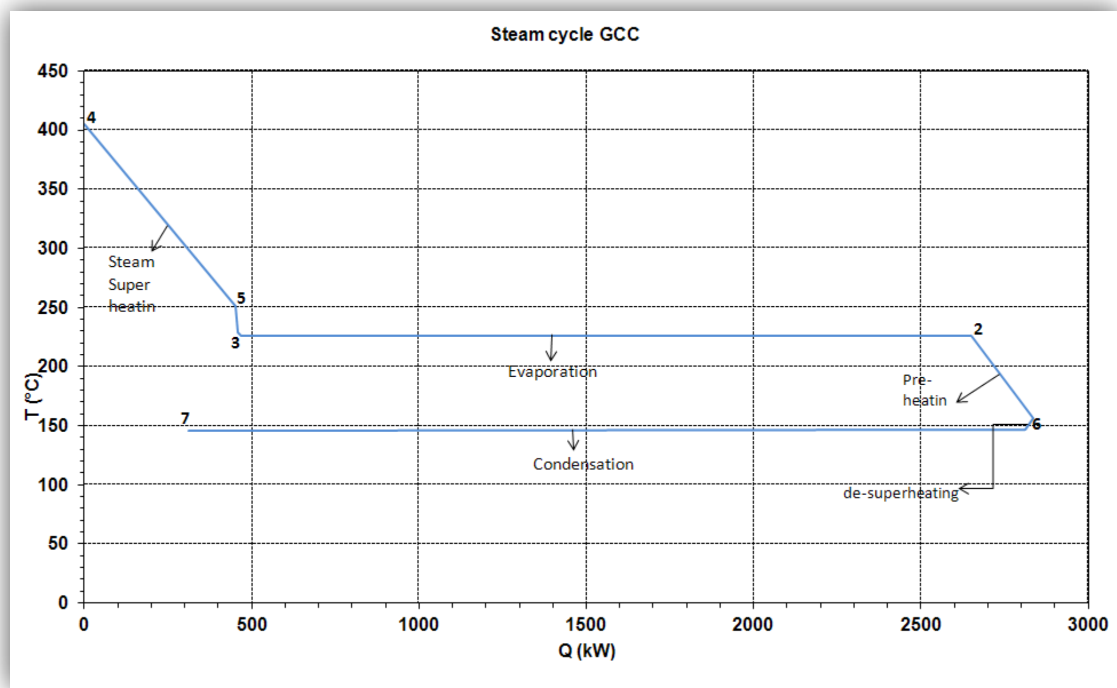

Figure 4.5 Steam cycle GCC

We start with a case in which steam evaporation pressure is equal to 25 bar (the value used today for the steam network) and 400°C as maximum superheating end-temperature. The calculations proceed by investigating the case in which steam can still be delivered at 5 bar to a nearby process. We also assume that the minimum temperature difference ΔT_{min} = 10K. The resulting steam mass flow rate for maximum power generation is equal to 1,2 kg/s. The thermal streams associated with the steam cycle are shown in *Table Table 4.2.* The resulting foreground/background representation is shown in Figure 4.7.

Table 4.2 Steam cycle

Type	Tstart [°C]	Ttarget [°C]	ΔT/2 [K]	ΔH [kJ/kg]	m [kg/s]	Q [kW]
Condensation	151,9	151	5	2107	1,185	2496
Preheating	151	224	5	321	1,185	380
Evaporation	223,9	224	2	1841	1,185	2181
Superheating	224	400	5	436	1,185	516
desuperheating	256	151,9	5	226	1,185	268

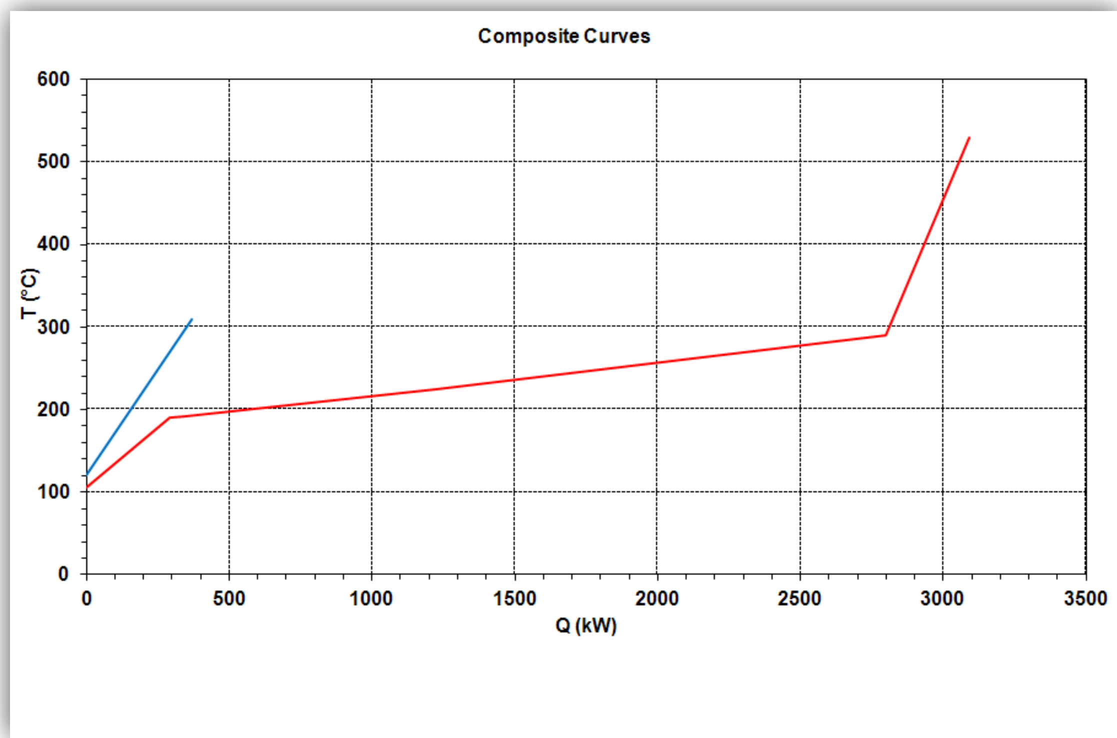

Figure 4.6 Composite Curve (formaldehyde plant)

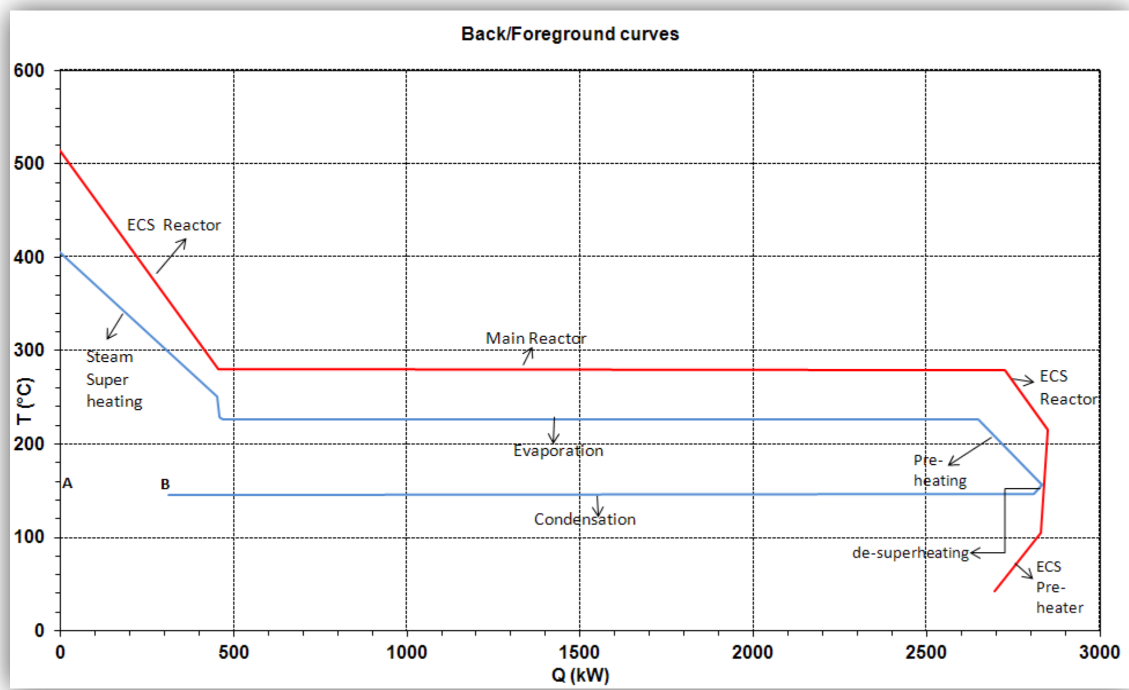

Figure 4.7 *Back/Foreground curve at 25 bar and 400°C (Back pressure turbine)*

The amount of power cogenerated can be read from Figure 4.7. This corresponds to the reading of the heat load value of the final point of steam condensation (distance between points A and B). For this case it corresponds to 312 kW.

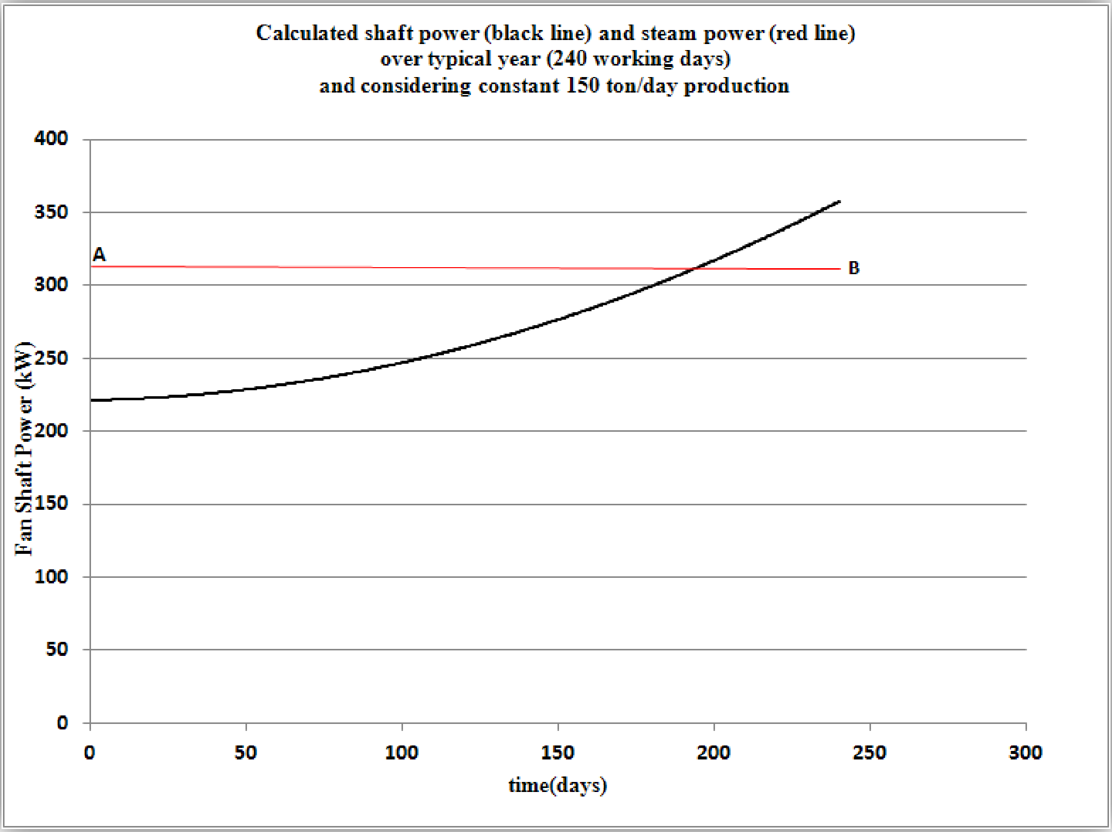

Figure 4.8 Estimation of power production potential through pinch technology

From *Figure 4.8*, through the Pinch technology estimation of power production potentials from the steam turbine is 312kW. Only we need a small size of motor, when the catalyst is getting old in order to compensate the remaining load of the recirculation blowers after 200 days.

In a sensitivity analysis, the power production potential was checked out with different sets of inlet temperatures and pressures for the back pressure turbine. By activating a second utility pinch we didn't get much big difference in compare to the power production potential as shown in *Figure 4.9*, *Figure 4.10*.

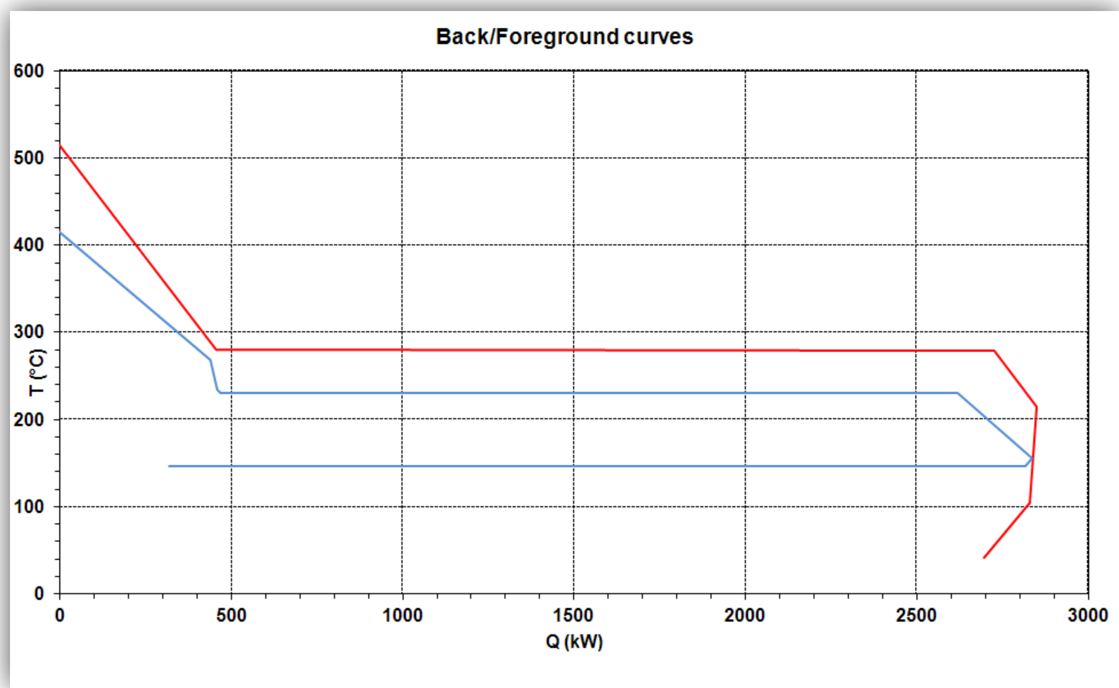

Figure 4.9 Back/Foreground curve at 28 bar and 410°C (Back pressure turbine)

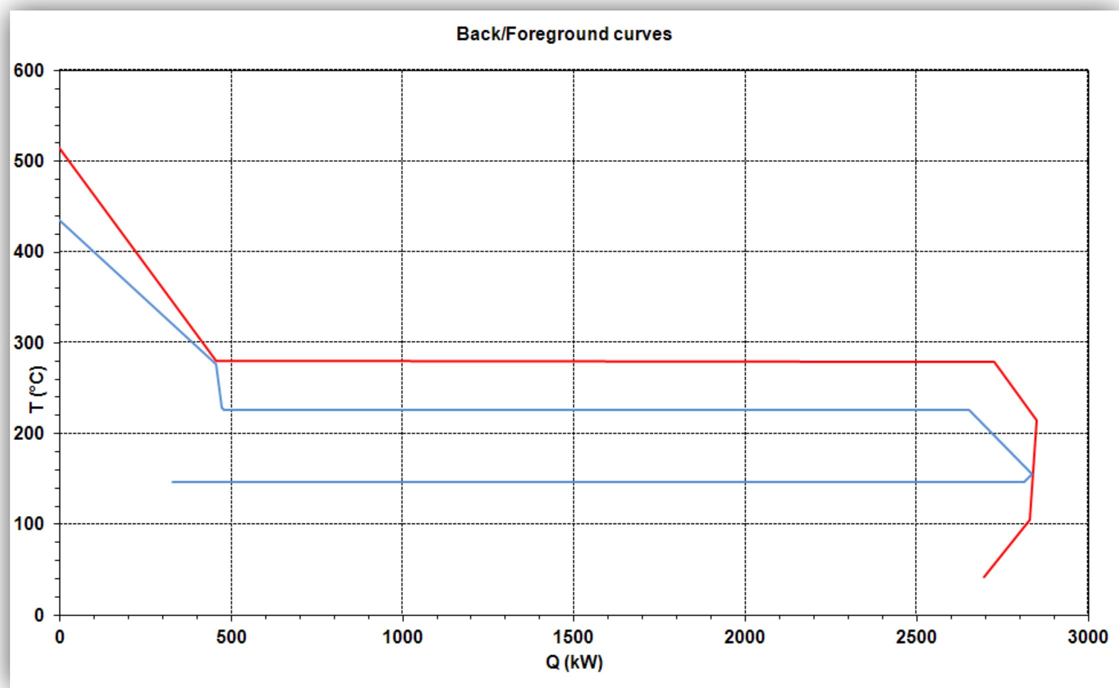

Figure 4.10 Back/Foreground curve at 25 bar and 430°C (Back pressure turbine)

The power production potential in case of condensing steam turbine at 25 bar and 400°C is 585 kW (distance between points A and B) for 0.1 bar pressure in the condenser as shown in the figure below.

