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ENERGY RECOVERY IN DRYING

BY ADSORPTION HEAT PUMPING.

by

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Submitted in partial fulfilment of the requirements for the Degree of Doctor of Philosophy of Loughborough University of Technology.

1982.

Supervisor and Director of Research:
Professor D. C. Freshwater.

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PAGE NUMBERS CUT OFF IN ORIGINAL
DEDICATION

Dedicated to the memory of Dad, a man of great vision, whose life remains a burning inspiration and to Mum, to whom I owe so much.
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<tr>
<td>A</td>
<td>Cross sectional area, drying area</td>
<td>$(m^2)$</td>
</tr>
<tr>
<td>a</td>
<td>Specific surface area</td>
<td>$(m^2/m^3)$</td>
</tr>
<tr>
<td>$C_h$</td>
<td>Humid heat capacity of air</td>
<td>$(kJ/(kg\cdot°C))$</td>
</tr>
<tr>
<td>$C_{max}$</td>
<td>Maximum humid heat capacity of air</td>
<td>$(kJ/(kg\cdot°C))$</td>
</tr>
<tr>
<td>$C_{min}$</td>
<td>Minimum humid heat capacity of air</td>
<td>$(kJ/(kg\cdot°C))$</td>
</tr>
<tr>
<td>$C_{pg}$</td>
<td>Specific heat of air</td>
<td>$(kJ/(kg\cdot°C))$</td>
</tr>
<tr>
<td>$C_{pv}$</td>
<td>Specific heat of water vapour</td>
<td>$(kJ/(kg\cdot°C))$</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of performance</td>
<td></td>
</tr>
<tr>
<td>COP$_{max}$</td>
<td>Ideal coefficient of performance</td>
<td></td>
</tr>
<tr>
<td>D$_{AB}$</td>
<td>Gas phase diffusivity</td>
<td>$(m^2/hr)$</td>
</tr>
<tr>
<td>D$_L$</td>
<td>Liquid phase diffusivity</td>
<td>$(m^2/hr)$</td>
</tr>
<tr>
<td>DPF</td>
<td>Dryer performance factor</td>
<td>$(kg^2/(kJ\cdot hr))$</td>
</tr>
<tr>
<td>$E_{comp}$</td>
<td>Compression energy</td>
<td>$(kJ)$</td>
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<tr>
<td>$E_P$</td>
<td>Primary energy</td>
<td>$(kJ)$</td>
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<tr>
<td>$E_{rec}$</td>
<td>Energy recovery in condenser</td>
<td>$(kJ)$</td>
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<tr>
<td>$E_{vap}$</td>
<td>Energy for vaporisation of moisture</td>
<td>$(kJ)$</td>
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<tr>
<td>$E_x$</td>
<td>Exergy</td>
<td>$(kJ)$</td>
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<tr>
<td>f</td>
<td>Fractional saturation of air in dryer</td>
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<td>f$_d$</td>
<td>Relative drying rate, $N/N_c$</td>
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<tr>
<td>f$_z$</td>
<td>Fraction of MTZ unused.</td>
<td></td>
</tr>
<tr>
<td>G</td>
<td>Air flowrate</td>
<td>$(kg/hr)$</td>
</tr>
<tr>
<td>$G_v$</td>
<td>Air mass flow velocity</td>
<td>$(kg/(m^2\cdot hr))$</td>
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</table>
HTU  Heat transfer units.

\( h_{1000} \)  Air humidity \( (\frac{\text{lbs}}{1000\text{ft}^3}) \)

\( h_c \)  Convective heat transfer coefficient \( (\frac{\text{KJ}}{\text{m}^2\cdot\text{hr} \cdot \circ \text{C}}) \)

\( h_r \)  Radiative heat transfer coefficient \( (\frac{\text{KJ}}{\text{m}^2\cdot\text{hr} \cdot \circ \text{C}}) \)

\( h_t \)  Total heat transfer coefficient \( (\frac{\text{KJ}}{\text{m} \cdot \text{hr} \cdot \circ \text{C}}) \)

\( j_D \)  Colburn factor.

\( k \)  Thermal conductivity \( (\frac{\text{KJ}}{\text{m} \cdot \text{hr} \cdot \circ \text{C}}) \)

\( k_g \)  Mass transfer coefficient \( (\frac{\text{m}}{\text{hr}}) \)

\( K \)  Breakpoint capacity correlation constant

\( K_T \)  Trouton's ratio \( (\frac{\text{KJ}}{\text{kg} \cdot \circ \text{C}}) \)

\( K_z \)  Moisture distribution ratio.

\( L \)  Length of continuous dryer \( (\text{m}) \)

\( L_T \)  Length of a transfer unit \( (\text{m}) \)

\( LUB \)  Length of unused bed \( (\text{m}) \)

\( l \)  Slab thickness \( (\text{m}) \)

\( M_s \)  Mass of solid adsorbent \( (\text{kg}) \)

\( M_Z \)  Adsorption bed height modulus

\( \text{MER} \)  Specific moisture extraction rate \( (\frac{\text{kg}}{\text{KJ}}) \)

\( \text{MTU} \)  Mass transfer units.

\( N \)  Drying rate \( (\frac{\text{kg}}{\text{m}^2 \cdot \text{hr}}) \)

\( N_c \)  Constant drying rate \( (\frac{\text{kg}}{\text{m}^2 \cdot \text{hr}}) \)

\( N_{\text{Reb}} \)  Reynold's number

\( N_r \)  Number of transfer units

\( n \)  Breakpoint capacity correlation constant.
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<th>Description</th>
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<tr>
<td>p</td>
<td>Partial pressure</td>
<td>mm Hg</td>
</tr>
<tr>
<td>P&lt;sub&gt;o&lt;/sub&gt;</td>
<td>Vapour pressure</td>
<td>mm Hg</td>
</tr>
<tr>
<td>P&lt;sub&gt;ob&lt;/sub&gt;</td>
<td>Saturation vapour pressure at bed temperature</td>
<td>mm Hg</td>
</tr>
<tr>
<td>P&lt;sub&gt;s&lt;/sub&gt;</td>
<td>Surface vapour pressure</td>
<td>mm Hg</td>
</tr>
<tr>
<td>P&lt;sub&gt;sat&lt;/sub&gt;</td>
<td>Saturation partial pressure</td>
<td>mm Hg</td>
</tr>
<tr>
<td>P&lt;sub&gt;w&lt;/sub&gt;</td>
<td>Wet-bulb partial pressure</td>
<td>mm Hg</td>
</tr>
<tr>
<td>Q</td>
<td>Heat input</td>
<td>KJ</td>
</tr>
<tr>
<td>Q&lt;sub&gt;abs&lt;/sub&gt;</td>
<td>Absorber heat</td>
<td>KJ</td>
</tr>
<tr>
<td>Q&lt;sub&gt;cond&lt;/sub&gt;</td>
<td>Condenser heat</td>
<td>KJ</td>
</tr>
<tr>
<td>Q&lt;sub&gt;reg&lt;/sub&gt;</td>
<td>Regeneration heat</td>
<td>KJ</td>
</tr>
<tr>
<td>Q&lt;sub&gt;T&lt;/sub&gt;</td>
<td>Total heat input</td>
<td>KJ</td>
</tr>
<tr>
<td>R</td>
<td>Universal gas constant</td>
<td>KJ / kg mole⁻¹ K⁻¹</td>
</tr>
<tr>
<td>r</td>
<td>Recycle ratio to dryer</td>
<td></td>
</tr>
<tr>
<td>r&lt;sub&gt;A&lt;/sub&gt;</td>
<td>Rate of adsorption</td>
<td>kg m⁻² hr⁻¹</td>
</tr>
<tr>
<td>r&lt;sub&gt;D&lt;/sub&gt;</td>
<td>Dryer recycle ratio in AHP</td>
<td></td>
</tr>
<tr>
<td>S</td>
<td>Entropy</td>
<td>J / °C</td>
</tr>
<tr>
<td>s</td>
<td>Approach to saturation</td>
<td>%</td>
</tr>
<tr>
<td>SER</td>
<td>Specific energy requirement</td>
<td>KJ / kg</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
<td>°C</td>
</tr>
<tr>
<td>T&lt;sub&gt;b&lt;/sub&gt;</td>
<td>Boiling temperature</td>
<td>°C</td>
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<tr>
<td>T&lt;sub&gt;c&lt;/sub&gt;</td>
<td>Critical temperature</td>
<td>°C</td>
</tr>
<tr>
<td>T&lt;sub&gt;f&lt;/sub&gt;</td>
<td>Feed temperature</td>
<td>°C</td>
</tr>
<tr>
<td>T&lt;sub&gt;r&lt;/sub&gt;</td>
<td>Reduced temperature</td>
<td></td>
</tr>
<tr>
<td>T&lt;sub&gt;s&lt;/sub&gt;</td>
<td>Drying surface temperature</td>
<td>°C</td>
</tr>
<tr>
<td>T&lt;sub&gt;sb&lt;/sub&gt;</td>
<td>Temperature of adsorption bed at breakpoint</td>
<td>°C</td>
</tr>
<tr>
<td>T&lt;sub&gt;sd&lt;/sub&gt;</td>
<td>Temperature of desorption bed</td>
<td>°C</td>
</tr>
</tbody>
</table>
\( T_{SI} \) Initial temperature of adsorption bed \((^\circ C)\)

\( T_{Sat} \) Saturation temperature \((^\circ C)\)

\( T_{W} \) Wet-bulb temperature \((^\circ C)\)

\( U \) Overall heat transfer coefficient \(\frac{KJ}{m\cdot hr\cdot ^\circ C}\)

\( V_{G} \) Gas volume \((m^3)\)

\( V_{L} \) Liquid volume \((m^3)\)

\( v \) Gas flow velocity \(\frac{m}{hr}\)

\( W \) Adsorbate loading on adsorbent \(\frac{kg}{hr}\)

\( W_{b} \) Adsorbent breakpoint capacity \(\frac{kg}{hr}\)

\( W_{ba} \) Adiabatic breakpoint capacity \(\frac{kg}{kg}\)

\( W_{bi} \) Isothermal breakpoint capacity \(\frac{kg}{kg}\)

\( W_{eD} \) Initial capacity of desorption bed \(\frac{kg}{kg}\)

\( W_{i} \) Initial capacity of adsorption bed \(\frac{kg}{kg}\)

\( X \) Moisture content of drying solids \(\frac{kg}{kg}\)

\( X_{b} \) Relative humidity of air

\( X_{bD} \) Adsorption bed relative humidity at breakpoint

\( X_{C} \) Critical moisture content \(\frac{kg}{kg}\)

\( X_{e} \) Equilibrium moisture content \(\frac{kg}{kg}\)

\( X_{eD} \) Equilibrium bed relative humidity

\( X_{eD} \) Desorption bed equilibrium relative humidity

\( X_{f} \) Modified relative humidity, \(X\exp(\frac{\pi}{500})\)

\( x \) Distance inside a slab (m)
\[ y \quad \text{Air humidity} \quad \left( \frac{kg}{kg} \right) \]

\[ y_{sat} \quad \text{Saturation air humidity} \quad \left( \frac{kg}{kg} \right) \]

\[ y_w \quad \text{Wet-bulb air humidity} \quad \left( \frac{kg}{kg} \right) \]

\[ Z \quad \text{Height of adsorbent bed} \quad (m) \]

\[ Z_s \quad \text{Thickness of slab} \quad (m) \]

**General Subscripts**

Conventional Dryer:

0 \quad \text{Ambient conditions}

1 \quad \text{Dryer inlet}

2 \quad \text{Dryer outlet}

Experimental Adsorption Column:

i \quad \text{inlet conditions}

e \quad \text{exit conditions}

Adsorption Heat Pump:

0 \quad \text{Ambient conditions}

Di \quad \text{Dryer inlet}

De \quad \text{Dryer exit}

Ai \quad \text{Adsorber inlet}

Ae \quad \text{Adsorber exit}

Ri \quad \text{Desorber inlet}

Re \quad \text{Desorber exit.}
### Greek Letters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>( a )</td>
<td>Dimensionless time constant</td>
</tr>
<tr>
<td>( \beta )</td>
<td>Humid Heat ratio, ( \frac{C_{ho}}{C_{h2}} )</td>
</tr>
<tr>
<td>( \Delta T )</td>
<td>Temperature lift ( ({}^\circ C) )</td>
</tr>
<tr>
<td>( \Delta y )</td>
<td>Humidity change ( (kg) )</td>
</tr>
<tr>
<td>( \delta )</td>
<td>Polar parameter</td>
</tr>
<tr>
<td>( \varepsilon )</td>
<td>Bed void fraction</td>
</tr>
<tr>
<td>( \varepsilon_r )</td>
<td>Thermal effectiveness</td>
</tr>
<tr>
<td>( \lambda )</td>
<td>Latent heat ( (\frac{kJ}{kg}) )</td>
</tr>
<tr>
<td>( \lambda_A )</td>
<td>Heat of adsorption ( (\frac{kJ}{kg}) )</td>
</tr>
<tr>
<td>( \lambda_f )</td>
<td>Latent heat at feed temperatures ( (\frac{kJ}{kg}) )</td>
</tr>
<tr>
<td>( \lambda_m )</td>
<td>Molar latent heat ( (\frac{kJ}{kgmol}) )</td>
</tr>
<tr>
<td>( \lambda_w )</td>
<td>Latent heat at wet-bulb ( (\frac{kJ}{kg}) )</td>
</tr>
<tr>
<td>( \eta )</td>
<td>Dryer thermal efficiency ( (%) )</td>
</tr>
<tr>
<td>( \eta_{conv} )</td>
<td>Thermal efficiency of conventional dryer ( (%) )</td>
</tr>
<tr>
<td>( \eta_{EM} )</td>
<td>Efficiency of AHP predicted from Equilibrium Model ( (%) )</td>
</tr>
<tr>
<td>( \eta_{ideal} )</td>
<td>Ideal instantaneous efficiency of dryer ( (%) )</td>
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<tr>
<td>( \eta_R )</td>
<td>Efficiency of combined dryer/heat exchanger ( (%) )</td>
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<tr>
<td>( \eta_{RH} )</td>
<td>Efficiency of AHP predicted from BRH Model ( (%) )</td>
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<tr>
<td>( \eta_T )</td>
<td>Thermodynamic efficiency ( (%) )</td>
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<tr>
<td>( \phi )</td>
<td>Characteristic moisture content</td>
</tr>
<tr>
<td>( \Theta )</td>
<td>Drying time ( (hrs) )</td>
</tr>
<tr>
<td>( \Theta_b )</td>
<td>Breakpoint time ( (hrs) )</td>
</tr>
<tr>
<td>( \Theta_{ba} )</td>
<td>Adiabatic breakpoint time ( (hrs) )</td>
</tr>
<tr>
<td>( \Theta_{bi} )</td>
<td>Isothermal breakpoint time ( (hrs) )</td>
</tr>
</tbody>
</table>
\( \Theta_c \)  Constant drying time (hrs)

\( \Theta_d \)  Dead time (hrs)

\( \Theta_e \)  Equilibrium time (hrs)

\( \Theta_f \)  Falling rate drying time (hrs)

\( \Theta_T \)  Total residence time (hrs)

\( \rho_g \)  Air density (kg/m³)

\( \rho_s \)  Solids density (kg/m³)

\( \mu \)  Fluid viscosity (kg/m·hr)

\( \Pi \)  Total pressure (atmosphere)

\( \Pi_H \)  Dimensionless heat transfer potential

\( \Pi_M \)  Dimensionless mass transfer potential

\( \Omega \)  Collision integral

\( \sigma \)  Surface tension (N/m)

\( \sigma \)  Molecular diameter (Å)

\( \psi \)  Particle shape factor
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ABSTRACT

Drying is one of the most energy intensive operations of the chemical industry and accounts for about 6% of the total energy used by U.K. industry and about 2.5% of the overall energy demand of the U.K. Measurements taken on a typical industrial tumble dryer confirmed that generally convective dryers operate at thermal efficiencies less than 40% and that over 50% of the energy input is lost as sensible heat content of the moist exhaust air. Any significant improvement in the thermal performance of dryers would therefore require a means of gainfully recovering the heat lost in the exhaust air.

Some conventional heat recovery methods have been considered. Recirculation of exhaust air was shown to improve thermal efficiency but at the cost of reduced drying rates. Heat recovery by heat exchange was found unattractive because very large heat exchange surface areas would be required. A new type of adsorption heat pump (AHP) which may be operated either as a temperature swing cycle (TSC) or a pressure swing cycle (PSC), has therefore been proposed here for heat recovery from dryers.

An experimental rig was built and used to investigate the adsorption of moisture on silica-gel in a 1.5 m. high, 0.25m. diameter column under conditions that simulate an industrial dryer, including high temperatures and humidities. Correlation of the experimental data led to the derivation of a polynomial function, similar to the system equilibrium equation, which relates the breakpoint capacity of the adsorbent to the breakpoint bed relative humidity. This function was used to develop a new theoretical model for predicting the performance of the proposed heat pump dryer. The predictions of this model enabled similar predictions obtained from two other models synthesized from various proposals put forward by other investigators previously to be tested against experimental results. This new model was found to be the most appropriate for the conditions encountered and was therefore considered to be the most suitable for predicting the performance of the adsorption heat pump drying system.
Theoretical predictions based on this model indicate that the heat pump dryer may be attractive for low temperature (<100°C) drying. Under high temperature drying conditions low thermal efficiencies and unfavourably large adsorption bed size requirements make the heat pump unattractive.
CHAPTER ONE

INTRODUCTION AND GENERAL BACKGROUND
1. INTRODUCTION AND GENERAL BACKGROUND

1.1. Background

1.1.1. The Energy Problem

The last decade witnessed two successive major global energy crises:

1. 1973/74 crisis, following the Middle-East war;
2. 1979 crisis, arising from the political upheaval and the subsequent revolution in Iran.

These two successive crises have served to show how dependent the fortunes of the industrial society are on the future of world energy supplies. It is hard to overstate the importance of energy to the sustenance of society. Most facets of human activity depend directly or indirectly on energy and this dependence becomes greater as machines are used more and more to replace animal power and human power.

The energy crises can be traced to a number of factors which include:

1. Excessive reliance on fossil fuels, especially oil and gas, as major sources of primary energy.
2. The sudden escalation of fuel prices in the aftermath of the 1973 Middle-East war.
3. Gross mismatch in terms of geographical location, of primary energy availability and its demand.
4. The finite availability of fossil fuels, especially oil.

1.1.2. Input Forms

Primary Energy: The final form in which energy is required for use is often not the form in which it is available naturally. Therefore, it often has to be converted from one form to another in order to be used. For instance, to obtain energy for househeating, oil or coal may be fired in a boiler. Electricity for lighting may be generated from a nuclear reactor. Each of these clearly involves transformation of a primary energy input, be it coal or nuclear fuel, into a convenient form in which it is required. The major primary energy
inputs are essentially oil, natural gas, coal, nuclear fuel and hydro-power.

Fossil Fuels: Fossil fuel refers to primary energy sources which are formed by geo-degradation of biological material in the earth's crust over the years. Conventionally, this includes oil, natural gas and coal.

1.1.3. Trends

Tables 1-1 to 1-3 show the total and primary energy inputs for the whole world, U.S.A. and Western Europe respectively over the period 1965-1980. Table 1-4 shows the same data for U.K. between 1950 and 1980. The statistics show that there was a consistent upward trend in total energy consumption which subsequently levelled-off during the last decade as a result of the energy crisis. Evidently oil is the most important primary energy input. Its proportion in the energy mix has increased considerably while that of coal has shown a steady decline. The proportion of natural gas has also increased consistently. Overall it is a reflection of the dominance of oil and gas in the energy mix that together they account for over 60% of world energy consumption. The trends are more clearly illustrated in figs. 1-1 and 1-2.

Obviously, between 1950 and 1970 there was a clear switch from coal to oil in the U.K; as the proportion of oil continued to rise at the expense of coal. Following the 1973/74 crisis, the proportion of oil started to fall and the slack was taken up more by natural gas than coal which more or less remained steady.

Several reasons have been adduced to explain this trend, including cheapness, less environmental problems associated with extraction and use, ease of handling and transportation (especially oil) and ease of conversion to heat and electricity. Whatever the reason, the significance is that any adverse development in the supply or price of oil and gas is immediately transmitted, with very considerable impact, to the overall energy situation, as was the case during most of the last decade.
### TABLE 1-1: WORLD PRIMARY ENERGY CONSUMPTION
(1965-1980)

<table>
<thead>
<tr>
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<td>TONNES</td>
<td>TONNES</td>
<td>TONNES</td>
</tr>
<tr>
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<td>%</td>
<td>OIL EQ. x10^9</td>
<td>%</td>
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<tr>
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<td>1.53</td>
<td>39.0</td>
<td>2.28</td>
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<td>0.96</td>
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<tr>
<td>COAL</td>
<td>1.50</td>
<td>38.0</td>
<td>1.62</td>
<td>31.0</td>
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<tr>
<td>NUCLEAR</td>
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<td>6.5</td>
<td>0.02</td>
<td>0.5</td>
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<tr>
<td>HYDRO</td>
<td>0.31</td>
<td>6.0</td>
<td>0.36</td>
<td>6.0</td>
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<tr>
<td>TOTAL</td>
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<td>100</td>
<td>5.19</td>
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### TABLE 1-2: U.S.A. PRIMARY ENERGY CONSUMPTION
(1965-1980)

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<tr>
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<td>OIL EQ. x10^9</td>
<td>%</td>
<td>OIL EQ. x10^9</td>
<td>%</td>
</tr>
<tr>
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<td>COAL</td>
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<td>23.1</td>
<td>0.33</td>
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<tr>
<td>NUCLEAR</td>
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<td>3.8</td>
<td>0.07</td>
<td>4.2</td>
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<tr>
<td>HYDRO</td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TOTAL</td>
<td>1.34</td>
<td>100</td>
<td>1.66</td>
<td>100</td>
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</table>
TABLE 1-3 : WEST EUROPEAN PRIMARY ENERGY CONSUMPTION  
(1965-1980)

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<tr>
<td></td>
<td>OIL EQ. x 10^9</td>
<td>%</td>
<td>OIL EQ. x 10^9</td>
<td>%</td>
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<td>0.63</td>
<td>58.9</td>
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<tr>
<td>GAS</td>
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<td>2.4</td>
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<tr>
<td>TOTAL</td>
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<tr>
<td>1980</td>
<td>0.7</td>
<td>0.6</td>
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And projections 1975-2000 (HILLION TONNES OF FUEL EQUIVALENT)

Trends in U.K. Energy Consumption 1950-80
Projections of U.K. energy demand up to the end of this century, adapted from Energy Paper No. 11 published by U.K. Department of Energy, are presented in Table 1-5 and indicated in figs. 1-1 and 1-2 by dashed strokes. The main assumptions of the forecasts are:

1. A low economic growth rate of 1 - 2% in U.K. up to the end of the century.
2. Energy prices would rise more slowly during the rest of the century than in the last decade (see section 1.1.4).
3. Increased implementation of energy saving measures would yield about 15% reduction in energy consumption below what it would have otherwise reached.

Clearly, the highlight of these forecasts is a continued rise in the total energy consumption, though slower than was experienced in the 1950's and 1960's. There would be a switch to coal and nuclear power at the expense of oil and gas.

1.1.4 Costs

For a long time energy costs remained very low. The relevant U.K. energy price indices up to 1977, corrected for inflation, originally presented by Kantyka and extended by Grant, are reproduced in fig. 1-3. The graph clearly shows the sharp rise in fuel prices between 1973 and 1974 relative to other prices. Similar data for U.S.A. and the world, indicating both actual transaction and inflation corrected real prices, are also presented in Table 1-6.

While energy remained cheap the evolving industrial society fashioned itself accordingly, taking advantage of the situation, with more energy driven machinery and consumer goods being introduced. It is no surprise therefore that the sudden escalation of fuel prices came as a severe shock. More than anything else it revealed the inherent dangers associated with the excessive reliance on petroleum resources for energy and the need to develop means of reducing energy consumption without possibly reducing industrial production.
Fig. 1-3. U.K. Price Indices, Including Fuels (Inflation Corrected) 1960-76.
<table>
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<th>YEAR</th>
<th>CURRENT U.S. PRICE (Dollars/bbl)</th>
<th>U.S. CONSUMER PRICE INDEX (1957-1959=100)</th>
<th>REAL U.S. price in 1957-59 (Dollars/bbl)</th>
<th>WORLD PRICE estimated actual transactions price</th>
<th>REAL WORLD PRICE 1957-59 (Dollars/bbl)</th>
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<tr>
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<td>1890</td>
<td>.77</td>
<td>32</td>
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<td>150</td>
<td>2.26</td>
<td>1.84</td>
<td>1.23</td>
</tr>
<tr>
<td>1973</td>
<td>3.89</td>
<td>154</td>
<td>2.53</td>
<td>2.91</td>
<td>1.89</td>
</tr>
<tr>
<td>1974</td>
<td>6.74</td>
<td>175</td>
<td>3.85</td>
<td>10.77</td>
<td>6.15</td>
</tr>
<tr>
<td>1975</td>
<td>7.67</td>
<td>187</td>
<td>4.10</td>
<td>10.72</td>
<td>5.73</td>
</tr>
<tr>
<td>1976</td>
<td>8.11</td>
<td>202</td>
<td>4.01</td>
<td>11.51</td>
<td>5.70</td>
</tr>
<tr>
<td>1977</td>
<td>8.22</td>
<td>211</td>
<td>3.90</td>
<td>13.12</td>
<td>6.22</td>
</tr>
</tbody>
</table>
1.2 Energy Use Pattern

1.2.1 Overall Consumption

A breakdown of the energy consumption in 1980 by all U.K. end users, in terms of sectors, is presented in Table 1-7 below.

**TABLE 1-7: SECTORAL ENERGY CONSUMPTION IN U.K. (1980)**

<table>
<thead>
<tr>
<th>SECTOR</th>
<th>Energy Consumption</th>
<th>Therms x10^6</th>
<th>KJx10^12</th>
<th>%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Industry</td>
<td></td>
<td>19,173</td>
<td>2,023</td>
<td>33.9</td>
</tr>
<tr>
<td>Transport</td>
<td></td>
<td>14,103</td>
<td>1,488</td>
<td>25.0</td>
</tr>
<tr>
<td>Domestic</td>
<td></td>
<td>15,784</td>
<td>1,665</td>
<td>27.9</td>
</tr>
<tr>
<td>Public Administration</td>
<td></td>
<td>3,545</td>
<td>374</td>
<td>6.3</td>
</tr>
<tr>
<td>Agriculture</td>
<td></td>
<td>594</td>
<td>63</td>
<td>1.0</td>
</tr>
<tr>
<td>Others</td>
<td></td>
<td>3,342</td>
<td>353</td>
<td>5.9</td>
</tr>
<tr>
<td><strong>TOTAL</strong></td>
<td></td>
<td><strong>56,547</strong></td>
<td><strong>5,966</strong></td>
<td><strong>100.0</strong></td>
</tr>
</tbody>
</table>

**TABLE 1-8: BREAKDOWN OF U.K. INDUSTRIAL ENERGY CONSUMPTION (1980)**

<table>
<thead>
<tr>
<th>GROUP</th>
<th>Energy Consumption</th>
<th>Therms x10^6</th>
<th>KJx10^12</th>
<th>%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Iron and Steel</td>
<td></td>
<td>2,867</td>
<td>302.5</td>
<td>15.0</td>
</tr>
<tr>
<td>Engineering, Metals</td>
<td></td>
<td>3,738</td>
<td>394.4</td>
<td>19.5</td>
</tr>
<tr>
<td>Food, Drinks, Tobacco</td>
<td></td>
<td>1,875</td>
<td>197.8</td>
<td>9.8</td>
</tr>
<tr>
<td>Chemicals and Allied</td>
<td></td>
<td>4,140</td>
<td>436.8</td>
<td>21.6</td>
</tr>
<tr>
<td>Textiles, Leather, Clothing</td>
<td></td>
<td>940</td>
<td>99.2</td>
<td>4.9</td>
</tr>
<tr>
<td>Paper, Printing, Stationery</td>
<td></td>
<td>1,102</td>
<td>116.3</td>
<td>5.7</td>
</tr>
<tr>
<td>Bricks, Tiles, Fireclay &amp; other Building Materials</td>
<td></td>
<td>454</td>
<td>47.9</td>
<td>2.4</td>
</tr>
<tr>
<td>Ceramics, Glass</td>
<td></td>
<td>634</td>
<td>66.9</td>
<td>3.3</td>
</tr>
<tr>
<td>Cement</td>
<td></td>
<td>787</td>
<td>83.0</td>
<td>4.1</td>
</tr>
<tr>
<td>Other</td>
<td></td>
<td>2,636</td>
<td>278.1</td>
<td>13.7</td>
</tr>
<tr>
<td><strong>TOTAL</strong></td>
<td></td>
<td><strong>19,173</strong></td>
<td><strong>2,022.9</strong></td>
<td><strong>100.0</strong></td>
</tr>
</tbody>
</table>
Clearly, industry, which accounts for about 34% of total demand, is the single most important energy user. A further breakdown of energy usage within industry, (Table1-8), shows that among the different industrial groups the Chemical and Allied Products Industry is the most prominent user, accounting for 21.6% of industrial consumption.

1.2.2. Energy Use in the Chemical Industry

The chemical industry energy consumption consists essentially of two components:

(i) fuel, power and others;

(ii) feedstocks.

The fuel demand comprises the primary energy inputs - coal, oil and gas - used essentially to provide process heating, including generation of steam, while the power demand is mainly in form of purchased electricity, which is used to provide motive power for driving mechanical equipment such as compressors, pumps, grinders/crushers, conveyors and agitators/mixers. There is also an extra demand for energy, mainly in form of electricity, for other purposes such as lighting and electrolysis which are strictly neither process heating nor motive power. In addition to purchased electricity the chemical industry generates significant quantities of electricity and power internally. The primary energy for this comes mainly from heat generated in the combustion of primary fuels. The feedstock component of the energy requirement consists of naphtha, LPG and natural gas which serve as basic raw materials for various petrochemical processes.

The U.K. Department of Energy\textsuperscript{2} statistics on overall and industry sector energy consumption, reproduced in Tables 1-7 and 1-8, do not account for the primary energy used to generate purchased electricity. Besides, they include only a fraction of the chemical industry feedstock energy input. Alternative data from Conseil Européen des Fédérations de l'Industrie Chimique (CEFIC) sources give the following breakdown of U.K. chemical industry energy consumption after the feedstock input and the primary fuel equivalent of purchased electricity have been put into consideration.
TABLE 1-9 : U.K. CHEMICAL INDUSTRY ENERGY CONSUMPTION (1979)\(^7\)

<table>
<thead>
<tr>
<th>Energy Consumption</th>
<th>therms x 10(^6)</th>
<th>KJ x 10(^{12})</th>
<th>% of Fuel</th>
<th>% of TOTAL</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Fuel : coal and other solid fuels</strong></td>
<td>158.3</td>
<td>16.7</td>
<td>3.7</td>
<td>50.3</td>
</tr>
<tr>
<td>oil &amp; other liquid fuels</td>
<td>1348.7</td>
<td>142.3</td>
<td>31.8</td>
<td></td>
</tr>
<tr>
<td>gaseous fuels</td>
<td>1166.7</td>
<td>123.1</td>
<td>27.6</td>
<td></td>
</tr>
<tr>
<td>purchased electricity</td>
<td>1561.9</td>
<td>164.8</td>
<td>36.9</td>
<td></td>
</tr>
<tr>
<td><strong>Sub-total : Fuel</strong></td>
<td>4234.9</td>
<td>446.9</td>
<td>100.0</td>
<td>49.7</td>
</tr>
<tr>
<td><strong>Feedstock: Liquid fuels</strong></td>
<td>3200.6</td>
<td>337.7</td>
<td>76.5</td>
<td></td>
</tr>
<tr>
<td>gaseous fuels</td>
<td>985.7</td>
<td>104.0</td>
<td>23.5</td>
<td></td>
</tr>
<tr>
<td><strong>Sub-total : Feedstocks</strong></td>
<td>4186.3</td>
<td>441.7</td>
<td>100.0</td>
<td>49.7</td>
</tr>
<tr>
<td><strong>TOTAL</strong></td>
<td>8421.2</td>
<td>888.6</td>
<td>100.0</td>
<td>100.0</td>
</tr>
</tbody>
</table>

Apparently, the fuel and feedstock energy demands of the chemical industry are about 50% each. However, it will be inappropriate to make a direct comparison between the data of Table 1-9 and those of Tables 1-7 and 1-8 owing to the different underlying assumptions. The comparison can be facilitated by adding the fuel energy demand of the chemical industry to the industry and overall consumption figures as well as discounting the primary fuel equivalent of purchased electricity in the chemical industry fuel energy input. Using a fuel to electricity conversion factor of 0.3, the data of Table 1-10A are obtained. On this basis therefore, the chemical industry would account for about 33% of U.K. industry sector energy consumption and 12% of overall consumption by all end users.
### TABLE 1-10A: COMPARISON OF U.K. CHEMICAL INDUSTRY, ALL INDUSTRY, AND TOTAL ENERGY CONSUMPTION (1979) (Electricity Generation Losses not included)

<table>
<thead>
<tr>
<th>Energy Consumption</th>
<th>therms x 10^6</th>
<th>KJ x 10^{12}</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chem. Ind: Fuel</td>
<td>3,131.9</td>
<td>331.5</td>
</tr>
<tr>
<td>Feedstock</td>
<td>4,186.3</td>
<td>441.7</td>
</tr>
<tr>
<td>(a) Total Chem. Ind.</td>
<td>7,328.2</td>
<td>773.2</td>
</tr>
<tr>
<td>(b) All Industry</td>
<td>22,360.9</td>
<td>2,359.3</td>
</tr>
<tr>
<td>(c) Overall U.K.</td>
<td>59,732.7</td>
<td>6,302.4</td>
</tr>
<tr>
<td>(a)/(b)</td>
<td>32.8%</td>
<td>32.8%</td>
</tr>
<tr>
<td>(a)/(c)</td>
<td>12.3%</td>
<td>12.3%</td>
</tr>
</tbody>
</table>

Alternatively, the data may be compared after electricity generation losses have been included. The U.K. Department of Energy gives the electricity demands by U.K. industry and by all consumers in 1979 as 2,533 and 8028 x 10^6 therms respectively. Based on this, the chemical industry demand would come down to about 29% of all industry energy consumption and 10.6% of total U.K. consumption (Table 1-10B).

### TABLE 1-10B: COMPARISON OF U.K. CHEMICAL INDUSTRY, ALL INDUSTRY AND TOTAL ENERGY CONSUMPTION (1979) (Electricity Generation Losses Included)

<table>
<thead>
<tr>
<th>Energy Consumption</th>
<th>therms x 10^6</th>
<th>KJ x 10^{12}</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chem. Ind: Fuel</td>
<td>4,234.9</td>
<td>446.9</td>
</tr>
<tr>
<td>Feedstock</td>
<td>4,186.3</td>
<td>441.7</td>
</tr>
<tr>
<td>(a) Total Chem. Ind.</td>
<td>8,421.2</td>
<td>888.6</td>
</tr>
<tr>
<td>(b) All Industry</td>
<td>29,365.0</td>
<td>3,098.3</td>
</tr>
<tr>
<td>(c) All U.K.</td>
<td>79,558.3</td>
<td>8,394.2</td>
</tr>
<tr>
<td>(a)/(b)</td>
<td>28.7%</td>
<td>28.7%</td>
</tr>
<tr>
<td>(a)/(c)</td>
<td>10.6%</td>
<td>10.6%</td>
</tr>
</tbody>
</table>
Grant\textsuperscript{4} presented approximate estimates of the broad end-use pattern of U.K. chemical industry non-feedstock (fuel) energy, which is reproduced in Table 1-11. This shows that nearly 75\% of the fuel energy input goes into process heating, which includes a wide range of heat-using operations, such as direct heating of process streams, distillation, evaporation, drying etc. It is noted however that this data includes purchased electricity used in electrolysis and for electrical drives for which the generation losses are not included.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Therms x 10^6</td>
<td>KJ x 10^{12}</td>
</tr>
<tr>
<td>Heat</td>
<td>2469</td>
<td>260.5</td>
</tr>
<tr>
<td>Electrolysis</td>
<td>224</td>
<td>23.6</td>
</tr>
<tr>
<td>Mechanical drives</td>
<td>458</td>
<td>48.3</td>
</tr>
<tr>
<td>Electrical drives</td>
<td>670</td>
<td>70.7</td>
</tr>
<tr>
<td>Mechanical drives</td>
<td>212</td>
<td>22.4</td>
</tr>
<tr>
<td>Mechanical drives</td>
<td></td>
<td></td>
</tr>
<tr>
<td>TOTAL</td>
<td>3363</td>
<td>354.8</td>
</tr>
</tbody>
</table>

1.2.3. Energy Use in Drying

Information on a breakdown of non-feedstock energy consumption in the chemical industry in terms of the actual unit operations in which they are used is difficult to come by, given the tendency in industry to assemble information on energy usage on a lumped basis. At best, most industries tend to identify plant areas which are more energy intensive but hardly go the whole length of detailed logging to obtain specific data in relation to the individual major operations carried out. However, it is well recognised that of the unit operations, distillation, evaporation drying, size reduction, mixing and materials handling (compression, pumping and conveying) are among the most energy intensive. In addition to being a long established operation in the chemical industry, drying has also found very extensive application in a broad range of other industrial sectors. These include food processing, textiles, pulp/paper, ceramics
building materials (cement, timber, bricks) and ores/minerals dressing. Besides, there are also some other strictly non-industrial sectors, such as laundering and crop drying, in which drying may also be an important operation.

**TABLE 1-12 : WATER REMOVED IN DRYING IN SOME MAJOR U.K. INDUSTRIES (1975)**

<table>
<thead>
<tr>
<th>Material</th>
<th>Water removed tonnes x 10^6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Paper and board</td>
<td>9.2</td>
</tr>
<tr>
<td>Bricks</td>
<td>2.4</td>
</tr>
<tr>
<td>Milk, dried</td>
<td>1.85</td>
</tr>
<tr>
<td>Milk, condensed</td>
<td>.85</td>
</tr>
<tr>
<td>Gypsum</td>
<td>.74</td>
</tr>
<tr>
<td>Plaster and plasterboard</td>
<td>1.0</td>
</tr>
<tr>
<td>Textiles</td>
<td>.4</td>
</tr>
<tr>
<td>China clay</td>
<td>.35</td>
</tr>
<tr>
<td>Fertilisers</td>
<td>.12</td>
</tr>
<tr>
<td>Timber, softwoods</td>
<td>.12</td>
</tr>
<tr>
<td>Timber, hardwoods</td>
<td>.05</td>
</tr>
<tr>
<td>Dyestuffs</td>
<td>.05</td>
</tr>
<tr>
<td>Vitrified china clay pipes</td>
<td>.11</td>
</tr>
<tr>
<td>Tiles, pottery and sanitary ware</td>
<td>.15</td>
</tr>
<tr>
<td><strong>TOTAL</strong></td>
<td><strong>17.4</strong></td>
</tr>
</tbody>
</table>

Reay\(^8\) presented data on the quantity of water removed from the products of some of these industries in U.K. (Table 1-12). Notably, the data covers most of the pulp/paper, ceramics, textiles and building materials industries, as well as sections of the chemical and food industries. However, it is estimated that the industries for which data is not available may account for about 40% of all the water evaporated in drying operations in U.K. industry\(^8,9\). On this basis therefore, a total of about 29-30 million tonnes of water is removed. Using a latent heat of evaporation of water of 2350 KJ/kg, the annual energy requirement for drying will then be about \(70 \times 10^{12}\) KJ. But for most
industrial dryers only a fraction of the total heat supplied is effectively used up in water removal. If a nominal efficiency of 50% is assumed, which is rather optimistic since quite often it is as low as 25-30%, annual energy consumption in drying would therefore be about $140 \times 10^{12}$ KJ. To put this in perspective, it is noted that compared with the data published by U.K. Department of Energy for 1975, drying alone would account for about 2.4% of total energy demand and 6.0% of energy consumption in all industry in U.K.

