Recovery of flue gas energy in heat integrated IGCC power plants using the contact economizer system

Vhutshilo Madzivhandila[a], Thokozani Majozi[a,b*], and Thoshko Zhelev[c]

[a] Department of Chemical Engineering
University of Pretoria, Pretoria 0002, South Africa

[b] Modelling & Digital Science
Council for Scientific and Industrial Research, Pretoria 0002, South Africa

[c] Department of Chemical and Environmental Science
University of Limerick, Ireland

[*]Corresponding Author's E-mail: thoko.majozi@up.ac.za

Keywords: IGCC; Contact economizer system; Flue gas; Mickley's graphical technique

Abstract

In this work, the contact economizer system is used to recover low potential heat from the gas turbine exhaust (flue gas) stream of a heat integrated IGCC plant adopted from previous studies. Recovery of this heat was demonstrated to be capable of further increasing the thermal efficiency of the plant. Application of Mickley's graphical technique for dehumidification in a contact economizer system following a slightly different procedure that allows calculation of the exact ratio between the liquid-phase heat transfer coefficient and gas-phase mass transfer coefficient demonstrated a possible increase in thermal efficiency from 54% to 55%. This method however is incapable of obtaining the optimum thermal efficiency due to the monotonous relationship between the boiler feed water temperature and the thermal efficiency of the plant.

1. Introduction

The IGCC (Integrated Gasification Combined Cycle) is one of the innovative power generation technologies that promises to provide a large share of the future world's energy needs in an economical and environmentally friendly way. This technology combines two primary systems, gasification and the combined cycle made up of the gas turbine and the steam turbine. The process followed by IGCC plants can be viewed as a two path process, the first path being the gas path and the second path being the steam path. These two paths are integrated by a gas cooler (referred to as a boiler) and a heat recovery steam generator (HRSG) to form one system. Five IGCC plants are currently operating commercially worldwide and the number of demonstrations is increasing rapidly.

A lot of research on IGCC plants is currently on the way and a lot of researchers have managed to improve the process in various aspects to obtain higher thermal efficiencies. Most of the research conducted in this field has focused on the improvement of equipment and optimizing the operating parameters. Only a few researchers have focused on optimizing the use of energy within the plant. Considering the amount of energy available for use in this system, this should be one the areas of focus.

The objective of this contribution is to recover low potential heat from the gas turbine exhaust stream of heat integrated IGCC plants using a contact economizer system.

The basis of this contribution was that an increase in the boiler feed water temperature will result in an increase in the overall efficiency of IGCC plants as demonstrated in this paper.

2. Theory

2.1 The integrated gasification combined cycle

It is assumed at this stage that the reader is familiar with the fundamental structure and concepts of the IGCC shown in Figure 1. The stream of concern in Figure 1 is the gas turbine (GT) exhaust stream leaving the HRSG *en route* to stack. More details on this core stream will be given in the following sections.