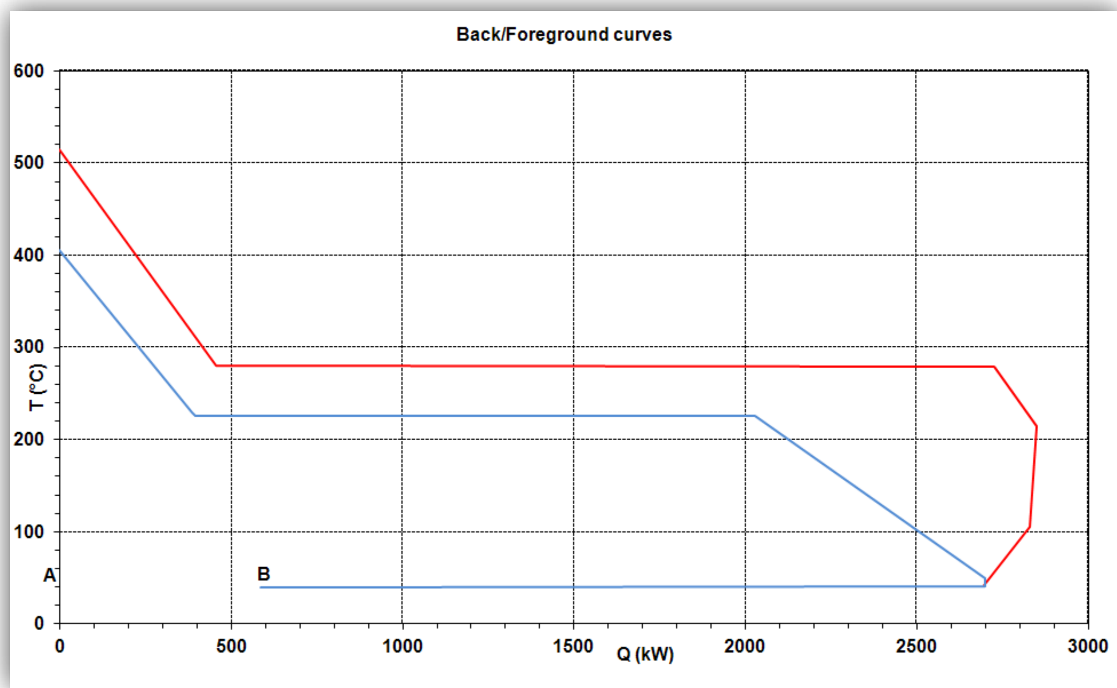

Figure 4.11 Back/Foreground curve at 25 bar and 400°C (Condensing steam turbine)

In a Figure 4.7 we can observe there is still a potential for increasing power production as the steam evaporation pressure (temperature) can be increased to activate another pinch point. In addition we observe that the superheating temperature can be easily increased theoretically up to 520°C. The maximum evaporation pressure corresponds to the case in which steam evaporation overlaps the thermal streams of the process at 280°C (T^{int}). This corresponds to a steam pressure of 60 bar[1].

Due to possible technical constraints, a maximum superheating end-temperature of 450°C was assumed. The result of this doesn't correspond to the maximum pressure as a pinch point constraints the mass flow rate to low values. Therefore we try with another set of values at 50 bar and 400°C.

From the *Figure 4.12* at 50 bar and 400°C the value of power results apparent. This corresponds to the reading of the heat value of the final point of steam condensation (point A & B). For this case it corresponds to 388 kW which means that we are producing extra power from the steam turbine compared with the needs of the recirculation blowers.

[1] 60 bar corresponds to the saturation pressure at 275°C (Real)

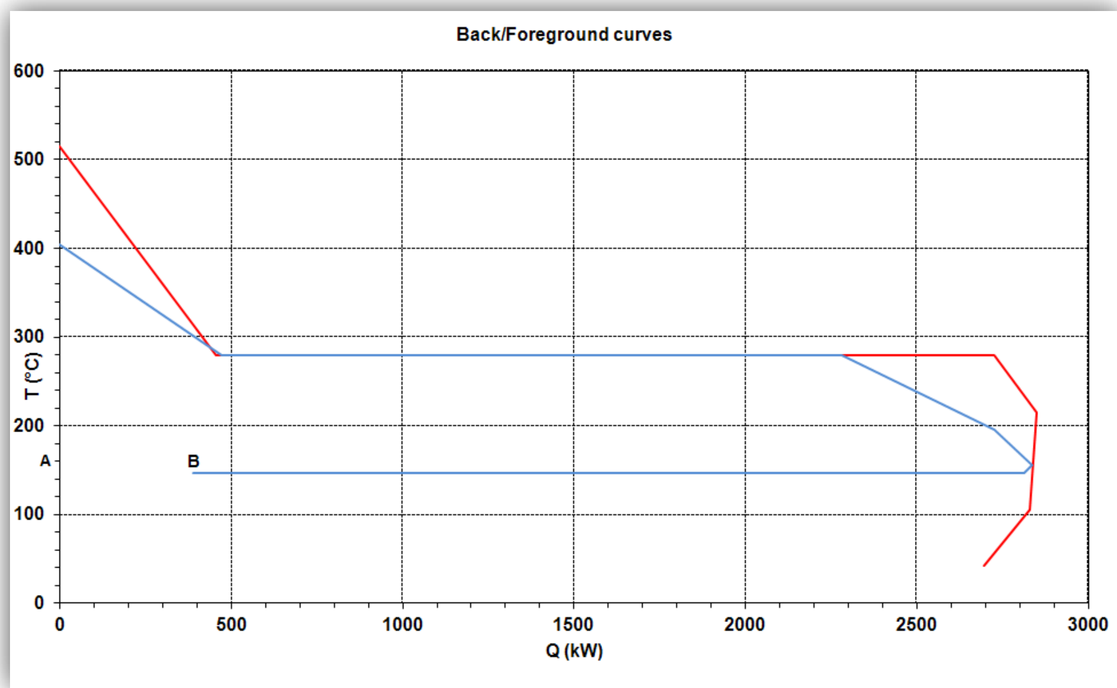

Figure 4.12 Back/Foreground curve at 50 bar and 400°C (Back pressure turbine)

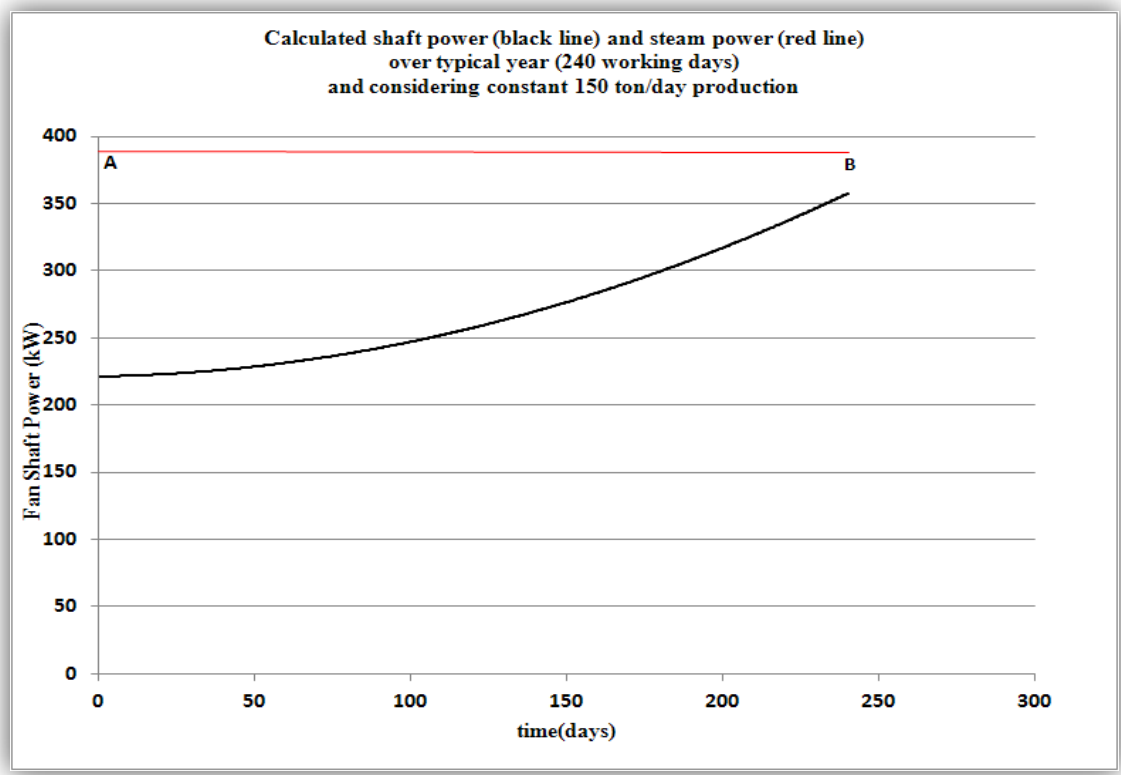

Figure 4.13 Technical maximum power generation from steam turbine

For condensing steam turbine at 43 bar and 450°C the maximum power generation is 634 kW (distance between points A and B) as shown in *Figure 4.14.*

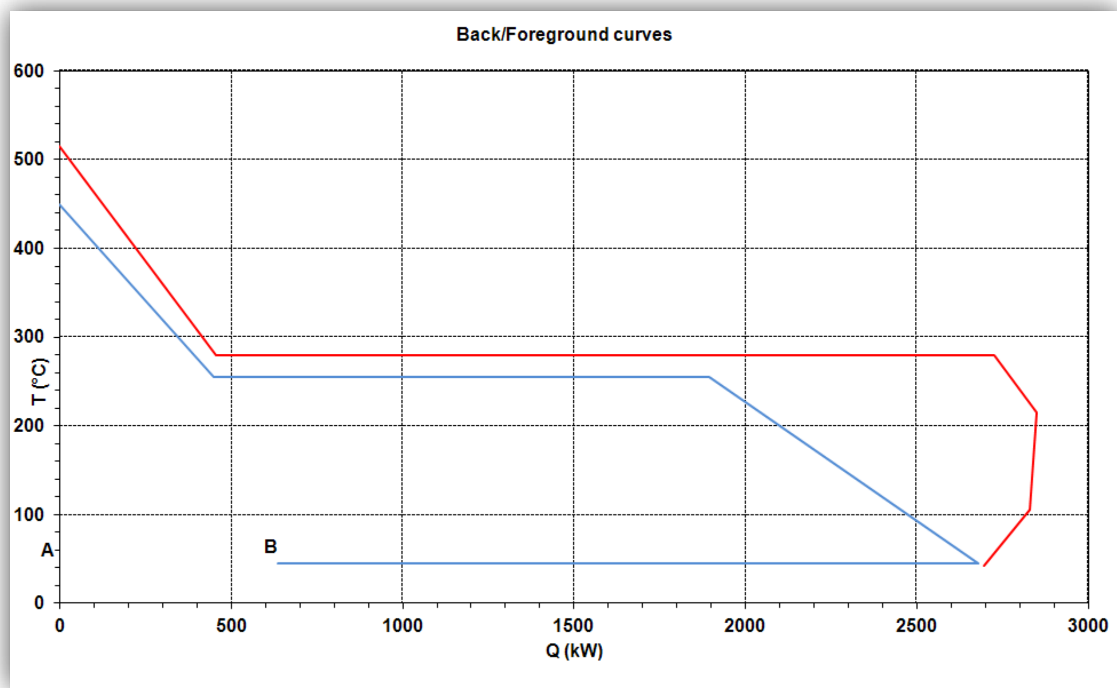

Figure 4.14 Back/Foreground curve at 43 bar and 450°C (Condensing steam turbine)

4.1.4 Theoretical electricity savings opportunities

Table 4.3 Summary of electric power output for the different sets of steam data

Type	Turbine inlet [bar]	Turbine outlet [bar]	Steam Temperature [°C]	Power [kW]	Electricity Savings [kWh/year]
Back pressure steam turbine	25	5	400	312	2’620’000
Back pressure steam turbine	28	5	410	317	2’660’000
Back pressure steam turbine	25	5	430	326	2’740’000
Back pressure steam turbine	50	5	400	388	3’260’000
Condensing steam turbine	25	0.1	400	585	4’900’000
Condensing steam turbine	43	0.1	450	634	5’300’000

The GCC of the process from ECS reactor, main reactor and ECS pre-heater does not change over time because these heat loads are constant under the assumption of constant maximum plant production.

From the theoretical heat recovery steam cycle calculated with pinch analysis we get 312 kW as shown in Figure 4.7.

The theoretical electricity savings that can be achieved with such steam turbine power cycle are around 2’620’000 kWh per year.

It is worth pointing out that from the interpretation of the process GCC, at least 4 heat exchangers (e.g. one between the steam superheating and ECS Reactor, one between water evaporation and main reactor, one between water preheating and ECS reactor and 4th between de-superheating and ECS pre-heater) appear necessary.

5 CONCEPTUAL DESIGN FOR PRACTICAL WORK RECOVERY IN THE FORMALDEHYDE PLANT

In this chapter a preliminary selection of turbo machinery can be done by means the well-known Balje diagrams approach [CITE] (Turbomachines). These diagrams are based on the turbo machinery similarity concept.

The primary objective for the turbo machine selection is to find a configuration that will give the highest efficiency that is achieving the desired compression with the minimum expansion power or the maximum compression with the available expansion power.

The network for steam and water flow in the new design is shown in *Figure 5.1*.

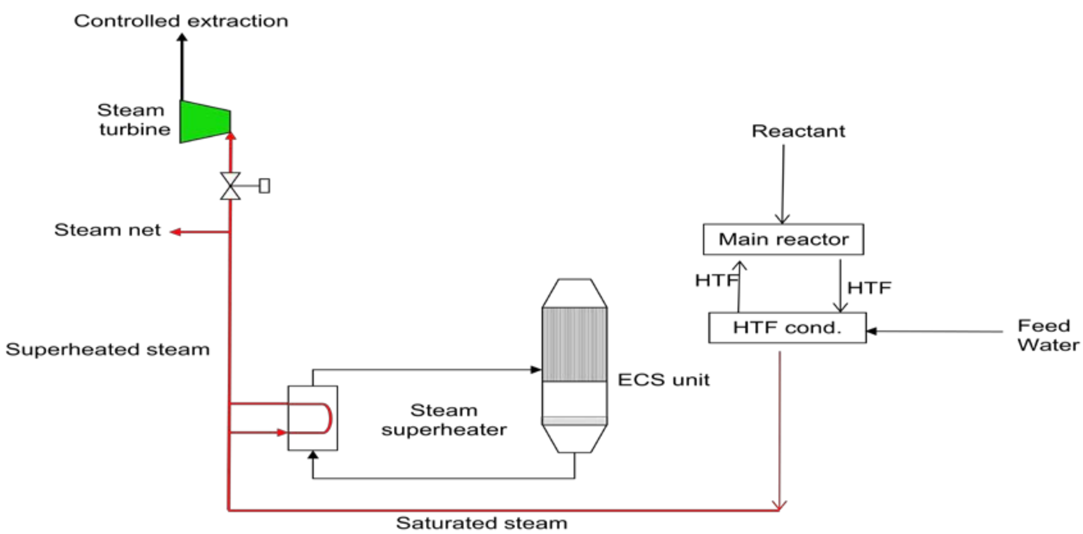

Figure 5.1 *Steam and water flow in the new design*

5.1 Method for preliminary selection of turbo machinery

In the case of turbine driven compressors (e.g. motor vehicle engine turbochargers) the revolutionary speed of the expander and of the compressor must be the same as they share the same shaft.