1.3 Project Objective

Deeply rooted in the principles of material and energy balances, the chemical industry in general has always been a relatively efficient user of energy. However, the challenges posed by the energy crisis have inspired renewed interest and effort in even more efficient utilisation of the increasingly scarce energy resources. It has been estimated that the potential exists for savings up to 103 million tonnes coal equivalent in U.K. chemical industry by the end of this century if all potential energy saving measures are fully implemented. These energy saving measures range from minor 'good housekeeping' measures, through modifications to existing plants or processes, to changes in design of new plants or devising of new process routes, in an increasing order of capital cost, implementation time and energy saving potential.

The minor 'housekeeping' measures applicable to most processes and unit operations, including drying, are generally well understood and identified and are increasingly being implemented. The present study is therefore geared toward possible energy saving measures in drying, either at the level of modification of existing installations, or involving innovation in the design of new plant. The scope of work includes:

(i) Examination of pattern of energy consumption in drying and identification of major sources of energy wastage.

(ii) Investigation of application of more conventional heat recovery methods.

(iii) Investigation of more innovative methods, like heat pump application.

In particular, development of a solids absorbent based heat pump cycle with possible application in heat recovery from dryers.
CHAPTER TWO

DRYING PROCESSES
2. DRYING PROCESSES

2.1 Scope of Application

As used in this context, drying refers to the removal of liquid from a solid containing liquid in an unbound form. Thus drying in this sense includes liquid removal by mechanical means, such as pressing and centrifugation, and also by heat and mass transfer processes (diffusion and evaporation), but excludes the removal of moisture from gasses by dehumidification or adsorption processes. (However, liquid removal by mechanical means is relatively ineffective in extracting a substantial amount of the liquid and is therefore almost always followed by a heat and mass transfer process when used. Indeed, heat and mass transfer is by far the most common method of drying and the consideration of drying here will therefore be confined to this method.)

(Though the above definition refers to any liquid, in practice evaporation of water is most frequently encountered. This will therefore provide the basis for much of the consideration. Besides, systems that do not involve evaporation of water almost invariably involve evaporation of volatile organic solvents which generally have much lower latent heats of vaporisation than water (see Table 2.1), with the result that they tend to be less energy intensive and therefore of less importance from the standpoint of energy utilisation in drying.

### Table 2.1. Latent Heats of Water and Organic Solvents

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>2,240</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>Benzene</td>
<td>381</td>
<td>5.88</td>
<td>5.88</td>
</tr>
<tr>
<td>Methanol</td>
<td>1,013</td>
<td>2.21</td>
<td>2.21</td>
</tr>
<tr>
<td>Ethanol</td>
<td>851</td>
<td>2.63</td>
<td>2.63</td>
</tr>
<tr>
<td>Acetone</td>
<td>461</td>
<td>4.86</td>
<td>4.86</td>
</tr>
<tr>
<td>Ethyl ether</td>
<td>293</td>
<td>7.64</td>
<td>7.64</td>
</tr>
<tr>
<td>Ethyl chloride</td>
<td>314</td>
<td>7.13</td>
<td>7.13</td>
</tr>
<tr>
<td>Carbon disulfide</td>
<td>272</td>
<td>8.24</td>
<td>8.24</td>
</tr>
<tr>
<td>Carbon tetrachloride</td>
<td>182</td>
<td>12.31</td>
<td>12.31</td>
</tr>
</tbody>
</table>
The breadth and scope of application of drying in industry have already been outlined in Chapter 1. The reasons for drying and its role in processing could be as diverse as the materials which may be processed. Obviously however, products must be in the most suitable form for storage, for subsequent processing, for handling or to meet product specification for marketing purposes, and drying often provides one of the most convenient means of preparing them to meet these requirements.

2.2 General Drying Behaviour

2.2.1 Equilibrium Relationships

When a moist solid is exposed (for a fairly long period) to a constant draught of low relative humidity air, the moisture content of the solid will approach an equilibrium value. This equilibrium moisture content will depend on the prevailing conditions, such as temperature, nature of the solids and relative humidity of the air. For certain materials, under most conditions all the moisture held by the solid is vaporised on attainment of equilibrium and hence equilibrium moisture content is practically zero, while for other substances there may remain a residual moisture content at equilibrium. These are referred to as hygroscopic substances.

Wet solids may contain moisture either in the bound or unbound form. (Wet sand, for instance, contains essentially unbound moisture; cotton, starch and leather on the other hand may contain both bound and unbound moisture. In general coarse granular solids tend to contain mostly unbound moisture and have equilibrium moisture values of zero, while others, especially fine, porous substances usually contain bound moisture and therefore tend to be hygroscopic.\textsuperscript{15,16} )

(The importance of equilibrium moisture content in drying lies in the fact that it represents the limiting moisture content for the given temperature and humidity at which the drying is carried out.) Fig. 2.1 below shows the equilibrium relationship between moisture content of solid and the corresponding moisture content of the air for hygroscopic and non-hygroscopic materials.
2.2.2 Drying Periods.

(If the moisture content of a drying solid is recorded with time, a general characteristic behaviour is obtained as shown in Fig. 2.2, which applies to most solids.) Though Fig. 2.2 indicates that the rate of drying (moisture evaporation) is subject to variation with time or moisture content, this variation can be better illustrated if the drying rate, obtained by differentiating the moisture content versus time curve, is plotted against moisture content or time as shown in Figs. 2.3 and 2.4. These rate curves clearly show that the process of drying is not a smooth continuous one but rather consists of several distinct stages. This phenomenon has long been recognised and has been extensively investigated by several workers.17,18) Thus, the drying period can be formally divided into three periods:

(i) an initial phase, represented by segment AB of the rate curves, during which the solid temperature adjusts to its steady-state value. It is usually so short that it is ordinarily ignored.

(ii) Constant-rate period, represented by section BC, and

(iii) falling rate period, represented by section CDE.

2.2.21. Constant Rate Period

In the constant-rate regime, drying proceeds at a fixed rate which represents the maximum possible rate attainable under the prevailing conditions. The entire exposed surface of the drying solid is saturated and drying is thought to take place by diffusion of vapour from the
Fig. 2.2. Moisture Content vs Drying Time.

Moisture content $X$

$X_E = \text{equilibrium moisture content.}$

Fig. 2.3. Drying Rate vs Moisture Content

Drying rate $N$

Fig. 2.4. Drying Rate vs Time
saturated surface across a stagnant air film into the environment. Moisture movement within the solid is rapid enough to maintain a constant saturated condition at the surface. The rate of drying is controlled by the rate of heat transfer to the evaporating surface and is completely independent of the nature of the solid. This is well-illustrated by the observation made by Sherwood and Comings in which they pointed out that under similar drying conditions many substances tend to dry at approximately the same rate in the constant-rate regime.

**TABLE 2.2 : MAXIMUM EVAPORATION RATES FROM MATERIALS IN A PAN WITH CROSSFLOW AIR AT 3 m/s and 50°C.**

<table>
<thead>
<tr>
<th>Material</th>
<th>Rate of Evaporation kg/m² - hour</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>0.27</td>
</tr>
<tr>
<td>Whiting pigment</td>
<td>0.21</td>
</tr>
<tr>
<td>Brass filings</td>
<td>0.24</td>
</tr>
<tr>
<td>Brass turnings</td>
<td>0.24</td>
</tr>
<tr>
<td>Fine sand</td>
<td>0.20-0.24</td>
</tr>
<tr>
<td>Clay</td>
<td>0.23-0.27</td>
</tr>
</tbody>
</table>

The rate of mass transfer balances the rate of heat transfer in the regime and the temperature of the saturated surface remains constant. Thus the drying rate will be given in terms of mass transfer as:

\[
N_c = \frac{dx}{d\Theta} = K_g A (p_s - p) - 2.1
\]

where

- \( N_c \) = constant rate of drying
- \( x \) = moisture content of solid
- \( \Theta \) = time
- \( A \) = area for evaporation
- \( K_g \) = mass transfer coefficient
- \( p \) = partial pressure of vapour in air
- \( p_s \) = vapour pressure of moisture at surface temperature of solid.
and in terms of heat transfer as:

\[ N_c = -\frac{dX}{dt} = h_t \frac{A}{\lambda} (T - T_s) \]  \hspace{1cm} 2.2

where \( h_t \) = total heat transfer coefficient

\( \lambda \) = latent heat of vaporisation at surface temperature

\( T \) = air temperature

\( T_s \) = solid surface temperature.

If heat is transferred solely by convection and in the absence of other heat effects, the surface temperature of the solid equals the prevailing wet-bulb temperature of the air under constant-rate drying conditions and \( h_t \) is the convective heat transfer coefficient \( h_c \). If the heat is also supplied by radiation, as is sometimes the case in very high temperature drying, \( h_t \) will be the sum \( h_c + h_r \), where \( h_r \) is the radiation coefficient and \( T_s \) will be higher than the wet-bulb temperature. For conductive drying through hot surfaces, the solids temperature approaches the boiling point temperature rather than the wet-bulb temperature.

During constant-rate drying, it is established that the rate of drying depends on the velocity, and hence flowrate, of air. For air flowing parallel to the solid surface in cross-circulation batch dryers, extensive experimental results have been satisfactorily correlated by the relationship \[ h_t = 0.0128 G^{0.8} \]  \hspace{1cm} 2.3

where \( G \) is airflowrate.

For airflow perpendicular to the drying surface Molstad et al.\(^{22}\) proposed the relationship

\[ h_t = 0.37 G^{0.37} \]  \hspace{1cm} 2.4

Treybal\(^{23}\) has suggested that equation 2.3 above can be reliable only within the range

500 < \( G \) < 6,000 lbs/hr - ft\(^2\)
2 < \( v \) < 25 ft/sec

and for equation 2.4

800 < \( G \) < 4,000 lbs/hr - ft\(^2\)
3 < \( v \) < 15 ft/sec.

where \( v \) is air velocity.
2.2.22. Falling-Rate Period

The point C in Figs 2.2-2.4 represents the transition from the constant rate period to the falling-rate period. It is referred to as the critical point and the moisture content at that juncture is called the critical moisture content. The falling-rate period can be divided into two zones:

(i) the first falling rate period CD during which the entire exposed surface can no longer be maintained saturated due to slower moisture movement within the solid, unsaturated surface drying takes place, and drying rate falls as drying proceeds, followed by (ii) the second falling rate period DE when moisture no longer gets to the surface, the plane of evaporation recedes into the solid and evaporation occurs from the interior of the solid.

The falling rate regime is characterised by an increasing surface temperature. Drying stops at the end of the second falling rate period when the equilibrium moisture content of the solid for the prevailing air relative humidity has been attained. During the unsaturated surface drying the rate of drying is a point function of the moisture content and varies linearly with it. This portion of the curve may be virtually absent or may constitute virtually the whole of the falling rate period, depending on the nature of the material being dried. For these cases when virtually the whole falling-rate curve is linear drying rate can be expressed as:

\[ \frac{N}{dX} = \frac{N_c}{X - X_E} \]

where \( X_c \) = critical moisture content
and \( X_E \) = equilibrium moisture content.

2.2.23. Critical Moisture Content

Critical moisture content depends on the ease of moisture movement through the solid and hence on the pore structure of the solid in relation to the drying conditions.) The complicated and uneven structure of porous solids therefore makes prediction of critical moisture content rather difficult. In general granular or crystalline materials have been observed
to exhibit relatively low critical moisture content and thus are dried predominantly at constant rate. On the other hand, the amorphous, fibrous or gel-like solids tend to show only very short constant rate periods ending at high critical moisture contents. (In addition, critical moisture content is believed to be influenced by other operating conditions, such as drying rate used and depth of the solids bed.)

2.3 Theories of Drying

The mechanism of liquid movement and consequently the rate of this movement vary markedly with the structure of the solid itself and is of particular importance in determining the rate of drying, especially in the falling-rate regime. For many solids, the drying mechanism is such that the falling-rate is a point function of the humidity difference \( \Delta \) and moisture content \( X \) and can often be expressed by a mathematical equation. Several theories have been postulated in order to correlate these relationships.

2.3.1 Liquid Diffusion Mechanism

(In relatively homogenous solids, such as fibrous materials, gel-like substances and porous cakes, it is believed that the movement of moisture from the interior to the drying surface is mainly by molecular diffusion of liquid.\textsuperscript{27,28} The rate of moisture movement can therefore be expressed by a modified form of Fick's Law thus:–)

\[
\frac{dX}{d\Theta} = D_L^* \frac{\partial^2 X}{\partial x^2} \quad \text{1.6}
\]

\[D_L^* = \text{Liquid phase diffusion coefficient applicable for movement through the solid ft}^2/\text{hr (m}^2/\text{hr)}\]

\[x = \text{Distance measured along the depth of material.}\]

For the simplest case in which,

(i) \( D_L^* \) is assumed to be constant,

(ii) drying is presumed to occur from the top surface of a slab the sides and bottom of which are insulated, and

(iii) initial moisture content is evenly distributed throughout the slab.
Newman and Sherwood have obtained a simplified solution to equation 2.6:

$$\frac{X - X_e}{X_c - X_e} = \frac{8}{\pi} e^{-D_L^* \theta (\frac{\pi}{l})^2}$$

where $l$ = slab thickness.

This can be readily differentiated to obtain the drying rate as:

$$\frac{dX}{d\theta} = -\frac{\pi^2 D_L^*}{l} \left( X - X_c \right)$$

Since liquid movement by diffusion is a relatively slow process, the drying rate curves associated with many solids to which it is applicable show almost no constant rate period but rather are often characterised by very dominant falling-rate regimes.

2.3.2. Capillary Theory

The capillary theory of drying postulates that for particulate solids and other substances with a large open-pore structure, movement of liquid within the solid capillaries results from a net force arising from differences in hydrostatic head and in surface tension effects. The basic importance of capillarity in granular materials was first highlighted by Slitcher in connection with the movement of moisture in soils. His work was elucidated and subsequently applied to the problem of drying by Ceaglske and Hougen and by Wheat and MacLeod.

As drying proceeds, at first moisture moves by capillarity to the surface sufficiently rapidly to maintain a uniformly wetted surface and the rate of drying remains constant. Subsequently, owing to an interplay of surface tension and gravitational effects in the capillaries, the proportion of the total surface that is wetted starts to decrease, marking the beginning of the unsaturated-surface drying period. By the end of this period, the sub-surface reservoirs eventually dry up and the liquid recedes into the capillaries and the second falling-rate period then starts.

Generally, the drying rate can be expressed with fair accuracy over the required range of moisture contents by an equation of the form

$$\frac{dX}{d\theta} = -K_1 (X - X_e)$$
where \( K_1 \) is a function of the constant drying rate as follows:

\[
K_1 = \frac{(dX/dt)_c}{X_c - X_e} \quad -\quad -\quad -\quad -\quad -\quad 2.10
\]

but

\[
N_c = \frac{(dX)}{d\theta}_c = \frac{h}{\lambda} (T - T_s) \quad -\quad -\quad -\quad 2.2
\]

From equations 2.2 and 2.10

\[
K_1 = \frac{h}{\lambda} \frac{(T - T_s)}{X_c - X_e} \quad -\quad -\quad -\quad -\quad -\quad 2.11
\]

and hence,

\[
\frac{dX}{d\theta} = \frac{h}{\lambda} \frac{(T - T_s)}{X_c - X_e} \quad -\quad -\quad -\quad -\quad 2.12
\]

This relationship is believed to hold for many materials which conform to the capillary mechanism of drying and represents the case in which most of the drying in the falling-rate regime consists of unsaturated surface drying and the falling-rate curve is entirely linear, to a fair approximation. 25

2.3.3 Receding Plane Model

The receding plane model for drying assumes that moisture evaporates from a sharp front on the drying surface which progressively withdraws into the body as it dries out and then diffuses as moisture vapour to the surface. It is assumed that this diffusion constitutes the rate-controlling step of the process. 31

By studying the temperature fields in drying brick-like materials Heertjes and Tuider 32 have shown that after an initial period has elapsed, two distinctive thermal regions appear. The temperature of the solid was observed to drop from a surface value of about dry-bulb temperature of the air to about the wet-bulb value at a distance away from the surface and this temperature persists thereafter throughout the remainder of the body as shown in Fig. 2.5.
Evidently, the discontinuity in temperature gradient represents the position of an evaporation front within the solid. Further evidence for the existence of the evaporation front separating a dry region from a wet one within a drying solid is provided from tests on drying terylene bobbins by Nissen et al\textsuperscript{24}.

Vapour-diffusion control of drying rate is most likely to be tenable when heat is applied to one surface of a solid while drying proceeds from another, as has been suggested by Treybal\textsuperscript{23}. In such situations moisture may evaporate beneath the surface and diffuse out as a vapour. The evaporation front recedes further into the solid as drying progresses.

Bates\textsuperscript{33} has developed an analytical treatment of receding plane drying, following his work on the drying of slabs made up of layers of cloth. Both heat and water vapour are deemed to be transported through a layer of dried material of thickness $x$ and at equilibrium moisture content. At any instant, the thickness of the dried layer is related to the average moisture content of the entire bed $X$ by the relationship

$$x = \frac{Z_s (X_c - X)}{X_c - X_e} \quad \text{------------------- 2.13}$$

If drying proceeds from only one flat surface of the slab $Z_s$ represents total depth of the slab, but when it proceeds from two opposite surfaces $Z_s$ is one-half of the total depth.

Heat transfer inside the slab is by conduction while transfer across the air film adjacent to the solid surface is by convection. Thus the overall heat transfer coefficient for the system will be given by:
\[
\frac{1}{U} = \frac{1}{h_c} + \frac{x}{k} \tag{2.14}
\]

where \( U \) is overall heat transfer coefficient,
\( h_c \) is convective heat transfer coefficient.
\( k \) is thermal conductivity.

\[
U = \frac{h_c k}{k + h_c x} \tag{2.15}
\]

but,
\[
N = \frac{U}{\lambda} (T - T_s) \tag{2.2}
\]

combining equations 2.13, 2.15 and 2.2,
\[
N = \frac{h_c k (X_c - X_e) (T - T_s)}{[k (X_c - X_e) + h_c Z_s (X_c - X)]} \tag{2.16}
\]

However, if conductive transfer inside the slab is considered to be rate limiting, then
\[
u = \frac{k}{x} \tag{2.17}
\]

Hence, the drying rate will be
\[
N = \frac{k (X_c - X_e) (T - T_s)}{Z_s (X_c - X)} \tag{2.18}
\]

2.3.4. Wetted Surface Model.

This model assumes that the drying rates in the falling rate regime lessen due to the smaller fraction of the solid surface that is wetted. It thus follows that if a relative drying rate \( f_d \) is defined as:
\[
f_d = \frac{N}{N_c} \tag{2.19}
\]

then \( f_d \) is equal to the fractional wetted area. To a first approximation, it has been suggested that the area available for evaporation varies as the square of an equivalent wet length, whereas the moisture concentration would vary as this length cubed. Hence, if a characteristic moisture content \( \phi \) is defined as:
\[
\phi = \frac{X - X_e}{X_c - X_e} \tag{2.20}
\]

then,
\[
f_d = \frac{\phi}{3} \tag{2.21}
\]
However, considering that the evaporative interface may recede into the body, the exponent on $D$ could be expected to be somewhat less. Therefore for a more general application equation 2.21 may be modified as:

$$f_d = \phi^n \quad 0 < n < \frac{2}{3} \quad - - - - 2.22$$

The value of the exponent $n$ is determined empirically and has been found to be 0.6 in the drying of Calsawood slabs up to 9.5 mm in thickness. Even with this modification this expression is only satisfactory to the extent that the material being dried is thin, and the average moisture content is approximately equal to the surface moisture content. Otherwise for thick materials, equation 2.21 would have to be corrected by a 'thickness factor' which depends on the ratio of the moisture content at the surface $X_o$ to the mean moisture content $X$. Then,

$$f_d = \left(\frac{X - X_o}{X - X_e}\right)^{\frac{2}{3}} \phi^{\frac{2}{3}} \quad - - - - 2.23$$

The model has been corroborated in the drying of moist granular beds and has been successfully applied to calculation of moisture content profiles in rotary drying of gypsum and in fluidized bed drying of pharmaceuticals.

2.3.5. Other Models

Various models, including the flow through porous media concept, the simultaneous transport approach and pressure mechanism have also been proposed for describing falling rate drying. These have however had more limited application than the previously considered models.

2.4. Performance of Dryers

2.4.1. Drying Time

Once the rate characteristics of drying a material are determined, the time required for drying can be readily calculated. Since drying is controlled by different factors at different stages of the process, the drying rate curve must be considered in its major sections in evaluating drying time.
Drying rate is related to the moisture content of the solid by the equation:

\[ N = - \frac{W_s}{A} \frac{dX}{d\theta} \quad \text{--- 2.24} \]

\( W_s \) = weight of dry solid

\( A \) = drying area

Rearranging and integrating

\[ \int_{0}^{\Theta} d\Theta = - \frac{W_s}{A} \int_{X_1}^{X_2} \frac{dX}{X_1} \quad \text{--- 2.25} \]

where \( X_1 \) = bulk moisture content at time 0

and \( X_2 \) = bulk moisture content at time \( \Theta \).

For the constant rate period, \( N_c \) is independent of \( X \) and hence equation 2.25 can be integrated readily within appropriate limits to obtain \( \Theta \). However, for the falling-rate regime, an appropriate rate expression, depending on the prevailing mechanism, may be substituted in equation 2.25 in order to facilitate integration.

2.4.2 Design of Dryers

The preliminary problem that confronts the process engineer in designing a dryer is to identify the nature of the material to be processed and the conditions under which it is to be processed in order to enable him to choose the most suitable type of drying equipment to be used. Once the type of dryer to be used has been chosen, the problem reduces to a question of application of basic principles to size the equipment. However, in addition to the difficulty of predicting equilibrium relationships, drying rate curves, and hence drying times, further difficulties arise concerning such aspects as variation of drying conditions through the dryer, non-uniform drying of solids, differences in area for heat transfer and for mass transfer, gas flow pattern, and configurations of material handling facilities.
Consequently, despite recent advances in the scientific understanding of drying, most of the practice is still very much of an art. In most instances preliminary tests necessarily have to be carried out under simulated production conditions. Even then, generalised techniques for scaling up are often lacking and frequently empirical procedures have to be resorted to. Poersch has highlighted these various steps involved in dryer design and presented them in an illustrative flowchart (fig. 2.6).

![Flowchart for Dryer Design](image)

**Fig. 2.6. Flowchart for Dryer Design.**

2.4.21 Batch Dryers

Most commonly batch drying is carried out in cabinet or tray dryers where the stock is held stationary in trays. This is by far the simplest type of dryer. If the initial adjustment period which precedes the constant-rate period can be neglected the total drying time is the sum of $\theta_c$ and $\theta_r$. To this must also be added the estimated dead time $\theta_d$ required for loading and unloading the dryer at the start and at the end of each batch charge. Hence, total batch time will be given by:

$$\theta_b = \theta_c + \theta_r + \theta_d \quad 2.26$$
Once the tray size and the batch charge have been chosen, the number of trays required can be readily estimated and hence the overall size of the dryer.

2.4.22. Continuous Dryers

Continuous drying carried out in either parallel or counter flow in tunnel, conveyor or rotary equipment could be amenable to fairly straightforward analytical treatment. However, many types of drying operations are so complicated by such factors as backmixing, crossflow and several other problems peculiar to specific types of drying equipment that it is sometimes exceedingly difficult to analyse them with any degree of certainty. For example, in spray dryers, though normal heat and mass transfer concepts are applicable they are of limited importance because they do not constitute the dominant controlling factor in the process. A successful spray dryer design will depend rather on analysis of:

(a) atomisation of the feed into droplets by a nozzle or rotating disc;
(b) evaporation times for a droplet of decreasing surface area and decreasing velocity relative to a medium of increasing humidity and decreasing temperature;
(c) trajectory of a droplet of decreasing mass.

On the other hand, for fluidised bed dryers the main peculiarity of design is the need to ensure that optimum conditions for particle fluidisation exist, in addition to provision of adequate heat transfer surface area to maintain the desired evaporation rate. The overall size of the bed must also be determined so as to provide sufficient residence time for mass transfer.

Several design methods have been proposed for continuous dryers in parallel or counter flow arrangements. Most of these methods are based on two fundamental simplifying assumptions:

(i) evaporation of moisture takes place at the adiabatic saturation temperature throughout the dryer;
(ii) instantaneous drying rate is a point function of the drying condition and is independent of the previous history of the material.
Furthermore, they all involve a general preliminary procedure of establishing all the end conditions (temperature and humidity) by an overall material and energy balance. These design procedures include:

(i) Residence Time Method
(ii) Heat Transfer Unit (HTU) Method, and
(iii) Mass Transfer Unit (MTU) Method.

An ideal continuous counterflow dryer can be represented schematically as in fig. 2.7 and the associated temperature and concentration profiles for air and solid going through such a dryer are shown in figs. 2.8 and 2.9. For purposes of analysis, the dryer can be considered to consist of three distinct sections, I, II and III, such that all the moisture is evaporated in section II, while sections I and III are concerned primarily with pre-heating and post-heating the solid. The residence time design approach, outlined by Foust et al\(^{39}\), involves the determination of residence time of material in each section and hence obtaining the overall residence time as a sum of the separate residence times:

\[
\theta_T = \theta_I + \theta_{II} + \theta_{III} \quad \text{(2.27)}
\]

where \(\theta_T\) is overall residence time. Several empirical relationships exist for various dryer types which enable the dryer size to be estimated then.

Alternatively, the dryer can be sized by calculating the number of HTUs and the length of each transfer unit. For the ideal dryer (fig. 2.7), conforming to the profiles of figs. 2.8 and 2.9, the number of HTUs in any section is given by:

\[
N_t = \left[ \frac{dT}{T-t} \right] \quad \text{(2.28)}
\]

which on integrating

\[
N_t = \frac{T_{in} - T_{out}}{(\Delta T)_m} \quad \text{(2.29)}
\]

where \((\Delta T)_m\) is log-mean temperature difference.
Fig. 2.7: Ideal Continuous Counter Flow Drying

Fig. 2.8: Temperature profiles in Counter Flow Drying.

Fig. 2.9: Concentration Profiles in Counter Flow Drying.
The total length of the dryer may then be obtained as:

\[ L = (N_t L_t)_I + (N_t L_t)_II + (\xi_t L_t)_III \]  

where \( L_t \) is length of a HTU.

Tsao et al.\(^{15}\) have suggested empirical equations for evaluation \( L_t \) for through circulation tunnel and rotary dryers.

The MTU approach considers the ideal continuous dryers to consist of two sections, as shown in figs. 2.10 to 2.12. Evidently drying in section I is at a constant rate while drying in section II is at a falling rate. The number of MTUs in I can then be given as:

\[ N_{t_I} = \int \frac{dy}{y_w - y} \]  

and in section II,

\[ N_{t_II} = \frac{dy}{(X - X_e)(y_w - y)} \]  

Empirical equations, similar to those applicable to the HTU method, are available for evaluating the length of each MTU. Hence, total dryer length can be obtained as:

\[ L = (N_t L_t)_I + (N_t L_t)_II \]  

2.4.23 Indirect Dryers

Indirect drying is done without direct contact between the feedstock and the drying medium. Heat is supplied to the stock mainly by conduction. In many other aspects indirect drying processes are similar to contact drying ones. However, with indirect drying, the drying rate is controlled by the rate of heat supply, thus:

\[ q = U A_h \Delta T \]  

\( A_h \) is the surface area available for heat transfer, which may differ from the surface area for drying. The temperature driving force, \( \Delta T \) is the difference between the temperature of the heating medium and the temperature of the stock. When unbound moisture is being removed, the stock temperature will be approximately equal to the boiling point of the moisture, but in other cases the stock temperature will be higher and is more difficult to estimate. Since drying is controlled by heat transfer, indirect dryers are sized generally on the basis of heat transfer area requirements.
Fig. 2.10. Ideal Continuous Parallel Flow Drying.

Fig. 2.11. Temperature profiles in Parallel Flow Drying.

Fig. 2.12. Concentration Profiles in Parallel Flow Drying.
2.5 Energy Requirements in Drying

The total energy requirements for drying solids consists of several distinct components, which include:

(i) energy required to provide latent heat for evaporating moisture from the solids, $Q_1$;
(ii) sensible heat lost in the exhaust air stream, $Q_2$;
(iii) energy absorbed by the dryer wall and associated structures, $Q_3$;
(iv) energy lost through the walls of the dryer to the environment, $Q_4$;
(v) energy associated with the hot solids as they leave the dryer, $Q_5$;
(vi) energy for circulation of air through the dryer, $Q_6$;
(vii) energy required to rotate or vibrate dryer, $Q_7$;
(viii) energy for operation of materials handling equipment, $Q_8$.

The energy inputs (i) - (v), which are basically thermal energy inputs, are intrinsically associated with the actual process of drying and can therefore be regarded as process energy demands. On the other hand, inputs (vi) - (viii) are not intrinsically associated with the process. They are usually supplied in the form of electrical energy which is used to produce mechanical motion. They could therefore be regarded as mechanical energy inputs.

The importance of each input varies considerably, depending on the conditions and the types of dryer in consideration. For instance, $Q_2$ is usually very small in indirect drying, whereas it will be a dominant item for direct drying. Furthermore, while rotation energy $Q_7$ will be important in rotary dryers, it will be largely irrelevant to through circulation tunnel dryers, where material handling energy demands $Q_8$ could be significant.

2.5.1 Ideal Instantaneous Energy Efficiency

(Considering that the main purpose of drying is to vaporize moisture from wet solids, the energy efficiency of a dryer may be defined as the ratio of energy required to vaporize moisture to the total energy supplied to the dryer,)

\[
\eta = \frac{\text{energy required for moisture evaporation}}{\text{total energy supplied}}
\]
hence, if
\[ Q_T = \sum_{i=1}^{8} Q_i \]  \hspace{1cm} 2.35

then
\[ \eta = \frac{Q_1}{Q_T} \] \hspace{1cm} 2.36

Slight differences exist between the various interpretation of this definition in literature. Several workers\textsuperscript{10,44,45} do not consider it very essential that the energy requirement for vaporisation should be evaluated in relation to the fact that while the solids are fed at low temperature the vaporized moisture is exhausted at relatively high temperature. Ashworth\textsuperscript{46} however has suggested that operation at high temperature should be penalized by defining the vaporisation energy relative to the temperature at which the solids are fed.

An ideal energy efficiency can be defined for a perfectly insulated, leakproof and well-operated dryer in which the mechanical energy demands and the losses associated with the solids and with the dryer walls are all very negligible and can be considered insignificant compared with both the energy requirement for vaporisation and the sensible heat lost in the exhaust air. Thus,
\[ Q_T = Q_1 + Q_2 \] \hspace{1cm} 2.37

and hence ideal efficiency will be given by
\[ \eta_{\text{ideal}} = \frac{Q_1}{Q_1 + Q_2} \] \hspace{1cm} 2.38

Dryers may be operated in two basic configurations:
- straight-through flow
- air recirculation flow.

Consider an ideal convective dryer in which a fraction of the exhaust gas is recycled as shown in fig. 2.13. An expression for the energy efficiency can be derived from heat and mass balances about the dryer.

Thus, suppose heat is transferred adiabatically between the circulating air and the solids, the rate of heat transfer will be given by : -
Fig. 2.13. Convective Dryer with Air Recirculation.
\[ Q_G = \dot{G} \cdot C_{\text{pg}} (T_1 - T_2) \quad - - - - \quad 2.39 \]

\[ C_{\text{hi}} = C_p + C_{pv} y_i \quad - - - - \quad 2.40 \]

where \( C_p \) = specific heat of air

\( C_{pv} \) = specific heat of water vapour

\( G \) = air mass flow velocity

\( A \) = cross-sectional area perpendicular to air flow direction.

In addition to vaporisation of moisture, this energy is also used for sensible heating of moisture, among others. An appropriate basis for estimating the fraction of the energy actually used up for vaporisation will therefore be to compare the latent heat of vaporising the moisture at the feed temperature \( T_f \) and the enthalpy difference between the cold moisture and the exit vapour. Hence,

\[ Q_1 = Q_G \frac{\lambda_f}{\lambda_f + C_{pv}} \frac{(T_2 - T_f)}{(T_2 - T_f)} \quad - - - - \quad 2.41 \]

where \( T_f \) = solids feed temperature

then,

\[ Q_1 = C_{h1} \cdot \dot{G} \cdot A \cdot (T_1 - T_2) \quad - - - - \quad 2.42 \]

A mass balance on the heater gives,

\[ r = \frac{y_1 - y_o}{y_2 - y_o} \quad - - - - \quad 2.43 \]

and an energy balance on the heater gives,

\[ Q_{T} = \dot{G} \cdot A \left[ \frac{C_{h1} (T_1 - T_2) + C_{h0} (1 - r) (T_2 - T_o)}{C_{h0} (T_1 - T_2) (y_2 - y_o)} \right] \quad - - - - \quad 2.44 \]

Hence, the ideal energy efficiency will be given by,

\[ \eta_{\text{ideal}} = \frac{1}{1 + C_{h0} \left( \frac{T_2 - T_o}{T_2 - T_f} \right) \left( y_2 - y_1 \right)} \quad \frac{\lambda_f}{\lambda_f + C_{pv} \left( \frac{T_2 - T_f}{T_2 - T_f} \right)} \quad 2.45 \]

If the dryer is operated in straight flow,

\[ \eta_{\text{ideal}} = \frac{1}{1 + C_{h0} \left( \frac{T_2 - T_o}{T_2 - T_f} \right) \left( y_2 - y_1 \right)} \quad \frac{\lambda_f}{\lambda_f + C_{pv} \left( \frac{T_2 - T_f}{T_2 - T_f} \right)} \]
Now, supposing:

(i) the exhaust air leaves at a reasonably low temperature, such that the need for penalizing for high temperature operation is obviated;

(ii) temperature and humidity conditions are low enough for the humid heat capacity of the air to remain fairly constant,

then, equations 2.45 and 2.46 respectively reduce to

\[ \eta_{\text{ideal}} = \frac{1}{1 + \frac{(T_2-T_0)(Y_2-Y_1)}{(T_4-T_2)(Y_2-Y_0)}} = \frac{T_1-T_2}{(T_4-T_2)+(1-r)(T_2-T_0)} \]  \hspace{1cm} - - - - 2.47

and

\[ \eta_{\text{ideal}} = \frac{1}{1 + \frac{T_2-T_0}{T_4-T_2}} = \frac{T_1-T_2}{T_2-T_0} \] \hspace{1cm} - - - - - 2.48

Equations 2.47 and 2.48 have been proposed by Hawtin and Reay and by Hodgett for estimation of the efficiency of convective dryers.

Examination of equations 2.45-2.48 above indicates that efficiency will be maximised at low exhaust temperatures and high extent of recirculation. In theory, efficiencies of nearly 100% are attainable at 100% recirculation of air. However, it must be recognised that this represents a trivial situation in that no drying would take place at all. Besides, the degree of recirculation is very closely inter-related to the temperature and humidity limits within which the dryer operates, as has been noted by Ashworth. A simple conclusion regarding its effect on energy efficiency may therefore be incorrect, except if the overall psychometric relationship is fully taken into consideration. By considering the dimensionless heat transfer potential \( \Pi_H \) and mass transfer potential \( \Pi_M \) of the circulating air, defined respectively by

\[ \Pi_H = \frac{T - T_s}{T_1 - T_s} \] \hspace{1cm} - - - - - - - - - - - 2.49

and

\[ \Pi_M = \frac{Y - Y_s}{Y_1 - Y_s} \] \hspace{1cm} - - - - - - - - - - - 2.50
Ashworth\textsuperscript{47} proposed that efficiency can be regarded to be approximately proportional to the moisture carrying capacity of the air. Clearly, when the air leaves saturated the dimensionless potentials become zero

\[ \Pi_H = \Pi_M = 0 \]  

Then, the moisture carrying capacity is maximum, which represents the conditions for the maximum attainable efficiency.

Furthermore, Ashworth suggested that increasing the wet bulb temperature (saturation temperature for air/water system) improves the efficiency of the dryer. This is closely related to the observation made by Reay\textsuperscript{8} in which he proposed that the efficiency of a dryer is maximised when the enthalpy of the exhaust air per unit mass of vapour is minimised and went on to show that the higher the wet-bulb temperature the lower the exhaust air enthalpy.

2.5.2. Ideal Overall Efficiency

Equations 2.45-2.48 predict the instantaneous energy efficiency of convective dryers. However, in practice changes often occur in the inlet conditions to and outlet conditions from a dryer, with time. It therefore becomes necessary that a means of averaging the efficiency with respect to time should be found in order to obtain an overall efficiency. Hence, if \( \eta^* \) is overall energy efficiency, then:

\[
\eta^* = \frac{\int_0^\Theta T \frac{Q_1}{Q_T} \, d\Theta}{\int_0^\Theta T \frac{Q_T}{Q_T} \, d\Theta} \quad \text{2.52}
\]

where \( \Theta_T \) is residence time.

Therefore, for the general configuration involving recycle of air,

\[
\eta^* = \frac{\int_0^\Theta T \left[ \frac{C_{h_1} (T_1 - T_2)}{\lambda_f + \frac{C}{P_f} (T_2 - T_f)} \right] \, d\Theta}{\int_0^\Theta T \left[ \frac{C_{h_1} (T_1 - T_2) + C_{h_0} (T_2 - T_0)}{(Y_2 - Y_o)} \right] \, d\Theta} \quad \text{2.53}
\]

The nature of equation 2.53 is such that it may not be readily amenable to integration, which makes it very difficult for overall efficiencies to be calculated in this way. For most practical purposes therefore a less tedious but more approximate procedure given by:

\[
\eta^* = \frac{1}{C_n} \int_0^{\Theta_T} \eta \text{ ideal} \, d\Theta \quad \text{2.54}
\]

could be used.
The significance of overall energy efficiency could depend on the type of dryer in question, mode of operation and arrangement of airflow distribution relative to the solids. For example, in batch dryers the outlet conditions remain constant during constant-drying but start to vary as falling-rate drying sets in. Thus, the instantaneous and overall efficiencies are initially identical but start to diverge later. On the other hand the outlet conditions from a continuous dryer are fairly constant for the entire duration of drying and the instantaneous and overall efficiencies would be expected to remain close.

2.5.3 Actual Efficiency

The actual efficiency of a dryer is given by

\[ \eta = \frac{Q_1}{\sum_{i=1}^{8} Q_i} \quad 2.55 \]

It is estimated that for a well-designed modern dryer the overall impact of the other energy demands of the dryer, \( Q_3 - Q_8 \), should be such that \( \eta \) should not be more than 10-20% less than the ideal overall efficiency \( \eta* \).

2.6 Measurement of Thermal Efficiency of Tumble Dryer

In order to determine the actual thermal efficiency of a functioning industrial dryer, measurements were taken on a convective tumble dryer (Type Ibis 5.162 Kamsin) used for drying batch charges of textiles and fabric materials in a local laundry and dyeing works. The tumbler consists essentially of a rotating cylindrical chamber in which the charge is placed and is designed to handle a batch charge of about 45.4 kgs (100 lbs) dry weight fabrics in a batch time of about 15 minutes. The drying air is heated by steam batteries located at the rear of the machine and circulated by a motor-driven centrifugal blower. Steam supply to the batteries is from a central generating mains which serves the entire works.