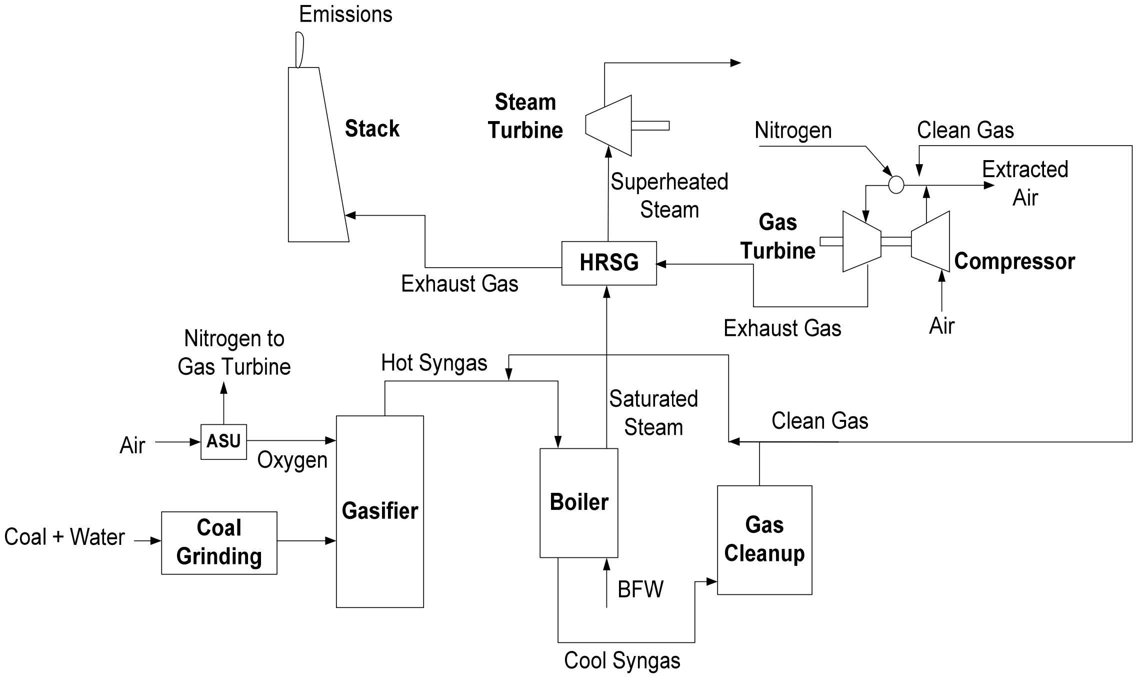

Figure 1: Basic flow sheet of an IGCC plant

2.2 Mickley's graphical technique

Mickley (1949) presented a simple and improved graphical method for the design of forced draft air conditioning equipment. This method is an extension of the "enthalpy potential" method proposed by Merkel (1925) and is recognized as the most convenient method for determining the size of the equipment for direct contact systems. All operating conditions of the equipment can be quickly determined by this method, and the danger of fog formation ascertained. The design equations for this graphical enthalpy potential method are given below, with a large portion of their development presented elsewhere (Mickley 1949).

$$H_G = c_s + (T_G - T_0) + \lambda_0 x \tag{1}$$

$$GdH_G = k_G a_M (H_i - H_G) dZ \tag{2}$$

$$\frac{H_{G1} - H_{G2}}{T_{L1} - T_{L2}} = \frac{Lcp_L}{G} \tag{3}$$

$$\frac{H_G - H_i}{T_L - T_i} = -\frac{h_L}{k_G} \tag{4}$$

$$\frac{dH_G}{dT_G} = \frac{H_G - H_i}{T_G - T_i} \tag{5}$$

a_M - ***mass transfer area***
c_s - ***humid heat***
G - ***flowrate of the gas***
H_G - ***enthalpy of the gas***
h_L - ***liquid-phase heat transfer coefficient***
k_G - ***gas-phase mass transfer coefficient***
T_i - ***temperature at liquid-gas interface***
T_L - ***liquid phase temperature***
T_0 - ***reference temperature***
X - ***humidity of the gas***
Z - ***tower height***
λ_0 - ***latent heat of vaporization of water at reference temperature*** T_0
1 - ***top of the packed bed column***
2 - ***bottom of the column***

Equation 1 defines the enthalpy of humid gas while equation 2 relates this enthalpy to the tower height. Equation 3 and equation 4, the operating line and the tie line respectively, govern the dehumidification/humidification operation with the help of the equilibrium curve and equation 5. These equations operate under the following assumptions:

- At the water-gas interface, the gas is saturated at the interface temperature T_i.
- The change in the water flowrate due to evaporation or condensation is negligible.
- The heat transfer area is equal to the mass transfer area. This assumption holds if and only if the interfacial area is fully wetted.

Figure 2 represents typical results of Mickley's graphical technique applied to a dehumidification process. Curve 'EF' in this figure represents the equilibrium line or saturation curve. Line 'AB' is the operating line constructed from equation 3 with point 'A' corresponding to the entering bulk-gas enthalpy and the leaving bulk-water temperature. The slope of this line is L/G as indicated in equation 3.