This means that the selection of turbine and the compressor stage is a compromise between the performances of the two machines to be solved through iterative procedures.

Accordingly, expected turbo machinery performances are a function of non-dimensional parameters such as the specific diameter d_s (which is a measure of the size of the machine) and the specific speed n_s (which is a measure of the capacity of a machine to convert work into kinetic energy or vice-versa).

The two non-dimensional parameters are function of the machine geometric and operating parameters and on fluid characteristics.

An overview of the expressions of the specific diameter and specific speed is given below. (Turbomachines)

Table 5.1 Specific speed ns and specific diameter ds of turbine and compressor

	Turbine	**Compressor**
Specific speed, ns	$ns = \frac{\omega.\sqrt{Vo}}{(\Delta His)^{0.75}}$	$ns = \frac{\omega.\sqrt{V1}}{(\Delta His)^{0.75}}$
Specific diameter, ds	$ds = \frac{D.(\Delta His)^{0.25}}{\sqrt{Vo}}$	$ds = \frac{D.(\Delta His)^{0.25}}{\sqrt{V1}}$

The so-called Balje diagram provides estimates of the total to static efficiencies for compressors and expansion stages. Lower efficiencies (about 3 to 7 %) result for “first built” units. For the convenience of application, a distinction is made between the machines using gases and liquids. Additionally, lines of constant optimum geometry can be plotted for constant values of hub ratio λ (e.g. the lines of constant λ values for axial turbines). *Figure 5.2Figure 5.2* shows lines of constant total to static efficiencies for different turbine and expander types as a function of specific speed and specific diameter operating with compressible media at a machine Reynolds number of $2*10^6$.

Higher efficiencies are obtained at specific speeds between 0.5 and 1.0. Single stage axial as well as radial full admission turbines cover this regime equally well. At higher or lower specific speeds the efficiency potential tends to decrease. For specific speeds below 0.1, partial admission turbines offer a higher efficiency potential than full admission turbines. Terry turbines have a lower efficiency potential than the partial admission axial turbines. Rotary displacement type expanders (multilobe) show higher efficiency in the low specific speed regime than dynamic machines. In general, for the dynamic type turbine the specific diameter tends to be higher than displacement type machines. Since the Mach number has a little influence on turbine so the same efficiencies can be obtained with higher or lower Mach number. There is also small increase in efficiency with increasing Reynolds number. The cross flow turbine covers a wide range of specific speeds, thus large specific speeds can be accommodated with small specific diameter. (Turbomachines)

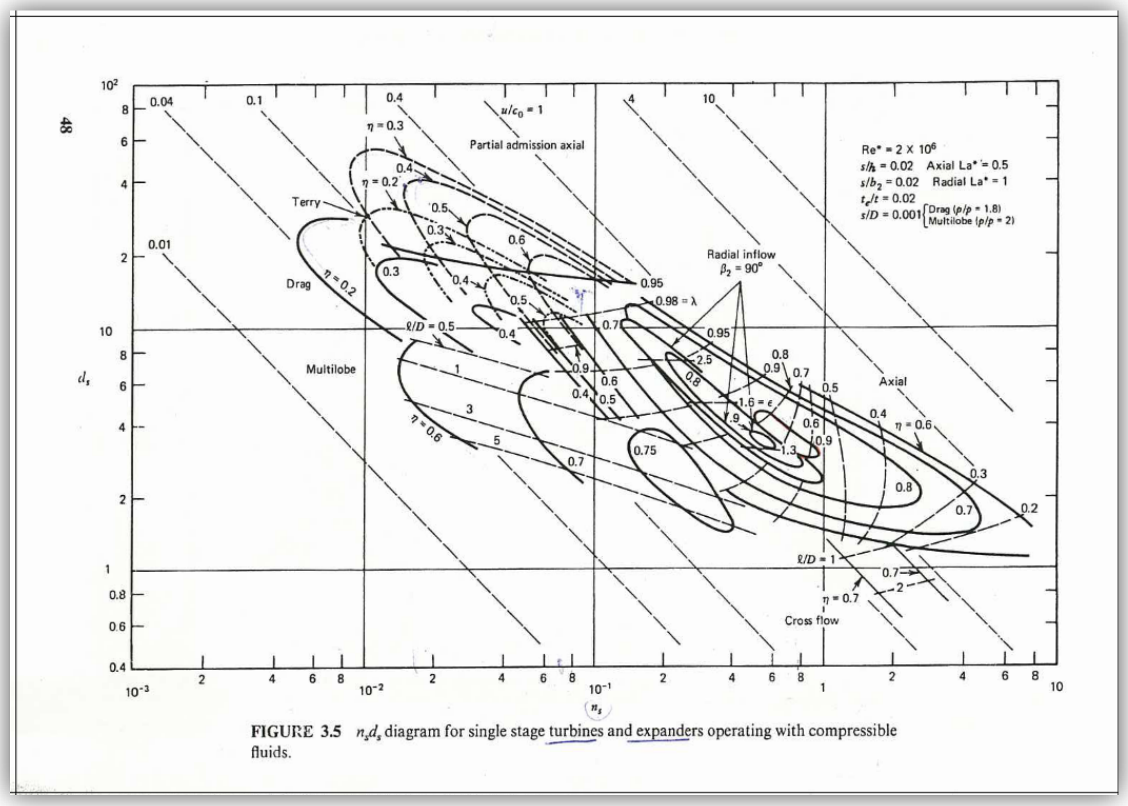

Figure 5.2 *$n_s d_s$ diagram for single stage turbine and expanders operating with compressible fluids* (Turbomachines)

A similar diagram is shown also for compressor stage in *Figure 5.3*. For dynamic machines highest efficiencies are obtained at specific speeds between 0.5 and 1.0 and the efficiency decreases generally with increasing or decreasing specific speeds. In the high specific speed regime axial compressor have higher efficiencies. Where-as radial compressors give better performance at low specific speeds. Mixed flow type compressors cover the specific speed regime between 1.0 and 2.0 best. For low specific speeds (n_s<0.2), partial emission compressors merit consideration, however they only offer a limited efficiency potential. There is definite limit exists for the applicability of single stage dynamic type machines. This limit can be identified by limiting head coefficient. When head coefficient exceeds values of 1.0 then usually extremely low efficiencies are obtained. This limit is identified by a dashed line in *Figure 5.3*, meaning that in the operating regime to the left of the dashed line, positive displacement machines or hybrid machines (drag pumps) offer a better performance. At very high specific speeds the cross flow blower offers a fair efficiency potential. (Turbomachines)

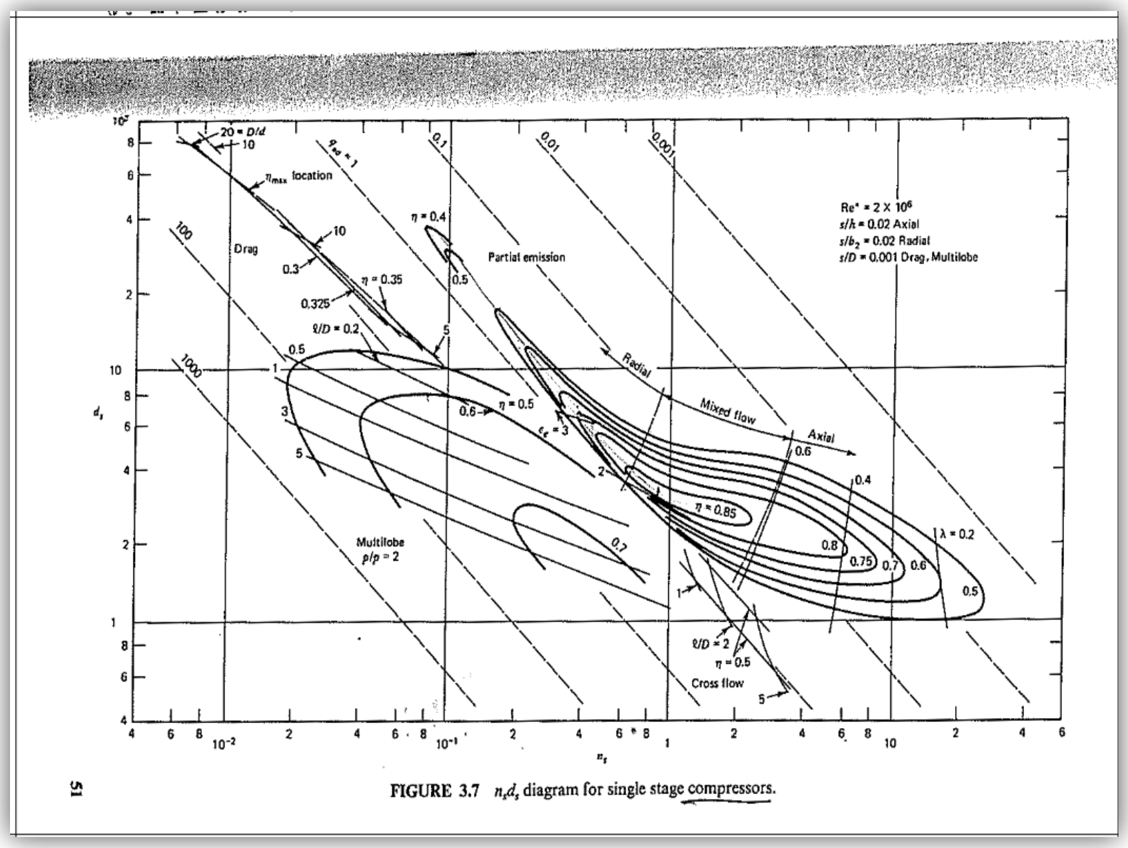

Figure 5.3 *$n_s d_s$ diagram for single stage compressor* (Turbomachines)

5.2 Estimation of practical potential of electricity generation

In the following section practical potential of power generation from the steam turbine and different configurations of steam driven blowers will be discussed.

5.2.1 Super heater

The steam for power production application must be superheated before expansion in order to avoid water droplets at the turbine outlet and in order to exploit a larger enthalpy drop for the given pressure ratio.

The following assumptions and data were used to calculate the practical potential of superheated steam according to the new process arrangement as shown in *Figure* 5.1. The data below is obtained from mass and energy balance,

Mass flow of feed water, m = 631 kg/hr
Specific heat of water, Cp= 4,226 kJ/kg.K
Feed water inlet temperature, Tin= 105℃
Feed water outlet temperature, $Tout$= 224℃
Pressure of feed water, P= 26 bara
Heat of vaporization, ΔH = 1840 kJ/kg (Wester, Oktober 1996).

$$Q = m * Cp * \Delta T + m * \Delta H \tag{5.1}$$

Heat load of Super heater is 410 kW.

Saturated steam coming from HTF Condenser (from the mass and energy balance);
Inlet temperature of Saturated steam, *Tin*= 224℃
Inlet pressure of Saturated steam, *Pin*= 25 bara
Enthalpy of Saturated steam, *hin*= 2802 kJ/kg (at *Pin* and *Tin)*
Mass flow of Saturated steam, *m*= 3390 kg/hr

$$Q = m * \Delta H \tag{5.2}$$
$$Q = m * (hout - hin) \tag{5.3}$$
$$hout = \frac{Q}{m} + hin \tag{5.4}$$

Outlet enthalpy comes out to be 3237 kJ/kg
The outlet temperature of superheated steam from steam library at (*Pin, hout*) is 399℃.

5.2.2 Concise overview of steam turbines

There are some advantages of back-pressure turbine over a condensing turbine, (Steam Turbine and Turboexpanders - chapter-8, 1997).

- Lowest in capital cost
- Most suitable for high speeds
- Simplest in construction and more reliable

There are also some advantages of condensing steam turbine which are;

- Control is easier because it requires less change in live steam for different loads
- The enthalpy drop across the turbine is larger so it requires less live steam for a given power output

The condensing turbine has some disadvantages as compared to back-pressure turbine,

- Longer blades due to high steam volumes
- Lower over-all reliability because of the need to provide condenser
- High initial cost due mainly to two factors:
    a) Large turbine due to high specific volume
    b) Extra cost of condenser
- Poor operating cost because two third of the steam heat content is used in heating condenser cooling water
- More costly boiler feed water treatment to remove chlorides, salts and silicates which would otherwise produce deposits or corrosion of the blades
- Failure of blades

5.2.3 Preliminary estimates of steam turbine power

The superheated steam from the super heater enters into the steam turbine. The estimated power from the steam turbine is 247 kW. The detailed calculation for estimation of steam turbine power is given in Appendix 9.2.

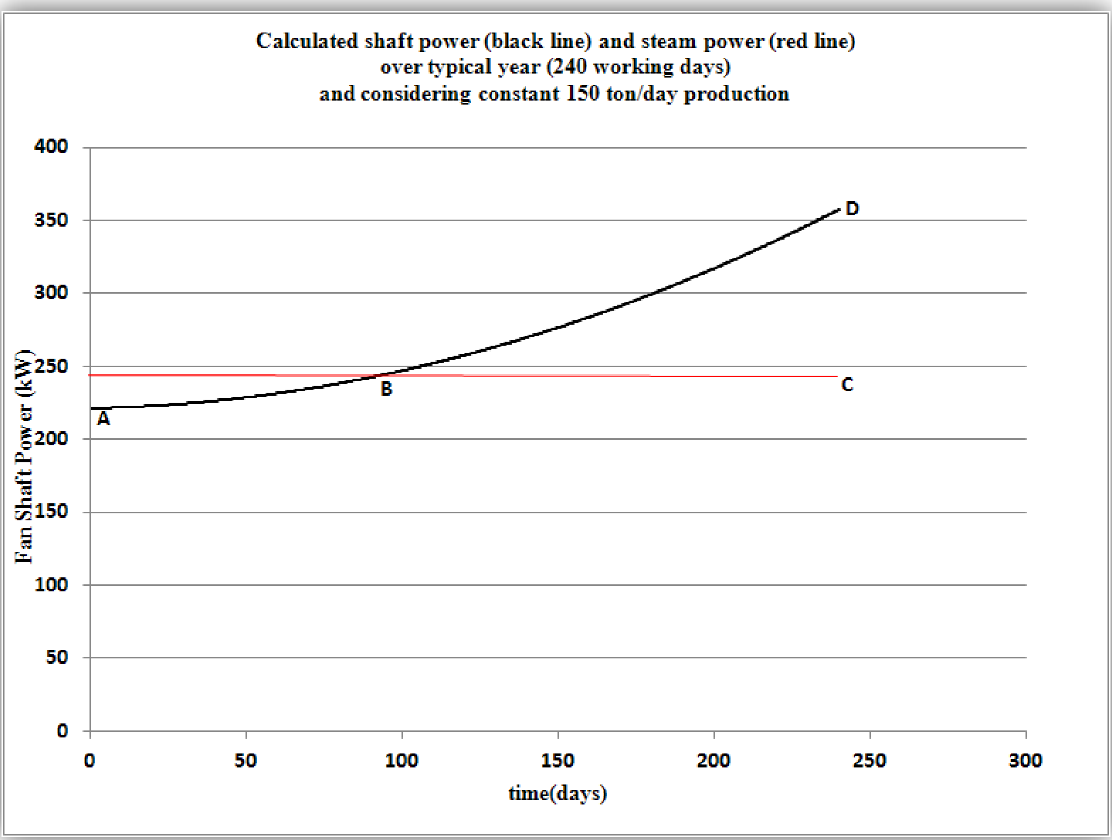

Figure 5.4   Estimate of practical potential of power output from the steam turbine against with the blower power consumption profile.

It can be seen from *Figure 5.4* that the power requirement is not always higher than the power generated by steam turbine. This means that if electricity storage in the batteries is not considered and if electricity is not sold, the steam power production must balance the power requirement. This means that the steam power available during the first 100 days cannot be fully exploited as the required blower power is less than 247kW

After 100 days, according to our hypothesis on catalyst behavior, the steam power is instead insufficient to fulfill the compression power requirement. From this time until catalyst reloading an extra electrical motor is required.