In order to facilitate the investigation, the equipment layout was slightly modified by incorporating a t-junction and valve in the condensate return line. This enabled the condensate to be collected. Since measurements necessarily needed to be taken during the normal
operation of the dryer, care was exercised to arrange them to interfere minimally with normal operation. The procedure involved the following steps:

(i) Recording of steam delivery pressure on the installed line pressure-gauge;

(ii) Diversion of steam condensate through the fitted t-junction to collect steam delivery over a timed interval and hence obtain steam flow rate.

(iii) Weighing wet batch charge before feeding into the dryer and then dry fabric at the end of the drying period.

(iv) Measurement of temperature of fabric before and at the end of the drying period.

(v) Recording, at regular time intervals, of temperature of hot air stream as it enters the drying chamber and of the moist air as it leaves the chamber.

(vi) Measurement of prevailing ambient wet and dry bulb temperatures.

The data obtained from several batch measurements enabled the thermal performance characteristics of the dryer to be evaluated. A sample calculation is outlined in Appendix B. Table 2.3 shows a comparison between the actual performance and that specified by the equipment vendor. While the machine is designed for about 48% thermal efficiency and 2.5 kg steam/kg moisture specific steam consumption, the actual performance is about 39% thermal efficiency and 3.0 kg/kg specific steam rate. This represents an 80% approach to design specification, which is reasonably impressive for an equipment that has been consistently in use for over ten years.

<table>
<thead>
<tr>
<th>TABLE 2.3 : PERFORMANCE OF TUMBLE DRYER</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>SPECIFIED PERFORMANCE</strong></td>
</tr>
<tr>
<td>STEAM PRESSURE (Psia)</td>
</tr>
<tr>
<td>STEAM RATE (KG/Min)</td>
</tr>
<tr>
<td>MOISTURE EXTRACTION RATE (KG/Min)</td>
</tr>
<tr>
<td>SPECIFIC STEAM RATE (KG/KG)</td>
</tr>
<tr>
<td>THERMAL EFFICIENCY (%)</td>
</tr>
</tbody>
</table>
A breakdown of heat use in the dryer is shown in Table 2.4 below.

TABLE 2.4 : BREAKDOWN OF ENERGY CONSUMPTION IN TUMBLE DRYER.

<table>
<thead>
<tr>
<th>HEAT USE</th>
<th>% OF HEAT SUPPLY</th>
</tr>
</thead>
<tbody>
<tr>
<td>EVAPORATION</td>
<td>39</td>
</tr>
<tr>
<td>EXIT AIR</td>
<td>56</td>
</tr>
<tr>
<td>HOT FABRICS</td>
<td>1</td>
</tr>
<tr>
<td>DRYER WALL/STRUCTURES</td>
<td>4</td>
</tr>
</tbody>
</table>

Evidently, of the total heat losses from the dryer (61%), the highest proportion of heat (56%) is lost as sensible heat in the exhaust air stream. To a large extent this loss is unavoidable since it is inherent in the basic nature of the drying process that the moist air is exhausted at temperatures above ambient. A practical problem which invariably contributed to this huge loss is the question of over-drying of fabrics. Owing to the difficulties involved in measuring moisture in solids, instrumentation is not provided for monitoring the moisture content of fabrics in the tumble dryer. Consequently, intuition and experience have to be resorted to in order to determine when the fabric is sufficiently dry. The result is that often over-drying occurs - the dryer is kept running while an insignificant amount of moisture or even none is being extracted from the fabrics - and the effect is a net reduction in the thermal efficiency of the dryer.

It is noted that the other losses (those associated with the dry fabrics and exchange through the dryer walls) are limited to about 5%. This can be regarded as a tolerable loss level. It must be pointed out that the key factor for this low level of loss through the walls is the fact that the dryer shell is double-walled and well insulated. The breakdown of energy demand is further illustrated in a Sankey diagram in Fig 2.14.
FIG. 2.14. SANKEY REPRESENTATION OF ENERGY USE IN TUMBLE DRYER.
2.7 Energy Conservation

2.7.1 Housekeeping

Like in most other industrial operations, efficiency is severely hampered if care is not taken in regular operation and maintenance of drying equipment and if the simple, good housekeeping measures are not adopted to curtail losses from the system. Most of these measures, though basic and well understood, are sometimes surprisingly neglected. A checklist of the measures will include:

- optimum choice of operating conditions, including temperatures, humidities and air rate,
- avoidance of air leakage by regular checks and replacement of packings and seals,
- maintenance and replacement of wall insulations,
- optimum setting of fuel/air ratio of burners for direct-fired systems,
- regular cleaning of burners and heat exchange surfaces,
- minimisation of downtime in batch drying,
- regular maintenance and proper operation of mechanical and electrical equipment, such as motors, blower and conveyors,
- avoidance of over-drying and avoidance of non-uniform drying of solids by ensuring uniform solid feedrate, uniform loading across dryer,
  low feed moisture content, narrow particle-size range, uniform airflow distribution and uniform agitation of solids,
- reduction of moisture content of feedstock by less energy intensive mechanical processes such as filtration, centrifuging etc,
- proper maintenance and care of control devices and instruments.

Though the list spans through most of the common measures that could be adopted to improve efficiency, it cannot be regarded as exhaustive. Numerous other measures exist which if appropriately adopted in specific situations will also help improve efficiency.
2.7.2 **Practical Constraints**

Although improved efficiencies can be attained by optimising drying conditions this can be hampered by a number of practical constraints:

(a) Solid materials could be thermally sensitive, arising from a number of reasons, such as degradation, discoloration and melting. The sensitivity will therefore fix an upper limit for the solids temperature and hence the air inlet temperature.

(b) There could be dangers of fire or explosion hazards at high temperatures especially if considerable dusting of fines is involved.

(c) If the exhaust humidity is too close to saturation, condensation of moisture may occur on the dryer walls or outlet ductworks causing corrosion problems.

2.7.3 **Heat Recovery**

It is clear however that owing to the nature of drying, even with the most careful operation and best housekeeping measures, substantial energy losses will continue to be incurred in the form of sensible heat content in the exhaust air, with the large latent heat content of the associated vapour. This therefore suggests that the best scope for substantial improvement in the energy efficiency of drying systems will be in devising means of recovering heat from the exhaust air.
CHAPTER THREE

CONVENTIONAL HEAT RECOVERY METHODS
3. CONVENTIONAL HEAT RECOVERY METHODS IN DRYING

3.1. Exhaust Air Recirculation

Recirculation of a portion of the effluent air from a convective dryer has long been practiced, mainly for purposes of controlling humidity levels and drying rates, especially when temperature sensitive materials are being dried. More recently, with the need for energy savings, the technique is being increasingly adopted also as a means of recovering energy from dryer exhaust air streams. However, as has been noted in the preceding chapter, a simple conclusion on the energy saving potential of recirculating air could be misleading, except if it is considered in terms of the overall psychometric relationship in the dryer.

To analyze the effect of air recirculation on performance of a dryer, it is important that it is done in the context of clearly specified operating conditions, in order to provide a basis for comparing the performance at different levels of recirculation. For the present study operation of a dryer under the following two conditions will be considered:

(i) constant fractional saturation of air,
(ii) constant air wet-bulb temperature.

3.1.1. Constant Fractional Saturation

A schematic representation of an air recycle convective dryer is shown in fig. 3.1. Using the notation of the diagram, the fractional saturation of air in a drying process is defined by the ratio:

\[ f = \frac{y_2 - y_1}{y_w - y_1} \]  \hspace{1cm} (3.1)

where \( f \) is fractional saturation and \( y_w \) is wet-bulb humidity.

The dryer can be operated in such a way that the amount of air recirculated varies, but \( f \) as defined above, is maintained constant. Then, from equation 3.1:

\[ y_2 = f y_w + (1 - f) y_1 \]  \hspace{1cm} (3.2)
Fig. 3.1: Air Recycle Convective Dryer.
If the fraction of air recirculated to the dryer is \( r \) the moisture balance equation for the mixing of recycled air and fresh make-up air can be written as:

\[
y_1 = y_2r + y_0(1-r) \tag{3.3}
\]

Hence,

\[
y_2 = \frac{y_1 - y_0}{r}(1-r) \tag{3.4}
\]

Combining equations 3.1 and 3.4,

\[
\frac{y_1 - (1-r)y_0}{r} = f y_w + (1-f)y_1 \tag{3.5}
\]

and thus:

\[
y_1 = \frac{(1-r)y_0 + f r y_w}{1 - r (1-f)} \tag{3.6}
\]

Now, the wet-bulb partial pressure of moisture in the air can be expressed in terms of \( y_w \) as:

\[
p_w = \frac{29 y_w}{18 + 29 y_w} \tag{3.7}
\]

where \( \Pi \) is total pressure.

Hence, using Antoine's phase equilibrium equation for water, wet bulb temperature will be,

\[
T_w = \frac{3883}{18.4 - lnp_w} - 230 \tag{3.8}
\]

Furthermore, by heat balance on the dryer,

\[
c_{h_1} (T_1 - T_2) = \frac{\lambda_f}{\lambda_f + c_p(T_2 - T_f)} = \lambda_w (y_2 - y_1) \tag{3.9}
\]

where \( c_{h_1} \) is humid heat capacity of inlet air

\( \lambda_f \) is latent heat of vaporisation of moisture at solids feed temperature \( T_f \).

\( \lambda_w \) is latent heat of vaporisation of moisture at wet-bulb temperature.
By replacing $T_2$ and $y_2$ with $T_w$ and $y_w$ respectively in equation 3.9, and rearranging,

$$T_w = \frac{\lambda_f c_h T_1 + \lambda_w \left( c_{p_v} T_f - \lambda_f \right) (y_w - y_1)}{\lambda_f c_h + \lambda_w c_{p_v} (y_w - y_1)}$$  \hspace{1cm} (3.10)$$

Solving equations 3.2, 3.6, 3.7, 3.8 and 3.10 iteratively enables the values of $y_1$ and $y_2$ to be obtained. Then $T_2$ can be obtained thus:

$$T_2 = \frac{\lambda_f c_h T_1 + \lambda_w \left( c_{p_v} T_f - \lambda_f \right) (y_2 - y_1)}{\lambda_f c_h + \lambda_w c_{p_v} (y_2 - y_1)}$$  \hspace{1cm} (3.11)$$

The instantaneous thermal efficiency of the dryer can then be obtained using equation 2.45 derived previously in Section 2.5 thus:

$$\eta = 1 + \frac{c_{h_0}(T_2 - T_0)(y_2 - y_1)}{c_{h_1}(T_1 - T_2)(y_2 - y_1)}$$  \hspace{1cm} \frac{\lambda_f + \lambda_{p_v}(T_2 - T_f)}{\lambda_f + \lambda_{p_v}(T_2 - T_f)}$$  \hspace{1cm} (2.45)$$

The instantaneous thermal efficiency of a convective recycle dryer, operated with a fixed fractional saturation, has been predicted theoretically for air inlet temperatures in the range 50 - 200°C and the results are plotted in figs 3.2 and 3.3. It is clear from fig. 3.2 that the thermal efficiency of the dryer is improved considerably as more of the exit air is recirculated, increasing more rapidly as fractional saturation approaches unity, and attains a maximum, in theory, when all the air is recirculated (i.e. $r = 1$). This is consistent with Ashworth's observation, discussed in section 2.5. The results also enable other general inferences to be drawn on the operational characteristics of dryers. Fig 3.2 shows efficiencies for two levels of fractional saturation, 0.4 and 0.8, and obviously higher thermal efficiencies would be expected with high fractional saturations. Over the range of inlet temperatures considered it is also evident that an enhancement of the thermal efficiency would be expected for high temperature operation, especially if a high fractional saturation is also maintained.
Fig. 3.2: Thermal Efficiency as a Function of Recycle Ratio, Fractional Saturation and Temperature.

$T_i$ is air inlet temperature.

Efficiency $\eta$ (%)

Recycle ratio $r$. 

$T_i = 200^\circ C$

$T_i = 150^\circ C$

$T_i = 100^\circ C$

$T_i = 50^\circ C$

---

 fractional saturation $f_{SAT}$

 fractional saturation $f=0$
Fig. 3.3: Thermal Efficiency and Wet-Bulb Temperature as Functions of Recycle Ratio and Fractional Saturation.

Air Inlet Temperature
\( T_1 = 100^\circ C \)

Efficiency
\( \eta \)

(\%)

Recycle ratio \( r \)

\( f = 0.4 \)

\( f = 0.6 \)

\( f = 0.8 \)

Wet-bulb temperature
\( T_w (^\circ C) \)

\( f \) is fractional saturation

--- thermal efficiency

--- wet bulb temperature
Fig. 3.3 clearly illustrates the dependence of air wet-bulb temperature on the recirculation ratio. As recirculation is increased, $T_w$ increases gradually initially, but rises exponentially as recirculation tends to unity. Furthermore, it is observed that overall higher wet-bulb temperatures are also associated with high fractional saturations. Since the same conditions would tend to favour both thermal efficiency and wet-bulb temperature, these observations are noted to be consistent with those of Reay, highlighted in section 2.5.

### 3.1.2 Constant Wet-Bulb

A convective dryer can alternatively be operated with a fixed wet-bulb temperature while the fraction of the exhaust air recycled is varied. For a set wet-bulb temperature, the corresponding wet-bulb partial pressure of moisture in the air can be obtained using Antoine's equation thus:

$$p_w = \exp \left( 18.4 \frac{3863}{T_w + 230} \right) \quad (3.12)$$

Hence, the wet-bulb humidity of the air will be:

$$y_w = \frac{18}{29} \left( \frac{p_w}{P - p_w} \right) \quad (3.13)$$

Using the adiabatic saturation equation, the humidity of the inlet air will then be obtained as:

$$y_1 = y_w - \frac{c_p}{\lambda_w} (T_1 - T_w) \quad (3.14)$$

From equation 3.3 above

$$y_2 = y_1 - y_o (1 - r) \quad (3.3)$$

and hence from equation 3.11

$$T_2 = \frac{\frac{\lambda_f}{\lambda_w} c h_1 T_1 + \frac{\lambda_f}{\lambda_w} \left( \frac{c_p}{c_p} \frac{T_f - \lambda_f}{c_p} \right) (y_2 - y_1)}{\frac{\lambda_f}{\lambda_w} c h_1 + \frac{\lambda_f}{\lambda_w} \frac{c_p}{c_p} (y_2 - y_1)} \quad (3.11)$$

This thus enables instantaneous thermal efficiency to be calculated using equation 2.45.
The performance of a dryer operated in this mode has been investigated for the range of wet-bulb temperatures 30-60°C and for various air inlet temperatures between 50°C and about 400°C. The results are presented in figs. 3.4 and 3.5. Evidently, unlike in operation with constant fractional saturation, when operating with a constant wet-bulb temperature thermal efficiency would decrease as the extent of recirculation of effluent air is increased. This effect appears to be more pronounced at low recycle levels when thermal efficiency drops very sharply for a small increase in recycle ratio. In further contrast to constant fractional saturation drying, in this mode of operation use of low rather than high air inlet temperatures would lead to higher efficiencies.

The psychometry of the system is such that for a given wet-bulb temperature and air inlet temperature, there is a minimum recycle ratio that must be maintained in the dryer. This can be expressed as:

\[ r_{\text{min}} = \frac{y_1 - y_0}{y_w - y_0} \]  

(3.15)

At \( r_{\text{min}} \) the exhaust air would be expected to be saturated and the maximum possible thermal efficiency under the operating conditions is attained.

Operating with high wet-bulb temperatures enables higher thermal efficiencies to be attained, as shown in fig. 3.6, where thermal efficiency and fractional saturation are plotted as functions of wet-bulb temperature and recirculation ratio, for a fixed air inlet temperature of 120°C. Evidently too, while increasing recirculation ratio reduces the fractional saturation, increasing wet-bulb temperature enhances it. It is therefore apparent that high fractional saturation would be associated with high thermal efficiency.

At this stage, the following deductions about the recirculation of effluent air as a means of recovering heat in drying can therefore be made:
Fig. 3.4: Thermal Efficiency as a Function of Recycle Ratio and Temperature
(Wet-Bulb Temperature $T_w=40^\circ C$)

$T_1$ is inlet temperature.

Efficiency $\eta$ (%)

Recycle ratio $r$. 
Fig. 3.5: Thermal Efficiency as a Function of Recycle Ratio and Temperature
(Wet-Bulb Temperature $T_w = 60^\circ C$).

$T_1$ is inlet temperature.
Fig. 3.6: Thermal Efficiency and Fractional Saturation as functions of Recycle and Wet-Bulb Temperature.
(1) Both high and low thermal efficiencies can be obtained by recirculating air, depending on the mode of operation. If recirculation is attended by an increase in wet-bulb temperature or in fractional saturation, then recirculation would improve thermal efficiency. Conversely, the effect on thermal efficiency is unfavourable if recirculation is attended by a fall in fractional saturation or wet-bulb temperature.

(2) Operating with constant fractional saturation, it will be most desirable to recirculate a large proportion of the effluent air but for constant wet-bulb temperature operation, it is most beneficial to keep the recirculation ratio low.

3.1.3. Effects on Drying Rate

The rate of drying in the constant rate regime is expressed as:

\[ N_c = \frac{h_t A}{\lambda_w} (T_1 - T_w) \]  

(2.2)

The heat transfer coefficient \( h_t \) is related to the flowrate of air and correlations have been proposed for this relationship for different process conditions and configurations, as has been noted previously in section 2.2. For through circulation of air in a batch dryer, the relationship is expressed as:

\[ h_t = 0.37 G^{0.37} \]  

(2.4)

where \( h_t \) is in \( \text{Btu/ft}^2 - \text{F} - \text{hr} \).

and \( G \) is in \( \text{lb} / \text{ft}^2 - \text{hr} \).

In metric units, the equation can be rewritten as:

\[ h_t = 11.37 G^{0.37} \]  

(3.16)

where \( h_t \) is in \( \text{KW/m}^2 - \text{C} - \text{hr} \).

and \( G \) is in \( \text{g/m}^2 - \text{hr} \).

Assuming that the drying surface area and the dryer cross section area normal to air flow are equal, as would be expected in most through circulation batch dryers, the drying rate can also be expressed in terms of airflow and humidity as:
Thus, from equations 2.2, 3.16 and 3.17,

$$\frac{11.37G}{\lambda_w (y_2-y_1)} (T_1-T_w) = G(y_2-y_1) \tag{3.18}$$

then,

$$G = \left[ \frac{11.37 (T_1-T_w)}{\lambda_w (y_2-y_1)} \right]^{1.59} \tag{3.19}$$

where G is in kg/m²-hr.

hence,

$$N_c = A \left[ \frac{11.37(T_1-T_w)}{\lambda_w (y_2-y_1)} \right]^{1.59} (y_2-y_1) \tag{3.20}$$

If a dryer is operated with a constant fractional saturation, for a given air inlet temperature T₁, as the recirculation ratio is increased the inlet humidity y₁ will also tend to increase. In addition, the wet-bulb temperature and humidity T_w and y_w will also tend to increase. From the definition of fractional saturation,

$$f = \frac{y_2 - y_1}{y_w - y_1} \tag{3.1}$$

it follows therefore that the outlet humidity y₂ must also increase in order to maintain f constant. By referring to equation 3.19 it is obvious then that there will be a fall in the flowrate G. In practice this implies that the flowrate of air must be decreased to maintain a constant fractional saturation if the recycle ratio in the dryer is increased. Consequently, the rate of drying will also decrease.

It becomes apparent then that for this mode of operation while recirculation of exhaust air improves the thermal efficiency of a dryer it reduces the rate of drying. As has been suggested by Noden^{107} energy costs account for up to 65% of the cost of operating a dryer. Any significant improvement in thermal efficiency of a dryer is therefore transmitted directly to the operating cost. On the other hand, a decrease in the rate of drying can be interpreted as a need for a larger sized equipment in order to handle a specific amount of charge, and hence extra capital investment. Recirculation of air can therefore be optimised for a set of drying conditions in terms of drying rate and thermal efficiency by considering capital and running costs and
determining the ratio that gives the shortest payback time.

Reliable cost information on dryers, both capital and operating, is difficult to obtain. Therefore, an alternative optimisation approach based only on technical performance considerations is proposed here. The specific moisture extraction rate in a drying system (MER) can be defined as the amount of moisture removed per unit of energy input, i.e.

\[
\text{MER} = \frac{n}{\lambda_w} \text{kg/KJ} \quad (3.21)
\]

Clearly, it will be desirable to operate with as high MER as possible. Now, if it is assumed that the essential performance characteristics of a dryer are energy consumption and drying rate, then a drying performance factor DPF can be defined as the product of the MER and the drying rate thus:

\[
\text{DPF} = \text{MER} \times N_c \quad (3.22)
\]

DPF is in units of \((\text{kg})/\text{(KJ)/(hr)}\)

The optimum performance of a dryer can therefore be obtained by maximising DPF (i.e. maximum removal of water with minimum energy and in minimum time).

The specific moisture extraction rate, the drying rate and the drying performance factor are shown in fig. 3.7 as functions of fractional saturation and air recirculation ratio for air inlet temperature 100°C. The locus of the optimum points has been traced over the range of fractional saturation 0.4-0.8. Evidently, high recirculation ratios will give optimum performance for dryers operated with low fractional saturation while low recirculation ratios will optimize high fractional saturation operation. The predicted performance of the dryer at optimum conditions is tabulated in Table 3.1A. Clearly, the indication is that the best performance, as reflected in the value of DPF, will be obtained for operation with low fractional saturation.
Fig. 3.7: Optimization of Recycle Ratio in terms of Efficiency and Rate (Air inlet Temperature $T_i=100^\circ C$)

Performance Factor

$$P = \frac{I_2^2}{LJ\cdot hr \times 10^3}$$

$f$ is fractional saturation
$x$ specific moisture extraction rate $N_c$
$\circ$ constant drying rate $N_c$

Recycle ratio $r$.
Optimization of Recycle Ratio in terms of Efficiency and Rate (Fractional saturation $f=0.6$)

- Locus of optima.
- $T_1 = 200^\circ C$
- 150
- 100
- 50

$T_1$ is inlet temperature.

- Specific moisture extraction rate $x$
- Constant drying rate $N_c$

Recycle ratio $r$.

Drying rate $N_c$ ($\frac{kg}{hr}$)
Fig. 3.8: Optimization of Recycle Ratio in terms of Efficiency and Rate (Fractional saturation $f=0.6$)

- Performance factor $\left(\frac{\nu^2}{\nu^2 \text{hr x } 10^3}\right)$
- Drying rate $\left(\frac{kg}{\text{hr}}\right)$

$T_1$ is the inlet temperature.

- Specific moisture extraction rate $MCR$.
- Constant drying rate $N_c$.

Recycle ratio $r$. 

Locus of optima.
TABLE 3.1A : DRYER OPTIMUM PERFORMANCE \( (T_1 = 100^\circ C) \)

<table>
<thead>
<tr>
<th>( f )</th>
<th>( \eta ) (%)</th>
<th>( \text{MER} ) ( \left( \frac{\text{kg}}{\text{kJ}} \right) \times 10^3 )</th>
<th>( N_c ) ( \left( \frac{\text{kg}}{\text{hr}} \right) )</th>
<th>( \text{DPF} ) ( \left( \frac{\text{kg}}{\text{kJ}-\text{hr}} \right) \times 10^3 )</th>
<th>( r )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4</td>
<td>52.5</td>
<td>0.215</td>
<td>1.75</td>
<td>0.38</td>
<td>0.70</td>
</tr>
<tr>
<td>0.6</td>
<td>53.0</td>
<td>0.225</td>
<td>0.40</td>
<td>0.09</td>
<td>0.41</td>
</tr>
<tr>
<td>0.8</td>
<td>57.5</td>
<td>0.24</td>
<td>0.15</td>
<td>0.04</td>
<td>0.00</td>
</tr>
</tbody>
</table>

TABLE 3.1B : DRYER OPTIMUM PERFORMANCE \( (f=0.6) \)

<table>
<thead>
<tr>
<th>( T_1 ) ( (\circ C) )</th>
<th>( \eta ) (%)</th>
<th>( \text{MER} ) ( \left( \frac{\text{kg}}{\text{kJ}} \right) \times 10^3 )</th>
<th>( N_c ) ( \left( \frac{\text{kg}}{\text{hr}} \right) )</th>
<th>( \text{DPF} ) ( \left( \frac{\text{kg}}{\text{kJ}-\text{hr}} \right) \times 10^3 )</th>
<th>( r )</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>53.5</td>
<td>0.215</td>
<td>0.17</td>
<td>0.03</td>
<td>0.47</td>
</tr>
<tr>
<td>100</td>
<td>54.5</td>
<td>0.225</td>
<td>0.40</td>
<td>0.09</td>
<td>0.43</td>
</tr>
<tr>
<td>150</td>
<td>54.5</td>
<td>0.220</td>
<td>0.68</td>
<td>0.15</td>
<td>0.35</td>
</tr>
<tr>
<td>200</td>
<td>51</td>
<td>0.210</td>
<td>1.04</td>
<td>0.22</td>
<td>0.13</td>
</tr>
</tbody>
</table>

Similar curves have also been plotted for a dryer operated with fractional saturation 0.6 and air inlet temperatures 50-200\(^\circ C\) (fig. 3.8) and the corresponding optimum performance data are presented in Table 3.1B. The optimum recirculation ratio decreases as the air inlet temperature is increased, as shown by the locus of optimum points in fig. 3.8. Given that high operating temperatures support both high efficiencies and faster drying rates, understandably, the DPF data in Table 3.1B show that the best performance would be expected to be obtained with high temperatures.

In the case of operation with constant wet-bulb temperature, the air inlet conditions (temperature and humidity) are fixed, hence an increase in the air recirculation ratio automatically implies that the exhaust air humidity must be reduced. In practice this can only be achieved by increasing the flow of air through the dryer as is reflected in equation 3.19 above. Consequently, the rate of drying also increases. Clearly therefore, with this mode of operation,
Fig. 3.9: Optimization of Recycle Ratio in terms of Efficiency and Rate. (Wet-bulb temperature $T_w=40^\circ C$)

$\text{LOG DPF}$

$\left( \frac{k_r^2}{L_{j-\text{hr}}} \times 10^3 \right)$

$T_1 = 160^\circ C$

$\text{LOG KER}$

$\left( \frac{k_r}{L_{j}} \times 10^3 \right)$

$T_1 = 160^\circ C$

Recycle ratio $r$

$T_1$ is inlet temperature.

$\times$ specific moisture extraction rate $K\text{ER}$

$\circ$ constant drying rate $N_0$
while recirculation of air impairs the thermal efficiency of the dryer it enhances the drying rate. The specific moisture extraction rate, the constant drying rate and the dryer performance factors are shown as logarithmic functions of air inlet temperature and recirculation ratio for a wet-bulb temperature of 40°C in fig. 3.9. The plots show that while the MER falls gradually as recirculation is increased, the rate of drying increases very rapidly, in a manner disproportionate with the decrease in MER. This implies therefore that the DPF is maximized for the maximum recirculation ratio attainable, which in theory is 1.0.

In conclusion, it is obvious then that increase in air recirculation ratio does not always lead to a fall in drying rate as is often believed. Depending on the method of operation, the present calculations have shown that the reverse effect could also be obtained.

3.2. Heat Exchange Recuperation

An alternative approach to the recovery of heat from dryer exhaust streams is to incorporate a heat exchange device, such as air/air heat exchanger, heat pipe, thermal wheel or run-around coil, into the system, as shown in fig 3.10.

3.2.1 Efficiency of Combined System

Without incorporating a heat recovery device, the thermal efficiency of the dryer is expressed by equation 2.45, derived in Section 2.5, as:

\[
n = \frac{1}{1 + \frac{c_p \gamma_2 (T_2 - T_0)}{c_h_1 (T_1 - T_2)} \left( \frac{y_2 - y_1}{y_2 - y_0} \right)}
\]

\[
\lambda_f + \frac{\gamma_2 (T_2 - T_f)}{p_v \gamma_2 (T_2 - T_f)}
\]

On incorporating the device, efficiency of the combined system using the notations of fig. 3.10 can then be rewritten as:

\[
\eta_R = \frac{1}{1 + \frac{c_h_0 (T_h - T_2)}{c_h_1 (T_1 - T_2)} \left( \frac{y_2 - y_1}{y_2 - y_0} \right)}
\]

\[
\lambda_f + \frac{\gamma_2 (T_2 - T_f)}{p_v \gamma_2 (T_2 - T_f)}
\]

Design of heat exchange devices is based on sound heat balance calculations, derived from the 1st Law of Thermodynamics. Since all the heat given up by the hot stream is deemed to be taken up by the cold stream (neglecting losses) all heat exchangers will therefore
Fig. 3.10: Combined Dryer/Heat Exchanger
operate with 100% efficiency. This however does not indicate the ability of the device to transfer all the heat potentially available in the hot stream to the cold stream.

Now, for any process stream entering or leaving a process stage 'availability' or 'exergy' can be defined as the potential of the stream for doing work, in a reference environment; that is the maximum amount of work that can be derived from the stream by a reversible process. Exergy can be expressed as:

\[ E_x = H - T_0 S \]  
(3.24)

where \( E_x \) is exergy of stream
\( H \) is enthalpy of stream
\( S \) is entropy of stream
and \( T_0 \) is temperature of the surroundings.

For finite changes:
\[ \Delta E_x = \Delta H - T_0 \Delta S \]  
(3.25)

An exergy balance can be performed for any real item of process equipment and it is found that owing to irreversibilities associated with real processes, overall, work potential is lost, i.e. real systems act as an exergy sink. Thus:

\[ \sum E_{x_{in}} = \sum E_{x_{out}} + E_{x_{lost}} \]  
(3.26)

where \( E_{x_{lost}} \) is the lost work potential.

A measure of the efficiency of the process, often referred to as effectiveness, which reflects the loss of work - doing capacity in the irreversible process, can then be defined as:

\[ \text{effectiveness} = \frac{E_{x_{out}}}{E_{x_{in}}} \]  
(3.27)

For a counterflow heat exchange device, the maximum available energy will be obtained when heat transfer proceeds reversibly and the hot stream leaves at the same temperature as the cold stream enters. The exchanger effectiveness can then be simply expressed in terms of energy thus:

\[ \varepsilon_r = \frac{\text{actual heat transfer}}{\text{maximum possible heat transfer}} \]  
(3.28)

The actual heat transfer may be obtained by calculating the energy lost
by the hot fluid or the energy gained by the cold fluid. If the two streams do not possess the same heat capacity, given by the product of mass flowrate and specific heat, it is customary to calculate the maximum possible heat transfer using the one that has the lower heat capacity since it alone can possibly undergo a temperature change equal to the maximum temperature difference in the exchanger, which is the difference between the inlet temperatures of the hot and cold streams.

For the scheme shown in fig. 3.10, the make-up fresh air flowrate must be equal to the exhaust air flowrate. Hence, the maximum available energy will be determined by the lower of the humid specific heats of the two streams. Since the make-up fresh air is cool and relatively dry, the fresh air humid specific heat is the smaller and hence thermal effectiveness of the recuperator will be:

\[ \varepsilon_r = \frac{T_3 - T_0}{T_2 - T_0} \]  

(3.28)

Now, by energy balance on the recuperator,

\[ c_{h_0} (T_3 - T_0) = c_{h_2} (T_2 - T_4) \]  

(3.29)

\[ T_4 = T_2 - \frac{c_{h_0}}{c_{h_2}} (T_3 - T_0) \]  

(3.30)

From 3.28

\[ T_3 = T_0 + \varepsilon_r (T_2 - T_0) \]  

(3.31)

Combining 3.30 and 3.31,

\[ T_4 = T_2 - \frac{c_{h_0}}{c_{h_2}} \left( \varepsilon_r (T_2 - T_0) \right) \]  

(3.32)

Let

\[ \frac{c_{h_0}}{c_{h_2}} = \beta \]  

(3.34)

then,

\[ T_4 - T_0 = (T_2 - T_0) \left( 1 - \beta \varepsilon_r \right) \]  

(3.35)
Fig. 3.11: Thermal Efficiency of Combined Dryer/Heat Exchanger as a Function of Temperature and Fractional Saturation.

$\eta$ is fractional saturation.

Efficiency $\eta$ (%)

Inlet Temperature $T_1$ (°C)

- Effectiveness $e_r = 0$
- Effectiveness $e_r = 0.6$
Fig. 3.12: Thermal Efficiency of Combined Dryer/Heat Exchanger as a Function of Recycle Ratio and Fractional Saturation.

\( \eta \) is fractional saturation.

- \( f = 0.8 \)
- \( f = 0.6 \)
- \( f = 0.4 \)
- \( f = 0.2 \)

Effectiveness \( \varepsilon_r = 0.6 \)

Effectiveness \( \varepsilon_r = 0 \)
Similarly,
\[ T_4 - T_f = (T_2 - T_f) - \varepsilon_r (T_2 - T_0) \]  
(3.36)

Putting 3.35 and 3.36 in 3.23 enables the efficiency of the combined system to be expressed thus:
\[
\eta = \frac{1}{1 + \frac{c_{h_0}}{c_{h_1}} (T_2 - T_0) (y_2 - y_1) (1 - \varepsilon_r)} \cdot \frac{\lambda + \varepsilon_r c_{FV} (T_2 - T_f) - \varepsilon_r (T_2 - T_0)}{c_{FV} (T_1 - T_2)} \cdot \frac{\rho f}{(y_2 - y_0)}
\]  
(3.37)

The thermal efficiency of the combined dryer/heat-exchanger system has been computed for a non-recycle dryer and a heat exchanger with a rated thermal effectiveness of 0.6. The results are plotted in fig. 3.11. Clearly, over a wide range of fractional saturations and air inlet temperatures, the combined system offers very considerable advantage over the conventional dryer. The advantage of using the combined system for a range of air recycle ratios has also been examined, using the same exchanger thermal effectiveness and the results are presented in fig. 3.12. As has been noted previously, for a given fractional saturation and inlet temperature, thermal efficiency of a dryer would be expected to increase with recirculation ratio. Understandably therefore, there is less scope for improvement in the thermal efficiency of the conventional dryer at higher recirculation ratios. Clearly, greater advantage would be expected if the combined system is operated with very low recirculation ratios.

3.2.2 Heat Transfer Units

The number of heat transfer units HTU in a heat exchanger is defined by:
\[
HTU = \frac{UA}{C_{\text{min}}}
\]  
(3.38)

where
- \( U \) is overall heat transfer coefficient
- \( A \) is heat transfer area
- \( C_{\text{min}} \) is minimum of the heat capacities of the two streams.
Obviously therefore, HTu is a direct measure of the surface area requirement for heat transfer and hence the size of the device. For a counterflow heat exchanger, HTu is related to thermal effectiveness by:

\[ \varepsilon_r = \frac{1 - \exp \left[ -HTu \left( 1 - \frac{C_{\text{min}}}{C_{\text{max}}} \right) \right]}{1 - \frac{C_{\text{min}}}{C_{\text{max}}} \exp \left[ -HTu \left( 1 - \frac{C_{\text{min}}}{C_{\text{max}}} \right) \right]} \]  

hence,

\[ HTu = \frac{C_{\text{max}}}{C_{\text{max}} - C_{\text{min}}} \ln \left[ \frac{1 - \varepsilon_r \frac{C_{\text{min}}}{C_{\text{max}}}}{1 - \varepsilon_r} \right] \]  

To illustrate the effect of thermal effectiveness on the size requirement of the heat exchanger and the performance of the dryer, calculations have been done for the number of HTu's and the specific energy requirement, SER, which is the net energy input required to evaporate unit mass of moisture, for various values of effectiveness and fractional saturation. The results are presented in fig. 3.13.

Evidently, HTu is independent of fractional saturation but increases exponentially with effectiveness. This implies that the penalty for improving energy recovery will be increased heat exchanger size. If effectiveness is increased towards unity, extremely large heat exchangers would therefore be required. From its definition, it is obvious that specific energy requirement is a reciprocal function of thermal efficiency of the combined system and can be expressed as:

\[ \text{SER} = \frac{\lambda w}{\eta_R} \]  

Using a heat exchanger with a large effectiveness rating would be expected to enable more energy to be recovered, improving the cycle efficiency and hence reducing the energy input required. However, from the plots it is observed that apart from operation with very low fractional saturation (0.2), energy requirement is only very little reduced when effectiveness is increased. Therefore a high effectiveness rating, and hence very large heat exchanger, would be required to substantially reduce energy requirement.
Fig. 3.13: Energy Requirements and Heat Transfer Units in Combined Dryer/Heat Exchanger

Inlet Temperature $T_i = 150^\circ C$

$\theta$ is fractional saturation.

$\times$ specific energy requirement SZR

o heat transfer units HTU

Specific Energy Requirement SZR $(\text{kJ/kg})$

Heat transfer units HTU

Thermal effectiveness $\varepsilon_r$. 
This phenomenon can be explained by considering the log mean temperature difference (LMTD), which provides the driving force for heat transfer in the heat exchanger. Except for drying with very low fractional saturation or very high temperature, the air leaving a drying chamber is usually of relatively low temperature and therefore of low energy quality. Consequently, even the maximum recoverable energy from it under reversible conditions is fairly low. By using a large heat exchange effectiveness not much heat would therefore be recovered, while the temperature approaches at the ends of the exchanger are substantially narrowed down. A low temperature driving force would therefore be obtained, hence necessitating a large heat exchanger.

In addition to the low quality of the exhaust air, the application of heat exchange devices to heat recuperation in drying is further constrained by the very low overall heat transfer coefficients that exist in air/air heat transfer systems (see Table 3-2). By referring to the HTU equation

$$HTU = \frac{UA}{C_{min}}$$

(3.38)

it is clear that such low heat transfer coefficients will result in even larger heat exchanger size requirements.

**TABLE 3-2 : TYPICAL OVERALL HEAT TRANSFER COEFFICIENTS FOR VARIOUS PROCESS SYSTEMS**

<table>
<thead>
<tr>
<th>System</th>
<th>$\frac{2U}{(KJ/^\circ C-m^-hr)}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air/Air</td>
<td>200</td>
</tr>
<tr>
<td>Air/Hydrocarbon Gases</td>
<td>715</td>
</tr>
<tr>
<td>Air/Ammonia</td>
<td>2,250</td>
</tr>
<tr>
<td>Air/Water</td>
<td>400-1,000</td>
</tr>
<tr>
<td>Kerosene/Water</td>
<td>600-1,300</td>
</tr>
<tr>
<td>Sulphur dioxide/water</td>
<td>3,000-4,000</td>
</tr>
<tr>
<td>Steam/Feedwater</td>
<td>8,000-20,000</td>
</tr>
</tbody>
</table>
CHAPTER FOUR

HEAT PUMP DRYING
4. HEAT PUMP DRYING

A heat pump is essentially a device which, by putting work into its working fluid, extracts heat from a low temperature source and rejects it to a high temperature sink. Thus, it can upgrade heat from a low quality to a high quality. By coupling a heat pump to a dryer it is therefore possible to recover low quality heat from the dryer exhaust air and upgrade it for further use. This enables the exhaust air to be dehumidified and recycled to the dryer.

4.1 Closed Cycle Mechanical Vapour Compression Heat Pump

The closed cycle mechanical vapour compression heat pump, like the mechanical refrigerator, is a simple thermodynamic machine consisting of a compressor, a condenser, an expansion valve and an evaporator, with a circulating working fluid (fig. 4.1). Essentially, it is a reversed heat engine cycle and differs from the refrigerator only in application. A refrigeration plant is usually installed solely for cooling while a heat pump is used either for heating alone or for both heating and cooling. To carry out a heat pumping duty, the heating temperature (i.e. condenser temperature) has to be above that at which heat is to be used and the heat extraction temperature (i.e. evaporator temperature) has to be below that of the heat source.

