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Optimal Integration of Humid Air Cycle in Energy Intensive Industries.

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ABSTRACT

Humid Air Turbine cycle (HAT) is characterised by its high single cycle efficiency. The HAT cycle is typically constrained by a pinch point at low temperature. This indicates that additional heat in the range 100 °C to 200 °C can be utilised with high marginal efficiency. At the same time energy intensive industries (for example refineries, Cement production plants and Steel works) typically have a surplus of heat from around 250 °C to 300 °C and down. This study is aimed at the integration of HAT Cycle into the industrial process plant where the complementary features can be exploited. The present paper has two main objectives. The first objective is to present a general approach for integration analysis. The approach is based on conceptual design using targeting procedures (e.g. Pinch Analysis). The second objective is to find an optimum integration scheme for specific heat sources available from industrial sites. To illustrate both objectives a case study based on real refinery data is discussed.

Keywords

Humid Air Turbine, Process Integration, Industrial co-generation.

Introduction

In the energy intensive industries (Refineries, Cement production, steel works etc.) a vast amount of heat is available from about 250 °C. The heat is often discharged because no economical utilisation is feasible. The amount of heat and the available temperature levels can easily be identified in a Process Integrating study (e.g. Pinch Analysis). The site pinch point is often placed well above 200 °C for the above mentioned industries. Steam raising is commonly used for transforming waste heat to a useable heat medium for power production or for direct mechanical drives. The drawback is, however, that steam raising results in a major exergy loss because of the phase change, that takes place at constant temperature (often at minimum 200 °C). It is possible to introduce multiple steam levels, which reduces the exergy loss, but increase the overall capital cost and plant complexity.

The system proposed in this paper is based on a modified Humid Air Turbine (HAT) cycle. The approach used throughout this study is based on the approach suggested by Nielsen (1995), which in the case of the HAT cycle optimisation is similar to the approach

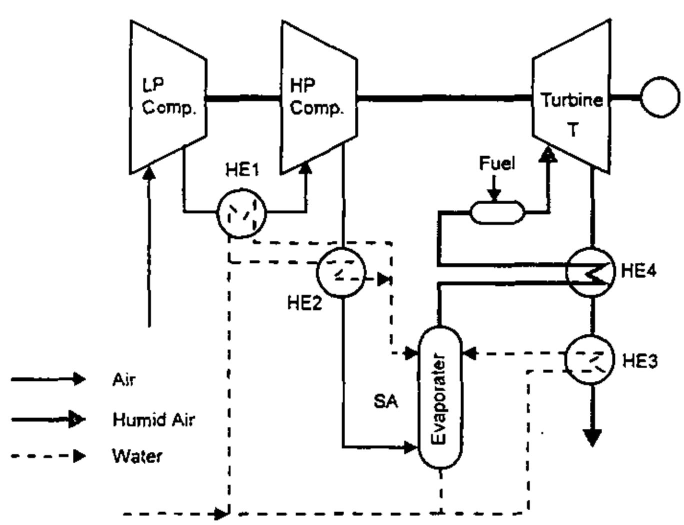

Figure.1 HAT cycle in typical form (After Stecco et al. 1993).

