CHAPTER 3: EXERGY ANALYSIS

Exergy is the tool, which indicates how far the system departs from equilibrium state. The concept of exergy was put forward by Gibbs in 1878. It was further developed by Rant in 1957.

3.1 QUALITY OF ENERGY

Quantitative evaluation of energy in a cycle or in a process can be done using the first law of thermodynamics. The direction of flow of heat or work is known from the second law of thermodynamics. However, it is equally important to assign the quality to the energy. As discussed in Chapter 1, energy can be broadly classified into (1) High grade and (2) Low grade energy. High grade form of energy are highly organised in nature and conversion of such energy to some other high grade form \((W \rightarrow W)\) is not dictated by the second law of thermodynamics. Conversion of high grade energy to low grade energy is not desirable. However, there may be some conversion to low grade energy as work is converted into other useful form. This is because of dissipation of heat due to friction (example: mechanical work \(\rightarrow\) Electricity, some losses are there due to the friction in bearing of machineries). Thus both the first and the second law of thermodynamics are to be considered for analysis.

Low grade energy such as heat due to combustion, fission, fusion reactions as well as internal energies are highly random in nature. Conversion of such form of energy into high grade energy \((Q \rightarrow W)\) is of interest. This is due to the high quality of organised form of energy obtained from low quality energy. Second law of thermodynamics dictates that conversion of 100% heat into work is never possible. That part of low grade energy which is available for conversion is termed as available energy, availability or exergy. The part, which according to the second law of thermodynamics, must be rejected is known as unavailable energy. Exergy analysis helps in finding the following:

- It can be used to determine the type, location and magnitude of energy losses in a system
- It can be used to find means to reduce losses to make the energy system more efficient

At this point, it is worth mentioning that the environment plays an important role in evaluating the exergy (composite property).
3.2 IMPORTANCE OF EXERGY ANALYSIS

Let us take an example. In case of coal fired power plant, the first law indicates that the condenser greatly effects the power plant efficiency as large amount of heat is transferred to the cooling water without providing any clue on the real usefulness of this relatively low temperature fluid. Also, energy balances do not provide information about the internal losses such as throttling valve and heat exchanger. Second law or exergy balance, however indicates that there is hardly 1% exergy loss in the condenser with more than 60% in the boiler. The contribution in the boiler exergy loss accounts for irreversibilities associated with combustion and finite temperature differences. Hence, analysis of exergy plays a deterministic role in identification of processes and rectifying the components.

3.3 AVAILABLE ENERGY REFERRED TO A CYCLE

Consider a cyclic heat engine as shown in Fig. 3.1. The maximum work output obtainable from this engine is the available energy \( AE \). The minimum energy that has to be rejected to the sink (as per 2nd law of thermodynamics) is called the unavailable energy \( UE \).

![Fig. 3.1 Available and unavailable energy in a cycle](image)

From energy balance

\[
Q_1 = AE + UE
\]  \hspace{1cm} (3.1)

Again,

\[
W_{\text{max}} = AE = Q_1 - UE
\]  \hspace{1cm} (3.2)

For the given source and sink temperatures \( T_1 \) & \( T_2 \) respectively
\[ \eta_{\text{rev}} = 1 - \frac{T_i}{T_{1}} \]  

(3.3)

If \( T_i \) is given, \( \eta_{\text{rev}} \) will increase with decrease of \( T_{1} \). The lowest practical temperature of heat rejection is the temperature of the surroundings, \( T_0 \).

\[ \therefore \eta_{\text{max}} = 1 - \frac{T_0}{T_{1}} \]  

(3.4)

And,

\[ W_{\text{max}} = \left[ 1 - \frac{T_0}{T_{1}} \right] Q_1 \]  

(3.5)

3.4 AVAILABILITY IN A FINITE PROCESS

Let x-y be a finite process in which heat is supplied reversibly to a heat engine. Taking an elementary cycle (shaded area in Fig. 3.2), let \( dQ_1 \) be the heat received by the engine reversibly at temperature \( T_1 \). Then

\[ dW_{\text{max}} = \frac{T_{1}}{T_{1}} - \frac{T_0}{T_{1}} dQ_1 = dQ_1 - \frac{T_0}{T_{1}} dQ_1 = AE \]  

(3.6)

For the whole process x-y,

![Diagram showing available and unavailable energy](image_url)

