Site-wide low-grade heat recovery with a new cogeneration targeting method

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**Abstract**

One of the key performance indicators for designing site utility systems is cogeneration potential for the site. A new method has been developed to estimate cogeneration potential of site utility systems by a combination of bottom-up and top-down procedures, which allows systematic optimization of steam levels in the design of site utility configurations. A case study is used to illustrate the usefulness of the new cogeneration targeting method and benefits of optimizing steam levels for reducing the overall energy consumptions for the site. Techno-economic analysis has been carried out to improve heat recovery of low-grade waste heat in process industries, by addressing a wide range of low-grade heat recovery technologies, including heat pumping, organic Rankine cycles, energy recovery from exhaust gases, absorption refrigeration and boiler feed water heating. Simulation models have been built for the evaluation of site-wide impact associated with the introduction of each design option in industrial energy systems in the context of process integration. Integration of heat upgrading technologies within the total site has been demonstrated with a case study for the retrofit scenario.

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**Keywords:** Process integration; Cogeneration; Site utility systems; Low-grade heat; Heat recovery

1. Introduction

Low grade heat is often wasted without being recovered in the process industries, although potential economic benefits from waste heat recovery can be significant. The industrial application of low grade heat recovery is relevant to wide range of industries, including the chemical, petroleum, pulp and paper, food and drink, manufacturing, iron and steel, and cement industries (Pellegrino et al., 2004).

To avoid unnecessary capital expenditure for oversized equipment and to enhance controllability of the energy systems, characteristics of energy supply and demand, along with integration with energy recovery technology must be incorporated into the energy study in a systematic and holistic manner. The implementation of these integrated energy saving projects within or beyond the plant may not be favoured, due to practical constraints. For example, considerable civil and piping work might be required, or legislative limitations imposed, or there might be different energy utilisation patterns between sources and sinks, etc. Therefore, it is vital to quantify the economic benefits of employing low grade energy recovery and its impact on the industrial site.

2. New cogeneration targeting method

The extent of heat recovery and cogeneration potential is closely related to the configuration of site-wide energy distribution systems in an industrial site, in which multiple levels of steam pressure are introduced. For example, VHP (very high pressure), HP (high pressure), MP (medium pressure) and LP (low pressure) might be typically used. Steam pressure is an important design variable as this can be adjusted to either minimize the fuel requirement or maximise profits by exploiting system-wide trade-off between heat recovery and power generation. Optimization of levels of steam mains is based on the manipulation of a targeting model for calculating cogeneration potential of site utility systems.

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Given steam levels, inlet superheat of VHP steam, process load, BFW, Condensate temperature

Isentropic efficiency

Calculate superheat temperatures at subsequent lower steam level using isentropic efficiencies (Equation 2)

Starting from the lowest level, calculate the mass flow rates using Equation 1.

Add flow rates to determine the overall flow rates through each level (bottom up)

\[
\text{LP superheat temperature} > \text{LP saturation temperature} + \Delta T^* \\
\text{NO}
\]

\[
\text{YES} \\
\text{STOP}
\]

Fig. 1 – Algorithm for new method based on isentropic expansion.

The performance of the energy systems needs to be optimized, either to obtain the best system design for grassroots design, or to obtain the optimum operating conditions for an existing design. Considering the part load performance of the equipment is necessary, together with design options to be considered in the optimization of site utility systems, including the number and conditions of steam levels, network configuration of energy generation and distribution systems, energy demands of downstream processes, and different economic scenarios. The simulation and optimization of the utility systems require an accurate estimation of the cogeneration potential for the total site analysis as it aids the evaluation of performance and profitability of the energy systems. The overall cost-effectiveness of power (electricity) and heat from the site is heavily influenced by the optimum management and distribution of steam between various steam levels, and the strategy for electricity import and export. Also, acceptability for implementing any energy saving projects such as recovery of low grade heat, will be strongly influenced by operating and design conditions of existing energy systems. Therefore, the accurate evaluation of cogeneration potential with full consideration of system-wide implications is essential for performing a meaningful economic evaluation of the design options considered for heat upgrading and/or waste heat recovery.

A number of methods are currently available in the literature for estimating the cogeneration potential of utility systems. The ideal shaftpower can be calculated as the exergy change of the steam passing through the turbine (Dhole and Linhoff, 1993). In this work, the exergetic efficiency was considered to be independent of the load and inlet-outlet conditions, and assumed to be a constant value. The steam conditions were approximated by saturated conditions, but the superheat in the inlet and outlet steam conditions were neglected (Kundra, 2005). There is a difference of up to 30% in cogeneration potential when compared with simulations based on THM (turbine hardware model) method developed by Mavromatis and Kokossis (1998).

