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ENHANCING CONVECTIVE HEAT TRANSFER

IN ROTARY KILNS

by

G. HART

Presented for the Degree of Doctor of Philosophy at the University of Surrey, Department of Chemical and Process Engineering, Guildford, England.

December 1985
The Fuels and Energy Research Group at the University of Surrey (FERGUS) have been investigating the heat transfer and energy utilisation within rotary cement kilns since the early 1970's. One such project has been the study of chain systems in cement kilns. The work presented in this thesis is an investigation of a chain system for dry process cement kilns.

A review of the literature has highlighted that there is a general paucity of experimental work on cement kilns, and in particular, very little on the role of chain systems. This lack of applicable results is due to the fact that experimentation on this type of full-sized plant is exceedingly difficult.

In modelling the convective heat transfer in a chained dry process cement kiln the analysis of Gardeik and Jeschar (1979) has been applied. These authors have developed an expression which enables the regenerative nature of a chained rotary kiln wall to be quantified. The similarity criteria utilised to scale down a chain system is that of equal voidages between the model chainbank and its industrial counterpart. This modelling criteria was developed by Patterson (1980) through air and water modelling experiments on different chain systems.

In order to generate the basic heat transfer data a 1/8th scale, indirect fired, model of the chained section of Chinnor No. 1 Kiln of the Rugby Portland Cement p.l.c. has been constructed. Clean low momentum flue gas from a vortex combustor has been used to heat, countercurrently, raw meal cement nodules for a range of gas and solid flowrates at different kiln rotational speeds. As with most experimental rotary kilns working at elevated temperatures (400°C) many difficulties have had to be overcome, with a consequent alterations to the original design. The problem of continuous monitoring of temperature within the model kiln has been solved by the use of a programmable data recorder and a gold slip ring assembly external to the kiln. This system has simplified the acquisition, storage and
retrieval of temperature data from a rotating kiln. Experiments have been performed with the model kiln operating without and with a chain system to quantify the swirl produced by the combustor and the regenerative action of the chainbank, respectively.

The direct heat transfer coefficients between gas and wall have been determined as well as the overall heat transfer coefficient. Correlations of the data have been presented and where possible the results have been compared with the literature and to commonly used equations. For the model kiln operating with a chain system the gas to wall heat transfer has been correlated to yield:

\[ Nu = 0.00661 \ \text{Re}^{0.15} \ \text{Pr}^{0.33} \]

and the overall heat transfer coefficient by:

\[ K_{\text{eff}} = 0.002188 \ \text{Re}^{1.28} \]

Significant enhancement in heat transfer to a moving bed of solids has been achieved by using swirling flows in tandem with a chain system.
ACKNOWLEDGEMENTS

Mr. F.D. Moles (supervisor) for his support and encouragement over the last three years.

Dr. B.G. Jenkins and Dr. M.C. Patterson for their help in the initial stages of the project. Mr. D. Arnall for the development of the multiplexer system and Dr. A. Tate for his computing expertise and original design of the vortex combustor.

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Mrs. C.W. Milton for the typing of this thesis.

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1 THE CEMENT INDUSTRY

1.1 Introduction

The cement industry is one of the world's most energy intensive industries, being affected to a great extent by fluctuations in the energy price market.

The idea of "conservation" is very much a late twentieth century phenomenon. Until 1973, the cement industry was more concerned with labour costs than the price of energy. With the advent of the 1973 Oil Crisis this situation had changed dramatically. The sharp increase in fuel prices caused energy costs to be the dominant factor in the manufacture of cement. Energy conservation had become a global issue.

During the early 1970's, Mr. F.D. Moles of the Fuels and Energy Research Group of the University of Surrey (F.E.R.G.U.S.) had proposed an intensive research program for the cement industry, the ultimate objective being a reduction in fuel consumption by the industry.

This total research program had highlighted that the aerodynamics and flame "control" within the rotary cement kiln were the most significant factors related to increasing the fuel efficiency of the process. Other studies pursued by FERGUS included the function of chains in wet process kilns (Patterson, 1980), the aerodynamics and heat transfer within grate coolers (Manuelpillai, 1982) and the acid/alkali flow visualization studies in both Lepol Grates (Costen, 1983) and Suspension Preheater Kilns (Tosunoglu, 1984).

1.2 The Manufacture of Portland Cement

Cement can be manufactured by two main routes, wet or dry, depending on the level of moisture in the raw materials. Two types of raw materials are required: one, rich in calcium, such as limestone, chalk or marl, the other rich in silica, such as clay or shale. Developed from these routes are two more processes, semi-wet and semi-dry (Figure 1.1).
Figure 1.1 Types of Cement Kiln and Cooler used in the Manufacture of Portland Cement (after Patterson (1980)).
Nearly all cement clinker is produced in huge rotary kiln systems. The rotary kiln is a high temperature refractory-lined, cylindrical steel shell (3 - 8 m dia., 50 - 230 m long), equipped with an electrical drive which rotates it at 1 - 3 rpm. It is a countercurrent heating device slightly inclined to the horizontal, so that material feed into the upper end travels slowly by gravity to be discharged onto the clinker cooler at the firing end. The burner at the firing end produces a current of hot gases which heats the clinker and the calcined and raw materials in succession as it passed upward toward the feed end.

Within a rotary cement kiln there are three distinct temperature zones. At the feed end of the kiln the material initially encounters gas temperatures of up to 300°C, this is the drying/preheating zone, where physical removal of moisture is accomplished. On moving to the next zone, at about 900°C, carbon dioxide is evolved from the material. This reaction is endothermic and is termed calcination. A partial phase change, referred to as sintering, occurs in the hottest part of the kiln at around 1450°C. It is in this zone that the most important cement making reaction, from the quality viewpoint, takes place, and is the exothermic formation of tricalcium silicate. The final sintering reaction results in the formation of partly fused nodules known as clinker. The clinker nodules are discharged from the kiln onto an appropriate clinker cooler. After exiting the cooler the material is then mixed with about 5% of gypsum and the product ground to fine grey powder.

The following sections briefly outline the various methods of cement manufacture, and attempt to describe the impressions gained by the cement industry on the role of chains in cement kilns.

1.2.1 The Wet Process

In a long wet-process kiln, the slurry introduced by the feed undergoes simultaneous heating and drying. The refractory lining is alternately heated by the gases when exposed and cooled by the slurry when immersed; thus the lining serves to transfer heat as do the gases themselves. Because large quantities of water (about 0.8 l/kg of clinker produced) must be evaporated, most wet kilns are equipped with chains to maximize heat transfer from the gases to the slurry. Large,
dense chain systems have permitted energy savings of up to 1.7 MJ/kg clinker produced in exceptionally favourable conditions (Anon, 1975). The chain system also serves to break up the slurry into nodules that can flow readily down the kiln without forming mud rings.

1.2.2 The Dry Process

The dry process utilizes a dry kiln feed rather than a slurry. Early dry process kilns were long, and the substantial quantity of waste heat in the exit gases from such kilns was frequently used in boilers for electric power generation. In some cases, the kiln has been lengthened to nearly the extent of long wet-process kilns, and chains have been added. Refractory heat-recuperative devices, such as crosses, lifters and trefoils were also installed (Duda, 1977). Thus equipped, the long dry kiln is capable of good energy efficiency. Other than the need for evaporation of water, its operation is similar to that of a long wet kiln.

The most recent development in cement manufacture is the dry process "Suspension Preheater". In this system intimate contact of hot kiln exhaust gases and finely ground raw material occurs within a series of cyclones, before entering a shortened rotary kiln. In the lowermost cyclones an extra source of heat is added, either by the direct injection of fuel or the installation of independently controllable burners. This modification enhances the evolution of CO$_2$, and precalcining takes place prior to the material entering the rotary kiln.

1.3 Kiln Chain Systems

Kiln chain system designs are based upon empirical and comparative engineering. The idea of a universal optimum chain system is not feasible, because the effectiveness of a chain system will depend on the raw material characteristics and their moisture content, as well as the proper operation of the kiln. (Debus and Nerzynski, 1966).