The maximum power of this motor should compensate the final power requirement of air compression with old catalyst which is around 112 kW.

Under this hypothesis, the mechanical energy produced by the steam turbine during the one catalyst charge cycle (240 days) is equal to the area under the surface ABC (1’400’000 kWh) in *Figure 5.4*. The electricity provided by the motor during the one catalyst charge cycle (240 days) is equal to the area of BCD (175000 kWh).

The net electricity required by blower system which must be provided with the electrical drivers corresponds to 175000 kWh over a typical load.

5.3 Screening of the concepts

According to the preliminary calculations shown above, the following turbo-machinery configurations are suggested.

5.3.1 Steam driven blowers

5.3.1.1 Double shaft option

In this arrangement one recirculation blower is driven by a turbine and the other recirculation blower is driven by an electrical motor. The possible configuration of this system is shown in *Figure 5.5.*

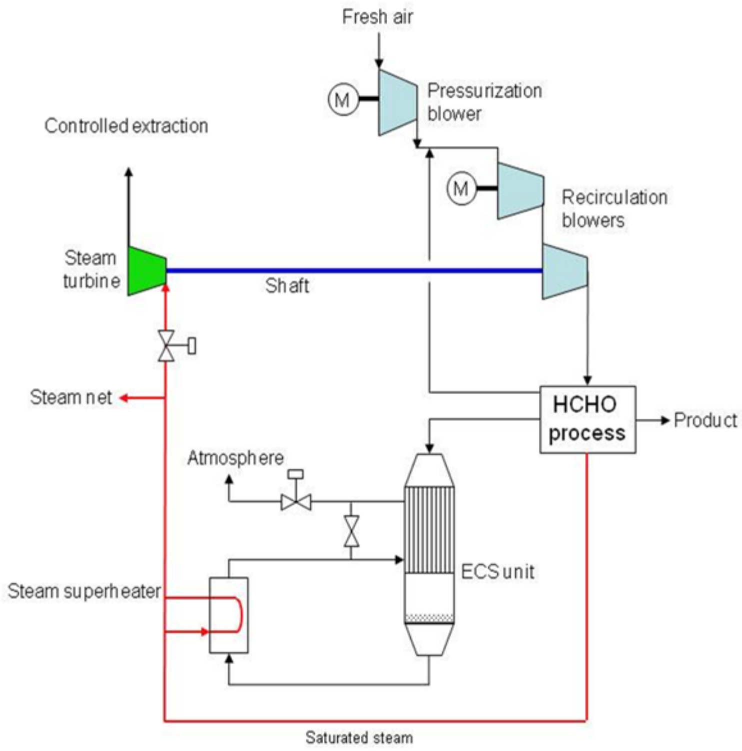

Figure 5.5 Sketch of double shaft configuration

5.3.1.2 Single shaft configuration

In this arrangement, the steam turbine is connected to one or more blowers in the same shaft (running at the same speed) and a motor is also coupled at the other end of the shaft as shown in *Figure 5.6.*

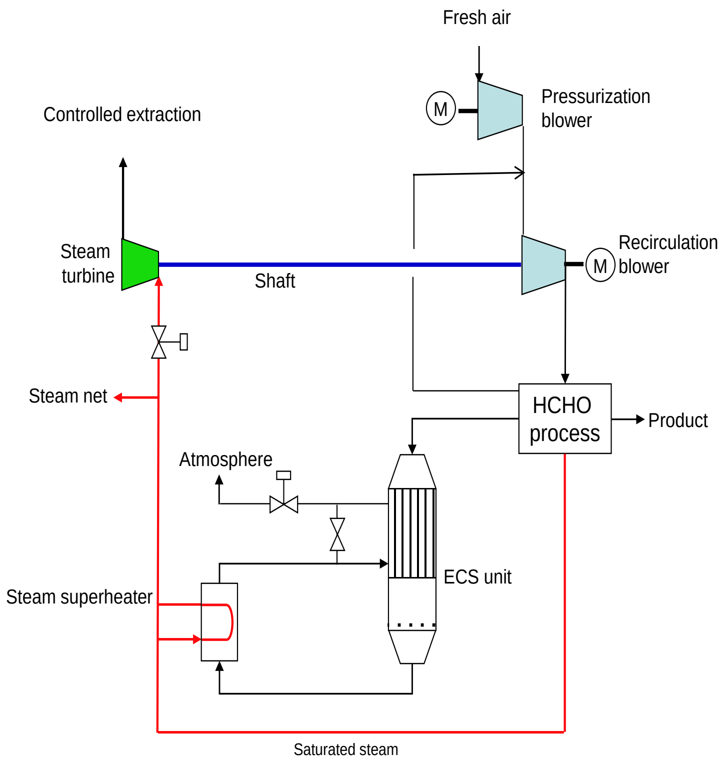

Figure 5.6 Sketch of single shaft configuration

5.3.2 Separate steam turbine set

A third configuration in which steam a turbine is connected with gear box, generator and transformer, as shown in *Figure 5.7*, appears also among the possible candidate solutions.

The investment cost for this system was estimated from (Chemical Engineering Revamps Boosting Capacity, April 2012) to be roughly double the investment cost for the other two configurations. At 150 ton/day we have low amount of steam available from the process. The possibility of using this configuration is not beneficial for this system. The power production potential from steam turbine is not good enough when we compare savings in terms of revenues. For the time being this configuration is discarded. In future if there is large amount of steam is available from the process, this configuration could be beneficial in terms of higher revenues.

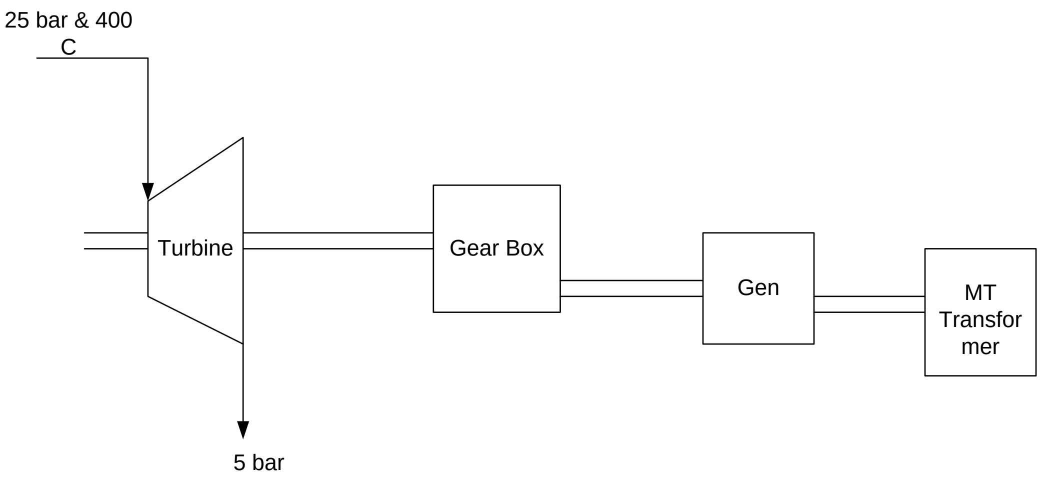

Figure 5.7 Example of Steam turbine drives the gear box

Table 5.2 Summary about all the configurations

	Double Shaft Configuration	Single shaft configuration	Separate steam turbine set
Investment cost	200’000 €	334’000 €	650’000 €
Revenues	94’000 €/year	146’000 €/year	182’000 €/year
Pros.	Invest cost is less	Higher efficiencies of turbo machinery, Less actual power consumption of electrical motor, More revenues in terms of electricity savings	Beneficial for large plant capacity (e.g. high amount of steam)
Cons.	Less efficiency of turbo machinery, More power consumption of electrical motor required for the first stage	Investment cost is higher	Investment cost is too high

6 SOLUTION FROM VENDORS

Purchasing cost of equipment was obtained by direct contact with vendors. The different vendors were provided with full specifications for the two possible steam-driven configurations (double shaft and single shaft).

Initially there were around 8 vendors contacted in which 5 vendors provided the estimated quote. The option suggested by the two vendor A and vendor B, were eventually selected as they appeared more technically reliable in terms of the quoted efficiencies for blowers and turbine compared to other vendors.

To obtain precise quotation by the vendors, three plant characteristic operating points were provided (OP1, OP2, and OP3). In reality the performance of the plant changes around the whole year so we took normal gas flow (OP1) as an operating point. Even though the plant performance varies but this operating point will accommodate all the changes. This is the maximum case (OP1). The other two cases related to electrical guarantee gas flow (OP2) and minimum gas flow (OP3).

Table 6.1 Characteristic operating point

Operating conditions	**Mass flow [kg/hr]**	**OP1 Inlet density [kg/m^3]**	**OP2 Static pressure inlet [bara]**	**OP3 Discharge [bara]**
Normal gas flow (+10%), max dP	23200	1.43	1.313	2.013
Electrical guarantee gas flow	21000	1.27	1.313	1.763
Min gas flow	12100	1.27	1.163	1.463

While enough indications about performances of the blower stages were attached to such quotations, only few non-relevant indications were provided for the turbine stages. So for the turbines we had to estimate the performances through preliminary considerations. Basically we took mass flow rate of steam and speed of compressor and from the Balje diagrams we estimate specific speed and efficiency of steam turbine.

For this reason, the expected turbine efficiency from the vendor's quote was verified with Balje diagrams, and accordingly the power requirement profiles for blowers and power production profile from the steam turbine estimated for a typical load.

6.1 Vendor A (Double Shaft Configuration)

The possible configuration of this system is shown in *Figure 5.5*.

Table 6.2 Data for Centrifugal compressor and turbine from vendor A

	Compressor	Turbine
Efficiency	75%	40%
Speed	3500 rpm	3500 rpm

6.1.1 Estimation of the blower nominal performances based on the quoted turbo-machinery speed

First we consider single stage compressor

The design conditions for Compressors are

P_{out}= 1.313+0.7=2.013 bar

P_{in}= 1.313 bar

ρ = 1.43 kg/ m^3

m= 21000 kg/hr, V_1= 4.08 m^3/sec

T_1= 323K

For air, k= 1.4 & Cp = 1.04kJ/kg.k

$$\Delta His = Cp.T1.\left[\left(\frac{pout}{pin}\right)^{k-1/k} - 1\right] \tag{6.1}$$

ΔHis= 43.4 kJ/kg

For chosen (from the vendor A quoted speed), n= 3500 rpm

$$\omega = \frac{2\pi n}{60} \tag{6.2}$$

ω= 366.34 rad/sec

$$ns = \frac{\omega.\sqrt{V1}}{(\Delta His)^{0.75}} \tag{6.3}$$

ns = 0.25→ Efficiency of Compressor is 50% (It is too low when we compare efficiency with the vendor's quote).

Accordingly, two stages must be chosen in order to distribute the desired pressure increase to achieve better performances at the same speed. This confirms the choice of two stages and is at the base of the two shaft configuration. Now if for 2 stage compressor,

ΔHis= 43.4/2 = 21.7 kJ/kg

$$ns = \frac{\omega.\sqrt{V1}}{(\Delta His)^{0.75}} \tag{6.4}$$

ns = 0.41→ Efficiency of Compressor is 75%.

Such value confirms the blower performances quoted by vendor A from *Table 6.2*.

6.1.2 Estimation of the blower power requirement profile based on the quoted characteristic curves

In order to estimate the actual power profile of the two blower stages along the one catalyst charge cycle (240 days), the performance characteristics provided by the vendor for the three typical plant operating point were “collapsed” into single performance curve according to the similarity theory.

This corresponds to plot the non-dimensional head coefficient ψ against the non-dimensional flow coefficient φ where

$$\psi = \frac{DHis}{n^2D^2} \tag{6.5}$$

$$\varphi = \frac{V}{nD^3} \tag{6.6}$$

According to similarity concepts the non-dimensional head coefficient and the machine efficiency assume equal numerical values in dynamically similar operating point (that is when all fluid velocities at corresponding points within the machine are in the same direction and proportional to the blade speed), resulting in the following type of diagram:

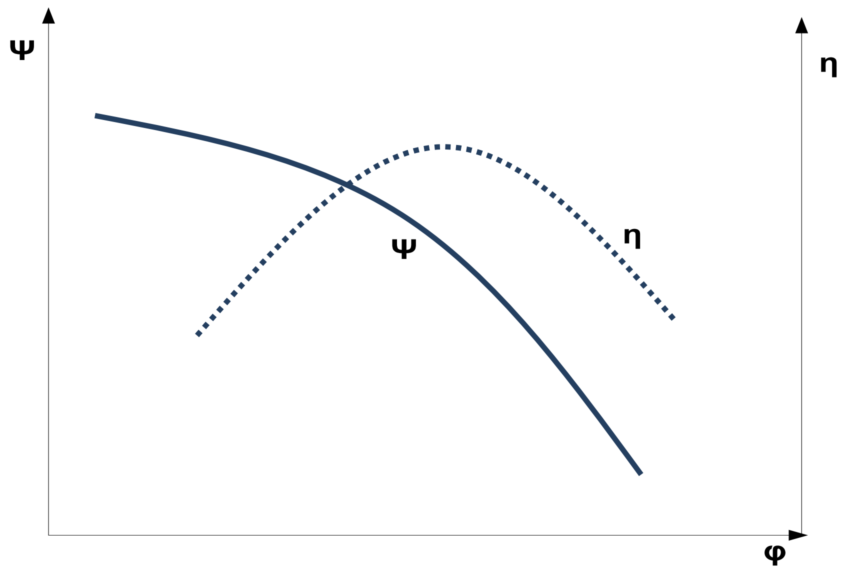

Figure 6.1 Non-dimensional flow coefficient φ against non-dimensional head coefficient ψ and efficiency η

In quotation from vendor A, the two stages are different so two different non-dimensional curves were built.

In addition, they suggest operating the two stages in such a way that the first stage follows the major part of the increasing total head while the second stage operates almost at constant pressure ratio.

By following such indications, the operating points of the two blowers were estimated. In particular as the required head changes over the time, the speed of the blower’s increases and therefore the non-dimensional flow coefficient diminish as the volumetric flow rate is constant (constant gas flow for nominal plant operation) but the turbo machinery speed increases. Accordingly the efficiency of the machines also changes during the typical year.

It can be seen in the *Table 6.3* that the design of double shaft configuration is particularly inefficient when the catalyst is fresh but the performances increases at higher speeds when the catalyst gets older.

Due to increased head of the first stage, the gas density at the second stage inlet progressively increases thus reducing the volumetric flow rate. This causes the second stage performance to be influenced by the first stage.

The speed of revolution of the blower also increases due to higher power requirement, and it is calculated as shown in *Table 6.3*.