Salisbury (1942) observed that the specific enthalpy of steam (i.e. enthalpy per unit mass flow) is approximately constant for all exhaust pressure values. There is a linear correlation between specific power \( w \) (power per unit mass flowrate of steam) produced in the turbine and the outlet saturation temperatures. The specific power can be conceptually interpreted as the area of the rectangle on a graphical representation of the inlet and outlet saturation temperatures of the turbine with respect to the heat loads of steam (Raissi, 1994). This methodology, often known as TH cogeneration model, is based on the following assumptions: specific load \( \beta \) of steam is constant with variation in exhaust pressure, and specific power is linearly proportional to the difference of inlet and outlet saturation temperatures.

Mavromatis and Kokossis (1998) proposed a new shaft-power targeting tool called the turbine hardware model (THM) based on the principle of the Willans’ line. The Willans’ line approximates a linear relationship between steam flowrate and the power output. The THM has limitations as addressed by Varbanov, 2004, such that the effect of back pressure is not taken into account. Also modelling assumptions for part-load performance are too simplistic, as the model assumes a linear relationship over the entire range of operation.

Sorin and Hammache (2005) introduced a different targeting method based on thermodynamic insights for the Rankine cycle. The ideal shaftpower was a function of outlet heat loads,
and difference in Carnot factor between the heat source and heat sink. The deviation of the actual expansion from the ideal expansion was defined in terms of isentropic efficiency.

2.1. New method

Cogeneration targeting in utility systems is used to determine fuel consumption, shaftpower production and cooling requirements before the actual design of the utility systems (Sorin and Hammache, 2005). The methods available in the literature have the following drawbacks. The TH model does not consider the contribution of superheat in the inlet and the outlet steam in the power generation. THM parameters are based on regression parameters derived from a small sample of steam turbines, and consequently are not applicable for all the possible sizes of turbines.

The TH model for targeting does not include the superheat conditions at each level, which results in significant error for estimating cogeneration potential. The THM model uses an iterative procedure based on specific heat loads to calculate the mass flowrate for the turbines. The calculation of flowrate in Sorin’s methodology is based on the flow of energy, and power produced by the system is estimated with the isentropic efficiency, available heat for power generation and inlet and outlet temperatures of Rankine cycle. However, there is no justification for the assumption that thermodynamic behaviour of all the steam turbines to be used acts as that of the Rankine cycle.

The new algorithm calculates the minimum required flowrate from a steam generation unit (e.g. boiler) and the levels of superheat at each steam main based on the heat loads specified by site profiles of heat sources and sinks. The algorithm for the new procedure is explained in Fig. 1. The calculation of superheat temperature at each steam level is made, starting with a certain superheat temperature of the steam from the boiler. The procedure is based on the assumption that the steam supplied to the site utility systems from a boiler is at the superheated conditions required as VHP steam level. Fig. 2 shows the temperature-entropy diagram for the expansion of steam at two different pressure levels. The initial conditions of superheated steam at higher pressure and temperature level are represented by Point 1. The steam at lower pressure level for an isentropic expansion is shown as Point 2' on the curve. Isentropic expansion with $x\%$ efficiency is used to determine the enthalpy at point 2. It is assumed during the targeting stage that all the steam turbines are operating at their full load.

The cogeneration potential of the system is dependent on the expansion efficiency of $x$. This parameter is dependent on the capacity of the turbine and details of the calculation are given below. Steam properties are calculated for the given entropy and pressure at the lower steam level. If the degree of superheat in the resulting LP steam main is less than required, then operating conditions of VHP is updated and then iterates until the acceptable superheated conditions for LP steam main is met.

In the bottom-up procedure, the temperature of the lowest steam level pressure is first used to calculate the steam mass flowrate for the expansion of steam between the lowest steam level and the higher pressure next to the lowest one. This procedure is sequentially repeated until the expansion interval for the highest steam pressure level. Flowrates at the higher levels are determined from the flowrate in the lower levels. The flowrate of steam for each expansion interval is a function of the heat load at that level and the enthalpy change to the condensate temperature at the given level. Superheated steam is condensed and supplied to downstream processes at condensate temperature of the steam.

$$\dot{m} = \frac{\dot{Q}}{\Delta H} \quad (1)$$

where $\dot{m}$, mass flow rate; $\dot{Q}$, heat load for a given level; $\Delta H$, enthalpy change from superheat conditions at the given level to condensate conditions at that pressure.