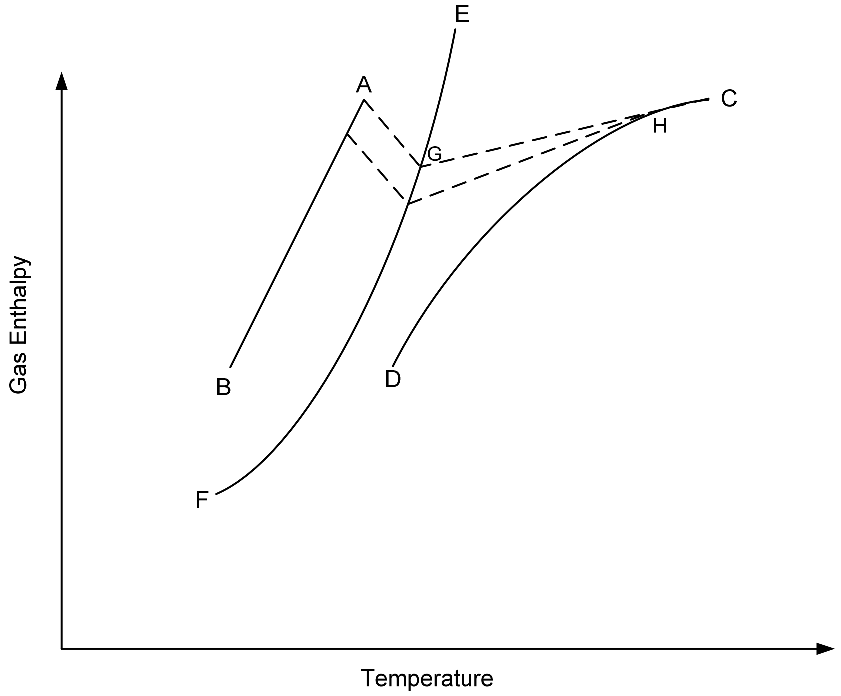

Figure 2: Enthalpy-Temperature diagram indicating the dehumidification process

The construction of the dehumidification path begins at point 'C' which represents the gas-water vapour mixture at the bottom of the tower. A tie line with slope $-h_L / k_G$ is drawn from point 'A' to intersect the equilibrium line at point 'G'. The coordinates of this intersection point are (T_i, H_i). A straight line drawn from point 'C' to point 'G' then gives the direction of the initial tangent to the gas path. The slope of this line is $(H_i - H_G)/(T_i - T_G)$, the ratio of the enthalpy driving force to the temperature driving force. By virtue of equation (5), the slope of line 'CG' also represents dH_G / dT_G , the rate of change of bulk-gas enthalpy with bulk-gas temperature. Assuming this slope is constant over a small interval, point 'H' represents the bulk-gas enthalpy and temperature at a short distance above the bottom of the tower. The construction is extended with a new tie line and a new direction of the path line tangential to point 'H'. This exercise is repeated until a complete gas path (CD) is achieved.

2.3 The contact economizer system

The contact economizer system (CES) is a low potential heat recovery system allotted to explore the simultaneous management of heat and mass transfer between a gas stream and a desiccant stream