There are essentially two different types of chain systems (Figure 1.2). These are the chain curtain and the chain garland system. In the curtain system one end of the chain is allowed to hang free, the other being attached to the kiln circumference. The hangers to which
the chains are attached may be in parallel rows perpendicular to the kiln axis or arranged in a helical path along the axis. The former is termed "cross-sectional suspension", and the latter "helical suspension". In the garland chain system both ends of the chain are attached to the kiln shell, the exact point of attachment depending on the required "sweep" of the chain. As for the curtain system both the cross-sectional and helical suspensions are used in practice.

Figure 1.2 Kiln Chain Systems (After Duda [1977]).

1.3.1 Chaining Practice

In the wet process kiln the chain system can be divided into three distinct zones, corresponding to the physical state of the material in each zone, i.e. slurry, plastic and granular. The method of chain suspension as well as the type of chain used should be designed specifically for each of these zones.

A dry process system essentially consists of a single zone, i.e. granular. As no drying zone is required, the exit gas temperature will be about 800°F, compared with 400°F for a wet process kiln. This results in higher gas velocities, which in turn create higher dust carry-over. Traditionally, the dry process route has used curtain hung chains. Current practice is to use a mixture of garland and curtain chain systems. The garland section acting as a dam, this obstruction to material flow creates a higher material bed which allows the chains to transfer more heat.
Site experiments with mixed chain systems have been reported in the literature by Bhatia (1979) and Ansari and El-Hossary (1982). In the latter of the above two reports fuel consumption was reduced. This lead to a 19% reduction in the velocity of gases through the kiln, this factor in turn causing a drop in dust losses from the kiln. Bonifay (1979) also reported the benefits of using a mixed chain system, these results being a reduction of 16% in fuel consumption, an increase in production of 13% and dust losses reduced by 57%.

1.3.2 Link Design and Chain Life

The chain link configuration has a considerable influence on the success of a kiln chain system. Robinson and Fuller (1973) comment on link design and have highlighted the following four key points:

(a) The links must be circular in shape to enhance even wear and self cleaning.

(b) A round cross-section is most desirable because it yields a higher surface area to weight ratio and has demonstrated the greatest ability to resist material buildup.

(c) As the diameter of a round cross-section link increases, strength increases but the surface area to weight ratio decreases.

(d) The most popular link size is 3/4" x 3" i.d. round link kiln chain.

Feiser (1964) also concluded that the ring pattern provided longer life and less down time in kiln operation.

The service life of chains in temperature zones of 550°C and above are not determined primarily by wear, but by their resistance to scaling. Chains located nearer the feed end of the kiln, usually are replaced due to wear rather than scaling. In the hot zones reactions between the gases and the chain lead to corrosion of the chain, the diameter of the chain link then decreases, the chain surface area to kiln internal volume also decreasing, these factors resulting in a reduction of heat transfer between gases, chains and the charge.
1.4 Scope of the Work

If the efficiency of the cement making process is to be optimized, then there is a need to increase the overall heat transfer by utilizing all the available convective energy to the greatest extent in the smallest possible volume.

Costen (1983) found employing a vortex combustion chamber increased the process efficiency of Lepol grate kilns. His work utilized a natural gas fired vortex combuster to provide clean low momentum flue gas. It has long been established that swirling the flow will improve heat transfer in duct flow. This particular project aims to quantify the degree of enhancement in heat transfer by the use of swirling flows passing through a 1/8th scale model of the chained section of Chinnor No. 1 kiln.

The aim of this work is to minimise the kiln back-end gas enthalpy losses. Traditionally, this waste heat gas has always been recovered external to the kiln.
2 LITERATURE SURVEY

2.1 Heat Transfer in Rotary Kilns

2.1.1 Convective Heat Transfer

Few experimental studies of heat flow in rotary kilns have been reported in the literature. This lack of information is related to the fact that making measurements on a rotating vessel is exceedingly difficult, especially under industrial conditions.

The heat exchange from gas to material in a rotary kiln takes place through several parallel heat transfer paths, in contrast to a standard counterflow heat exchanger in which the heat flow from hot to cold fluid passes through a number of resistances in series.

With the introduction of chains in a rotary kiln the system of heat flow paths becomes far more complicated. There is no reported information with the exception of Patterson (1980) concerning experimental investigations on chained kilns. Patterson highlights the heat flow paths as depicted in figure 2.1.

![Figure 2.1 Heat flow paths in a chained rotary kiln with its electrical resistive analogue.](image)
where the heat flow paths are

- $Q_1 = \text{gas to chain}$
- $Q_2 = \text{solids mixing: conduction from chains to solids}$
- $Q_3 = \text{gas to wall}$
- $Q_4 = \text{solids mixing: conduction from solids to wall}$
- $Q_5 = \text{gas to solids}$
- $Q_L = \text{shell losses}$

One of the first practical investigations of heat transfer in rotary kilns was undertaken by Gygi (1936) on a wet process cement kiln. Gygi assumed the temperature distribution of the rotating wall in a circumferential direction. Then the longitudinal wall and material temperature profiles were obtained from heat balances and experimental measurement of average gas temperatures. The material temperature was assumed uniform at any given cross-section.

Heiligenstaedt (1951) departed from this procedure in that he attempted to calculate the circumferential wall temperature distribution instead of assuming it. He replaced the cylinder by a slab insulated on one face and alternately heated by the gas and cooled by the well mixed charge on the other face.

Folliot's (1955) investigation on an experimental rotary kiln ($L = 45\text{m}$, $d = 2.75\text{m}$) and subsequent analysis of his measurements did not agree with the findings of Gygi (1936). Folliot attributed this discrepancy to gas radiation, due to the dust content of the gas stream. Folliot demonstrated that at a gas temperature of $1100^\circ\text{C}$ the measured value of the gas emissivity is 0.83, whereas the calculated value is only 0.26. Folliot also emphasized the importance of solids mixing which he termed 'solids convection'. He stated that the heat flow to the bed is a two stage process, involving heat transfer to a hot, thin layer of particles at the surface followed by mixing of this layer into the bulk of the bed. Bowers and Read (1965) reached a similar conclusion. They reviewed the industrial literature specifically for processes that utilized rotary kilns. The data that they obtained from limestone calcination, dry process cement manufacture, dolomite calcination and shale expansion were used to calculate overall heat fluxes. They reported that one of the factors influencing the heat flux was the
particle size distribution. From the processes reviewed, they found that the limestone process had a higher heat transfer rate than the dry cement process, both kilns having similar dimensions. They accounted for the difference by noting that the limestone process used a much larger feed than the dry cement process. They considered the bed to be a slab and they cited the following equation:

\[ \varepsilon = \frac{F_0 \cdot \frac{r^2 \cdot \rho \cdot c}{k_e}} \]

where

- \( \varepsilon \) = time
- \( F_0 \) = relative time ratio, Fourier number
- \( r \) = one-half the average thickness of kiln bed
- \( \rho \) = solids bulk density
- \( k_e \) = bed thermal conductivity due to conduction and radiation between particles
- \( c \) = solids heat capacity

This relationship gave a relative measure of kiln bed thermal acceptance or the ability of the bed to absorb the heat from the hot kiln gases and kiln wall. The greater the thermal acceptance the shorter the heating time \( \varepsilon \). The finer the bed, the lower the kiln bed thermal acceptance. Coarser particles were stated to enhance the thermal acceptance by virtue of their ability to mix with other particles.

Imber and Paschkis (1960, 1962) provided one of the first rigorous analysis of heat transfer in rotary kilns. Their work is noteworthy in its attempt to present an analytical procedure where both the circumferential temperature distribution is calculated not assumed and the longitudinal temperature distribution is calculated by the use of a classic heat exchange analysis. They considered the hot gases and the material charge as the hot and cold fluids, respectively, of a counterflow heat exchanger and developed a kiln equation that expresses average gas and material temperatures versus kiln length. These authors although aware of more heat flow paths only considered the following three:
(i)  gas to charge  \((gc)\)

(ii) gas to wall \((gw)\)

(iii) wall to charge \((wc)\)

The last heat flow, path (iii) is essentially regenerative in nature and was analysed later by Gorog et al (1982). Imber and Paschkis, assuming no variation of heat transfer coefficient with temperature or position along the kiln length, proceeded to:

(i) calculate the circumferential temperature distribution of a thin rotating ring in terms of gas and material temperatures, assuming the gas and charge each to be well mixed,

(ii) derive a kiln equation relating gas and material temperature to kiln length for a short section of the kiln,

(iii) extend (1962) the investigation to the material bed temperature, only, by assuming that the temperature on the surfaces of an unmixed charge are those of the gas and average wall temperatures found for the perfectly mixed charge condition. This latter two-dimensional transient boundary value problem with a simplified bed configuration (rectangular) was solved to show that the mid-temperature of bed depth is substantially lower than the prescribed surface temperature.