2.2. Isentropic efficiency calculation

It is at the designer’s discretion to which value to use for the isentropic efficiency for the cogeneration targeting method. Information of isentropic efficiency available in the literature can be also used. Mavromatis and Kokossis (1998) developed a thermodynamic model to estimate the isentropic efficiency of single and multiple extraction turbines. Varbanov et al. (2004) presented equations to determine the parameters in terms of saturation temperature. Medina-Flores and Picón-Núñez (2010) modified the correlations of Varbanov et al. (2004) to obtain the regression parameters as a function of inlet pressure. Table 1 shows the regression parameters obtained by Varbanov et al. (2004), which are based on the turbine data of Peterson and Mann (1985).

$$\eta_s = \frac{W_{\text{max}}}{W_{\text{is,max}}} \quad (2)$$

$$W_{\text{max}} = \left( \frac{W_{\text{is,max}} - A}{B} \right)$$

Table 1 - Regression coefficients for single extraction turbines (Varbanov et al., 2004).

<table>
<thead>
<tr>
<th>$b_1$ (MW)</th>
<th>$b_2$ (MW $\cdot^{{\circ}C}$)</th>
<th>$b_3$ ($^{{\circ}C}^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00108</td>
<td>1.097</td>
<td>0.00172</td>
</tr>
<tr>
<td>0.000538</td>
<td>1.155</td>
<td>0.000538</td>
</tr>
</tbody>
</table>

\[ A = b_0 + b_1 \cdot \Delta T_{\text{sat}} \]
\[ B = b_2 + b_3 \cdot \Delta T_{\text{sat}} \]

where \( n_{\text{iso}} \), isentropic efficiency; \( \Delta H_{\text{sat}} \), isentropic enthalpy change; \( A, B, b_0, b_1, b_2, b_3 \), regression coefficients.

The results are investigated with the STAR®, software for the design of utility systems for a single process or a group of processes involving power (electricity) and heat (steam) generation, and associated heat exchange and distribution units. The design procedure of utility systems in STAR® requires information about steam flowrates, heat supply and loads, VHP (very high pressure) steam specification (e.g. VHP steam generation capacity and temperature at the outlet of the boiler). At the initial targeting stage, some of these design parameters are not known. The parameters, such as steam flowrate from the boiler, steam level conditions, must be specified for the detailed design in STAR®. The information required for the calculation of cogeneration potential from the utility systems is current flowrate of steam generated, maximum and minimum flowrates of equipment, thermodynamic model and efficiency of steam turbines, steam demand and surplus for each steam main, superheat condition of steam generated from the boiler, etc. STAR® has both the isentropic and the THM model for the calculation of power generation of steam turbines, while can use either the TH or THM model for cogeneration targeting.

### 3. Optimization of steam levels

As explained before, the choice of steam level in the design of site utility systems is critical to ensure cost-effective generation of heat and power, and its distribution in the site. In a new design, pressures of steam levels can be readily optimized. However, for the retrofitting of existing systems, opportunities for the change of steam level conditions are limited. The mechanical limitation for the steam mains limits a significant increase in steam pressure. However, long term investment with a proper optimization of the steam levels may be economically viable, in spite of the fact that the short term investment cannot be justified (Smith, 2005). VHP steam generation in the boiler and hence the fuel costs in the utility boilers can be decreased by increasing the number of steam mains, which increases the heat recovery in the total site. The number of steam mains has a significant impact on the cogeneration potential. Therefore, to minimize fuel cost with maintaining high cogeneration potential, the design of utility systems should be thoroughly investigated.

#### 3.1. Optimization framework

In this study, optimization for determining the cost-effective conditions of steam mains for the site utility systems has been carried out, incorporating the new cogeneration targeting method. The optimization model is formulated in an NLP (non-linear programming) problem and the details of the optimization model are as follows.