Table 6.3 Calculated blower power requirements for 1st and 2nd stage

		1st stage			2nd stage			Total blower power
days	Δp [bar]	n [1/min]	power [kW]	Eff.	n	η	power [kW]	[kW]
0	0,47	3184	154	0,69	3168	0,71	106	261
10	0,471	3186	155	0,69	3168	0,71	106	261
20	0,472	3192	155	0,69	3169	0,71	106	262
30	0,475	3201	157	0,69	3170	0,71	107	263
40	0,479	3214	158	0,69	3172	0,71	107	265
50	0,485	3231	160	0,69	3174	0,71	107	267
60	0,491	3252	163	0,69	3176	0,71	108	270
70	0,499	3276	166	0,69	3179	0,71	108	274
80	0,508	3303	169	0,69	3182	0,71	108	278
90	0,519	3334	173	0,69	3185	0,72	109	282
100	0,530	3369	178	0,70	3188	0,72	110	287
110	0,543	3406	182	0,70	3191	0,72	110	293
120	0,557	3447	187	0,70	3195	0,72	111	298
130	0,572	3491	193	0,70	3199	0,72	112	305
140	0,589	3538	199	0,70	3206	0,73	113	311
150	0,606	3587	205	0,71	3214	0,73	114	319
160	0,625	3639	211	0,71	3222	0,73	115	326
170	0,645	3694	218	0,71	3230	0,74	116	334
180	0,666	3751	225	0,72	3238	0,74	117	342
190	0,689	3810	233	0,72	3245	0,74	118	351
200	0,713	3871	241	0,72	3251	0,74	119	359
210	0,738	3934	248	0,73	3260	0,75	120	368
220	0,764	3998	257	0,73	3270	0,75	121	378
230	0,791	4064	265	0,73	3281	0,75	122	387
240	0,820	4131	273	0,74	3290	0,76	124	397

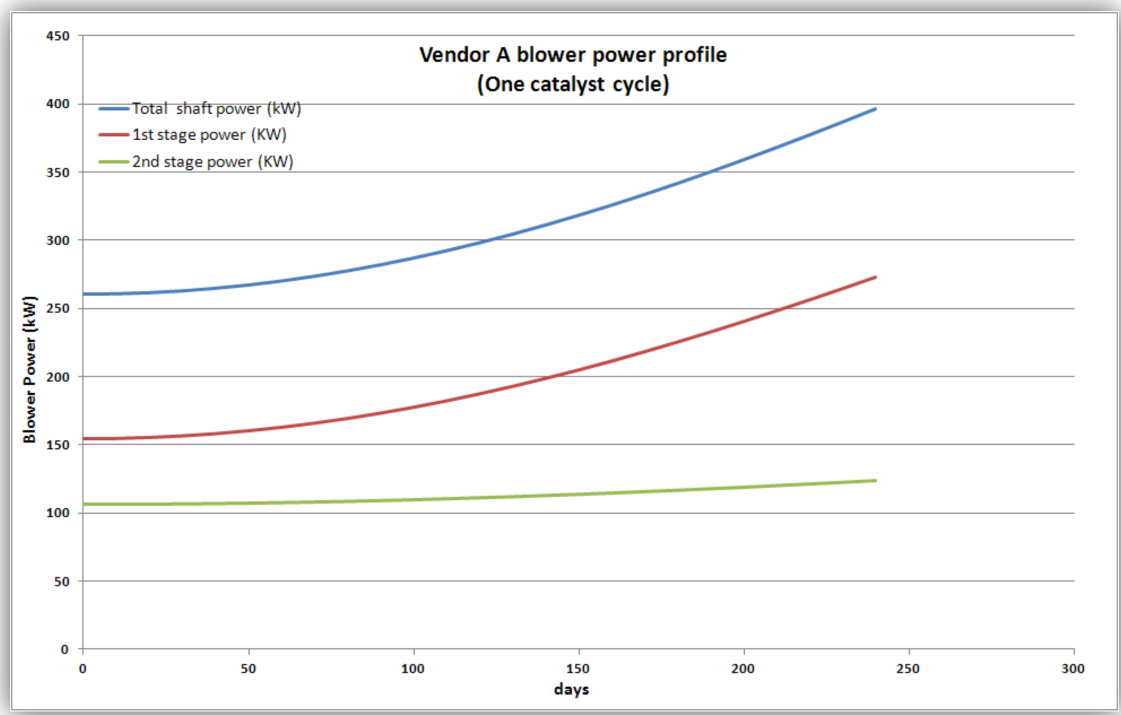

Figure 6.2 Blower power profile for a typical load

6.1.3 Preliminary estimation of the turbine performance based on the quoted nominal speed

The type of steam turbine used by vendor A is partial admission axial type and this is confirmed by looking at the Balje diagrams for the corresponding specific speed and specific diameters.

Design conditions for Turbine

P_{out}= 5 bar

P_{in}= 25 bar

T_1= 400°C

ρ at (5bar & 270°C) = 2.03 kg/ m^3

m= 3390 kg/hr, V_1= 0.464 m^3/sec

From the Mollier Diagram ΔHis= 405 kJ/kg

For a preliminary value of the nominal speed of 3500 rpm

$$\omega = \frac{2\pi n}{60} \tag{6.7}$$

ω= 366.34 rad/sec

$$ns = \frac{\omega.\sqrt{V1}}{(\Delta His)^{0.75}} \tag{6.8}$$

ns = 0.015→ Efficiency of Steam Turbine is 40%.

This very low turbine efficiency value is confirmed from the Balje diagrams for vendor A as shown in *Table 6.2*.

The procedure for the preliminary design of turbine is summarized in the table below. In particular it was assumed that the nominal rotational speed is the maximum rotational speed for the blower stage coupled with the turbine (3290 rpm).

Under this assumption, the isentropic spouting velocity c0 is calculated and the ratio between the tangential velocity u (calculated by selecting the appropriate machine diameter from Balje diagrams) and c0 estimated.

Table 6.4 Preliminary turbine design

average n blowers	3208				
max n blower	3289				
n_design	3290		ω [rad/s]	344,5	
p0 [bar]	25		h0 [kJ/kg]	3239	
p2 [bar]	5		s0[kJ/kgK]	7,0	
T0 [degC]	400		h2is[kJ/kg]	2835	
			Δhis[kJ/kg]	404	
m [kg/s]	0,94				
			c0 [m/s]	899	
			1 stage		
			guess values		
check the calculations with results from Balje			$\eta$is	0,4	
			Δh [kJ/kg]	161	
			h2 [kJ/kg]	3078	
			s2 [kJ/kg]	7,4	
			T2 [degC]	306,5	
			ρ [m3/kg]	1,89	
			V2 [m3/s]	0,49	
			ns	0,01	
			0,015149	1	
			0,015149	100	
			from Balje		
			ds	35	
			$\eta$is	0,4	
			D [m]	0,97	
for values higher than 300 check stress !			u [m/s]	168	

			u/c0	0,18	
			c* [m/s]	585	
check flow is choked			Mach guess	1,53	
			objective: keep the same u/c0		
			to keep high the turbine efficiency		

6.1.4 Estimation of the turbine power based on the obtained blower shaft speed

The ratio (u/c0) defines the operating point of maximum turbine performance and should be therefore kept constant during turbine operation.

As the shaft speed changes according to the increasing head that the blower has to provide, the isentropic spouting velocity c0 must therefore be adjusted to keep the turbine in high performance regimes.

This must be done by adjusting the inlet pressure p0 available at the turbine inlet. In particular, if this is equal to 25 bar when the turbine is operating at maximum power (old catalyst), the pressure at the turbine inlet is much lower at lower powers (new catalyst).

This also causes the specific work of the turbine stage to change as a consequence of the changing available theoretical head, so the required power can be provided by admitting more or less steam.

In fact, at the turbine inlet the flow is chocked (M=1), which means that for each inlet nozzle a given constant steam flow rate is admitted. Accordingly, the mass flow rate can be changed only by adjusting the partial admission that is by opening or closing some inlet nozzles.

The remaining mass flow rate that is not required for power generation is throttled directly to the low pressure and used for heating purposes.

In *Table 6.5* the turbine operation during the typical load is described.

Table 6.5 Turbine power for 2nd stage

	turbine						
			from blower	turbine	throttle to heating		
days	n [rev/min]	p0 [bar]	power req [kW]	m [kg/s]	m [kg/s]	Q [kW]	Heating [kWh]
0	3168	22	106	0,71	0,23	2487	0
10	3168	22	106	0,71	0,23	2487	596880
20	3169	22	106	0,71	0,23	2487	596880
30	3170	22	107	0,71	0,23	2487	596880
40	3172	22	107	0,71	0,23	2486	596640
50	3174	22	107	0,71	0,23	2486	596640
60	3176	22	108	0,71	0,23	2486	596640
70	3179	22	108	0,71	0,23	2485	596400
80	3182	22	108	0,72	0,23	2485	596400
90	3185	22	109	0,72	0,22	2484	596160
100	3188	22	110	0,72	0,22	2484	596160
110	3191	22	110	0,72	0,22	2483	595920
120	3195	22	111	0,73	0,21	2482	595680
130	3199	23	112	0,73	0,21	2482	595680
140	3206	23	113	0,73	0,21	2481	595440
150	3214	23	114	0,74	0,21	2480	595200
160	3222	23	115	0,74	0,20	2479	594960
170	3230	23	116	0,74	0,20	2478	594720
180	3238	24	117	0,74	0,20	2477	594480
190	3245	24	118	0,75	0,19	2476	594240
200	3251	24	119	0,75	0,19	2475	594000
210	3260	24	120	0,76	0,19	2474	593760
220	3270	24	121	0,76	0,18	2472	593280
230	3281	25	122	0,76	0,18	2471	593040
240	3290	25	124	0,76	0,18	2470	592800
						total	14288880

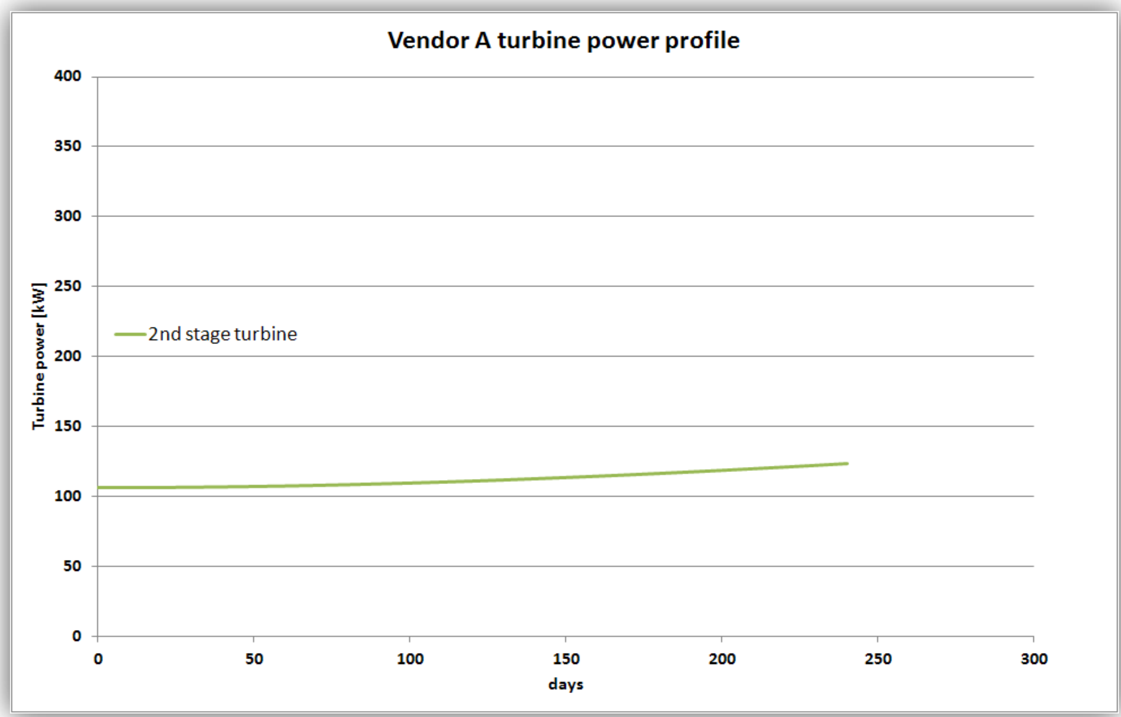

Figure 6.3 Steam turbine power profile (Blower 2nd stage is connected with the steam turbine)

6.1.5 Electricity savings

The theoretical electricity savings and the required power of the electrical motor for the first stage are given in *Table 6.6*. Initially we did all the calculations for one catalyst cycle (240 days). In order to calculate annual electricity savings for 350 days we multiply the theoretical electricity savings by a factor of (350 days/240 days) 1.46. The total theoretical electricity savings is then 940’000 kWh/year. The total electricity consumption is 2’600’000 kWh/year.

Table 6.6 Electricity savings for the double shaft case over a typical load

Days	Total Shaft Power (kW)	Steam Power for 2nd stage (kW)	Theoritical Electrical Saving (kWh)	Electricity Consumption (kWh)	Actual power consumption of Electrical Motor for 1st stage (kW)
0	260	106	0	0	154
10	261	106	25440	62640	155
20	261	106	25440	62640	155
30	263	106	25440	63120	157
40	265	107	25680	63600	158
50	267	107	25680	64080	160
60	270	107	25680	64800	163
70	273	108	25920	65520	165
80	277	108	25920	66480	169
90	282	109	26160	67680	173
100	287	109	26160	68880	178
110	292	110	26400	70080	182
120	298	111	26640	71520	187
130	304	112	26880	72960	192
140	311	113	27120	74640	198
150	318	113	27120	76320	205
160	326	114	27360	78240	212
170	333	115	27600	79920	218
180	342	116	27840	82080	226
190	350	118	28320	84000	232
200	359	119	28560	86160	240
210	368	120	28800	88320	248
220	378	121	29040	90720	257
230	387	122	29280	92880	265
240	397	124	29760	95280	273
			648240	1792560	

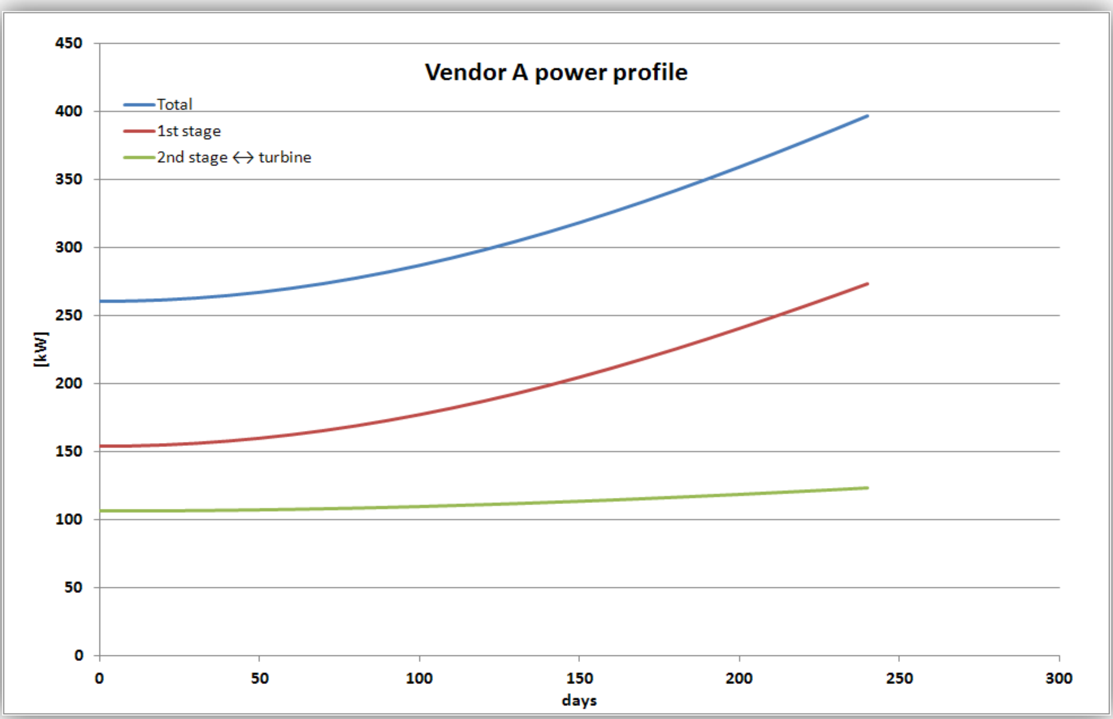

Figure 6.4 Turbo machinery power profile over a typical load

Table 6.7 Summary about double shaft configuration

	Double shaft configuration
Steam available for sale in the reference plant (For a steam price of 20€/ton of steam) Appendix 9.4	675’000 €/year
Steam available for sale in the double shaft configuration (For a steam price of 20€/ton of steam) Appendix 9.4	645’000 €/year
Electricity generation	940’000 kWh/year
Cash flows	94’000 €/year

6.1.6 Profitability analysis

6.1.6.1 Investment cost

The investment cost was obtained from the direct contact with vendors. The investment cost for the double shaft configuration was estimated to be 200’000 €/unit. This investment cost includes the super heater, piping, valves, engineering cost, controls, and contingency.

6.1.6.2 Annual savings

The annual savings is calculated from the available steam turbine power. The price of electricity is assumed to be 0.1 €/kWh. The savings for the double shaft case is 94’000 €.

6.1.6.3 Payback time

The time elapses from the start of the project to the breakeven point. The shorter is the payback time the more attractive is the project.

$$PBT = \frac{Investment}{ACF\ (Annual\ Cash\ Flow)} \tag{6.9}$$

6.1.6.4 Net present value

The effect of time on the value of money is taken into account by discounting the annual cash flow A_{CF} with the rate of interest to obtain annual discounted cash flow A_{DCF}. The sum of ΣA_{DCF} over 10 years is known as NPV.

$$NPV = \sum_{yera=0}^{10} Discounted\ cash\ flow_{year} - Initial\ Investment\ cost \tag{6.10}$$

The greater the positive *NPV* for a project, the more attractive is the investment.