Fig. 3.2 Unavailable energy by 2\(^{nd}\) law of thermodynamics
Hence, maximum work obtainable is

\[ UE = Q_{x-y} - W_{\text{max}} \]  

\[ UE = T_0 \left( s_y - s_x \right) \]  

### 3.5 AVAILABLE ENERGY FROM A FINITE ENERGY SOURCE

Consider hot gas of mass \( m_g \) at temperature \( T \) when the temperature of environment is \( T_0 \). Let the gas be cooled at constant pressure to \( T_0 \) from state 1 and the heat given up by the gas \( Q_1 \) is utilized in heating up reversibly a working fluid of mass \( m_{wf} \) from state 3 to 1 along the same path so that the temperature difference between the gas and working fluid at any instant is zero and hence entropy increase of the universe is also zero.

The working fluid expands reversibly and adiabatically in an engine or turbine from state 1 to 2 doing work \( W_E \), and then rejects heat \( Q_2 \) reversibly and isothermally to return to the initial state 3 to complete the heat engine cycle.

Here,

\[ Q_1 = m_g \ C_{pg} \ (T - T_0) = m_{wf} \ C_{pwf} \ (T - T_0) = \text{Area 1-4-5-3-1} \]  

![Fig. 3.3 Available and unavailable energy in finite processes](image-url)
\[ \therefore m_s C_{pg} = m_{wf} C_{pwf} \quad (3.11) \]

Now

\[ \Delta s_{gas} = \int_{T_0}^{T} m_s C_{pg} \frac{dT}{T} \quad (3.12) \]

( negative as \( T_0 < T \) )

\[ \Delta s_{wf} = \int_{T_a}^{T} m_{wf} C_{pwf} \frac{dT}{T} = m_{wf} C_{pwf} \ln \frac{T}{T_0} \quad (3.13) \]

\[ \therefore \Delta s_{univ} = \Delta s_{gas} + \Delta s_{wf} = 0 \quad (3.14) \]

\[ Q_2 = T_0 \Delta s_{wf} = T_0 m_{wf} C_{pwf} \ln \frac{T}{T_0} = \text{Area 2-4-5-3} \quad (3.15) \]

\[ \therefore \text{Available energy} \]

\[ W_{max} = Q_1 - Q_2 \quad (3.16) \]

\[ = m_s C_{pg} (T - T_0) - T_0 m_s C_{pg} \ln \frac{T}{T_0} \]

\[ = \text{Area 1-2-3-1} \]

Hence, exergy of a gas of mass \( m_s \) at temperature \( T \) is

\[ AE = m_s C_{pg} \left[ (T - T_0) - T_0 \ln \frac{T}{T_0} \right] \quad (3.17) \]

### 3.6 Demonstration of Quality of Energy Based on Exergy

Consider a hot gas of mass \( m \) flowing through a pipeline (Fig. 3.4). Due to the heat loss to the surroundings, the temperature of the gas decreases continuously from the inlet state 'a' to the exit state 'b'. Thus, the process is irreversible. However, for sake of simplicity, let us consider the process to be reversible and isobaric between the inlet and outlet. For an infinitesimal reversible process at constant pressure,
Fig. 3.4 Energy quality degrades along the flow direction

\[ ds = \frac{m C_p dT}{T} \]  

or,

\[ \frac{dT}{dS} = \frac{T}{m C_p} \]  

Thus the slope \( \frac{dT}{dS} \) depends on temperature \( T \). With increase in \( T \), \( \frac{dT}{dS} \) increases and vice versa.

Let, \( Q \) unit of heat lost to the surroundings as temperature of gas decreases from \( T_1' \) to \( T_1' \), \( T_1' \) being average of the two temperatures.

Heat loss at section 1-1
\[ Q = m C_p \left( T'_1 - T'_1 \right) = T_1 \Delta S_1 \]  \hspace{1cm} (3.20)

Exergy lost with this heat loss at temperature \( T_1 \) is

\[ W_1 = Q - T_0 \Delta S_1 = T_1 \Delta S_1 - T_0 \Delta S_1 = (T_1 - T_0) \Delta S_1 \]  \hspace{1cm} (3.21)

At section 2-2, let heat loss be same as in section 1-1, \( Q \left( T_2 < T_1 \right) \)

\[ Q = m C_p \left( T'_2 - T'_2 \right) = T_2 \Delta S_2 \]  \hspace{1cm} (3.22)