##### 3.1.1. Objective function

The objective function is to minimize the amount of hot utility to be supplied from the steam generation unit (e.g. boiler). It should be noted that minimizing the amount of fuels to be used in the utility systems does not always correspond to minimum overall cost of site utility systems. However, it is extremely difficult to generalize the capital investment to be required in the conceptual stage of process design, and the current study focuses on the maximizing energy efficiency for the site utility systems, which provides sufficient information and reliable guidance for achieving cost-effective design in the later stage of detailed design. However, another point to be noted is that the method presented in this paper is generic for being able to accommodate different objective functions, for example overall fuel cost, operating profit, etc., as long as the relevant cost parameters are available.

\[
\text{minimise} \quad H_{\text{shifted, sink}} - H_{\text{heat, source}} \quad \text{VHP} 
\]

\[
H_{\text{shifted, sink}} \quad \text{Enthalpy of shifted heat sink for VHP} 
\]

\[
H_{\text{heat, source}} \quad \text{Enthalpy of heat source for VHP} 
\]

#### 3.1.2. Optimization variables

\[ P_i \quad \text{Pressure at} \; i \in \text{Steam levels} \quad (\text{VHP, HP, MP, LP}) \]

Four steam mains are used in the current optimization model, as this is most common in the large-scale industrial plant. A different number of steam mains, for example three levels (HP, MP and LP) can be considered, based on the needs and operating characteristics on the plant.

#### 3.1.3. Model constraints

Total source and sink profiles are generated from steam data for the site (Raissi, 1994). The design procedure for manipulating steam data to generate the site profiles is not a part of this study and those details can be found from Smith (2005) and Klemes et al. (2010). In order to maintain feasibility of heat recovery across steam mains, a constraint between sink and source site profiles is needed. First, the sink is shifted until the enthalpy of heat source at either of the steam levels is the same as the enthalpy of heat source corresponding to the pinch point, and then enthalpy difference at each steam levels is always greater than zero.

\[
H_{\text{shifted, sink, i}} - H_{\text{heat, source, i}} \geq 0 \quad i \in \text{Steam levels} 
\]

\[ \text{(VHP, HP, MP, LP)} \]

#### 3.1.4. Mass balance

The mass flow rate of steam between steam levels is given as:

\[
\dot{m}_{i\rightarrow j} = \dot{m}_{j\rightarrow k} + \frac{\dot{Q}_j}{\Delta H_j} 
\]

where \( \dot{m}_{i\rightarrow j} \), mass flow rate of steam through a turbine between \( i \) and \( j \) steam levels; \( \dot{m}_{j\rightarrow k} \), mass flow rate of steam through a turbine between \( j \) and \( k \) steam levels; \( \dot{Q}_j \), heat duty at \( j \) steam level; \( \Delta H_j \), enthalpy extracted by process from superheated steam at \( j \) level to reach condensate conditions.

Power is calculated base on the new design algorithm as shown in Fig. 1.

Fig. 3 shows the model for the determination of optimal steam pressure levels for site utility systems. The change in the steam pressure levels shifts the site sink and surplus profiles, along with heat demand and supply. Cogeneration potential for the site composite is calculated from the new algorithm. The process is repeated until optimum pressure levels corresponding to minimum value of objective function are found for the site.
4. Case study 1

An illustrative case study is used to test different cogeneration targeting methodologies. The four steam levels considered in this example are very high pressure (VHP), high pressure (HP), medium pressure (MP) and low pressure (LP) at 120, 50, 14 and 3 bar(a) respectively. The heat demand at HP, MP and LP steam levels is 50, 40 and 85 MW respectively. The efficiency of the boiler is assumed to be 100% for simplicity, which can be updated, according to boiler data available, and steam at a temperature of 575 °C is produced. Water supplied to the boiler and the condensate returns are both assumed to be at a temperature of 105 °C.

In this work, cogeneration targeting methods have been applied to the case studies with back pressure turbines only. However, it can be easily extended to condensing turbines. One of the additional constraints on condensing turbine is a maximum wetness permitted at the exhaust. Wetness factor in the condensing turbine can be controlled by adjusting the superheat in the steam mains, as similarly treated in the consideration of level of superheat in LP steam.

The isentropic efficiency is calculated as given in Eq. (2), while the mechanical efficiency was assumed to be 100%. Overall shaftpower target from different methods are compared in Table 2, and for TH model, the value of conversion factor (CF) is assumed to be 0.00135. Fig. 4 shows a schematic illustration of shaftpower target obtained from the new method. The main difference between the new method and existing TH and THM models is the calculation of superheat temperature for each steam main, as explained previously. Superheat temperature of the outlet LP steam should be greater than the saturation temperature of LP steam, in order to avoid excessive condensation in the LP mains or excessive wetness in the outlet of the turbine, and thereby reduced performance and efficiency. The amount of superheat in VHP steam determines the superheat in LP steam. In the new algorithm, the superheat in VHP steam from the boiler is treated as a variable, which is adjusted to ensure the required degree of superheat in the LP steam.