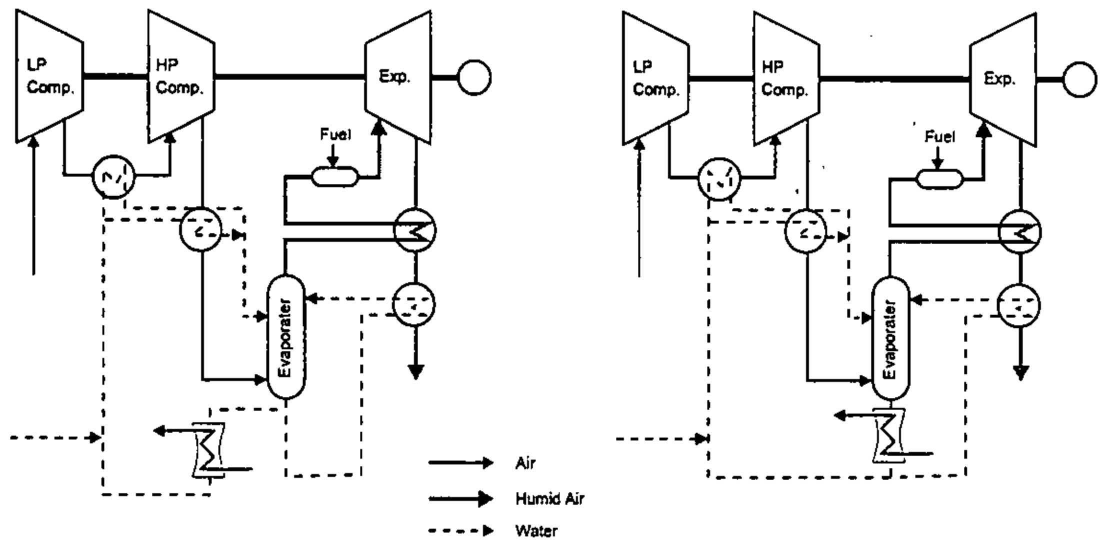

Figure 2. Modified HAT cycle structures as proposed by Stecco et al. (1993).

suggested by De Ruyck et al. (1995). The key features of the approach is that it is based on a conceptual description using targeting procedures for heat recovery and water evaporation. This is in contrary to the conventional approach, which is a parametric study based on fixed plant structure. The principal advantage of the conceptual description and targeting approach is that it does not become caught in a topology trap. It also allows for fast screening of different alternative conceptual set-up, which is discussed in the subsequent chapters.

HAT Cycle concept

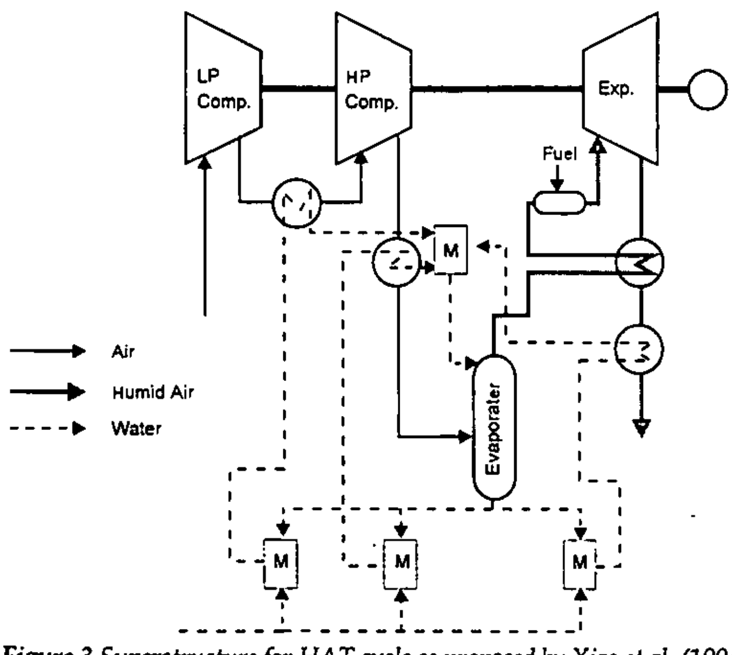

Figure 3 Superstructure for HAT cycle as proposed by Xiao et al. (1994). M denotes mixer units.

The Humid Air Turbine (HAT) cycle was patented by Flour Daniel (Rao 1989). The cycle uses a working medium of humidified air. The cycle, which in its general form is quite simple compared with other innovative power cycles (e.g. Kalina et al. 1991), is based on recent advances in gas turbine technology. With no top or bottoming cycles the HAT cycle is reported to give efficiencies of over 60% (LHV). The cycle has been given a considerable interest over the last couple of years. Several studies have aimed at optimising the cycle performance using different process layouts (Xiao et al. 1994, Stecco et al. 1993a,1993b). These studies have been based on a fixed or semifixed conceptual layout followed by a parametric optimisation. The approach in this work is based on a general top down analysis (Nielsen 1995). This analysis approach can be applied to the studied cycle with a considerable gain in the optimisation work. With simple rules based on second law insight and pinch technology an optimisation study can be carried out using a standard process simulator within a few days.