4.1.1. Coefficient of Performance (COP)

The ideal heat pump can be considered to be a reversed Carnot heat engine, operating on the cycle shown in fig. 4.2, such that work is done on the working fluid, enabling an amount of heat $Q_c$ to be absorbed at a low temperature $T_c$ from a reservoir and an amount of heat $Q_h$ to be delivered at a higher temperature $T_h$. The efficiency of the heat pump, customarily referred to as coefficient of performance (COP), can be expressed as the ratio of the heat delivered to the heat equivalent of the mechanical work done on the working fluid thus:

$$\text{COP} = \frac{Q_h}{W} = \frac{Q_h}{Q_h - Q_c}$$

(4.1)

For the ideal Carnot cycle operation COP can also be expressed in terms of the absolute temperatures thus:
FIG. 4.1: CLOSED CYCLE MECHANICAL VAPOUR COMPRESSION HEAT PUMP
FIG. 4.2: IDEAL CARNOT HEAT PUMP CYCLE

FIG. 4.3: REAL HEAT PUMP CYCLE
The Carnot cycle however is merely a hypothetical conception which cannot be realised practically, owing to several thermodynamic constraints. These include:

(i) The non-isentropic nature of compression,
(ii) heat transfer at the condenser and evaporator may occur at variable temperatures owing to superheating, subcooling and pressure drops,
(iii) non-reversible, adiabatic expansion,
(iv) finite temperature differences between cold and hot fluids in the condenser and evaporator.

The real COP of an actual heat pump is therefore always considerably less than that of the Carnot heat pump. Practical heat pumps operate as reversed Rankine cycles, with isenthalpic expansion rather than adiabatic expansion. The cycle is best illustrated as a pressure-enthalpy plot as shown in fig. 4.3.

Perhaps a more realistic measure of the performance of a heat pump is the primary energy ratio (PER):

$$\text{PER} = \frac{Q_h}{E_p}$$

where $E_p$ is the primary energy equivalent of the work input. Hence if $\eta_c$ is the efficiency of the mechanical device which puts work into the heat pump relative to primary energy input, then:

$$\text{PER} = \eta_c \cdot \frac{Q_h}{W} = \eta_c (\text{COP})$$

It follows therefore that for a heat pump to be energetically more efficient than an ordinary fuel fired heating system, the PER must be larger than 1.0

4.1.2 Thermodynamic Efficiency

Coefficient of performance is based on 1st Law of thermodynamics analysis, which considers energy in terms of quantity. An extension
of the analysis in terms of the 2nd Law of thermodynamics indicates the quality or the work potential of energy. Hence, for a work consuming device, such as the heat pump, while the 1st Law efficiency will be defined as:

$$\eta = \frac{\text{energy output}}{\text{work input}}$$ (4.5)

the 2nd Law efficiency or thermodynamic efficiency will be defined as the ratio of the ideal work (i.e. minimum work) input to the system to the actual work put into the system. Thus:

$$\eta_T = \frac{\text{ideal work input}}{\text{work input}}$$ (4.6)

It is obvious then that thermodynamic efficiency is similar to the concept of effectiveness discussed in Section 3.2. and is also a measure of the irreversibility of real processes.

The ideal work is the work input required by a reversible Carnot heat pump. If such a heat pump operates between temperatures $T_h$ and $T_c$ and delivers an amount of heat $Q_h$, ideal work can then be expressed as:

$$W_{\text{ideal}} = Q_h \left( \frac{T_h - T_c}{T_h} \right)$$ (4.7)

From 4.6 and 4.7, thermodynamic efficiency can thus be given by:

$$\eta_T = \frac{Q_h}{W} \left( \frac{T_h - T_c}{T_h} \right)$$ (4.8)

Since the Carnot heat pump efficiency is the maximum attainable under given conditions, it is evident from equations 4.2 and 4.5 and 4.6, that $\eta_T$ can also be expressed as:

$$\eta_T = \frac{\text{COP}}{\text{COP}_{\text{max}}}$$ (4.9)

where COP refers to a real cycle and COP$_{\text{max}}$ refers to a Carnot cycle.
4.2 The E.C.R.C. Heat Pump Dryer

The U.K. Electricity Council Research Centre (E.C.R.C.) has developed a commercial scale mechanical heat pump which has been adapted successfully to wood drying. A schematic flow diagram of the cycle is shown in fig. 4.4. Moist exhaust air from the dryer is passed through the evaporator whereby it is contacted with cold working fluid at relatively low pressure. The air is cooled below its dewpoint enabling the moisture to be condensed and removed while the heat so released (sensible and latent) is used to evaporate the working fluid. The dehumidified air is then contacted with compressed working fluid at high temperature in the condenser. The working fluid condenses and the heat released is transferred to the air. The air, heated and dehumidified, is then recycled to the wood drying chamber. Usually the dryer exhaust air is split into two streams. One portion goes through the evaporator and is subsequently mixed with the rest before being passed to the condenser. This is necessary because the amount of heat liberated at the condenser is usually too much to be carried away by the dehumidified air stream alone.

The overall performance of the heat pump dryer may be measured by its specific moisture extraction rate (MER), which is the amount of water condensed in the evaporator per unit work put into the compressor. This enables the heat pump dryer to be compared directly with a conventional dryer. For a conventional dryer the MER can be obtained from equation 3.21 thus:

\[
MER = \frac{n}{\lambda_w} \tag{3.21}
\]

and hence,

\[
n = \lambda_w (MER) \tag{4.10}
\]

where \( MER \) is in kg/KJ and \( \lambda_w \) is in KJ/kg.
FIG. 4.4 : CLOSED CYCLE MECHANICAL HEAT PUMP DRYER
For an electrically driven heat pump compressor, MER is usually expressed in kg/KWh. Therefore, if the primary fuel conversion efficiency of electricity is $\eta_e$, then:

$$\eta = \frac{\eta_c \lambda_m \text{MER}}{3600}$$

Equation 4.11 thus enables the thermal efficiency of a heat pump dryer to be evaluated, which can then be compared directly with the thermal efficiency of a conventional dryer.

The temperature of the air recirculated to the dryer depends on the condenser temperature. Most available working fluids possess such thermodynamic properties that with reasonably attainable pressures they condense at relatively low temperatures. Consequently, initially the E.C.R.C. heat pump dryer was limited to operation with a maximum dryer chamber inlet temperature of about 50°C. With subsequent development temperatures of about 80°C are now attainable, with the possibility of going up to 100°C. Lawton presented data obtained from theoretical modelling and experimental investigation of the machine and this is reproduced in fig. 4.5. The theoretical calculations were done for temperature differences between the condenser and evaporator of 40°C and 50°C while practical values were obtained for temperature differences of 41-46°C, with air inlet temperature to the dryer about 80°C and outlet relative humidity 60%. Evidently the operational characteristics of the system are such that increased MER is obtained with increasing dryer temperature and decreasing temperature difference between the condenser and evaporator. With the currently available machine, MER up to about 3-4 kg/KWh has been obtained. Assuming a latent heat of vaporisation of moisture of 2400 KJ/kg and efficiency of primary fuel conversion to electricity of 30%, from equation 4.11, for an MER of 4 kg/KWh, cycle efficiency will be 80%. Compared with efficiencies of less than 40% obtainable with conventional drying, this indicates that there is substantial scope for reducing energy requirement in drying by using the heat pump to recover exhaust heat.

4.3 **Open-Cycle Vapour Recompression**

An open-cycle heat pump differs from a closed-cycle heat pump in that it functions without an external working fluid. The use of open-cycle heat pumping, often referred to as mechanical vapour
FIG. 4.5: PERFORMANCE OF ECRC MECHANICAL HEAT PUMP DRYER

Experimental data
$41^\circ C \leq \Delta T \leq 46^\circ C$

$T = 40^\circ C$

$T = 50^\circ C$

Chamber Temperature

$60^\circ C$ $80^\circ C$ $100^\circ C$

$\frac{\text{kg} \, \text{H}_2\text{O}}{\text{kWh}}$

2.0

3.0

4.0
recompression (MVR), for improving the energy performance of processes, especially in evaporators and to a more limited extent distillation, is fairly well advanced. The adaptation of this technique to heat recovery in drying is considered here.

4.3.1. Air Recompression

A cycle arrangement involving the recompression of dryer effluent air is shown in fig. 4.6. The effluent air is split into two streams. One stream is compressed to a high pressure and temperature and cooled against the other stream in a heat exchanger enabling some of the moisture content to be condensed and removed. The uncompressed stream is simultaneously heated up to a high temperature and subsequently mixed with the compressed stream before the air is recirculated to the dryer.

The dryer is considered to be operated with a given fractional saturation $f$ and air inlet temperature $T_1$. For any split ratio $r_s$ between the compressed and uncompressed streams the procedure previously outlined in section 3.1. is used to find the temperature and humidity conditions in the dryer.

For the preliminary estimation of the compression energy demand $E_{comp}$, it is assumed that the compressor operates under the following conditions:

(i) compression process is adiabatic,
(ii) mechanical efficiency of compressor is 100%,
(iii) compressor is electrically powered,
(iv) efficiency of generating electricity from primary fuels is 30%,
(v) compression is single stage.

Since the compression energy is the only energy input, the cycle efficiency can be expressed as:

$$\eta_{comp} = \frac{E_{vap}}{E_{comp}} \quad (4.12)$$

where $E_{vap}$ is energy used for evaporation of moisture.

Calculations have been carried out for the air recompression cycle for the following set of conditions:

$0.4 < f < 0.8$
$50 < T_1 < 200$
$0.2 < r_s < 0.8$
FIG. 4.6: OPEN CYCLE AIR RECOMPRESSION DRYER
FIG. 4.7 (a) & (b): AIR RECOMPRESSION CYCLES

(A)

Dryer

Heat Exchanger

Compressor

(B)

Dryer

Heat Exchanger

Compressor
The results obtained show that efficiencies of 20-35\% would be expected from the cycle over the range of operating conditions considered. The data presented earlier in Chapter 3 indicate that efficiencies in excess of 50\% could be expected from conventional drying under comparable conditions. It is clear therefore that in general the air recompression cycle would be unfavourable for heat recovery. It is also observed that these figures obtained here compare poorly with the practical efficiency value of about 40\% obtained for the tumble dryer discussed in Section 2.6. Similar calculations have also been done for various other possible air recompression cycle arrangements which involve bleeding of some of the circulating air and admission of fresh make-up air (fig. 4.7) and similar results have also been obtained.

4.3.2 Steam Recompression

Dry superheated steam can be used directly to dry solid materials in an open-cycle steam recompression cycle, as shown in fig. 4.8. As the dry superheated steam passes through the dryer heat is transferred from the steam which raises the solid temperature above the boiling point of water, thus evaporating moisture from the solid. Owing to the heat transfer, steam leaves the dryer at a reduced temperature and its quantity is increased by the addition of the evaporated moisture. On exit, an amount of steam equal to that added in the dryer is withdrawn, recompressed to a high temperature and pressure and then passed through the heat exchanger where it is contacted with the main stream of circulating steam. The high pressure steam is condensed while simultaneously the circulating stream is heated to an appropriate high temperature before it is returned to the drying chamber.

Extensive theoretical modelling of the system carried out at the E.C.R.C. indicates that potentially MERs of 3.5-4.5 kg/KW H can be obtained, with the added advantage that some of the mechanical power for recompression is recovered as high grade heat in the condensate, which can be readily used as process heat. This performance will correspond to efficiencies of 70-90\%, which seems very attractive.
FIG. 4.8: OPEN-CYCLE STEAM RECOMPRESSION DRYER
One characteristic of the system which contributes measurably to its high efficiency is that moisture evaporation takes place at boiling point and there is therefore no wet-bulb depression. The discussions of Section 3.1 have clearly highlighted that drying efficiencies are usually enhanced with low wet-bulb depression.

Preliminary development of this cycle is currently going on at the E.C.R.C. Progress to the pilot plant stage using a more complex version of the cycle has already been made elsewhere in France.\(^{113}\)

4.4. Absorption Heat Pumps (AbHP)

If water vapour is contacted with a concentrated aqueous solution of an alkali, say NaOH or KOH, maintained at a higher temperature (> 100°C) the vapour is condensed and absorbed in the solution, giving up its latent heat. The temperature of the solution is raised and its concentration diluted simultaneously. This would appear to contradict the 2nd Law of thermodynamics, since heat seems to pass of itself from a colder to a hotter body. However, this is not so. In fact this is a mass transfer controlled process for which the driving force is provided by the difference in the vapour pressures of the vapour and solution phases. Condensation and dilution will therefore continue until the two vapour pressures are equal, which occurs at the elevated boiling point of the dilute solution, before the process stops. This phenomenon is the principle of absorption heat pumping.

4.4.1 Honigman's Locomotive

One of the earliest applications of absorption heat pumping in the process industry dates back to last century in Germany when Honigman built a caustic cycle with which he drove factory locomotives.\(^{118}\). A vessel containing hot concentrated caustic solution was surrounded by a hot water jacket maintained under considerable pressure. When the regulator was opened, steam flashed off the water, passed through the locomotive cylinders and the exhaust steam was bubbled into the caustic solution. The caustic was heated up, enabling it to continue to boil off water in the surrounding jacket. Condensation of steam stopped when the caustic solution became very dilute and the locomotive was then taken to an evaporation plant where the caustic was re-concentrated.
FIG. 4.9: LYLE'S CYCLE

- **VAPOUR**
  - 140°F
  - 5.0 psi

- **E V A P O R A T O R**
  - 24°F

- **T H I N L I Q U O R**
  - CONDENSATE

- **T H I C K L I Q U O R**
  - CONDENSATE

- **C O A L B O I L E R**
  - 170°F

- **C O N D E N S A T E**
  - 60°F

- **W A T E R B O I L E R**
  - 180°F

- **W A T E R**
  - 60°F

- **C O N D E N S A T E**
  - 120°F

- **S T E A M**
  - 330°F

- **R. V.**
  - 10 psi
  - 240°F

- **S T E A M B O I L E R**
  - 180°F

- **K O H**
  - 170°F

- **C A U S T I C C O N D E N S E R**
  - 24°F

- **V A C U U M P U M P**
Lyle\textsuperscript{118} suggested a closed-cycle arrangement, operating on the same principles as Horigman's cycle, but in which both the dilution and reconcentration halves are directly connected, as shown in fig. 4.9. Coal would be fired in a boiler to raise steam for reconcentration of the dilute caustic. The vapour driven off during reconcentration is used to run an evaporator while the hot concentrated caustic is used to generate steam for possible use in driving machinery.

### 4.4.2. Absorption Refrigeration Cycle

Refrigeration cycles, based on absorption systems, such as NH\textsubscript{3}/H\textsubscript{2}O and LiBr/H\textsubscript{2}O, have also been available for long. These operate essentially on the same principles as the Horigman and Lyle cycles described above and can be adapted to heat pumping duties also. The similarity between this type of device and the closed cycle mechanical heat pump (fig. 4-1) is evident from the cycle arrangement shown in fig 4.10. Essentially the compressor of the mechanical heat pump is replaced by a secondary circuit consisting of an absorber and a generator, in which the absorbent is circulated by a pump.

### 4.4.3 The ERG/ERSU Heat Pump

The Energy Research Group (ERG) of the Open University, in conjunction with the Energy Research Support Unit (ERSU) of Rutherford Laboratories is currently developing an AbHP cycle based on absorption of water vapour on H\textsubscript{2}SO\textsubscript{4}.\textsuperscript{119} A prototype experimental unit consisting only of the absorber/evaporator half of the heat pump has been constructed and demonstrated. A diagram of this is shown in fig. 4.11. Water is fed to the evaporator (vessel 2), where it is vaporised under vacuum by circulating water at about 10\textdegree{}C. The vapour flows through the connecting duct to the absorber (vessel 1) containing concentrated H\textsubscript{2}SO\textsubscript{4} in which it is absorbed, releasing its latent heat and so raising the acid temperature to 50-100\textdegree{}C, as well as diluting it. The dilute acid is collected in a dump tank. The input heat exchanger is used to transfer low grade heat from water in the evaporator, while the output heat exchanger serves to extract the high grade heat generated in the absorber. Having demonstrated the absorption/evaporation half of the cycle, development of the heat pump is now intended to proceed to the reconcentration (regeneration)/condensation half.\textsuperscript{119}
FIG. 4.10 : ABSORPTION HEAT PUMP
Experimental H₂SO₄ heat pumping system

To avoid problems of heat exchange and mixing in static reservoirs, thin surface layers of H₂O and H₂SO₄ are used, fed continuously from the storage vessels.
Other suggestions have been made about various other combinations of chemical substances which could possibly be used in AbHP cycles. Notably, Weiner\textsuperscript{120}, as well as Davis and Manchanda\textsuperscript{77}, have proposed systems with ethylene glycol, diethylene glycol or triethylene glycol as absorbent and water as the volatile substance.

4.4.4 Coefficient of Performance (COP)

The COP of an ideal heat driven AbHP is best illustrated by referring to Carnot cycle diagrams (fig. 4.12). An ideal AbHP is represented by a Carnot heat pump driven by a Carnot engine. The Carnot engine absorbs an amount of heat $Q_R$ at temperature $T_R$ and delivers an amount of work $W$, represented by the shaded portion on the left hand diagram. An amount of heat $Q_{he}$ is rejected at temperature $T_h$. In the heat pump cycle on the right hand diagram, the work $W$ is utilised to extract $Q_c$ units of heat at temperature $T_c$ and deliver $Q_{hp}$ units of heat at temperature $T_h$. The overall COP of the ideal heat driven heat pump is the ratio between the total heat provided for heating and the heat supplied for driving the engine, i.e.

$$\text{COP} = \frac{Q_{he} + Q_{hp}}{Q_R} \quad (4.13)$$

For the Carnot heat engine,

$$W = Q_R \left( \frac{T_R - T_h}{T_R} \right) \quad (4.14)$$

therefore

$$Q_{he} = Q_R - W = Q_R \left( \frac{T_h}{T_R} \right) \quad (4.15)$$

For the Carnot heat pump,

$$Q_{hp} = W \left( \frac{T_h}{T_h - T_c} \right) \quad (4.16)$$

From 4.14 and 4.16,

$$Q_{hp} = Q_R \left( \frac{T_R - T_h}{T_R} \right) \left( \frac{T_h}{T_h - T_c} \right) \quad (4.17)$$
FIG. 4.12: CARNOT CYCLE REPRESENTATION OF HEAT DRIVEN ABSORPTION HEAT PUMP.
Therefore, provided that the work output of the heat engine is the same as the work input of the heat pump \((W)\) and the rejection temperature of the engine and the delivery temperature of the heat pump are equal \((T_h)\), from equations 4.13, 4.15 and 4.17 COP of an ideal heat driven heat pump can be expressed in terms of temperatures thus:

\[
\text{COP} = \left( \frac{T_h}{T_h - T_c} \right) \left( \frac{T_R - T_c}{T_R - T_c} \right)
\]  

Examination of equation 4.18 shows that in general COP will be slightly sensitive to the reservoir temperature \(T_c\) but will depend considerably on \(T_h\) and \(T_R\). Evidently, high COP's will be obtained with large values of \(T_R\) and low values of \(T_h\). For instance, a high COP of 3.25 will be obtained with \(T_R = 200^\circ C\), \(T_h = 50^\circ C\) and \(T_c = 10^\circ C\).

Following the discussions of Section 4.1, it is obvious that real AbHP cycles cannot attain this ideal performance. In a real cycle the heat output consists of the condenser heat and the absorber heat, while the heat input will be the regenerator heat. Since each of these involves change of phase, COP of a real system can be readily estimated. Haselden in considering the LiBr/H\(_2\)O system suggested that typically the heat effects of mixing and dilution which are associated with absorption and regeneration will be about 20% of the latent heat of condensation or vaporisation. Thus, in the absence of other losses, COP would be given by:

\[
\text{COP} = \frac{\text{condenser heat} + \text{absorber heat}}{\text{regenerator heat}} = \frac{2.2}{1.2} = 1.8
\]  

It is clear therefore that even though ideal COP's in excess of 3.0 are potentially possible, in practice COP's will always be less than 2.0. The thermodynamic efficiency of AbHP's can then be expressed as the ratio of the real COP to the ideal COP thus:

\[
\eta_T = \frac{\text{COP}}{\text{COP}_{\text{max}}} = \frac{T_R (T_h - T_c) (\zeta_{\text{cond}} + \zeta_{\text{abs}})}{T_h (T_R - T_c) \zeta_{\text{reg}}}
\]  

where \(\zeta_{\text{cond}}\) is condenser heat

\(\zeta_{\text{abs}}\) is absorber heat

and \(\zeta_{\text{reg}}\) is regenerator heat.
Since the ERG/ERSU $\text{H}_2\text{SO}_4/\text{H}_2\text{O}$ cycle has not been closed its performance cannot be expressed in terms of COP. Alternatively, performance data has been expressed in terms of the rate of energy output from the absorber. An output of 2.5-3 kw. of heat has been reported from test runs on the prototype half-cycle 119.

4.4.5. **Chemical Heat Pump (CHP)**

The chemical heat pump (CHP), a special type of AbHP in which the volatile and absorbing fluid pair is replaced by two materials with stronger chemical affinity which enables them to undergo chemical reaction, is also being developed. In such a system a high temperature endothermic reaction constitutes the generator-condenser combination, while a lower temperature exothermic reaction constitutes the evaporator-absorber combination. Offenherz 122 has proposed a $\text{CaCl}_2/\text{CH}_3\text{OH}$ system in which the following reactions take place:

$$\text{CaCl}_2 \cdot 2\text{CH}_3\text{OH} \overset{120-130^\circ\text{C}}{\longrightarrow} \text{CaCl}_2 + 2\text{CH}_3\text{OH}$$  (generation)

$$\text{CaCl}_2 + 2\text{CH}_3\text{OH} \overset{50-60^\circ\text{C}}{\longrightarrow} \text{CaCl}_2 \cdot 2\text{CH}_3\text{OH}$$  (absorption)

He estimated that the cycle would function with a COP of about 1.5-1.6. Other similar systems which have been suggested include $\text{SnO}_\text{S}/\text{H}_2\text{O}$, $\text{CaO}_\text{S}/\text{H}_2\text{(g)}$ and $\text{CO}_\text{(g)}/\text{H}_2\text{(g)}$ 123.

4.5. **Concept of Adsorption Heat Pump**

Despite the potential advantages of the closed cycle mechanical heat pump dryer, its most obvious limitation is that it cannot be applied to high temperature drying. Considerable problems exist in devising suitable working fluids with such thermodynamic properties as to enable air to be recirculated to the dryer at reasonably high temperatures ($> 100^\circ\text{C}$). Besides, at high temperatures further problems are encountered, as most working fluids:

(i) tend to be chemically unstable,

(ii) readily dissolve in compressor lubricants,

(iii) easily interact with metal surfaces with which they are in contact.
Regarding the open-cycle steam recompression technique, it is obvious that it cannot be feasibly applied to drying below about 150°C since the steam must be substantially superheated. On its part, practical application of AbHP to drying would also be beset by several severe problems, including:

(i) many of the suitable solutions, notably the H₂SO₄, KOH and NaOH are very highly corrosive and therefore create special handling problems;

(ii) the solutions tend to be entrained or volatize, giving rise to liquid losses and problems of contamination in the dryer;

(iii) at high temperatures crystallisation boundaries may be approached leading to solidification problems in the circuit or unfavourably high fluid viscosities may be obtained.

In view of the foregoing limitations an adsorption heat pump (AHP) cycle is proposed here for heat recovery from dryers. The AHP, which is similar to the AbHP and CHP, consists of twin columns of solid adsorbent material, such that while moisture is being adsorbed on one from the dryer exhaust air the other is being desorbed. When the processes come to completion (i.e. the adsorbing bed is used up and the desorbing bed is stripped out), the columns are interchanged and the processes continue, with the stripped bed adsorbing and the used bed desorbing. The adsorption process which is exothermic replaces the absorber while the desorption process replaces the generator of an AbHP. The air is simultaneously heated by the heat generated in the adsorbing column as it is being dehumidified and is recycled to the dryer.
5. ADSORPTION

5.1 Introduction

5.1.1 Nature of Adsorption

Certain solids possess the unique property of preferentially concentrating onto their surfaces one component of a mixture from fluid solutions with which they are contacted. Adsorption is concerned with the exploitation of this property to effect separation of components of a liquid or gas mixture. The adsorptive process is specific and selective in nature. On contacting the solid and fluid phases one or more constituents of the fluid may be selectively removed from the stream while the rest pass through. The solid is referred to as adsorbent while the material adsorbed is the adsorbate.

5.1.2 Types of Adsorption

It is necessary to distinguish between two types of adsorption phenomena:

- physical adsorption
- chemical adsorption (chemisorption).

(i) Physical Adsorption:- This results from intermolecular forces of attraction between molecules of the adsorbent and the adsorbate. The balance of these intermolecular forces is such that adsorbate is transferred from the fluid phase to the adsorbent phase. The process continues until the partial pressure of the adsorbed material equals that of the contacting fluid phase when equilibrium is deemed to be attained. By lowering the pressure of the fluid phase or by raising the temperature the process could be reversed and the adsorbent stripped of adsorbed material.

(ii) Chemisorption :- Unlike physical adsorption, chemisorption results from chemical interaction between the adsorbent and adsorbate molecules and is frequently irreversible. The heat effects associated with this type of adsorption are usually large relative to those associated with the physical adsorption process. Chemisorption is generally unimportant in most industrial adsorption processes but often plays a significant role in catalysis.
5.1.3 Properties of Adsorbents

Adsorbents are generally characterised by a highly porous solid structure, providing large surface areas per unit weight of material. The pores are usually very small, ranging from about $0.1 \mu m$ to $50 \mu m$, but their large numbers furnish enormous surface for adsorption. The performance of an adsorbent does not only depend on the nature of its constituent material, it also depends substantially on the method by which it is manufactured as its crystalline structure and particle size are affected by the method of manufacture. Performance may also depend on the previous history of an adsorbent since most are prone to deterioration with constant use. Adsorbents are made in three forms - powder, granules and beads. The powder has a very high specific surface area but gives rise to excessive pressure drops when packed in beds and is prone to dusting and entrainment. Though the beads on the other hand possess relatively low contact area, they are usually hard and rigid, enabling them to provide desired structural strength to withstand attrition. In general, the granular form possesses properties which lie between those of the powder and the beads. This commends it specially for widespread industrial use, in particular for operations involving gases.

The main adsorbents in general use industrially include clays, bone char, activated carbon, activated alumina, silica-gel and molecular sieves (synthetic zeolite). Table 1 below shows the essential properties of three of these adsorbents (granular form).

<table>
<thead>
<tr>
<th>PROPERTIES</th>
<th>Activated Carbon</th>
<th>Activated Alumina</th>
<th>Silica Gel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Specific surface area ($m^3/kg \times 10^{-3}$)</td>
<td>1100</td>
<td>390</td>
<td>850</td>
</tr>
<tr>
<td>Pore volume ($m^3/kg \times 10^3$)</td>
<td>0.56</td>
<td>0.35</td>
<td>0.46</td>
</tr>
<tr>
<td>Average pore climate ($\AA$)</td>
<td>19</td>
<td>48</td>
<td>22</td>
</tr>
<tr>
<td>Bulk Density ($kg/m^3$)</td>
<td>480</td>
<td>800</td>
<td>720</td>
</tr>
<tr>
<td>Specific Heat ($kJ/kg\cdot{^\circ} C$)</td>
<td>0.84</td>
<td>0.92</td>
<td>1.05</td>
</tr>
</tbody>
</table>
5.2 Adsorption Equilibrium Relations

The capacity of an adsorbent for an adsorbate is limited by the static equilibrium relationship between the concentration of the adsorbate in the fluid phase and in the adsorbed phase. Static equilibrium in the system can only be attained if contact between the two phases is sustained indefinitely. In practice, when fluid flows continuously through a fixed bed of adsorbent static equilibrium cannot be achieved but rather dynamic equilibrium is established.

5.2.1 Presentation of Equilibrium Relations

The loading of adsorbate in the adsorbent at equilibrium is believed to be a function of the partial pressure or concentration of adsorbate in the fluid and the prevailing temperature only,

\[ W = f(p, T) \quad \text{(5.1)} \]

where \( W \) = loading,
\( p \) = partial pressure
and \( T \) = temperature.

When \( p \) is varied and \( T \) kept constant, a plot of \( W \) against \( p \) is called the adsorption isotherm (fig. 5.1),

\[ W = f(p) \quad T = \text{constant} \quad \text{(5.2)} \]

When \( p \) is constant and \( T \) is varied, the plot is the adsorption isoobar (fig. 5.2)

\[ W = f(T) \quad p = \text{constant} \quad \text{(5.3)} \]

The plot of \( p \) versus \( T \) at constant loading is the adsorption isostere (fig. 5.3).

\[ p = f(T) \quad W = \text{constant} \quad \text{(5.4)} \]

The presentation of adsorption equilibrium data can be simplified by plotting the loading against the relative humidity of the adsorbate in the fluid (fig. 5.4), such that relative humidity is defined by the ratio

\[ X = \frac{p}{p_o} \quad \text{(5.5)} \]

where \( p_o \) is saturation vapour pressure of the adsorbate at the prevailing temperature. This, therefore, combines the effects of \( p \) and \( T \) on \( W \).
FIG. 5.1 - ADSORPTION ISOTHERMS

FIG. 5.2: ADSORPTION ISOBERS

FIG. 5.3: ADSORPTION ISOSTERES

FIG. 5.4: LOADING vs RELATIVE HUMIDITY
5.2.2 Monolayer Theory

The monolayer theory is one of the theories that have been deduced to explain the mechanism of adsorption. It postulates that in the process of adsorption adsorbate molecules are retained as a surface layer on the adsorbent, one molecule thick. For very low concentrations of adsorbate in fluid, it is believed that the isotherm may be linear, similar to Henry's law equation. Then,

\[ W = H y \]  \hspace{1cm} (5.6)

where \( y \) = adsorbate concentration in fluid

and \( H \) = Henry's constant.

Following his empirical studies, Freundlich\(^{52}\) suggested that for monolayer adsorption over a wide range of \( y \), the isotherm is such that it is concave to the \( y \)-axis (fig. 5.5) and hence he proposed an equilibrium relationship of the form

\[ W = K y^n \]  \hspace{1cm} (5.7)

where \( K, n \) are empirical constants.

\[ \text{FIG. 5.5: FREUNDLICH'S ISOTHERM} \]
Consequently, the Henry's Law relationship can be regarded as a special case of Freundlich's exponential equation, when $n = 1$. Freundlich's equation has been found to be more appropriate for correlating the equilibrium adsorption of liquids than vapours.

From theoretical considerations Langmuir\textsuperscript{53} developed an alternative equation for this isotherm:

$$ W = \frac{W_m K y}{1 + K y} \quad (5.8) $$

where $W_m$, $K$ are constants.

On comparing experimental values for adsorption of various vapours with calculated values using Langmuir's and Freundlich's equations better agreement was obtained with the former. However, at higher pressures experimental values fail to agree so well with Langmuir's equation.\textsuperscript{50}

5.2.3 Multilayer Theory

It has been observed that in some cases the adsorption curve is S-shaped and increases to infinity as the partial pressure approaches the vapour pressure.\textsuperscript{55} This has been explained by the formation of a multimolecular layer of adsorbate molecules, of definite thickness. This type of equilibrium has been observed for the adsorption of water vapour on carbon black and of nitrogen on iron catalyst\textsuperscript{59} and is similar to the monolayer curve for low and intermediate partial pressures. Brunauer, Emmett and Teller\textsuperscript{56} derived an isotherm equation for multimolecular layers that is similar to Langmuir's equation for unimolecular layers thus:

$$ W = \frac{W_m C y \left[1-(n+1)y^n+ny^{n+1}\right]}{1-y \cdot \left[1+(c-1)y-Cy^{n+1}\right]} \quad (5.9) $$

where $n$ = number of layers

- $C$ is a constant given by $C = \frac{e^{(E_1-E_L)/RT}}{RT}$
- $E_1$ is heat of adsorption
- $E_L$ is normal heat of condensation
- $R$ is $\text{g} \cdot \text{s}^{-1}$ constant
When \( n = \infty \) equation (5.9) reduced to
\[
W = \frac{W_m C_y}{(1-y)[1+(c-1)y]} \tag{5.10}
\]
In the other extreme, when \( n = 1 \) (only one layer adsorbed on surface) equation (5.9) reduces to Langmuir's equation (5.8).

5.2.4 Capillary Condensation

Following his studies on the adsorption of water on silica-gel, Zsigmondy\(^5\) concluded that the adsorption process is a function of the capillary properties of silica-gel. This provided the basis for the theory of capillary condensation which has been used to explain certain adsorption phenomena. Provided that the fluid wets the adsorbent, leading to the formation of a concave meniscus, the vapour pressure \( p_c \) over the condensate confined in a capillary is related to the saturation vapour pressure \( p_v \) in free space, at the same temperature, by the Kelvin expression:
\[
\ln \frac{p_c}{p_v} = - \frac{2 \sigma V}{rRT} \tag{5.11}
\]
- \( \sigma \) is surface tension of condensate
- \( V \) = molecular volume in the liquid state
- \( r \) = radius of capillary
All the quantities of the right hand side of equation 5.11 are positive, and hence

\[ P_C < P_0 \]

On contacting the fluid and adsorbent it is thought therefore that adsorbate condenses in the capillary pores of the adsorbent, provided the pore radii are less than that calculated for \( P_c \). It is believed that usually capillary condensation augments the adsorption taking place on flat surfaces of adsorbents, and the loading attains a finite maximum value at a relative humidity of unity, as shown in fig. 5.7.

On examination of the characteristic equilibrium curve for capillary condensation augmented adsorption, it could be observed to consist of a combination of two Langmuir type processes, as was suggested by Rabinowitch and Wood. This has led to the development of the Double-Langmuir form of adsorption equilibrium expression thus:
\[ W = \frac{a_1 b_1 X_{f_1}}{1 + b_1 X_{f_1}} + \frac{a_2 b_2 X_{f_2}}{1 + b_2 X_{f_2}} \]  

(5.12)

\[ X_f = \left( \frac{p}{p_0} \right)^{\gamma/A} \]  

(5.13)

\( a_1, b_1, a_2, b_2 \) and \( A \) are constants.

5.2.5. Water Adsorption Equilibrium

Several researchers have investigated the important special case of the equilibrium relations in the adsorption of water vapour on various adsorbents. Working in the temperature range 18-38°C and saturation 5-50%, Carter and Barret\textsuperscript{59} correlated the equilibrium for adsorption of moisture from air using activated alumina with:

\[ W = 0.143 \left( \frac{Y}{y_{sat}} \right)^{0.46} \]  

(5.14)

where \( y_{sat} \) is saturation humidity of air at average temperature of bed. On bead type silica-gel, they obtained:

\[ W = 0.62 \left( \frac{Y}{y_{sat}} \right) \]  

(5.15)

and on molecular sieve:

\[ W = 0.203 \left( \frac{Y}{y_{sat}} \right)^{0.116} \]  

(5.16)

Jury and Horng\textsuperscript{63} working over a wider range of conditions, obtained a B.E.T. type multilayer adsorption equation for activated alumina:

\[ W = \frac{a_1 b_1}{1-X_f} \left\{ \frac{X_f}{1 + (b_1-1) X_f} \right\} \]  

(5.17)

For silica-gel, Jury and Edwards\textsuperscript{60} fitted the data with a Double-Langmuir type expression similar to equation, 5.12. Comparing values obtained using their equation with the experimental equilibrium data obtained by Hubard\textsuperscript{61} for the same system, they obtained an agreement to within 3-5%. Working with locally made samples of silica-gel, Mukesh and Rao\textsuperscript{12} also fitted the equilibrium data for adsorption of moisture from air with a Double-Langmuir expression, similar to that obtained by Jury and Edwards.
Simpson and Cummings\textsuperscript{64} on their part proposed a pair of linear equations for the water/silica-gel system, thus:

\begin{align*}
W &= 1.567 + 0.5836X \quad (0 < X < 0.50) \\
\text{and} \quad W &= 24.63 + 0.136X \quad (0.50 < X < 1.00)
\end{align*}

(5.18a) \quad (5.18b)

5.2.6. **Hysteresis**

Certain adsorbents, especially those that possess high degrees of capillarity, are such that the equilibrium relationship obtained in adsorption is not exactly the same as that obtained in the reverse process of desorption, as shown in fig. 5.8. This phenomenon is referred to as hysteresis. For such materials therefore, different equations need be obtained for the forward and reverse processes. Jury and Edwards\textsuperscript{160} have noted however that for water/silica-gel system the hysteresis effect is negligible and the equilibrium equation is applicable to the two processes.
5.3 Breakthrough Phenomenon

When a fluid containing an adsorbable component is continuously passed through an adsorption column, the layer of adsorbent directly in contact with inlet air at first adsorbs the adsorbate rapidly and effectively. The adsorbate concentration in the effluent air at the outlet end of the bed is almost, if not completely, removed. The layer of adsorbent at the inlet end is practically saturated while the rest of the bed is virtually unused. It is believed that the bulk of the adsorption takes place over a relatively narrow 'adsorption zone' or 'mass transfer zone' (MTZ), in which the adsorbate loading in the adsorbent changes rapidly. As the fluid continues to flow through the bed the adsorption zone travels as a wave along the bed, though at a much slower velocity relative to the air. This continues until the front end of the adsorption zone reaches the outlet end of the bed. There is a sudden rise in the adsorbate concentration in the effluent fluid and the system is then deemed to have attained breakpoint. The effluent concentration subsequently rises rapidly until the initial concentration of the feed is attained. Then, the entire bed is saturated and there can be no more adsorption. The change in effluent concentration with time can be represented by a breakthrough wave as shown in fig. 5.9 below.

Since adsorption is exothermic, the associated heat causes a temperature wave to flow through the bed in a way similar to the adsorption wave, but with the latter lagging the former. The fluid, in contact with the adsorbent, takes away some of the evolved heat, thereby undergoing a temperature increase also. If heat transfer conditions are favourable and the system attains thermal equilibrium the temperatures of the air and the adsorbent at the adsorption zone (MTZ) will become equal.

The temperature of effluent air reaches a maximum and stays fairly constant while the temperature of the bed depends on the position of the wave front. The attainment of breakpoint is preceded by a drop in effluent temperature and peaking of the temperature of the layer of adsorbent at the outlet of the bed, since the adsorption wave lags the
FIG. 5.9: THE ADSORPTION WAVE AND BREAKTHROUGH
FIG. 5.10: BREAKTHROUGH TEMPERATURE AND CONCENTRATION CURVES

Time 0

Concentration

Temperature

FIG. 5.11: ADSORPTION BED TEMPERATURE PROFILES

Time 0

Temperature

FIG. 5.12: DEPLETION TEMPERATURE AND CONCENTRATION CURVES

Time 0

Concentration

Temperature

FIG. 5.13: DESORPTION BED TEMPERATURE PROFILES

Time 0
temperature wave. Fig 5.10 shows characteristic effluent fluid concentration and temperature curves for the process. If bed temperature is measured along the length of the bed over a time interval, the typical bed temperature profile that would be expected is shown in fig. 5.11. Similar curves to those shown in figs 5.10 and 5.11 have been obtained by several investigators working on the adsorption of water vapour from air using different adsorbents.

The regeneration process is roughly the reverse of adsorption. The depletion point is deemed to be attained when the adsorbate concentration in the effluent from the desorbing column becomes essentially negligible and remains fairly steady at that level subsequently. Depletion would normally be attended by an increase in temperature of the effluent and of the adsorbate to maximum levels, at which they remain fairly steady. Typical desorption concentration and temperature curves, similar to those obtained by Der
t et al and by Zwiebel et al, for regeneration using hot, dry air, are shown in figs. 5.12 and 5.13.