Once the steam levels, and the heat surplus and deficit are known, a detailed design procedure is used for the optimal design of the utility systems or to find the optimum operating conditions for an existing design. However, as discussed before, the detailed design requires additional parameters, such as flowrates and superheat steam temperatures. These additional parameters are specified by trial and error. STAR® was used to test the targeting potential against the actual production from the steam turbine. The shaftpower was calculated by the isentropic model with isentropic efficiency calculated as shown in Eq. (1). The utility systems consist of a boiler supplying VHP steam at 575 °C. The steam is passed from the boiler to the higher pressure steam main to lower pressure levels.

Fig. 3 – Flowchart to determine optimum steam pressure level.

Fig. 4 – Case study 1: Cogeneration potential obtained from new method.

Fig. 5 – STAR® simulation isentropic efficiency.
Table 2 – Comparison of cogeneration targeting results.  

<table>
<thead>
<tr>
<th>Pressure (bars)</th>
<th>VHP</th>
<th>HP</th>
<th>MP</th>
<th>LP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Saturation temperature (°C)</td>
<td>120</td>
<td>50</td>
<td>14</td>
<td>3</td>
</tr>
<tr>
<td>Heat demand (MW)</td>
<td>324.7</td>
<td>264</td>
<td>195.1</td>
<td>133.6</td>
</tr>
</tbody>
</table>

Table 3 – Problem data parameters.  

<table>
<thead>
<tr>
<th>Methodology</th>
<th>Overall (MW)</th>
<th>VHP-HP (MW)</th>
<th>HP-MP (MW)</th>
<th>MP-LP (MW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sorin’s methodology</td>
<td>41.43 (±5.9%)</td>
<td>18.2</td>
<td>14.46</td>
<td>8.77</td>
</tr>
<tr>
<td>TH model in STAR®</td>
<td>33.02 (±15.6%)</td>
<td>14.35</td>
<td>11.62</td>
<td>7.06</td>
</tr>
<tr>
<td>THM model in STAR®</td>
<td>14.1 (±64.0%)</td>
<td>9.4</td>
<td>4.7</td>
<td>0</td>
</tr>
<tr>
<td>STAR® simulation – constant isentropic efficiency</td>
<td>39.0 (±0.3%)</td>
<td>14.85</td>
<td>14.78</td>
<td>9.37</td>
</tr>
</tbody>
</table>

Fig. 6 – Sink and source profiles for a given site.

steam main, via a steam turbine. Any unused steam can be passed through the vent. The process cooling and heating duty at each steam main level is specified as given in Table 3. Fig. 5 shows the site utility systems modelled in the STAR® environment. The overall turbine shaftpower is 39.12 MW.

Discrepancies in the shaftpower targets shown in Table 2 are mainly due to the assumptions used in these models. The shaftpower target obtained from the new method of 39.12 MW is only 0.31% different from the detailed design procedure in STAR®.

4.1. Optimization of steam levels

Site data was taken from an example available in the literature (Perry, 2009). Site sink and source profiles are shown in Fig. 6. Four steam mains are available at very high pressure (VHP), high pressure (HP), low pressure (LP) and medium pressure (MP) respectively. The sink profile is shifted by the minimum enthalpy difference between the source and sink profiles, which identifies a site pinch point for the utility system.

The site utility grand composite curve (SUGCC) plots the difference between the hot and the cold composite curves as shown in Fig. 7. The heat generation and use at individual steam level is shown in Figs. 7–10, which shows the cogeneration potential between different steam levels. The power output for these zones for the optimized case, based on the new algorithm, is found to be 7.69 MW.

The objective function is the minimization of the utility cost. The hot utility is supplied as VHP steam from the boiler. The optimization framework described in previous section and the model calculations are performed in Microsoft Excel®. The size of the model and the optimization problem is small and therefore the solver function in Microsoft Excel® can be effectively used for the minimization of the utility cost. Levels of steam pressures affect both heat recovery and the cogeneration potential, via the steam turbine network (Smith, 2005).