Exergy loss due to this heat loss at temperature \( T_2 \) is

\[ W_2 = Q - T_0 \Delta S_2 = (T_2 - T_0) \Delta S_2 \]  \hspace{1cm} (3.23)

Considering the exergy loss at temperatures \( T_1 \) and \( T_2 \)

\[ W_1 = Q - T_0 \Delta S_1 = T_1 \Delta S_1 - T_0 \Delta S_1 = (T_1 - T_0) \Delta S_1 \]  \hspace{1cm} (3.24)

\[ W_2 = Q - T_0 \Delta S_2 = (T_2 - T_0) \Delta S_2 \]  \hspace{1cm} (3.25)

As \( T_1 > T_2 \), \( \Delta S_1 < \Delta S_2 \)

\[ \therefore W_1 > W_2 \]  \hspace{1cm} (3.26)

Interpretations:

1. For the same heat loss at two different temperatures, exergy loss is more with higher temperature. For instance, exergy loss is more at 1000 K than that at 300 K for 1 kJ of heat loss.
2. The more the temperature, the more is the quality of energy. For example, quality of energy of a gas at 1000 K is superior to that at 400 K, since the gas at 1000 K has the capacity of doing more work than the gas at 400 K.

3.6.1 SOME MORE EXAMPLES OF QUALITY DEGRADATION

(A) THROTTLE PROCESS

For frictionless, steady flow process of an ideal gas between sections (1)-(1) and (2)-(2) in a pipe (Fig. 3.5),
Fig. 3.5 Throttled flow through a pipe

\[ h_1 = h_2 \quad (3.27) \]

Again,

\[ T \, ds = dh - v \, dp \quad (3.28) \]

\[ \therefore h_1 = h_2 \quad \text{and} \quad p \, v = R \, T \]

\[ \therefore ds = -\frac{v \, dp}{T} = -\frac{R \, dp}{p} \quad (3.29) \]

or,

\[ s_2 - s_1 = -R \left[ \frac{dp}{p} \right]_1 = -R \left[ \ln p \right]_1 \quad (3.30) \]

or,

\[ s_2 - s_1 = -R \ln \frac{p_2}{p_1} = R \left[ \ln \left( 1 - \frac{\Delta p}{p_1} \right) \right] \quad (3.31) \]

\[ S_{\text{gen}} = m \, R \ln \frac{\Delta p}{p_1} \quad (3.32) \]

\[ \therefore \dot{I} = m \, R \, T_0 \ln \frac{\Delta p}{p_1} \quad (3.33) \]
Equation (3.33) is the expression for the irreversibility or lost work in the system comprising of the sections (1)-(1) and (2)-(2). It is obvious from the expression that quality degrades logarithmically with pressure drop between the sections as well as with the mass flow rate for an initial pressure $p_1$ and surrounding temperature $T_o$.

**(B) FLOW WITH FRICTION**

Consider a steady and adiabatic flow of an ideal gas through the segment of a pipe. Applying the first law of thermodynamics between sections (1)-(1) and (2)-(2) (Fig. 3.6)

![Diagram of adiabatic steady flow of an ideal gas in a straight pipe](image)

Fig. 3.6 Adiabatic steady flow of an ideal gas in a straight pipe

$$h_1 = h_2 \quad \text{(3.34)}$$

Applying $Tds$ relationship

$$Tds = dh - vdp \quad \text{(3.35)}$$

or,

$$ds = \frac{dh}{T} - \frac{v}{T} dp \quad \text{(3.36)}$$

or,

$$\int_1^2 ds = - \int_1^2 \frac{vdp}{T} = - \int_1^2 \frac{R}{p_1} dp \quad \text{(3.37)}$$

$$\therefore S_{gen} = \int_1^2 mds = -mR \int_1^2 \frac{dp}{p_1} = mR \ln \frac{\Delta p}{p_1} \quad \text{(3.38)}$$
where \[ \ln \left[ 1 - \frac{\Delta p}{p_i} \right] = -\frac{\Delta p}{p_i} \] since \( \frac{\Delta p}{p_i} < 1 \) and higher order terms are neglected.

Hence, lost work or irreversibility is
\[
W_{\text{lost}} = m \left[ (h_1 - T_0s_1) - (h_2 - T_0s_2) \right] = mT_0 \left[ s_2 - s_1 \right] = T_0 S_{\text{gen}} = mRT_0 \frac{\Delta p}{p_i}
\]

Hence decrease in exergy is proportional to the pressure drop as well as mass flow rate.