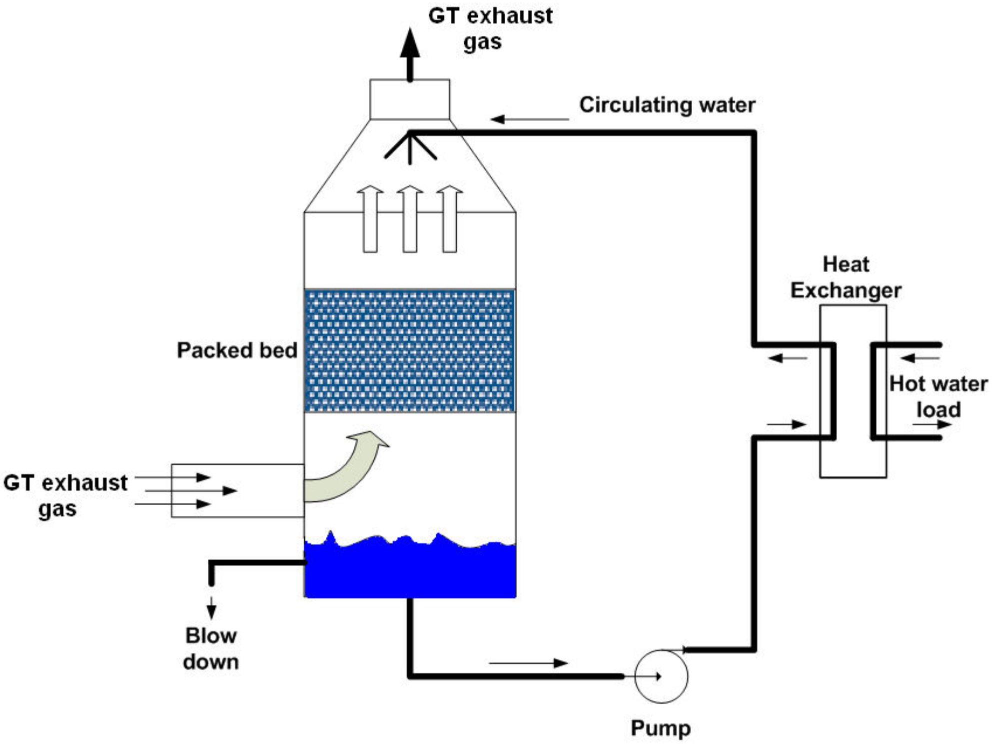

Figure 3: Basic structure of the contact economizer system

Figure 3 represents a typical CES for the recovery of heat from the GT exhaust stream using water as a desiccant. The CES process involves direct heat transfer between the hot gas stream and a cold circulating water (desiccant) stream accompanied by dehumidification of the flue gas, in a packed bed column. The heated desiccant that leaves at bottom of the packed bed column can then be used as a source of heat for other operations.

Literature (Mickley 1949) has shown that the circulating water can only be heated to a point where its temperature equals the wet-bulb temperature of the gas provided an infinite heat exchange area is available. At this point, the gas temperature enthalpy curve crosses the equilibrium line and fog formation is a possibility. Consequently, the circulating water should be heated to a certain temperature "ΔT" degrees lower than the wet-bulb temperature of the gas. This allows for thermal driving forces and also ascertains that fogging conditions will not be attained in the tower. "ΔT" values as low as 2.5°C are feasible for packed bed columns and have been previously demonstrated in literature (Zhelev and Semkov 2002).

3. Methodology

Application of the contact economizer system in conjunction with the graphical construction of the dehumidification path line according to Mickley is discussed below, but firstly the basis of this research is demonstrated. Equation 3 was adopted from previous studies (Madzivhandila, Majozi and Zhelev, 2009) and represents how the amount of energy available ($Q_{available}$) within the IGCC system can be used up by the steam path to improve the thermal efficiency as described in that study.

$$Q_{available} = \dot{m}\left(H_{sl} - H_{BFW}\right) + \dot{m}\lambda_{v} + \dot{m}\left(H_{sp} - H_{ss}\right) \tag{3}$$

$\boldsymbol{H_{BFW}}$ ***- enthalpy of the boiler feed water***
$\boldsymbol{H_{sl}}$ ***- enthalpy of the saturated liquid in the boiler***
$\boldsymbol{H_{sp}}$ ***- enthalpy of the superheated steam***
$\boldsymbol{H_{ss}}$ ***- enthalpy of the saturated steam***
$\boldsymbol{m}$ ***- maximum boiler feed flowrate***
$\boldsymbol{\lambda_V}$ ***- latent heat of vaporization of water***

Manipulation of equation 3 by replacing $\boldsymbol{\lambda_V}$ by H_{ss} - H_{sl} results into equation 4.

$$Q_{available} = \dot{m}H_{sp} - \dot{m}H_{BFW} \tag{4}$$

Rearranging equation 4 by writing mH_{sp} as the subject of the formula results in equation 5, the energy balance of the plant. Stated in words, equation 5 shows that the energy carried by the superheated steam to the steam turbine (Q_{sp}) is equal to the sum of the energy available within the IGCC system ($Q_{available}$) and the energy carried by the boiler feed water ($mH_{BFW} = Q_{BFW}$) into the system.

$$\dot{m}H_{sp} = Q_{available} + \dot{m}H_{BFW} = Q_{sp} \tag{5}$$

$$Q_{sp} = Q_{available} + Q_{BFW}$$

Given that $Q_{available}$ is constant, it is evident from equation 5 that an increase in H_{BFW} will result in an increase in Q_{sp}. An increase in Q_{sp} in turn results in an increase in the overall thermal efficiency (η_{IGCC}) according to equation 6 where W_{ST}, the power output of the steam turbine, is given by equation 7.