5.4 Kinetics of Adsorption

The fixed-bed adsorption breakthrough curve may be predicted from the basic system kinetics and equilibrium. Such mathematical analysis essentially consists of formulation and solution of equations for the conservation of mass, energy and momentum. Given the complexity of the processes that are associated with adsorption, a number of simplifying assumptions are often made in order to facilitate analytical treatment. These include:

(1) The fluid mixture is dilute, implying isothermal operation, since amount of material is assumed to be small enough for associated heat effects to be neglected.

(2) Pressure drop through adsorbent bed is small, therefore the velocity and density of the fluid are constant.

(3) Adsorbent particles are of regular shape and sufficiently small so that the change in fluid-phase concentration over a length equal to diameter of a particle can be neglected.

(4) Fluid flows in the axial direction, such that the velocity profile is flat and longitudinal diffusion is negligible.
(5) Both solid and fluid phases have radial symmetry.

(6) There is no direct interparticle mass transfer.

As a result of assumptions (1) and (2), energy and momentum balance equations can be excluded from the analysis. Subject to these assumptions, for a single adsorbable component fluid mixture, the differential mass balance for an element of the column will be:

\[ \nu \frac{\delta y}{\delta z} + \frac{\delta y}{\delta \Theta} + \left( \frac{1-e}{e} \right) \frac{\delta w}{\delta \Theta} = 0 \]  

(5.19)

where \( \nu \) = fluid flow velocity  
\( z \) = distance along the column length  
\( \Theta \) = time  
\( e \) = bed void fraction.

The initial and boundary conditions for a step change in feed at time zero would be:

\[ y \left( Z = 0, \Theta \right) = y_0 \]  

(5.20a)

\[ y \left( Z, \Theta =0 \right) = 0 \]  

(5.20b)

\[ w \left( Z, \Theta =0 \right) = 0 \]  

(5.20c)

The derivative \( \delta w/\delta \Theta \) in equation (5.19) represents the rate of mass transfer from the fluid to the adsorbent. To evaluate this rate, knowledge of the controlling resistance to mass transfer is essential. The process may be visualized as one consisting of the following steps:

(1) transport of the adsorbate from the bulk fluid phase to the interface of a stagnant fluid film surrounding the adsorbent particle;

(2) external mass transfer through the stagnant film to the external surface of the particle;

(3) internal diffusion within the particle pores;

(4) adsorption on particle active sites;

(5) reaction with active sites.
Chemical reaction at the active site is often irrelevant to most adsorption processes, except when chemisorption is of importance. However, even for processes that do not involve chemisorption, Thomas proposed a model which involves the assumption of an imaginary chemical reaction as the rate limiting step. Particularly for systems exhibiting Langmuir or linear isotherm, this results in rate expressions of relative mathematical simplicity. It is also believed that in practice, transport in the bulk fluid phase (step 1) is usually extremely rapid and may be assumed without serious error to be instantaneous. Any of the other three steps or a combination of them may therefore be important and thus constitute the rate limiting step.

5.4.1 External Diffusion Control

When external film diffusion is the rate limiting step, the external film mass transfer coefficient \( k_f \) may be regarded as an empirical lumped parameter approximation to the overall resistance. The rate expression can then be written as:

\[
\frac{dw}{dt} = k_f \ a \ (y - y^*)
\]  

(5.21)

where \( a \) is specific surface area for mass transfer and \( y^* \) is fluid-phase concentration in equilibrium with the average solid-phase concentration. \( y \) is related to \( W \) by a suitable equilibrium equation of the form:

\[
W = f(y^*)
\]  

(5.22)

as discussed in Section 5.2 above.

Solving equations 5.19 - 5.22 provides the breakthrough curve for external diffusion control. Hougen and Marshall obtained a solution for the case of a linear isotherm type, which they expressed in form of Bessel functions. The corresponding problems, with Langmuir and Freundlich type isotherms, have also been solved.
Following their theoretical and experimental studies, Carter and Barrett found that for silica-gel adsorbent, external diffusion constitutes the controlling resistance, with pore diffusion contributing less than 10% to the adsorbed phase transport.

5.4.2 Internal Pore Diffusion Control

If the controlling resistance is the internal diffusion in the pores of the adsorbent then equations 5.21 and 5.22 may be replaced by equations 5.23 and 5.24 respectively:

\[
\frac{dw}{dt} = k_s \cdot a \cdot (w^* - w) \quad (5.23)
\]

\[
w^* = f(y) \quad (5.24)
\]

where \(k_s\) is the solid-film mass transfer coefficient and \(w^*\) is solid phase concentration in equilibrium with the average fluid phase concentration.

For non-linear isotherms and intraparticle-diffusion control, several researchers have considered prediction of the breakthrough curve, assuming the intraparticle-diffusion coefficient \(k_s\) to be independent of adsorbate concentration. However, for many systems \(k_s\) could be highly concentration dependent. Glueckauf outlined a method for estimating \(k_s\) for such systems if basic diffusivity data are available. Solutions of the breakthrough curve have been presented for intra-particle diffusion control, taking into account the concentration dependence of diffusivity and non-linear equilibrium relationships, by Garg and Ruthven and by Carter and Husain. The internal diffusion step is believed to constitute the rate controlling step for most activated alumina and molecular sieve adsorbents.

5.4.3 Physical Adsorption Control

Vermulen considered the case of physical adsorption control. Based on Thomas's original theory of fixed bed performance, he assumed that physical adsorption can be represented by a second order surface kinetics of the form:

\[
A + \text{sorbent} \rightarrow A, \text{Sorbent}
\]

and proposed solutions to the model, applicable to systems with both
linear and Langmuir type isotherms. More recently, Masamune and Smith have also outlined a solution based on the same rate controlling mechanism.

5.4.4 Combined Resistances

For some systems more than one of the adsorption steps make significant contributions to the overall mass transfer process and no single step can therefore be deemed to be the controlling step. Possibly, any combination of two of the three major steps could be controlling or even a combination of the three. Rosen obtained a solution for the case of combined external and internal diffusion control. Masamune and Smith, on the other hand, solved the equations for all three possible two-resistance cases. They also presented a solution for the combined three-resistance case. Davis and Manchanda, on their part, suggested that for any combination of resistances, an overall mass transfer coefficient calculated from the additive law of resistance, could be used to solve the equation. Though not strictly correct this has proved a useful practical tool.

5.5 Adiabatic Adsorption

The assumption of dilute adsorbent concentration in the fluid for the description of the adsorption process implies that the associated heat effects are insignificant and hence the process could be taken to be isothermal. However, if the fluid concentration is high, substantial heat effects may be involved. In the limiting case, if all the heat generated is retained in the system, then the process is deemed to be adiabatic. Assuming adiabatic conditions and subject to the other restrictions previously outlined, the differential fluid phase and adsorbent-phase heat balances for an element of an adsorption bed will respectively be:

$$v \frac{c_p}{\delta Z} \frac{\delta T_f}{\delta Z} + \frac{c_p}{\delta \Theta} \frac{\delta T_f}{\delta \Theta} = \frac{h}{\rho_f} (T_s - T_f) \quad (5.25)$$

and

$$\left(1 - \epsilon \right) \frac{c_p}{\rho_s} \frac{\delta T_s}{\delta \Theta} = -h (T_s - T_f) + r \lambda \lambda \quad (5.26)$$
where

\[ \lambda_A \] is heat of adsorption

\[ r_A \] is rate of adsorption

\[ \rho \] is specific gravity

\[ c_p \] is specific heat

and subscripts \( f \) and \( s \) refer to fluid and solid respectively.

The solution of the conservation equations for the adiabatic case has generally received less attention than the isothermal case. Hougen and Marshall\(^7\) extended their consideration of isothermal adsorption to the adiabatic case and proposed a graphical method of obtaining the time-position-temperature conditions in both fluid and solid phases during adsorption. Acrivos\(^8\) considered heat and mass transfer problems in fixed bed operations and proposed in general terms the possibility of using the method of characteristic technique for adiabatic adsorption. Sanlaville studied the heat and mass transfer for air drying over activated alumina and concluded that for short beds the process is controlled by the rate of heat transfer. Carter\(^8\) considered adiabatic adsorption for the case when mass transfer is limited both by external and internal diffusion, heat transfer is controlled by external diffusion, and for systems exhibiting linear isotherm. He outlined a numerical method for solving the equations.

In many practical situations, actual adiabatic conditions are not attained owing to heat losses to the surroundings and absorption of heat by the walls of the adsorber column. Most adsorption systems in practice are therefore deemed to operate under semi-adiabatic or non-isothermal conditions. Adiabatic conditions are most closely approached in large diameter beds, with low thermal conductivity of vessel wall and low fluid residence time. Meyer and Weber\(^9\) investigated the removal of methane from helium by non-isothermal adsorption on activated carbon experimentally and theoretically. The mathematical model, applicable to systems with non-linear, Langmiur type isotherms as well as internal and external diffusion control, was solved numerically and the solutions were found to agree measurably well with experimental data.
5.6 Desorption

The regeneration of a spent bed at the end of adsorption can be achieved by one of several methods, which include:

- heat conduction
- heat convection
- heatless

Heat Conduction: Coils, usually heated electrically, are placed in the adsorbent bed to provide heat for regeneration. A flow of dry air is maintained through the bed, to sweep out water vapour in the course of regeneration.

Heat Convection: An external heater is used to heat up dry air before it is blown through the adsorbent bed. The hot air transfers heat to the adsorbent by convection, providing the required latent heat for evaporation of moisture and simultaneously sweeping out the generated water vapour.

Heatless: This is accomplished without heating the adsorbent bed. Operation is based on maintaining a lower pressure in the desorbing half of the cycle relative to the adsorbing half. The moisture evaporated, following the lowered pressure, is removed in a stream of dry regenerating air flowing through the bed.

Desorption, using a heated flow or a flow containing a low concentration of adsorbate or a combination of both, can be described essentially by the same type of heat and mass transfer equations as used for the adsorption process, which can be solved using the appropriate initial and boundary conditions:

\[ y (z = 0, 0) = 0 \]  \hspace{1cm} (5.27a)

\[ y (z, \theta = 0) = y_o \]  \hspace{1cm} (5.27b)

\[ w (z, \theta = 0) = w_o \]  \hspace{1cm} (5.27c)
Using the external film model to describe the mass transfer mechanism and Langmuir type isotherm to account for the phase equilibrium. Zwiebel et al.\(^6\) developed a model for the desorption process. From their solution, they observed the non-symmetrical nature of the depletion curve relative to the breakthrough curve and imputed this primarily to effects associated with isotherm non-linearity and to a less extent on the prevailing temperature effects. Besides, differences between the breakthrough and depletion curves may also be related to hysteresis effects, since separate isotherm equations may be required for both processes. Carter\(^9\) obtained a similar model for the system and showed that the shape of the depletion curve could be affected by such factors as extent of saturation of bed before desorption, temperature and flow of desorbing air. Garg and Ruthven\(^7\) as well as Ruthven\(^9\) have also variously considered the kinetics of desorption for both linear and non-linear isotherms.

5.7 MTZ Approach

In fixed bed sorption processes, it is believed that as the mass transfer zone (MTZ) moves through the sorption bed, it may approach a steady-state length after an initial period of expansion or it may continue to expand without limit. If the MTZ attains a steady-state length it is said to be stable.\(^7\),\(^9\) Considerable attention has been devoted by several investigators to the application of this concept of stable MTZ to the quantitative description of sorption processes. Of very significant importance is the mathematical approach originally developed by Michaels\(^9\) (1952) for fixed-bed ion-exchange and subsequently adapted to fixed-bed adsorption by Treybal\(^2\). In addition to the general simplifying assumptions that could be applicable even in unsteady state situations, outlined earlier, the method is based on the following further restrictions:

(i) stable MTZ exists,
(ii) dilute concentration of adsorbate in feed,
(iii) isothermal conditions,
(iv) favourable isotherm, concave to the fluid concentration axis,
(v) height of adsorbent bed is large relative to height of adsorption zone.
At breakpoint, the bed is considered to consist of a saturated zone in equilibrium with the inlet air and the MTZ in which the adsorbent is only partially used. If breakthrough data is available, Treybal outlined a procedure for estimating the length of an MTZ \( L_a \) based on the notion of number of transfer units and height of a transfer unit in the zone, and for obtaining the fraction of the adsorbent in the MTZ that is unused (i.e. free of adsorbate) \( f_z \). He then expressed the overall loading of adsorbent on the adsorbate at breakpoint \( W_b \) as:

\[
W_b = W_e \left( \frac{Z - f_z Z_a}{Z} \right)
\]

where \( W_e \) is equilibrium loading in the saturated zone and \( Z \) is total height of adsorption bed.

Nutter and Burnet\(^{94} \) investigated the application of the MTZ concept to isothermal adsorption of water from air using molecular sieve. By combining a breakthrough equation, based on solid shell resistance, proposed by Eagleton and Bliss\(^{97} \) and an empirical expression obtained from their experimental data, they derived a relationship between inlet air humidity and height of MTZ for the system. In examining the isothermal drying of low humidity air using silica-gel, Simpson and Cummings\(^{64} \) proposed that to predict the bed loading at breakpoint, a combination of equations would be needed and they obtained appropriate empirical equations for water/silica-gel to meet this need. One would predict \( W_e \), another would predict \( Z_a \) and a third would predict the loading in the MTZ.

Closely related to the idea of MTZ, Collins\(^{95} \) enunciated the concept of length of unused bed/equilibrium section, for isothermal fixed-bed adsorption. He considered that at breakthrough the adsorption bed would consist of an equivalent length of unused bed (LUB) portion which compensates for the presence of an MTZ, and an equilibrium section with a uniform loading. Hence if length of the equilibrium section is \( Z_e \), then:

\[
Z = Z_e + \text{LUB}
\]

(5.29)
If breakthrough data is available, he outlined a procedure for calculating $Z_e$, and LUB, thus enabling $W_b$ to be easily obtained.

In a remarkable contribution, Leavitt\textsuperscript{96} extended the MTZ approach, which had hitherto almost exclusively been confined to isothermal conditions, to adiabatic fixed bed adsorption. In deriving his equations he made the usual assumptions regarding the stability of the MTZ and its height in relation to the bed height but he was not limited by such other restrictions as low adsorbate concentrations and favourable isotherms. He postulated that during adiabatic adsorption two transfer zones tend to be formed. These zones are deemed to arise at the entrance end of the bed and move towards the exit end at different speeds. Given sufficient time the faster one would be separated from the slower one and pull well ahead of it. Between them a long expanding region which he referred to as the inter-zone would be formed, in which concentration and loading are uniform and the loading is essentially in equilibrium with the local concentration at the local temperature. He then derived heat and mass balance equations by which he related the inter-zone conditions to the conditions in the end regions. The region ahead of the faster zone is deemed to be at equilibrium with the outlet air while the region preceding the slower zone is at equilibrium with the inlet air.

Leavitt experimentally investigated the adsorption of carbon dioxide from a nitrogen stream by molecular sieve in a 5 ft. high and 8 ins. diameter column. His results showed the occurrence of two breakthroughs, therefore corroborating the formation of two transfer zones. Further evidence of the occurrence of a double breakthrough phenomenon has also been provided by Getty and Armstrong\textsuperscript{98} following their experiments on adiabatic adsorption of moisture from air on silica-gel adsorbent.
5.8 Other Semi Steady-State Methods for Non-Isothermal Adsorption

The KTV approach employs the use of the concept of breakpoint capacity and presumes the availability of breakthrough data. When such data are not available several other semi-steady-state methods, similarly employing the notion of breakpoint capacity, have also been proposed.

5.8.1 Breakpoint Capacity

Lightfoot et al.\(^71\) proposed a method of predicting breakpoint capacity \(W_b\) based on an equilibrium model, for adiabatic adsorption and outlined a graphical solution using an enthalpy-concentration diagram. However, since enthalpy-concentration data for most systems is most often not available the method has been of limited application.

Lee and Cummings\(^99\) outlined a more general procedure in which they essentially sought to correlate between analytical solutions for isothermal adsorption and experimental data from non-isothermal adsorption, in order to predict \(W_b\) under non-isothermal conditions. Since adsorption on silica-gel is controlled by external diffusion, by approximating the water/silica-gel non-linear equilibrium relation by a series of straight line sections of different gradients, they obtained an analytical solution for the system using the Hougen-Marshall model\(^72\). Thus they were able to calculate breakpoint and equilibrium times for isothermal operations, \(\theta_b\) and \(\theta_e\) respectively. They considered that owing to non-isothermal conditions the adsorbent would undergo a decrease in capacity, which they expressed by a dimensionless time ratio \(a\) defined by:

\[
a = \frac{\theta_e - \theta_{bi}}{\theta_e - \theta_{bi}}
\]  

(5.30)

where \(\theta_{bi}\) is breakpoint time for non-isothermal adsorption.

Following experimental studies on non-isothermal air drying over silica-gel in a 6 ft. high, 4 ins. diameter bed, they subsequently obtained a correlation between \(a\) and air inlet humidity \(y_i\) thus:
\[
a = 1.0 + 1.66 y_1^{-0.87} \quad (y_1 < 0.1) \quad (5.31a)
\]
\[
a = 3.0 \quad (y_1 > 0.1) \quad (5.31b)
\]

Dehler\(^{100}\), on his part, proposed that breakpoint capacity could be obtained as the mean of the equilibrium loadings corresponding to the inlet and outlet fluid streams. If adiabetic conditions obtain, he suggested that allowance would be made for the temperature effect by assuming that it modifies the relative humidity of the exit fluid.

5.8.2 Effects of Operating Variables on \( W_b \)

Several investigators have studied the effects of different operating variables on break-point capacity. Grayson\(^{68}\), as well as Getty and Armstrong\(^{98}\), working on adsorption of water vapour from air on bead silica gel and activated alumina respectively showed that increasing depth of adsorption bed increases breakpoint capacity while increasing air flowrate tends to reduce breakpoint capacity. Simpson and Cummings\(^{64}\), and Worthington\(^{67}\), have also presented results that show an adverse effect of flowrate on \( W_b \). On the other hand, Moss and MacLaughlin\(^{101}\), and Miller and Roberts\(^{102}\), produced experimental data for adsorption on silica gel and alumina respectively, which indicate that breakpoint capacity would be independent of air flowrate.

Worthington\(^{67}\) investigated the influence of inlet humidity on the breakpoint capacity during non-isothermal adsorption of moisture from air on silica-gel. He concluded that higher humidities lead to reduced capacities. Grayson\(^{68}\) arrived at a similar conclusion following his experiments using bead type silica-gel. In addition, he indicated that under certain conditions, such as large bed depths, the reverse effect might also be obtained. Miller and Roberts\(^{102}\) on the other hand, suggested that breakpoint capacity would be independent of inlet absolute humidity.

Defining the breakpoint bed relative humidity \( X_b \) as the ratio of adsorbate partial pressure in the feed to the saturation vapour pressure of the adsorbate at the mean temperature of the bed during operation, it has been suggested that a relationship exists between \( X_b \) and \( W_b \). For the water/activated alumina system, Miller and Roberts\(^{102}\)
FIG. 5.14: MILLER/ROBERTS DATA

FIG. 5.15: WORTHINGTON'S DATA
as well as Coulson and Richardson\textsuperscript{103} correlated $X_b$ and $W_b$ with a linear relationship. The data of Miller and Roberts are reproduced in fig. 5.14. Worthington obtained data for water/silica-gel, reproduced in fig. 5.15, which indicate a non-linear relationship between $X_b$ and $W_b$.

5.8.3. Temperature Change

Since the temperature wave is believed to precede the adsorption wave, it may be considered that while the bed attains mass transfer breakpoint, thermal equilibrium is established. This implies that heat transfer between the fluid and the adsorbent is substantially complete at breakpoint and thus the fluid and the solid attain about the same average temperature\textsuperscript{65,102}. Based on this, Miller and Roberts derived an equation for the temperature increase associated with non-isothermal adsorption, thus:

$$\Delta T = \frac{W_b \lambda A \Delta y}{W_b c_p + c_p \Delta y}$$  \hspace{1cm} (5.32)

where $\Delta y$ is change in adsorbate concentration.

If the influent fluid is relatively dilute,

$$W_b c_p \gg c_p \Delta y$$

and hence,

$$\Delta T = \frac{\lambda A \Delta y}{c_p}$$  \hspace{1cm} (5.33)

Equation 5.33 has been proposed by Carter\textsuperscript{65} and by Coulson and Richardson\textsuperscript{103}.

Based on his experimental studies on adsorption of water on silica-gel, Worthington\textsuperscript{67} suggested the following empirical equation for obtaining $\Delta T$ for the water/silica-gel system:

$$\Delta T = 70 h_{1000}$$

where $h_{1000}$ is the moisture adsorbed in lbs/1000 ft\textsuperscript{3} air and $\Delta T$ is temperature rise in °F.

Similarly, Mantell\textsuperscript{50} suggested that the adsorption of 4.9 grains of water on silica-gel results in a temperature rise of about 10° F. For the same system, Rathmell and Bateman\textsuperscript{106} obtained the following relation:
\[ \Delta T = 2950y_i - 3.5 \quad (5.35) \]

provided that inlet humidity \( y_i \) lies in the range \( 0.003-0.015 \) kg/kg and \( \Delta T \) is expressed in °C. For water/activated alumina, and within the same range of inlet humidity they obtained:

\[ \Delta T = 3500y_i \quad (5.36) \]

5.8.4 Regeneration

Fre9 studied the conductive regeneration of silica-gel and alumina beds using embedded electrical heating coils extensively and suggested that the desorption requirements can be predicted from the movement of the desorption zone. He was able to correlate the speed of the desorption zone with the purge air flowrate for both types of adsorbent using linear relationships. For convective regeneration, Carter65, Mantell50, as well as Hougen and Dodge104, have outlined closely related calculation procedures based on heat balances. The heat requirements for regeneration are deemed to comprise essentially:

(i) heat for vaporizing adsorbate at the temperature at which most of the desorption takes place;

(ii) sensible heat required to heat the bed to the regeneration temperature.

In convective regeneration, these heat demands are evidently provided by the sensible heat loss of the desorbing air. Geankopolis105 has suggested that unlike in drying of solids in shallow beds, the temperature at which regeneration of thick beds takes place may be considerably higher than the air wet-bulb temperature. Rathmell and Bateman106 indicated that this temperature, and hence the humidity of the effluent air, are directly related to the initial capacity of the used bed and the results presented by Worthington67 for regeneration of silica-gel beds using hot, dry air in the temperature range 100-250°C agreed substantially with this.
CHAPTER SIX

THE ADSORPTION HEAT PUMP (AHP)
6. THE ADSORPTION HEAT PUMP (AHP)

6.1. System Description

The AHP may be operated either on a:

(i) Temperature-Swing Cycle (TSC), or
(ii) Pressure-Swing Cycle (PSC).

6.1.1. TSC

Operating on a TSC, adsorption takes place at low temperature while desorption takes place at an elevated temperature, both columns being maintained at the same pressure, usually atmospheric. The dryer exhaust air is passed into the adsorption bed directly. A portion of the dehumidified air is returned to the dryer while the rest is heated to a high temperature and used to regenerate the other bed. A schematic diagram of the cycle is presented in fig. 6.1, while the successive steps involved in the cycle are illustrated on a p-T diagram in fig. 6.3.

6.1.2. PSC

This is slightly more complicated than the TSC in that it incorporates a compressor, a cooler and an expansion valve. The dryer exhaust air is compressed and passed through the cooler, where some of the moisture is condensed and removed, before it goes to the adsorption column. After adsorption the air is expanded and while a fraction is returned to the dryer the rest is used to regenerate the other column at the reduced pressure (atmospheric). The need for air heating before desorption is thereby eliminated. Figs. 6.2 and 6.4 are a schematic and p-T representation of the system respectively.
FIG. 6.1: TSC HEAT PUMP DRYER
FIG. 6.2: PSC HEAT PUMP DRYER.
FIG. 6.3: p-T REPRESENTATION OF TSC HEAT PUMP DRYER

1 - 2 - 3 : mixing of fresh air intake and recycle air to dryer.
2 - 4 : heating air mixture in the dryer heater.
4 - 5 : drying
5 - 3 : adsorption
3 - 6 : heating regeneration air
6 - 7 : regeneration
FIG. 6.4: p - T REPRESENTATION OF PSC HEAT PUMP DRYER

1-2-3 : mixing of fresh air intake and recycle air to dryer.
2-4 : heating air mixture in the dryer heater
4-5 : drying
5-6 : compression
6-7 : cooling/condensing
7-8 : adsorption
8-3 : expansion
3-9 : regeneration
It is observed that there is a break between the air intake and air exhaust points 1 and 7 in fig. 6.3 and 1 and 9 in fig. 6.4, which may suggest that the cycle is not closed. This is not so. Basically, since the equivalents of the adsorption section and the generation section of an analogous closed-cycle absorption heat pump are present (adsorption and desorption respectively) the cycle can be considered to be closed. The moisture and carrier air stream is not brought back to the initial conditions at the end of the cycle however, as would be expected in a closed cycle, because there are no condenser and evaporator components. The peculiar nature of this application of the heat pump is such that the moisture from the dryer is already vaporised as it enters the heat pump and that removed in the desorption process is substantially degraded in energy, thereby obviating the need of an evaporator and a condenser.

6.2. Performance Evaluation

The performance of the AHP in terms of its ability to upgrade the low quality energy in the air leaving the dryer is measured by the COP while the overall performance of the AHP dryer is measured by the cycle thermal efficiency.

6.2.1. COP

Similar to an AbHP** operating between temperatures $T_R$, $T_h$ and $T_c$, the ideal COP of an AHP operating between the same temperature limits will be expressed as:

$$COP_{max} = \left( \frac{T_h}{T_h - T_c} \right) \left( \frac{T_R - T_c}{T_R} \right) \quad (6.1)$$

where $T_R$ represents the temperature of regeneration,

$T_h$ represents the temperature of air leaving the adsorption bed, and

$T_c$ is temperature of air leaving the drying chamber.

However, for a real cycle, COP must be expressed as:

**Absorption Heat Pump
\[ \text{COP} = \frac{\text{energy absorbed in the adsorber}}{\text{energy input for regeneration}} \quad (6.2) \]

and hence, thermodynamic efficiency of the system will be:

\[ n_T = \frac{\text{COP}}{\text{COP}_{\text{max}}} \quad (6.3) \]

### 6.2.2. Thermal Efficiency

The overall thermal efficiency of the AHP dryer differs from equation 6.2 in that this refers to the evaporation of moisture from the dryer with respect to the net energy input and hence will be expressed as:

\[ n = \frac{\text{heat used for evaporating moisture}}{\text{net energy input}} \quad (6.4) \]

From equation 3.21 the specific MER will be given by:

\[ C_{E} = \frac{n}{\lambda} \quad (6.5) \]

### 6.3 Equilibrium Model

In the simplest form, the AHP dryer can be considered to be such that:

(i) there are no heat losses from the system and hence the processes are adiabatic;

(ii) heat and mass transfer processes in the system go to completion and hence no consideration is given to equilibrium constraints.

Thus, the system may be described by simple adiabatic heat and mass balances, enabling the maximum energy efficiency of the system to be determined.

Consider the TSC dryer of fig. 6.1. such that the temperature and humidity conditions are as shown in the diagram.
6.3.1. Dryer

Let the amount of air recirculated to the dryer for 1 kg. air through the dryer per unit time be $r_D$. By overall moisture balance on the cycle.

$$ (1 - r_D) (y_{R_e} - y_o) = y_{D_e} - y_{Di} \quad (6.6) $$

$$ r_D = \frac{(y_{R_e} - y_o) - (y_{D_e} - y_{Di})}{(y_{R_e} - y_o)} \quad (6.7) $$

moisture balance on dryer heater,

$$ y_{D_i} = (1 - r_D) y_o + r_D y_{Ae} \quad (6.8) $$

$$ r_D = \frac{y_{D_i} - y_o}{y_{Ae} - y_o} \quad (6.9) $$

from 6.7 and 6.9

$$ \frac{y_{D_i} - y_o}{y_{Ae} - y_o} = \frac{(y_{R_e} - y_o) - (y_{De} - y_{Di})}{(y_{R_e} - y_o)} \quad (6.10) $$

$$ y_{De} = y_{D_i} + \frac{(y_{Ae} - y_{Di}) (y_{R_e} - y_o)}{(y_{Ae} - y_o)} \quad (6.11) $$

The fractional saturation of air in the dryer $f$ is expressed as:

$$ f = \frac{y_{De} - y_{D_i}}{y_{sat} - y_{Ae}} \quad (6.12) $$

where $y_{sat}$ = saturation humidity

$$ y_{De} = (1 - f) y_{D_i} + f \ y_{sat} \quad (6.13) $$

from 6.11 and 6.13,

$$ y_{D_i} - \frac{(y_{R_e} - y_o) (y_{Ae} - y_{Di})}{(y_{Ae} - y_o)} = (1 - f) y_{D_i} + f \ y_{sat} \quad (6.14) $$
hence,

\[ y_{D_i} = \frac{f(y_{s,a} - y_o) - y_{e} (y_{R_e} - y_o)}{f(y_{e} - y_o) - (y_{R_e} - y_o)} \] (6.15)

saturation pressure in the dryer is obtained from:

\[ p_{sat} = \frac{29 \pi y_{sat}}{18 + 29 y_{sat}} \] (6.16)

where \( \pi \) = total pressure.

hence, from Antoine's equation,

\[ T_{sat} = \frac{3883}{18.4 - \ln p_{sat}} - 230 \] (6.17)

Also, by simultaneous heat and mass balance on the dryer, operating under adiabatic saturation conditions:

\[ T_{sat} = T_{D_i} \frac{\lambda_w}{c_{p_g}} (y_{sat} - y_{D_i}) \] (6.18)

and \[ T_{D_e} = T_{D_i} - \frac{\lambda_w}{c_{p_g}} (y_{D_e} - y_{D_i}) \] (6.19)

where \( \lambda_w \) is latent heat of evaporation

and \( c_{p_g} \) is specific heat of air

6.3.2 Adsorption

By heat and mass balance in the adsorbing column,

\[ T_{e} = T_{A_i} + \frac{\lambda_w}{c_{p_g}} (y_{A_i} - y_{e}) \] (6.20)

but \( T_{e} = T_{D_e} \)

and \( y_{A_i} = y_{D_e} \)

hence,

\[ T_{e} = T_{D_e} + \frac{\lambda_w}{c_{p_g}} (y_{D_e} - y_{e}) \] (6.21)
6.3.3. Regeneration

Suppose air is finally discharged 100% saturated at a humidity \( Y_{Re}^* \) from the desorbing column, then the partial pressure of moisture in exhaust air will be given by:

\[
P_{Re}^* = \frac{29 Y_{Re}^*}{18 + 29 Y_{Re}^*}
\]

(6.22)

and the temperature by:

\[
T_{Re}^* = \frac{3883}{18.4 - \ln P_{Re}^*} - 230
\]

(6.23)

If all the heat given up by the hot air is used in desorbing moisture, by heat and mass balance on the regenerating column:

\[
T_{Re}^* = T_{Ri} - \frac{\lambda}{c_p g} (Y_{Re}^* - Y_{Ri})
\]

(6.24)

Now if the air is actually discharged with an approach to saturation of \( S \% \), then actual partial pressure of moisture in exhaust air will be:

\[
P_{Re} = \frac{P_{Re}^* S_{Re}}{100}
\]

(6.25)

and

\[
Y_{Re} = \frac{18}{29} \left( \frac{P_{Re}}{2} - P_{Re}^* \right)
\]

(6.26)

hence,

\[
T_{Re} = T_{Ri} - \frac{\lambda}{c_p g} (Y_{Re} - Y_{Ri})
\]

(6.27)

6.3.4. Cycle Performance

Following the definition of equation 6.4 and noting that total heat input to the system consists of heat supplied to the dryer and regeneration heaters, the thermal efficiency of the cycle can be expressed as:

\[
\eta_{EM} = \frac{c_p g (T_{Di} - T_{De})}{c_p (T_{Di} - T_{De}) + c_p (1 - r_D) (T_{De} - T_0) + c_p g (1 - r_D) (T_{Ri} - T_{Re})}
\]

(6.28)
The ideal COP of the TSC as a heat pump will be:

\[
\text{COP}_{\text{max}} = \frac{\left(\frac{T_{Ae}}{T_{R1}}\right)}{\left(\frac{T_{R1}}{T_{De}}\right)}
\]

and the actual COP will be:

\[
\text{COP} = \frac{T_{Ae} - T_{De}}{(1-r_D)(T_{R1} - T_{Ae})}
\]

### 6.3.5. PSC Heat Pump Dryer

The heat and mass balance equations derived for the TSC heat pump are also essentially applicable to the PSC heat pump. However, the expression for thermal efficiency differs significantly from that of the TSC in view of the inclusion of a compressor and cooler and elimination of the regeneration heater in the PSC heat pump. In this regard, therefore, the thermal efficiency of the PSC heat pump dryer will be defined by:

\[
\text{COP} = \frac{c_p (T_{Ae} - T_{De})}{E_{\text{comp}} - E_{\text{rec}}}
\]

where \( E_{\text{comp}} \) is energy input to compressor and \( E_{\text{rec}} \) is energy recovered in the cooler. Similarly, the COP of the cycle will be expressed as:

\[
\text{COP} = \frac{c_p (T_{Ae} - T_{A1})}{E_{\text{comp}} - E_{\text{rec}}}
\]

The ideal COP using the notation of fig. 6.2 will be:

\[
\text{COP}_{\text{max}} = \frac{\left(\frac{T_{Ae}}{T_{De}}\right)}{\left(\frac{T_{R1}}{T_{De}}\right)}
\]
6.4 Lee-Cummings Procedure

For this model, the equations previously derived in respect of the dryer in section 6.3.1. will also be applicable, noting however that humidification of air may not take place along an adiabatic path. The equations for evaluation of performance, outlined in Section 6.3.4. will also hold.

6.4.1 Adsorption

The processes taking place in the adsorption column may be described by an adaptation of the procedure proposed by Lee and Cummings discussed in Chapter 5. The calculation procedure consists of the following steps:

1) Calculation of bed Reynolds number $N_{Re_b}$ from:

$$N_{Re_b} = \frac{D \mu}{G_v} \left( \frac{1}{6(1-\varepsilon)\psi} \right)$$  \hspace{1cm} (6.35)

where $D$ is average adsorbent particle diameter, $\mu$ is fluid viscosity, $\varepsilon$ is bed void fraction, $\psi$ is particle shape factor, and $G_v$ is air mass flow velocity.

2) Calculation of Colburn factor $J_D$:

$$J_D = \begin{cases} 0.91(N_{Re_b})^{-0.51} & (N_{Re_b} < 50) \\ 0.61(N_{Re_b})^{-0.41} & (N_{Re_b} > 50) \end{cases}$$  \hspace{1cm} (6.36a)

or

$$J_D = \begin{cases} 0.91(N_{Re_b})^{-0.51} & (N_{Re_b} < 50) \\ 0.61(N_{Re_b})^{-0.41} & (N_{Re_b} > 50) \end{cases}$$  \hspace{1cm} (6.36b)

3) Calculation of gas phase mass transfer coefficient $k_{ga}$ from:

$$k_{ga} = J_d \frac{v a}{D_{AB} \kappa_g} \left( \frac{\nu}{D_{AB} \kappa_g} \right)^{2/3}$$  \hspace{1cm} (6.37)

where $v$ is superficial air velocity, $D_{AB}$ is gas phase mass diffusivity, $\kappa_g$ is air density.

4) Calculation of modified distribution ratio $K_\varepsilon$.  


\[ K_e = \frac{M \cdot \ell_s}{y_{sat} \cdot g} \quad (6.36) \]

\( \ell_s \) is packed bed density
\( M \) is slope of linear equilibrium isotherm in a plot of capacity vs percent humidity
and \( y_{sat} \) is saturation humidity evaluated at air inlet temperature to the adsorber \( T_{Ai} \).

5) Calculation of dimensionless time ratio \( \alpha \)
\[
\alpha = 1.0 + 1.66 \left( 100 \Delta y \right)^{0.87} \quad (\forall y < 0.01) \quad (6.39a)
\]
or
\[
\alpha = 3.0 \quad (\forall y > 0.01) \quad (6.39b)
\]

6) Calculation of constants \( C_1 \) and \( C_2 \) given by:
\[
C_1 = 4.66 \alpha - 2.30 \quad (6.40)
\]
and
\[
C_2 = 0.07 \alpha + 1.32 - \frac{K_a}{K_e} \theta_b \quad (6.41)
\]
where \( \theta_b \) is breakpoint time.

7) Calculation of bed height modulus \( M_z \)
\[
M_z = \frac{1}{2} \left[ \left( C_1^2 - 2 C_2 \right) + C_1 \sqrt{C_1^2 - 4 C_2} \right] \quad (6.42)
\]

8) Calculation of bed height \( Z \)
\[
Z = \frac{M_z \cdot v}{K_e a} \quad (6.43)
\]

This procedure enables the size of the bed to be determined, and hence the breakpoint capacity of the bed, but not the temperature condition of the outlet air \( T_{Ai} \). Now, if it is considered that thermal equilibrium is approximately attained between the exit air and the solids in the adsorption bed, then:
\[
T_{Ai} = T_{xb} \quad (6.44)
\]
where \( T_{xb} \) is temperature of solids at breakpoint.

By a lumped heat balance on the column at breakpoint:
Ms \, \frac{c_s}{p_s} \, (T_{A_e} - T_{s_i}) + G \Theta_b \, \frac{c_p}{p_g} \, (T_{A_e} - T_{s_i}) = G \Theta_b \, \Delta y \lambda_A \quad (6.45)

where \Delta y = y_{A_i} - y_{A_e} \quad (6.46)

and \, \frac{c_s}{p_s} \, is \, heat \, capacity \, of \, solids

G \, is \, flowrate \, of \, air

\lambda_A \, is \, heat \, of \, adsorption

\Theta_b \, is \, breakpoint \, time

T_{s_i} \, is \, initial \, bed \, temperature.

By moisture balance

M_s \cdot \Delta W = G \cdot \Theta_b \Delta y \quad (6.47)

where \, \Delta W = W_b - W_i \quad (6.48)

and \, W_i, W_b \, are \, loadings \, of \, moisture \, on \, adsorbent \, initially \, and \, at \, breakpoint \, respectively;

M_s \, is \, mass \, of \, the \, bed.

If the bed is initially fully desorbed, then

M_s = \frac{G \Theta_b \Delta y}{W_b} \quad (6.49)

From 6.45 and 6.49

\begin{align*}
T_{A_e} &= \frac{W_b \left( \Delta y \lambda_A + \frac{c_p}{p_g} \, \frac{T_{s_i}}{T_{A_e}} \right) + \Delta y \, \frac{c_s}{p_g} \, T_{s_i}}{\Delta y \cdot \frac{c_p}{p_g} + W_b \, \frac{c_s}{p_g}} \\
& \quad (6.50)
\end{align*}

This equation, which is similar to equation 5.32 proposed by Miller and Roberts\textsuperscript{102}, can thus be used in conjunction with Lee-Cummings procedure to predict the performance of the adsorption column.
6.4.2 Regeneration

Following the discussion of the nature of the temperature, humidity and loading changes and of equilibrium relations in desorption (chapter 5), the process can be described by a combination of an appropriate equilibrium relationship and a lumped heat balance analysis.