Imber and Paschkis (1962) conclude by stating that their results agreed well with best available interpretation of Gilbert's values (1936). In their comparison Imber and Paschkis (1960) took the gas to wall heat transfer coefficient to be 85 W/m² °C and the wall to charge to be five times this value i.e. 425 W/m² °C. They obtained overall heat transfer coefficients utilizing Gilbert's 1936 data on slurry driers of about 75 W/m² °C in the preheating zone and about 115-153 W/m² °C in the firing zone.

Vaillant (1965) provides the most detailed analysis of the heat transfer in a direct fired kiln. The analytical equations developed by Vaillant were essentially based on the fundamental treatise of Carslaw and Jaeger (1959). Vaillant used a passive network analogue computer to solve some of the equations that were derived. Vaillant reports no experimental work on direct fired kilns, but does introduce the concept
of a partially mixed charge. He also comments on the work of Imber and Paschkis in that their calculations on a thin walled kiln seem better suited to the analysis of indirect fired kilns, rather than direct fired kilns since wall temperatures in the radial direction are uniform for the former but not for the latter.

Saas (1957) has provided a simplified model of the heat transfer phenomena in a rotary kiln. Five heat transfer paths were identified and a corresponding set of heat transfer equations formed in which five modified heat transfer coefficients are used ($\alpha_i$), each defined as:

$$\alpha_i = h_i A_i$$

where $h_i = \text{heat transfer coefficient}$

$A_i = \text{heat transfer surface per unit length of kiln}$

The relationships used are summarised in Table 2.1. The predicted temperature profiles made by Saas were in good agreement with the temperature profile data of Gilbert. Even so doubts about the validity of some of these relationships, given in Table 2.1, have been expressed (Kaiser and Lane, 1968). These reservations include the recommendation that the term $fE_g$ in $h_3 Rad$ should be replaced by the expression of Eckert and Drake (1959):

$$\frac{1}{E_s} + \frac{A_3 \text{ Rad}}{D_1} \left[ \frac{1}{E_w} - 1 \right]^{-1}$$

A similar set of equations have been employed by Riffaud et al (1972) in their modelling studies of an alumina kiln.

Wingfield et al (1974) and Pearce (1973) used the method of heat balances in stages to calculate the temperature distribution in a rotary kiln for reducing iron ore, and, for dead burning magnesia, dolomite and limestone respectively. Pearce obtained a value for the overall heat transfer coefficient of about 90-170 W/m² °C for dead burning of magnesia, about 250-530 W/m² °C for dead burning of dolomite and around 550-770 W/m² °C for burning limestone and dolomite.
TABLE 2.1 Heat-transfer coefficient correlation after Saas (1967)

<table>
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<tr>
<th>Heat-transfer path</th>
<th>( h_i ) \text{ Btu/h ( \circ )F ( ft^2 )}</th>
<th>( A_i ) \text{ ft}^2/\text{ft}</th>
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<td>Gas to inner kiln wall</td>
<td>( h_1 = 0.05(G_g/S_x)^{0.67} + 0.173 \times 10^{-8} ) ( \frac{E_g(T_g^4 - T_w^4)}{(T_g - T_w)} )</td>
<td>( A_1 = \pi[1 - (\theta/360)]D_i )</td>
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<td>Gas to solid</td>
<td>( h_2 = 0.05(G_g/S_x)^{0.67} + 0.173 \times 10^{-8} ) ( \frac{E_g(T_g^4 - T_o^4)}{(T_g - T_o)} )</td>
<td>( A_2 = D_i \sin(\theta/2) )</td>
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| Inner kiln wall to solid | \( h_3 = h_{3cc} + h_{3Rad} \) | \( A_{3cc} = \pi D_i (\theta/360) \)
| Inner wall to outer wall | \( h_4 = 0.2k/(D_o - D_t) \) | \( A_4 = \pi(D_o - D_i)/2.303 \log(D_o/D_i) \) |
| Outer wall to ambient air | \( h_5 = 2.5* \) | \( A_5 = \pi D_o \) |

Temperatures in \( ^\circ \)R

* Approximation for initial estimate of outer-wall temperature

Value of \( f \) taken as 0.5 for two kilns examined

\( G_g = \) mass flow rate of wet gas, \( \text{lb/h} \)
\( T_s = \) solid temperature, \( \circ \)F
\( T_g = \) gas temperature, \( \circ \)F
\( S_x = \) cross-sectional area of kiln, \( \text{ft}^2 \)
\( T_w = \) inner kiln wall temperature, \( \circ \)F
\( E_g = \) gas emissivity
\( E_s = \) solids emissivity
\( D_o = \) outer kiln diam., ft
\( D_i = \) inner kiln diam., ft
\( f = \) correction factor
\( k = \) thermal conductivity of insulating brick, \( \text{Btu/lb \circ \)F \( \text{ft} \)
Kern (1974) analysed the heat flow paths in an unflighted rotary cooler; a charge of wide size range entered the cooler at 800-1000°C and was cooled down to about 200°C by a countercurrent stream of air, here radiative exchange was assumed to be negligible. The solids temperature was assumed to be independent of radial position which is a 'well-mixed condition' (Imber and Paschkis, 1960). For this particular process where the charge was stated to have a wide size range, the 'well-mixed' condition is not a valid assumption as preferential segregation takes place (Richards, 1966). The most important assumption made by Kern is the existence of a linear relationship between the differential heat flux densities and the gas and charge temperatures $T_g$ and $T_s$ respectively:

\[
\frac{dT_s}{dz} = A_1 x T_s + A_2 x T_g + A_3 \quad 2.2
\]

\[
\frac{dT_g}{dz} = B_1 x T_s + B_2 x T_g + B_3 \quad 2.3
\]

The coefficients $A_n^*$ and $B_n^*$ were stated to be functions of the mass flow rates, the rotational frequency and contain the physical properties of the materials. In order to solve the above equations the coefficients $A_n^*$ and $B_n^*$ were assumed to be independent of $T_g$, $T_s$ and the axial coordinate $z$.

Vaillant (1965) proved the existence of such a linear relationship as used by Kern (1974) and evaluated the coefficients by the use of a passive network analogue computer.

Sunavala (1977) performed a mathematical analysis of the temperature distribution of the charge and gas in a dry process cement kiln. Analytical expressions were drawn from the theory of counterflow recuperators. By extending the theory for counterflow equipment Sunavala presented intermediate temperatures of the gas ($T_H$) and charge ($T_C$):

\[
T_H = T_{H1} - \frac{a (1 - z)}{(a - z_0)} (T_{H1} - T_{C1}) \quad 2.4
\]

and

\[
T_C = T_{H1} - \frac{1}{(1 - z_0)} (T_{H1} - T_{C1}) \quad 2.5
\]
where

\[ a = \frac{W_c C_p}{\eta \ W_h C_p H_h} \]

\[ W = \text{mass flow rate} \]
\[ C_p = \text{average specific heat} \]
\[ \eta = \text{overall thermal efficiency of rotary cement kiln (54\%)} \]
\[ T_{H1} = \text{temperature of hot gas at distance L from inlet of hot gas} \]
\[ T_{C1} = \text{temperature of charge at inlet to rotary kiln} \]
\[ T_c = \text{temperature of charge at distance L from inlet of hot gas} \]

\[ z = \exp \left[ -\frac{h A_n (a - l)}{W_c C_p H_h} \right] \]

\[ z_o = \exp \left[ -\frac{h A_o n (a - l)}{W_c C_p H_h} \right] \]

\[ A_o = \text{total exposed area of charge surface in the kiln} \]
\[ A = \text{exposed area of charge up to distance L} \]
\[ h = \text{overall heat transfer coefficient} \]

Using the same notation Sunavala (1977) simplified the complicated structure of the equations derived by Imbe and Paschkis (1962) for the well mixed case:

\[ T_h = T_{H1} - \frac{a (z - z_o) (T_{H1} - T_{C1})}{z (a - z_o)} \]

\[ T_c = T_{H1} - \frac{(az - z_o) (T_{H1} - T_{C1})}{z (a - z_o)} \]

In calculating the temperature distribution of gas and charge Sunavala varied the overall heat transfer coefficient \( h \) from 100 to 500 \( W/m^2 \) K. A value of 350 \( W/m^2 \) K was found to give the closest agreement between the theoretical predictions and measured temperature distributions.