$$\eta_{IGCC} = \frac{W_{GT} + W_{ST}}{Q_{COAL}} \tag{6}$$

$$W_{ST} = \eta_{ST} Q_{sp} \tag{7}$$

$\boldsymbol{\eta_{ST}}$ ***- thermal efficiency of the steam turbine***
$\boldsymbol{W_{GT}}$ ***- gas turbine power output***
$\boldsymbol{Q_{COAL}}$ ***- calorific value of coal***

The increase in η_{IGCC} by increasing Q_{sp} as discussed above is the basis of the application of the contact economizer system. The application of the aforementioned methodology with reference to Figure 4 is as follows:

Given the conditions of the GT exhaust:

1. Determine T_{BFW2} required for a certain increase in η_{IGCC} . This can be done by substituting equations 5 and 7 into equation 6 and writing H_{BFW} in terms T_{BFW2} .
2. Draw the equilibrium curve on the "H_G" vs. "T_L" graph.

3. Select a "ΔT_{min}" for the column and specify T_{L2} to be at least "ΔT_{min}" lower than the wet-bulb temperature (T_w) of the gas to allow for thermal driving forces. At the same time, make sure that T_{L2} is a certain "ΔT" degrees higher than T_{BFW2} in step 1 to allow for heat exchange in the heat exchanger. If there is no feasible "ΔT" for the heat exchanger (if T_{BFW2} is greater than T_{L2} or if the difference is too small), it is necessary to increase the moisture content of the gas so as to increase its wet-bulb temperature; hence increasing T_{L2}.

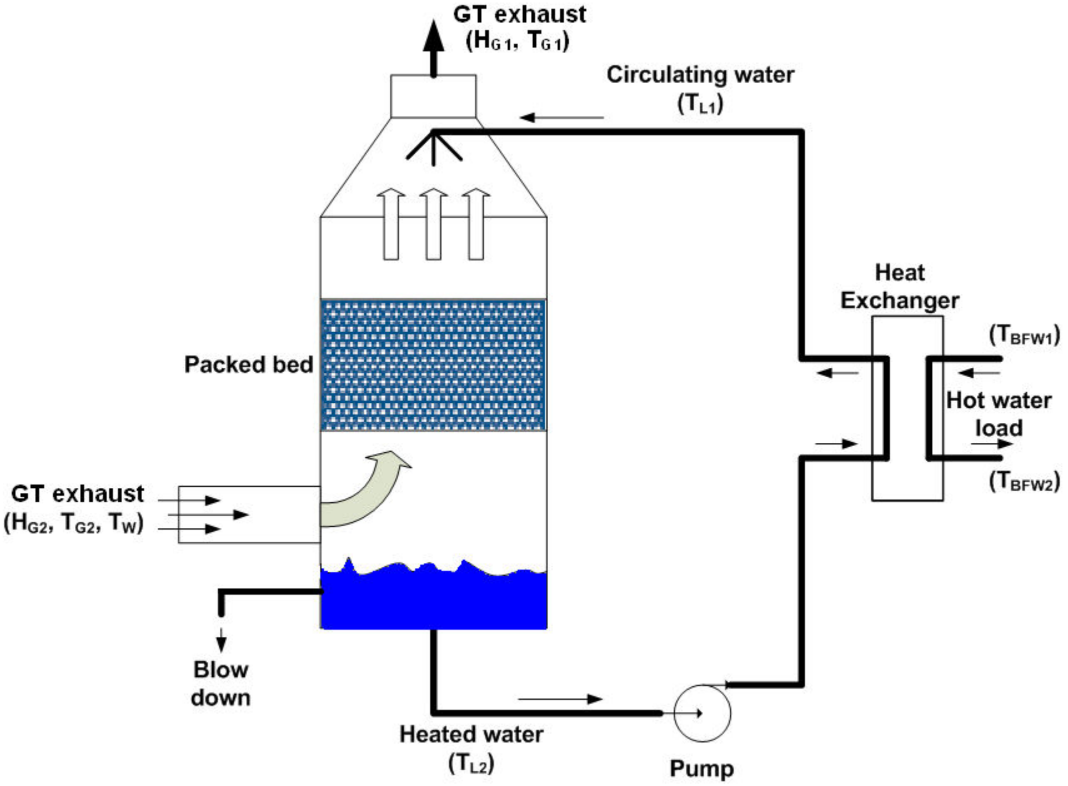

Figure 4: Necessary stream data for the application of the contact economizer system

