

# 1

## Fundamentals of Energy Analysis of Dryers

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### 1.1

#### Introduction

Drying is a highly energy-intensive process, accounting for 10–20% of total industrial energy use in most developed countries. The main reason for this is the need to supply the latent heat of evaporation to remove the water or other solvent. There are thus clear incentives to reduce energy use in drying: to conserve finite resources of fossil fuels, to reduce carbon footprint and combat climate change, and to improve process economics, but it is a challenging task facing real thermodynamic barriers.

Effective analysis of current energy use is a vital first step in identifying opportunities for savings. An initial lower bound of dryer energy needs is provided by calculating the evaporation load for the amount of water to be removed (Section 1.3). This shows how much energy is inherently required and, by comparing with current measured energy usage, what opportunities exist to reduce energy consumption. These fall into three main categories;

- 1) Reduce the evaporation load – for example, by upstream dewatering to reduce initial moisture content, or avoiding overdrying.
- 2) Increase the dryer efficiency – for example, by improving insulation and reducing heat losses, installing heat recovery or changing operating parameters.
- 3) Improve the energy supply (utility) system – for example, by increasing boiler efficiency, or using combined heat and power (CHP), heat pumps, waste incineration, or other alternative low-cost fuels.

Frequently, the evaporation load will be less than 50% of the actual process energy consumption in terms of fuel supplied. The numerous causes for this difference include:

- Additional energy required to break bonds and release bound moisture
- Heat losses in the exhaust (particularly for convective dryers) or through the dryer body
- Heating solids and vapor to their discharge temperature
- Steam generation and distribution losses and condensate losses

- Losses in non-routine operation, for example, startup, shutdown or low load periods
- Ancillary steam use, for example, trace heating, steam ejectors and turbine drives

In addition, there will be power consumption for fans, vacuum pumps, chillers, mechanical drives and other general uses. The dryer's energy use must also be seen in the context of the complete process and, indeed, of the site as a whole.

One key tool is pinch analysis (Section 1.4.2), which shows the temperatures at which the dryer heat load is required and where heat can be recovered from the exhaust vapor, and places this in the context of the overall production process. This shows the feasibility of heat recovery, CHP, heat pumps and process changes, and helps to generate realistic targets for how much energy the process should be using. Practical methods to help achieve these targets are reviewed in Section 1.5, and the whole analysis methodology is shown in action in the case study in Section 1.6.

An engineer – maybe a reformed gambler – once restated the three laws of thermodynamics (conservation of energy, increasing entropy and increasing difficulty of approaching absolute zero) as follows:

- 1) You can't win – you can only break even.
- 2) You can't break even – you can only lose.
- 3) You can't get out of the game.

Energy consumption, like taxes, is an unavoidable fact of life. Nevertheless, it is sensible, and feasible, to use our ingenuity to reduce it as far as possible.

## 1.2 Energy in Industrial Drying

Industrial dryers are major energy users. A survey by Wilmshurst (1988) (reported by Bahu, 1991) estimated that drying processes accounted for at least 10% of industrial energy demand in the UK and Europe – not just 10% of process engineering, but of all industrial consumption. Since then, if anything, the figure has increased; a similar survey by Kemp (1996) for the UK Government's Department of Energy evaluated the figure at 12–15% of total industry energy use. Similar figures are thought to apply for most developed countries.

Why should this be, in an era of increasing focus on energy efficiency and work by manufacturers to improve their equipment? The answer is that drying processes have an unavoidable constraint – they must supply enough heat or energy to provide the latent heat of evaporation for all the vapor which is removed – over  $2000 \text{ kJ kg}^{-1}$  for the most common solvent, water. All industries have been working to reduce their energy consumption, but many have been able to make energy savings more easily than drying, which is limited by its thermodynamic barrier. Furthermore, dryers tend to have inherently low thermal efficiency (often below 50% for convective dryers) and many new products requiring drying have appeared on the market (e.g., special foods, pharmaceuticals, videotapes).

In the 1970s and 1980s, energy prices were high, and this provided the major cost incentive for installing energy-saving projects – an incentive that was markedly reduced when oil prices fell to much lower levels in the following years. Now, in the twenty-first century, energy is recognized to be only part of the bigger picture of sustainability. Major associated benefits of energy reduction include reducing CO<sub>2</sub> and other greenhouse gases, and pollutants and acid gases including SO<sub>x</sub> and NO<sub>x</sub>. With oil prices volatile again, and the principles of dryer energy reduction better understood than in the past, it is an excellent time to revisit the challenge of making drying systems more energy efficient. This should involve the entire process, including the energy supply systems, rather than treating the dryer in isolation.

An economic point which is often overlooked is that energy is a direct cost, so that a saving of £1000 (GBP) goes directly onto the bottom line and appears as £1000 extra profit. In contrast, a £1000 increase in sales is diluted by a corresponding increase in production costs, including raw materials, transport and, of course, energy itself. Nevertheless, the tight constraints on budgeting and economic return on energy-saving schemes make it essential that a clear analysis of the principles is made before embarking on a project.

### 1.3 Fundamentals of Dryer Energy Usage

#### 1.3.1 Evaporation Load

We will use the common definition of a drying process as being one where liquid is removed from a solid specifically by evaporation. This excludes mechanical dewatering processes such as filtration and centrifugation. Hence, to achieve drying, the latent heat of evaporation must be supplied to turn each kilogram of moisture into vapor. Thus the absolute minimum amount of heat or other energy,  $E_{v,\min}$  (J), which must be supplied for a drying process is:

$$E_{v,\min} = M_v \Delta H_v \quad (1.1)$$

It is often more convenient to use the corresponding heat supply rate,  $Q_{v,\min}$  (J s<sup>-1</sup> or W), which is given by:

$$Q_{v,\min} = W_v \Delta H_v \quad (1.2)$$

For a continuous process it is

$$Q_{v,\min} = W_s (X_{in} - X_{out}) \Delta H_v \quad (1.3)$$

and for a batch process (at any instant)

$$Q_{v,\min} = M_s \left( \frac{-dX}{dt} \right) \Delta H_v \quad (1.4)$$

Latent heat varies with temperature. For the most common solvent, water, the latent heat of evaporation is  $2501 \text{ kJ kg}^{-1}$  at  $0^\circ\text{C}$  and  $2256 \text{ kJ kg}^{-1}$  at  $100^\circ\text{C}$ . At ambient temperatures, around  $20^\circ\text{C}$ , a figure of  $2400 \text{ kJ kg}^{-1}$  is a good working approximation. So, for a drying process which requires the evaporation of  $1 \text{ kg s}^{-1}$  of water from the solid, an absolute minimum of  $2400 \text{ kJ s}^{-1}$  ( $2400 \text{ kW}$ ) must be supplied to the process in some way. Note, however, that if the liquid enters with the solid at one temperature, and emerges as a vapor at a higher temperature, additional sensible heat will be needed to achieve this, in addition to the latent heat at a fixed temperature.

### 1.3.2

#### Dryer Energy Supply

The evaporation load is the minimum energy demand for drying, but this energy has to be transferred to the solids in a practical way; for example, from hot air (convective drying), a hot wall or surface (contact or conduction drying), or by absorbing electromagnetic radiation (infrared, radiofrequency or microwave drying). The process of supplying heat typically consumes significantly more energy than the latent heat of evaporation.

For a continuous convective (hot air) dryer, the heater duty for the inlet air heat exchanger (excluding heater losses) is given by:

$$Q_{\text{heater}} = W_g c_{p_g} (T_{g,\text{in}} - T_{g,\text{a}}) \quad (1.5)$$

Here  $T_{g,\text{in}}$  is the inlet temperature to the dryer and  $T_{g,\text{a}}$  is the temperature at which the air is supplied. Conversely, when the hot air is supplied to the dryer, the exhaust emerges at a mean temperature of  $T_{g,\text{out}}$ . A simple heat balance on a continuous dryer (as developed for debottlenecking by Kemp and Gardiner, 2001) gives:

$$W_g c_{p_g} (T_{g,\text{in}} - T_{g,\text{out}}) \approx W_s (X_{\text{in}} - X_{\text{out}}) \Delta H_v + W_s c_{p_s} (T_{g,\text{out}} - T_{g,\text{in}}) + Q_{\text{loss}} \quad (1.6)$$

that is, heat given up by hot air  $\sim$  evaporation load + sensible heating of solids + heat losses.

Combining Eqs. 1.5 and 1.6 we find that, to a first approximation:

$$Q_{\text{heater}} = \frac{(T_{g,\text{in}} - T_{g,\text{a}})}{(T_{g,\text{in}} - T_{g,\text{out}})} [W_s (X_{\text{in}} - X_{\text{out}}) \Delta H_v + Q_{s,\text{sens}} + Q_{\text{loss}}] \quad (1.7)$$

This would be the heat energy required to run a perfect adiabatic dryer, and also the amount of fuel needed if the heat was supplied by a perfect energy conversion system with zero losses. We see that, compared to the basic evaporative load, there are additional terms for heat losses in the exhaust gas, sensible heating of the solids and heat losses from the dryer body.

In some cases, filter cakes can be dewatered by blowing ambient air through them, so that evaporative cooling occurs and the damp exhaust air emerges below ambient. No heat is then required, but there is a significant pressure drop across the cake, so

power is needed to run the fans. Moreover, the process is very slow – typically taking many hours or days – because the driving forces are so low.

Likewise, if heat is supplied by conduction, there is no need for a large air flow to transmit heat, and the heat requirement with no carrier gas would fall to:

$$Q_{\text{heater}} = [W_s(X_{\text{in}} - X_{\text{out}})\Delta H_v + Q_{s,\text{sens}} + Q_{\text{loss}}] \quad (1.8)$$

However, a partial pressure or humidity driving force is needed to carry the vapor away from the solids, otherwise the local air becomes saturated with vapor and drying rates fall towards zero. A flow of carrier gas is required, or a vacuum must be pulled. This requires additional electrical power for fans or pumps, or steam for ejectors.

Finally, if a system incorporates water adsorption by zeolites, as described in Chapter 5, this reduces the evaporation load. However, energy will then be needed to regenerate the zeolite to its dry state for reuse (or to manufacture new zeolite if it cannot be recycled).

Hence, in practice, the actual energy which must be supplied is normally considerably greater than the evaporation load calculated in Section 1.3.1. The various additional energy penalties can be broken down into the following categories:

- 1) Thermal inefficiencies in the dryer: exhaust heat content in convective dryers, sensible heating of solids, heat losses from dryer body.
- 2) Thermal inefficiencies in the utility (heat supply) system: steam generation efficiency, steam leaks and mains losses.
- 3) Additional energy demands: power for solids transport, vacuum pumps and air fans.

These will now be illustrated by a detailed practical example.

### 1.3.3

#### Evaluation of Energy Inefficiencies and Losses: Example

Assume a continuous process with a flowrate  $1 \text{ kg s}^{-1}$  of dry solid, being dried from 12 to 2% moisture (dry basis) so that  $\Delta X = 0.1 \text{ kg kg}^{-1}$  and the evaporation rate  $W_v = 0.1 \text{ kg s}^{-1}$ . Hence, from Eq. 1.2,  $Q_{v,\text{min}} = 240 \text{ kW}$ .

A psychrometric chart provides a convenient and rapid way to estimate enthalpies and outlet conditions. Either a Grosvenor (temperature–humidity) or Mollier (enthalpy–humidity) chart can be used. It allows for additional factors, such as the extra heat required to heat water vapor from  $20^\circ\text{C}$  to exhaust temperature. By reading off the exhaust humidity, the required airflow can be calculated using a mass balance on the solvent; alternatively, if dryer airflow is known, the maximum evaporation rate can be found. Assuming negligible leaks, the mass balance is:

$$W_g(Y_{\text{out}} - Y_{\text{in}}) = W_v = W_s(X_{\text{in}} - X_{\text{out}}) \quad (1.9)$$

For convective dryers, the heat is supplied by hot air. Assume the inlet air is at  $150^\circ\text{C}$ , which is heated by steam from ambient ( $20^\circ\text{C}$ ), and the ambient humidity is  $7.5 \text{ g kg}^{-1}$  ( $0.0075 \text{ kg kg}^{-1}$ , corresponding to a dewpoint of  $10^\circ\text{C}$ ). Either by

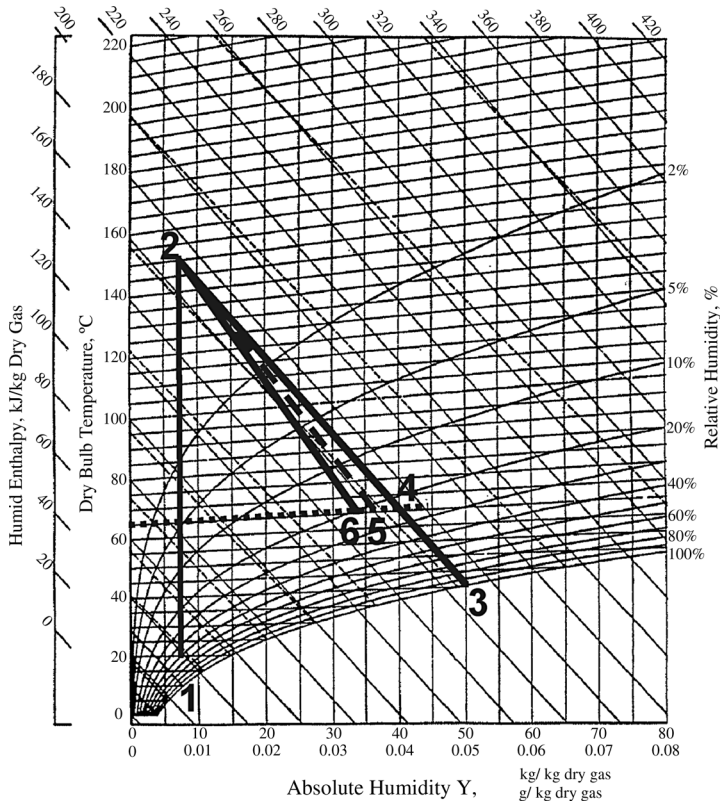


Fig. 1.1 Mollier psychrometric chart with operating lines for an indirect-heated convective dryer.

calculation or by reading from a psychrometric chart (see Fig. 1.1), the enthalpy is approximately  $40 \text{ kJ kg}^{-1}$  for the ambient air (point 1 on the chart) and  $170 \text{ kJ kg}^{-1}$  for the inlet air (point 2), so that  $130 \text{ kJ kg}^{-1}$  must be supplied in the air heater. For illustrative purposes, these and other enthalpy figures will be rounded throughout the following calculations. Also, by substituting the appropriate numbers into Eq. 1.9, we obtain the useful relationship  $W_g = 0.1/(Y_{\text{out}} - 0.0075)$ .

### 1.3.3.1 Dryer Thermal Inefficiencies

**Exhaust Heat Losses** The absolute minimum exhaust temperature is the adiabatic saturation temperature  $T_{\text{as}}$ . For inlet air at  $150^\circ\text{C}$  and humidity  $7.5 \text{ g kg}^{-1}$ ,  $T_{\text{as}}$  is approximately  $40^\circ\text{C}$  (point 3), reading along the adiabatic saturation line on the psychrometric chart (or, as a good approximation at low to moderate humidity, using lines of constant enthalpy which are roughly parallel to adiabatic saturation lines). Hence there is about  $40 \text{ kJ kg}^{-1}$  of sensible heat in the exhaust compared to  $20 \text{ kJ kg}^{-1}$  in the ambient air, and of the  $130 \text{ kJ kg}^{-1}$  supplied to heat the air, at most  $110 \text{ kJ kg}^{-1}$  could be used for evaporation. This can be roughly confirmed from the adiabatic

saturation humidity which is about  $50.5 \text{ g kg}^{-1}$ ; hence,  $43 \text{ g kg}^{-1}$  has been evaporated and, with a latent heat of about  $2400 \text{ kJ kg}^{-1}$ , requires  $103 \text{ kJ kg}^{-1}$ ; the additional heat above this is needed to raise the water vapor to exhaust temperature.