Table 4 shows the base case conditions for the four steam levels. Optimum steam level pressure and temperature along with heat load at each level is shown in Table 5. The minimum VHP steam generation required from the boiler is 70.22 MW, while the VHP steam flowrate requirement from the boiler is 88.16 t/h. Steam generation required at VHP mains has been reduced from 105.20 MW to 70.22 MW for the optimized case. However, the cogeneration potential was reduced from 8.8 MW for base case to 7.67 MW for the optimized case. Therefore, increasing the heat recovery reduces the steam generation from the boiler as well as the cogeneration potential for this particular example. If power generation in the site should be increased, then additional VHP steam should be generated to pass through steam mains. Alternatively, condensing power generation can be introduced.

This optimization framework can be extended to accommodate other economic scenarios (e.g. to minimize the fuel costs with maintaining the same cogeneration potential) or practical constraints (e.g. the maximum number of steam levels allowed).

5. Utilisation of low grade heat in the process industries

Low grade heat source can be very useful to provide energy to the heat sink by upgrading low-grade energy (e.g. low pressure steam). The upgrade of low grade heat can be carried out by heat pumping, absorption refrigeration, thermo compressors, etc., by recovering and/or upgrading waste heat from various sources (e.g. gas turbine exhaust), then utilising them in the wide range of applications (e.g. drying and boiler feedwater heating).

The integration of low grade heat upgrading technologies with an existing site utility system is evaluated as given in Fig. 11. The characteristics of low grade energy, such as available heat sources for the potential use in the heat pumping, ORC (Organic Rankine Cycle) and boiler feed water heating, is obtained from total site sink and source profiles. Aspen HYSYS® simulation is used to obtain the performance indicators, such as coefficient of performance (COP), energy efficiency and other necessary information, for example size of equipment. Different characteristics of available waste heat are reflected through Aspen HYSYS® simulation, of which results are combined to the optimization framework for calculating the site-wide and integrated performance of
energy systems, subject to low-grade heat upgrading or waste heat recovery.

For the simulation and optimization of the utility systems, the optimization framework developed by Aguillar (2005) has been adopted for the purpose of optimization, which determines the most appropriate configuration of utility systems with optimal operating conditions. Aguillar’s method has been extended to systematically incorporate the impact associated with the introduction of low-grade heat recovery in the site (see the details of modelling and optimization framework in the reference of Aguillar, 2005).

Within the optimization framework, linear models have been used for all the energy equipment, so that an MILP solver in the GAMS® environment could be used for the optimization.

### Table 4 – Base case steam levels (Perry, 2009).

<table>
<thead>
<tr>
<th>Pressure (bar)</th>
<th>Temperature (°C)</th>
<th>Heat load (MW)</th>
<th>Saturation temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>180</td>
<td>625</td>
<td>105.20</td>
<td>357.14</td>
</tr>
<tr>
<td>50</td>
<td>458.74</td>
<td>137.01</td>
<td>264.09</td>
</tr>
<tr>
<td>10</td>
<td>322.1</td>
<td>125.29</td>
<td>180.04</td>
</tr>
<tr>
<td>2</td>
<td>143.63</td>
<td>81.98</td>
<td>120.36</td>
</tr>
</tbody>
</table>

### Table 5 – Optimized steam levels.

<table>
<thead>
<tr>
<th>Pressure (bar)</th>
<th>Temperature (°C)</th>
<th>Heat load (MW)</th>
<th>Saturation temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>180</td>
<td>625</td>
<td>70.22</td>
<td>357.14</td>
</tr>
<tr>
<td>46.65</td>
<td>449.21</td>
<td>113.45</td>
<td>259.79</td>
</tr>
<tr>
<td>12.26</td>
<td>308.08</td>
<td>107.57</td>
<td>189.09</td>
</tr>
<tr>
<td>2.25</td>
<td>214.48</td>
<td>55.34</td>
<td>124.30</td>
</tr>
</tbody>
</table>

### 6. Case study 2

Various design options for low-grade heat upgrade and recovery are evaluated in the second case study. The base case design taken from (Aguillar, 2005) is shown in Fig. 12, which consists of four boilers (with capacity of 40 kg/s for each boiler), four back-pressure turbines between VHP and HP levels, and one back-pressure turbine between HP and LP steam levels. Two multi-stage turbines are available for the expansion of steam between HP-MP and MP-LP respectively, while there are four mechanical pumps to be driven by either steam turbines or electric motors, and an electric motor is used for the supply of the feed water to the boiler.
Site data for heat load, electricity demands, pump electricity demand, condensate return and cooling water is shown in Table 6. The site operating periods are divided into two major categories as summer and winter, with 67% of year given as winter. The ambient temperature, relative humidity, electricity natural gas and fuel oil price is shown in Table 7. The total number of working hours for the site is assumed to be 8600 h per year. The latent heat values for fuel oil and natural gas are 45 and 50.24 MJ/kg respectively.