(C) MIXING OF TWO FLUID STREAMS

Two fluid streams 1 and 2 of an incompressible fluid or ideal gas mixing adiabatically at constant pressure as shown in Fig. 3.7.

![Fig. 3.7 Mixing of two fluid streams](image)

Mass balance:
\[
m_1 + m_2 = m_3 = m \quad \text{(say)}
\]

Let,
\[
x = \frac{m_1}{m_1 + m_2}
\]

By first law of thermodynamics,
\[
m_1 h_1 + m_2 h_2 = (m_1 + m_2)h_3
\]

or,
\[
xh_1 + (1 - x)h_2 = h_3
\]
Since,

\[ h = h(T) \]  \tag{3.44}

From Eq. (3.43)

\[ xT_1 + (1-x)T_2 = T_3 \]  \tag{3.45}

\[ \therefore \frac{T_3}{T_1} = x + (1-x)\frac{T_2}{T_1} = x + (1-x)\tau \]  \tag{3.46}

where,

\[ \tau = \frac{T_2}{T_1} \]  \tag{3.47}

By second law of thermodynamics,

\[ S_{\text{gen}} = m_3 s_3 - m_1 s_1 - m_2 s_2 \]  \tag{3.48}

\[ \therefore \frac{S_{\text{gen}}}{m} = (s_3 - s_2) + x(s_2 - s_1) \]  \tag{3.49}

\[ \therefore \frac{S_{\text{gen}}}{m} = C_p \ln \frac{T_3}{T_1} + xC_p \ln \frac{T_2}{T_1} \]  \tag{3.50}

or,

\[ \therefore \frac{S_{\text{gen}}}{mC_p} = \ln \left( \frac{T_3}{T_1} \right) \left( \frac{T_2}{T_1} \right)^x \]  \tag{3.51}

Let, \( N_s \) = Entropy generation number = \( \frac{S_{\text{gen}}}{mC_p} \)
Now, substituting $\frac{T_3}{T_1} = x + (1 - x)\tau$ from Eq. (3.46) in Eq. (3.52), we get

$$N_s = \ln \frac{x + \tau(1 - x)}{\tau^{1-x}}$$

(3.53)

$$\therefore N_s = N_s(x, \tau)$$

(3.54)

For, $x = 1$, $N_s = 0 \Rightarrow m_2 = 0$, the system is with single stream.

For, $\tau = 1$, $N_s = 0$, temperature of both the streams are same.

Rate of exergy loss due to mixing is,

$$W_{\text{lost}} = \dot{I} = T_0 S_{\text{gen}}$$

(3.55)

$$W_{\text{lost}} = T_0 m \cdot C_p \ln \frac{x + \tau(1 - x)}{\tau^{1-x}}$$

(3.56)

### 3.7 EXERGY BALANCE FOR CLOSED AND OPEN SYSTEM

Exergy balance for the closed and open systems can be used to determine the locations, types and magnitude of losses of potential energy resources (fuels) and ways can be found to reduce such losses for making the energy system more efficient.

#### 3.7.1 CLOSED SYSTEM

For a closed system (Fig. 3.8), exergy or availability transfer occurs through heat and work interactions. No mass is transferred across the system boundary.
Ist Law of thermodynamics:

\[ E_2 - E_1 = \frac{1}{T_1} \int^2_1 dQ - W_{1-2} \]  

(3.57)

2\textsuperscript{nd} Law of thermodynamics:

\[ S_2 - S_1 - \frac{1}{T_1} \int^2_1 dQ = S_{\text{gen}} \]  

(3.58)

or,

\[ T_0 (S_2 - S_1) - T_0 \left[ \frac{1}{T_1} \int^2_1 dQ \right] = T_0 S_{\text{gen}} \]  

(3.59)

Subtracting Eq. (3.59) from Eq. (3.57), we get,

\[ E_2 - E_1 - T_0 (S_2 - S_1) = \frac{1}{T_1} \int^2_1 dQ - W_{1-2} - T_0 \left[ \frac{1}{T_1} \int^2_1 dQ \right] - T_0 S_{\text{gen}} \]  

(3.60)

Now, Defining availability function as

\[ A = E + p_0 V - T_0 S \]  