The typical layout for the HAT cycle is shown in Figure 1. The air is compressed in two stages with an intercooler (HE1). The intercooler gives a lower temperature into the second stage compressor and thus a smaller power consumption. The removed heat is used to heat up some water for the evaporator. After the second compression stage the gas is cooled once more (HE2) before it enters the evaporator (SA). Here the gas meets the hot water in countercurrent flow. This unit gives a simultaneous mass and heat transfer where the water is evaporated at variable temperature. The humidified air is then heated in a regenerator (HE4), where heat from the turbine exit is used to heat the humidified air. In the combuster fuel is injected and burned. The hot gas now enters the gas turbine (T). The tail gas is cooled in two steps. First in the regenerator and after that in a water heater (HE3).

Stecco et al. (1993a,b) propose two variations where a cooling tower is applied to decrease the temperature inlet to second compression stage (Figure 2). Xiao et al. (1994) propose an altemative layout based on a superstructure that embed several different heat exchanger configurations (Figure 3). Stecco et al. (1993a,b) get their results by varying the key parameters for each configuration. In this case they varied the compression ratios for both compressors and the relative humidity at the evaporator outlet. Xiao et al. (1994) optimised the system with an algorithmic approach applying an infeasible path method with a process simulator.

Nielsen (1995) proposed a simple global approach which has been used in several industrial process integration studies. The global approach is as follows:

1. Obtain base case configuration.
2. Remove all utility streams (non kernel process streams).
3. Select basic utility concept(s).
4. Set initial/Update kernel process parameters and key utility parameters.
5. Serve all process streams with simple utilities.
6. Calculate pinch targets (minimum approach temperature).
7. Repeat step 4-6 while optimising energy recovery, material conversion or power production.
8. Design heat exchanger network for optimised process and utility concept.

The main feature is the two stage design of the heat exchanger network. Individual heat exchangers are removed and replaced with utility exchangers. After that a pinch analysis calculates energy targets (or temperature approach targets for exothermic processes). At that stage the actual heat exchanger network design is not fixed, but the result shows the feasibility of the proposed process design. Now key process parameters can be varied in the search for optimum performance. For each conceptual process design a pinch analysis can show whether a feasible heat exchanger network exists (whether minimum approach temperature is over or below 0). The major benefit by this general approach is the limited work on heat exchanger network synthesis. In this case the actual network design is carried out as the last step when most other parameters are fixed. One major advantage is that the same conceptual model can be used throughout the analysis.

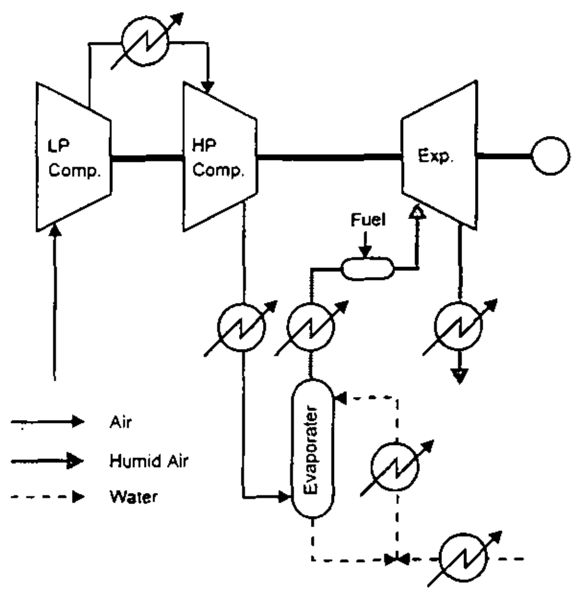

Figure 4 Conceptual model for HAT cycle evaluation.

For the HAT cycle a modified approach is suggested based on the particularity of this cycle. If the objective is to maximise the thermal efficiency, some basic exergy consideration can be utilised. The following operations in the HAT power cycle cause exergy losses:

1. Mixing fluids with different temperatures.
2. Heat transfer with temperature driving forces.
3. Irreversibility in the humidifier.
4. Combustion irreversibility.