Grand composite curves (GCC) of the individual process are modified by removing the pockets corresponding to additional heat recovery within the process. These modified process GCC are then combined together to form the total site sink and source profile (Fig. 13). The site utility grand composite curve (SUGC) represents the horizontal separation between the source and the sink. Steam demand at

![Fig. 9 – Site profile targets for steam generation and steam usage.](image)

![Fig. 10 – Site profile with cogeneration potential area.](image)

![Fig. 11 – Algorithm for evaluation of low grade heat upgrade technology.](image)
VHP, HP, MP and LP levels are 110.8, 21.4, 9.3 and 73.6 MW respectively. Power generation potential is represented as areas in the SUGCC with VHP-HP, HP-MP and MP-LP cogeneration potential of 79.8, 58.4 and 49.1 MW respectively (Fig. 13c).

6.1. Integration of a heat pump

A model of heat pump has been simulated in Aspen HYSYS®. It consists of four items of equipment: an evaporator, a compressor, a condenser and a throttle valve. Refrigerant R112-a is used as a working fluid. Low grade heat is supplied in the evaporator at a temperature of 115°C. Electrical energy is used in the compressor to raise the pressure of the vapour. LP steam is generated from the condenser at a temperature of 150°C. A throttle valve is used to reduce the pressure of the vapour liquid mixture from the condenser.

Purchase cost of a heat pump cycle is approximated based on a linear correlation between the cost and the evaporator duty.

\[ \text{PC}_{\text{heat pump}} = A \times \Delta H_{\text{eva}} + B \]  
(6)

where \( \text{PC}_{\text{heat pump}} \), purchase cost of heat pump; \( \Delta H_{\text{eva}} \), evaporator duty (MW); \( A, B \), regression coefficients (\( A = 0.1 \text{M$/\text{MW}; B = 1.15 \text{M$/}} \)).

The total site source and sink profile before and after integration of the heat pump is shown in Fig. 14 and Table 8. LP steam demand changes from 73.62 to 18.67 MW in summer and from 88.54 to 33.59 MW in winter. To maintain a minimum temperature difference of 10°C in the evaporator of the heat pump, low grade heat is only extracted above 115°C. The COP of the heat pump as calculated from the simulations is 3.3, and the external electricity consumption from the site increases as shown in Table 9 from 68.82 to 85.42 MW in summer and from 62.2 to 78.8 MW in winter.

From the operational optimization of site utility systems with the integration of heat pump, it is found that external power cost increases from 22.67 M$ to 35.03 M$ after integration of heat pump, while fuel cost decreases from 93.08 to 82.09 M$. Total annual cost increases to 118.67 M$/yr from 116.32 M$/yr after integration of the heat pump. Therefore, with current cost parameters, it is not economic to introduce a heat pump for the utilization of low-grade heat.

6.2. Integration of Organic Rankine Cycle (ORC)

Aspen HYSYS® is used to calculate the efficiency and the purchase cost function for the ORC. The ORC set up consists of an evaporator, a turbine, a condenser and a pump. Benzene is used as the organic working fluid in this case. Low grade heat at 110°C is used to vaporize benzene at elevated pressure (1.145 bar). Benzene vapour is used to drive a turbine along with reduction in pressure (0.145 bar). The vapour stream from...
the turbine at a low pressure is condensed in the condenser (27 °C). A pump is used to pump the low pressure organic liquid stream to elevated pressure (1.145 bar) before being fed to the evaporator.

The efficiency of the ORC is 11.10% from the simulation. The purchase cost of the ORC is given by the total cost of equipment such as condenser, evaporator and turbine. The cost of the evaporator and condenser is obtained from Milligan (2003), while turbine cost is obtained from Peters et al., 2003. Purchase cost of the ORC is approximated based on a linear correlation between the cost and the evaporator duty.

\[ P_{C^{\text{ORC}}} = A \times \Delta H_{\text{eva}} + B \]  

where \( P_{C^{\text{ORC}}} \), purchase cost ORC; \( \Delta H_{\text{eva}} \), evaporator duty (MW), A, B, regression coefficients (\( A = 0.01 \text{ M$/MW}; B = 25.1 \text{ M$/} \)).