Mukherjee and Ghosh (1977) experimented with a direct oil fired rotary kiln of 3.3 m length and 0.3 m inside diameter. The charge used was limestone of particle size -5 +10 mesh. They measured the material
temperature using a cup-type pyrometer. This essentially consisted of a hemispherical stainless steel cup 5 cm in diameter, welded to this cup was an 11 ft long stainless steel tube assembly of outside diameter 2.22 cm. The tube assembly consisted of two concentric tubes, through the inner tube was drawn insulated chromel-alumel thermocouple wire.

Gas temperatures were measured with a suction pyrometer. The analysis used to evaluate their temperature measurements was identical to that of Imber and Paschkis (1962). Their computed and measured temperature profiles agree well with each other. The relative values of the various heat transfer coefficients they measured are given in Table 2.2.

**TABLE 2.2.**

\[
\begin{align*}
  h_{gw} &= 18.6 \text{ k cal/m}^2 \text{ h} \quad ^\circ \text{C} \\
  h_{wc} &= 93.23 \\
  h_{gc} &= 50.5 \\
  h_{wa} &= 9.75^* \\
\end{align*}
\]

* this value taken after Gilbert (1936)

It can be seen that the value of \(h_{wc}\) is approximately 5 x \(h_{gw}\), i.e. the rule of thumb criteria used by Imber and Paschkis (1962).

Brimacombe and Watkinson (1978a, 1978b) experimented with a natural gas direct-fired kiln. The charge used was a 2:1 ratio mixture of silica flint shot and Ottawa sand with a mean particle diameter of 0.58 mm. The kiln \((L = 5.5 \text{ m}, \text{i.d.} = 0.406 \text{ m})\) was fitted with fifty-two thermocouples. These thermocouples rotated with the kiln, millivolt readings were taken off the rotating shell through a slip ring assembly. This assembly unit was thermally insulated from the kiln shell and welded concentrically to the outside of the kiln. On the rotating assembly unit were series of electrically insulated copper strips connected to appropriate thermocouples, a fixed brush contact being used to transmit the millivolt signal from the copper strips to a multichannel recorder. Patterson (1980) experimented with such a slip ring system and abandoned this type of system, on account of the electrical noise being picked up by the copper rings from the
electrical motor used to drive the kiln, dust contamination and oxidation of the copper rings. Patterson then went on to develop a noise free, data gathering, system.

Brimacombe and Watkinson (1978b) employed equations similar to Saas (1967) but due to the low Reynolds number used (2000) the convective coefficient is different, the gas to wall flux density being calculated from:

\[
\frac{q_{gw}}{A} = \frac{\sigma}{1 + \frac{1}{E_g} + \frac{1}{E_w} - 1} (T_g^4 - T_w^4) + h_{cw} (T_g - T_w)
\]

2.10

where

\( \sigma \) = Stefan Boltzman constant
\( E_g \) = gas emissivity
\( E_w \) = wall emissivity

and

\[
h_{cw} = 1.26 \frac{K}{D} \left[ \frac{Re \ Pr}{2 (L - x)/D} \right]^{0.4334}
\]

2.11

These equations apply to the case of convection from a gas to the wall of a non-rotating duct (Rohsenow and Hartnett, 1973), where the velocity and temperature profiles are still developing. The gas to solids heat flux was measured to be up to ten times greater than the heat flux between the gas and wall. Burden-side and gas-side convective heat transfer coefficients were calculated from the heat flux data to have values ranging from 700 to 1200 and 120 - 240 W/m² K respectively. Again, the rule of thumb, values of Imber and Paschkis (1962) is apparent. Brimacombe and Watkinson (1978b) failed to predict the gas to solids heat flux successfully.

A study made by Tscheng and Watkinson (1979) was purely convective in nature. A 7.6 kW electrical furnace was used to heat air before it entered the kiln. The kiln dimensions were \( L = 2.44 \) m and i.d. = 0.1905 m. The air inlet temperature was in the range 373 - 650 K, and Reynolds number varied from 1600 to 7800. In this temperature range Tscheng and Watkinson give a gas to wall radiation coefficient of 0.07 W/m² K, a factor of 35 to 70 times less than gas to wall convective coefficient. They found the effect of particle size (1.26, 0.73,
0.5 mm) had a very slight effect on the gas to solids coefficient, they attributed this to the narrow size range of particles used. The degree of fill (6.5 - 17%) on the gas to wall heat transfer coefficient was also found to be insignificant. Local heat transfer coefficients for gas to solid were in the range 10 - 40 W/m² K and for gas to wall 3 - 10 W/m² K.

Gardeik and Jeschar (1979a, 1979b) distinguished between the direct transfer of energy from the gas to the charge and the indirect (regenerative) transfer via the rotating wall of the tube. They highlighted the complexity of the heat transfer in rotating tubes by breaking down the direct and indirect heat transfer modes, as depicted in Figure 2.2, into their various sub-processes.

![Figure 2.2 Energy transport through a cross-section of a rotary kiln](image)

where the direct transfer comprises:

- **(A₁)** heat transfer by convections and radiations from the gas to the surface of the charge
- **(A₂)** heat transfer by convection and radiation from the gas to the part of the tube wall not covered by the material and by re-radiation to the material
- **(A₃)** heat transport into the feed material by:
  1. conduction in the material
  2. radiation between material particles
  3. solid-state convection by mixing
and, indirect transfer of heat comprises:

(B1) heat transfer by convection and radiation from the gas to the part of the tube wall not covered by the material

(B2) enthalpy transport (due to the rotary motion of the tube wall) of that portion of the heat flow transferred between gas and wall which is not re-radiated by the wall to the material

(B3) heat transfer by conduction, convection and radiation from the wall of the tube to the adjacent feed material

(B4) heat transport into the feed material by:
   (i) conduction in the material
   (ii) radiation between material particles
   (iii) solid-state convection by mixing

Gardiek and Jeschar obtained temperature distributions in the gas and charge phases by treating the rotating tube as an adiabatic recuperator. This arrangement results in an increase in the enthalpy of the feed and a decrease in enthalpy of the gas. The two processes being coupled to each other. They obtained, as in the case of Kern (1974), two coupled differential equation from the energy balances for the gas and charge phases:

\[
\frac{dT_g}{dx} + \frac{K_{\text{eff}} \cdot A_{s,g}}{L} \cdot (T_g - T_s) = 0 \quad 2.12
\]

\[
\frac{dT_s}{dx} + \frac{K_{\text{eff}} \cdot A_{s,g}}{L} \cdot (T_s - T_g) = 0 \quad 2.13
\]

where

- \( K_{\text{eff}} \) is the overall heat transfer coefficient between gas and solid
- \( A_{s,g} \) denotes the total available heat transfer surface area
- \( L \) is the length of the recuperator
- \( \dot{w} = \dot{m} \cdot c_p \)

Rather than solve these coupled differential equations (Kern)

Gardiek and Jeschar (1979a) decoupled them:

\[
\frac{d^2T_g}{dx^2} + \left[ 1 + \frac{\dot{W}_s}{\dot{W}_g} \right] \cdot \frac{K_{\text{eff}} \cdot A_{s,g}}{\dot{W}_g \cdot L} \cdot \frac{dT_g}{dx} = 0 \quad 2.14
\]
\[ \frac{d^2 T_s}{dx^2} + \left(1 + \frac{\dot{W}_g}{\dot{W}_s}\right) \cdot \frac{K_{eff} \cdot A_{s,g}}{\dot{W}_s L} \cdot \frac{dT_s}{dx} = 0 \tag{2.15} \]

The solutions of the above equations being given as:

\[ \frac{T_g(x) - T_g(x=0)}{T_s(x=0) - T_g(x=0)} = \frac{1}{1 + \frac{\dot{W}_g}{\dot{W}_s}} \left[ 1 - \exp \left( - \left(1 + \frac{\dot{W}_g}{\dot{W}_s}\right) \cdot \frac{K_{eff} \cdot A_{s,g}}{\dot{W}_s} \right) \right] \left( \frac{x}{L} \right)^j \tag{2.16} \]

The expression for the temperature distribution in the charge can be obtained by transposing the subscripts in the above equation (i.e. replace \( g \) for \( s \)). The above equations are for a parallel-flow recuperator and the authors stated that they are valid for counterflow situations if the capacity flow \( \dot{W}_g \) which is contrary to the positive \( x \)-direction is substituted as a negative quantity.