Thus if dryer efficiency  $\eta$  is expressed as latent heat of evaporation divided by actual heat supplied to the air, this is  $(103/130)$  or approximately 80%, so that 20% is inherently lost in the exhaust. This efficiency will vary with inlet air temperature, falling further for lower inlet temperature.

Likewise, from Eq. 1.9,  $W_g = (0.1/0.0043) = 2.33 \text{ kg s}^{-1}$ , and from Eq. 1.5,  $Q_{\text{heater}} = 2.33 \times 1.0 \times (150 - 20) = 302.3 \text{ kW}$ , so that  $\eta = 240/302.3 = 79.4\%$ , as above.

**Exhaust Air Temperature Above Dewpoint** The ideal efficiency above can only be achieved for a dryer where the exhaust gas reaches equilibrium with the solids. This implies extremely good heat transfer because the driving forces for heat and mass transfer will be very low. It can be closely approached in a batch fluidized bed during constant rate drying, as heat and mass transfer in the bed is excellent and the number of transfer units is high (typically 5–10). For most dryers, however, to reduce dryer size and avoid condensation, exhaust temperature will be significantly above its dewpoint. Taking a typical value of  $25^\circ\text{C}$ , the exhaust temperature is now  $65^\circ\text{C}$  (point 4), the sensible heat component is about  $65 \text{ kJ kg}^{-1}$ , the exhaust humidity is  $40.5 \text{ g kg}^{-1}$ , and this corresponds to a latent heat component of  $(33/1000 \times 2400)$  or  $79 \text{ kJ kg}^{-1}$ . Dryer efficiency has now fallen to just over 60%.

To achieve the dryer duty with the lower exhaust humidity, airflow  $W_g$  must rise to  $3.03 \text{ kg s}^{-1}$ , from Eq. 1.9, and heater duty to  $394 \text{ kW}$ , from Eq. 1.5, hence the lower efficiency.

**Heating of Solids** If the solids enter at ambient ( $20^\circ\text{C}$ ) but are heated to  $50^\circ\text{C}$  during the falling-rate section of the drying process to remove bound moisture, and the specific heat capacity of the solids is  $1 \text{ kJ kg}^{-1} \text{ K}^{-1}$ , the sensible heat supplied is  $W_s c_{ps} \Delta T = 1 \times 1 \times 30 = 30 \text{ kW}$ . The total  $Q_v$  has risen to  $270 \text{ kW}$ , just over 10% above the base value. Conversely, of the  $79 \text{ kJ kg}^{-1}$  calculated above, about 10% is used to heat solids rather than for evaporation, so the evaporation load falls to  $70 \text{ kJ kg}^{-1}$  and the efficiency to 54%. The result of this is that the outlet humidity falls to approximately  $37 \text{ g kg}^{-1}$  for the same exhaust temperature (point 5), and the dryer operating line is no longer parallel to an adiabatic saturation line or a line of constant enthalpy.

Using Eq. 1.9 again,  $W_g = 3.39 \text{ kg s}^{-1}$ , and from Eq. 1.5,  $Q_{\text{heater}} = 441 \text{ kW}$ .

**Heat of Wetting of the Solids** When removing bound moisture, extra heat will be required to break the bonds between the water and the substrate. This appears as an increase above the normal latent heat of evaporation, becoming greater as the moisture content falls. This is material-specific and is not included in these calculations, but should be borne in mind. It can sometimes be rolled in with heat losses (below).

**Heat Losses from the Dryer Body** These are typically 5–10%, but can be greater if the dryer is poorly insulated or of small capacity (higher surface-to-volume ratio). As well as convection and radiation from the outer surfaces, heat conduction along the

Tab. 1.1 Outlet conditions, airflow and heater duty for different scenarios.

Condition	$T_{g,out}$ (°C)	$Y_{out}$ (kg kg <sup>-1</sup> )	$\Delta H_{latent}$ (kJ kg <sup>-1</sup> )	$W_g$ (kg s <sup>-1</sup> )	$Q_{heater}$ (kW)	$\eta$ (%)
Adiabatic saturation	40	0.0505	103.2	2.33	302	79.4%
Exhaust approach 25 °C	65	0.0405	79.2	3.03	394	60.9%
Including solids heating	65	0.037	70.8	3.39	441	54.5%
Including heat losses	65	0.034	63.6	3.77	491	48.9%

supporting framework must also be taken into account. Losses may be expressed in various ways – an absolute value, a percentage of inlet air enthalpy or a percentage of evaporation load. Assuming for this case a loss of 10% of evaporation, the evaporation load has now fallen to 63 kJ kg<sup>-1</sup> and efficiency to 49%. A further reduction in outlet humidity also takes place, to 34 g kg<sup>-1</sup> assuming the same exhaust temperature of 65 °C is maintained (point 6 on the psychrometric chart). Hence  $W_g = 3.77 \text{ kg s}^{-1}$  and  $Q_{heater} = 491 \text{ kW}$ .

Overall, we see that a typical convective dryer, even if well-designed and well-operated, can be less than 50% efficient. Table 1.1 summarizes the outlet conditions, air flow requirement and heater duty for the different situations. The various operating lines are shown on the Mollier psychrometric chart in Fig. 1.1.

### 1.3.3.2 Inefficiencies in the Utility (Heat Supply) System

**Boiler Efficiency** Assuming the dryer is heated by steam, this must be raised in a boiler where fuel is burned in an air stream. Significant heat is lost in the exiting flue gases. Modern boilers will recover as much heat as possible from the flue gas, for example, by economizers which heat boiler feedwater and the incoming air. However, even then, the flue gases will leave at 120–150 °C and the maximum boiler efficiency will be about 80–85%. Boiler efficiency is usually stated on the air side, as the ratio between heat passed to the process fluid and the heat released from combustion of the fuel. Lower flue gas temperatures would give condensation in the exhaust gas, which can lead to stack corrosion due to acid gases, even with relatively clean fuels such as natural gas.

**Boiler Feedwater Heating** The steam will be condensed in the process to provide heat to the dryer and will release its latent heat. For example, 10 bara steam (10 bar absolute pressure, 9 bar gage) condenses at 180 °C with a latent heat of 2015 kJ kg<sup>-1</sup>. Hot condensate emerges. If this is returned at the same temperature and pressure to the boilers to act as feedwater, with no leaks or temperature losses, the same amount of heat (2015 kJ kg<sup>-1</sup>) can be supplied in the boiler to produce steam. However, in practice, this never happens. Some water is blown down to avoid build-up of salts, and must be made up with cold water which requires additional heating. Some plants do not have a pressurized condensate return system, which limits the boiler feedwater return temperature to about 85–90 °C before cavitation (boiling) occurs. Others



**Tab. 1.2** Heat required to raise steam from boiler feedwater at different temperatures.

Fluid	Temperature (°C)	Enthalpy (heat content) (kJ kg <sup>-1</sup> )	Heat required to raise 10 bar steam (kJ kg <sup>-1</sup> )	Fuel for boiler at 80% efficiency (kJ kg <sup>-1</sup> )
10 bara steam	180	2778	0	0
10 bara water	180	763	2015	2519
Water at 90 °C	90	377	2401	3001
Water at ambient	20	84	2694	3368

return only part of their condensate, or none at all. If all boiler feedwater is raised from ambient (20 °C) to 180 °C, this gives an additional 679 kW heat requirement, and a loss of 25% (because 2694 kJ kg<sup>-1</sup> heat must be supplied but only 2015 kJ kg<sup>-1</sup> is recovered on condensation). If condensate is heated from 90 °C, the additional heat requirement is 386 kW and the loss is 16%. See Tab. 1.2, which also includes the loss due to boiler efficiency. Thus, for a boiler which is 80% efficient in raising 10 bara steam, but is supplied with feedwater at ambient temperature which is not heated by flue gas in an economizer, the actual heat delivered to the process is only 60% of the fuel used (2015/3368).

**Steam Distribution Losses** Steam is passed from the boiler to the dryer along steam mains. Ideally, these will be short, well-insulated and well-maintained, and heat losses and steam leakage will be below 5%. However, many large and old sites have extensive networks of steam mains which suffer significant losses from long pipe runs, missing or damaged insulation, redundant sections which have not been blanked off, steam leaks and poorly maintained steam traps. In an extreme case (Kemp, 2007), half the steam generated at the boilers was unaccounted for.

Assuming a modest steam distribution loss of 10% and a condensate return temperature of 90 °C, Tab. 1.3 (left-hand side) and Fig. 1.2 show how the different losses add up for a typical steam-heated convective dryer, combining the dryer losses from Section 1.3.3.1 and the utility system losses from Section 1.3.3.2. The evaporative load is less than 30% of the gross calorific value of the fuel, illustrating starkly how a large number of apparently unimportant losses can add up to a major overall penalty. Even if the mains distribution losses were only 5% and the condensate return system heat loss was 10%, so that most condensate is returned above 100 °C in a pressurized system, the overall system efficiency is still only 33.5%, as shown in the right-hand side of Tab. 1.3; or, putting it another way, the amount of fuel required in the boiler in heating terms is three times the minimum heat required for the required evaporation duty.

In Fig. 1.2, the top right-hand segment represents losses in the dryer, the top left-hand segment is losses in the utility system, while the bottom solid region is the actual evaporation load.

Overall, it can be seen that for an indirect-heated convective dryer it will be extremely rare to get an overall thermal efficiency greater than 50% (expressed as

Tab. 1.3 Breakdown of fuel use for a typical convective steam-heated dryer.

Situation	Steam Distribution Loss 10%, 90 °C Condensate Return			Steam Distribution Loss 5%, Condensate Return Loss 10%		
	Heat Required	Marginal Heat, kW	Marginal %	Heat Required	Marginal Heat, kW	Marginal %
Minimum evaporation load	2400	2400	29.6	2400	2400	33.5
Adiabatic saturation	3023	623	7.7	3023	623	8.7
Exhaust approach 25 °C	3939	916	11.3	3939	916	12.8
Including solids heating	4407	467	5.8	4407	467	6.5
Including heat losses	4906	499	6.1	4906	499	7.0
Steam distribution loss	5451	545	6.7	5164	258	3.6
Condensate losses	6495	1044	12.9	5740	576	8.0
Boiler efficiency (80%)	8118	1623	20.0	7174	1434	20.0

latent heat of evaporation compared to gross calorific value of fuel), and this will require a highly efficient steam system as well as an efficient dryer. For a contact dryer, exhaust airflow and exhaust heat losses are much lower, but the maximum practicable figure is likely to be 70%. Neither of these figures include power for vacuum pumps, fans, and so on, which are covered in Section 1.3.3.3.

**Water and Thermal Oil Systems** Some dryers are heated by circulating loops containing hot water (at lower temperatures) or thermal fluid (at high temperatures). These still have to be heated by furnaces, and heat losses in these are similar to those in boilers. Flue gas heat losses will depend on the circulating fluid temperature,

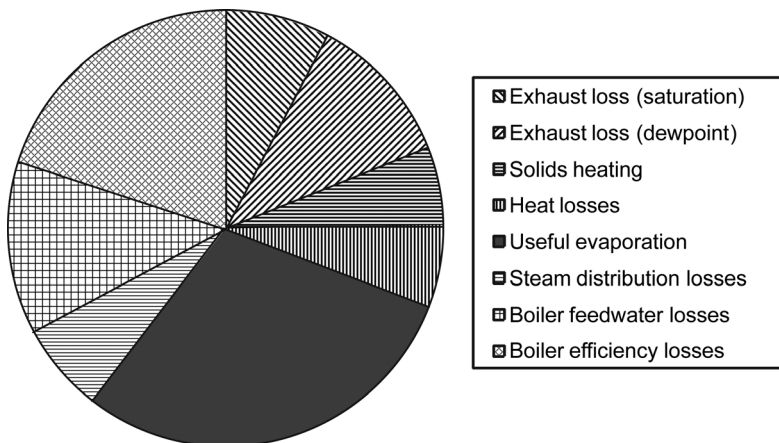
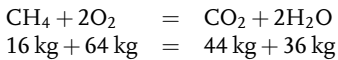


Fig. 1.2 Breakdown of overall usage of heat supplied as fuel for a typical steam-heated convective dryer.

although the flue gas can still be used to heat the incoming air to the boiler. Condensate losses should be lower, as all the heat transfer fluid is recirculated, but heat losses from the pipework must still be allowed for.

**Direct-Fired Dryers** In some dryers, fuel can be burned and the combustion gases can be used directly as the hot inlet gas for a convective dryer. This eliminates the losses involved in the steam-raising boilers. The system can only be used where it is acceptable for the product to come into direct contact with the combustion gases, and the inlet air has a higher humidity due to the additional water produced in combustion. Normally, natural gas is used and the additional water added to the air is given by:



Typically, the gross calorific value of natural gas is  $54\,000 \text{ kJ kg}^{-1}$  and the net calorific value  $47\,000 \text{ kJ kg}^{-1}$ . The latter is more convenient for calculations here, as the water generated by combustion ends up as water vapor. Hence 1 GJ heat is released from 21.3 kg fuel and generates 47.9 kg water vapor. In this case, where 130 kJ per kg air needs to be added, an additional 0.0062 kg water vapor is added and the inlet humidity rises from 7.5 to  $13.7 \text{ g kg}^{-1}$ . The new enthalpy is  $190 \text{ kJ kg}^{-1}$  instead of  $170 \text{ kJ kg}^{-1}$ , because of the extra energy contained in the latent heat of the water vapor – this in effect has been supplied by the difference between the gross and net calorific values.

The psychrometric chart in Fig. 1.3 shows a direct-fired system working with the same air inlet temperature and exhaust  $\Delta T$  as an indirect system. The dryer now works between point 7 (inlet) and 8 (exhaust), as against points 2 and 6 for the dryer heated by an indirect heat exchanger. Both cases allow for heat losses and solids heating, as before. The useful heat released as a percentage of heat in the fuel can be expressed as  $(47\,000/54\,000)$  in terms of calorific value or  $(170 - 40)/(190 - 40) = 130/150$  in terms of inlet enthalpy, both equating to an efficiency of 87%; the remaining 13% is used to heat the additional water vapor. This is better than the boiler efficiency of an indirect steam heated system, even if a small additional allowance is made for heat losses from the burner (usually well under 5%). In addition, all steam distribution and condensate return losses are eliminated entirely.

Table 1.4 shows the outlet conditions and burner duty, and can be compared with Table 1.1. To a first approximation, the required airflow will be the same as for the indirect heater, but the outlet humidity is higher by  $0.0062 \text{ kg kg}^{-1}$  throughout and the burner duty is greater than the heater duty.

Likewise, we can generate Tab. 1.5 showing a breakdown of the energy losses, equivalent to Tab. 1.3, and the corresponding pie chart, Fig. 1.4. In this case, we have broken out the heat required for the water vapor from combustion as a separate entity, and added a notional 5% heat loss from the burner itself. Despite these, the overall efficiency is more than 40%, which is considerably better than an equivalent steam-heated system. Comparing Fig 1.4 with Fig 1.2, the top left segment corresponding to utility system losses is much smaller.