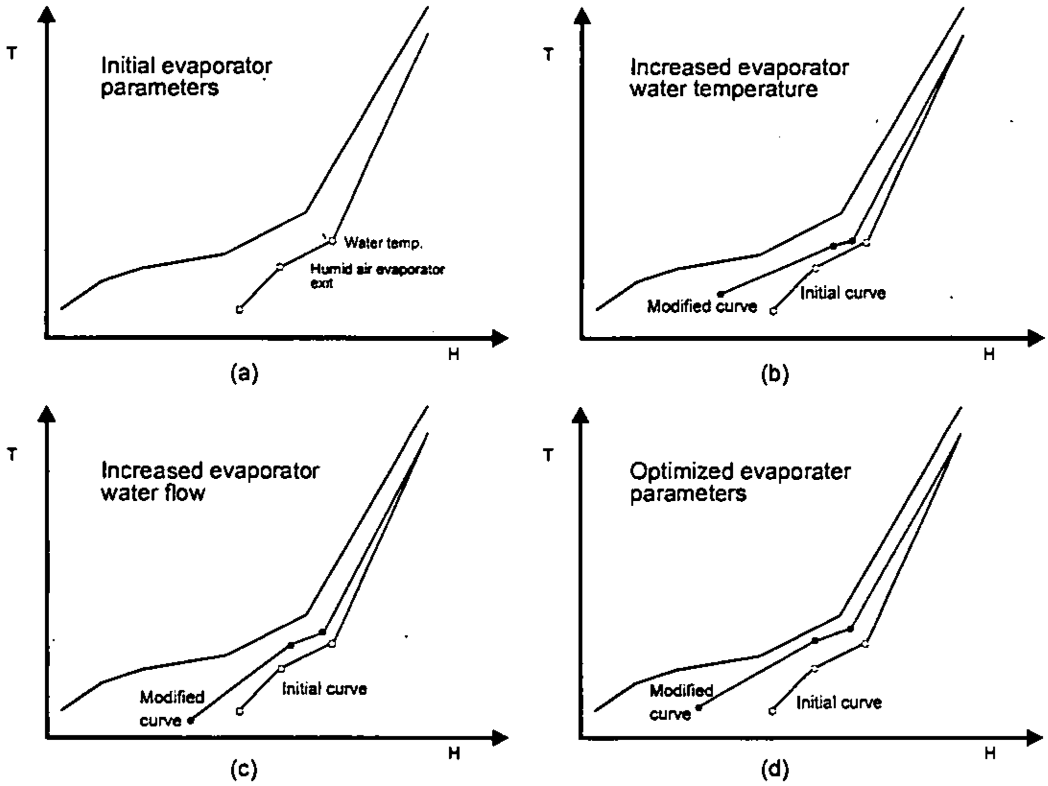

Figure 5 Effect of changing the flow rate and temperature of water to the evaporator.

In this example the irreversibility in the combustion, which can be improved by recycling off-gases (Harvey et al. 1992), will not be considered. Without any quantification it is postulated that optimum performance is obtained when all fluid mixing is done isothermally, heat transfer is carried out with a constant minimum approach temperature and irreversibility in the humidifier is minimised. The irreversibility in the humidifier is caused by a combination of fluid mixing and heat transfer with excessive temperature difference.

A conceptual flowsheet for the HAT cycle is shown in figure 4. In this flowsheet no restriction is imposed on placement of individual heat exchangers. It is assumed that all process streams are served by the best available heat source. In the flowsheet, this assumption is represented by an external utility. With a given set of process parameters (pressure ratios, humidifier parameters) a targeting analysis can predict the feasibility of the process. With pinch technology the composite curves can be used to identify thermal inefficiencies (excessive temperature differences). The information can then be used to update the process parameters.

To obtain maximum efficiency of the conceptual model outlined in figure 4, the following parameters are optimised

Pressure ratios stage 1 and 2
Evaporator condition (water flow and temperature)
Evolving air flow (kg air/kg fuel)

Simple second law thermodynamic implies that the recuperated temperature should be as close as possible to the turbine exit temperature (i.e. equal to minimum approach temperature). Further we want to keep the composite curves as close as possible over the whole temperature range. Based on this simple insight we are ready to tailoring the approach for HAT cycle optimisation. The optimisation is based on a fixed fuel flow.