(3.61)

\[ \therefore A_2 - A_1 = E_2 - E_1 + p_0 (V_2 - V_1) - T_0 (S_2 - S_1) \]  

(3.62)
\[
\begin{align*}
&= \int \left[ 1 - \frac{T_0}{T_{\sigma}} \right] dQ - W_{1-2} - T_0 S_{\text{gen}} + p_0 (V_2 - V_1) \\
&= \int \left[ 1 - \frac{T_0}{T_{\sigma}} \right] dQ - \left[ W_{1-2} - p_0 (V_2 - V_1) \right] - T_0 S_{\text{gen}}
\end{align*}
\]

(3.63)

\[
\frac{A_2 - A_1}{\text{Change in exergy}} = \int \left[ 1 - \frac{T_0}{T_{\sigma}} \right] dQ - \left[ W_{1-2} - p_0 (V_2 - V_1) \right] - T_0 S_{\text{gen}}
\]

In the form of rate equation,

\[
\frac{dA}{d\tau} = \sum_j \left[ 1 - \frac{T_0}{T_j} \right] \dot{Q}_j - \left[ W - p_0 \frac{dV}{d\tau} \right] - \dot{I}_{\text{ex}}
\]

(3.64)

where, \( T_j = \) instantaneous temperature at the boundary

\[
\frac{dV}{d\tau} = \text{rate of change of system volume.}
\]

For an isolated system,

\[
A_2 - A_1 = \Delta A = -\dot{I}
\]

(3.65)

Since, \( \dot{I} > 0 \), the only processes allowed by the second law of thermodynamics are those for which the exergy of the isolated system decreases. In other words,

**The exergy of an isolated system can never increase.**

(Counterpart of entropy principle, which states that entropy of an isolated system can never decrease.)
3.7.2 OPEN SYSTEM: EXERGY BALANCE FOR A STEADY FLOW SYSTEM

Consider an open system (Fig. 3.9). Applying the first law of thermodynamics:

\[
H_1 + \frac{mV_1^2}{2} + mgz_1 + Q_{1\rightarrow 2} = H_2 + \frac{mV_2^2}{2} + mgz_2 + W_{1\rightarrow 2}
\]

(3.66)

Second law of thermodynamics:

\[
S_1 + \int_1^2 \left[ \frac{dQ}{T} \right]_\sigma - S_2 = S_{gen}
\]

(3.67)

or,

\[
T_0 \left( S_1 - S_2 \right) + T_0 \int_1^2 \left[ \frac{dQ}{T} \right]_\sigma = T_0 S_{gen} = I
\]

(3.68)

From Eqs. (3.66) and (3.68),

\[
H_2 - H_1 - T_0 \left( S_2 - S_1 \right) + \frac{mV_2^2 - V_1^2}{2} + mg \left( z_2 - z_1 \right) = \int_1^2 \left( 1 - \frac{T_0}{T_\sigma} \right) dQ - W_{1\rightarrow 2} - I
\]

(3.69)

or,

\[
A_2 - A_1 = \int_1^2 \left( 1 - \frac{T_0}{T_\sigma} \right) dQ - W_{1\rightarrow 2} - I
\]

(3.70)
In the form of rate equation at steady state,

\[ \sum_{j} \left[ 1 - \frac{T_j}{T_0} \right] q_j - \dot{W}_{cv} + m \left( a_{j_1} - a_{j_2} \right) - I_{cv} = 0 \]  

(3.71)

where,

\[ a_{j_1} - a_{j_2} = \left( h_1 - h_2 \right) - T_0 \left( s_1 - s_2 \right) + \frac{V_1^2 - V_2^2}{2} + g \left( z_1 - z_2 \right) \]  

(3.72)

For a single stream entering and leaving, the exergy balance is given by

\[ \left[ 1 - \frac{T_0}{T_o} \right] \frac{\dot{Q}}{m} + a_{j_1} - \frac{\dot{W}}{m} - a_{j_2} = \frac{\dot{I}}{m} \]  

(3.73)

### 3.8 TOOLS FOR EXERGY ANALYSIS

The following are the tools for exergy analysis

1. Value/Grassman/Shankey diagram
2. Pinch point technology

In general, conversion of energy or production of chemical substances in a plant requires determination of all the exergy flows that are transferred between distinguished apparatuses or unit operations of the plant. The resulting exergy losses can provide useful information with regard to the overall performance of the plant. However, it is generally difficult to judge the thermodynamic losses without any reference. Evaluation of plant performance will usually require a comparison of the thermodynamic performance of specific apparatuses or unit operations with available data from previously built plants. Thus, exergy efficiencies have appeared to be more useful, for large plants or integrated plants. It bears no significance to components. Exergy efficiencies combined with exergy flow diagrams can provide the thermodynamic performance of even for a complex system.