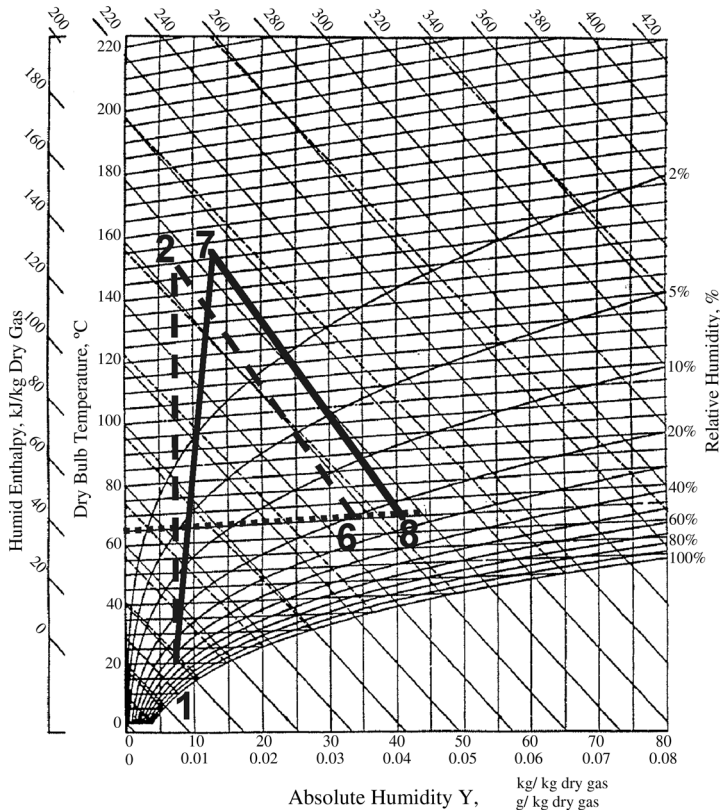


Fig. 1.3 Psychrometric chart for direct-fired and indirect-heated convective dryers.

**Electrical Heating** Electrical heating can be used, for example, by infrared heater or RF and microwave systems. However, the electricity must be generated somewhere. For typical thermal power stations, efficiencies are typically around 30–40% (though over 50% for modern combined cycle systems). Hence, allowing for capital costs and distribution losses, it is a reasonable rule of thumb that electrical power typically costs three times more per kWh than fuels in most countries. The exceptions are where

Tab. 1.4 Outlet conditions, airflow and heater duty for direct-fired dryer.

Condition	$T_{g,out}$ (°C)	$Y_{out}$ (kg kg <sup>-1</sup> )	$\Delta H_{latent}$ (kJ kg <sup>-1</sup> )	$W_g$ (kg s <sup>-1</sup> )	$Q_{burner}$ (kW)	$\eta$ (%)
Adiabatic saturation	40	0.0567	103.2	2.33	349	68.8
Exhaust approach 25 °C	65	0.0467	79.2	3.03	454	52.8
Including solids heating	65	0.0432	70.8	3.39	508	47.2
Including heat losses	65	0.0402	63.6	3.77	566	42.4

**Tab. 1.5** Breakdown of fuel use for a typical convective dryer with direct-fired burner.

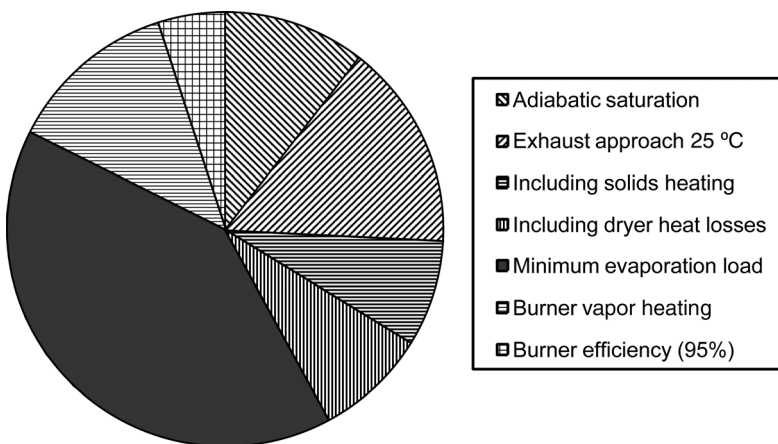
	Heat Required	Marginal Heat, kW	Marginal %
Minimum evaporation load	2400	2400	40.3
Adiabatic saturation	3023	623	10.5
Exhaust approach 25 °C	3939	916	15.4
Including solids heating	4407	467	7.8
Including dryer heat losses	4906	499	8.4
Burner vapor heating	5660	755	12.7
Burner efficiency (95%)	5958	298	5.0

cheap hydro-electric power is available (see also Section 1.3.4.2). Moreover, losses also occur in electrical heating; the magnetrons used for microwave heating typically only have an efficiency of 50% (delivered microwave energy compared to input power).

### 1.3.3.3 Other Energy Demands

These are as electrical power, and it must again be remembered that this typically costs three times as much as heat per unit energy (kW or kWh).

- 1) Solids transport costs, for example, screw feeders, bucket or belt conveyors, or fans for pneumatic conveying systems; also rotary discharge valves, and so on.
- 2) Vacuum pump power. For conductive (contact) dryers, where vacuum is almost invariably used to give better driving forces and achieve acceptable drying times.
- 3) Steam ejectors. These are used as an alternative method of pulling vacuum. Obviously they eliminate the power required for vacuum pumps, but extra steam is required (which cannot be recovered as condensate). As ejectors often work

**Fig. 1.4** Breakdown of fuel usage for a typical convective dryer with direct-fired burner.

best with ratios of around 1:1 for driver steam to vacuum flow, the steam consumption of the ejectors can be similar to that used in the dryer itself, thus almost doubling steam use.

- 4) Fan power. Depending on the system pressure drop, the exhaust fan power for the calculated airflow of  $3.8 \text{ kg s}^{-1}$  ( $13\,700 \text{ kg h}^{-1}$ ) will typically be in the range 50–100 kW. This is a significant demand compared with steam usage (bearing in mind the price differential).
- 5) Boiler feedwater pumps and other liquid pumping duties. Usually small.

#### 1.3.4

### Energy Cost and Environmental Impact

#### 1.3.4.1 Primary Energy Use

Dryers typically use both heat and electrical power. Sometimes these have been lumped together and quoted as a total energy consumption for the unit operation or site. This is often inappropriate. It is often more helpful to state energy use in terms of primary energy, total energy cost or, in these days of climate change, carbon footprint (total carbon dioxide released into the atmosphere).

Primary energy is the usage of the original source of fuel. Hence, for a dryer heated by steam, the primary energy use is the fuel burned in the boilers, which will typically be at least 20% higher than the heat delivered from the steam. However, the difference between point-of-use and primary energy consumption becomes far more significant for electrical power. If, as in most countries, this is generated from fossil fuels, the typical primary energy consumption can be about three times higher than the power use. So supplying 1 kW of evaporation by steam heating may require 1.2 kW of primary energy, but doing so by electrical heaters requires no less than 3 kW of primary energy.

#### 1.3.4.2 Energy Costs

Relative costs of energy sources tend to reflect the primary energy use, so the cost of electric power is typically three times that of fuel. The exception comes where power is mainly generated as hydro-electricity, in which case the fossil fuel use is zero and the generating cost is also very low. The charge made for hydro-electric power is usually based on amortization of the high capital costs, with a small amount for operating costs (principally labor and maintenance). Likewise, nuclear power tends to have relatively low fuel cost but high capital charges, which should also allow for final decommissioning costs.

For energy cost estimation, normally one should use fuel and power prices applicable at the specific site studied. If for any reason there are no data available, the following values can be used (2010 figures) for a ballpark costing only – remembering that energy prices can fluctuate widely:

Fuel £6 per GJ = £21.6 per MWh

Power £18 per GJ = £64.8 per MWh

### 1.3.4.3 Carbon Dioxide Emissions and Carbon Footprint

Carbon footprint varies between different fossil fuels because of their different ratios of carbon to hydrogen and calorific value. Typical figures for CO<sub>2</sub> produced per kWh of energy for different fuel sources are:

Natural gas: 0.184 kg kWh<sup>-1</sup>

Diesel oil and fuel oil: 0.25 kg kWh<sup>-1</sup>

Coal: 0.324 kg kWh<sup>-1</sup>

The values for oil and coal vary with grade, and for natural gas the figure depends on the proportion of other hydrocarbons and gases mixed with the main constituent, methane (CH<sub>4</sub>).

The value for electric power will depend on how it is generated, which will be a mix of technologies varying with the country. Using the general rule of thumb that electricity requires three times as much primary energy per kW as heat, for a country which generates all its power from fossil fuels, the expected figure would be roughly 0.55 kg kWh<sup>-1</sup> if natural gas is the main source, 0.75 kg kWh<sup>-1</sup> for oil and 0.95 kg kWh<sup>-1</sup> for coal. This will be reduced if a significant proportion of a country's power is hydroelectric or nuclear power, which have virtually zero CO<sub>2</sub> emissions.

In practice, typical values are 0.4–0.6 kg kWh<sup>-1</sup> for Europe, 0.6 kg kWh<sup>-1</sup> for North America, 0.8–1.0 kg kWh<sup>-1</sup> for developing countries, but with significant exceptions. The highest values are for countries using a high proportion of coal, such as Australia (0.953 kg kWh<sup>-1</sup>). As expected, values are substantially lower for countries which generate much of their power from renewable sources (hydroelectricity, wind, etc.) or nuclear, both of which give an effectively zero carbon footprint. For example, for France (where over 70% of electricity is nuclear) the 2010 figure is only 0.088 kg kWh<sup>-1</sup>. Table 1.6 and Fig 1.5 show comparative figures for a range of major energy-using countries.

**Tab. 1.6** Carbon footprint for fuels and electric power for selected countries; Data from National Energy Foundation, UK, 2010, all figures in kgCO<sub>2</sub> kWh<sup>-1</sup>.

Fossil Fuels	Natural Gas	0.184
	Fuel oil	0.25
	Diesel oil	0.25
	Coal	0.324
Electricity	France	0.088
	Germany	0.458
	UK	0.541
	USA	0.613
	China	0.836
	India	0.924
	Australia	0.953
CHP	Gas engine	0.27

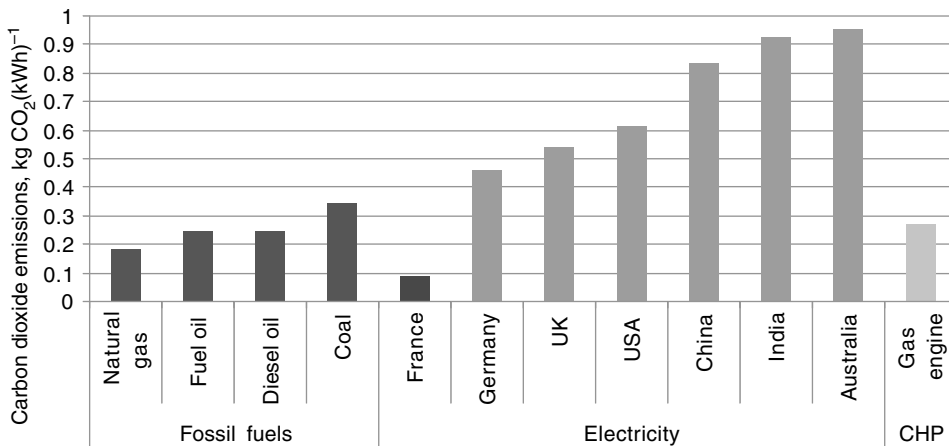


Fig. 1.5 Carbon footprint for fuels and electric power for selected countries.

## 1.4

### Setting Targets for Energy Reduction

#### 1.4.1

##### Energy Targets

Logically, we should aim to achieve a significant energy reduction by setting a target to aim at. How do we evaluate these energy targets? There are three levels:

- 1) “Management” or “arbitrary” targets – aiming for a specified percentage reduction year-on-year, say 5 or 10%. Kemp (2007) points out that this takes no account of the reality of the process, and unfairly penalizes efficient ones! However, these targets can be appropriate for “good housekeeping” type measures, especially if there are very little process data available.
- 2) Rigorous targets for the existing process – based on a calculation of what it should be using, given a specified evaporation load and expected dryer efficiency. To calculate this, reliable values of key process heat loads are needed, preferably obtained from a consistent heat and mass balance.
- 3) Further reduced targets for an improved process – specifically redesigned to inherently use less energy, by reducing the evaporation requirement or substituting a more efficient dryer.

In the case study at the end (Section 1.6) we can see how these are applied in a specific case.

Key tools for analysis include:

- 1) Heat and mass balance on the dryer; need not be precise, but must be consistent.



- 2) Overall energy consumption data for the process plant, including heat and power supplied (fuel, steam, imports from grid) and local usage (broken down by process).
- 3) Pinch analysis; see Section 1.4.2.

#### 1.4.2

### Pinch Analysis

#### 1.4.2.1 Basic Principles

In the last 30 years, pinch analysis (also known as pinch technology or process integration) has been shown to be a vital tool for assessing minimum required energy consumption and setting rigorous targets, and hence identifying energy saving opportunities. It allows a systematic analysis of the overall plant, and has developed into the broader subject of process synthesis.

This section summarizes the basic principles. The overall methodology is described in detail by Kemp (2007), Smith (2005), Linnhoff *et al.* (1982) and in ESDU Data Items (1987, 1989, 1990). Kemp (1991, 2005, 2007) and Smith (2005) have applied the method specifically to dryers.

Pinch analysis examines the flows and unit operations in processes which require or release heat. These are categorized into “hot streams” (which give up heat, e.g., a hot dryer exhaust stream as it cools and condenses) and “cold streams” (which require heat, e.g., the wet solids entering the dryer which need to be heated to evaporate off the moisture). All the heating requirements could be fulfilled by hot utilities (e.g. steam, hot water, furnace gases) and likewise the cooling needs could be fulfilled by cold utilities (e.g., cooling water, chilled water or refrigeration). However, heat can be recovered between hot streams at a higher temperature and cold streams at a lower temperature. All heat exchange reduces both hot and cold utility use, and hence reduces fuel and power use and emissions. Pinch analysis allows rigorous *energy targets* to be calculated for how much heat exchange is possible, and hence the minimum possible levels of hot and cold utility use.

Most processes have a *pinch temperature*. Above this temperature they have a net heat requirement; below the pinch, there is net waste heat rejection. Heating below the pinch, cooling above the pinch, or heat exchange across the pinch all incur an energy penalty. Conversely, heat pumps only achieve a real energy saving if they work backwards across the pinch, upgrading useless below-pinch waste heat to useful above-pinch heat. Hence, a pinch analysis of a system is an important prerequisite of any energy saving project, to ensure that it will achieve its aims.

Streams are characterized by their temperature and heat load (kW), the latter being calculated as:

$$Q_{\text{stream}} = Wc_p(T_{\text{in}} - T_{\text{out}}) \quad (1.10)$$

In terms of potential heat recovery, the main hot and cold streams in a typical dryer are:

Hot streams

H1. The exhaust gas from the dryer. This includes both sensible and latent heat.