The main aim of Gardeik and Jeschar (1979a) was the evaluation of the term \( K_{eff} \). They obtained an expression for \( K_{eff} \) by considering further heat balances for a volume element of the wall in the gas phase and then in contact with the charge. They gave the following expression for \( K_{eff} \):

\[ K_{eff} = \alpha_{gs, eff} + \frac{\rho_w \cdot c_w \cdot \omega (R_o^2 - R_i^2) \cdot E}{2 \cdot R_i \sin \left( \frac{\phi_s}{2} \right)} \tag{2.17} \]

where

\[ \alpha_{gs} = \text{direct heat transfer coefficient} \]
\[ \rho_w = \text{density of wall} \]
\[ c_w = \text{specific heat of wall} \]
\[ \omega = \text{angular velocity of kiln} \]
\[ R_{i,o} = \text{radius of kiln} (i = \text{inner, o = outer}) \]
\[ \phi_s = \text{included angle of bed} \]
\[ E = (1 - E_{gg}) \cdot (1 - E_{ss})/(1 - E_{gs}) \]
\[ E_{gg} = \exp (-St_{w,g} \cdot \phi_g) \]
\[ E_{ss} = \exp (-St_{w,s} \cdot \phi_s) \]
The first term in the above equation for \( \kappa_{\text{eff}} \) characterizes the direct heat transfer between gas and charge, while the second term takes into account the indirect (regenerative) transfer of heat.

Patterson (1980) provides the only published experimental work on chain systems used in the wet process of cement manufacture of portland cement. The small experimental kiln (\( L = 4 \text{ ft} \), i.d. = 1 ft) was operated in the range 1 - 4 rpm. The Reynolds number varied between 1000 - 46000, at an average gas inlet temperature of about 100°C. Slurry flowrates were in the range 200-410 kg/hr. Patterson found that effect of using dam rings to maintain a constant bed depth was ineffective. The overall effect of kiln rotational speed was found to be insignificant.

2.1.2 Conductive Heat Transfer: Bed to Wall Heat Transfer

The wall to solids heat transfer rate measurements in rotary kilns with a definite particle-wall contact time has been described with the aid of a penetration mathematical model (Wachters and Kramers, 1964). This model assumes a pseudo-homogeneous fixed bed, no gas convection in the interparticle voids, negligible radiation, the same wall and bed temperature at the contact surface and a heat conduction transfer process according to Fourier's law provided that the contact time is long and the particle diameter small enough.

Wachters and Kramers experimented with a copper cylinder (\( L = 0.475 \text{ m} \), i.d. = 0.152 m). They interpreted their results in terms of unsteady-state heat transfer from the cylinder wall during a contact time \( t_c \), and a region at the wall, of thickness \( d \), in which conduction takes place. Their proposed relationship between the heat transfer rate and wall layer thickness is shown in Figure 2.3.
FIGURE 2.3 Heat transfer as a function of the dimensionless layer thickness, $\delta$, at wall of rotary kiln
(after Wachters and Kramers (1964))

$a = \lambda/\rho C_p$, heat diffusivity, m$^2$/s
$\alpha = \text{average heat transfer coefficient during time of contact, } t_c, \quad \text{W/m}^2 \degree C$
$d = \text{thickness of wall layer, } m$

Wachters and Kramers (1964) highlighted three special cases

(i) no separate layer at the wall ($\delta = 0$), this gives an average heat transfer coefficient $h_{gw}$

$$h_{gw} = \frac{2 k_s}{\gamma \pi \alpha t_c}$$  \hspace{1cm} 2.18

$k_s = \text{thermal conductivity of the bed}$
$\alpha = (k_s/\rho C_p)$, thermal diffusivity
$t_c = \text{contact time}$

(ii) a dimensionless layer of thickness $\delta > 2$

$$h_{gw} = \frac{2 k_s}{3 \gamma \pi \alpha t_c}$$  \hspace{1cm} 2.19
(iii) an intermediate region for \( S < 0.6 \), Tscheng and Watkinson (1979) expressed their results for this case

\[
h_{ws} = \left[ \frac{\sqrt{\pi} \alpha t_c}{2k_s} + \frac{\sqrt{\pi} d'}{2k_s} \right]^{-1}
\]

where the wall layer thickness, \( d' \) was determined to be:

\[
d' = 0.00112 \sqrt{\beta}
\]

\( \beta \) = central angle of the sector occupied by the solid bed in radians.

Tscheng and Watkinson re-expressed equation 2.18 in a dimensionless form

\[
\frac{h_{sw} f'_w}{k_s} = 2 \sqrt{2} \left( n R^2 \beta / \alpha \right)^{1/2}
\]

where

\[
f'_w = \text{covered wall circumference}
\]
\( n \) = rotational speed
\( R \) = kiln radius

They also reviewed the data on the bed to wall heat transfer path and expressed the data in terms of \( h_{sw} f'_w/k_s \) versus \( n R^2 \beta / \alpha \) by the following equation

\[
\frac{h_{sw} f'_w}{k_s} = 11.6 \left( n R^2 \beta / \alpha \right)^{0.3}
\]

for \( n R^2 \beta / \alpha < 10^4 \)

Wes et al (1976) also used the simple penetration model and confirmed its validity. Values for \( h_{sw} \) ranged between 100 - 300 W/m² K for starch particles of 15 - 100 \( \mu \)m mean particle diameter for a rotational speed range of 1 to 10 rpm.
2.1.3 Radiative Heat Transfer

In a direct fired kiln, two distinct regions of heat transfer can be identified, the flame zone and the non-flame zone. In the flame zone radiation is the predominant heat transfer process.

Cross and Young (1976) provide the simplest radiation model used in the radiative analysis of a rotary kiln. One of the most notable experimental works was undertaken by Jenkins (1976). Jenkins' results were taken using an extremely well instrumented cement kiln. This being the first case where the instruments (gas sampling and suction pyrometers) actually revolved with the kiln. The mathematical model of the heat transfer mechanism for a large enclosed flame in a rotary cement kiln was based on the zone method of Hottel and Sarofim (1967). The zone method was modified to take into account the specific firing conditions for a large cement kiln. With this model both temperature and heat-flux distributions were predicted and then compared with reasonable success to the values measured in the full size kiln.

More recent theoretical models have been made available (Gorog, 1981, 1983), but these models have not included or been compared with any experimental measurements.

2.2 Residence Time and Solids Flow in Rotary Kilns

The mean residence time of the charge in a rotary kiln and its flow behaviour, well-mixed, non-mixed or intermediate depends on several variables. These variables assume the angle of inclination, speed of rotation and dimensions of the kiln and the degree of fill and dynamic angle of repose of the material. Due to temperature gradients which exist in a rotary kiln the rate of passage along a kiln is not constant, but varies with the physical and chemical transformations which the initial charge undergoes. The fraction of the kiln volume occupied by the solids, termed holdup, will also strongly influence the rate of heat transport from those surfaces of the charge exposed to heat into the innermost core of the bed.
2.2.1 Mean Residence Time

One of the first attempts to find a correlation between these variables were made by Sullivan et al (1927) of the U.S. Bureau of Mines, who experimented with small inclined kilns and relatively small feed rates. The kilns used by Sullivan et al were all seven feet in length but with various diameters (3, 6, 111/2, 193/4 inches). They derived an empirical formula for the mean residence time through kilns:

\[
T = \frac{1.77 L (\theta)^{1/2}}{2 R N \alpha^0}\min
\]

where

- \( N \) is the rotary speed of the kiln
- \( L \) is the length of the kiln
- \( R \) is the kiln radius
- \( \alpha^0 \) is the inclination of the kiln to the horizontal axis
- \( \theta \) is the dynamic angle of repose of material

Sullivan et al developed other formulae to take into account obstructions to flow and ring formation. Bayard (1945) re-correlated this data and expressed the data in the form of nomograms.