#### 3.8.1 VALUE DIAGRAM

Consider the heat transfer between two fluids in a recuperative type of heat exchanger. Temperature versus heat flow \((T - \dot{Q})\) between the two fluids has been presented in Fig. 3.10.
Fig. 3.10 Heat transfer process represented in a Q-T diagram

This diagram may be more informative if modified as follows:
Abcissa: indicates the heat transfer between the two streams
Ordinate: temperature $T$ is replaced by the term $\left(1 - \frac{T_0}{T}\right)$. At $T = T_0$, the ordinate is zero and at $T = \alpha$, ordinate value is unity (Fig. 3.11).
If it is assumed that an infinitesimal small amount of heat $dQ$ is transferred from the secondary flow, the flow that is cooled down in the heat exchanger, the resulting decrease in temperature $dT_s$ may be neglected. For the energy of this amount of heat can be written as

$$dEx_s = \left(1 - \frac{T_0}{T}\right) dQ \tag{3.74}$$

In the value diagram (Fig. 3.11) the area 1-3-4-6-1 equals the amount of heat $dQ$, whereas the area 1-2'-5'-6-1 equals the exergy of this amount of heat. The term $\left(1 - \frac{T_0}{T}\right)$ indicates which part of the considered heat can be converted into work and can in principle be converted into work and can be seen as the exergy fraction of this amount of heat. The total exergy from the secondary flow van be determined by integrating Eq.(3.74) from the inlet temperature $T_{s,in}$ to the outlet temperature $T_{s,out}$

$$Ex_s = \int_{T_{s,in}}^{T_{s,out}} \left(1 - \frac{T_0}{T}\right) dQ \tag{3.75}$$
This amount of exergy equals the whole area below the temperature curve in the value diagram. Within the heat exchanger the heat $dQ$ is transferred to the primary flow. The exergy of the heat supplied to this flow is
\[ dEx = \left(1 - \frac{T_o}{T_s}\right) dQ \tag{3.76} \]

In the value diagram, this exergy is represented by the area 1-2-5-6-1. The area 2-2'-5'-5-2 indicates the exergy lost due to temperature difference necessary to transfer heat from the secondary to the primary flow. The total exergy absorbed by the primary flow is
\[ Ex_p = \int_{T_{p,m}}^{T_{p,pu}} \left(1 - \frac{T_o}{T_p}\right) dQ \tag{3.77} \]

where subscript $p$ indicates the primary flow. The absorbed exergy by the primary flow equals the area below the temperature curve for this flow in the value diagram. The total exergy loss due to heat transfer, $Ex_s - Ex_p$, is represented by the area between the two temperature curves.

Thus, the amount of exergy as well as the exergy loss can be easily presented with the help of value diagram.

### 3.8.2 PROCESS INTEGRATION: PINCH POINT TECHNOLOGY

- A method of using the overall thermal balance of a process
- Theoretically predicts minimum energy consumption
- Predicts the construction costs of the plant using a heat recovery system.
- First put into practical use in 1984, at Linhoff March Co.

Pinch point technology, or process integration, is the name given for a technique developed by Prof. Linnhof and co-workers (1978) at Leeds university, UK to optimize the heat recovery in large complex plants with several hot and cold streams of fluids. To illustrate the basic principle take a case of a plant with two hot and two cold streams, as shown in Table 3.1.
Table 3.1 Data for 4 (four) fluid streams