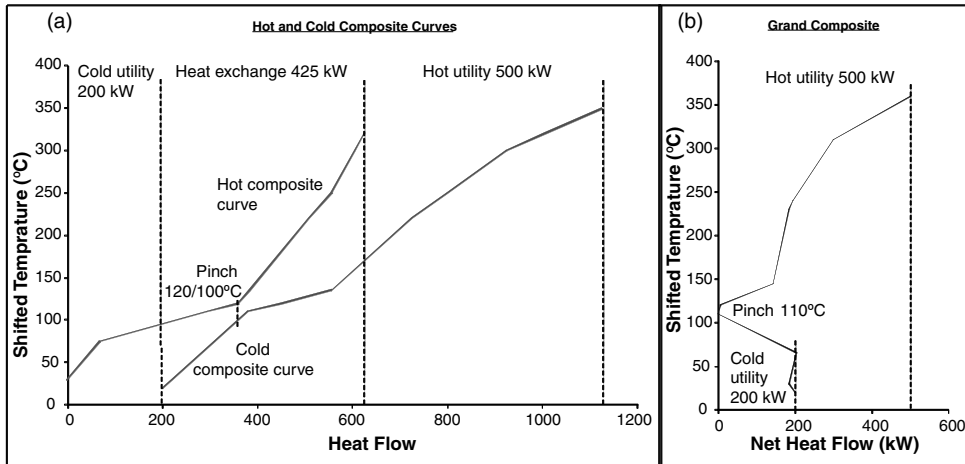


Fig. 1.6 (a) Composite and (b) grand composite curves for a typical liquid-phase process.

H2. The hot solids emerging from the dryer. A sensible cooling load, normally much less than H1.

#### Cold streams

- C1. (For convective dryers) Heating the drying air, usually from ambient to dryer inlet temperature.
- C2. (For contact dryers) Heat supply to the dryer via the jacket.
- C3. Preheating the solids before they enter the dryer.

C1 or C2 are normally the dominant heat loads, as these will supply the heat used for evaporation.

The hot and cold streams can be effectively represented on a temperature–heat load diagram, as shown in Fig. 1.6 for a typical liquid-phase process. Where there are multiple hot and cold streams, their heat loads can be summed together to produce composite curves. The hot composite curve is the sum of the heat loads of all the hot streams over the temperature ranges where each one exists. Likewise, the cold composite curve is the sum of the heat loads of all the cold streams. A minimum temperature difference for heat exchange,  $\Delta T_{\min}$ , must be selected, and has been chosen as 20 K. This gives the vertical distance between the curves. The point of closest vertical approach is the pinch, which here corresponds to a temperature of 100 °C for the cold streams and 120 °C for the hot streams. The region of overlap between the hot and cold composite curves shows the opportunity for heat exchange, recovering heat from hot to cold streams. The remaining heating and cooling in the non-overlapping region must be supplied by heating or cooling utilities.

A further useful calculation is to subtract the total heat required for the cold streams from that available from the hot streams at any temperature, to give the net requirements for hot or cold utility (external supply of heating or cooling) at any temperature. This gives the grand composite curve (GCC), shown in Fig 1.6b. Above

the pinch, hot utility is required; below the pinch, cold utility is needed. To allow for the minimum temperature difference for heat exchange  $\Delta T_{\min}$ , hot stream temperatures must be reduced by half this amount ( $10^\circ\text{C}$ ) and the cold stream temperatures increased, to give “shifted temperatures”. The grand composite curve shows the exact location of the pinch more clearly than the composite curves.

A heat exchange system can then be designed to try to achieve the energy targets. It must be remembered that pinch analysis targets give the maximum feasible heat recovery and that some aspects of this, particularly heat exchangers with small loads, may be uneconomic. Very often, there is a capital-energy tradeoff where some potential heat recovery is sacrificed to give a cheaper, simpler project with a better economic rate of return.

#### 1.4.2.2 Application of Pinch Analysis to Dryers

Virtually all dryers use air as the carrier gas and water as the solvent to be evaporated. The high heat requirement of dryers is almost entirely due to the latent heat of evaporation of the water. Much of the heat supplied to the dryer emerges as the latent heat of the vapor in the exhaust gas, which can only be recovered by condensing the water vapor from the exhaust. However, as saturation humidity increases almost exponentially with temperature, the dewpoint of exhaust air is generally  $50^\circ\text{C}$  or lower. It is very rare for this to be above the process pinch and the heat must thus be wasted. Hence, it is usual to simply vent the dryer exhaust from a stack, possibly recovering a small proportion of the heat as sensible heat.

Heat supply to convective dryers is in the form of hot air. Ambient air is drawn in and heated in either a direct-fired or indirect-fired furnace. The heat load of the dryer therefore plots as a sloping line. Composite curves for a typical dryer are shown in Fig. 1.7. It is clear that the scope for heat recovery in the basic system is limited. Some heat can be recovered from the dryer exhaust to the cold feed air. Usually this is only a

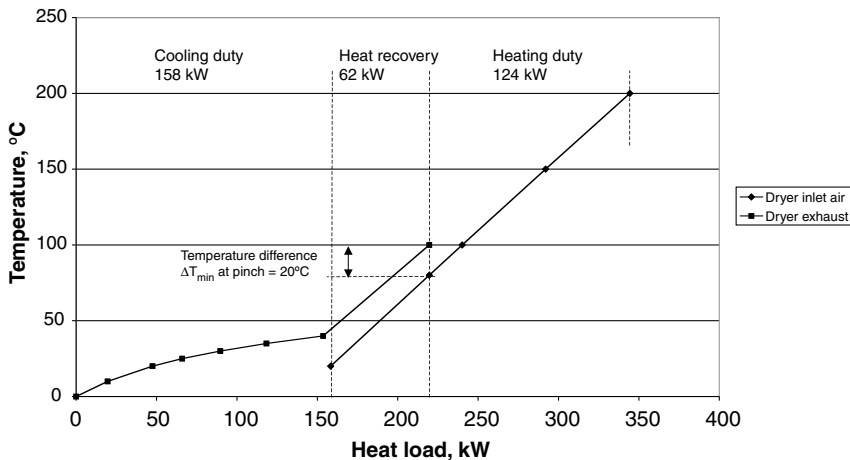


Fig. 1.7 Hot and cold composite curves for a simple convective dryer.

Tab. 1.7 Stream data for simple convective dryer example.

Hot Stream H1 (Dryer Exhaust)			Cold Stream C1 (Inlet Air Heating)		
Temperature (°C)	Heat Load (kW)	Adjusted (kW)	Temperature (°C)	Heat Load (kW)	Adjusted (kW)
100	229	220	20	45	158
40	163	154	80	107	220
30	99	90	100	127	240
20	56	47	150	179	292
0	9	0	200	231	344

small proportion of that available, but nevertheless, the actual cost savings can be significant because dryers are so energy-intensive.

Let us take a simple practical example. A convective dryer operates with  $1 \text{ kg s}^{-1}$  of dry airflow at an inlet humidity of  $0.01 \text{ kg kg}^{-1}$ . This hot air is heated from 20 to  $200^\circ\text{C}$  and used to evaporate moisture from solids. The exhaust gas temperature is  $100^\circ\text{C}$ . From calculation, or a psychrometric chart, we can see that the enthalpy of the inlet air rises from  $45 \text{ kJ kg}^{-1}$  to  $230 \text{ kJ kg}^{-1}$ . Ignoring both heat losses and the sensible heating of the solids, the exhaust air emerges with a humidity of  $0.048 \text{ kg kg}^{-1}$  and a dewpoint of  $40^\circ\text{C}$ . Hence  $0.038 \text{ kg s}^{-1}$  of water is evaporated, and (taking latent heat as  $2450 \text{ kJ kg}^{-1}$ ) the inherent requirement is approximately  $93 \text{ kJ s}^{-1}$  ( $93 \text{ kW}$ ). In fact, approximately  $185 \text{ kW}$  has been used to heat the incoming air, so the dryer is barely 50% efficient – or significantly less when heat losses and the heating of the inlet solids are included. Table 1.7 tabulates the resulting temperature–heat load data for the key streams.

The “adjusted” data shifts the heat load figures so that the hot stream begins at zero heat load and the cold stream lies below it.  $\Delta T_{\min}$  has again been chosen as 20 K. The pinch is calculated as  $80^\circ\text{C}$  for the cold streams and  $100^\circ\text{C}$  for the hot streams, at an adjusted heat load of  $220 \text{ kW}$ . The resulting plot is shown in Fig. 1.7. The range over which the hot and cold streams overlap, where heat can be recovered from the hot exhaust stream to the cold inlet air, can be seen at a glance.

Likewise the grand composite curve (Fig. 1.8) shows the net heating and cooling requirements for the dryer. The heat loads of the dryer exhaust and inlet air do not quite match over the range  $30\text{--}90^\circ\text{C}$  (shifted temperature) because the exhaust has a higher humidity, and hence a higher specific heat capacity. The cooling requirement is calculated to condense all the water in the exhaust and reduce its temperature to  $0^\circ\text{C}$ , but in practice this will not normally be necessary; indeed, in many cases the dryer exhaust can be discharged directly at its final temperature after all useful heat has been recovered by heat exchange, and no cooling is required at all.

Compare with the composite and grand composite curves of the typical liquid-phase process shown in Fig 1.6. The overlap region for the liquid process composite curves is much greater than for a typical dryer, showing that there are proportionately more opportunities for heat recovery within the process.

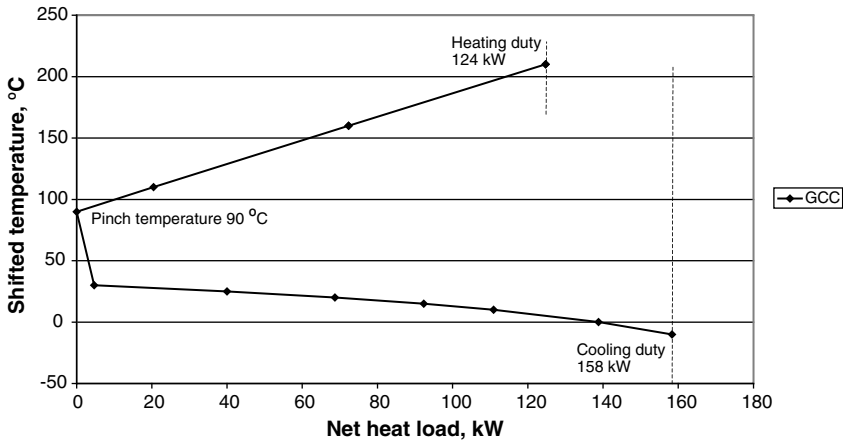


Fig. 1.8 Grand composite curve for a convective dryer.

#### 1.4.2.3 The Appropriate Placement Principle Applied to Dryers

Pinch analysis enables us to see not only how much energy the dryer is using, but whether it is able to exchange with other parts of the process, and whether these opportunities can be increased.

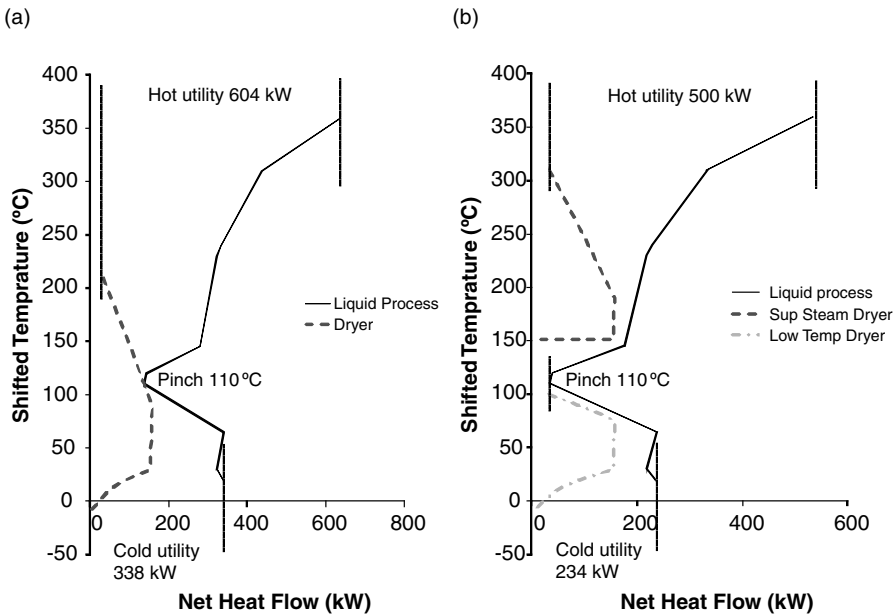
Let us consider the general process whose composite and grand composite curves are shown in Fig. 1.6, and the dryer shown in Figs. 1.7 and 1.8. If these are on the same site, are there possibilities for heat exchange between them? And are the operating conditions of the dryer and background process (particularly temperatures) such that this heat recovery is maximized? Or can we change the operating conditions to increase the potential for heat exchange between them?

The Appropriate Placement principle for a unit operation or a utility states that, to minimize energy use, it should ideally be placed so that it releases all its heat above the pinch temperature and above the GCC of the process, or receives all its heat below the pinch and below the GCC. Visually, if the heat demands of a unit operation, such as a dryer, are plotted on the same graph as the remaining “background process”, the dryer should fit either entirely above or entirely below the GCC. This means that it can exchange all its heat with the rest of the process, rather than requiring a separate supply.

To see whether this is the case, we can “split the grand composite curve”, plotting two separate lines for the dryer and the background process, as shown in Fig. 1.9a. The dryer GCC is reversed to allow this. The total target for the separate processes is 624 kW (124 kW for the dryer and 500 kW for the liquid process), while that for the combined processes is 604 kW, so that only 20 kW extra can be recovered by heat exchange.

It is clear that the dryer is working across the pinch; it is not so clear what can be done about it. The only possibilities are:

- 1) Reduce the temperature at which the dryer requires heat.



**Fig. 1.9** Split grand composite curves for (a) standard convective dryer, (b) pressurized superheated steam or low-temperature convective dryers.

- 2) Raise the temperature at which the dryer exhaust stream releases heat.
- 3) Alter the background process so that more of it fits above or below the dryer heat profile.

The first can be achieved by using a low-temperature dryer extracting heat below the pinch as warm air or warm water. The reduced temperature driving forces would normally cause a huge increase in the size and capital cost of the dryer. However, if a dispersion dryer (e.g., a fluidized bed or cascading rotary dryer) can be substituted for a layer dryer (e.g., an oven or tray unit), the much enhanced heat transfer coefficients may allow low-temperature drying with a small unit. Warm air can be fed directly to the dryer; warm water can heat it indirectly via internal coils. Alternatively, some preheating of the wet feed solids may be carried out in a pre-dryer working on below-pinch waste heat. This has particular advantages for sticky or temperature-sensitive materials.

The second option is virtually impossible with conventional air dryers as the dewpoint cannot be altered significantly. Recycling exhaust gases and raising the humidity will raise the dewpoint; however, it may adversely affect drying. In any case no heat can be recovered above the boiling point, 100 °C, unless the entire system is placed under high pressure – an extremely expensive option. However, in some cases a heat transformer has been used to absorb moisture from the exhaust gas and recover some of its heat.

If, instead, the superheated form of the solvent being evaporated is used as the carrier gas instead of air, a very different picture emerges. The recovered vapor can

then be condensed at high temperature, above the pinch. The commonest case is superheated steam drying, which also has the advantage of a better heat transfer coefficient between vapor and solids than for air. The steam is recirculated and reheated; a bleed equal to the evaporation rate is required, and this steam can be condensed to yield useful heat – at 100 °C when working at atmospheric pressure, or significantly above 100 °C if operating at elevated pressures. Superheated steam drying has previously been advocated for heat transfer or safety reasons, but it clearly has energy advantages too. The main drawback is that a large fan or compressor is required to recirculate the steam, and the power consumption of this can cancel out the savings from heat recovery. An interesting solution to this problem is the airless dryer (Stubbing, 1993, 1999), where no gas recirculation is used; the water driven off from the solids in the early stages of drying forces the air out of the system to create the superheated steam atmosphere. This system works at atmospheric pressure. In contrast, large-scale continuous superheated steam dryers used, for example, for pulp and paper processing, typically operate at high pressures and temperatures.

Figure 1.9b illustrates the placement of a pressurized superheated steam dryer above the process GCC, or a low-temperature dryer below the GCC.