The initial loading of moisture on the desorbing bed will be expected to be the breakpoint capacity obtained in the preceding adsorption and is considered to be in equilibrium with the exhaust air. If the desorption relative humidity is defined as a function of the exhaust air humidity and temperature \( y_{Re} \) and \( T_{Re} \), respectively, then any suitable system equilibrium relationship can be represented by:

\[
W_b = \frac{W_{eD}}{V_{eD}^*} = f (x_D) = f (y_{Re}, T_{Re})
\]  

(6.51)

where \( W_{eD} \) is the initial loading on the desorbing bed. By assuming thermal equilibrium between the desorbing bed and exhaust air,

\[
T_{Re} = T_{SD}
\]

(6.52)

where \( T_{SD} \) is temperature of desorbing bed.

A lumped heat balance on the bed therefore gives,

\[
G_{eb} \cdot C_p \cdot \left( T_{Ri} - T_{Re} \right) = M_s \cdot C_p \cdot \left( T_{Re} - T_{si} \right) + \lambda_A \cdot G_{eb} \cdot \left( y_{Re} - y_{Ri} \right)
\]

(6.53)

hence,

\[
y_{Re} = y_{Ri} + \frac{G_{eb} \cdot C_p \cdot \left( T_{Ri} - T_{Re} \right) - M_s \cdot C_p \cdot \left( T_{Re} - T_{si} \right)}{\lambda_A \cdot G_{eb}}
\]

(6.54)

Combination of equations 6.51 and 6.54 would enable the exhaust air conditions to be determined and hence performance of the system can then be readily evaluated.
6.5. **Dehler Method**

The concept of length of unused bed (LUB) and equilibrium section in isothermal adsorption enunciated by Collins, has been discussed in Chapter 5. With Dehler's empirical deduction, which was also discussed in Chapter 5, this concept can be extended to non-isothermal conditions. He proposed that under non-isothermal conditions half of an adsorption bed may be deemed to be at equilibrium with the inlet air while the other half, comprising the LUB, will be at equilibrium with the outlet air. This can therefore be alternatively used to represent the adsorption section of the AHP under consideration.

The relative humidity of inlet air to the adsorber can be expressed as:

\[
Xe_i = \frac{p_{Ai}}{p_{Ai}^o}
\]

where \(p_{Ai}\) is partial pressure of moisture in inlet air, and \(p_{Ai}^o\) is saturation vapour pressure of moisture at air inlet temperature \(T_{Ai}\). Thus, using an appropriate equilibrium equation, the loading of moisture at the inlet half of the bed may be given by:

\[
W_{e_i} = f (Xe_i) = f (y_{Ai}, T_{Ai})
\]

At the outlet end, relative humidity would be affected by the temperature effects associated with adsorption, which need to be known. The temperature rise may therefore be calculated using a procedure such as the Worthington empirical equation:

\[
\Delta T = 70 h_{1000}
\]

where \(h_{1000}\) is the moisture adsorbed in lbs/1000 ft\(^3\) air and \(\Delta T\) is temperature rise across the bed in °F. Expressing this in terms of actual humidities and in °C, then,

\[
\Delta T = \frac{12.25 \times 10^3 \Delta y}{(T_{Ai} + 273)}
\]
Hence, $T_{a,e} = T_{a,i} + \left(12.25 \times 10^3 \Delta y\right) \frac{T_{a,i}}{T_{a,i} + 273}$ \hspace{1cm} (6.58)

Thus, in a similar manner as the inlet end, relative humidity of exit air will be:

$$x_{e,e} = \frac{p_{a,e}}{p_{a,e}^0}$$ \hspace{1cm} (6.59)

and hence the loading at the outlet half of the bed will be expressed as:

$$w_{e,e} = f(x_{e,e}) = f(y_{a,e}, T_{a,e})$$ \hspace{1cm} (6.60)

Breakpoint capacity of the bed can then be obtained using Dehler's equation thus:

$$W_b = \frac{1}{2} (W_{e,i}^0 + W_{e,e}^0)$$ \hspace{1cm} (6.61)
CHAPTER SEVEN

EXPERIMENTAL
7. EXPERIMENTAL

7.1. Objectives

An experimental programme was undertaken, with the following objectives:

(1) Demonstration of the simultaneous drying of humid air and temperature lifting (heat pumping) in an adsorption column, under conditions that simulate those that may be expected from a solids dryer.

(2) Observation of the temperature and humidity profiles in the fluid and solid phases during the adsorption process and during the subsequent regeneration of the bed.

(3) Investigation of the effects of operating parameters, such as
   - air flowrate
   - air temperature
   - air humidity
   - bed height
   on the adsorption process, especially under conditions of high temperature and humidity.

(4) Correlation of results to enable theoretical predictions from the models outlined in the previous chapter, which were derived from various proposals made by other investigators, to be tested against the experimental data obtained.

7.2. Design of Apparatus

The experimental apparatus, which was purpose designed and erected for the study, consists essentially of three main items of equipment:

- air blower
- air heater
- adsorption column.
The blower is a purchased item while the heater and the column were in-house designed and fabricated. Air is taken in by the blower, passed through the heater and then into the adsorption column. A steam junction is provided before the entrance to the column, to enable steam from the central mains to be mixed with the inlet air, in order to increase the air humidity, if desired. A schematic flow diagram of the apparatus is shown in fig. 7.1.

7.2.1. Air Blower

The blower is an electrically driven, centrifugal device, with the following speed and power specifications:

- speed : 9500 rpm
- power : 0.75 hp (0.56 kw)

The operating characteristics, flowrate and pressure head, are shown in the Appendix (C1).

7.2.2. Air Heater

This consists of a 40 x 40 x 48 cm. rectangular steel box in which 3 electrical heating elements, each of a power rating of 1 kw., are mounted. By connecting the elements in series, the temperature to which the air is heated can be readily controlled. The walls of the box are lined with 8 mm. thick asbestos sheets which are intended to serve as insulators, in order to curtail heat loss from the heater to the environment.

7.2.3. Adsorption Column

The adsorption column consists basically of a 1.5 m. high, 23.5 cm. internal diameter and 10 mm. thick mild steel cylindrical shell made from a standard pipe section in which the solid adsorbent is packed. The solid is held on a perforated support grid which is mounted 25 cm. from the bottom of the column. The grid is supported on three cylindrical legs which rest on the bottom flange, thereby providing the structural strength that enables it hold the weight of the bed. A 7.5 cm. charge hole is provided on the top flange for feeding in the adsorbent and two 4 cm. bore holes are fitted on the shell, 15 cm. from the top and bottom flanges respectively, to serve as moist air inlet and dehumidified air outlet. Five 6 mm. bores are also fitted to the shell, at 25 cm. intervals, for temperature measurements along the bed. A 10 mm. diameter vent at the top flange
Plate A: EXPERIMENTAL APPARATUS
Fig. 7.2: Adsorption Column
is designed for air venting in case of accidental pressurization of the column while a 12 mm. diameter drain line is located at the bottom flange to enable any accumulation of water to be drained away. A thin gauze sheet is also placed on top of the bed to prevent entrainment of the solid particles in the air stream.

7.3. Instrumentation and Procedure.

7.3.1. Flow Measurement

The flowrate of air is measured with a Metrix X series, 65X tube size rotameter, with a type A duralumin float. A calibration chart provided by the manufacturer, which is shown in the Appendix (C2), is used to calibrate the rotameter.

The flow of air is controlled by a damper which is situated at the suction side of the blower and by a globe valve fitted between the blower and the heater. A valve is also installed between the heater and the steam inlet, which stops moisture from drifting back into the heater, thereby preventing possible electrical hazards. The admission of steam is detected by means of a pressure gauge located near the inlet point. A main steam inlet valve and a bleed valve enable the flow to be controlled.

7.3.2. Temperature Measurement

Chrome-alumel thermocouple wires are fitted to the adsorption column by passing them through rubber bungs which are used to plug the temperature measurement tap points provided on the shell. Care is taken to place the ends of the thermocouples at the centre of the column axis and adsorbent is carefully packed around them. The other ends of the thermocouple wires are connected to a multipoint digital thermometer which in turn is connected to a switch selector unit. The thermocouples are thus used to measure the temperature of the bed at different positions along the length of the column during operation. The air inlet and outlet temperatures are also measured using thermocouples fitted at the air inlet and outlet lines.
7.3.3. Humidity Measurement

Many types of devices are available commercially for measuring the concentration of water vapour in gases. These include the dew point cup, resistance, capacitance, and electrolytic methods. A study of these instruments undertaken by Smith and Free showed that the use of most of the instruments is constrained by the difficulty of calibrating them, in addition to other problems, such as measurement range and time of response. They require gaseous phase standards for calibration, and these in themselves are difficult to produce and standardize.

Free proposed an alternative gas-liquid chromatography method, using a carbowax 1500 (polyethylene glycol) column on a teflon support, a katharometer detector and hydrogen as the carrier gas. The instrument is readily calibrated by the injection of aqueous solutions of organic compounds, such as methanol and ethanol, concentrations of which can be readily determined. The instrument was however found suitable for measurement of very low water vapour concentrations and unsuitable for high humidity conditions, such as envisaged in the present study.

Several modern electronic digital instruments which can be appropriately used for this purpose are now commercially available, but owing to cost constraints they could not be acquired for use in the study. It was therefore decided that most appropriately a wet bulb thermometer be used since it is cheap, does not require to be calibrated and if carefully used can give fairly accurate readings.

The use of a wet-bulb thermometer for measuring humidities also requires a measurement of dry bulb temperature and combining these with an appropriate psychometric chart. To ensure that the wet and dry bulb temperature readings are directly comparable the two thermometers must be tested by putting them in a beaker of cold water and taking their readings simultaneously at regular time intervals, while the water is heated. If the readings are plotted, one on the abscissa and the other on the ordinate, the pair that gives readings that lie on a straight line as close to the 45° line as possible is deemed to be most suitable.
Fig. 7.3: Wet/Dry Bulb Thermometer for Humidity Measurement.
The instrument was made up by drilling two holes in a rubber bung through which the two thermometers were passed. A short plastic tube was put around the stem of one of the thermometers, directly above the bung, to create a cavity in which a water column can be held. The cavity was sealed at the base of the tube using a rubber ring. A thick, porous wick sheath was then slipped around the bulb of the thermometer. Two such instruments were made up and used to measure the humidity of the air at the inlet and the outlet, by inserting the bung holding the instruments in holes drilled along the appropriate conduit lines.

7.3.4. Procedure

For each experimental run, the heater is switched on to give the appropriate level of heating desired. The blower is turned on and the flow of air is adjusted to the required level. Steam is then immediately let in by turning and adjusting the steam valves. It is always ensured that the flow of air precedes the injection of steam in order to avoid the electrical hazards that may be associated with a backward drift of steam into the heater when there is no air flow. The following readings are taken at time intervals as the run proceeds:

- air inlet and outlet temperatures,
- air inlet and outlet wet-bulb temperatures,
- bed temperatures at different heights.

From time to time the air flowrate is read to ensure that a steady flow is maintained. Given the general practical difficulty of direct measurement of steam flow, the humidity of the inlet air is set by a careful and intuitive manipulation of the steam valves in co-ordination with the readings of the wet-bulb thermometer as well as the indication of the pressure gauge. Under conditions of high air flow rate, low humidity and high temperature, the removal of water from the wick proceeds very rapidly, exceeding the rate at which it is replaced by capillary flow from the water cavity above. It becomes necessary therefore to remove the instrument from position, wetting the wick and reinserting the instrument before a reading is taken.
At the end of each adsorptive run the adsorbent is regenerated by a flow of dry hot air. The procedure for the regenerative runs is similar to that of the adsorptive runs except that there is no steam injection to increase air humidity in this case. Air at ambient humidity is used.

7.3.5. Materials

Granular type silica-gel adsorbent was used for the study. The properties of the adsorbent, as furnished by the suppliers are:

- Average Particle Diameter: 3.5 mm.
- Surface Area: 0.85 m$^2$/kg.
- Pore Volume: $0.35 \times 10^{-3}$ m$^3$/kg.
- Bulk Density: 1200 kg/m$^3$.
- Specific Heat: 1.05 KJ/kg$^{-0}{\text{C}}$.

7.4. Results

The series of experimental runs was carried out over the following range of operating conditions:

- Air flowrate: 0.5 - 2.0 kgs/min.
- Air inlet temperature: 30 - 105°C.
- Air inlet humidity: 0.005 - 0.05 kg/kg.
- Bed height: 0.6 - 1.0 m.

Typical temperature and humidity profiles for a run are presented in fig. 7.4. Fig. 7.4a. shows bed temperature at different times; Fig. 7.4b is a plot of air inlet and outlet temperatures against time, as well as a plot of inlet and outlet humidities versus time. These observations confirm that the adsorption front controls the breakpoint and is evidently preceded by the heat transfer front. These are also fairly typical of most of the runs. The double breakthrough suggested by Leavitt$^{96}$ was not observed. This indicates therefore that the double peak phenomenon does not occur in the adsorption of water vapour on silica-gel, at least in this apparatus.
Fig. 7.4a: Adsorption Bed Temperature Profiles (Run 37)

- $T_3$: 62.5 cm. from air inlet
- $T_4$: 37.5 cm. from air inlet
- $T_5$: 12.5 cm. from air inlet
Fig. 7.4b: Adsorption Air Temperature and Humidity Profiles (Run 37)

Temperature (°C)

Humidity (kg/kg)

Time θ (mins)
Fig. 7.4b: Adsorption Air Temperature and Humidity Profiles (Run 37)

Temperature ($^\circ$C) vs. Time (mins)

- $T_0$ (upper line)
- $T_1$ (middle line)
- $y_1$ (lower line)

Humidity ($k_g/k_c$) vs. Time (mins)

- $y_e^*$ (lower line)

Key:
- $\Delta T$
- $\Delta y$
The dehumidifying effect is illustrated in fig. 7.4b. by $\Delta y$ which is the difference between the humidities of the inlet and outlet air streams. The heat pumping effect is also demonstrated by the temperature increase of the air $\Delta T$, as shown in fig. 7.4b. Fig 7.4a. shows clearly that the temperature in the bed varies with distance from the air inlet at any time. Evidently the layers close to the inlet attain a maximum temperature faster and fall off earlier. The temperature of the exit air at any time lies between the maximum and minimum temperatures in the bed at that time.

The obtained results are tabulated in Table 7.1, where

- $\theta_b$ is breakthrough time
- $G$ is air flowrate
- $T_i$ is air inlet temperature
- $T_e$ is air outlet temperature
- $y_i$ is air inlet humidity
- $y_e$ is air outlet humidity.
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7.5. Calculation Procedure

If the average temperature in the bed and of the exit air is $T_e$, then the temperature rise across the column up to breakpoint will be given by:

$$\Delta T = T_e - T_i$$

(7.1)

Humidity drop across the column will be:

$$\Delta y = y_i - y_e$$

(7.2)

The amount of moisture extracted, up to breakpoint will then be:

$$M_b = G \theta_b \Delta y$$

(7.3)

and hence the adsorptive capacity of the bed at breakpoint will be

$$\omega_b = \frac{M_b}{M_s}$$

(7.4)

where $M_s$ is mass of solid adsorbent held in the bed.

The partial pressure of moisture in inlet air can be obtained from the humidity, thus:

$$P_i = \frac{29 \Pi y_i}{18 + 29 y_i}$$

(7.5)

where $\Pi$ is the total pressure.

Saturation vapour pressure of moisture evaluated at average bed temperature can be expressed by the Antoine equation:

$$P_{ob} = \exp \left( 18.4 \frac{-3883}{T_e + 230} \right)$$

(7.6)

Hence, from its definition bed relative humidity at breakpoint $X_b$ will be:

$$X_b = \frac{P_i}{P_{ob}}$$

(7.7)

A sample calculation, starting from raw experimental data, is presented in the Appendix and the calculated results are shown in Table 7.2.
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<th>$\Delta y$ (kg/kg)</th>
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<th>$X_b$ (%)</th>
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7.6. **Effects of Variables**

7.6.1. **Flowrate**

Several runs were carried out at an average air inlet humidity of 0.0060 kg/kg and at inlet temperature levels of about 35°C and 75°C, while the flowrate of air was varied between 0.65 and 2.0 kgs/min. The experimental observations showed that the breakpoint time and to a lesser extent the bed capacity were influenced by the air flowrate at both temperature levels. The slight fall in bed capacity as flow-rate is increased as illustrated in fig. 7.5. These data are fairly consistent with those of several earlier investigators [64, 67, 68, 98] which indicated a measure of dependence of capacity on flowrate. They do not support the suggestions of Miller and Roberts [102], and of Moss and McLaughlin [101], that capacity is unaffected by flowrate. Since most earlier observations were made under low temperature conditions, it is of particular interest to note that the effect of flowrate on capacity at high temperature is similar to that for the lower temperature situation.

The slight drop in bed capacity as flowrate is increased can be explained in terms of the reduced contact time between the adsorbent granules and the air, associated with a high flow. With reduced contact time, moisture is less able to permeate the pores of the adsorbent particles.

The effect of flowrate on bed relative humidity is similar to that on bed capacity and is also shown in fig. 7.5.
Fig. 7.5: Effect of flowrate on Skpt. Capacity and Bed Rel. Hum.

- $35^\circ\text{C}$
- $75^\circ\text{C}$

Bed rel. humidity $X_b$(%)  

Bed pt. cap. $W_b$(%)  

Flowrate $G$ (kgs/min)
7.6.2. Temperature

The adverse effect of temperature on the process of adsorption is well recognised. However, little previous work has been done on the nature of this effect, especially with regard to inlet temperature of moist air. Experiments were conducted at atmospheric humidity and at two air flowrates, with inlet temperatures varying between 30 and 95°C. The predictable fall of adsorptive capacity and bed relative humidity as inlet temperature is increased is illustrated in fig. 7.6. A more interesting observation is the effective drop in temperature of the air as it passes through the adsorption column (i.e. $\Delta T < 0$) under high temperature conditions. This phenomenon can be explained by considering the heat transfer processes that take place. If the released heat of adsorption is insufficient to raise the temperature of the solids to the inlet temperature of the air, then there could be a net transfer of heat from the gas phase to the solid phase, resulting in a decrease in temperature of the air.

Obviously, high temperature conditions would be expected to be accompanied by shorter breakthrough times. This is even more significant if moisture is adsorbed on a hot bed from a hot air stream. Consequently, under such conditions, very low breakpoint capacities are obtained. However, following the discussion of the nature of heat transfer taking place, high temperature increases in the air would be expected. This is best illustrated by considering Run 39. Though most of the runs were carried out with initial bed temperatures of 20-30°C, this was performed with an initial bed temperature of 105°C. A bed capacity of 1.23% was obtained, with a high temperature rise $\Delta T$ of 57°C, as compared with typical values of bed capacity of 3-8% and temperature rise less than 30°C, for lower bed temperatures.
FIG. 7.7: Effect of inlet temperature on temperature change

- $1.25 \text{ kgs/min}$
- $2.00 \text{ kgs/min}$

Temperature change $\Delta T$ vs. Inlet temperature $T_i$
7.6.3. **Humidity**

The manner in which the humidity of inlet air affects the breakpoint capacity is twofold. First, an increase in air humidity results in increased vapour pressure and hence higher adsorption capacity. However, since adsorption is intrinsically exothermic, the associated temperature increase tends to have an opposite effect.

The dependence of capacity and bed relative humidity on air humidity is presented in fig. 7.8. At a low air inlet temperature \( (35^\circ C) \), the adverse effect of temperature seems to be dominant and hence capacity decreases slightly as humidity is increased. This appears to be consistent with the observations of Grayson \(^{68}\) and Worthington \(^{67}\). On the other hand, at higher inlet temperatures, it was observed that the breakpoint capacity increased with humidity up to about 0.02 kg/kg and subsequently levelled off. This also agrees with the earlier suggestions of Grayson \(^{68}\), that capacity may also increase with humidity under certain conditions, and of Miller and Roberts \(^{102}\), that capacity may be independent of absolute humidity, for adiabatic adsorption.

The relationship between the temperature and humidity changes of the air across the column is plotted in fig. 7.10. The plots lend support to the general concept of a linear relationship between \( \Delta T \) and \( \Delta y \), up to a \( \Delta y \) value of about 0.01 kg/kg. Such linear relationships have been proposed by Rathmell and Bateman \(^{106}\) and by Worthington \(^{67}\), as noted in chapter 5. However, their linear equations do not take into account the possible effects of inlet temperature and flow rate on this relationship, which are clearly obvious from the plots of fig. 7.10.

It is noted also that these plots reinforce the previous observation that negative values of \( \Delta T \) may occur under a combination of high temperature and low humidity conditions.
Fig. 7.8: Effect of inlet humidity on breakpoint capacity.

- $\times 33^\circ C$, 2.0 kgs/min.
- $\circ 62^\circ C$, 2.0 kgs/min.
- $\triangle 72^\circ C$, 2.0 kgs/min.
- $\triangledown 88^\circ C$, 1.25 kgs/min.

Inlet humidity $y_i$ (kg moisture/kg air).
Fig. 7.9: Effect of inlet humidity on bed rel. hum.

- $33^\circ C$, 2.0 kgs/min.
- $62^\circ C$, 2.0 kgs/min
- $72^\circ C$, 2.0 kgs/min
- $88^\circ C$, 1.25 kgs/min.

Bed rel. hum. $X_b$ (%)

Inlet humidity $Y_i$ (kg moisture/kg air)
Fig. 7.10: Temperature change $\Delta T$ vs humidity change $\Delta y$

- $33^\circ C$, 2.0 kgs/min
- $62^\circ C$, 2.0 kgs/min
- $72^\circ C$, 2.0 kgs/min
- $88^\circ C$, 1.25 kgs/min
7.6.4. Bed Height

Experimental runs were carried out with bed heights of 0.6, 0.75 and 1.0 m. under fairly similar conditions. Predictably, it was observed that higher breakpoint capacities and bed relative humidities resulted from increased bed heights (fig. 7.1). The influence of bed height can be attributed also to contact time. A larger contact time would be associated with increased bed height, which thus allows the adsorbed moisture to penetrate to the inner pores of the adsorbent particles, enabling adsorption of more moisture.
Fig. 7.11: Effect of bed height on bkpt. capacity and bed relative humidity

- $x$ 2.0 kgs/min.
- $\circ$ 1.25 kgs/min.

$W_b$ ---

$x_b$ ---

Bkpt. cap. $x_b$ (%)

Bed rel. hum. $x_b$ (%)

Bed height $Z$ (m)
7.7. Correlation of Data

The calculated breakpoint capacity is plotted against bed relative humidity for each run in fig. 7.12. These data show some similarity with the data of Miller and Roberts for semi-isothermal adsorption of moisture on activated alumina (fig. 5.14) in that there is an indication of a relationship between breakpoint capacity and bed relative humidity. However, while their data depict a straight line relationship, these data do not. The data of fig. 7.12 are also comparable with those of Worthington, for non-isothermal adsorption of moisture on silica-gel (fig. 5.15). It is noted however that a considerable difference exists between the two sets of data. While Worthington's plot appears to be convex to the \( X_b \) axis, the present data appear to be concave to the same axis.

The shape of the plot suggests that the data can be correlated with a polynomial function of the form:

\[ W_b = K X_b^n \]  

(7.8)

A log-log plot of \( W_b \) against \( X_b \) is presented in fig. 7.13. Using the method of least squares, the data was correlated by the straight line equation:

\[ \ln W_b = 0.07 + 0.81 \ln X_b \]  

(7.9)

hence,

\[ W_b = 1.07 X_b^{0.81} \]  

(7.10)

Alternatively, the data was correlated by a pair of straight line equations, one suitable in the range 0 to 6\% relative humidity and the other for relative humidities above 6\%. Thus:

\[ W_b = 0.8 X_b \quad 0 < X_b < 6.0 \]  

(7.11a).

\[ W_b = 0.375 X_b + 2.75 \quad X_b > 6.0 \]  

(7.11b)

It is noted however that the investigation was carried out under adiabatic and atmospheric conditions for which bed relative humidity values are low. For higher relative humidities, which may be attained under isothermal or high pressure conditions, equations 7.10 and 7.11 may therefore not be applicable. Worthington suggested that a maximum bed capacity of about 20\% is attained at high bed relative humidities for high pressure adsorption of water vapour on silica-gel.
Fig. 7.12: Correlation of bkpt. capacity $W_b$ and bed relative humidity $X_b$. 
Fig. 7.13: Log-log correlation of bkpt capacity $W_b$ and bed relative humidity $X_b$. 

![Graph showing log-log correlation of bkpt capacity $W_b$ and bed relative humidity $X_b$.](image)
7.8. Comparison of Breakpoint Capacity and Equilibrium Capacity

The silica-gel / water vapour adsorption equilibrium has been discussed in Chapter 5. The concave shape of the plot (fig. 7.12), and the nature of the corresponding equations (7.10 and 7.11) indicate that there is a resemblance between the breakpoint data obtained and the silica-gel / water vapour equilibrium relationship. In this regard therefore, it may be useful to compare the values of breakpoint capacity predicted from equations 7.10 and 7.11, for both isothermal and adiabatic operations, with equilibrium capacities obtained from some of the proposed equilibrium equations.

A silica-gel adsorption bed is shown in fig. 7.14. Humid air is deemed to be passed into the bed from a convective dryer as shown and conditions are as indicated in the diagram. The calculation procedure consists of:

(1) Dryer design calculations to obtain the outlet temperature and humidity conditions $T_2$ and $y_2$ and hence the inlet conditions of air to the adsorber.

(2) Calculation of equilibrium relative humidity $X_e$, evaluated at inlet temperature and humidity, $T_3$ and $y_3$, to the adsorption bed.

(3) Calculation of equilibrium bed capacity $W_e$ using the following equations:

(a) Simpson-Cummings\textsuperscript{64} Equations -

\[ W_e = 1.57 + 0.58X_e \quad (0 < X_e < 50) \quad (7.12a) \]

\[ W_e = 24.63 + 0.14X_e \quad (50 < X_e < 100) \quad (7.12b) \]

(b) Jury - Edwards\textsuperscript{60} Correlation -

If $X_f$ is defined by

\[ X_f = X_e \exp \left( 1.8 \frac{T_3}{500} \right) \quad (7.13) \]

where $T$ is in °K.
Fig. 7.14: Scheme for Evaluating Silica-Gel Equilibrium and Breakpoint Capacities.
then,  
\[ \frac{2X}{1 + 13.33X} + \frac{10.15X}{1 + 46.7X} = \frac{3.3}{3.3} \]  
(7.14)

(c) Mukesh-Rae equation -
\[ \frac{0.92X}{1 + 1.603X} + \frac{0.784}{2.603} \]  
(7.15)

(4) Calculation of breakpoint bed capacity \( W_{b1} \), for a bed operated isothermally, say by insertion of cooling coils in the bed, from equation 7.10 and 7.11, but using the equilibrium bed relative humidity \( X_e \), rather than the breakpoint humidity \( X_b \). This is necessitated by the fact that the bed temperature and the air outlet temperature would remain at about air inlet temperature for isothermal operation.

Thus:
\[ W_{b1} = 1.07X_e \]  
\[ (7.16) \]

or
\[ W_{b1} = 0.83X_e \quad (0 < X_e < 6) \]  
\[ (7.17a) \]
\[ W_{b1} = 0.375X_e + 2.75 \quad (X_e > 6) \]  
\[ (7.17b) \]

(5) Calculation of air outlet temperature from the adsorption bed \( T_4 \) and hence the breakpoint bed relative humidity \( X_b \) and breakpoint capacity \( W_{b1} \), for non-isothermal operation, using the bed heat balance equation 6.50 and either equation 7.10 or 7.11 in combination.

Using the procedure outlined above, a computer programme was set up and the results of the computation of the various capacities and relative humidities over a range of dryer operating conditions, and hence inlet conditions to adsorber, are presented in Table 7.3.
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</table>

<table>
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<th>(%)</th>
<th>(%)</th>
<th>(%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>( q_{\text{M}} )</td>
<td>( q_{\text{M}} )</td>
<td>( q_{\text{M}} )</td>
<td>( q_{\text{M}} )</td>
<td>( q_{\text{M}} )</td>
<td>( q_{\text{M}} )</td>
<td>( q_{\text{M}} )</td>
<td>( q_{\text{M}} )</td>
<td>( q_{\text{M}} )</td>
</tr>
</tbody>
</table>

Table 7.2: Comparison of Equilibrium and Breakpoint Capacities
For each set of entries in Table 7.3, $X_e$ is seen to be larger than $X_b$. This arises because of the increase in air temperature as it leaves the adsorber. However, smaller $X_e$ than $X_b$ could be obtained, if conditions are such that there is a fall in air temperature across the bed.

Among the three equilibrium equations considered, the Mukesh-Rao equation evidently predicts a much higher equilibrium capacity than the other two, which agree to a reasonable extent. A closer examination however indicates that Jury-Edwards correlation would generally predict slightly lower values than Simpson-Cummings equation.

For both isothermal and adiabatic adsorption, the results show that the two empirical equations obtained for breakpoint capacity, equations 7.10 and 7.11, agree fairly reasonably, especially for low and intermediate capacities. Overall, equation 7.11 tends to give slightly lower predictions up to bed capacities of about $6\%$ and slightly higher predictions for bed capacities greater than $6\%$.

Overall, predicted equilibrium capacity values are larger than isothermal breakpoint capacities, especially in relation to the use of Mukesh-Rao equilibrium equation, and to a less extent the Simpson-Cummings equation. Theoretically, since a fraction of the adsorbent in the MTZ is unused at breakpoint, it would be expected that breakpoint capacity would be less than the equilibrium capacity. The difference between $W_e$ and $W_b$ would however be expected to diminish under high temperature, and hence low bed relative humidity conditions, since the heat and adsorption fronts would tend to travel faster and the breakthrough would be sharper. Evidently therefore, equations 7.10 and 7.11 will predict breakthrough capacities fairly satisfactorily.

Following the relationship between $X_e$ and $X_b$, it is obvious that adiabatic breakpoint capacity will be smaller than both isothermal breakpoint capacity and equilibrium capacity. However, if conditions are such that $X_e < X_b$, then the reverse situation will apparently hold.
7.9. **Regeneration**

Various flows in the region 1.0 - 1.5 kgs/min. and temperatures of 110-140°C were used for the regeneration of the used bed. Typical gas phase and solid phase temperature and humidity profiles for a regeneration run are presented in figs. 7.15 and 7.16. These evidently support the wave theory of sorption. Clearly the profiles show that desorption at any level in the bed takes place at a fairly steady temperature which subsequently rises when desorption is complete at that level. It is also noted that the desorption temperature at the air inlet end of the bed is nearest to the air inlet wet-bulb temperature and increases as the wave moves further along the bed.

The initial temperature of the used bed was observed to exert a considerable influence on the mode of desorption. The adsorption run is usually stopped when the bed is saturated at a high temperature. If the bed is allowed to cool, the equilibrium is shifted and the bed loading is less than saturation. Thus, initially the flow of hot, relatively dry air may result in adsorption. Subsequently, the temperature of the bed rises rapidly. The equilibrium is then re-established and desorption then commences. If regeneration is commenced while the bed is hot, there is no initial period of adsorption since the equilibrium favours desorption from the start. Fig 7.15 represents regeneration of a hot bed while fig. 7.16 represents the regeneration of a cold bed.
\[ \text{Temperature (°C)} \]

\[ \text{Time (min)} \]

- \( T_1 \) : 62.5 cm from inlet
- \( T_2 \) : 37.5 cm from inlet
- \( T_3 \) : 12.5 cm from inlet

Desorption bed temperature profiles (run 26)
$T_5: 12.5 \text{ cm. from air inlet}$

$T_4: 37.5 \text{ cm. from air inlet}$

$T_3: 62.5 \text{ cm. from air inlet}$
CHAPTER EIGHT

PREDICTION OF PERFORMANCE
8. PREDICTION OF PERFORMANCE

8.1. Conditions of Air from Conventional Dryer

Before proceeding to evaluate the performance of the AHP dryer using the various models outlined, in conjunction with the results of the experimental investigation, it is important that the nature and state of moist air delivered from a dryer, under various operating conditions, are clearly stated. By considering a convective dryer operated with air inlet temperature $T_{Di}$ in the range 50-200°C and fractional saturation $f$ of 0.2 - 0.8, the outlet temperature and humidity from the dryer ($T_{De}$ and $Y_{De}$ respectively) and hence delivery conditions to an AHP, $T_{Ai}$ and $Y_{Ai}$, have been calculated. Ambient conditions are assumed to be $10^\circ$C and 0.005 kg/kg humidity.

The data are presented in Table 8.1 and in fig. 8.1. Clearly at constant dryer air inlet temperature as fractional saturation is increased, outlet temperature $T_{De}$ falls while outlet humidity $Y_{De}$ increases. Hence, the more moisture laden the air, the lower its temperature. On the other hand, operating at constant fractional saturation, $T_{De}$ and $Y_{De}$ increase simultaneously with the inlet temperature since a higher inlet temperature would naturally imply a higher outlet temperature and the moisture holding capacity of air increases with temperature. These relationships are clearly illustrated in fig. 8.1.
<table>
<thead>
<tr>
<th>$T_{D_1}$ (°C)</th>
<th>$f$ (fractional saturation)</th>
<th>$T_{De}$ (°C)</th>
<th>$Y_{De}$ (kg/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>.2</td>
<td>44.39</td>
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<td>.4</td>
<td>38.83</td>
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</tr>
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</tr>
<tr>
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<td>59.57</td>
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<td>.0424</td>
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<td>134.93</td>
<td>.0314</td>
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<td>104.43</td>
<td>.0446</td>
</tr>
<tr>
<td></td>
<td>.8</td>
<td>75.18</td>
<td>.0578</td>
</tr>
</tbody>
</table>
Fig. 8.1: Air Conditions from Convective Dryer.

$T_{Di}$ is dryer air inlet temperature.

$f$ is air fractional saturation.

Exit air temp. $T_{De}$ ($^\circ$C)

Exit air humidity $Y_{De}$ (kg H$_2$O/kg air)
8.2. Equilibrium Model.

The Equilibrium Model, which has been previously outlined (Section 6.3), was used to predict the performance of the ADP dryer, operating in the TSC mode (Fig. 6.1). The results are plotted in Figs. 8.2 and 8.3.

8.2.1. Drying conditions

It is observed that the drying conditions, air inlet temperature $T_{Di}$ and fractional saturation $f$, have virtually no effect on the thermal efficiency of the TSC dryer. Operating at a higher drying temperature leads to a high outlet temperature $T_{De}$ and consequently high influent temperature to the adsorber $T_{Ae}$ at any fractional saturation. Also, this gives a higher dryer outlet humidity $Y_{De}$ and hence a lower recycle $r_D$. From equation 6.29,

$$\eta_{EM} = \frac{T_{Di} - T_{De}}{(T_{Di} - T_{Ae}) + (1-r_D)(T_{Ri} - T_o)}$$

(6.29)

It is evident that increased $T_{Di}$ and $T_{Ae}$ would improve the cycle thermal efficiency $\eta_{EM}$ while increased $T_{De}$ and reduced $r_D$ would tend to lower it. Thus, the net effect is that $\eta_{EM}$ is almost independent of $T_{Di}$. The reduction in $T_{De}$ and increase in $Y_{De}$ associated with an increase in $f$, for a given temperature, have opposing and counterbalancing effects on $T_{Ae}$. The lower $T_{De}$ and the attendant reduction in $r_D$ consequently keep $\eta_{EM}$ constant.

8.2.2. Regeneration Temperature

Examination of equation 6.29 suggests that lower regeneration temperatures $T_{Ri}$ will give higher efficiencies. However, it is noted that a low $T_{Ri}$ implies also a low exhaust humidity $Y_{Re}$ and hence increased demand for regeneration air. Consequently, hot air recirculation to the dryer $r_D$ falls and $\eta_{EM}$ falls too. This effect is illustrated by the plots for regeneration at 150°C and 200°C in fig. 8.3.
Fig. 8.2: Thermal Efficiency of TSC Dryer
(Equilibrium Model)

$T_{Di}$ is dryer air inlet temperature

$\times$ efficiency of conventional dryer, $\eta_{\text{conv}}$

$\circ$ efficiency of TSC dryer, $\eta_{\text{EM}}$
Fig. 8.3: Effects of Regeneration Temperature and exhaust condition on Thermal Efficiency of TSC Dryer (Equilibrium Model)

- $\eta_{EM}$ is efficiency of TSC dryer
- $\eta_{conv}$ is efficiency of conventional dryer
- $T_{RI}$ is regeneration temperature
- $S_{Re}$ is exhaust saturation

![Graph](image-url)
8.2.3. Exhaust Conditions

The closer to saturation the air is when finally exhausted, the higher the cycle efficiency, as shown in the plots for 60%, 80% and 100% approaches to saturation in fig. 8.3. Though this is not directly reflected in equation 6.29, it arises indirectly through the increased recirculation of hot air which accompanies exhaustion at higher saturation levels.

8.2.4. Comparison with Conventional Dryer.

It may be instructive to relate the performance of the TSC dryer to that of a comparable non-recycle conventional dryer which is considered to operate with the same temperature and fractional saturation. The results indicate that considerable advantage may be derived by TSC drying, especially if the air is exhausted at a sufficiently high level of saturation. This advantage is even more distinct when fractional saturation is low since the thermal efficiency of the comparable conventional dryer $\eta_{\text{conv}}$ is strongly dependent on $f$ (see fig. 8.2). Hence, it may be inferred that ordinarily, operating a dryer with as high a fractional saturation as possible provides the simplest way to improve the thermal efficiency. However, if for other constraints, such as effect on drying rates, it is desirable to operate with low fractional saturation, considerable scope may exist for improving the thermal performance of the dryer by combining it with an AHP operating on the TSC mode.
8.2.5. **Pressure Swing Cycle (PSC)**

Similar computations have been done for a PSC dryer (fig. 6.2) and the results are presented in figs. 8.4 and 8.5. The energy consumption in the compressor $E_{\text{comp}}$ was obtained in the way previously outlined in section 4.3, while the energy recovery in the cooler $E_{\text{rec}}$ was calculated by heat balance on the cooler. Whereas the efficiency of the TSC dryer is virtually independent of the $T_{Di}$ and $T_{Ai}$, the efficiency of the PSC dryer is found to increase with both parameters as shown in fig. 8.4.

As the compression ratio is increased, the energy requirement for compression increases. Though the heat available for recovery at the condenser increases simultaneously, this is insufficient to offset the increased energy requirement. This is because the efficiency of converting primary fuels to electricity is low, which makes $E_{\text{comp}}$ very large. The net effect is a decrease in the efficiency of the PSC dryer at high compression ratios. With the predominance of compression energy in the total energy demand, it is also observed that lowering the cooler temperature $T_{Ai}$, thereby recovering more heat at the cooler, has only a limited effect on cycle efficiency.

8.2.6. **Comparison of TSC and PSC**

Like the TSC dryer, the highest cycle efficiency of the PSC dryer would be expected when air is exhausted at the highest possible saturation level. However, in relation to TSC drying under identical conditions, PSC drying would clearly provide a less favourable thermal efficiency. While efficiencies of up to 60% may be expected using a TSC, the PSC dryer may not attain efficiencies greater than 25%.

Similarly, relative to comparable conventional drying, PSC drying offers a poor thermal performance, as is shown in figs. 8.4 and 8.5, over the range of conditions considered. The indication therefore is that the PSC will not be suitable for enhancing the thermal performance of dryers.
Fig. 8.4: Thermal Efficiency of PSC Dryer (Equilibrium Model)

Pressure $P = 6$ atm
$T_{D1}$ is dryer air inlet temperature.

- Efficiency of PSC dryer, $\eta_{EM}$
- Efficiency of conventional dryer, $\eta_{conv}$

Temperature:
- $T_{D1} = 50^\circ C$
- $T_{D1} = 100^\circ C$
- $T_{D1} = 150^\circ C$
- $T_{D1} = 200^\circ C$

Efficiency $\eta$ vs. fractional saturation $f$. 