6.1.6.5 Electricity prices

Different prices of electricity were taken into account in order to estimate the profitability of the different configurations if they were to be implemented in different countries.

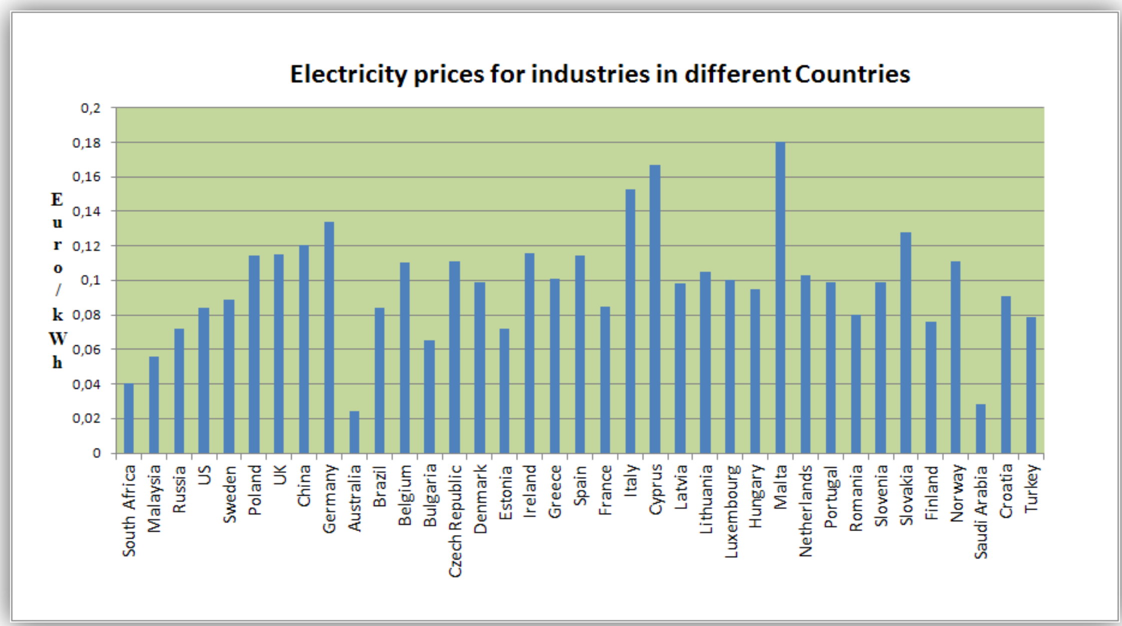

Figure 6.5 Electricity prices in different countries (Portel, 2011)

Without considering the economic value of heat sales from the double shaft case, the profitability analysis is given in Figure 6.6. The revenues in the table below came from the electricity savings.

Table 6.8 Investment and Revenues for double shaft configuration

Price of electricity	0.1 €/kWh
Electricity savings	940’000 kWh/year
Investment	200’000 €
Revenues	94’000 €/year

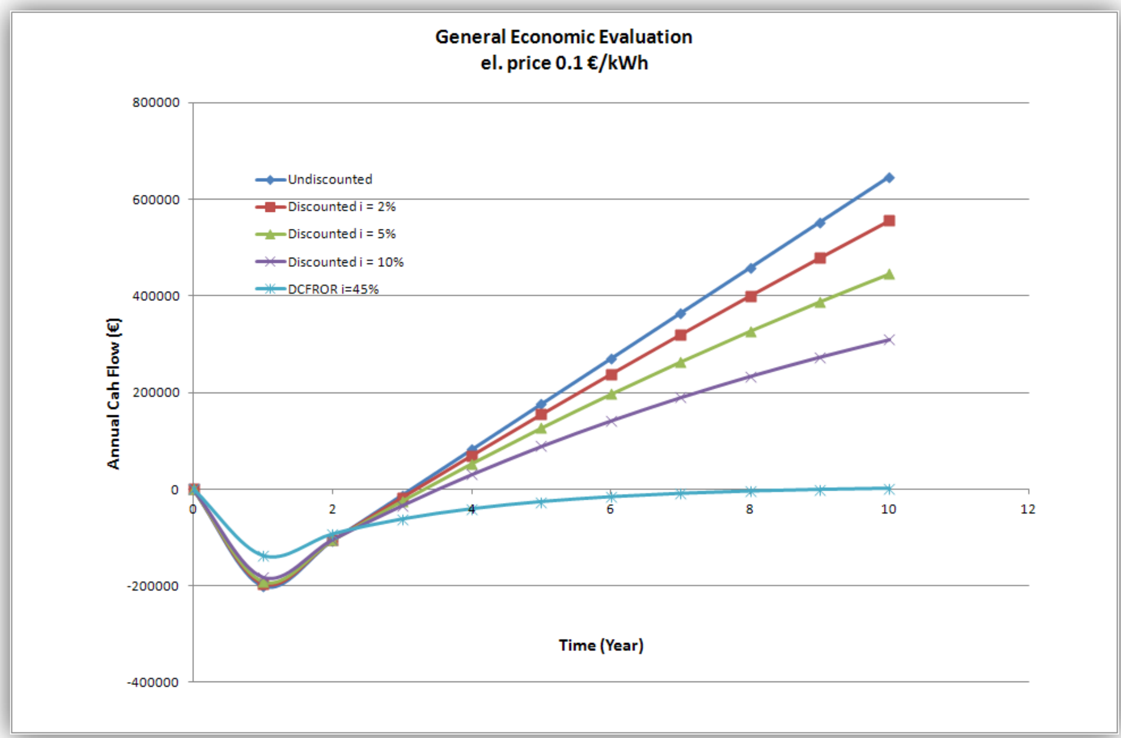

Figure 6.6 Profitability analysis without heat sales

If we consider economic value of heat sales, as a comparison of the double shaft case with the standard case the difference will be the loss of money that is 30000 €/year. Appendix 9.4

So the revenues from the double shaft configuration is 64000 €/year, shown in *Figure 6.7*.

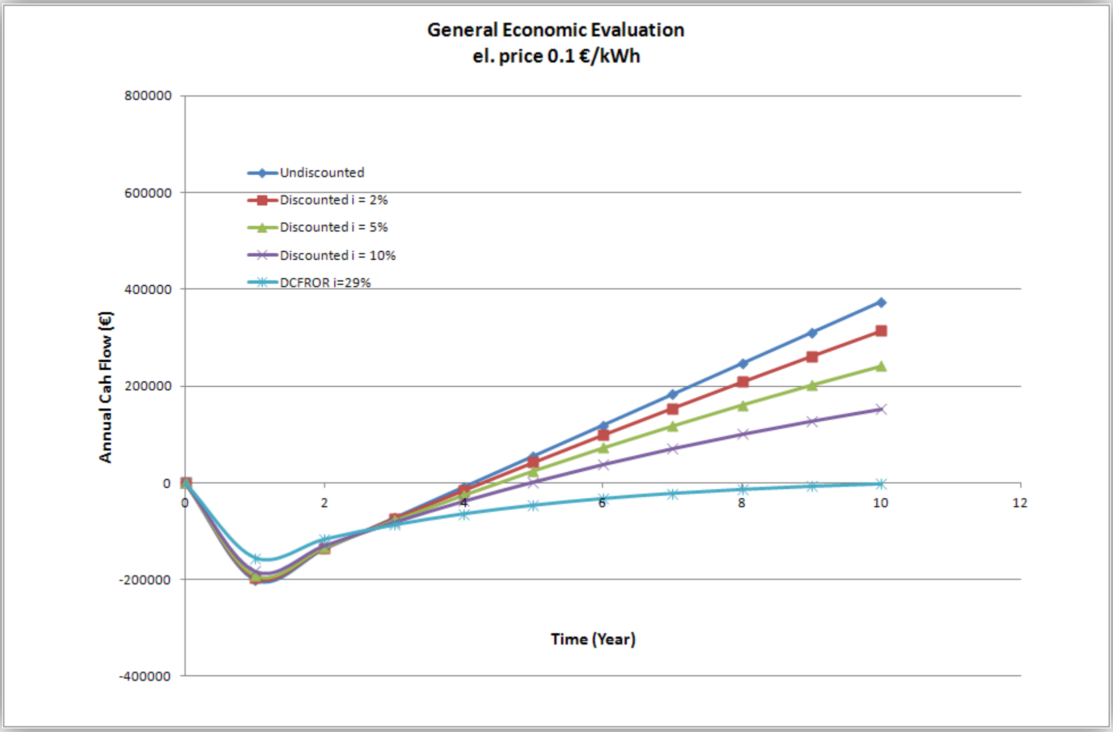

Figure 6.7 Profitability analysis with considering heat sales

6.2 Vendor B (Single Shaft Configuration)

The method from (Turbomachines) used to verify the performances of compressors and turbine for double shaft is also used to verify the performances for single shaft configuration and to estimate the actual system operation during the one catalyst charge cycle (240 days). The possible configuration of this system is shown in *Figure 5.6*. The turbo machinery speed in the single shaft case is much higher than in the double shaft case, so higher efficiencies are expected from the Balje diagrams.

Table 6.9 Data for Centrifugal compressor and turbine from vendor B

	Compressor	Turbine
Efficiency	85%	60%
Speed	12000 rpm	12000 rpm

6.2.1 Estimation of the blower nominal performances based on the quoted turbo-machinery speed

First we consider single stage compressor

Design conditions for Compressors

P_{out}= 1.313+0.7=2.013 bar

P_{in}= 1.313 bar

ρ = 1.43 kg/ m^3

m= 21000 kg/hr, V_1= 4.08 m^3/sec

T_1= 323K

For air, k= 1.4 & Cp = 1.04kJ/kg.k

$$\Delta His = Cp.T1.[\left(\frac{pout}{pin}\right)^{k-1/k} - 1] \tag{6.11}$$

ΔHis= 43.4 kJ/kg

For chosen (from the vendor B quoted speed), n= 12000 rpm

$$\omega = \frac{2\pi n}{60} \tag{6.12}$$

ω= 1256 rad/sec

$$ns = \frac{\omega.\sqrt{V1}}{(\Delta His)^{0.75}} \tag{6.13}$$

ns = 0.84→ Efficiency of Compressor is 85%. Such value is confirmed by vendor B as shown in *Table 6.9.*

6.2.2 Estimation of the blower power requirement profile based on the quoted characteristic curves.

In quotation from vendor B, they have only one stage so only one dimensional curve was built. The speed of revolution of the blower is also quite high, so they can handle all the head requirements in a single stage.

In addition, they suggest operating the single stage in such a way that blower, turbine and motor are connected on a single shaft. In particular as the required head changes over the time, the speed of the blower increases and therefore the non-dimensional flow coefficient diminish as the volumetric flow rate is constant (constant gas flow for nominal plant operation) but the turbo machinery speed increase. Accordingly the efficiency of the machines also changes during the typical year.

It can be seen in the *Table 6.10*, that single shaft design is particularly efficient both when the catalyst is fresh and when the catalyst gets older, due to their high revolutionary speed and efficiencies and the blower power is calculated as shown in table below.

Table 6.10 Calculated blower power requirements

days	Δp [bar]	n [rev/min]	η	Total blower power [kW]
0	0,470	10212	0,789	227
10	0,470	10215	0,790	228
20	0,472	10230	0,790	228
30	0,475	10256	0,790	230
40	0,479	10295	0,791	231
50	0,485	10330	0,791	233
60	0,491	10385	0,792	236
70	0,499	10445	0,793	239
80	0,508	10520	0,794	243
90	0,519	10595	0,794	247
100	0,538	10680	0,795	252
110	0,543	10780	0,796	257
120	0,557	10880	0,796	263
130	0,572	11000	0,796	269
140	0,589	11110	0,796	276
150	0,606	11240	0,796	283
160	0,625	11370	0,796	291
170	0,645	11510	0,795	299
180	0,666	11660	0,795	308
190	0,689	11800	0,793	317
200	0,713	11970	0,792	327
210	0,738	12120	0,790	338
220	0,764	12290	0,788	349
230	0,791	12460	0,786	361
240	0,820	12635	0,784	373

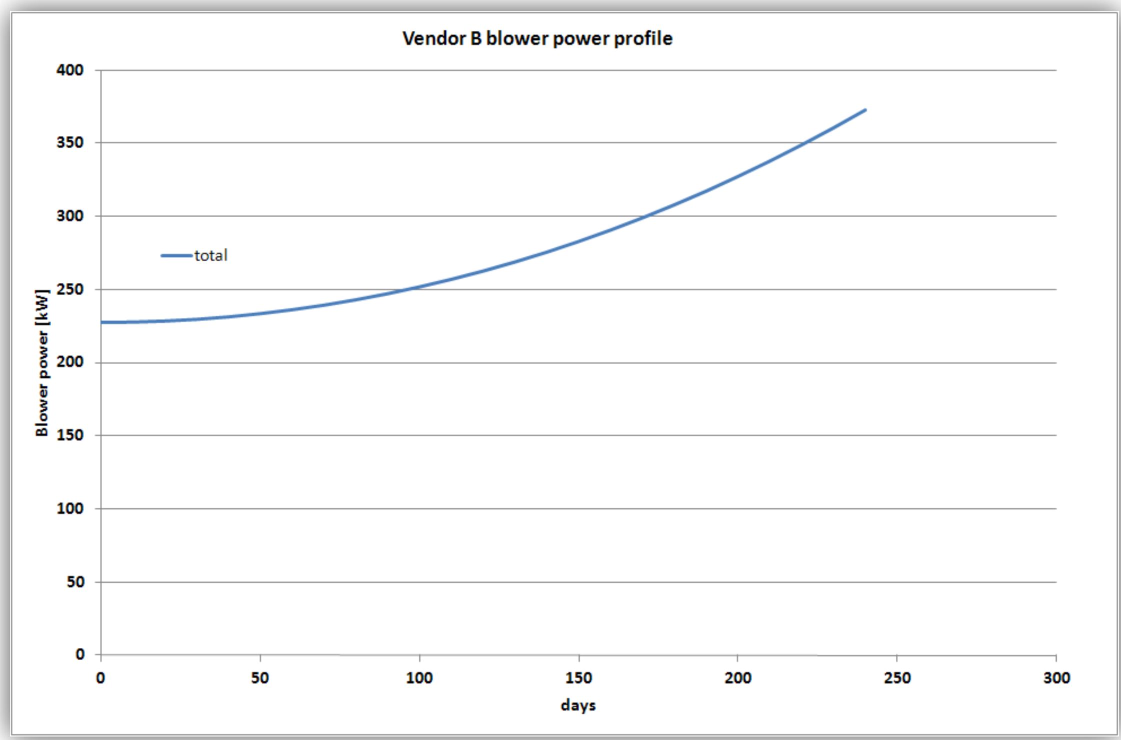

Figure 6.8 Blower power profile

6.2.3 Preliminary estimation of the turbine performance based on the quoted nominal speed

The type of steam turbine used by vendor B is partial admission axial type.

Design conditions for Turbine

P_{out}= 5 bar

P_{in}= 25 bar

T_1= 400°C

ρ at(5bar & 270°C) = 2.03 kg/ m^3

m= 3390 kg/hr, V_1= 0.464 m^3/sec

From the Mollier Diagram ΔHis= 405 kJ/kg

For n= 12000 rpm

$$\omega = \frac{2\pi n}{60} \tag{6.14}$$

ω= 1256 rad/sec

$$ns = \frac{\omega.\sqrt{V1}}{(\Delta His)^{0.75}} \tag{6.15}$$

ns = 0.053→ Efficiency of Steam Turbine is 60%. This turbine efficiency value is confirmed from the Balje diagrams for vendor B.