The total site source and sink profiles after integration of the ORC are shown in Fig. 15. 62.11 MW of low grade heat is available above a temperature of 105 °C. Cold utility requirement is reduced by 62.11 MW. The amount of electrical energy is reduced from 68.82 to 61.99 MW during summer and from 62.2 to 55.39 MW during winter. The purchase cost of the ORC at the given evaporator duty is 17.13 M$.

### 6.3. Absorption refrigeration

Simulation flowsheet for absorption refrigeration systems is illustrated in Fig. 16. Heat is released at a temperature of 32 °C to the surrounding at a pressure of 13 bar in the condenser. Ammonia vapour is passed through a throttle valve to reduce the pressure to 0.145 bar before absorbing heat from the surroundings at a low temperature (−5 °C) as the refrigeration load in the evaporator. Ammonia vapour is absorbed with the
lean solution of ammonia in the absorber. Heat is released to
the surroundings from the absorber. A concentrated solution
of ammonia in water is pumped from 0.1459 bar to 13 bar into
the generator. Low grade heat is used in the generator to sep-
erate ammonia from the concentrated solution to produce a
lean solution of ammonia water. Heat is exchanged between
outgoing lean solution of ammonia water and incoming strong
solution.

Low grade heat is recovered through absorption refriger-
ation, and the generated cold energy replaces the current
refrigeration load in the site, as illustrated in Fig. 17. The

performance of the simulated absorption refrigeration is that
around 26 units of cooling can be provided when 100 units of
low grade heat are recovered.

6.4. Boiler feed water (BFW) heating

Low grade heat can also be used to raise the tempera-
ture of make-up water from 25 °C to 101.3 °C. With the
available low grade heat in the case study, 177.29 m³/s and
197.58 m³/s of make-up water can be heated up for sum-
mer and winter respectively. This reduces the cost of fuel
consumed in the boiler from 93.08 M$/yr to 80.57 M$/yr. The
benefits of BFW heating depend on condensate recycling
process and condensate management. BFW heating does
not change the hot utility requirement from the base case.
However, the cost of fuel required to supply the hot util-
ity decreases from 93.08 to 80.57 M$/yr, due to decrease
in the heating required for boiler feed water. The overall
energy cost decreases from 117.83 M$/yr in the base case to
107.63 M$/yr.

Table 10 shows overall comparison between the various
low grade heat upgrade options. It should be noted that
care must be taken to interpret these results, as the calcula-
tion is based on the particular cost parameters, specified
site conditions, and fixed operating conditions of upgrading

Fig. 16 – Aspen HYSYS® model of absorption refrigeration.

Fig. 17 – Recovery of low grade heat for absorption
refrigeration.
7. Conclusions

The selection of steam level conditions is important for achieving cost-effective heat and power production in industrial sites. A new cogeneration targeting model has been developed in this work, as existing models have been shown to give misleading results, compared to detailed design procedure. The new model is based on isentropic expansion and the results obtained from the new model agree well with the results from the detailed isentropic design methods. The new method has been incorporated in an optimization study, which systematically determines the levels of steam mains, subject to economic parameters and constraints.

Wide range of technologies is available for upgrading or recovering low-grade heat in process industries, and its techno-economic impact has been addressed with the aid of simultaneous consideration of site utility systems optimization and performance of upgrading technologies. It should be emphasised that the selection and design of low-grade heat upgrading or recovery should be made in the simultaneous consideration of the system-wide environment and constraints, as the best heat upgrade technology is strongly dependent on the site fuel and electricity cost, condensate management system, and characteristics of low grade heat (quality and size).

The methodology developed needs to be further extended to accommodate the integration of renewable energy sources, such as solar, wind, geothermal, etc. to the total site and to consider over-the-pence process integration between process sites and community energy systems. Another work to be considered in future is to improve the applicability and practicality of the proposed design method in practice, for example, engineering limitation, geographical constraints and regulatory barriers existed in the implementation of low grade heat recovery and utilisation.

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<thead>
<tr>
<th>Options</th>
<th>Table 10 – Techno-economic evaluation of low grade heat upgrade technologies</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>Hot utility (MW)</td>
</tr>
<tr>
<td>Base case</td>
<td></td>
</tr>
<tr>
<td>Hot utility</td>
<td>252.17</td>
</tr>
<tr>
<td>Hot pump</td>
<td>201.23</td>
</tr>
<tr>
<td>District heating</td>
<td>252.17</td>
</tr>
<tr>
<td>Absorption refrigeration</td>
<td>252.17</td>
</tr>
<tr>
<td>BFW heating</td>
<td>252.17</td>
</tr>
</tbody>
</table>