Friedman and Marshall (1949) related the residence time (\(T\)) to the holdup (\(X\)) by the following equation:

\[
T = \frac{L X}{K F}\]

where

- \( F \) = solids flow (ft\(^3\)/h ft\(^2\))
- \( K \) = a constant

They also reviewed a number of previous studies, and found that there was no general agreement with the formula of Sullivan et al.

Saeman (1951) analysed the tumbling behaviour of solids in rotary kilns in order to quantify the empirical constant (1.77) of Sullivan et al in equation 3.23. Saeman (1951) gave the following residence time equation:
\[ T = \frac{L \sin \theta}{2 \pi R N (\alpha + \psi \cos \theta)} \]  \hspace{1cm} (2.25)

where
\[
\begin{align*}
\alpha &= \text{slope of the kiln in radians} \\
\psi &= \text{angle between bed and kiln axis in radians}
\end{align*}
\]

For lightly loaded kilns the bed depth along the kiln is essentially uniform, for such cases the angle \( \psi \) in equation 2.25 is zero.

\[ T = \frac{L \sin \theta}{2 \pi R N \alpha} \]  \hspace{1cm} (2.26)

Re-expressing equation 2.26 to have \( \alpha \) in degrees

\[ T = \frac{L \sin \theta}{2 \pi R N \left[ \frac{\pi}{180} \times \alpha^0 \right]} \]  \hspace{1cm} (2.27)

For many practical purposes \( \theta \), the dynamic angle of repose of the material is between 35° and 45°. Saeman used a value of \( \theta = 37° \) in equation 2.27, with the following result

\[ T = \frac{10.97 L}{2 R N \alpha^0} \]  \hspace{1cm} (2.28)

With \( \theta = 37° \) is substituted into equation 2.23 resulting in

\[ T = \frac{10.77 L}{2 R N \alpha^0} \]  \hspace{1cm} (2.29)

This close comparison is confirmation of the theoretical analysis presented by Saeman.

As can be seen from the general equation of Friedman and Marshall (1949), the holdup \( (X) \) is an important variable in determining the mean residence time. Dimensional analysis has been used to relate the holdup to some operating variables [Varentsov and Yufa (1961), Zablotny (1965), and Whatley (1977)].

Zablotny derived the following groups:

\[ X = C \left[ \frac{Q_v}{N \cdot D^3} \right]^a \left[ \frac{\alpha}{\beta} \right]^b \]  \hspace{1cm} (2.30)
The dimensionless expression \( Qv/ND^3 \) is the ratio of the quantity of material feed in a unit of time per unit cross-section of the kiln to the rate of rotation of the kiln. Zablotny working on quartzite sand of size range (1.5 - 0.34 mm), of angle of repose (\( \beta = 32^\circ \)) in a rotary kiln of length 3.55 m and diameter 0.352 m, where angle (\( \alpha \)) of inclination of the kiln was varied from (1° - 6°), evaluated the constant \( C \) and exponents \( a \) and \( b \), viz:

\[
C = 0.55 \\
a = 1.0 \\
b = -0.85
\]

Akerman et al (1966) have obtained results on industrial kilns which indicate that under favourable conditions, with relatively free-flowing materials, solids transport can approximate to plug flow. Luoto and Rotkirch (1958) using a potassium tracer obtained a full distribution curve for clinker issuing from a cement kiln, as shown in figure 2.4.

![Figure 2.4](image)

**FIGURE 2.4** Residence time distribution of clinker in a rotary cement kiln (After Luoto & Rotkirch).

### 2.2.2 Flow of Solids in Rotary Kilns

To understand the heat transfer mechanisms from the gas to solids bed, some considerations must be given to the bed motion. Rutgers (1965) has described regimes of bed motion in a rotating cylinder as a function of the ratio \( N/N_c \), where \( N_c \) is the critical angular speed of rotation at which centrifuging starts.
The schematic description of the path of a particle which is transported through a rotating cylinder has been proposed by Vahl and Kingma (1952) and Saeman (1951). The bulk of material has no relative motion with respect to the motion of the kiln. Thus a particle (see figure 2.5) at a distance \( r \) from the kiln axis describes a critical path in a plane perpendicular to the axis with the same angular velocity as the kiln wall. After it has reached the surface of the material, it cascades down along a straight path having a dynamic angle of repose (\( \phi \)) with respect to a horizontal plane. The latter movement has two components, one perpendicular to the kiln axis (\( k \)) and one parallel to it (\( s \)). The major contribution to the transport of material through the kiln comes from \( s \) — known as kiln action.

![FIGURE 2.5. Cross-section of a rotary kiln (after Saeman 1951).](image)

Kiln action can be broken down further into two modes of transport:

(i) cascading/rolling
(ii) kilning/sliding
Case (i) provides a well mixed bed, and (ii) results in a stagnant core. These two modes depend heavily on the nature of the material and its size range.

The ratio between $s$ and $k$ depends on the angle of repose ($\beta$) of the material, the slope of the kiln ($\tan \alpha$) and the gradient of the bed surface with respect to the kiln axis ($-\frac{dh}{dx}$). This relationship has been shown by Saeman (1951) to be:

$$\frac{s}{k} = \frac{\tan \alpha}{\sin \beta} \frac{\cos \beta}{\sin \beta} \frac{dh}{dx}$$

It is valid for small values of $\alpha$ encountered in practice and for small values of $\frac{dh}{dx}$.

Kramers and Croockewit (1952) developed the following formula for the volumetric flow of material through any cross section of the kiln:

$$\Phi_v = \frac{4}{3} \pi R^3 N \left[ \frac{\tan \alpha}{\sin \beta} \frac{dh}{dx} \cot \beta \right] \left[ \frac{2h - \frac{h^2}{R}}{R^2} \right]^{3/2}$$

where $h$ is the bed height at a distance $x$ from the feed end. Kramers and Croockewit, working with a kiln ($L = 1.78$ m, $D = 0.197$ m) and using river sand (0.6 mm - 1.2 mm) and ground code (0.5 mm - 1.5 mm), found good agreement between their theoretical values and experimental results.

A corresponding equation was developed by Saeman (1951) in parallel with Kramers and Croockewit (1952):

$$\Phi_v = \frac{4}{3} \pi N \left[ \frac{\alpha + \Psi \cos \beta}{\sin \beta} \right] \left[ \frac{R^2 - r_o^2}{r_o^2} \right]^{3/2}$$

where

- $\alpha$ - slope of kiln/radians
- $\Psi$ - angle between bed surface and kiln axis/radians
- $r_o$ - minimum radius in bed
- $N$ - rate of rotation/rpm
Referring to figure 2.5, the bed height can be related to the angle subtended by the bed (ϕ) and the chord length c by the following equations:

\[
\sin \left( \frac{\phi}{2} \right) = \frac{(c/2)}{R} \quad ; \quad \cos \left( \frac{\phi}{2} \right) = \frac{R_0}{R}
\]

squaring and adding the results in:

\[
c/2 = (R^2 - r_o^2)^{1/2}
\]

or

\[
\frac{D \sin (\phi/2)}{2} = (R^2 - r_o^2)^{1/2}
\]

substituted into equation 2.33 results in:

\[
\phi_v = \pi N D^3 \sin^3 \left( \frac{\phi}{2} \right) \left( \psi \cos \beta + \alpha \right) / 6 \sin \beta
\]

2.34

For mass flow \((W_g)\) equation 2.34 becomes:

\[
W_g = \rho_d \cdot \phi_v
\]

2.35

where \(\rho_d\) = bulk density of the bed.

Tscheng and Watkinson (1979) have also reported close agreement with equation 2.34.

2.2.3 Bed Height Along the Kiln

Equation 2.22 derived by Kramers and Croockewit (1952) is a differential equation which relates the height of material, \(h\) with distance \(x\) from the feed end for given conditions of operation.