1. Set initial/update pressure ratios.
2. Set initial/update evaporator conditions.
3. Set initial/update Air flow.
4. Set initial/update recuperator temperature.
5. Calculate combustion temperature.
6. Repeat 4 and 5 until combustion temperature constraint is reached.
7. Repeat 3-6 until $T_{rec} = T_{exit} - \Delta T_{min}$.
8. Perform ΔT_{min} targeting.
9. Use composite curves to close the gaps between the hot and cold composite curves. This is done by updating evaporator conditions. Repeat step 2 - 9 until the curves are as parallel as possible.
10. Repeat step 1 - 9 over a reasonable

pressure range.

11. Design heat exchanger network for selected design parameters.

Although the procedure is quite iterative part of it can be done automatically using a standard process simulator. This is the case for step 3 to 6. Updating of the evaporator condition is a mixture of flow and temperature changes. The choice is dependent of the form of the composite curves. The effect of changing the temperature and massflow rate respectively is illustrated on figure 5.

For each set of pressure ratios the only two variables that have to be considered are the water flow rate and temperature. The design problem has thus been reduced significantly.

The problem dimension can be reduced even further by using a targeting model for the saturator (Nielsen 1995). Instead of simulating the heat and mass transfer in the saturator an alternative model can be used. Water is mixed directly with the combustion air giving a two phase flow. When this mixture is heated, the water gradually evaporates according to its partial pressure. The lay-out of this conceptual model is outlined in figure 6. The added water represents the amount of water that has to be transferred in the humidifier. De Ruyck *et al.* (1995) proposed a similar approach, where they use a black box representation of the combined heat transfer and evaporation system.

The proposed modified conceptual model is outlined in figure 6. The heat curve for the resulting gas/water heater is shown in figure 7. The benefit in using this alternative model is a simplification in the synthesis. In the original formulation it is necessary to include a model of the absorption process. With the new formulation the absorption process and heating of working gas and water is combined in a simple model where water and working gas are mixed and subsequently heated. With the new formulation only water flow rate (absorbed) and turbine inlet temperature has to be specified. It is quite simple to retrieve the full model from the targeting model. This can be done graphically - figure 7. The slope of the fresh water heating is determined by the amount of fresh water. The best heating of the rest can be determined by looking at the saturation curve along with the process heat release curve. If the pinch is located at the saturation curve the line has to be drawn through the pinch point in order to maintain the minimum approach temperature.