$\eta_{EM}$ and $\eta_{conv}$ are plotted against $f$.
Fig. 8.5: Effect of Pressure on Efficiency of PSC Dryer (Equilibrium Model)

Dryer air inlet temperature $T_{Di} = 150^\circ C$.

- $P$ is pressure
- $\eta_{EM}$ efficiency of PSC dryer
- $\eta_{conv}$ efficiency of convective dryer

Efficiency $\eta$ (%)

Fractional saturation $f$
8.3. Evaluation of Adsorption Models

Consideration of the Equilibrium Model for TSC and PSC dryers is only useful as an indication of the kind of thermal efficiency that could be expected and for providing a simple basis for comparing the performances of the two cycles. To make a more realistic prediction it is necessary to perform calculations based on the other models outlined in Chapter 6 (Lee-Cummings Procedure and Dehler Method) which take into account the dynamic effects of the solid adsorbent ignored in the Equilibrium Model.

Before proceeding to consider the overall system performance it is considered necessary that the Lee-Cummings and Dehler Models be applied to the adsorption bed, under identical conditions. This would enable the models to be compared on a common basis and to examine how they are affected by changes in operating conditions.

The computation is based on the scheme illustrated in fig. 8.6. Moist air is passed into the adsorption bed from the dryer. The calculation steps are outlined below:

1. Dryer design calculation, to obtain the influent temperature and humidity conditions to the adsorber, \( T_{Ai} \) and \( Y_{Ai} \).
2. Calculation of gas and solid phase temperature and humidity changes in the adsorption bed using the Lee-Cummings Procedure.
3. Similar application of the Dehler Method to the adsorption column.

8.3.1. Lee-Cummings Procedure

The data obtained from theoretical prediction of non-isothermal adsorption by the Lee-Cummings Procedure are presented in Table 8.2. Essentially, in consideration of an adsorption bed for heat pumping, the most important parameter is the increase in temperature of air as it passes through the bed \( \Delta T \), for any given inlet temperature and humidity conditions, since this reflects the temperature lifting capacity of the system. In addition the breakpoint capacity of the adsorbent is also very important and enables the size of the bed required to be evaluated.
Fig. 8.6: Scheme for Evaluation of Adsorption Models
### Table 8.2: Prediction of Adsorption by Lee-Cummings Procedure

<table>
<thead>
<tr>
<th>$T_{Di}$ ($^\circ$C)</th>
<th>$f$</th>
<th>$T_{ Ae}$ ($^\circ$C)</th>
<th>$W_b$ (%)</th>
<th>$X_b$ (%)</th>
<th>$\Delta T$ ($^\circ$C)</th>
<th>$\Delta y$ ($k_r/k_F$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>.2</td>
<td>52.66</td>
<td>2.38</td>
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<td>8.27</td>
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<tr>
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<td>.6</td>
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<td>3.72</td>
<td>13.63</td>
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<td>4.97</td>
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<td>25.91</td>
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<tr>
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<td>105.20</td>
<td>4.05</td>
<td>7.10</td>
<td>30.02</td>
<td>.0558</td>
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</table>
Fig. 8.7. shows the prediction of $\Delta T$ over a range of drying conditions. Clearly $\Delta T$ improves as $f$ is increased, with a stronger dependence at higher temperature levels. Therefore at relatively low values of $f$, low temperature operation would give a larger $\Delta T$ but at higher values of $f$ the situation would tend to be reversed. The increase in $\Delta T$ with $f$ obviously stems from the attendant increase in humidity as fractional saturation becomes larger and hence a larger heat release on adsorption.

From fig. 8.8, the relationship between breakpoint capacity $w_b$ and fractional saturation is similar to that shown between $\Delta T$ and $f$. Thus, $W_b$ would be expected to increase with saturation but to show more sensitivity at higher temperature levels than at lower temperatures.

From its definition the breakpoint bed relative humidity $X_b$ can be expressed as:

$$X_b = \frac{P_{Ai}}{P_{Ae}}$$  \hspace{1cm} (8.1)

where $P_{Ai}$ is pressure of moisture in inlet air and $P_{Ae}$ is saturation vapour pressure of moisture at outlet temperature. Now, $P_{Ai}$ is a direct function of inlet humidity $Y_{Ai}$ while $P_{Ae}$ is a direct function of outlet temperature $T_{Ae}$. By operating with a higher fractional saturation it is therefore possible to either increase or decrease $X_b$. From figs. 8.7 and 8.8, when the drying temperature is low ($T_{Di} = 50^\circ C$) both $\Delta T$ and $W_b$ increase gradually with $f$. Consequently, the humidity effect on $X_b$ dominates and $X_b$ increases with $f$. Hence $W_b$ and the corresponding $X_b$ increase in the same direction. However with higher drying temperatures ($100^\circ C$ and $150^\circ C$) $\Delta T$ rises more rapidly with $f$ than $W_b$ does. $X_b$ becomes more dependent on the temperature effect and thus decreases with $f$. The net result is that $X_b$ decreases while $W_b$ is increasing. These effects are clearly illustrated in fig. 8.9.
Fig. 8.7: Temperature lift vs fractional saturation

(Lea Cummings Procedure)

$T_{Di}$ is dryer air inlet temperature.

$T_{Di}=50^\circ C, 100^\circ C, 150^\circ C$

Temperature lift $\Delta T$ vs fractional saturation $f$.
**Fig. 8.8 : Bkpt. capacity vs fractional saturation**

*Lee-Cummings Procedure*

$T_{Di}$ is dryer air inlet temperature

$W_b$ (\%) is the break point capacity.

$T_{Di} = 50^\circ C$

Fractional saturation $f$. 

---

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Fig. 8.9: Bkpt. capacity vs bed rel. hum. (Lee-Cummings Procedure)

$T_{Di}$ is dryer air inlet temperature
Fig. 8.10: Temp. lift vs humidity change

(Ted's Chumming Procedure)

$T_{Di}$ is dryer air inlet temperature.

humidity change $\Delta y$
(kgs H$_2$O/kg air).
The variation of temperature change $\Delta T$ with amount of moisture removed in the adsorber $\Delta y$ is presented in fig. 6.1C. $\Delta T$ evidently rises very rapidly with $\Delta y$ owing to increased heat release as more moisture is adsorbed. It is noteworthy also that values of $\Delta T$ less than zero ($\Delta T < 0$) are predicted for conditions of high temperature and low humidity (see fig. 8.7 also).

The data presented in table 8.2 and figs. 8.7 - 8.1C were obtained by solving the Lee-Cummings Model for initial bed temperature $T_{si} = 30^\circ C$ and superficial air velocity $V = 1250$ m/hr. To illustrate the effect of initial bed temperature further calculations have been done for $T_{si} = 60^\circ C$ and the results are shown in Table 8.3. The Lee-Cummings procedure seeks to correlate breakpoint capacity for adiabatic adsorption through the amount of moisture adsorbed (i.e. humidity) and does not take into direct consideration the air outlet temperature and the bed initial and final temperatures. Consequently the values of $\chi_{b}$ predicted by this method are completely independent of $T_{si}$. However since $\Delta T$ is obtained from the heat balance relation

$$T_{ae} = \frac{\chi_{b} \left( \Delta y \lambda A + \frac{c_p T_s}{A_i} \right) + \Delta y \frac{c_p T_s}{A_i}}{\Delta y \frac{c_p T_s}{A_i} + \chi_{b} \frac{c_p T_s}{A_i}}$$

(6.50)

it will show sensitivity to $T_{si}$. If the bed is initially hot, less heat is demanded to raise the temperature of the solids and hence $\Delta T$ increases. This is consequently attended by a reduction in the value of bed relative humidity $\chi_{b}$.

Predictions for velocities of 750, 1250 and 1800 m/hr. are shown in Table 8.4. Evidently there is a slight increase in $\chi_{b}$ with velocity. Also there is a slight rise in $\Delta T$, and hence a fall in $\chi_{b}$, with velocity.
<table>
<thead>
<tr>
<th>( T_{si} = 30^\circ{C} )</th>
<th>( T_{si} = 60^\circ{C} )</th>
</tr>
</thead>
<tbody>
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<td>( \Delta T ) (°C)</td>
<td>( W_b ) (%)</td>
</tr>
<tr>
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<td>41.85</td>
<td>5.21</td>
</tr>
<tr>
<td>30.02</td>
<td>4.05</td>
</tr>
</tbody>
</table>
**TABLE 8.4: EFFECT OF VELOCITY ON ADSORPTION**

*(Lee Cummings Procedure)*

<table>
<thead>
<tr>
<th>$v = 750$ m/hr.</th>
<th>$v = 1250$ m/hr.</th>
<th>$v = 1800$ m/hr.</th>
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</thead>
<tbody>
<tr>
<td>$\Delta T$</td>
<td>$W_b$ (%)</td>
<td>$X_b$ (%)</td>
</tr>
<tr>
<td>$(^\circ C)$</td>
<td>(%)</td>
<td>(%)</td>
</tr>
<tr>
<td>7.80</td>
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<tr>
<td>26.33</td>
<td>3.72</td>
<td>8.08</td>
</tr>
</tbody>
</table>
8.3.2. Dehler Method

In outlining the Dehler Method of describing adsorption, it was proposed that temperature rise in the bed will be obtained from Worthington's equation:

\[ \Delta T = 70 \, h_{1000} \]  

The predicted values of \( \Delta T \) are plotted against fractional saturation \( f \) in fig. 8.11, for various dryer temperatures. Essentially \( \Delta T \) would increase with \( f \) as was the case with the Lee-Cummings Model. However these results differ considerably from those obtained from the Lee-Cummings Model in that over the entire range of fractional saturation larger temperature lifts \( \Delta T \) would be obtained by operating at a higher temperature than at a lower temperature. From the nature of equation 5.34, it is obvious also that only positive values of \( \Delta T \) would be predicted. Overall, it is observed that much larger values of \( \Delta T \) are obtained here than with the Lee-Cummings Model. Furthermore, from equation 5.34 it might have been expected that the plot of temperature rise \( \Delta T \) against humidity change \( \Delta y \) will be a unique characteristic curve. However, since the humidity difference term \( h_{1000} \) is defined in terms of air volume, it is slightly sensitive to temperature and hence separate curves would be obtained for different temperature levels. (fig. 8.14).
| $T_{Di}$  
$^{\circ}$C | $f$ | $T_{oE}$  
$^{\circ}$C | $W_{b}$  
(%) | $X_{b}$  
(%) | $eT$  
$^{\circ}$C | $\gamma$  
(kg/kg) |
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**Fig. 8.11: Temp. lift vs fractional saturation**

(Dehler Method)

$T_{Di}$ is dryer air inlet temp.

- $20^\circ$C
- $150^\circ$C
- $100^\circ$C
- $T_{Di}=50^\circ$C
The Jury-Edwards equation for silica-gel/water vapour equilibrium was used to obtain the equilibrium loadings at the inlet and exit conditions. Thus, from equations 6.55 and 6.59, at the inlet,

\[
X_{i} = \frac{P_{A_i}}{P_{A_e}}
\]  

(6.55)

and at the exit,

\[
X_{e} = \frac{P_{A_i}}{P_{A_e}}
\]  

(6.59)

Defining \( X_f \) by

\[
X_f = X_e \exp \left( \frac{1.8 T_A}{500} \right)
\]

(8.2)

then,

\[
W_{ei} = \frac{2X_{fi}}{1 + 13.33X_{fi}^{3.3}} + \frac{10.15 X_{fi}^{3.3}}{1 + 46.7X_{fi}^{3.3}}
\]

(8.3)

and

\[
W_{ee} = \frac{2X_{fe}}{1 + 13.33X_{fe}^{3.3}} + \frac{10.15 X_{fe}^{3.3}}{1 + 46.7X_{fe}^{3.3}}
\]

(8.4)

\( W_b \) was then obtained from Dehler's equation (6.61) thus:

\[
W_b = \frac{1}{2} (W_{ei} + W_{ee})
\]

(6.61)

Over the range of conditions considered the prediction of this model is that larger breakpoint capacities \( W_b \) would be obtained at low temperatures or high fractional saturations (fig. 8.12). In relation to the Lee-Cummings Model, the values of \( W_b \) obtained here are considerably larger. With the prediction of large temperature rises also small values of bed relative humidity \( X_b \) are obtained and the plots of \( W_b \) against \( X_b \) would therefore tend to be very steep (fig. 8.13). Generally, \( W_b \) would be expected to increase as \( X_b \) increases at all temperatures, unlike the case with Lee-Cummings Model.
Fig. 8.12: Bkpt. capacity vs fractional saturation (Dehler Method).

\( T_D \) is dryer air inlet temperature

\( T_D = 50^\circ C \)

\( T_D = 100^\circ C \)

\( T_D = 150^\circ C \)

\( T_D = 200^\circ C \)

fractional saturation \( f \)
Fig. 8.13: Bkpt. capacity vs bed rel. humidity. 
(Dehler Method)

$T_D$ is dryer air inlet temperature.
Fig. 8.14: Temperature lift vs humidity change.

(Dehler Method)
8.3.3. Modified Dehler Method

Following the apparent over-estimation of $\Delta T$ using Worthington's equation, it is considered that the Dehler Model could be modified by using the heat balance equation (6.50) in conjunction with Dehler's proposal. This would enable $\Delta T$ and $W_b$ to be estimated simultaneously by an iterative procedure.

The results predicted on this basis are shown in Table 8.6 and figs. 8.15-8.18. In general, lower temperature lifts and hence higher bed relative humidities $X_b$ are obtained but $W_b$ is virtually unaffected. With the high temperature and low humidity conditions at the exit end, $W_{ee}$ would tend to be very small. From Dehler's equation above (6.61), $W_b$ would then be virtually determined by $W_{ei}$ which obviously depends on the outlet conditions from the dryer. Since the modification of the model is in no way concerned with the conditions in the dryer, it is understandable that $W_b$ remains unaffected.
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<th>f</th>
<th>$T_{Ac}$ (°C)</th>
<th>$W_b$ (%)</th>
<th>$X_b$ (%)</th>
<th>$\bar{X}$ (°C)</th>
<th>$\Delta y$ (kg/kg)</th>
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Fig. 8.15: Temperature lift vs fractional saturation
(Modified Dehler Method)

\( T_{D_i} \) is dryer air inlet temperature.

Temperature lift \( \Delta T \)
\(^{\circ}\text{C} \)

fractional saturation \( f \)
Fig. 8.16: Bkpt. capacity vs fractional saturation (Modified Dehler Method)

$T_{Di}$ is dryer air inlet temp. $T_{Di} = 50^\circ C$

$W_b$ (%) vs fractional saturation $f$ at $100^\circ C$, $150^\circ C$, and $200^\circ C$
Fig. 8.17: Bkpt. capacity vs bed rel. hum. (Modified Dehler Method)

$T_{D_i} = 50^\circ C$

$T_{D_i}$ is dryer air inlet temperature
Fig. 6.18: Temperature lift vs humidity change (Modified Desler Method)

\[ T_{D1} \text{ is dryer air inlet temperature.} \]
8.4. **Bed Relative Humidity (BRH) Model**

The discussions of the previous section indicate that the Lee-Cummings and Dehler Models give measurably disparate predictions of adsorption under the kind of conditions encountered. It is considered that perhaps these disparities arise because even though the concepts embodied in the model were developed for non-isothermal conditions they may not be appropriate for the unusually high temperatures and humidities that are tenable in the present study but which scarcely arise in ordinary adsorption systems. Consequently, an alternative model has been derived using the experimental results presented in the last chapter. Since these were mostly obtained under high temperature and humidity conditions which simulate those that would be expected from a dryer this model would provide a suitable basis for testing the other models.
8.4.1. Outline of Model

Correlation of the data obtained from experiments led to the derivation of the following relationship between breakpoint capacity and breakpoint bed relative humidity for silica-gel/water system:

\[ W_b = 1.07X_b \quad (7.10) \]

The Bed Relative Humidity (BRH) Model, derived from the experimental data, is based essentially on the use of this equation in combination with the lumped heat balance equation for an adsorption column at breakpoint

\[ T_{Ae} = \frac{W_b (\Delta Y \lambda_A + c_p T_A)}{\Delta Y c_p x + W_b c_p x} \quad (6.50) \]

to obtain the outlet temperature of air and final loading of moisture on the adsorbent.

8.4.2. Evaluation of Model

Predictions of adsorption temperature lift and breakpoint capacity using this model have been done and the results are presented in Table 8.7 and figs. 8.19 - 8.22, in the same manner as with the Lee-Cummings and Dehler Models.

Fig. 8.19 shows the predicted temperature lift \( \Delta T \) against fractional saturation \( f \). Relating this to the predictions of the other models, it is clear that the BRH and Lee-Cummings Models show essentially similar characteristics in that low temperatures would give larger temperature lifts at low fractional saturation and smaller temperature lifts at high fractional saturation. Negative temperature lifts (i.e. temperature depressions) are obtained with high temperatures and low saturations. However, the data obtained from Lee-Cummings predictions show a stronger dependence on fractional saturation at high temperatures (> 100°C) with the result that the curves rise more steeply.

The predicted breakpoint capacity \( W_b \) is presented as a function of fractional saturation and temperature in fig. 8.20. The decrease in \( W_b \) as temperature is increased is a direct reflection of the unfavorable effect of temperature on adsorption, which has been noted earlier in chapter 7 from the experimental results. The predictions of Lee-Cummings Model differ from this since larger or smaller breakpoint
TABLE 8.7: PREDICTION OF ADSORPTION BY BED RELATIVE HUMIDITY MODEL

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<th>$T_{Di}$ (°C)</th>
<th>$T_{ae}$ (°C)</th>
<th>$W_{b}$ (%)</th>
<th>$X_{b}$ (%)</th>
<th>$\Delta T$ (°C)</th>
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Fig. 6.19: Temperature lift vs fractional saturation
(BRH Model)

\[ T_{Di} = \text{dryer air inlet temp.} \]

Temperature lift

\[ \Delta T \text{ (°C)} \]

Fractional saturation \( f \)

\(-60\) \( -50\) \( -40\) \( -30\) \( -20\) \( -10\) \( 0\) \( 10\) \( 20\) \( 30\) \( 40\)
Fig. 8.20: Bkpt. Capacity vs fractional saturation (BK: Model)

$T_{Di}$ is dryer air inlet temperature.

$T_{Di} = 50^\circ C$

Fractional saturation $f$. 
Fig. 8.21: Bkpt. capacity vs bed rel. humidity (BE Model)
Fig. 6.22: Temperature lift vs humidity change (ERH Mrdl)

$T_{Di} = 50^\circ C$, $100^\circ C$, $150^\circ C$, $200^\circ C$

Humidity change by
(kg H$_2$O/kg air)

$T_{Di}$ is dryer air inlet temperature.
capacities would be obtained with low temperatures, depending on the fractional saturation. To the extent that larger breakpoint capacities are predicted for low temperature operation over the entire range of fractional saturation, the predictions of the Dehler Model are similar to these. However, comparison of figs. 8.12 (or 8.16) and 8.20 shows that $W_b$ predicted from Dehler Model shows much more sensitivity to fractional saturation.

Since equation 7.10 is presumed to apply over a wide range of conditions the values of $W_b$ and $X_b$ predicted by this model understandably lie on a unique curve (fig. 8.21). In this regard therefore, the predictions of this model differ markedly from those of the other two models, especially the Lee-Cummings Model.

The temperature lifts obtained experimentally were presented as a fraction of humidity and temperature in fig. 7.10. Similar data, predicted from the derived model, are plotted in fig. 8.22 and they also show similar characteristics.

The effect of initial bed temperature $T_{si}$ on the prediction of breakpoint capacity and final temperature has been examined also. Results for an initial cold temperature ($30^\circ$C) and an initial hot temperature ($60^\circ$C) are presented in Table 8.8. Evidently, adsorbing on a hot bed gives higher temperature lifting capability, as has been noted previously in consideration of the Lee-Cummings Model. This is consequently attended however by a reduction in bed relative humidity $X_b$ and hence, from equation 7.10, a reduction in breakpoint capacity $W_b$ also.

Though the predictions obtained from the Lee-Cummings and Dehler Models variously agree with the experimental deductions in certain respects, very significant discrepancies have been noted in many other respects. In general the predictions of breakpoint capacity and temperature lift from Lee-Cummings Model tend to be lower than those obtained from the BRH Model. On the other hand the values predicted by the Dehler Model tend to be larger than those predicted by the BRH Model, especially at high fractional saturations. These observations therefore seem to indicate further that the Lee-Cummings and Dehler Models are inadequate for the severe temperature and humidity conditions considered. Besides, in correlating adiabatic adsorption by means of
TABLE 8.8: EFFECT OF INITIAL BED TEMPERATURE ON ADSORPTION

(BED RELATIVE HUMIDITY MODEL)

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<th>( X_b ) (%)</th>
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</table>
humidity, Lee-Cummings Model vitally neglects the direct effects of the final temperature conditions in the bed. Clearly, both humidity and temperature constitute the most important parameters that affect adsorption. Concentration on correlation in terms of humidity while neglecting temperature therefore renders the procedure defective. Regarding the Dehler Model, it is evident that the conditions of the influent air to the adsorption bed influence the predictions rather too strongly. In addition, the assumption that half of the bed is unused would seem to be rather arbitrary and is unlikely to hold for all conditions.

In view of these, since the BRH Model is derived from the experimental data obtained under more appropriate conditions with regard to heat pumping requirements and strongly reflects the effects of temperature and humidity, both initially and finally, it will provide the basis for further prediction of the overall performance of the AHP dryer.

8.5. Overall Cycle Prediction by Bed Relative Humidity Model

8.5.1 Thermal Efficiency.

The prediction of thermal efficiency of the TSC heat pump dryer $\eta_{RH}$ by this model is illustrated in fig. 8.23 in relation to the thermal efficiency of a comparable conventional non-recycle dryer, $\eta_{conv}$. The comparable conventional dryer is considered to be such that it operates at the same temperature and fractional saturation as the dryer incorporated in the heat pump dryer. Both efficiencies are evidently dependent on temperature and fractional saturation. While the efficiency of the heat pump dryer shows more sensitivity to temperature the efficiency of the conventional dryer depends more strongly on fractional saturation. It can therefore be deduced that AHP drying would provide considerable advantage over conventional drying under conditions of low temperature and low fractional saturation. At high temperatures and fractional saturations conventional drying would however be preferable. Thus, for any temperature $T_{Di}$, there exists a maximum fractional saturation $f$ at which the efficiency of the AHP dryer would exactly equal that of the conventional dryer. If thermal
### Table 6.9: Prediction of Performance of TSC Drier

**(BRH Model)**

<table>
<thead>
<tr>
<th>$T_D$ (°C)</th>
<th>$f$</th>
<th>$\eta_{RH}$ (%)</th>
<th>$\eta_{conv.}$ (%)</th>
<th>COP$_{max.}$ (KJ/KJ)</th>
<th>COP (KJ/KJ)</th>
<th>$r_T$ (%)</th>
<th>$K_s$ (kr silica-rel/kr air)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>.2</td>
<td>39.72</td>
<td>4.15</td>
<td>22.56</td>
<td>.43</td>
<td>1.86</td>
<td>.10</td>
</tr>
<tr>
<td></td>
<td>.4</td>
<td>47.88</td>
<td>23.03</td>
<td>11.08</td>
<td>.54</td>
<td>4.59</td>
<td>.14</td>
</tr>
<tr>
<td></td>
<td>.6</td>
<td>54.64</td>
<td>41.34</td>
<td>7.46</td>
<td>.63</td>
<td>8.50</td>
<td>.15</td>
</tr>
<tr>
<td></td>
<td>.8</td>
<td>59.31</td>
<td>59.22</td>
<td>5.73</td>
<td>.69</td>
<td>12.12</td>
<td>.16</td>
</tr>
<tr>
<td>100</td>
<td>.2</td>
<td>25.84</td>
<td>11.12</td>
<td>-37.30</td>
<td>-10</td>
<td>.28</td>
<td>.51</td>
</tr>
<tr>
<td></td>
<td>.4</td>
<td>34.94</td>
<td>27.57</td>
<td>12.25</td>
<td>.22</td>
<td>1.78</td>
<td>.60</td>
</tr>
<tr>
<td></td>
<td>.6</td>
<td>42.00</td>
<td>44.57</td>
<td>5.32</td>
<td>.42</td>
<td>7.80</td>
<td>.63</td>
</tr>
<tr>
<td></td>
<td>.8</td>
<td>48.14</td>
<td>60.50</td>
<td>3.61</td>
<td>.56</td>
<td>15.50</td>
<td>.62</td>
</tr>
<tr>
<td>150</td>
<td>.2</td>
<td>22.08</td>
<td>13.45</td>
<td>-2.46</td>
<td>-.87</td>
<td>35.35</td>
<td>1.13</td>
</tr>
<tr>
<td></td>
<td>.4</td>
<td>31.85</td>
<td>29.93</td>
<td>-12.71</td>
<td>-.13</td>
<td>1.02</td>
<td>1.21</td>
</tr>
<tr>
<td></td>
<td>.6</td>
<td>39.19</td>
<td>46.57</td>
<td>6.71</td>
<td>.22</td>
<td>3.33</td>
<td>1.22</td>
</tr>
<tr>
<td></td>
<td>.8</td>
<td>43.99</td>
<td>62.38</td>
<td>3.33</td>
<td>.42</td>
<td>12.56</td>
<td>1.24</td>
</tr>
<tr>
<td>200</td>
<td>.2</td>
<td>20.97</td>
<td>14.60</td>
<td>-0.54</td>
<td>-1.74</td>
<td>32.14</td>
<td>1.75</td>
</tr>
<tr>
<td></td>
<td>.4</td>
<td>31.34</td>
<td>31.56</td>
<td>-2.18</td>
<td>-.46</td>
<td>21.80</td>
<td>1.84</td>
</tr>
<tr>
<td></td>
<td>.6</td>
<td>38.19</td>
<td>47.88</td>
<td>31.84</td>
<td>.03</td>
<td>.11</td>
<td>1.86</td>
</tr>
<tr>
<td></td>
<td>.8</td>
<td>43.33</td>
<td>64.42</td>
<td>3.37</td>
<td>.33</td>
<td>9.67</td>
<td>1.85</td>
</tr>
</tbody>
</table>
**Fig. 6.23: Thermal Efficiency of TSC Dryer (BRH Model)**

- Efficiency of conventional dryer, $\eta_{\text{conv}}$
- Efficiency of TSC dryer, $\eta_{\text{Ri}}$

$T_{Di}$ is air inlet temperature to dryer.

<table>
<thead>
<tr>
<th>Temperature ($^\circ$C)</th>
<th>Efficiency $\eta$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td></td>
</tr>
<tr>
<td>100</td>
<td></td>
</tr>
<tr>
<td>150</td>
<td></td>
</tr>
<tr>
<td>200</td>
<td></td>
</tr>
</tbody>
</table>

$T_{Di} = 50^\circ$C

$T_{Di} = 100^\circ$C

$T_{Di} = 150^\circ$C

$T_{Di} = 200^\circ$C

fractional saturation $f$
efficiency is the only criterion for comparing the two dryers, this may be deemed to be the efficiency at which the AHP dryer must be replaced by a conventional dryer. The replacement efficiencies for various temperatures, deduced from the plots of fig. 8.23, are presented in Table 8.10.

<table>
<thead>
<tr>
<th>T_{Di} (°C)</th>
<th>f</th>
<th>Conv. dryer effcy. η (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>0.90</td>
<td>61.5</td>
</tr>
<tr>
<td>100</td>
<td>0.56</td>
<td>41</td>
</tr>
<tr>
<td>150</td>
<td>0.45</td>
<td>34</td>
</tr>
<tr>
<td>200</td>
<td>0.41</td>
<td>31</td>
</tr>
</tbody>
</table>

From the table, it is clear that the AHP dryer will be replaced by a conventional dryer at an efficiency of 61.5%, with a corresponding fractional saturation of 0.9, for a drying temperature of 50°C. But for 200°C, it will be replaced at only 31% efficiency, and 0.4 fractional saturation. A more succinct illustration is provided by plotting the loci of these replacement points in fig. 8.24. This enables values to be determined for intermediate temperatures. For example at 120°C, the replacement efficiency will be 38%, with 0.51 fractional saturation, as shown in the graph.
Fig. 8.24: Loci of Replacement Efficiency and Fractional Saturation of TSC Dryer.
8.5.2. Comparison with Mechanical Heat Pump Drying

Following the discussion of Section 4.2, the maximum MER that could be expected from a closed cycle mechanical heat pump dryer for the present maximum attainable temperature of 100°C is about 3-4 kg/KWH. Using an MER value of 3.5 in equation 4.7 the thermal efficiency of the system would be about 70%. Now, this can be related to the AHP dryer by referring to fig. 8.23. It is evident then that for a temperature of 100°C the AHP dryer on its part cannot be expected to exceed an efficiency of about 55%. However, it is interesting to note that the two systems in consideration have opposite performance characteristics, in relation to temperature. While lower efficiencies are attainable with low temperature in mech heat pump drying, higher efficiencies are attained in the AHP dryer. For the previous maximum attainable temperature of about 50°C and the reported associated MER of about 2.5 kg/KWH, the equivalent efficiency would be 50%. But from fig. 8.23, considerably higher efficiencies, up to 60-65%, could be expected for the AHP dryer operating at the same temperature. Hence it can be inferred that for low temperatures AHP drying will likely provide more scope for heat recovery than mechanical heat pump drying but for high temperatures the reverse may hold.

8.5.3. Coefficient of Performance

COP, as has been stated, is a measure of the heat pumping efficiency of a heat pump system. The theoretical maximum COP and the estimated actual COP of the TSC heat pump are presented in figs. 8.25 and 8.26. From equations 6.30,

$$\text{COP}_{\text{max}} = \left(\frac{T_{ae}}{T_{ae} - T_{de}}\right) \left(\frac{T_{ri} - T_{de}}{T_{ri}}\right)$$  \hspace{1cm} (6.30)

which was derived by analogy with a heat driven Carnot engine, it is clear that the theoretical COP may be very high when $T_{re}$ and hence $\Delta T$, is low and tends to infinity as $\Delta T$ approaches zero.
It has been shown that under certain conditions the inequality 
\( \Delta T < 0 \) may hold. Under such conditions then \( \text{COP}_{\text{max}} \) may also 
become negative. Similarly, from equation 6.31.

\[
\text{COP} = \frac{T_{\text{Ae}} - T_{\text{De}}}{(1-r_D)(T_{\text{Ri}} - T_{\text{ Ae}})} \tag{6.31}
\]

it is clear that the actual \( \text{COP} \) of the system may also become negative 
when \( \Delta T \) is negative. This phenomenon is illustrated in figs. 8.25 
and 8.26 by dashed strokes. However, it is noted that the magnitude 
of the negative values of \( \text{COP}_{\text{max}} \) and \( \text{COP} \) do not have any practical 
significance except merely indicating situations of net energy input 
to the system rather than output.

The predictions indicate that the actual cycle \( \text{COP} \) generally 
increases with fractional saturation but decreases with air inlet 
temperature, as shown in fig. 8.26. A comparison with fig. 8.23 
shows that \( \text{COP} \) is favoured by the same conditions that favour the 
overall thermal efficiency of the cycle. However, even under the 
most favourable conditions the highest attainable predicted \( \text{COP} \) is 
less than 0.8. This implies that even at best more energy actually 
needs to be put into the heat pump than it delivers to the dryer.

The values of theoretical \( \text{COP} \) and actual \( \text{COP} \) have been used to 
estimate the thermodynamic efficiency of the heat pump and the data 
are plotted in fig. 8.27. Evidently, since the conditions that 
produce negative values of \( \text{COP}_{\text{max}} \) and \( \text{COP} \) are the same, from equation, 
4.9 
\[
\eta_T = \frac{\text{COP}}{\text{COP}_{\text{max}}} \tag{4.9}
\]

positive values of thermodynamic efficiency may be obtained when both 
\( \text{COP}_{\text{max}} \) and \( \text{COP} \) are negative. Since such values of \( \text{COP} \) can only be of 
trivial significance, the corresponding values of \( \eta_T \) are also equally 
trivial. Values of \( \eta_T \) corresponding to negative \( \text{COP} \)'s are also 
indicated by dashed strokes in fig. 8.27. Bearing this in mind, it 
is observed that generally low values of effective \( \eta_T \), of the order 
5-20\%, are predicted. This poor performance can be interpreted to 
mean a very severe degradation of the high quality heat input to the 
desorbing column.
Fig. 8.25: Theoretical COP of TSC Heat Pumps

T_{Di} is air inlet temperature to dryer.

\( \times \) T_{Di} = 50°C
\( \circ \) T_{Di} = 100°C
\( \triangle \) T_{Di} = 150°C
\( \triangledown \) T_{Di} = 200°C
Fig. 8.26: Actual COP of TSC Heat Pump
Fig. 8.27: Thermodynamic Efficiency of TEC Heat Pump

- $\cdot T_{Di} = 50^\circ C$
- $\circ T_{Di} = 100^\circ C$
- $\triangle T_{Di} = 150^\circ C$
- $\nabla T_{Di} = 200^\circ C$

$T_{Di}$ is air inlet temperature to dryer.

Thermodynamic efficiency $\eta_T$
8.5.4. Adsorbent Bed Size

The amount of adsorbent and hence the size of the adsorption column required for any given conditions of operation can be estimated from the predicted adsorbent bed capacity and the rate of moisture removal in the column. Thus:

\[ M_s = \frac{GQ_b \Delta y}{W_b} \]  \hspace{1cm} (8.5)

where \( M_s \) is weight of adsorbent required and \( Q_b \) is the cycle time required for the bed to attain breakpoint before it is regenerated.

For a cycle time of 2 hours and airflow rate of 1 kg/hr, the silica-gel adsorbent requirement has been predicted for the range of conditions considered. The results are presented in fig. 8.28. Evidently, \( M_s \) will be virtually independent of the air fractional saturation in the dryer \( f \) and very heavily dependent on dryer air inlet temperature \( T_{Di} \). Following the discussions of sections 8.1 and 8.4, both breakpoint capacity \( W_b \) and moisture removal in the bed \( \Delta y \) increase as \( f \) and will thus tend to produce counter-balancing effects on \( M_s \), keeping it fairly constant as \( f \) varies. On the other hand, while \( \Delta y \) increases with \( T_{Di} \), \( W_b \) is adversely affected and hence \( M_s \) will increase very considerably with \( T_{Di} \). Since the smallest bed size would imply least capital expenditure, low temperature conditions would therefore be considered most favourable in this regard.

8.5.5. Initial Bed Temperature

Calculations have been done for a dryer operated under the two sets of conditions indicated in Table 8.11 and with initial bed temperatures \( (T_{Si}) \) of 70°C and 90°C respectively. The results are shown in the table against similar results for a bed initially cooled to 30°C. The analysis of section 8.4. has showed that if the adsorbent bed is not allowed to cool down sufficiently after regeneration, the effect is a higher temperature lift but a lower breakpoint capacity. Thermal efficiency would therefore be expected to be improved by adsorbing on a hot bed, though a severe penalty would be incurred by way of substantially increased bed size requirement.
Fig. 8.28: Adsorbent bed size for TSC dryer

Mass of solids $M_s$

(kg silica-gel) / kg air

Temperature:
- $T_{di} = 200$
- $150^\circ C$
- $100^\circ C$
- $50^\circ C$

Fractional saturation $t$
TABLE 8.11: EFFECT OF INITIAL BED TEMPERATURE
ON PERFORMANCE OF TSG DRYER

(BRH MODEL)

<table>
<thead>
<tr>
<th></th>
<th>$T_{Di}=100^\circ C$, $f=0.6$</th>
<th>$T_{Di}=150^\circ C$, $f=0.4$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$T_{si}=30^\circ C$</td>
<td>$T_{si}=70^\circ C$</td>
</tr>
<tr>
<td>$X_b$ (%)</td>
<td>6.40</td>
<td>4.55</td>
</tr>
<tr>
<td>$W_b$ (%)</td>
<td>4.59</td>
<td>3.48</td>
</tr>
<tr>
<td>$M$ (kg)</td>
<td>0.63</td>
<td>0.84</td>
</tr>
<tr>
<td>$\eta_{RH}$ (%)</td>
<td>42.00</td>
<td>43.68</td>
</tr>
<tr>
<td>$\eta_{conv}$ (%)</td>
<td>44.57</td>
<td>44.56</td>
</tr>
<tr>
<td>COP max</td>
<td>5.32</td>
<td>3.83</td>
</tr>
<tr>
<td>COP</td>
<td>0.42</td>
<td>0.60</td>
</tr>
<tr>
<td>$\eta_T$</td>
<td>7.80</td>
<td>15.67</td>
</tr>
</tbody>
</table>
For the latter set of conditions considered ($T_{Di} = 150^\circ C$, $f = 0.6$), it is observed that $m_s$ almost doubles while $n_{RH}$ increases by about 15%. The COP and thermodynamic efficiency of the heat pump are also improved when $T_{Si}$ is high.

8.5.6. Regeneration Temperature

Table 8.12 shows a comparison between regeneration at $200^\circ C$ and $150^\circ C$ for two sets of dryer operating conditions. Apparently $m_s$ is not affected by the temperature of regeneration $T_{Ri}$. Thermal efficiency would however be expected to diminish with lower regeneration temperature, in the same manner as has been discussed for the Equilibrium Model. Though $n_{RH}$ drops, COP and $n_T$ improve with lower $T_{Si}$. This therefore, implies that despite the decline in overall energy performance there is less degradation of heat in the system when regeneration is carried out at lower temperature.

| TABLE 8.12 : EFFECT OF REGENERATION TEMPERATURE ON PERFORMANCE OF TSC DRYER (BPH MODEL) |
|--------------------------------------------|--------------------------------------------|--------------------------------------------|--------------------------------------------|
| $T_{Di} = 100^\circ C$, $f = 0.6$ | $T_{Di} = 150^\circ C$, $f = 0.4$ |
| $T_{Ri} = 200^\circ C$, $T_{Ri} = 150^\circ C$ | $T_{Ri} = 200^\circ C$, $T_{Ri} = 150^\circ C$ |
| $X_b$ (%) | 6.40 | 6.27 | 3.04 | 3.11 |
| $W_b$ (%) | 4.59 | 4.61 | 2.54 | 2.55 |
| $M_s$ (kg) | 0.63 | 0.64 | 1.21 | 1.24 |
| $n_{RH}$ (%) | 42.00 | 37.82 | 31.65 | 29.58 |
| $n_{conv.}$ (%) | 44.57 | 44.47 | 29.93 | 30.61 |
| COP | 5.32 | 3.86 | -12.72 | -6.94 |
| COP | 0.42 | 0.46 | -1.13 | -1.16 |
| $n_T$ | 7.80 | 11.79 | 1.02 | 2.34 |
8.5.7. **PSC Dryer**

Since the analysis of the PSC dryer based on the Heat and Mass Balance Model has clearly indicated that it will not improve the thermal efficiency of drying processes and that it is less efficient than the TSC dryer, further consideration on the basis of the BRH Model cannot be expected to provide more encouraging results. Nevertheless, application of the latter to the PSC dryer is worth considering, in view of the fact that it enables the solid adsorbent requirement to be estimated.

Predictions for TSC and PSC drying systems, operating under the same drying conditions, are shown in Table 8.13. Though the data reinforce the superior thermal performance of TSC over PSC, the required bed size for the latter would clearly be much smaller. This arises directly from the higher bed relative humidity $X_b$ and thus higher breakpoint capacity $W_b$ which are associated with the high pressure condition in the adsorption bed. In this regard therefore the PSC dryer is seen to offer considerable advantage over the TSC dryer. Investigation of the impacts of the pressure $P$ and condenser temperature $T_{Ai}$ (Table 8.14) on bed size, also show that high pressure and low condenser temperature would tend to give larger $X_b$ and $W_b$ and hence lead to smaller bed size.