Conversely, if the operating conditions of the dryer cannot be changed, it may be possible to alter those of the rest of the process instead. An example (Kemp, 2007; Linnhoff *et al.*, 1982) is for a gelatin plant where a three-stage dryer (working at 60–80 °C) followed a three-stage evaporation system. The composite and grand composite curves are shown in Fig. 1.10, highlighting the heat loads due to the evaporator (E1-E3) and dryer (D1-D3). The pinch was initially at 40 °C and it was impossible to bring the dryer below that. Instead, the operating pressure of the evaporator train was raised so that it discharged vapor at a higher temperature; in effect, shifting the pinch upwards to 97 °C and bringing it above the temperature of the dryer heat loads. The net result was that the vapor from evaporator effect 2 was then hot enough to heat the dryer directly, giving an overall energy reduction of nearly a third, from 1517 to 1027 kW, as shown in Fig. 1.11. Although increased pressure

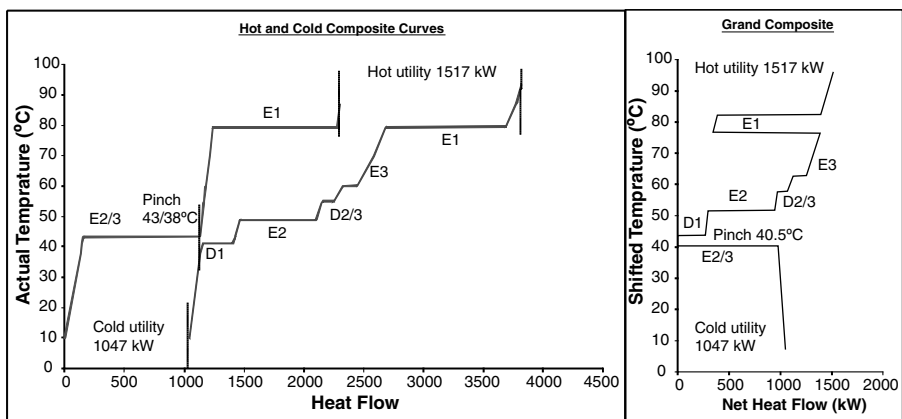


Fig. 1.10 Composite and grand composite curves for gelatin process, original form.

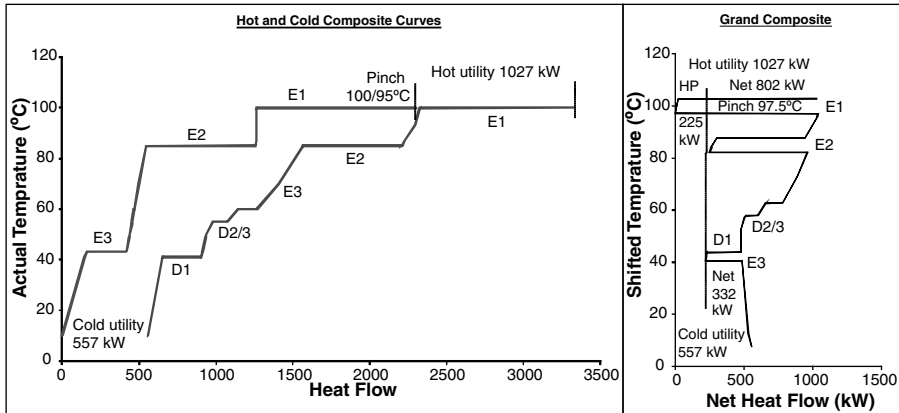


Fig. 1.11 Composite and grand composite curves for gelatin process, modified form.

normally entails increased equipment cost, in this case the evaporator train was working under vacuum, so all that was needed was to reduce the vacuum pulled (also reducing vacuum pump power and cost). The evaporator had been designed as a unit operation, with a multiple effect configuration over a range of temperatures; this was correct design taken in isolation, but not when a holistic analysis was made over the complete plant! There was also a thermocompressor across the first evaporator effect. Being a heat pump, this should be working across the pinch, but in the original configuration it was above the pinch; in the revised layout it was correct. This saves another 225 kW and brings the heat required from external utilities down to 802 kW, a total saving of no less than 46%. Another example of linking evaporators and dryers appears in the case study in Section 1.6.

#### 1.4.2.4 Pinch Analysis and Utility Systems

The grand composite curve is also helpful for optimizing configuration of the utility systems that supply the heating and cooling requirements. The operating line of the hot utility system needs to lie entirely above the process GCC, and the cold utility system below it. If heat is provided by condensing steam, this plots as a horizontal line at the condensation temperature; multiple steam levels may be used. Alternatively, heat may be supplied from a hot gas stream (air or flue gas); this releases sensible heat over a range of temperatures and plots as a sloping line. Both methods are illustrated in Fig. 1.12.

In many cases, the dryer inherently lies across the process pinch, and it is very difficult to reduce its energy consumption significantly using pinch technology. However, the net cost of supplying the heat can be substantially reduced by using a cogeneration (CHP) system; the exhaust from either a gas turbine or a reciprocating engine is hot enough to supply almost any hot gas dryer. The exact inlet temperature is easily controlled by adding a varying amount of cool dilution air. CHP is described further in Section 1.5.4.3. The main limitations on such a system are the capital cost and the cleanness of the exhaust; gas turbines and gas engines are more acceptable in the latter respect than diesels. The heat profile of the exhaust should again lie above



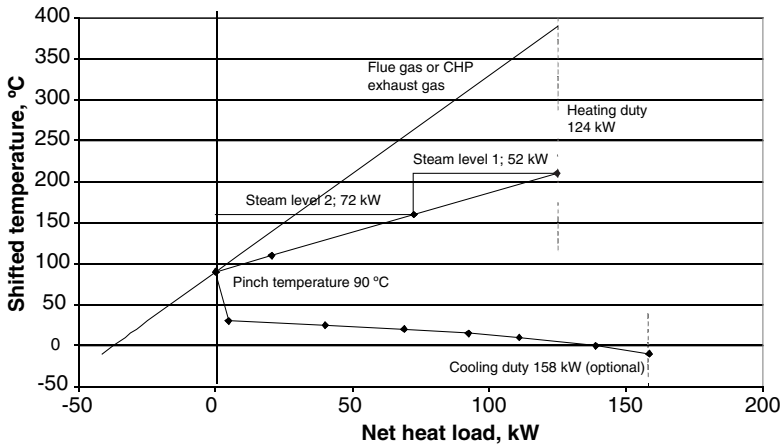


Fig. 1.12 Fitting utility heating to the process grand composite curve.

the process GCC as much as possible, under the Appropriate Placement principle, to minimize the overall heat requirement of the process plus utility system. See Figs. 1.12 and 1.14.

Heat pumps can be an option. Again, the Appropriate Placement principle applies, but in this case, as the heat is released at a higher temperature than the cooling, the heat pump should be placed so that the heat is released above the GCC, and any heat absorbed from the process should come below the GCC. In other words, the heat pump should work backwards across the pinch. For many dryers, the temperature lift is too high to achieve this. However, for dryers using a large air recycle with a low temperature lift, including many food and agricultural dryers, heat pumping may be economic. Further details are given in Section 1.5.4.4.

### 1.4.3

#### Drying in the Context of the Overall Process

Dryers are normally part of a larger solids processing operation, and two major types can be distinguished.

- 1) Insoluble solids. Typically formed by crystallization from solution, mechanically separated by filtration or centrifugation, and then dried. As much water as possible is removed in the mechanical separation step, but even if a high vacuum or pressure is used in the filter, or a high speed in a centrifuge, there will be a significant amount of both unbound and bound moisture which cannot be removed mechanically. The thermal energy required to dry this off is far greater than that used for mechanical separation.
- 2) Soluble solids. Here a chemical or biological entity may be formed in solution, concentrated by evaporation until handling or pumping becomes too difficult, and then dried. Even where the majority of water is removed in the evaporation, the energy consumption of that step can be greatly reduced by using multiple

effects and reusing the latent heat of evaporation. For drying, by contrast, the temperature drop is such that only a single-stage unit can normally be used. Moreover, evaporators can give efficiencies close to 100%, whereas dryers can frequently be at 50% or below, due to driving force losses, heat lost in exhaust air and additional energy required to remove bound moisture.

One of the best ways to reduce the energy load on a dryer is to reduce the initial moisture content (or increase the % solids) of the incoming feed.

The methods for doing this will obviously depend on the upstream process. Typical situations are:

- 1) Filtration or centrifugation. Probably the most common upstream unit operations, occurring in processes such as bulk and fine chemicals, primary pharmaceuticals and other situations where solids have been formed (e.g., by crystallization or precipitation) and are being concentrated as a slurry. Obviously, the inlet moisture to the dryer and the dryer heat load will be minimized by maximizing mechanical dewatering of the slurry in the filter or centrifuge.
- 2) Granulation, as in secondary pharmaceutical processes. Minimize the water required to achieve granulation by careful design, effective mixing and good choice of operating conditions such as agitation speed. In some cases, alternative granulation methods such as roller compaction can be considered which inherently use little or no added water.
- 3) Evaporation to a solid, slurry or paste, as in many food and mineral processes. Use multiple effects and temperature stacking in evaporators wherever possible, so that energy (particularly latent heat) can be reused. Mechanical and thermal vapor recompression (basically heat pumping using vapor) may also be applicable. As the solution becomes more concentrated and viscous, and boiling point rise becomes greater, all these techniques become harder to apply and a basic 1:1 condensation/evaporation system becomes the norm. Nevertheless, this is often still more thermally efficient than a dryer, so it is desirable to get the solids as dry as possible before transferring to the dryer.
- 4) Liquid processing or evaporation to a liquid feed form (slurry or solution), for feeding to a spray dryer, film-drum or thin-film (scraped-surface) dryer. Again, one would normally try to increase the concentration as far as possible by evaporation or (for a slurry) mechanical dewatering, as long as the feed remains sufficiently pumpable.

See also Section 1.5.2.1.

## 1.5 Classification of Energy Reduction Methods

An initial overview analysis of the energy requirements of the process, based on the evaporation load, shows how much energy is inherently required and, by comparing with current measured energy usage, what opportunities there are for savings.

Opportunities to reduce energy consumption can be classified into three main categories;

- a) Reduce the evaporation load – for example, by upstream dewatering to reduce initial moisture content, or avoiding overdrying.
- b) Increase the dryer efficiency – for example, by improving insulation and reducing heat losses, installing heat recovery or changing operating parameters.
- c) Improve the energy supply (utility) systems – for example, increase boiler efficiency, reduce distribution losses, install combined heat and power (CHP), heat pumps, waste incineration or other alternative low-cost fuels.

It is also useful to subdivide further:

- a) Reduce the evaporation load by:
  - 1) Reducing the inherent energy requirement for drying, for example, by dewatering the feed, or avoiding the need for drying altogether.
  - 2) Increasing the efficiency of the dryer, by reducing heat losses, total air flow or batch times.
- b) Increase the dryer efficiency by:
  - 3) Heat recovery within the dryer system, between hot and cold streams.
  - 4) Heat exchange between the dryer and surrounding processes.
- c) Improve the utility systems by:
  - 5) Using lower-cost heat sources to supply the heat requirement, for example, low-grade heat or renewable energy (including alternative fuels, biofuels and waste).
  - 6) Improving the efficiency of the energy supply system, for example, by reducing losses in the boiler or steam distribution system.
  - 7) Using CHP; co-generate power while supplying the heat requirement to the dryer.
  - 8) Using heat pumps to recover waste heat to provide dryer heating.

Hence, methods 1 and 2 can be categorized as ways of directly reducing the dryer heat duty, methods 3 and 4 use heat recovery to reduce the amount required from external utilities (heating and cooling systems), and methods 5–8 reduce the cost of the utilities or the primary energy requirement. The order of classification represents the logical order in which the steps should be investigated practically; there is little point in sizing a heat recovery scheme if it is possible to alter the dryer heat flows significantly. In all cases, the ultimate aim and benefit is the same; to reduce the net usage of fossil fuels and other non-renewable energy sources, and to minimize emissions of CO<sub>2</sub>, greenhouse gases, pollutants such as NO<sub>x</sub> and SO<sub>x</sub>, and other waste materials.

Likewise, from a pinch analysis viewpoint, heat duty reduction (methods 1 and 2) is a process change which reduces energy targets, methods 3 and 4 are heat recovery or heat exchange which help to achieve calculated targets, and methods 5–8 improve the efficiency or reduce the cost of the utility systems meeting the residual energy demands.

These eight methods will be considered in more detail in Sections 1.5.2–1.5.4.

## 1.5.1

**Reducing the Heater Duty of a Convective Dryer**

Referring back to Section 1.3.2, we derived two alternative expressions for the heater duty of a convective dryer:

$$Q_{\text{heater}} = W_g c_{p_g} (T_{g,\text{in}} - T_{g,a}) \quad (1.5)$$

$$Q_{\text{heater}} = \frac{(T_{g,\text{in}} - T_{g,a})}{(T_{g,\text{in}} - T_{g,\text{out}})} [W_s (X_{\text{in}} - X_{\text{out}}) \Delta H_v + Q_{s,\text{sens}} + Q_{\text{loss}}] \quad (1.7)$$

Here  $T_{g,\text{in}}$  is the inlet temperature to the dryer and  $T_{g,a}$  is the temperature at which the air is supplied. From Eq. 1.5, we see that to reduce the heat duty  $Q_{\text{heater}}$ , we will need to reduce the airflow  $W_g$ , decrease  $T_{g,\text{in}}$  or increase  $T_{g,a}$ . If we know the dryer heat load and the air inlet and outlet temperatures, we can calculate the required air flow by combining Eqs. 1.5 and 1.7, or rearranging Eq. 1.6:

$$W_g = \frac{W_s (X_{\text{in}} - X_{\text{out}}) \Delta H_v + Q_{s,\text{sens}} + Q_{\text{loss}}}{c_{p_g} (T_{g,\text{in}} - T_{g,\text{out}})} \quad (1.11)$$

Likewise, from Eq. 1.7, we have the following alternative ways to reduce  $Q_{\text{heater}}$  for a fixed production rate  $W_s$ :

- Reduce the inherent evaporative duty by reducing inlet moisture content  $X_{\text{in}}$ , increasing final moisture content  $X_{\text{out}}$  or reducing the latent heat of evaporation  $\Delta H_v$  (Section 1.5.2.1).
- Reduce heat loss  $Q_{\text{loss}}$  or change operating conditions by increasing inlet gas temperature  $T_{g,\text{in}}$  or decreasing outlet gas temperature  $T_{g,\text{out}}$  (Section 1.5.2.2).
- Preheat the air entering the heater (increase  $T_{g,a}$ ) by heat recovery or exhaust air recycle (Section 1.5.3).

We will continue to use our worked example from Section 1.3 as a base case:  $W_s$  (dry basis) =  $1 \text{ kg s}^{-1}$ ,  $X_{\text{in}} = 0.12 \text{ kg kg}^{-1}$ ,  $X_{\text{out}} = 0.02 \text{ kg kg}^{-1}$ ,  $T_{g,\text{in}} = 150^\circ\text{C}$ ,  $T_{g,a} = 20^\circ\text{C}$ ,  $T_{g,\text{out}} = 65^\circ\text{C}$ ,  $Q_{s,\text{sens}} = 30 \text{ kW}$ ,  $Q_{\text{loss}} = 30 \text{ kW}$ ,  $c_{p_g} = 1 \text{ kJ kg}^{-1} \text{ K}^{-1}$ .