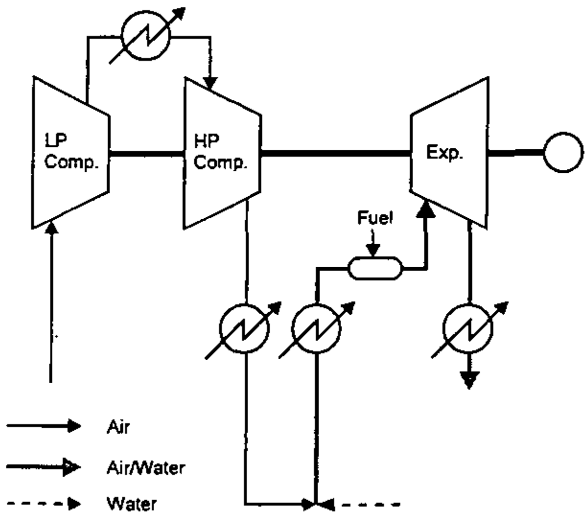

Figure 6. *A conceptual model of the HAT Cycle for conceptual optimisation.*

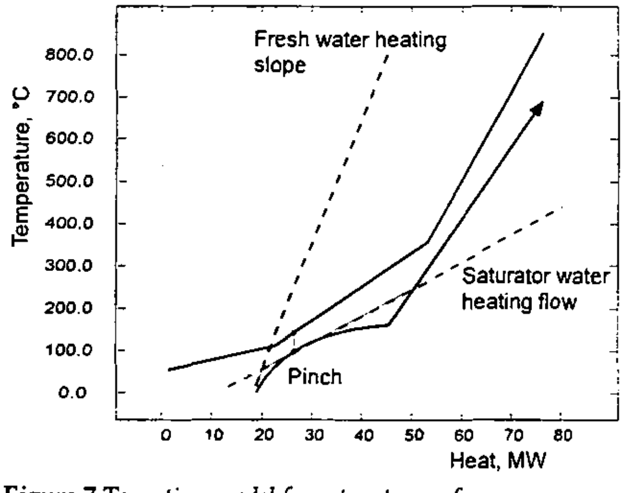

Figure 7 *Targeting model for saturator performance.*

With the conceptual model in figure 7 a simple study can be conducted. The objective of the study is to illustrate the approach. It is thus assumed that the pressure ratios and pressure losses are fixed. Further the combustion heat is added as a simple heat load.

Data for the subsequent case studies are given in table 1. In this paper the main issue is to illustrate the general approach. Therefore the condition in table 1 is assumed fixed, although subsequent optimisation and practical considerations may change the specific values.

Compressor isentropic efficiency	88 %
Turbine isentropic efficiency	90 %
Air and water temperature	20 °C
Intercooler temperature	25 °C
Aftercooler temperature	30 °C
Compressor I discharge pressure	3 bar
Compressor II discharge pressure	9 bar
Turbine inlet temperature	1200 °C
Pressure loss Intercooler	0.05 bar
Pressure loss combustion section	0.5 bar
Pressure loss tail gas section	0.04 bar
Minimum approach temperature	20 °C

Table 1 Basic assumptions for initial case study.

The exergy loss in the heat exchanging process is minimised when the process composite curves features two pinch points. With a process simulator, with an integrated pinch targeting tool (e.g. HYSIM, which is used in this study), it is straight forward to find the exact amount of water required to create a dual pinch point. With the assumption for this study the dual pinch situation occurs when 0.08 kg

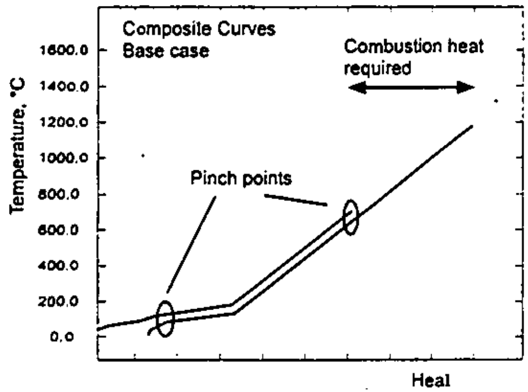

***Figure 8** Optimisation of the stand alone HAT cycle. Optimum is found when two pinch points co-exist.*

water is absorbed per kg evolved air. This results in a specific power output of 463.4 kJ/kg air and requires a fuel heat load of 754.3 kJ/kg air. This gives a thermal efficiency of 61.43 %. The resulting composite curve is outlined in figure 8.

Industrial integration of the HAT cycle

Many industrial sites have a vast amount of heat available from about 250 °C. The heat is often discharged because no economical utilisation is feasible. This can easily be identified by a process integrating study (e.g. Pinch Analysis) where a pinch temperature above 200 °C typically is identified. Steam raising is a commonly used mean for transforming waste heat to work or power. The drawback with steam is, however, that it results in a major exergy loss because of the phase change, that takes place at constant temperature (often at minimum 200 °C). It is possible to introduce multiple steam levels, which reduces the exergy losses. This does, however, increase the overall capital cost and plant complexity.

From the industrial point of view the modified HAT cycle offers a simple alternative to the complex and expensive steam systems (Steam drums, steam traps, dearaters etc.). In stead of raising steam the excessive process heat can be used for heating water for the HAT cycle and thus increase the power output from that cycle. It is thus possible to produce power utilising low temperature heat sources. There are thus two heat sources to the cycle - waste heat and fuel heat. Two different power efficiencies will be considered to describe the effective utilisation of the heat. The first describe the marginal power efficiency $\eta_{m,WH}$ of the waste heat Q_{WH} which can be expressed as

$$\eta_{m,WH} = \frac{P - \eta_{BC} \cdot Q_{fuel}}{Q_{WH}}$$

P is the total cycle power output. To get the marginal power from the waste heat it is necessary to subtract the power that could be produced by the fuel. In this paper the reference is the efficiency of the stand alone HAT cycle η_{BC}. Q_{fuel} is the required heat input from fuel firing. The marginal efficiency thus describes the actual efficiency for transforming waste heat to power. This efficiency can thus be compared with the efficiency of a steam cycle based on the waste heat.