Table 6.11 Preliminary turbine design

average n blowers	*11079*	*ω [rad/s]*	*1323*
max n blower	*12635*		
n_design	*12635*	*h0 [kJ/kg]*	*3239*
		s0 [kJ/kg-K]	*7,0*
p0 [bar]	*25*	*h2is [kJ/kg]*	*2835*
p2 [bar]	*5*	*Δhis [kJ/kg]*	*404*
T0 [degC]	*400*		
		c0 [m/s]	*899*
m [kg/s]	*0,94*		
		guess values	
		ηis	*0,61*
		Δh [kJ/kg]	*246*
		h2 [kJ/kg]	*2993*
		s2 [kJ/kg]	*7,3*
		T2 [degC]	*265*
		ρ [m3/kg]	*2,0*
		V2 [m3/s]	*0,46*
		ns	*0,05*
		from Balje	
		ds	*17*
		ηis	*0,61*
		D [m]	*0,45*
		u [m/s]	*302,6*
		u/c0	*0,33*
		c [m/s]*	*564,4*
		Mach guess	*1,59*

6.2.4 Estimation of the turbine power based on the obtained blower shaft speed

As the shaft speed changes according to the increasing head that the blower has to provide, the isentropic spouting velocity c0 must therefore be adjusted to keep the turbine in high performance regimes.

This must be done by adjusting the mass flow rate that gives inlet pressure p0 available at the turbine inlet. In particular, if this is equal to 25 bar when the turbine is operating at maximum power (old catalyst), the pressure at the turbine inlet is much lower at lower powers (new catalyst).

In fact, at the turbine inlet the flow is choked (M=1), which means that for each inlet nozzle a given constant steam flow rate is admitted. Accordingly, the mass flow rate can be changed only by adjusting the partial admission that is by opening or closing some inlet nozzles.

The remaining mass flow rate that is not required for power generation is throttled directly to the low pressure and used for heating purposes.

As the speed of revolution in case of single shaft is higher than double shaft, so steam turbine is generating more power. Due to their high revolutionary speed, their efficiencies are also higher than double shaft.

Table 6.12 Power from steam turbine

days	n [rev/min]	p0 [bar]	Turbine power [kW]
0	10212	13	152
10	10215	13	152
20	10230	13	152
30	10256	13	153
40	10295	14	154
50	10330	14	155
60	10385	14	157
70	10445	14	159
80	10520	14	161
90	10595	14	164
100	10680	15	166
110	10780	15	169
120	10880	15	172
130	11000	16	176
140	11110	16	180
150	11240	17	184
160	11370	17	188
170	11510	18	193
180	11660	19	198
190	11800	20	203
200	11970	21	209
210	12120	21	214
220	12290	23	220
230	12460	24	226
240	12635	25	233

How the turbine power profile varies with time is shown in *Figure 6.9*.

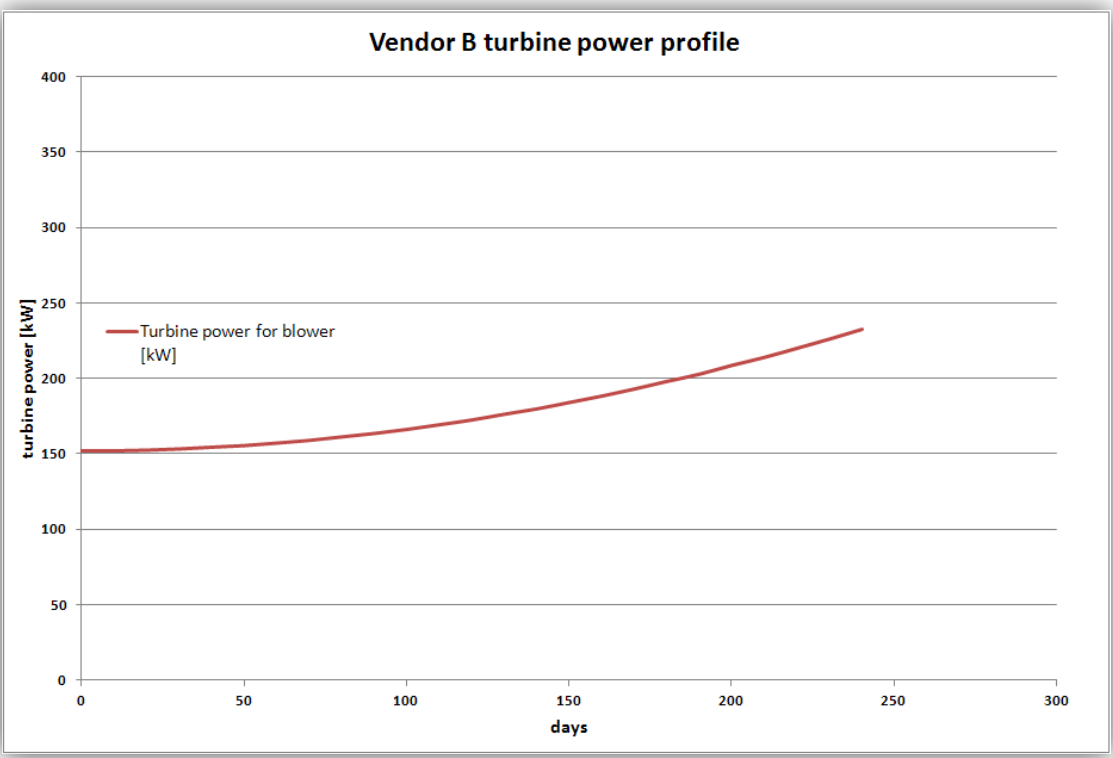

Figure 6.9 Turbine power profile

6.2.5 Electricity savings

The electricity savings and the power required by the electrical motor are given in *Table 6.6* and *Table 6.13*. Initially we did all the calculations for one catalyst cycle (240 days). In order to calculate annual electricity savings for 350 days we multiply the theoretical electricity savings by a factor of (350 days/240 days) 1.46. The total theoretical electricity savings from single shaft configuration is 1’460’000 kWh/year. The total electricity consumption for single shaft configuration is 2’240’000 kWh/year. The theoretical electrical savings are more in single shaft configuration than in the double shaft configuration; also with single shaft system actual power consumption of electrical motor is less than the double shaft system.

Table 6.13 Electricity savings for single shaft case over a typical load

days	Total shaft power (kW)	Turbine power for blower (kW)	Theoritical Electrical Saving (kWh)	Electricity Consumption (kWh)	Actual power consumption of electrical motor (kW)
0	227	152	0	0	75
10	228	152	36480	54720	76
20	228	152	36480	54720	76
30	230	153	36720	55200	76
40	231	154	36960	55440	77
50	233	155	37200	55920	78
60	236	157	37680	56640	79
70	239	159	37920	57360	80
80	243	161	38640	58320	82
90	247	164	39120	59280	84
100	252	166	39840	60480	86
110	257	169	40560	61680	88
120	263	172	41280	63120	90
130	269	176	42240	64560	93
140	276	180	42960	66240	96
150	283	184	44160	67920	99
160	291	188	45120	69840	102
170	299	193	46080	71760	106
180	308	198	47520	73920	110
190	317	203	48480	76080	115
200	327	209	49920	78480	119
210	338	214	51120	81120	124
220	349	220	52800	83760	129
230	361	226	54240	86640	135
240	373	233	55680	89520	140
			1039200	1602720	

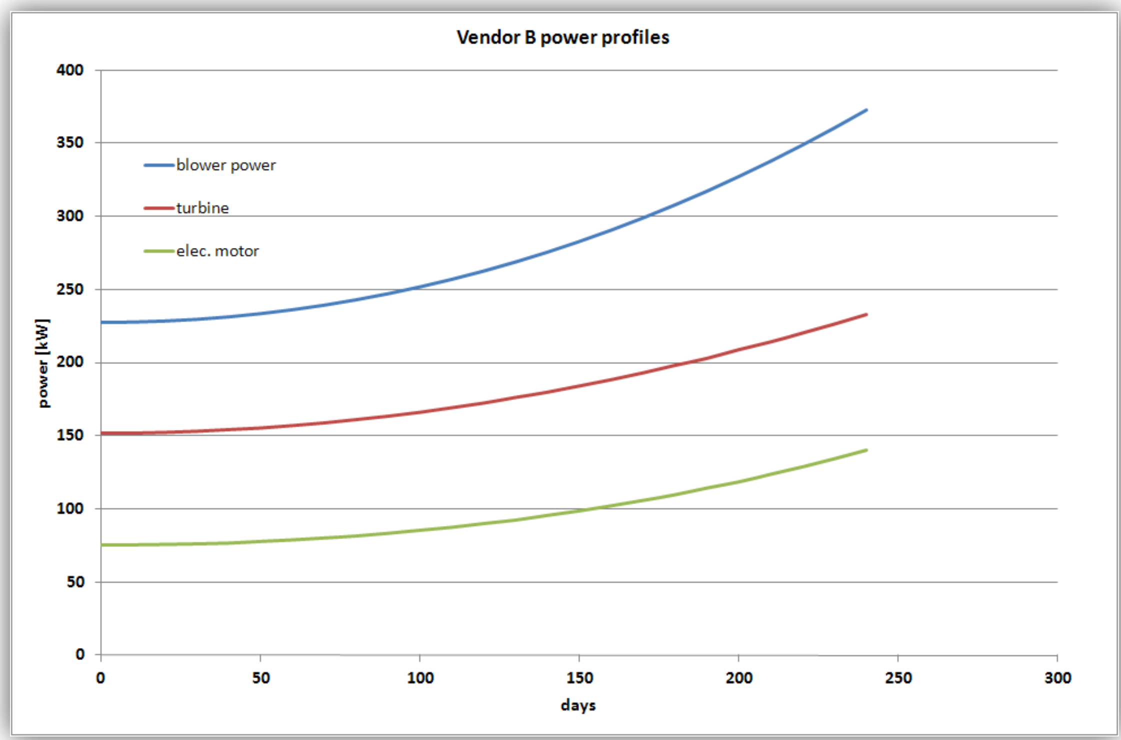

Figure 6.10 Turbo machinery power profile over a typical load

Table 6.14 Summary about Single shaft configuration

	Single shaft configuration
Steam available for sale in the reference plant (For a steam price of 20€/ton of steam) Appendix 9.4	675’000 €/year
Steam available for sale in the Single shaft configuration (For a steam price of 20€/ton of steam) Appendix 9.4	630’000 €/year
Electricity generation	1’460’000 kWh/year
Cash flows	146’000 €/year

6.2.6 Profitability analysis

Investment cost is obtained by the direct contact with vendors. The investment cost from the vendor is 334’000 €/unit. This investment cost includes the super heater, piping, valves, engineering cost, controls, and contingency. The annual savings is calculated from the available steam turbine power. The price of electricity is assumed to be 0.1 €/kWh. The annual savings in case of single shaft case is 146’000 €.

If we do not consider the economic value of heat sales from the single shaft case than profitability analysis is shown in *Figure 6.11*.

Table 6.15 Investment and Revenues for single shaft configuration

Price of electricity	0.1 €/kWh
Electricity savings	1’460’000 kWh/year
Investment	334’000 €
Revenues	146’000 €/year

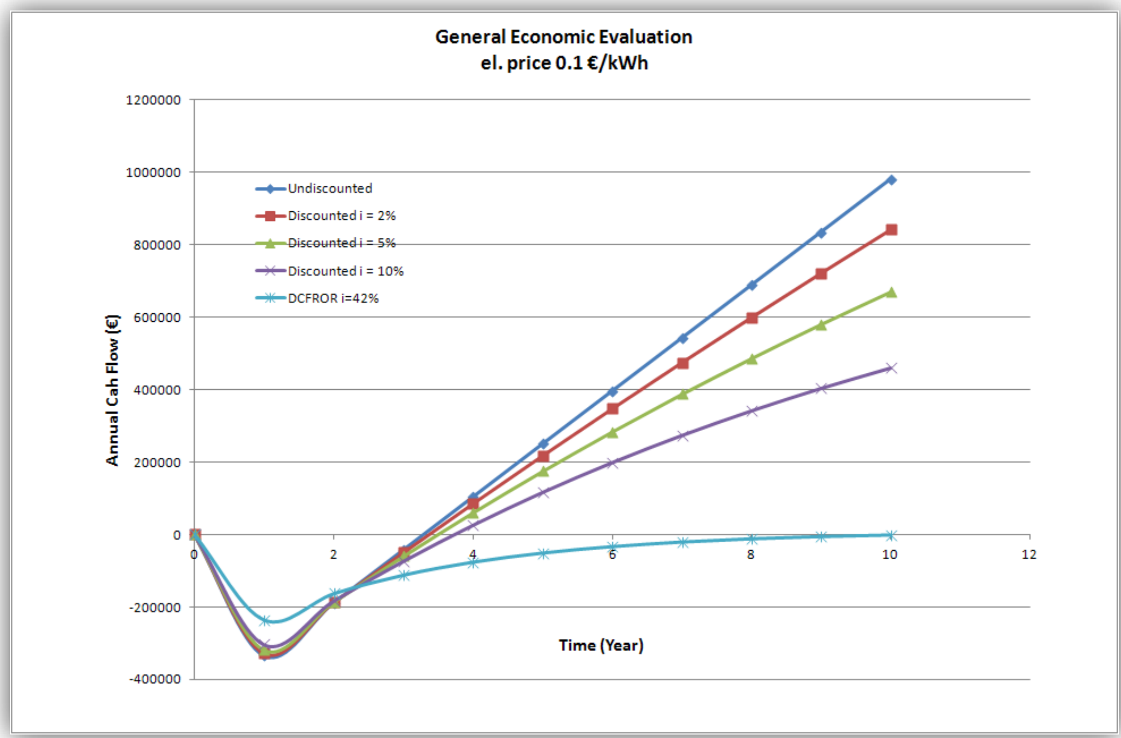

Figure 6.11 Profitability analysis without heat sales

If we consider the economic value of heat sales than 45700 € money is lost every year. So the revenues from the single shaft configuration is 100’000 €/year, which is given in *Figure 6.12*.

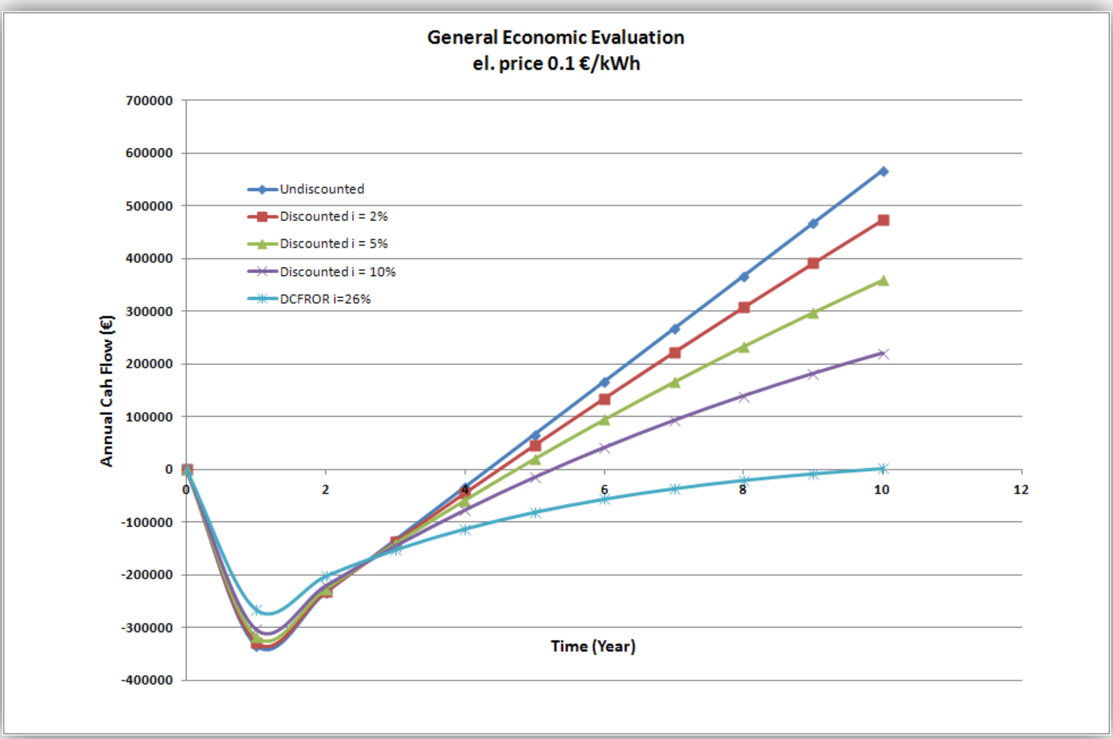

Figure 6.12 Profitability analysis with considering heat sales

7 SUMMARY OF RESULTS AND CONCLUSION

7.1 Electricity savings and profitability

Table 7.1 Yearly shaft work available from the turbine

Double shaft configuration (kWh)	Single shaft configuration (kWh)	Pinch analysis methodology (kWh)
940’000	1’460’000	2’620’000

Table 7.2 Profitability analysis without heat sales

	Double shaft configuration	Single shaft configuration
NPV (i=5%)	420’000 €	640’000 €
PBT (i=5%)	3.0 year	3.4 year
NPV (i=10%)	300’000 €	420’000 €
PBT (i=10%)	3.5 year	3.9 year

Table 7.3 Profitability analysis with heat sales

	Double shaft configuration	Single shaft configuration
NPV (i=5%)	240’000 €	350’000 €
PBT (i=5%)	4.2 year	4.6 year
NPV (i=10%)	160’000 €	220’000 €
PBT (i=10%)	5.0 year	5.2 year

Table 7.4 Summary about all configurations

	Double Shaft Configuration	Single shaft configuration
Lifetime of plant	10 years	10 years
Investment cost	200’000 €	334’000 €
Revenues	94’000 €/year	146’000 €/year
Steam available for sale	645’000 €/year	630’000 €/year

7.2 Conclusion

In this thesis different configurations of turbo machinery have been investigated for power generation from the process excess heat. The energy analysis for this thesis was carried out using Pinch Technology in order to gather all the necessary stream data and investigate the potential for energy recovery.