Recalling equation 2.32:

\[
\Phi_v = \frac{4}{3} \pi N R^3 \left[ \tan \frac{x}{\sin \beta} - \frac{dh}{dx} \cot \beta \right] \left[ \frac{2h - h^2}{R^2} \right]^{3/2}
\]

2.32

Kramers and Croockewit replaced the function containing \(h/R\) by a linear one:

\[
\left[ \frac{2h - h^2}{R^2} \right]^{3/2} \approx 1.24 \frac{h}{R}
\]

2.36
They further introduced the following dimensionless combinations:
\[
N_\Phi = \frac{\Phi_v \sin \beta}{N R^3 \tan \alpha} \quad \text{and} \quad N_k = \frac{R \cos \beta}{L \tan \alpha}
\]  
2.37

with the condition
\[
h = h_L \quad \text{for} \quad x = L
\]  
2.38

If no constriction is presented at the discharge side of the kiln, \( h_L = 0 \). After substitution of equations 2.37 and 2.38 into equation 2.32, they arrived at the following solution:
\[
\frac{h_L - h}{R N_\Phi} = \frac{0.193}{N N_\Phi} \ln \left( \frac{h_L}{h} \right) - 0.193 = \frac{L - x}{L N_k N_\Phi}
\]  
2.39

where
\[
L \quad \text{- length of kiln}
\]
\[
x \quad \text{- axial distance from feed end}
\]

Equation 2.39 is a dimensionless relationship between the height of the bed and its distance from the discharge end. Recalling Saeman's (1951) equation 2.33:
\[
\Phi_v = \frac{4 \pi N}{3} \left( \frac{\alpha + \psi \cos \beta}{\sin \beta} \right) \left( \frac{R^2 - r_o^2}{2} \right)^{3/2}
\]  
2.33

where \( r_o \) is the path in the bed of material having the shortest radius and is related to bed height (\( h \)) and kiln radius (\( R \)) by:
\[
R = r_o + h
\]
\[
r_o^2 = (R - h)^2
\]

and the slope of the bed, \( \psi \), can be expressed by \(-dh/dx\).

Following these substitutions into equation 2.33, the resulting equation being:
\[
\Phi_v = \left[ \frac{4 \pi N R^3}{3} \tan \alpha \frac{-dh}{dx} \cot \beta \right] \left( \frac{2h - h^2}{R R^2} \right)^{3/2}
\]  
2.40
which is identical to 2.32. The different results are due to the fact that different simplifications were used by Saeman and Kramers and Croockewit, but both are based on the same theoretical assumptions first developed by Vahl (1949).

2.2.4 Tapering Off of Depth of Load with Free Discharge

The dropping off effect can take place at the discharge end of a kiln, or when a kiln changes from a lesser to a greater diameter. Warner (1953) reviewed the data of Sullivan et al (1927), and concluded that for all practical purposes, the dropping off effect extends up to 5 diameters. This is borne out by the actual data of Sullivan et al quoted by Saeman (1951). However, the dropping off effects depend on the length to diameter ratio of a kiln, and the greater the normal depth, the further up the kiln the dropping-off effect extends. When the depth of a load is more than 17% of the kiln diameter, it can be considered to be a heavy bed (Warner).

2.2.5 The Criteria for Rolling/Sliding

It is essential that the nodule bed be well mixed during its passage down the kiln, so that it can be uniformly heated. Rutgers (1965) has discussed the many forms of bed behaviour - slipping, slumping, rolling, cascading, cateracting and centrifuging. These individual cases depend on variables such as rotational speed, kiln diameter, holdup and the physical characteristics of the bed.

As previously discussed (Imber and Paschkis (1962)) there are two main ways in which the bed might behave:

(i) well mixed case - rolling situation
(ii) non-mixed case - sliding situation.

Cross (1974) derived a model based on a differential moment analysis on a static bed and has shown that the nodule bed will move in the rolling mode provided that:

\[
\sin \beta < \left[ \frac{3\mu (\phi/2 - \sin \phi/2 \cos \phi/2)}{2 \sin^3 \phi/2} \right]^{2.41}
\]
where
\[
\beta \quad \text{is the static angle of repose} \\
\phi/2 \quad \text{is the half angle subtended by the nodule bed} \\
\mu \quad \text{is the static coefficient of friction between the kiln wall and the nodule bed.}
\]

The predictions of Henein et al (1983) follow the same basic analysis as used by Cross and arrive at the following criteria for the bed to be in the rolling mode:

\[
\tan \beta < \left[ \frac{3\mu (\phi/2 - \sin\phi/2 \cos\phi/2)}{2 \sin^3 \phi/2} \right] 2.42
\]

Henein et al state that their result is more logical than that of Cross because the tangent of the inclination angle of the bed rather than the sine appears on the left hand side of the equation. Henein et al mathematically characterise the different modes of bed behaviour highlighted by Rutgers (1965), and expressed their results in the form of a bed behaviour diagram (bed depth vs. rotational speed or holdup vs. Froude number).
3 PHYSICAL MODELLING APPLIED TO CHAINED ROTARY KILNS

3.1 Similarity Criteria in Models

The design and operation of any pyroprocessing equipment gives rise to complex problems which are not always soluble. Any modifications to such equipment to improve efficiency are costly ventures both financially and time-wise, with no real guarantee of an increase in process efficiency. The use of both physical and mathematical models to solve such problems provides a quick and cheap alternative. For an effective correlation between the results from units of different size it is necessary to simulate all the factors which could affect heat transfer in a rotary kiln.

The principle of similarity defines the relationships existing between systems of different size and it has proved most useful in the scaling up or down of physical and chemical processes (Johnstone and Thring, 1957). According to this principle, the configuration of a system in space and time can be defined by ratios of magnitudes within the system, and does not depend on the size or nature of the units of measurement.

The laws of similarity as applied to heat transfer, were obtained for the first time by Nusselt (1910, 1915), who derived the dimensionless parameters appropriate to both forced and free convection from the differential equations of fluid flow and heat transfer with their respective boundary conditions. The resulting dimensionless parameters are the now familiar Reynolds (Re), Nusselt (Nu), Peclet (Pe) and Prandtl (Pr) numbers.

3.1.1 The Physical Meaning of Dimensionless Parameters

The Reynolds number may be interpreted as a measure of the ratio of inertia forces to viscous forces in view of the relation

\[ \text{Re} = \frac{u L}{\nu} = \frac{\rho u^2}{\mu u/L} \]
where
L = a characteristic dimension of the system
u = velocity of fluid
\( \rho \) = density of fluid
\( \mu \) = dynamic viscosity
\( v = \mu/\rho \), kinematic viscosity

The Nusselt number assumes the existence of a stationary fluid layer with conductivity \( k \) adjacent to the solid boundary, in which layer there occurs the same temperature drop as that on which the heat transfer coefficient \( h \) is based, then the thickness of layer \( k/h \) and the Nusselt number can be expressed as

\[
Nu = \frac{h L}{k} = \frac{L}{k/h}
\]

and interpreted as the ratio of a characteristic length \( L \) to the thickness of the layer.

The Peclet number can be written as

\[
Pe = \frac{u L}{\alpha} = \frac{u \rho C_p}{k/L}
\]

\( C_p \) = specific heat of fluid
\( \alpha = k/C_p \rho \), thermal diffusivity of fluid

It provides a measure of the relative magnitude of heat transfer by convection to heat transfer by conduction.

The Prandtl number, \( Pr = v/\alpha \), compares two molecular transport properties, the kinematic viscosity (or momentum diffusivity), which governs the transport of momentum through friction, with the thermal diffusivity, \( \alpha \), which governs the transport of heat through conduction. The transport of momentum is due to velocity gradients, while the transport of heat is caused by temperature gradients. Thus the Prandtl number is characteristic of the relationship between the velocity and temperature fields.
3.1.2 Geometric Similarity

Geometric similarity implies that every linear dimension of the model bears the same ratio to the corresponding dimension of the full scale system. With a rotary kiln the percent holdup of the charge or the kiln loading in the model must be equal to that in the full scale system (Themelis et al, 1964).

3.1.3 Kinematic Similarity

Kinematic similarity implies that the fluid follows geometrically similar paths in corresponding intervals of time. This implies that there is a velocity scale ratio which is maintained constant between corresponding points of the model and the real system.

3.1.4 Dynamic Similarity

Dynamic similarity involves the forces which affect the movement of mass in a moving system. In this case, dynamic similarity also entails kinematic similarity, as the the motions of both the gas and solid phases are functions of the forces applied to the system.