4. Specify the circulating water flowrate (L) such that an $\frac{L}{G'}$ ratio is between 1.1 and 1.5 (considering a splash fill media is to be maintained in the packed bed column).
5. Draw the operating line using equation 2 and apply Mickley's graphical technique to determine the gas dehumidification path as follows:

   5.1 Specify T_{G1} and H_{G1} such that the gas is still above its condensation point.

   5.2 Choose a starting $-h_L/k_G$ and apply Mickley's graphical technique to determine the dehumidification path.

   5.3 Check if the dehumidification path intersects the point $(T_G, H_G) = (T_{G1}, H_{G1})$ in step 5.1 and proceed as follows:

   - If the dehumidification path intersects the point $(T_G, H_G) = (T_{G1}, H_{G1})$, then the design can be carried out with the specified $-h_L/k_G$ ratio.

- Otherwise repeat steps 5.2 and 5.3 while changing $-h_L/k_G$ in step 5.2 until the dehumidification path intersects the aforementioned point.

4. Case study

The CES was adopted into a heat integrated design of the Elcogas plant obtained from previous studies (Madzivhandila, Majozi and Zhelev 2009) and shown in Figure 5. Its (CES) application was on the gas turbine exhaust stream that leads to the stack, sitting at temperature of 90°C. The composition of the gas turbine exhaust stream was assumed to be the design composition of a typical IGCC. This stream was assumed to behave like air under these conditions. Consequently, enthalpy data was obtained from the air-water psychometric chart.

The five step procedure given in section 3 was followed in the application of the CES to increase the η_{IGCC} to 55%.

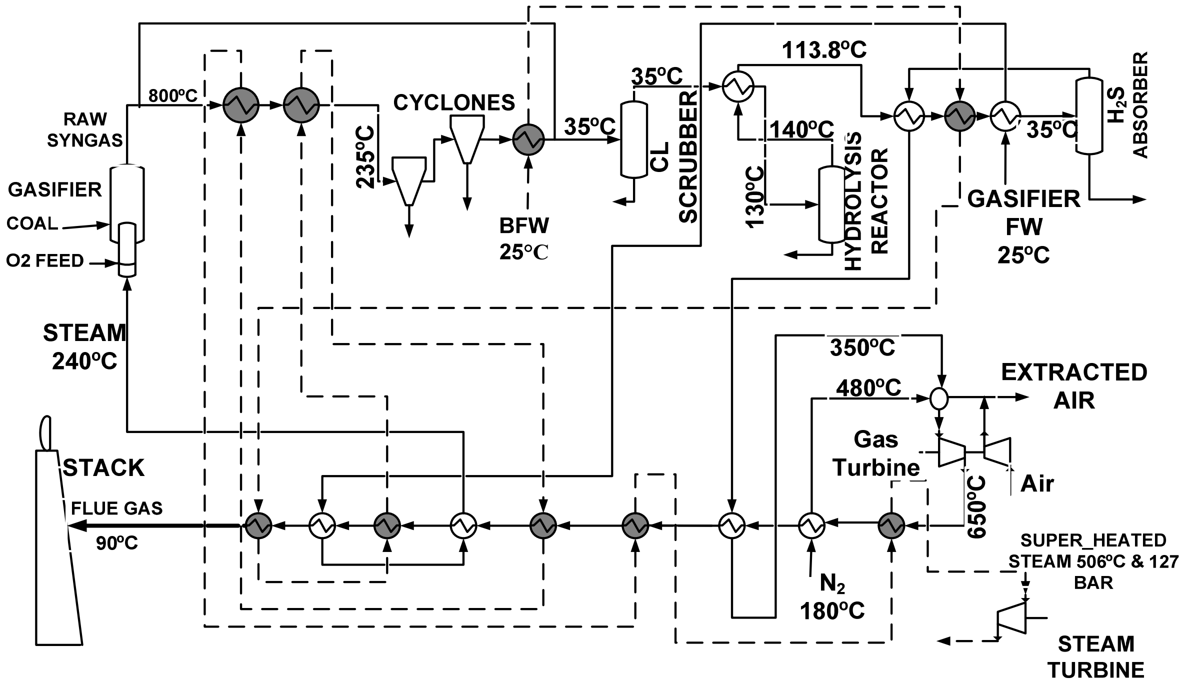

Figure 5: The heat integrated design of the Elcogas plant.