There is a complication with the latent heat of evaporation. We have generally taken  $\Delta H_v = 2400 \text{ kJ kg}^{-1}$ . However, the evaporated water enters with the solid at  $20^\circ\text{C}$ , with  $h_l = 84 \text{ kJ kg}^{-1}$  (from steam tables), and emerges with the exhaust air at  $65^\circ\text{C}$ , with  $h_v = 2618 \text{ kJ kg}^{-1}$ . So for use in Eq. 1.7, we need to use  $\Delta H_v = 2534 \text{ kJ kg}^{-1}$ . The difference is due to the additional sensible heat taken up by the vapor. Hence  $Q_{\text{heater}} = (313 \times 130/85) = 479 \text{ kW}$ . The slight difference from the value of 491 kW in Table 1.1 is due to the difference between the calculation and reading from a psychrometric chart. Heat of wetting, which would further increase  $\Delta H_v$  at low moisture content, is assumed zero (or rolled into  $Q_{\text{loss}}$ ).

From Eq. 1.11,  $W_g$  (dry basis) =  $(313/85) = 3.69 \text{ kg s}^{-1}$ . Using Eq. 1.5 as a cross-check,  $Q_{\text{heater}} = (3.69 \times 1.0 \times 130) = 479 \text{ kW}$ . This will be used as the base case for all the following calculations.

## 1.5.2

**Direct Reduction of Dryer Heat Duty****1.5.2.1 Reducing the Inherent Heat Requirement for Drying**

The dominant component is the evaporative heat load, and reducing this requires a reduction in the moisture removed in the dryer (or, in special cases, in the latent heat of evaporation).

- 1) Reduce the inlet moisture  $X_{in}$ ; this can only be achieved by altering the upstream process, and a holistic approach to overall solids process design as described by Kemp (2004) will be helpful. Typical methods include improved mechanical dewatering by centrifuging, vacuum or pressure filtration, or gas blowing of the filter cake supplied to the dryer. A promising development in recent years has been the use of superheated steam for filter cake dewatering, for example, by the Hi-Bar pressure filter manufactured by Bokela (Karlsruhe, Germany). This provides some heat for evaporation, but the more significant mechanism appears to be that the surface tension forces binding the water to the cake are reduced and substantially more liquid is removed in the mechanical dewatering step. A similar phenomenon has been observed in paper dryers, where heating the felt rollers allowed more water to be detached mechanically from the sheet surface. If  $X_{in}$  can be reduced from 12 to 10%,  $W_g = (2534 \times 0.08 + 30 + 30)/85 = (263/85) = 3.09 \text{ kg s}^{-1}$  from Eq. 1.11. From Eq. 1.5,  $Q_{heater} = (3.09 \times 130) = 402 \text{ kW}$ , a substantial reduction of 77 kW, or 16% of the base case heater duty.

Another method of reducing initial moisture content is by absorption or adsorption of some of the liquid in the material. However, the absorbent material needs to be regenerated and, as this will normally be done thermally, the latent heat of evaporation must still be removed. There can only be a net gain in energy usage if either (i) the absorbent can be regenerated at a low temperature using low-grade waste heat which would otherwise be thrown away, or (ii) the absorbent can be regenerated at a high temperature giving a more thermally efficient system than normal drying. (i) will never normally occur, but (ii) can occur for drying of foodstuffs where there are severe temperature limitations on the product; see Chapter 5 and also van Deventer (2002).

The absorbent solid must also be easily separable from the main product.

- 2) Increase the outlet moisture  $X_{out}$  by relaxing the product specification. This rarely makes a large difference to the heat balance; however, avoiding a stringent requirement for low final moisture content can substantially reduce drying time, and bring major benefits because drying efficiency is low in the tail end of the falling-rate drying period. A common consequence is a reduction in outlet gas temperature  $T_{g,out}$ . Hence  $(T_{g,in} - T_{g,out})$  is greater, the heat obtained from 1 kg of air increases proportionally, and from Eq. 1.11, the airflow can be reduced. For example, if  $X_{out}$  is relaxed to 2.5% ( $0.025 \text{ kg kg}^{-1}$ ), the evaporative heat duty only falls from 253.4 to 240.7 kW, a 5% saving. However, if  $T_{g,out}$  also falls from 65 to 55 °C,  $W_g = (302/95) = 3.17 \text{ kg s}^{-1}$ . From Eq. 1.5,  $Q_{heater} = (3.17 \times 130) = 412 \text{ kW}$ , a total saving of 67 kW or nearly 15%.

- 3) Reduce the latent heat of evaporation  $\Delta H_v$ , by substituting a solvent with lower latent heat. Bahu (1991) suggested displacing water by toluene in a pre-drying step. However, this approach has not been adopted in practice, because there are now very severe emissions limits on VOCs (volatile organic compounds) such as toluene. Hence, the solution is worse than the problem, as the potential environmental damage and gas cleaning costs from using toluene far outweigh the benefits from the energy savings.

### 1.5.2.2 Altering Operating Conditions to Improve Dryer Efficiency

- 1) Increase inlet gas temperature  $T_{g,in}$ ; usually  $T_{g,out} > T_{g,a}$ , so the first term in Eq. 1.7 will decrease. The limitation is the risk of thermal damage. From Eq. 1.5, to achieve a lower drying duty,  $W_g$  must fall. For example, suppose  $T_{g,in}$  is increased to 170 °C and  $T_{g,out}$  can be maintained at 65 °C. From Eq. 1.11,  $W_g = (313/105) = 2.98 \text{ kg s}^{-1}$ , and from Eq. 1.5,  $Q_{heater} = (2.98 \times 150) = 448 \text{ kW}$ , a saving of 31 kW. In practice, however, it is difficult to avoid an increase in  $T_{g,out}$ , which increases exhaust heat losses and reduces the gain in efficiency. If  $T_{g,out}$  rises to 70 °C,  $W_g = (313/100) = 3.13 \text{ kg s}^{-1}$ , and  $Q_{heater} = (3.13 \times 150) = 470 \text{ kW}$ .
- 2) Decrease outlet gas temperature  $T_{g,out}$ . From Eq. 1.11,  $W_g$  falls and exhaust heat losses are reduced. However, outlet humidity and relative humidity increase, along with dust concentrations and condensation problems, while temperature and humidity driving forces fall. Hence, drying times will tend to increase, and it may become difficult to achieve the final moisture specification.
- 3) Reduce heat losses  $Q_{loss}$  by adding insulation, removing leaks and so on. For example, if  $Q_{loss}$  is halved from 30 kW (~10%) to 15 kW (~5%), Eq. 1.11 shows that the required airflow also falls:  $W_g = (298/85) = 3.51 \text{ kg s}^{-1}$ , and  $Q_{heater} = (3.51 \times 130) = 456 \text{ kW}$ , saving 23 kW (5%).

Important benefits can also arise from improved control to ensure that a dryer is always working at its preferred design conditions, or to ensure that batch drying is stopped as soon as the product has reached its target moisture content.

Table 1.8 summarizes the various options for reducing heater duty, including heat recovery possibilities described in Section 1.5.3.

**Tab. 1.8** Effect of energy-saving changes on airflow and heater duty for indirect dryer.

Parameter	$T_{g,in}$ (°C)	$T_{g,out}$ (°C)	$Q_{loss}$ (kW)	$W_g$ ( $\text{kg s}^{-1}$ )	$Q_{heater}$ (kW)	Saving (%)
Operating Conditions						
Base case	150	65	30	3.69	479	0
Inlet moisture 10%	150	65	30	3.09	402	16
Outlet moisture 2.5%	150	65	30	3.54	460	4
Outlet 2.5%, lower $T_{g,out}$	150	55	30	3.17	412	14
Increased $T_{g,in}$	170	65	30	2.98	448	7
Increased $T_{g,in}$ and $T_{g,out}$	170	70	30	3.13	470	2
Reduced heat loss	150	65	15	3.51	456	5
Air preheat to 45 °C	150	65	30	3.69	387	19

## 1.5.3

**Heat Recovery and Heat Exchange****1.5.3.1 Heat Exchange Within the Dryer**

The heating and cooling profiles in Fig. 1.7 show that, for a typical dryer, only a small proportion of the exhaust heat can be recovered to heat incoming cold air, and none of the latent heat of evaporation as it is released below 40 °C. Hence dryers, especially convective dryers, tend to have low efficiencies, measured in terms of the external heat requirement, compared to that theoretically necessary to evaporate the moisture from the solids. In Fig. 1.7, heat recovery has reduced the dryer heat requirement by one-third, from 186 to 124 kW. However, this is still substantially greater than the evaporative heat load of 93 kW, and this is before taking into account the numerous other barriers to efficiency, such as heat losses, solids heating and utility system losses.

Moreover, the assumed minimum temperature difference  $\Delta T_{\min}$  of 20 K between the hot and cold streams is very optimistic for gas-to-gas heat exchange. For higher  $\Delta T_{\min}$ , the curves are pushed laterally apart, the overlap is reduced and the heat recovery falls, becoming zero for  $\Delta T_{\min} > 80$  K. Krokida and Bisharat (2004) presented a detailed analysis of heat recovery from dryer exhaust air in a form suitable for computation, including the effect of heat pumping.

For our worked example, let us assume that we do not wish to bring the exhaust gas below its dewpoint, to avoid condensation. For the base case, where  $T_{g,\text{out}} = 65$  °C and  $Y_{\text{out}} = 0.034$  kg kg<sup>-1</sup>, the psychrometric chart shows that the dewpoint is about 34 °C. However, this would give a  $\Delta T_{\min}$  of only 14 K. Hence, taking  $\Delta T_{\min} = 20$  K instead, the inlet air can be warmed from 20 to about 45 °C, and the heater duty becomes  $Q_{\text{heater}} = 3.69 \times (150 - 45) = 387$  kW, a saving of 92 kW or 19%. Again, if higher  $\Delta T_{\min}$  is used, heat recovery falls and becomes zero at  $\Delta T_{\min} = 45$  K.

A number of schemes for heat recovery from dryer exhaust gases have been installed and some were supported by the UK Government as demonstration projects in the 1980s and 1990s. However, the results were disappointing. The wet, dusty nature of dryer exhaust streams led to severe fouling problems on heat exchanger tubes (Kaiser *et al.*, 2002), and sometimes corrosion. Special units such as glass tube heat exchangers were tried, but the capital cost was high, while the relatively small proportion of heat being recovered meant that savings were only modest. When the price of energy fell from its early 1980s peak, plans for further schemes were discreetly abandoned.

An alternative heat recovery method is exhaust air recycle. However, to prevent build-up of humidity in the circuit, most of the water vapor must first be condensed out. Typically this reduces the air temperature to below 40 °C, which gives very little gain compared to once-through heating with ambient air, as can be seen from the temperature–heat load profile in Table 1.7 and Fig. 1.7. If, however, it is feasible to expel the air from the dryer and create an atmosphere of water vapor, as in “airless drying” systems, the vapor can be condensed at atmospheric pressure at 100 °C and the losses due to the heated air in the dryer exhaust also disappear.

### 1.5.3.2 Heat Exchange with Other Processes

There are two possibilities; either heat from below the dryer pinch can be used to heat other processes, or heat rejected from other processes can be used to heat the dryer. The GCC (Fig. 1.8) shows what heat can be recovered to or from an external process at any given temperature or range. Unfortunately, net heat sources well above 100 °C or sinks below 30 °C would be needed, and few industrial sites meet these criteria. For atmospheric pressure (and vacuum) dryers, the latent heat in the dryer exhaust is released at too low a temperature to be useful, as the dewpoint will be well below 100 °C.

Superheated steam dryers are significantly different. The exhaust will condense at or above 100 °C, at the saturation temperature corresponding to the applied pressure, and this may be hot enough to supply other moderate temperature processes on site. The economics of a superheated steam drying system may stand or fall on the ability to use this heat effectively. In a closed-loop system, the steam is recirculated and recompressed, and the vapor removed by drying is purged as steam, with a heat load corresponding to the latent heat of evaporation. If this steam can be condensed for process heating duties, replacing steam from standalone boilers, a major operating cost saving is achieved.

Conversely, waste heat from other site processes is rarely hot enough to supply conventional dryers. A special low-temperature dryer could be used, but the low driving forces will push up the required size, and hence the capital cost. Opportunities might arise where the dryer inherently has to work at low temperatures, for example, agricultural crop dryers, vacuum dryers heated by hot water, or even low-pressure superheated steam dryers (Devahastin *et al.*, 2004). However, it is rare to find agricultural dryers on the same site as high-temperature industrial processes, while low-temperature vacuum dryers tend to be batch dryers in intermittent use and with low energy consumption.

The remaining possibility is that, on a site with other furnaces or boilers present, it may be possible to use heat from the flue gases to heat the dryer. Often this is more effectively done as part of a CHP scheme, as described below. Energy recovery from incinerators is also possible.

A point to beware of when integrating between different processes (or between two sub-sections of the same process) is that heat recovery is only possible when both processes are operational. There can be some loss of flexibility. If one process has to be shut down, the other must also be shut down, or alternative utility heating or cooling must be supplied. The same can apply at process start-up, even with internal heat recovery within the process; until the exhaust streams have reached operating temperature, heat cannot be recovered from them to the inlet streams.

## 1.5.4

### Alternative Utility Supply Systems

The objective here is not to reduce dryer energy consumption directly, but to supply the heat load using low-cost utilities or by thermodynamically efficient methods that reduce primary energy consumption.



#### 1.5.4.1 Low Cost utilities

Providing heat at a high temperature may be significantly more expensive than at a lower temperature. For example, in a steam system, higher temperature levels require much higher pressures, and hence much more expensive boilers and pipelines. A hot water recirculating system may obviate the need for steam altogether for low-temperature heat duties: or it may be possible to utilize solar heating. The grand composite curve evaluates these opportunities. The utility heat must be supplied at or above the temperature of the cold stream it is heating (allowing for necessary temperature differences in heaters etc.). In Fig. 1.12, we see that it is possible to supply more than half the total heat duty (72 kW) at a temperature of 160 °C (6 bar steam) instead of 210 °C (20 bar).

To utilize lower temperature utilities, lower driving forces must be tolerated, requiring larger heat exchangers. Also, a low-temperature drying process such as solar heating, may need different processing methods from a higher-temperature one; for example, drying times will normally need to be lengthened, often substantially.

Renewable sources and waste products should be considered as fuel options. Where security of supply is insufficient, or the calorific value is too low, dual fuel boilers can be used, supplementing fossil fuels by alternative fuels. Emission levels need to be investigated, particularly odors from waste-derived fuel. When considering alternative fuels such as biofuels, sustainability and land use should be carefully checked. It would be inappropriate to cause irreversible deforestation by excessive timber burning, or to use a biofuel which is grown on land desperately needed for local food production. On the other hand, an unwanted by-product or waste product of that food production process, such as rice husk, makes a highly appropriate and sustainable fuel.

#### 1.5.4.2 Improving Energy Supply System Efficiency

Section 1.3.3.2 demonstrates many of the changes to heat supply systems which can make them more efficient. For steam boilers, efficiency is maximized by use of flue gas heat recovery and economizers, boiler feedwater heating and condensate return. Steam distribution systems should be well maintained, steam leaks promptly repaired, steam traps monitored for leakage, “dead legs” closed off promptly, and local meters installed and recalibrated regularly. Direct fired burners eliminate all costs associated with steam raising and distribution, and should be considered where the product quality is not impaired by coming into direct contact with exhaust gases. Modern burners, giving low emissions of nitrogen oxides (NO<sub>x</sub>) and sulfur oxides (SO<sub>x</sub>), should be used; retrofitting of older burners can be considered. Localized boilers are another way to eliminate steam mains, as are localized CHP generators, covered in the next section.