The practical aspect of energy recovery through steam-driven recirculation blowers were discussed in detail.

Turbo machinery similarity theory and, in particular, Balje diagrams were used to performed preliminary selection of the required machines and the quotation from vendors were used to make estimation of the steam-driven system over a typical year of plant operation.

An economic analysis for the different configurations was carried out.

In the current plant with a nominal capacity of 150 ton/day, the plant produces 3390 kg/hr saturated steam at 25 bar from the excess process heat. This steam, in the current standard design, is sold to nearby industrial process.

It is in the interest of the company to propose a design in which this steam is internally used.

A preliminary calculation shows that 247 kW power output from a steam turbine is possible in nominal plant condition.

From the Pinch Analysis consideration, a higher amount of steam appears potentially available (4300 kg/hr) including super heater which can possibly correspond to the 312 kW of power if the same steam pressure is used as in the standard plant, which is around 70 kW more than what is practically achievable if no major plant re-design is carried out.

The total blower power requirement in the case of single shaft is lower than double shaft. We need to have an electrical motor in order to meet the blower power requirement when the catalyst is getting old. Thus the size of an electrical motor is also smaller and the available power from the steam turbine is also higher in single shaft configuration than double shaft. This is due to the fact that turbo machinery in single shaft configuration operates at a higher speed thus allowing much higher efficiencies of their machines.

The economic evaluations of the thesis were carried out with different interest rates and an economic lifetime of 10 years. The investment cost for the different configuration of turbo machinery was taken from the vendors.

The pay-back period investigations indicate a minimum payback period of around 3 years (for single and double shaft configuration) for a electricity price of 0.1 €/kWh, but the net present value is higher in single shaft configuration than double shaft at the end of 10 year. Even though the investment cost is higher in single shaft configuration than double shaft configuration. The revenues in terms of electricity savings is also more in single shaft configuration therefore single shaft solution is preferred.

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9 APPENDIX

9.1 Estimation of Blower Power

1st stage calculations

Density of gas before recirculation blowers, ρo =kg/m^3

Inlet volumetric flow rate to the recirculation blowers, Vo = 4,08 m^3/sec

Inlet mass flow rate to the recirculation blowers, m = 5,8 kg/sec

Inlet pressure of gas before recirculation blowers, P_0 = 1,3 bara

Inlet temperature of gas before recirculation blowers, T_0 = 38°C

Heat capacity ratio for air, κ = 1,4

Specific heat of air, Cp = 1,009 kJ/kg

For fresh catalyst the outlet pressure from the 2nd stage, $P2$ (set) = 1,78 bara

Pressure increases across 2nd stage is

$$\Delta P2 = \frac{P2\ - Po}{1,8} + 0,005 \tag{9.1}$$

$\Delta P2$= 0,272 bara

Pressure increases across 1st stage

$$\Delta P1 = (P2 - Po) - \Delta P2 + 0,005 \tag{9.2}$$

$\Delta P1$= 0,213 bara

Outlet pressure from the 1st stage, P1

$$P1 = Po + \Delta P1 \tag{9.3}$$

$P1$ = 1,513 bara

For the adiabatic process, see *Figure 3.2*

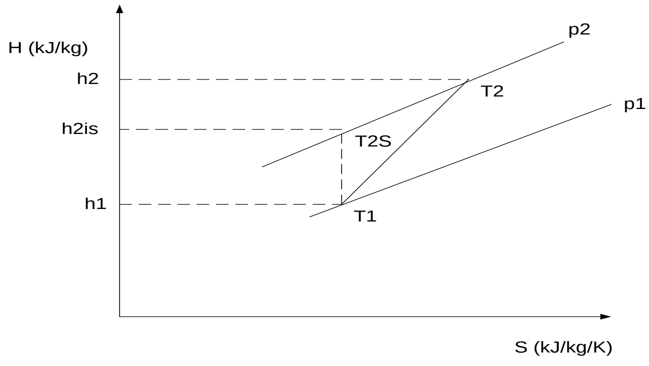

Figure 9.1 Adiabatic process

$$\frac{T1s}{To} = \left(\frac{P1}{Po}\right)^{\kappa-1/\kappa} \tag{9.4}$$

$$T1s = To\left(\frac{P1}{Po}\right)^{\kappa-1/\kappa} \tag{9.5}$$

$T1s$ = 51,8 °C

$$h1s = Cp * T1s \tag{9.6}$$

$h1s$ = 327,7 kJ/kg

$$ho = Cp * To \tag{9.7}$$

h_0 = 313,8 kJ/kg

$$\Delta H = h1s - ho \tag{9.8}$$

ΔH = 13,92 kJ/kg

The efficiency of the fan is assumed to be constant 79 % (including all bearings and couplings losses) and the shaft power of fan from the first stage comes out to be (Fans, 1997).

$$P = \frac{Vo * \Delta H * \rho o}{\eta} \tag{9.9}$$

P = 103 kW

2nd stage calculations

For the 2nd stage, the Inlet pressure, = Outlet pressure from 1st stage, $P1$ = 1,51 bara

$$\eta = \frac{Cp * (T1s - To)}{Cp * (T1 - To)} \tag{9.10}$$

The outlet temperature from the 1st stage will be

$$T1 = \frac{(T1s - To)}{\eta} + To \tag{9.11}$$

$T1$ = 55,5 °C after the 1st stage

The inlet density to the 2nd stage,

$$\rho 1 = \frac{P1 * To * \rho o}{T1 * Po} \tag{9.12}$$

ρ_1 = 1,576 kg/m^3

Volumetric flow rate to 2nd stage, $V_1 = m/\rho_1$

V_1 = 3,70 m^3/sec

The outlet pressure from the 2nd stage is

$$P2 = P1 + \Delta P2 - 0{,}005 \tag{9.13}$$

$P2$ = 1,78 bara

$$T2s = T1\left(\frac{P2}{P1}\right)^{\kappa-1/\kappa} \tag{9.14}$$

$T2s$ = 71,1 °C

$$h2s = Cp * T2s \tag{9.15}$$

h2s = 347,2 kJ/kg

$$h1 = Cp * T1 \tag{9.16}$$

h1 = 331,4 kJ/kg

$$\Delta H = h2s - h1 \tag{9.17}$$

ΔH = 15,73 kJ/kg

$$P = \frac{V1 * \Delta H * \rho 1}{\eta} \tag{9.18}$$

If the efficiency of fan is assumed to be constant at 79 % (including all bearings and couplings losses), As the efficiency does not vary substantially as the volumetric flow rate remains constant and the increase in pressure is achieved by increasing the rotational speed), the shaft power of the fan from the second stage comes out to be

P = 116 kW

The total shaft power of two recirculation blowers is the sum of 1^{st} and 2^{nd} stage powers.

$$Ptotal = Pstage1 + Pstage2 \tag{9.19}$$

Ptotal = 219 kW

By performing the same calculations procedure described above with an old catalyst for a pressure range (0.3-1.12 barg) and by assuming the 79% efficiency of fans, the power requirement for 2 stages of recirculation blowers comes out to be 351 kW.

$$Ptotal = Pstage1 + Pstage2 \tag{9.20}$$

$$Ptotal = 170 + 181 \tag{9.21}$$

Ptotal = 351 kW

9.2 Estimated power from the steam turbine

Mass flow of superheated steam to the steam turbine, *m*= 3390 kg/hr
Inlet pressure of the superheated steam to the steam turbine, *Pin*= 25 bara
Inlet temperature of the superheated steam to the steam turbine, *Tin*= 399℃
Inlet Enthalpy of superheated steam, *hin*= 3238 KJ/Kg From (Wester, Oktober 1996)at *Pin* and *Tin*
Inlet Entropy of the superheated steam, *Sin*= 7 kJ/kg/K (at *Pin* and *hin*)
Outlet pressure from the steam turbine, *Pout*= 5 bara
Outlet isentropic enthalpy, *hisout*= 2834 kJ/kg at *Pout* and *Sin*
Isentropic efficiency of steam turbine, η = 65% calculated from (Harvey, 2009), FIG. 11-4, Turbine efficiency design.

$$\eta = \frac{hin - hout}{hin - hisout} \tag{9.22}$$

$$hout = hin - \eta(hin - hisout) \tag{9.23}$$

The outlet enthalpy comes out to be 2975 kJ/kg

$$P = m * (hin - hout) \tag{9.24}$$

The estimated power from the steam turbine is 247 kW.

9.3 Blower and Steam Turbine Specifications

Table 9.1 For vendor's blower and steam turbine specifications

Blower Specifications			
MECHANICAL DESIGN			
Pressure	[bar g]	1.6	
Temperature	[ºC]	120	
FLUID / MEDIUM			
Medium		Recycle gas with traces of corrosive gases (composition see below) saturated with water	
		Formaldehyde:	200 ppm
		Carbon monoxide:	1.5%
		Dimethyl ether:	0.6%
		Methanol:	0.1%
Steam Turbine Specifications			
MECHANICAL DESIGN			
Pressure	[bar g]	26	
Temperature	[ºC]	430	
FLUID / MEDIUM		Superheated steam	
Operating Conditions (Available steam from process)			
Case 1:Normal gas flow (+10%), max dp [kg/hr]		Max 3390	
Case 2: Electrical guarantee gas flow [kg/hr]		Max 3390	
Case 3: Min gas flow [kg/hr]		Max 1090	
Case 4: Start-up flow [kg/hr]		To be discussed	

9.4 Consequence on available heat at the turbine outlet (Double shaft configuration)

For the case where a back-pressure turbine is installed, steam has still quite high temperature (260° C) and can be used for heating purposes the outlet of steam turbine (5 bar).

In a current standard plant the pressurized steam after the HTF condenser and ECS steam generator is being sold as a heat to nearby processes. To calculate the amount of heating values in a standard plant,

Total mass flow of steam is m= 4008 kg/hr

Condensate coming back at P= 3 bar, T= 100°C, h= 420 kJ/kg

Saturated steam going out to nearby process at P= 25 bar, T= 224°C, h= 2821 kJ/kg

$$\mathrm{Q} = \mathrm{m} * \Delta\mathrm{H} \tag{9.25}$$

The heating value is 2673 kW

As the plant operated 350 days/year, the total heating value is 2’250’000’0 kWh.

As ΔH= 2821-420 (kJ/kg)

Then ΔH will be 0.666 kWh/kg (1kWh= 3600 kJ)

1ton of steam= 666 kWh (1 ton =1000 kg)

$$\frac{tons\ of\ steam}{year} = \frac{total\ steam\ kWh}{666\ kWh} \tag{9.26}$$

There will be 33000 tons of steam/year. A typical price for steam is 20€/ton of steam, then the total economic value of the steam in the current reference plant is 675’700 €/year.

For the double shaft configuration the condensate coming back at P= 3 bar, T= 100°C, h= 420 kJ/kg

The saturated steam is delivered to a nearby process at P= 5 bar, h= 2747 kJ/kg

As ΔH= 2747-420 (kJ/kg)

Than ΔH will be 0.646 kWh/kg (1kWh= 3600 kJ)

1ton of steam= 646 kWh (1 ton =1000 kg)

$$\frac{\text{tons of steam}}{\text{year}} = \frac{\text{total steam kWh}}{646\ \text{kWh}} \tag{9.27}$$

So there is 32000 tons of steam. For a steam price of 20€/ton of steam, then the total economic value of the steam available for sale from the double shaft system is 645’500 €/year. When we compare the double shaft case with the standard case the difference will be the loss of money that is 30000 €/year.

9.5 Consequences on available heat at the turbine outlet (Single shaft configuration)

The steam at 5 bar outlet from the steam turbine can be sold to nearby processes.

For the BNI, the condensate is at P= 3 bar, T= 100°C, h= 420 kJ/kg

The saturated steam going out to nearby process at P= 5 bar, h= 2747 kJ/kg

As ΔH= 2747-420 (kJ/kg)

Than ΔH will be 0.646 kWh/kg (1kWh= 3600 kJ)

1ton of steam= 646 kWh (1 ton =1000 kg)

$$\frac{tons\ of\ steam}{year} = \frac{total\ steam\ kWh}{646\ kWh} \tag{9.28}$$

There is 31000 tons of steam/year. For a typical steam price 20€/ton of steam, than the total economic value of the steam available for sale from the single shaft system is 630’000 €/year. When we compare the single shaft case with the standard case the difference will be the loss of money that is 45700 €/year.

9.6 Economic Factors

Table 9.2 Undiscounted, Discounted Cash Flow for double shaft configuration

			i	0,02
year	**Undiscounted**		**Discounted**	
	cash flow [€]	cumulative cash flow [€]	cash flow	cumulative cash flow [€]
0	0	0	0	0
1	-200000	-200000	-196078	-196078
2	94000	-106000	90350	-105729
3	94000	-12000	88578	-17150
4	94000	82000	86841	69691
5	94000	176000	85139	154830
6	94000	270000	83469	238299
7	94000	364000	81833	320132
8	94000	458000	80228	400360
9	94000	552000	78655	479015
10	94000	646000	77113	556128

i	**0,05**	**i**	**0,1**	**i**	**0,45**
Discounted		**Discounted**		**Discounted - to find DCFROR**	
cash flow	cumulative cash flow [€]	cash flow	cumulative cash flow [€]	cash flow	cumulative cash flow[€]
0	0	0	0	0	0
-190476	-190476	-181818	-181818	-137931	-137931
85261	-105215	77686	-104132	44709	-93222
81201	-24015	70624	-33509	30834	-62389
77334	53319	64203	30695	21265	-41124
73651	126971	58367	89061	14665	-26459
70144	197115	53061	142122	10114	-16345
66804	263919	48237	190359	6975	-9370
63623	327542	43852	234210	4810	-4560
60593	388135	39865	274076	3318	-1242
57708	445843	36241	310317	2288	1046

Table 9.3 Undiscounted, Discounted Cash Flow for single shaft configuration

year	**Undiscounted**		**i**	**0,02**
			Discounted	
	cash flow [€]	cumulative cash flow [€]	cash flow	cumulative cash flow [€]
0	0	0	0	0
1	-334000	-334000	-327451	-327451
2	146000	-188000	140331	-187120
3	146000	-42000	137579	-49541
4	146000	104000	134881	85340
5	146000	250000	132237	217577
6	146000	396000	129644	347221
7	146000	542000	127102	474322
8	146000	688000	124610	598932
9	146000	834000	122166	721098
10	146000	980000	119771	840869

i	**0,05**	**i**	**0,1**	**i**	**0,43**
Discounted		**Discounted**		**Discounted - to find DCFROR**	
cash flow	cumulative cash flow [€]	cash flow	cumulative cash flow [€]	cash flow	cumulative cash flow[€]
0	0	0	0	0	0
-318095	-318095	-303636	-303636	-233566	-233566
132426	-185669	120661	-182975	71397	-162169
126120	-59549	109692	-73283	49928	-112241
120115	60566	99720	26437	34915	-77327
114395	174961	90655	117091	24416	-52911
108947	283908	82413	199504	17074	-35837
103759	387668	74921	274426	11940	-23897
98819	486486	68110	342536	8350	-15547
94113	580599	61918	404454	5839	-9708
89631	670230	56289	460743	4083	-5625

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