The second efficiency describes the fuel efficiency based on the assumption that waste heat is alternatively discharged. This efficiency is expressed as

$$\eta_{m,fuel} = \frac{P}{Q_{fuel}}$$

For illustration a complete example is outlined. The main objective is to demonstrate the approach, while the detailed practical problems, such as available machinery, is a subject to future work. As a case study a small refinery is considered. The refinery has an

Waste heat kJ/kg air	Water flow rate kg water/kg air	Power kJ/kg air	Heat input kJ/kg air	Marginal waste heat efficiency %	Fuel efficiency %
0 (Base case)	0.08	463	754.3	-	61.4 (Base)
100	0.12	506	793	19.1	63.8
200	0.16	549	832	19.0	66.0
400	0.24	634	920	17.2	68.9
600	0.32	720	1018	15.7	70.7
800	0.37	773	1078	13.9	71.7
1000	0.43	837	1138	13.8	73.6
2000	0.63	1050	1394	9.7	75.3

Table 2 Conceptual optimisation of the integrated HAT cycle.

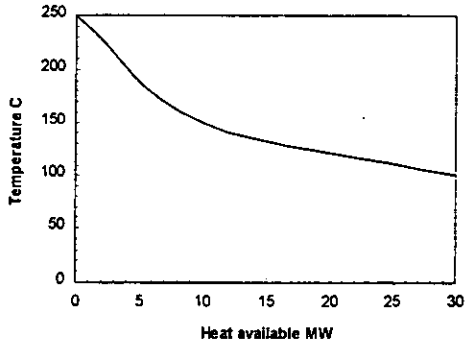

Figure 9 Heat available from small refinery site.

average crude intake of 8400 tpd. In a process integration study a total process composite curve is constructed (figure 9). The belonging grand composite curve show where excessive heat is available from the refinery site. For simplicity the site heat source is approximated in two sections. One section from 250 °C to 100 °C and one from 150 °C to 100 °C. This can be include in the integrated power cycle analysis by using two process streams. With the same approach and assumptions, as for the stand alone HAT cycle, an optimisation is carried out. For a given relative waste heat (kW per kg evolved air) the operation point that features two pinch points is identified. The results are outlined in table 2

With the example from the refinery a waste heat amount of 30 MW (250 °C to 100 °C) can yield a marginal power production of up to 5.7 MW (100 KJ/kg air waste heat) in an enhanced HAT cycle. This requires an additional fuel input of 240 MW. Alternatively an integrated cycle with 2000 kJ/kg air heat will result in a marginal power production of 2.9 MW with an additional fuel input of 20.9 MW. The advantage with the combination HAT cycle/process heat is that the extra investment (when the HAT cycle equipment is matured) will be rather small. With a high waste heat to air ratio a water content of up to 39 % is found.

Sub atmospheric HAT cycle

With high water content in the combustion gas it can be an advantage to run the expander to below the atmospheric pressure. In that case the expansion gas will be cooled down to well below the dew point. The condensate is separated and the remaining flue gas is re-compressed and discharged at atmospheric pressure. This process enhancement is examined with a waste heat to air ratio of 1000 kJ/kg air. Under the same conditions as before the optimum operation points are found for given subatmospheric pressures. The results are listed in table 3.

Turbo charger equipment

So far only state of the art turbo machinery is considered. The last case study examines the

Sub atmospheric pressure bar	Water flow rate kg water/kg air	Power kJ/kg air	Heat input kJ/kg air	Marginal waste heat efficiency %	Fuel efficiency %
0.74	0.43	931.7	1271	15.1	73.3
0.64	0.43	966.9	1324	15.4	73.0
0.54	0.43	1004.1	1385	15.4	72.5
0.44	0.43	1042.6	1454	9.2	71.7

Table 3 Conceptual optimisation of the heat integrated HAT subatmospheric cycle.

possibilities of producing a low cost HAT cycle based on turbo charger units. These units are typically produced in large numbers and thus available at moderate cost. Turbochargers do, however, not have state of the art efficiencies. In this new case study a simple HAT cycle is considered without intercooling.