<table>
<thead>
<tr>
<th>Stream No</th>
<th>Initial Temp (°C)</th>
<th>Final Temp (°C)</th>
<th>Mass flow rate (kg/s)</th>
<th>Sp. Heat capacity (kJ/kgK)</th>
<th>Heat capacity rate (kW/K)</th>
<th>Rate of enthalpy increase (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>205</td>
<td>65</td>
<td>2.00</td>
<td>1.00</td>
<td>2.0</td>
<td>-280</td>
</tr>
<tr>
<td>2</td>
<td>175</td>
<td>75</td>
<td>3.20</td>
<td>1.25</td>
<td>4.0</td>
<td>-400</td>
</tr>
<tr>
<td>3</td>
<td>45</td>
<td>180</td>
<td>3.75</td>
<td>0.80</td>
<td>3.0</td>
<td>+405</td>
</tr>
<tr>
<td>4</td>
<td>105</td>
<td>155</td>
<td>3.00</td>
<td>1.50</td>
<td>4.5</td>
<td>+225</td>
</tr>
</tbody>
</table>

The hot streams can be combined into an equivalent composite stream as follows: From Table 3.1, it is clear that both stream 1 and 2 are having common temperature drop between 175 to 175°C. For the common processes, we go for process integration by considering composite thermal capacity. The hot composite curve will consists of (1) stream 1 from 205 to 175°C with heat capacity 2 kW/K, (2) between temperature 175 to 75°C, a combined stream of thermal capacity rate (2+4)=6 kW/K (3) between temperature 75 to 65°C, stream 1 with heat capacity rate 2 kW/K. Similarly, the cold streams can be combined. The two composite streams are then plotted on a temperature-heat load graph as shown in Fig. 3.12. The pinch point is defined as the point where temperature difference between the two composite curves is minimum.

![Combined hot and cold streams diagram](image)

Fig. 3.12 Temperature verses rate of enthalpy change for composite hot and cold streams

The temperature difference at the pinch point depends on the design of the heat exchanger. Smaller the temperature difference, the more expensive is the heat.
exchanger. A high value of pinch point indicates high thermal losses due to external irreversibility.

Taking as an example a temperature difference of 20K, say, then the cold stream (composite) can be moved from left to right on the diagram horizontally on the diagram, keeping the hot composite curve fixed, until the temperature difference at pinch point is 20K. It is then seen then the external heating load of 90 kW and external cooling load of 140 kW are required for the system, all other energy changes can be achieved by the heat exchangers between the various streams. Required heating and cooling of each stream above and below the pinch point is shown in Table 3.2.

<table>
<thead>
<tr>
<th>Stream</th>
<th>Above Pinch Point</th>
<th>Below Pinch Point</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Cooling: $2(205-125)=160kW$</td>
<td>Cooling: $2(125-65)=120kW$</td>
</tr>
<tr>
<td>2</td>
<td>Cooling: $4(175-125)=200kW$</td>
<td>Cooling: $4(125-75)=200kW$</td>
</tr>
<tr>
<td>3</td>
<td>Heating: $3(180-105)=225kW$</td>
<td>Heating: $3(105-45)=180kW$</td>
</tr>
<tr>
<td>4</td>
<td>Heating: $4.5(155-105)=225kW$</td>
<td>Zero</td>
</tr>
<tr>
<td>Net external requirement</td>
<td>90 kW (Heating)</td>
<td>140 kW (cooling)</td>
</tr>
</tbody>
</table>

From Table 3.2, we can infer that for a pinch point of 20K, we require external heating load of 90kW and cooling load of 140kW above and below the pinch point respectively. The process integration above the pinch point is now as follows:

Above pinch point, stream 1 and stream 3 exchange 160 kW heat, stream 2 and stream 4 exchange 200 kW. Below pinch point, stream 2 and 3 exchanges 180 kW heat. Stream 2 and 4 are to be externally heated with 65 kW and 25 kW respectively to meet the deficit/demand of 90 kW. Similarly, stream 1 and 3 are to be cooled externally with 120 kW and 20 kW heat exchangers respectively to meet the demand of 140 kW. Possible processes are shown in Fig. 3.13.
Fig. 3.13 Possible plant to heat and cool four fluid streams for a minimum 20 K temperature difference

The following rules should be followed in process integration:

1. Do not transfer heat from one fluid to another across the pinch point
2. No external heating below pinch point
3. No external cooling above the pinch point
4. A heat exchanger should operate on one side of the pinch, either taking a heat supply from below the pinch, or rejecting heat to a fluid above the pinch
5. A heat pump should operate across the pinch from a cold stream below the pinch to a hot stream above the pinch.

Summary:

Exergy is the maximum work potential of a system
Exergy transfer with heat, work and mass
For an isolated system exergy always decreases
Exergy remains constant in a reversible process
Anything that generate entropy is responsible for decrease of exergy