The small bed size requirement generally associated with PSC drying may be deceitful however, considering that extra capital outlay will be incurred in incorporating a compressor and a cooler in the cycle. The cost of the cooler is partially offset though by the fact that the need for a regeneration heater is obviated. Increasing the pressure or reducing the condenser temperature, while leading to even smaller beds, will imply even more capital expenditure for a larger compressor or cooler. In addition, more capital cost will also be incurred in making the adsorption column into a pressure vessel.
TABLE 8.13: COMPARISON OF PERFORMANCE OF TSC and PSC DRYERS (BRH MODEL)

<table>
<thead>
<tr>
<th></th>
<th>TSC</th>
<th>PSC</th>
</tr>
</thead>
<tbody>
<tr>
<td>$X_b$ (%)</td>
<td>4.83</td>
<td>33.18</td>
</tr>
<tr>
<td>$W_b$ (%)</td>
<td>3.77</td>
<td>17.45</td>
</tr>
<tr>
<td>$M_s$ (kg)</td>
<td>1.22</td>
<td>.12</td>
</tr>
<tr>
<td>$n_{RH}$ (%)</td>
<td>39.19</td>
<td>9.34</td>
</tr>
<tr>
<td>COP$_{max}$</td>
<td>6.71</td>
<td>1.00</td>
</tr>
<tr>
<td>COP</td>
<td>.22</td>
<td>.06</td>
</tr>
<tr>
<td>$n_T$ (%)</td>
<td>3.33</td>
<td>6.22</td>
</tr>
</tbody>
</table>

TABLE 8.14: EFFECTS OF TOTAL PRESSURE AND CONDENSER TEMPERATURE ON PERFORMANCE OF PSC DRYER (BRH MODEL)

<table>
<thead>
<tr>
<th></th>
<th>$\pi = 6$ atms $T_{Ai} = 50^\circ$C</th>
<th>$\pi = 10$ atms $T_{Ai} = 50^\circ$C</th>
<th>$\pi = 10$ atms $T_{Ai} = 75^\circ$C</th>
</tr>
</thead>
<tbody>
<tr>
<td>$X_b$ (%)</td>
<td>33.18</td>
<td>50.52</td>
<td>26.79</td>
</tr>
<tr>
<td>$W_b$ (%)</td>
<td>17.45</td>
<td>26.45</td>
<td>14.73</td>
</tr>
<tr>
<td>$M_s$ (kg)</td>
<td>.12</td>
<td>.05</td>
<td>.27</td>
</tr>
<tr>
<td>$n_{RH}$ (%)</td>
<td>9.34</td>
<td>8.24</td>
<td>7.32</td>
</tr>
<tr>
<td>COP$_{max}$</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>COP</td>
<td>.06</td>
<td>.02</td>
<td>.06</td>
</tr>
<tr>
<td>$n_T$</td>
<td>6.22</td>
<td>2.47</td>
<td>5.73</td>
</tr>
</tbody>
</table>
CHAPTER NINE

CONCLUSIONS AND SUGGESTIONS FOR FURTHER WORK
9. CONCLUSIONS AND SUGGESTIONS FOR FURTHER WORK


In conclusion, the general deductions that may be drawn from this study are presented as follows:

1. Measurements taken on an industrial convective tumble dryer show that the dryer operates at a thermal efficiency less than 40% and that 55-60% of the heat supplied to the dryer is lost as sensible heat in the air exhausted from the dryer. This is considered typical of many industrial dryers and clearly indicates that the main scope for improvement of thermal efficiency of dryers is in recovering the heat lost in the exhaust air.

2. Two conventional methods of heat recovery have been considered, namely:

   (a) exhaust air recirculation,
   (b) heat exchange recovery.

(a) Calculations show that air recirculation can affect the thermal efficiency of a drying process either favourably or adversely, depending on the mode of operation. Operating with a constant fractional saturation of air, the best thermal efficiency would be obtained with the highest possible recirculation ratio, whilst the best efficiency for operation with fixed wet-bulb temperature would be obtained with the minimum permissible recirculation. It was shown however that the conditions which favour improved efficiency in either operational mode are also those that retard the rate of drying. A method of optimizing air recirculation in terms of thermal efficiency and drying rate has been suggested.

(b) Drying with a heat exchange recuperator helps improve thermal efficiency, especially under low fractional saturation and high temperature drying conditions, but in view of the very low heat transfer coefficients associated with air-air heat exchange and the low temperature driving force obtained, this necessitates intolerably large heat exchanger surface areas.
3. Experiments on adsorption of water vapour from air on silica-gel adsorbent affirmed that temperature and humidity are the most important factors that affect the process. In particular, the results show that with a combination of high temperature and fairly low humidity conditions a depression of air temperature as it passes through the adsorption column would be obtained, rather than temperature lifting. Hence under such conditions air left the column colder than it entered, i.e. $\Delta T < 0$.

4. By defining the bed relative humidity at breakpoint $X_b$ as a function of air inlet humidity $Y_{Ai}$ and average bed temperature $T_{Ae}$ thus:

$$X_b = f(Y_{Ai}, T_{Ae})$$

the adsorption process was found to be described by a polynomial function, similar to the system equilibrium relation, of the form:

$$W_b = KX_b^n$$

where $K$ and $n$ are constants and $W_b$ is the breakpoint loading of moisture on silica-gel.

Correlation of the experimental data produced the following equation:

$$W_b = 1.07 X_b^{0.81}$$

5. The correlation was used to set up the Bed Relative Humidity (BRH) Model for predicting the performance of the Adsorption Heat Pump (AHP) Dryer. The predictions of this model enabled similar predictions obtained from two other models synthesized from various proposals put forward by other investigators to be tested against experimental results. These other models, Lee-Cummings and Dehler, variously agreed with the BRH Model in certain respects but in general they were found to be inadequate for the unusually high temperature and humidity conditions encountered. The BRH Model was therefore considered to be most appropriate for predicting the system performance.
6. Comparison of predictions of performance of the AHP dryer operated as temperature swing cycle (TSC) and as pressure swing cycle (PSC) showed that owing to the large energy demand for compression, the TSC dryer would provide much higher cycle thermal efficiencies.

7. The performance characteristics of the TSC dryer showed that the best thermal efficiencies, up to 60%, would be expected if the dryer is operated with low temperature and high fractional saturation. However, in comparison with an equivalent non-recycle conventional dryer which is deemed to operate at the same temperature and fractional saturation, the TSC dryer would be expected to provide measurable advantage only at low temperatures (<100°C) or low fractional saturations (<0.5). This arises mainly because the efficiency of the conventional dryer was found to be much more sensitive to fractional saturation than that of the TSC dryer.

8. In relation to mechanical heat pump drying the AHP dryer was found to compare favourably only at low temperatures (<100°C).

9. The predicted coefficient of performance (COP) of the heat pump is such that more heat would be put into the cycle to regenerate the used bed than would be extracted from the dryer effluent air. In addition, very low thermodynamic efficiencies, less than 20%, are predicted, implying that the heat supplied for regeneration at high temperature would be substantially degraded in the system.

10. Very large adsorbent bed sizes are estimated for high temperature drying but smaller beds would be required for lower temperatures.

11. In view of both the poor thermal performance and the large bed size requirements, AHP drying will not be appropriate for the high temperature conditions for which it was initially conceived. Nevertheless, the cycle seems attractive for low temperature (<100°C) application.
9.2 Suggestions for Further Work

1. Following the above conclusions, though the original purpose of the AHP dryer is lost, it may be useful to investigate its applicability to low temperature drying further. In this regard, there seems to be sufficient justification for construction of a prototype AHP dryer for practical demonstration and testing of the performance of the overall unit.

2. Data obtained from practical tests will enable a more precise comparison to be made with conventional drying and with mechanical heat pump drying. This will also enable the appropriateness of the present theoretical prediction model to be tested. In particular the practical performance data will highlight other losses, including those associated with pressure drops across the adsorbent beds and along the conduits, which have not been sufficiently considered yet.

3. There may be need for more rigorous modelling of the system, especially in the adsorption bed, if practical data are unmatched by the present theoretical predictions.

4. The use of silica-gel adsorbent here was based mainly on its relative cost advantage. It may be interesting to extend the consideration to other types of adsorbent, especially molecular sieves.

5. The A.H.P. may have possible applications in domestic air conditioning and heating and it may be useful to investigate this further.
REFERENCES
REFERENCES


66, (1944), 1664.


APPENDIX
APPENDIX A

THERMODYNAMIC AND TRANSPORT PROPERTIES.

A1. Vapour Pressure

Clayperon developed a mathematical relationship between vapour pressure of a pure substance and temperature:

\[
\frac{dP}{dT} = \frac{\lambda}{T(V_G - V_L)} \quad \text{A1.1.}
\]

where \( P \) = vapour pressure
\( T \) = absolute temperature
\( \lambda \) = heat of vaporisation at temperature \( T \)
\( V_G \) = volume of gas
\( V_L \) = volume of liquid.

If the volume of liquid is negligible in relation to the gas volume and the ideal gas law is presumed to apply, the above relationship reduces to the Clausius-Clapeyron equation:

\[
\frac{dP}{P} = \frac{\lambda_m}{RT^2} \, dT \quad \text{A1.2}
\]

where \( R \) = gas law constant
and \( \lambda_m \) = molal heat of vaporisation.

Then, if the term \( \frac{\lambda_m}{R} \) is assumed to be constant and independent of temperature,

\[
\ln P = A - \frac{B}{T} \quad \text{A1.3}
\]

\( A, B \) are constants.

Obviously, equation A.1.3 can be reliable only when the vapour pressure is relatively low, such that it obeys the ideal gas laws and when the liquid phase volume is very small.
Besides, the temperature range over which it is applicable will also be narrow in order for molal heat of vaporisation to remain constant. Antoine proposed a modification of this equation which enables it to be applied over a wider range of pressures and temperatures:

\[ \ln p = A - \frac{B}{T + C} \]  

where C is also a constant.

It has been found that for most materials C is approximately \(-43^\circ K\),\(^{51}\) and hence,

\[ \ln p = A - \frac{B}{T - 43} \]  

From data provided by Hougen, Watson and Ragatz\(^{51}\), values of constants A and B for water would be 18.4 and 3883 respectively when vapour pressure is expressed in millimeters of mercury (mmHg) and temperature in \(^\circ K\). Hence vapour pressure of water may be expressed as:

\[ \ln p = 18.4 - \frac{3883}{T-43} \]  

If T is in \(^\circ C\), then

\[ \ln p = 18.4 - \frac{3883}{T+230} \]  

This equation has been used to find the vapour pressure of moisture at different temperatures.
A2. Heat of Vaporisation

The heat of vaporisation of a substance is a function of the temperature. As temperature is increased, heat of vaporisation decreases. At the critical point, there is no distinction between the liquid and vapour states, either in enthalpy or in physical properties, and heat of vaporisation therefore becomes zero.

According to Trouton, the ratio of the molal heat of vaporisation $\lambda_m$ of a substance at its normal boiling point to the absolute temperature $T_b$ is a constant. Thus,

$$\frac{\lambda_m}{T_b} = K_T$$  \hspace{1cm} A2.1

$K_T$ is termed Trouton's ratio. For many substances, especially non-polar liquids, this ratio is approximately equal to 21, where the latent heat is expressed in cals/gram mole and temperature is in °K. For polar liquids, values of this ratio tend to be much greater than 21.

Watson extended the Trouton concept and produced a relationship that would apply reasonably well to both polar and non-polar liquids. He found that the following empirical equation satisfactorily represents heats of vaporisation over a wide range of temperatures,

$$\frac{\lambda_2}{\lambda_1} = \frac{1 - T_{r_2}}{1 - T_{r_1}}$$  \hspace{1cm} A2.2

$T_{r_1}, T_{r_2}$ are reduced temperatures at states 1 and 2 and $\lambda_1, \lambda_2$ are corresponding heats of vaporisation at states 1 and 2. If heat of vaporisation at a standard temperature, say normal boiling point, is known, then heat of vaporisation at any given temperature can be readily obtained.
For water, with a normal boiling point of 100°C (373°K) and a corresponding heat of vaporisation of 2256 KJ/Kg, as well as a critical temperature of 647.3°K, then heat of vaporisation at any temperature would be:

\[ \lambda = 2256 \left( \frac{1 - \frac{T}{T_c}}{0.4238} \right) \]

This equation has been used to estimate the heat of vaporisation of moisture at different temperatures.

A3. **Heat of Adsorption**

Adsorption of a vapour on an adsorbent is accompanied by evolution of heat. If chemisorption takes place the heat evolved is usually of the order of the heat of reaction between the substances but if physical adsorption prevails, less heat, of the order of the heat of condensation, is evolved. In general however, the heat of physical adsorption is still measurably greater than that of normal condensation. This is believed to be mainly due to the surface interactions that take place during adsorption, resulting in the wetting of the adsorbent surface and the release of extra heat. The heat of wetting would depend on many interactive factors which include temperature, pressure, vapour phase concentration, adsorbent phase loading, and nature of adsorbate and adsorbent materials. Accurate generalized information on heats of adsorption is therefore very difficult to obtain and empirical deductions are often resorted to.

Working on the adsorption of water vapour from air on activated alumina at a temperature of 25-30°C, Miller and Roberts reported an average value of heat of adsorption of about 1.2 times the normal latent heat of condensation. Working at slightly higher temperatures, Free reported heats of adsorption in the range 2790-3370 KJ/kg (1200-1450 Btu/lb) for the same system. He also suggested that for adsorption of water vapour on silica-gel under comparable conditions the heat of adsorption is in the range 2326-2560 KJ/kg (1000-1100 Btu/lb) implying that the heat effect
associated with adsorption of water on activated alumina is greater than that associated with adsorption on silica-gel. He attributed this to the existence of greater attractive forces for water vapour in activated alumina, than in silica-gel. Free's findings agree with the work of Rathmell and Bateman, in which they proposed the following linear equations for correlating temperature effects:

\[ AT = 3500 A y \text{ for activated alumina.} \]
\[ AT = 2950 A y - 3.5 \text{ for silica-gel.} \]

Clearly, this implies that under comparable conditions the heat evolved by adsorbing moisture on silica-gel would be less than that evolved by adsorbing on activated alumina.

The average value of about 2450 KJ/kg suggested by Free for adsorption on silica-gel at about 40°C, is about 1.03 times the heat of condensation at that temperature. Bearing in mind also the Miller and Roberts suggestion of 1.2 times for activated alumina, it was decided in this study to use a value of heat of adsorption of moisture on silica-gel 1.05 times the heat of condensation at any temperature. Watson's equation (A2.3) was used to evaluate the heat of condensation at the temperature.

A.4 Viscosity

From consideration of elementary kinetic theory of gases in momentum transfer, Reid et al. related the transport property (i.e. viscosity) to the molecular properties of a gas. Thus they derived the equation:

\[ \mu = (\text{constant}) \frac{T^\frac{1}{2}}{\sigma^2} K^\frac{i}{2} \]
For a rigid, non-interacting molecular sphere model the value of the constant was obtained thus:

\[ \mu = 26.69 \frac{\sqrt{MT}}{\sigma^2} \]  

where

- \( \mu \) is viscosity in micropoise
- \( M \) is molecular weight
- \( T \) is temperature in °K
- \( \sigma \) is hard sphere molecular diameter in Å.

However, for most gases, intermolecular forces do exist between molecules. Chapman and Enskog considered this to arise mainly from collision between molecules and obtained a modified form of equation A4.2 thus:

\[ \mu = 26.69 \frac{\sqrt{MT}}{\sigma^2 \Omega} \]  

where \( \Omega \) is the collision integral.

\( \Omega \) will therefore be unity if the molecules do not attract each other and the Chapman-Enskog equation (A4.3) reduces to equation A4.2.

In order to obtain the viscosity of any gas, \( \Omega \) must therefore be known. At low pressures, Neufeld et al proposed an empirical equation which relates \( \Omega V \) to a dimensionless temperature \( T^* \) thus:

\[ \Omega \frac{V}{\Omega_V} = \left( \frac{A}{T^* B} \right) + \left( \frac{E}{\exp(DT^*)} + \left[ \frac{E}{\exp(FT^*)} \right] \right) \]  

A, B, C, D, E, F are constants and \( T^* \) is given by:

\[ T^* = \left( \frac{C}{A} \right) T \]  

The following values were suggested for the constants:

- \( A = 1.16145 \)
- \( B = 0.14874 \)
- \( C = 0.52487 \)
- \( D = 0.77320 \)
- \( E = 2.16178 \)
- \( F = 2.43787 \)
Equation A4.4 was however found to be suitable for only non-polar gases. For polar molecules Stockmayer suggested that the equation could be modified by addition of a polar parameter $\delta$. Then,

$$\Omega V(p) = \Omega V(np) + \frac{0.2 \delta^2}{T^*}$$  \hspace{1cm} A4.5

where $\Omega V(p)$ is collision integral for polar molecule and $\Omega V(np)$ is collision integral for non-polar molecule.

For air, the following values obtain:

$$\sigma = 3.71$$
and $$\frac{\epsilon}{k} = 78.6$$

and for water vapour,

$$\sigma = 2.641$$
$$\frac{\epsilon}{k} = 809.1$$

Also, for water, which is polar, $\delta = 1.0$.

Using these values, viscosities of air and water vapour can then be obtained.

The individual viscosities of air and water vapour were calculated by the above procedure. By considering moist air as a binary mixture of air and water vapour, Wilke's approximation was then used to obtain its viscosity from the individual pure component viscosities.

$$\mu_m = \frac{Y_1 \mu_1}{Y_1 + Y_2 D_{12}} + \frac{Y_2 \mu_2}{Y_2 + Y_1 D_{21}}$$  \hspace{1cm} A4.6
where \( \mu_m \) = viscosity of mixture (moist air)

\( \mu_1, \mu_2 \) = pure component viscosities (air, water vapour)

\( y_1, y_2 \) = mole fractions (air, water vapour)

and

\[
D_{12} = \frac{1 + \left( \frac{\mu_1}{\mu_2} \right)^{\frac{1}{2}} \left( \frac{M_2}{M_1} \right)^{\frac{1}{2}}} \left\{ 8 \left[ 1 + \left( \frac{M_1}{M_2} \right) \right] \right\}^{\frac{1}{4}} \tag{A.6.7}
\]

\[
D_{21} = \frac{D_{12}}{\mu_1} \frac{M_1}{M_2} \tag{A.6.8}
\]

A5. **Diffusivity**

The transport property associated with gaseous diffusion mass transfer, diffusivity, can be related to the molecular properties of a gas in a similar manner as viscosity. Thus, for binary diffusion of gases A and B,

\[
D_{AB} = 1.858 \times 10^{-3} \frac{T^{3/2}}{P} \left[ \frac{(M_A + M_B) / M_A M_B}^2 \right]^{\frac{1}{2}} \tag{A.5.1}
\]

where \( D_{AB} \) = binary diffusion coefficient, cm \(^2\)/s

\( T \) = temperature, °K

\( P \) = pressure, atm

\( \sigma \) = hard sphere molecular diameter, Å

\( \Omega_D \) = diffusion collision integral

and

\[
\sigma_{AB} = \frac{\sigma_A + \sigma_B}{2} \tag{A.5.2}
\]

Defining \( \varepsilon_{AB} \) by,

\[
\varepsilon_{AB} = (\varepsilon_A \varepsilon_B)^{\frac{1}{2}}
\]

and \( T^* \) by

\[
T^* = \left( \frac{k}{\varepsilon_A \varepsilon_B} \right) T \tag{A.5.3}
\]
then, $\Omega_D$ would be expressed as:

$$\Omega_D = \frac{A}{T^*} B + \frac{C}{\exp D T^*} + \frac{E}{\exp F T^*} + \frac{G}{\exp H T^*}$$

where the constants are:

- $A = 1.06036$
- $B = 0.15610$
- $C = 0.19300$
- $D = 0.47635$
- $E = 1.03587$
- $F = 1.52996$
- $G = 1.76474$
- $H = 3.89411$

With the same values of $\sigma$ and $\gamma/k$ used in estimating viscosities of air and water vapour, the binary diffusion coefficient of the air-water vapour system was calculated by this procedure.
APPENDIX B

SAMPLE CALCULATION OF TUMBLER EFFICIENCY.

Data

Condensate flow = 20.5 liters/6 mins.
Steam pressure = 65 psig.
Wt. of wet fabrics = 117 lbs.
Wt. of dry fabrics = 72 lbs.
Ambient wet bulb temp. = 18°C.
Ambient dry bulb temp. = 25°C.
Initial temperature of fabrics = 22°C
Final temperature of fabrics = 40°C.

TABLE B.1: TUMBLER AIR INLET AND OUTLET TEMPERATURES.

<table>
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<tr>
<th>θ (mins)</th>
<th>Tin (°C)</th>
<th>Tout (°C)</th>
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<td>82.5</td>
</tr>
<tr>
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<td>106</td>
<td>85</td>
</tr>
</tbody>
</table>
Calculation Procedure

The air inlet and outlet temperatures $T_{\text{in}}$ and $T_{\text{out}}$ are plotted in Fig. D1 and their mean values over the entire drying period graphically determined. Thus,

\[
\begin{align*}
T_{\text{in}} &= 104^\circ C \\
T_{\text{out}} &= 75^\circ C.
\end{align*}
\]

Moisture Extraction:

Moisture evaporated = 117 - 72 = 45 lbs.
Drying time, $\Theta$ = 17 mins.
Moisture Extraction Rate = \(\frac{45}{17}\) lbs/min.
\[= 1.20 \text{ kgs./min.}\]

Steam Rate:

Condensate flow = 20.5 lits/6 mins.
\[= 3.42 \text{ kgs/min.}\]

Flashing occurs as condensate expands through the valve on the outlet line. Therefore the condensate rate as measured is inaccurate and must be corrected by doing a flash calculation for the prevailing conditions.

\[\begin{align*}
F &\rightarrow \text{Y kg/min} \\
\text{kg/min} &\rightarrow X \text{ kg/min}
\end{align*}\]

Fig. B2 : Scheme for flash calculation
Let \( X \) = flowrate of condensate as measured
\( Y \) = flash steam rate
\( F \) = actual condensate flowrate.

By mass balance,
\[
F = X + Y \quad \text{B.1}
\]
but \( X = 3.42 \)
hence, \( F = 3.42 + Y \) \( \text{B.2} \)

By heat balance,
\[
F H_F = X H_X + Y H_Y \quad \text{B.3}
\]
where \( H \) is enthalpy.

At steam pressure 65 psig,
\( H_F = 272 \text{ Btu/lb.} \)

At atmospheric pressure,
\( H_X = 180 \text{ Btu/lb.} \)
and \( H_Y = 1150.5 \text{ Btu/lb.} \)

therefore,
\[
F(272) = 3.42 (180) + Y(1150.5) \quad \text{B.4}
\]

Solving B2 and B4,
\( F = 3.76 \text{ kgs/min.} = \text{actual steam rate.} \)

Specific Steam Rate:-

Steam rate = 3.76 kg/min.
Moisture removal rate = 1.20 kg/min.
hence, specific steam rate = \( \frac{3.76}{1.20} \)
\( = 3.15 \text{ kg steam/kg moisture.} \)

Thermal Efficiency:-

Latent heat of condensation of steam \( \lambda_s = 2115 \text{ KJ/kg} \)
Latent heat of vaporisation of moisture \( \lambda_m = 2400 \text{ KJ/kg.} \)
Assuming complete transfer of heat on condensation of steam, rate of energy consumption will be:-
\[
q_1 = \lambda_s F
= 3.76 \times 2115
= 7995 \text{ KJ/min}
\]
Energy use for vaporisation of moisture will be:

\[ q_2 = \frac{M}{m} \]

where \( M \) is moisture removal rate

\[ q_2 = 2400 \times 1.2 \]
\[ = 2880 \text{ KJ/min} \]

Therefore the thermal efficiency will be given by

\[ \eta = \frac{q_2}{q_1} \times 100 \]
\[ = 36\% \]

Air Flowrate:-

Dry bulb temperature, \( T_{db} = 25^\circ C \)

Wet bulb temperature \( T_{wb} = 18^\circ C \)

For these conditions, air inlet humidity \( y_{in} = 0.0120 \)

Now, let air flowrate \( = \dot{m} \), kgs/min

- air specific heat = \( c_p \)
- water vapour specific heat = \( c_v \)

Air humid heat will be given by

\[ C_h = c_p \dot{m} + c_v \dot{m} y \]

\[ h = \frac{c_p \dot{m} + c_v \dot{m} y}{\dot{m}} \]

by heat balance on steam batteries,

\[ G \left( T_{in} - T_{db} \right) = q_1 \]

\[ G = \frac{q_1}{C_h (T_{in} - T_{db})} \]

\[ = \frac{7995}{1.02 \times 79} \]
\[ = 95 \text{ kgs/min} \]
(iv) Loss Through Dryer Walls.

This is given approximately by the difference between total energy consumption and the sum of the inputs into other components. Hence, loss through walls will be:

\[
100 - (36 + 59.5 + 1.2)
\]
\[
= 3.3\%
\]

Ideal Efficiency:

The ideal thermal efficiency of the dryer will be given as:

\[
\eta_{\text{ideal}} = \frac{\text{heat used for evaporation}}{\text{heat used for evaporation} + \text{sensible heat lost in exit air}} \times 100
\]

\[
\eta_{\text{ideal}} = \frac{36}{36 + 59.5} \times 100
\]

\[
= 37.7\%
\]

Alternatively, this can be obtained by the concepts of equations 2.48 and 2.54 in section 2.5.

Mean inlet air temperature = 104°C
Mean outlet air temperature = 75°C

hence,

\[
\eta_{\text{ideal}} = \frac{104-75}{104-25}
\]

\[
= 36.7\%
\]
Blower Performance (Typical figures)

Exhauster Performance (Typical figures)

Note: Air Flow figures given are representative referred to intake conditions at normal temperature and pressure.
CALIBRATION CHART FOR AIR.
METRIC X SERIES TUBE SIZE 65X
DURALUMIN FLOAT TYPE A
SIZE 65
\[ d = 65.00 \quad \omega = 145.57 \text{ gm} \]

FLOW: LITRES/MINUTE AIR AT 15°C AND 760 mm. Hg. Abs.
APPENDIX D

SAMPLE CALCULATION OF EXPERIMENTAL RESULTS

Experimental Data (Run 45)

Air flowrate = 1.6 m$^3$/min.
Height of bed = 0.75 m.
Dry bulb ambient temperature = 180°C
Wet bulb ambient temperature = 11.5°C.

TABLE D1 : RAW DATA (EXPERIMENT 45)

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<th>$T_{iw}$ (°C)</th>
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<th>$T_4$ (°C)</th>
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<tr>
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<td>32.5</td>
<td>27</td>
<td>44</td>
<td>57.5</td>
<td>66.5</td>
<td>58.5</td>
<td>32.5</td>
</tr>
<tr>
<td>90</td>
<td>32.5</td>
<td>27</td>
<td>40</td>
<td>54.5</td>
<td>65</td>
<td>53</td>
<td>31</td>
</tr>
</tbody>
</table>
$T_3 = 0.375 \text{ m from inlet}$

$T_4 = 0.375 \text{ m from inlet}$

$T_5 = 0.625 \text{ m from inlet}$
**Calculation**

The bed temperature profiles are plotted in fig. D1, while the air inlet and outlet temperatures and humidities are plotted in fig. D2.

From fig. D2, the average air inlet temperature and humidity $T_i$ and $y_i$ are:

$$T_i = 32°C$$
$$y_i = 0.020 \text{ kg/kg}$$

Also from fig D2, the average air exit temperature $T_e$ is:

$$T_e = 75°C$$

Breakpoint was obtained for outlet humidity $y_e = 0.0010 \text{ kg/kg}$

hence,

$$\Delta T = T_e - T_i = 43°C$$
$$\Delta y = y_i - y_e = 0.0190 \text{ kg/kg}$$

Breakpoint time $Q_b = 35 \text{ mins.}$

Air flowrate $G = 1.6 m^3/\text{min} = 2 \text{ kgs/min.}$

Hence, amount of moisture extracted up to breakpoint will then be:

$$M_b = GQ_b \Delta y$$
$$= 1.33 \text{ kgs.}$$

Mass of adsorbent $M_s = \frac{720 \times \Pi \times (0.235)^2 \times 0.75}{4}$
$$= 23.4 \text{ kg.}$$

Thus adsorptive capacity of bed at breakpoint will be:

$$W_b = \frac{M_b}{M_s}$$
$$= 0.0568 \text{ kg/kg}$$
$$= 5.68\%.$$
The partial pressure of moisture in inlet air is expressed as:

\[ p_i = \frac{29 y_i}{18 + 29 y_i} \]

Now, \( \Pi = 760 \text{ mm Hg} \)

hence, \( p_i = 27.16 \text{ mm Hg} \)

By Antoine equation, saturation vapour pressure of water at the average bed temperature (or air exit temperature) will be:

\[ p_{ob} = \exp (18.4 - \frac{3883}{T_e + 230}) \]

\[ = 289.70 \]

Bed relative humidity at breakpoint \( X_b \) will therefore be:

\[ X_b = \frac{p_i}{p_{ob}} \]

\[ = 0.0819 \]

\[ = 8.19\% \]
APPENDIX 2

PREDICTION PROGRAMME - BRH MODEL

The computer algorithm which was set up for predicting the overall performance of the TSC heat pump dryer is outlined below:

1. Start

   Input data

   Set exhaust humidity $y_{Re}$

   Calculate dryer conditions; saturation temperature and humidity $T_s$, $y_s$; inlet humidity $y_{Di}$, outlet temperature and humidity $T_{De}$, $y_{De}$, recirculation ratio $r_D$.

   Call Dryer Design

   Calculate adsorption conditions: bed relative humidity $X_b$, breakpoint capacity $W_b$, outlet temperature $T_{Re}$, mass of adsorbent $M_s$.

   Call Adsorption-BRH

   Calculate desorption conditions: equilibrium capacity $W_R$, exhaust temperature $T_{Re}$.

   Call Desorption
APPENDIX E cont'd.

Call Evaluation

evaluate performance-efficiencies

print results

Stop
SUBROUTINE DRYER DESIGN

Call Dryer Design

Calculate \( T_{s1} \) from Antoine equilibrium

\( \text{Calculate } y_{Di} \text{ by overall material balance.} \)

\( \text{Calculate } T_{s2} \text{ from dryer air saturation equation.} \)

\( \text{if} \quad \left| T_{s1} - T_{s2} \right| < \text{error} \)

\( \text{Yes} \)

\( \text{Calculate } T_{De} \text{ and } y_{De} \text{ from dryer air saturation equation.} \)

\( \text{Calculate } r_{D} \text{ by moisture balance.} \)

Return.

Set \( y_s \)
SUBROUTINE ADSORPTION - BRH

Call Adsorption-BRh

Calculate $X_b$

Calculate $W_{b1}$ from BRH equation.

Calculate $W_{b2}$ from heat balance equation.

if $|W_{b1} - W_{b2}| < \text{error}$

No

Yes

Calculate $K_s$

Return
SUBROUTINE DESORPTION

Call Desorption

Calculate $T_{Re}$ from heat and mass balance.

Calculate $W_R$ from equilibrium relations.

Return
SUBROUTINE EVALUATION

Call Evaluation

Subroutine calculates the cycle thermal efficiency, heat pump maximum and actual COPs and thermodynamic efficiency.

Return
5LIST MARGE5549.2
10 REM PREDICTION OF PERFORMANCE OF TSD CARRIER BY ENH MODEL.
15 REM FOR EXPLANATION OF NOTATIONS SEE SUBROUTINE 1340
17 C000 4500
18 REM BASIS IS 1 KG/HF AIR WITH HALF CYCLE TIME 2 HPS
20 LET T1=250
30 LET F=0.4
40 LET Y4=0.962
50 LET T6=260
60 LET D2=10
70 PRINT
80 PRINT 'Y1', 'Y4', 'Y6', 'D2'
90 PRINT T1, F
100 PRINT
110 READ T6, 50
120 DATA 1.0, 0.5
130 READ C1, C2, C3, C4
140 READ P0, 18, 05, 4, 18
150 READ L9, T8, K9
160 DATA 225, 0.5, 0.973
170 DEF FNK(T)=P/(P0-P)*18/29
180 DEF FNP(T)=EXP(18*4-(1883/(T+23)))
190 DEF FNY(Y)=EXP(18*(4-3933/(T0+23)))
200 DEF FNY(P)=P/23.58
210 P=P0
220 Y0=FNY(P)
230 Y=Y0
240 K0=FNK(T)
250 T=T0
260 LET Y=0
270 LET Y2=0
280 LET T1=21
290 LET T6=0.0255+21
300 GCDUP 2000
310 PRINT 'Y1', 'Y2', 'Y5', 'Y5'
320 PRINT Y1, Y2, Y5, Y5
330 PRINT Y1, Y2, Y5
340 T5=T2
350 Y3=Y2
660 GOSUB 2000
610 PRINT ‵′T1′′, ‵′T4′′, ‵′W4′′, ‵′X4′′
620 PRINT T1, T4, W4, X4
630 PRINT
650 GOSUB 4000
655 PRINT
657 PRINT
660 J5=100*ABS(W4-W7)/W4
670 IF J5<2 GOTO 700
680 Z1=Z1+0.0005
690 GOTO 470
700 PRINT ‵′Y7′′, ‵′Y7′′, ‵′W7′′
710 PRINT Y7, T7, W7
720 PRINT
730 GOSUB 5000
800 PRINT ‵′C1′′, ‵′C6′′, ‵′N3′′
810 PRINT ‵′C5′′, ‵′C6′′, ‵′N3′′
820 PRINT P1, N1
840 PRINT C5, C6, N3
850 PRINT
900 END
1500 REM SUBROUTINE NOTATIONS
1502 REM F IS FRACTIONAL SATN
1504 REM T IS AIR TEMP
1506 REM Y IS AIR HUMIDITY
1508 REM P IS PARTIAL PRESS2 OF MOISTURE IN AIR
1510 REM K IS TEMP IN KELVIN
1512 REM X IS AIR REL HUM
1514 REM W IS ADSORBENT CAPACITY
1516 REM R IS SATN YAP PRESS
1518 REM THE FOLLOWING SUBSCRIPT NOTATIONS APPLY TO T, V, F, K, Y
1520 REM 0 IS AMBIENT CONDTNS
1522 REM 1 IS DRYER INLET
1524 REM 2 IS DRYER EXIT
1526 REM 3 IS ADSORBER INLET
1528 REM 4 IS ADSORBER EXIT
1530 REM 5 IS SATN CONDTN IN DRYER
1532 REM 6 IS REGTN INLET
1534 REM 7 IS REGTN EXIT
1540 REM C1 IS AIR SP HEAT
1542 REM C2 IS WATER VAPOR SP HEAT
1544 REM C3 IS SILICA GEL SP HEAT
1546 REM C4 IS WATER SP HEAT
1548 REM H IS HUMID HEAT OF AIR
1550 REM S0 IS AMBIENT SATN
1552 REM R1 IS RECYCLE RATIO
1554 REM Z Isl ARE AN ITERATION VARIABLE
1556 REM J Isl ARE AN ITERATION DUMMY FUNCTION
1558 REM C5 IS MAX CYCLE COP
1560 REM C6 IS ACTUAL CYCLE COP
1562 REM N2 IS THERMODYNAMIC EFFCY
1564 REM N1 IS THERMAL EFFCY
1566 REM M IS MASS OF SILICA GEL BED
1568 REM D2 IS INITIAL TEMP OF ADSORPTION BED
1570 REM D6 IS INITIAL TEMP OF DESORPTION BED
1572 REM E Isl ARE ENERGY TERMS IN EFFCY CALCULATION
1574 REM L IS LATENT HEAT
1576 REM K9 IS CRITICAL TEMP OF WATER
1578 REM T8 IS BPT TEMP OF WATER
1580 REM L8 IS LATENT HEAT AT T8
1582 REM P9 IS YAP PRESS AT AMBIENT
1585 RETURN
2000 REM SUBROUTINE DRYER  
2010 LET Z2=0  
2020 Y5=Y0+Z2  
2030 Y1=(F*Y5*(Y4-Y0)-Y4*(Y7-Y0))/(F*(Y4-Y0)-(Y7-Y0))  
2035 Y=Y1  
2036 H1=FNH(Y)  
2040 Y=Y5  
2050 P5=FNK(Y)  
2070 J1=(3883/(12.4-LOG(P5)))-220  
2080 K=J1  
2090 L5=FNK(K)  
2100 J2=(L0*H1+L5*(C2*70-L0)*(Y5-Y1))/(L0*H1+L5*(C2]*(Y5-Y1))  
2110 IF J1<0 GOTO 2150  
2120 IF J2<0 GOTO 2150  
2130 J0=100*ABS(J1-J2)/J1  
2140 IF J0<2 GOTO 2170  
2150 Z2=Z2+0.605  
2160 GOTO 2020  
2170 T5=J2  
2180 Y2=(1-F)*Y1+F*Y5  
2190 T2=(L0*H1+L5*(C2*70-L0)*(Y2-Y1))/(L0*H1+L5*(C2]*(Y2-Y1))  
2200 R2=(Y1-Y0)/(Y4-Y0)  
2250 RETURN  
3000 REM SUBROUTINE ADSORBER  
3010 LET Z3=2  
3020 T4=60+Z3  
3030 Y=Y3  
3040 P3=FNK(Y)  
3050 X4=EXP(18.4-3883/(230+T4))  
3070 X4=100*P3/P4  
3080 J3=1.07*(X4-.01)  
3090 T=T4  
3100 K4=FNK(T)  
3110 K=K4  
3120 L4=FNK(K)  
3130 J4=(100*C3*(Y3-Y4)*(T4-D3))/(1.03*L4*(Y3-Y4)-C1*(T4-T3))  
3140 IF J3<0 GOTO 3180  
3150 IF J4<0 GOTO 3180  
3160 J9=100*ABS(J3-J4)/J3  
3170 IF J9<2 GOTO 3200  
3180 Z3=Z3+0.5  
3190 GOTO 3020  
3200 W4=J4  
3210 W=100*2*(Y3-Y4)/W4  
3300 RETURN  
4000 REM SUBROUTINE DESORBER  
4010 D6=T4  
4020 K4=T4+273  
4045 K=K4  
4050 L4=FNK(K)  
4050 T7=(2*1-F1)+31+76*M*(C2*D6-1.05*L4+W4/200))/10*(C6+2+1-F1  
4054 Y=Y7  
4055 P7=FNK(Y)  
4060 P7=EXP(18.4-3883/(T7+230))  
4070 K7=T7+273  
4090 X7=P7/P7*(1.8*K7/500)  
4090 W7=100*(X2*X7)/(1+12.33*X7)+.16*15*X7*2W/1+0.7*X7  
4200 RETURN
REM SUBROUTINE EVALUATION
5010  Y=Y1
5020  H1=FINH(4)
5030  Y=Y4
5040  H4=FINH(4)
5050  E1=H1*(T1-T2)
5060  E2=H1*(T1-T4)
5070  E3=H3*(1-P1)*(T4-T0)
5080  E4=H4*(1-P1)*(T6-T4)
5090  E5=LO/(LO+2*(T7-T0))
5100  N1=100*E1*E5/(E2+E3+E4)
5110  E6=H6*(1-P2)*(T2-T0)
5120  E7=LO/(LO+2*(T2-T0))
5130  T=T2
5140  K3=FKK(T)
5150  T=T4
5160  K4=FKK(T)
5170  T=T6
5180  K5=FKK(T)
5190  C5=(K4/(K4+K3))*((K6-K5)/K5)
5200  C6=(T-T3)/((1-P2)*(T-T2))
5210  N2=100*C6/C5
5220  E2=UPW
OK.