Cold utilities are usually cheaper than hot utilities, but note that low-temperature refrigeration loops are very expensive and the circulating temperature should therefore be maximized, and cooling water used where temperatures allow, in preference to chilled water or refrigeration.

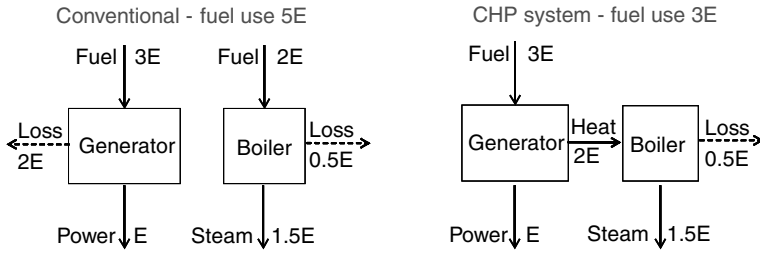


Fig. 1.13 Schematic representation of a typical CHP (combined heat and power) system.

#### 1.5.4.3 Combined Heat and Power

The dryer heat requirement is supplied in conjunction with power generation in a heat engine. This power can be used on the site or exported to the electricity supply grid. In effect, the power is generated at a marginal efficiency of nearly 100%, compared with 40% in a typical stand-alone power station. The principle is shown schematically in Fig. 1.13. The heat that would otherwise be rejected through the stack is usefully used in the process. In effect, instead of using 3E kW of fuel to generate power  $E$ , we have only used  $E$ , so the power has been generated at 100% marginal efficiency!

In practice, this is an oversimplification. Usually, the heat must be supplied from a CHP system at a higher temperature than it is rejected at in a stand-alone power station, so that power typically falls from 40 to 33% of fuel energy and heat rises from 60 to 67%. Nevertheless, the reduced power generation (or increased fuel to generate the same power) is far outweighed by the value of energy recovered from the exhaust heat (plus the substantial reduction in environmental emissions compared with separate systems). A very clear example of this is shown in the case study, in Section 1.6.3.

CHP is increasingly used to heat buildings via district heating schemes, but industrial applications require the heat to be generated at much higher temperatures. Table 1.9 lists the three main forms of industrial CHP.

Steam turbines are the earliest form of CHP system, and have been extensively used for decades in pulp and paper mills. However, they give a relatively low power output for a given heat production. Hence, they have been used infrequently in recent installations. They are usually large-scale systems, providing several megawatts of heat from the low-pressure steam and more than 1 MW of electricity. The

Tab. 1.9 Types of combined heat and power system.

CHP System	Scale	Power/Heat Ratio	Heat/Power Ratio	Main Heating Range (°C)
Steam turbines	Large	<0.2	>5	100–200
Gas turbines (natural gas or fuel oil)	Large	0.67–0.2	1.5–5	100–400
Diesel or gas reciprocating engines	Small	1.25–0.5	0.8–2	100–300, <80

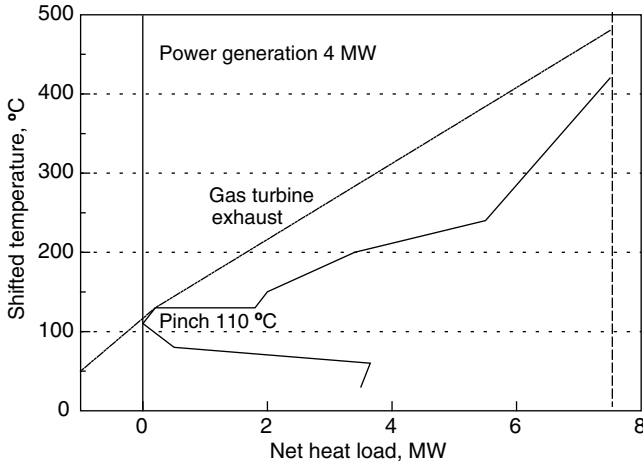


Fig. 1.14 Gas turbine CHP system fitted to a grand composite curve.

temperature and pressure of the steam levels can be optimized by using the GCC, as in Fig. 1.12.

Gas turbines are often the most effective form of CHP; they produce a good proportion of power, plus high-grade heat in the form of hot exhaust gas, which can be used to heat a dryer directly, or to raise steam for indirect heating. Again, the required heat load can be found from the GCC; the hot exhaust gases plot as a sloping line, as in Fig. 1.12. An excellent example is the 1982 installation by Scottish Grain Distillers at Port Dundas, Glasgow. Here a gas turbine generates 3.5 MW of power, sufficient for all the site needs, and the hot exhaust gases at approximately 450 °C are fed directly to a pneumatic conveying dryer, which dries the spent grain residues from the fermentation to produce animal feed. The grand composite curve for a similar system is shown in Fig. 1.14. Here, 4 MW of power and 7.5 MW of useful above-pinch heat is being generated, while about 2 MW of above-ambient exhaust heat is rejected as it is low-grade heat below the pinch and, therefore, useless.

Gas and diesel engines also produce a high power output, but less high-grade heat than gas turbines, with a large amount of waste heat at around 80 °C from the jacket cooling water. This is often useless for process duties, especially if the pinch is around 100 °C as the heat is released below the pinch. However, if there are nearby offices or warehouses, the jacket heat can provide central heating and domestic hot water to them. Conversely, if the pinch is at ambient, there must be some low-grade process heating duties, which can thus be fulfilled by the jacket water.

Scale is another factor; gas turbines are mainly used for large installations (2 MW power production and above, generating at least 3 MW heat) whereas gas engines are usually substantially smaller than 1 MW power output, giving heat loads well under 500 kW. An interesting small-scale example is a sewage sludge dryer developed around 2000 by NMA (Netherlands) with an integral gas engine; the power generated is used for the agitator drive, overcoming the high resistance of the sticky sludge in the early stages, and the exhaust heat is fed into the dryer for final thermal drying.

So, the big question; which CHP system is most appropriate for a dryer? As usual, the answer is, “It depends”. Key factors are the required temperature range, the shape of the GCC, the site power-to-heat ratio and the total power and heat requirements. For most small- and medium-sized chemical and process plants, gas engines are more appropriately sized than steam turbines or gas turbines. More than one gas engine is often used in parallel, to match site heat demand and give operating flexibility (see the case study, Section 1.6.3).

The relative costs of heat and power are also very important; obviously CHP schemes are more worthwhile in countries where power is mainly provided from fossil fuels and is relatively expensive, and are unattractive where cheap power sources such as hydro-electricity are available.

#### 1.5.4.4 Heat Pumps

A heat pump can recover heat from the exhaust gas to heat the dryer. In thermodynamic terms, this needs to work backwards across the pinch, recovering useless below-pinch waste heat for duties above the pinch. Again, the GCC shows how much heat can be upgraded at any temperature. The coefficient of performance (COP) is the heat upgraded per unit power supplied, and falls as the temperature lift increases. In most countries, electrical power is generated from fossil fuels in power stations, with an efficiency of 30–40%. Hence, a COP of 3 is required just to break even in primary energy terms, and to give some economic return for the cost of the heat pump, the COP needs to be at least 5 and preferably nearer 10. This will only be possible where the temperature lift is very low. Good opportunities are agricultural dryers and grain stores where exhaust air is recycled and reheated; drying is very slow, the heating duty is low and the temperature difference between exhaust and inlet air is only a few degrees. On the GCC, this would show up as a “sharp” pinch. In contrast, our example dryer in Fig. 1.8, where a temperature lift of over 100 °C is required to upgrade any significant amount of heat, is totally unsuitable for heat pumping. Inappropriate placement of a heat pump destroys any potential energy savings; Sosle *et al.* (2003) noted that a heat pump dryer for apple, although giving good product quality, had a higher energy consumption than an equivalent hot air dryer because the heat lost in the secondary condenser could not be usefully recovered. However, Krokida and Bisharat (2004) suggest that a heat pump can still be economic, even over a fairly wide pinch region.

In countries where electrical power is available cheaply, for example, from hydro-electric sources, the economics of heat pump systems are substantially better. Lower COPs can be tolerated and the use of heat pumps on industrial dryers becomes economically feasible in some more cases.

The analysis above is for conventional closed-cycle heat pumps taking in a given amount of exhaust heat and releasing a slightly greater amount at a higher temperature. There can be considerable differences for other systems, for example, open-cycle (mechanical or thermal vapor recompression), absorption heat pumps and heat transformers/splitters. In all cases, however, the temperature/heat load profiles of the heating and cooling stages should be compared with the process GCC to ensure that the most appropriate system is selected and that it is working over the optimum range of temperature and heat load.

## 1.6

### Case Study

A case study on a food processing plant gives an example to compare different dryer types, both in terms of energy performance and other operational and economic parameters.

#### 1.6.1

##### Process Description and Dryer Options

The plant is producing a dry final product with approximately 2% final moisture content from an initial solution with about 30% dissolved solids. The upstream process is of interest; in the early stages of evaporation, multiple-effect evaporators and either thermal or mechanical vapor recompression are used to reduce the energy consumption to a small fraction of the total evaporation latent heat load. As the concentration increases, viscosity rises and the solution also shows a substantial boiling-point rise (vapor temperature lower than solution temperature), reducing temperature driving forces so that multiple effects can no longer be used. Finally, a paste is formed and this must be dried from about 20% moisture wet basis, 25% dry basis. There are three technology options for this:

- Batch vacuum tray (oven) dryers
- Continuous vacuum band (belt) dryers
- Continuous spray dryer (working from a more dilute initial solution).

Vacuum tray dryers are the original historic process. They are simple to construct and operate, and operate at an absolute pressure of 70–100 mbar, which can be achieved by liquid ring vacuum pumps. However, they are of relatively small capacity (about 150 kg) and the manual loading and unloading is highly labor-intensive. Drying cycle time is 1 h and loading adds 20 min to the cycle. Hence up to 20 ovens in parallel are required to give a production rate of  $2 \text{ t h}^{-1}$ , as the effective throughput of each oven is little more than  $100 \text{ kg h}^{-1}$ .

Vacuum band dryers are the continuous equivalent of the batch tray dryers. They have a larger throughput, and feed and discharge are automatic, so that labor requirements are far lower. They can be operated under similar conditions of temperature and vacuum to the batch dryers, giving a drying time of about 50 min, as, unlike batch ovens, they do not need time for heating up or pulling and releasing vacuum at the start and end of each cycle. However, by pulling a higher vacuum, with absolute pressures down to about 30 mbar, falling-rate drying can be substantially accelerated and the drying time falls to about 30 min. The disadvantage is that to pull this level of vacuum, either a chilled water condenser (using power) or a steam ejector (using steam) is required, and the energy consumption of the latter is comparable to the latent heat load. Moreover, if the nozzles are worn, steam consumption becomes even higher with no gain in vacuum, so regular maintenance and replacement is necessary – yearly is recommended. Between 2 and 5 band dryers in parallel, depending on scale and vacuum conditions, could be used to meet a  $2 \text{ t h}^{-1}$  production requirement.

The third option is a spray dryer. This is a convective dryer working at atmospheric pressure, giving a significantly different temperature history from the vacuum dryers, so that product physical properties may be different. A single unit can easily achieve  $2 \text{ t h}^{-1}$ , or indeed far higher throughput. The feed must be pumpable and is, therefore, considerably more dilute than for the tray and band dryers, at approximately 50% solids. This reduces the need for evaporation equipment, saving capital cost, but the energy efficiency of the convective spray dryer is inherently lower than that of a comparable evaporator, even of single-stage type. For this product, tolerable inlet temperature is about  $180^\circ\text{C}$  and exhaust temperature  $100^\circ\text{C}$ , while ambient air supply averages about  $20^\circ\text{C}$  over the year. Some energy is lost in solids heating and post-drying rather than used for direct evaporation, so efficiency expressed in terms of evaporative heat load is around 40% rather than the 50% which might be expected from the temperatures. To produce  $2 \text{ t h}^{-1}$  of product, an evaporation rate of  $2 \text{ t h}^{-1}$  is required, giving an evaporative load of  $4000 \text{ MJ h}^{-1}$ . With a temperature drop of  $80^\circ\text{C}$  and air specific heat capacity of  $1 \text{ kJ kg}^{-1} \text{ K}^{-1}$ , this means that an airflow of no less than  $50 \text{ t h}^{-1}$  will be needed.

### 1.6.2

#### Analysis of Dryer Energy Consumption

For drying, the heat demand is significantly higher than the evaporation load, for several reasons:

- The solids need to be heated from their initial temperature to their final temperature in the dryer before discharge
- For batch ovens, the trays and ovens need to be reheated during each cycle, and some steam is supplied during loading and unloading
- For spray dryers, the whole of the airflow needs to be heated from ambient temperature to the final exhaust temperature

Measurements of steam consumption and comparison with calculated evaporation load reveal that:

- Vacuum tray ovens have typical efficiencies of about 30–40% for this product.
- Vacuum band dryers at the same conditions have higher efficiencies of 50–60%.
- Band dryers working at a higher vacuum have a substantial additional energy demand for the vacuum system.
- Spray dryer efficiency depends strongly on the inlet and exhaust temperatures; for this product, as noted above, the calculated efficiency is about 40%.

A pinch analysis shows that there is very little overlap between the composite curves (Fig. 1.15) and hence there is little opportunity for heat recovery within the plant or on the dryer. The dryer is working across the pinch, as shown by the grand composite curve (Fig. 1.16) and split GCC (Fig. 1.17) but it is not possible to alter the operating temperatures of either the dryer or other process units (notably the evaporators) to allow heat exchange between them. However, there may be opportunities to exchange heat with the utility system. In particular, below-pinch waste heat

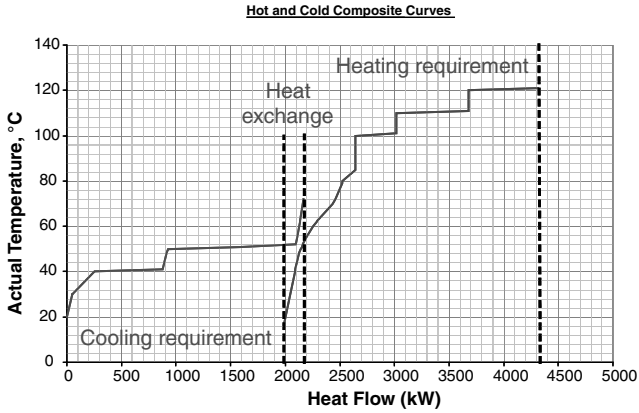


Fig. 1.15 Composite curves for a food process.

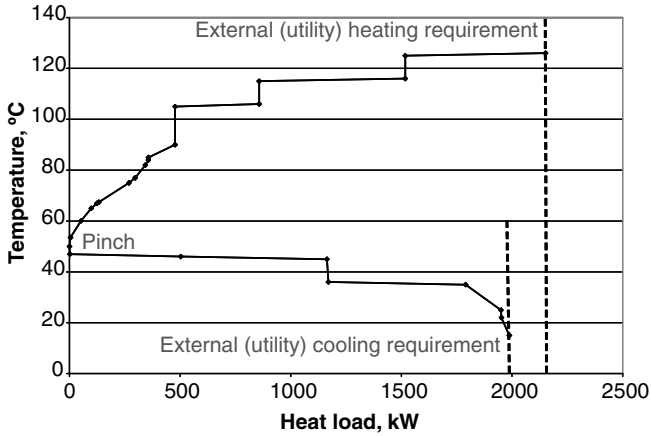


Fig. 1.16 Grand composite curve for a food process.