5. Results and discussion

Figure 5 shows the results of the case study after application of the graphical methodology in section 3. Mickley's graphical technique is shown in Figure 5a with the final gas path transposed into Figure 5b. The temperature (T_{BFW2} in Figure 4) required to increase η_{IGCC} to 55% as discussed in section 4 was found to be 57.3°C. A T_{L2} of 59°C at a column ΔT_{min} of 3°C after increasing the moisture content of the GT exhaust to 150 g/kg satisfied the required conditions for step 3 of the procedure. The ΔT_{min} maintained for the heat exchanger was 1.7°C. The outlet gas temperature and enthalpy $\left(T_{G1} \text{ and } H_{G1}\right)$ where specified to be (64°C, 450kJ/kg) as indicated by the small circles in

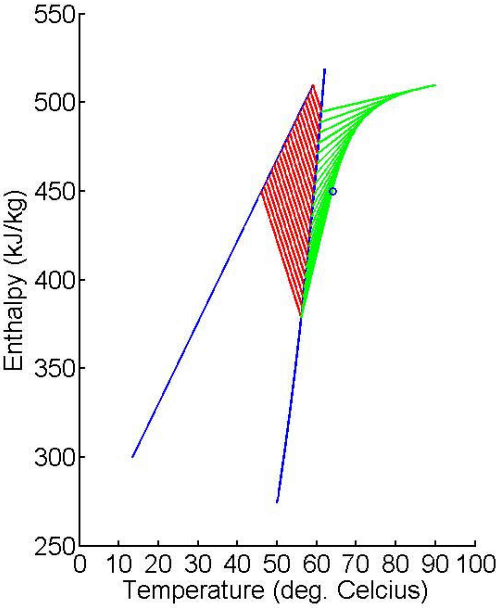

Figure 5a: Mickley's graphical technique

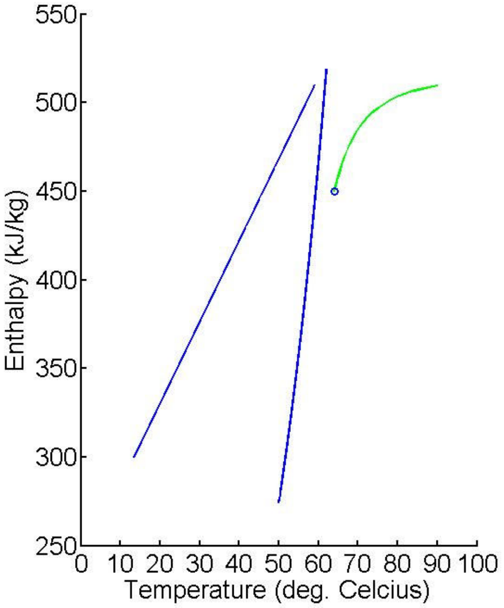

Figure 5b: Enthalpy-Temperature diagram indicating the operating line, equilibrium curve and the final gas path.

both Figure 5a and Figure 5b. The corresponding $-h_L/k_y$ that satisfied the conditions of step 5 of the procedure was 7.04 kJ/kgK.

6. Conclusion

Application of the CES proved effective in recovering the low potential heat from the GT exhaust stream in order to improve the IGCC plant's efficiency. The method used avoids the estimation of the ratio between the liquid-phase heat transfer and the gas-phase mass transfer. The approach followed is, however, incapable of obtaining the optimum thermal efficiency due to the monotony of the function relating the boiler feed water temperature (T_{BFW2}) to the thermal efficiency (η_{IGCC}) of the IGCC. Optimality can only be claimed if certain boundary conditions to T_{BFW2} are introduced.

References

1. Madzivhandila V., Majozi T., and Zhelev T. Process integration as an optimization tool in clean coal technology: A focus on IGCC, *Chemical Engineering Transactions*, **18**, 941-946, 2009.
2. Mickley, H.S., DESIGN OF FORCED DRAFT AIR CONDITIONING EQUIPMENT, *Chem. Engng. Prog.*, **45**, 739-749, 1949.
3. Zhelev T.K, and Semkov K.A., Cleaner flue gas and energy recovery through pinch analysis, *Journal of Cleaner Production*, **12**, 165-170, 2004.