Table 4	
Compressor efficiency	81.5 %
Turbine eff.	84 %
Air and water Temp.	20 °C
After cooler temp.	30 °C
Discharge pressure	3.5 bar
Turbine inlet temp.	750 °C
Pressure loss comb.	0.15 bar
Pressure loss exit	0.04 bar
Minimum approach temp.	20 °C

Table 4 Conditions for the turbo charger based HAT cycle.

The parameters used in the study are as listed in table 4

Waste heat load kJ/kg air	Water flow rate kg water/kg air	Power kJ/kg air	Heat input kJ/kg air	Marginal waste heat efficiency %	Fuel efficiency %
0	0.055	117.5	298.1		39.4
50	0.075	125.8	307.2	9.4	41.0
100	0.100	136.2	318.7	9.7	42.7
200	0.145	154.8	339.4	10.5	45.6
400	0.250	198.3	387.6	11.4	51.1
800	0.420	268.5	465.6	10.6	42.6

Table 5 Conceptual optimisation of the turbo charger based integrated HAT cycle.

With these values the refinery example is examined. The optimisation procedure is the same as for the prior case study. The results using turbocharger units are listed in table 5. The results given in table 5 show that it is possible to reach a reasonable efficiency with standard turbocharger units. Further the results show that it is possible enhance the power output using industrial waste heat. The turbo charger based cycle has a potential for the low end market, where the relatively low cost will compensate for the reduced efficiency.

Cycle sensitivities

The cases studies treated in this paper is based on the conditions stated in table 1 and 4. These constraints do not represent the real world constraints. As stated earlier the main objective is to illustrate the proposed approach. Several test has, however, been made to check wether the results are of general nature. The supplementary studies cover changed pressure ratios and increased temperature difference for the recuperator.

First the discharge pressures are increased to 3.5 bar and 12 bar for the atmospheric cycle. Secondly the temperature difference for the hot pinch is increased to 40 °C.

For the turbo charger based HAT cycle the hot pinch temperature difference approach is increased to 40 °C. Results for both supplementary studies are presented in table 6.

It is noticed that the increased hot pinch temperature approach decreases the efficiencies of the cycles as expected. The overall trend is, however, unchanged.

Conclusion and further work

The work presented in this paper deals primarily with the conceptual approach to industrial integration of the HAT cycle. Practical obstacles for implementation have not been considered. However, several important observations have been made. First of all it has been demonstrated that industrial sites can utilise a large of their amount of waste heat by integrating a HAT cycle into the process. Secondly it has been noticed that cycle syntheses are performed easily using a general conceptual targeting approach.

The major obstacle for application of HAT cycle power plant is the technological development. The study presented in this paper outline the possibility of using simple *of the shelf* turbo charger units for low cost power system where industrial waste heat can be utilised.

Case Atmospheric cycles	Waste heat load kJ/kg air	Marginal waste heat efficiency %	Fuel efficiency %
Increased pressure	0		61.3
Increased pressure	2000	9.1	74.2
Increased pressure + Increased hot pinch appr.	0		60.0
Increased pressure + Increased hot pinch appr.	2000	9.0	73.0
Turbocharger cycles			
Increased hot pinch appr.	0		37.1
Increased hot pinch appr.	400	10.3	47.0

Table 5 Supplementary analyses.

There are, however, a number of uncertainties that have to be examined further. The current goal of our research is to work further on the practical aspects and to examine the aspects of industrial integration further.

Normenclature

$\eta_{m,WH}$	Marginal efficiency for waste heat integration
$\eta_{m,fuel}$	Total fuel efficiency with waste heat utilisation.
η_{BC}	Reference efficiency stand alone utility system.
Q_{fuel}	Fuel firing for integrated cycle
Q_{WH}	Waste heat available

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