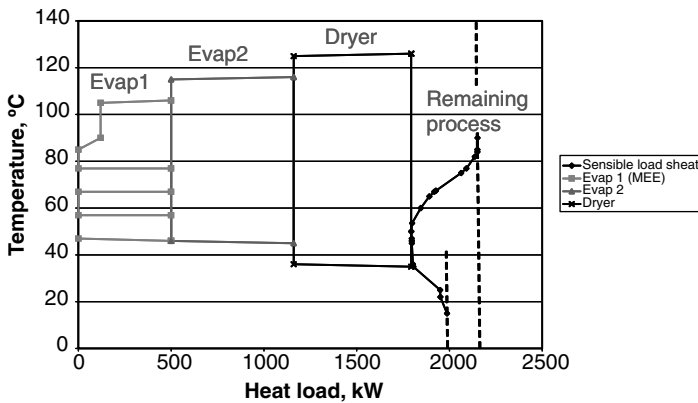


Fig. 1.17 Split grand composite curve for a food process.

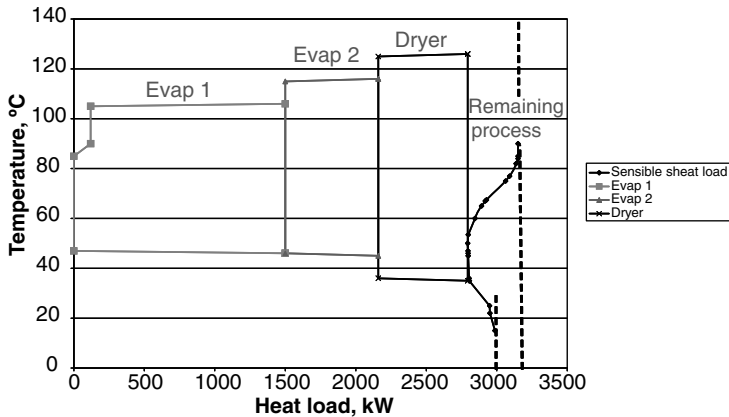


Fig. 1.18 Split GCC for a food process, with single-effect instead of multi-effect evaporator.

can potentially be used to preheat make-up water for the boiler feedwater system, as condensate return is well below 100%. The first-stage evaporators are multi-effect, which reduces their energy consumption substantially; compare Fig. 1.18 for the corresponding single-effect system, which shows that an extra 1000 kW of energy is required – a 45% penalty. As the solution is concentrated, the temperature driving forces are squeezed by the boiling point rise effect, where evaporated vapor is at a lower temperature than the solution. This, coupled with high viscosity, which reduces heat transfer coefficients, and the tight temperature limitations on the product, prevents the use of multiple effects in the second stage evaporators.

For the spray dryer, there is an opportunity to recover heat from the exhaust to preheat the inlet air. This can be done either with a direct air-to-air heat exchanger or by two separate exchangers with a circulating fluid to transfer heat, typically hot water. In both cases, it is desirable to use extended heat transfer surfaces to maximize the heat transfer coefficient. However, there is then a high risk of fouling for the dust-laden exhaust gas stream; either the heat transfer surfaces must be relatively plain for easy cleaning (giving a physically large exchanger) or the dust must be removed prior to the exchanger by a cyclone or bag filter, which incurs both extra capital cost and higher pressure drop. Exhaust heat recovery could bring exhaust temperature down from 100 to about 70 °C and improve efficiency from 40 to 55%.

Power use also needs to be considered. The power consumption of vacuum pumps is modest, and both primary energy use and costs are normally less than for steam ejectors. Spray dryers use substantial power for the fans, due to the high airflow (over 300 kW for the 2 t h<sup>-1</sup> unit here). Moreover, in countries with high humidity, the inlet airflow may need to be dehumidified, requiring chilled water, which in turn incurs a power cost for the refrigeration system.

Table 1.10 compares the alternative dryer systems and their energy usage, and Fig. 1.19 shows the breakdown of steam consumption; note that for the tray and band dryers, the actual dryer evaporation load is a very small proportion of the overall process steam consumption.



Tab. 1.10 Comparison of alternative dryer options for food processing plant.

Dryer Type	Vacuum Tray (Oven) Dryers	Vacuum Band Dryers	Band Dryers – High Vacuum	Spray Dryer, no Heat Recovery	Spray Dryer With Heat Recovery
Mode of operation	Batch	Continuous	Continuous	Continuous	Continuous
Mode of heating	Conduction	Conduction	Conduction	Convective	Convective
Initial solids (%)	80	80	80	50	50
Residence time (min)	80	50	30	1	1
Holdup/Capacity (kg)	140–170	300–500	300–500	30–40	30–40
Production rate (per unit) ( $\text{kg h}^{-1}$ )	100–130	400–600	660–1000	2000	2000
Number required	16–20	4–5	2–3	1	1
Vacuum equipment	Vacuum pump	Vacuum pump	Steam ejector	None	None
Pre-evaporator	Yes	Yes	Yes	No	No
Evaporator:					
Evaporation rate ( $\text{kg h}^{-1}$ )	1600	1600	1600	0	0
Efficiency (%)	88	88	88		
Steam consumption ( $\text{kg h}^{-1}$ )	1800	1800	1800		
Dryer:					
Evaporation rate ( $\text{kg h}^{-1}$ )	400	400	400	2000	2000
Efficiency (%)	33	50	50	40	55
Steam consumption ( $\text{kg h}^{-1}$ )	1200	800	800	5000	3700
Ejector steam use ( $\text{kg h}^{-1}$ )	0	0	800	0	0
Total steam use ( $\text{kg h}^{-1}$ )	3000	2600	3400	5000	3700
Steam use (kW)	1700	1500	1900	2800	2100
Electrical power use	Medium	Medium	Low	High	Very High
Capital cost	Medium	High	Low	Medium	High
Labor cost	High	Low	Low	Low	Low
Maintenance cost	Low	Low	High	Low	Low

Clearly, each dryer type has its advantages and disadvantages, and, for selection, the preference will be based on the relative importance of capital, energy and labor cost. In practice, the batch drying option is only economic in developing countries with very low labor costs, and can be eliminated from consideration elsewhere. It is noteworthy, however, that the heat energy consumption of the spray drying system is substantially higher than the vacuum dryers, and the main reason is that the spray dryer is much less efficient than the pre-evaporators for concentrating the liquid from 50 to 80% solids.

Power consumption can be a substantial factor whose importance can be overlooked. The fan power requirement of 300 kW is a further penalty for the spray dryer, but if this is expressed in terms of primary energy, it rises to about 900 kW, giving a

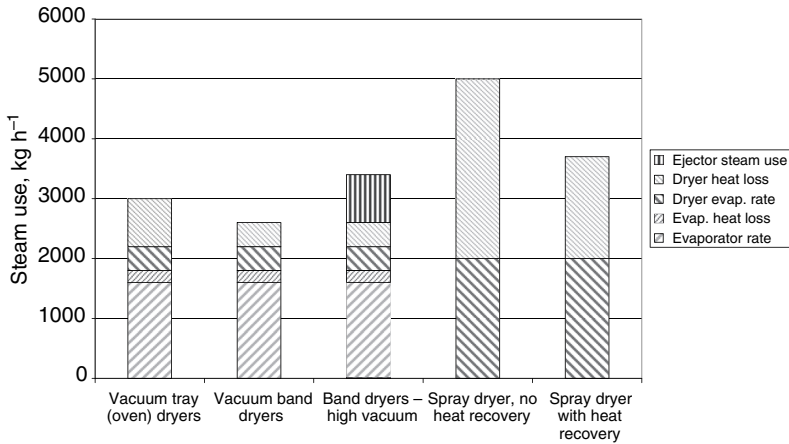


Fig. 1.19 Breakdown of steam consumption for alternative drying systems.

total energy use for the spray dryer of no less than 3700 kW. Although heat recovery from the exhaust lowers dryer steam consumption, there is an additional pressure drop through the heat exchanger and any additional dust collection equipment, giving an even higher fan power requirement. In addition, if inlet air dehumidification by chilled water is required during summer, particularly in tropical countries, the refrigeration power load can be several hundred kilowatts during those months.

Choice of the vacuum level for the band dryers also shows a clear capital-energy trade-off. In practice, careful temperature profiling in the dryer and combined pump/ejector systems can be used to optimize performance and reduce residence time while not incurring excessive steam costs. Hence, in practice, the preferred choice in normal circumstances will be one of the variants of vacuum band drying systems.

### 1.6.3

#### Utility Systems and CHP

Combined heat and power should always be considered as an option. The process is well suited for it; a gas engine is appropriately sized, the exhaust gas can be used to generate steam for the dryers and evaporators, and the jacket hot water can be used to heat parts of the upstream process (around 50–80 °C). The payback on a basic scheme is around 5–10 years, which is typical of CHP schemes in general.

However, for some sites in developing countries, the picture is very different. In some areas, rapidly increasing power demand has outstripped local generating capacity and power grid supply. For example, one site has random power outages for 25–30% of the day, and the site has five diesel generators which frequently need to be brought into action at a moment's notice. Because of the high cost of diesel fuel, this is an expensive method of power generation. Converting three of the diesel engines to gas or dual-fuel engines (cheaper than buying new gas-fueled generators), and installing heat recovery from the exhaust gases gives an excellent retrofit project

**Tab. 1.11** Energy use and carbon dioxide emissions with and without CHP.

	Original		With CHP	
	kW	t <sub>CO2</sub> y <sup>-1</sup>	kW	t <sub>CO2</sub> y <sup>-1</sup>
Natural gas use	1667	2700	3333	5400
Steam/hot water raised	1333		1333	
Losses to exhaust	333		1000	
Power generated	0		1000	
Power imported (70%)	700	3297	0	0
Power from diesel	300		0	
Diesel oil used	1021	1276	0	0
Total primary energy	4788		3333	
Total carbon dioxide		7273		5400

with a payback of barely 2 years, because of the large savings from eliminating diesel fuel. The primary energy and carbon footprint benefits are similarly considerable, as shown in Tab. 1.11. The figures are based on a power consumption of 1000 kW and to calculate annual emissions, a working year of 5000 h has been assumed.

For imported electricity, primary energy is taken to be 3 times the power use. Overall, the CHP system gives 30% savings in primary energy and 26% in carbon footprint. These figures would be similar even if all power was imported and none generated locally from diesel.

One factor which needs to be checked is whether the local gas supply infrastructure is able to supply the large gas flows required for a major CHP scheme. This has in the past ruled out some promising CHP schemes, even in developed countries.

Additional steam use is generated from coal-fired boilers. In some locations, alternative fuels such as rice husk can be considered as supplementary fuel. Also, the steam is generated and distributed at high pressure. Where a heating duty requires only low pressure steam, the steam can be let down through a small local steam turbine, instead of a let-down valve. Rather than generate a small amount of electrical power, requiring expensive alternators, the most cost-effective use of the shaft work is as direct drive to pumps and other process machinery on that process stage. Hence, if the stage is shut down and the steam is not required, the power load is also not needed and does not have to be separately supplied.

## 1.7

### Conclusions

Although dryers are intensive energy users, it is often difficult to find obvious major energy savings. Their energy use needs to be considered in the context of the overall process.

Pinch analysis of a typical dryer shows that heat recovery is severely limited by thermodynamic as well as economic considerations. Dryers usually have a high net heat demand above 100 °C; the main heat output is the latent heat of evaporation held in the vapor in the exhaust gas, which cannot usually be recovered except as low-grade waste heat. In a few specific situations, heat may be exchanged with a nearby process whose pinch is substantially above (or, more rarely, below) the dryer pinch temperature. Heat pumps are possible, but only in specific circumstances; there are often opportunities in agricultural drying, and solar heating is also an important possibility. In many industrial situations, the biggest opportunity comes from CHP (combined heat and power), particularly gas turbines on large sites and gas engines on small or medium ones. Reduction of the inlet moisture content to the dryer is the other major possibility, but the changes needed to achieve this must be made to the upstream process rather than the dryer itself.

Despite these difficulties, energy analysis of dryers is very worthwhile and can lead to major energy and cost savings. Key tools are the formation of a consistent heat and mass balance, and a systematic comparison between fuel use, steam generated and delivered to plants, and the energy actually required and consumed by the process itself.

#### Additional Notation Used in Chapter 1

$\Delta H_v$	specific enthalpy of evaporation	$\text{J kg}^{-1}$
$\Delta T_{\min}$	minimum temperature difference (at pinch)	K

#### Subscripts

burner	for a direct-fired burner
heater	for an indirect heater (heat exchanger)
latent	latent heat
loss	heat loss
sens	sensible heat
stream	for a stream (in pinch analysis)

#### Abbreviations

CHP	combined heat and power
COP	coefficient of performance
GBP	Great Britain pound
GCC	grand composite curve
RF	radio frequency
VOC	volatile organic compound

## References

- Bahu, R. E., 1991. Energy considerations in dryer design. *Proceedings of 7th International Drying Symposium (IDS'90)*, Prague, pp. 553–557.
- Devahastin, S., Suvarnakutura, P., Soponronnarit, S., Mujumdar, A. S., 2004. A comparative study of low-pressure superheated steam and vacuum drying of a heat-sensitive material. *Drying Technol.* **22**(8): 1845–1868.
- ESDU 1987–1990. ESDU Data Items 87030, 89001 and 90027; *Process integration and pinch technology*. Available by subscription from ESDU International plc, London, UK.
- Kaiser, S., Antonijevic, D., Tsotsas, E., 2002. Formation of fouling layers on a heat exchanger element exposed to warm, humid and solids-loaded air streams. *Exper. Therm. Fluid Sci.* **26**: 291–297.
- Kemp, I. C., 1991. Some aspects of the practical application of pinch technology methods. *Trans. Inst. Chem. Eng.* **69**(A6): 471–479.
- Kemp, I. C., 1996. Unpublished survey on energy use in industrial drying for ETSU (Energy Technology Support Unit), UK Department of Energy.
- Kemp, I. C., 2004. Drying in the context of the overall process. *Drying Technol.* **22**(1 & 2): 377–394.
- Kemp, I. C., 2005. Reducing dryer energy use by process integration and pinch analysis. *Drying Technol.* **23**(9–11): 2089–2104.
- Kemp, I. C., 2007. *Pinch analysis and process integration (User guide to process integration for the efficient use of energy)*, 2nd edn, Butterworth Heinemann, Elsevier, Oxford, UK and New York, USA.
- Kemp, I. C., Gardiner, S. P., 2001. An outline method for troubleshooting and problem-solving in dryers. *Drying Technol.* **19**(8): 1875–1890.
- Krokida, M. K., Bisharat, G. I., 2004. Heat recovery from dryer exhaust air. *Drying Technol.* **22**(7): 1661–1674.
- Linnhoff, B., Townsend, D. W., Boland, D., Hewitt, G. F., Thomas, B. E. A., Guy, A. R., Marsland, R. H., 1982. *User guide to process integration for the efficient use of energy*, 1st edn, Institution of Chemical Engineers, Rugby, UK. For 2nd edition see Kemp 2007.
- Smith, R., 2005. *Chemical process design and integration*, John Wiley & Sons.
- Sosle, V., Raghavan, G. S. V., Kittler, R., 2003. Low-temperature drying using a versatile heat pump dehumidifier. *Drying Technol.* **21**(3): 539–554.
- Stubbing, T. J., 1993. Airless drying: Its invention, method and application. *Trans. Inst. Chem. Eng.* **71**(A5): 488–495.
- Stubbing, T. J., 1999. Airless drying: Developments since IDS'94. *Drying Technol.* **17**(7&8): 1639–1651.
- van Deventer, H. C., 2002. Advanced drying concepts: Superheated steam and adsorption. Presentation at EFCE Drying Working Party meeting, University of Magdeburg.
- Wilmshurst, A., 1988. *Industrial energy use in drying with special reference to infra-red*. Diss., University of Cambridge, UK. Reported by Bahu (1991).