Dynamic similarity in the gas phase is usually met by ensuring that the Reynolds numbers are equal. For the moving bed of solids dynamic similarity can be ensured by using the same feed and having the same kiln loading between model and the full scale system. Thus, the average angle of material repose and the forces of friction which affect rolling, mixing and lateral movement of the particles through the kiln should be of the same magnitude for different kilns.

3.2 The Modelling Criteria for a Chain System

A kiln chain system can be thought as an aggregation of solid bodies arranged so that the void spaces between the bodies are connected and will permit through passage of a fluid. The fluid on its travel through these connected voids will undergo frequent expansions and contractions. Therefore, a chain bank can be idealized into a series of baffles of large voidage. Patterson (1980) extended the analysis of Coulson and Richardson (1977) on the fluid flow through converging-
diverging channels. Patterson found through air and water modelling experiments that for dynamic similarity the voidage of the chains in the model must be identical to that of the real system.

3.3 Scale-Down of Kiln Chain Zone

3.3.1 Chain Density

As a consequence of the requirement of equal voidage in the model and the real kiln, it becomes apparent that the chaining density in the model should equal that of the real kiln:

\[ e = \frac{V_{\text{kiln}} - V_{\text{chains}}}{V_{\text{kiln}}} \]

\[ e = \text{voidage in kiln} \]

\[ V_{\text{chains}} = \text{volume of chains} \]

\[ V_{\text{kiln}} = \text{volume of chained kiln} \]

Rearranging

\[ 1 - e = \frac{V_{\text{chains}}}{V_{\text{kiln}}} = \frac{(M_{\text{chains}}/\rho_s)}{V_{\text{kiln}}} = \frac{\rho_c}{\rho_s} \]

\[ M_{\text{chains}} = \text{mass of installed chains} \]

\[ \rho_c = \frac{M_{\text{chains}}}{V_{\text{kiln}}} \text{, chaining density} \]

\[ \rho_s = \text{density of steel} \]

Since

\[ |e_{\text{model}}| = |e_{\text{real}}| \]

\[ \left| 1 - \frac{\rho_c}{\rho_s} \right| \frac{\rho_s}{\rho_s_{\text{model}}} = \left| 1 - \frac{\rho_c}{\rho_s} \right| \frac{\rho_s}{\rho_s_{\text{real}}} \]

then

\[ |\rho_c|_{\text{model}} = |\rho_c|_{\text{real}} \]
3.4 Chain Representation

3.4.1 Traditional

The model chained kiln is based on Chinnor No. 1 kiln of the Rugby Portland Cement plc. It is current practice to represent the chain length and the pitch of the spiral hanger in terms of kiln diameter. The number of spiral starts of the chain hanger is three.

Hence, chain length = \( \frac{2}{3} \times \text{kiln diameter} \)
spiral pitch = \( 1 \times \text{kiln diameter} \)

It is a trial and error procedure to evaluate the correct chain stock size for a given distance between chain attachment points to give the required chain density. Figure 3.1 gives a schematized representation of the drawing needed for a chained section of a cement kiln.

---

**Figure 3.1** Schematic representation of three spiral start hangers.

3.4.2 Mathematical Alternative

With the growth of microcomputers and allied software packages, the traditional representation of chain installations (Figure 3.1) seems cumbersome and inflexible. With this in mind the following analysis is developed by the author to be used to represent the chain system on a computer.
The equation of a helix in vector notation is (Kuipers and Timman, 1969)

\[
\tau = \begin{bmatrix}
x = R \cos wt \\
y = R \sin wt \\
z = R \omega t \cdot \tan \alpha
\end{bmatrix}
\]

where

- \( R = \) radius of kiln
- \( \alpha = \) helix angle
- \( \omega = \) frequency
- \( t = \) time

The length of arc \((ds)\) between two points can be evaluated by:

\[
\left[ \frac{ds}{dt} \right]^2 = \tau \cdot \tau
\]

\[
s_{1,2} = \int_{t_1}^{t_2} \left[ \frac{dr}{dt} , \frac{dr}{dt} \right]^{1/2} \ dt
\]

where

\[
\frac{dr}{dt} = \begin{bmatrix}
-w R \sin wt \\
w R \cos wt \\
w R \tan \alpha
\end{bmatrix}
\]

and

\[
\left[ \frac{dr}{dt} , \frac{dr}{dt} \right]^{1/2} = w R \sqrt{1 + \tan^2 \alpha}
\]

for a frequency of \(2\pi\) radians in time \(t_{\text{rev}}\), then for one revolution the arc length \((s_{\text{rev}})\) is

\[
s_{\text{rev}} = \int_{0}^{t_{\text{rev}}} \left[ \frac{2\pi}{t_{\text{rev}}} \right] \ R \sqrt{1 + \tan^2 \alpha} \ dt
\]

\[
= nD \sqrt{1 + \tan^2 \alpha}
\]

From figure 3.1 it is seen that

\[
\tan \alpha = \frac{D}{nD}
\]
then \( s_{1rev} \) becomes

\[
s_{1rev} = \pi D \sqrt{1 + \frac{1}{\pi^2}}
\]

\[
= D \sqrt{\pi^2 + 1}
\]

Using the above analysis and the Gino graphics system at the University of Surrey, a graphics program has been written to draw a helix.
4.1 Introduction

In a rotary kiln, heat is transferred to the charge by two paths, across the exposed upper surface and the covered lower surface of the bed as depicted in figure 4.1. At the upper surface of the bed the charge receives heat directly by radiation and convection from the combustion gases. At the lower surface, heat flows by conduction from the wall to charge. The latter heat-transfer path is part of the regenerative cycle of the kiln wall.

FIGURE 4.1 Schematic diagram of a rotary kiln showing major heat flow paths to the charge (after Gorog et al, 1982).

With the introduction of chains in the cold end of a rotary cement kiln this regenerative cycle becomes important. In modelling this regenerative action, the chain system is considered to be a thin layer of the kiln wall. For thick-walled kilns this regenerative action has been shown to be a second order effect (Imber and Paschakis, 1960).
This analysis concentrates on the gas side of the system, with the development of an expression which enables the regenerative action of a chain system to be quantified. Much of the analysis used here has been applied from the work of Gardeik and Jeschar (1979).

4.2 Temperature Distributions of the Gas and Solids Phases

The following and subsequent sections are concerned with describing mathematically the heat transfer in a thin walled rotary kiln through the use of a simplified heat transfer model. In this treatment the regenerative transfer of heat from the wall to the adjacent feed material as a result of the chain system is more particularly considered, and is developed step by step. The object of the analysis presented here is the determination of an overall heat transfer coefficient for the transfer of heat between gas and solid.

The temperature distribution within the gas and solid phases can be established by considering energy balances for each stream (figure 4.2).

\[ \dot{m}_g c_p (\bar{T}_g + \frac{dT}{dx}) \quad \text{---} \quad \dot{m}_g c_p \bar{T}_g \]

\[ \dot{m}_s c_p \bar{T}_s \quad \text{---} \quad \dot{m}_s c_p (\bar{T}_s + \frac{dT}{dx}) \]

FIGURE 4.2 Cross section of a kiln showing gas and solid heat flows

The heat flow terms (\(dQ\)) are calculated by applying Newton's law of cooling. Energy balances for each phase gives two coupled differential equations:

\[ \dot{w} \frac{dT_g}{dx} + K_{eff} \cdot \frac{A_{gs} \bar{T}_s}{L} \cdot (\bar{m}_g - \bar{m}_s) = 0 \]  

4.1
\[
\dot{w} \frac{dT}{dx} + K_{\text{eff}} \cdot \frac{A_{s,g}}{L} \cdot (\bar{T}_s - \bar{T}_g) = 0
\]

where

\begin{align*}
\dot{w} &= \dot{m} \cdot c \\
\dot{m} &= \text{mass flowrate} \\
c &= \text{specific heat} \\
K_{\text{eff}} &= \text{effective heat transfer coefficient; takes into account both the direct and indirect transfer between gas and solid (after Gardeik and Jeschar (1979)).} \\
A_{s,g} &= \text{heat transfer surface area} \\
L &= \text{length of kiln} \\
\bar{T}_{g,s} &= \text{mean temperature of the gas (g) or solid (s) phase}
\end{align*}

Equations 4.1 and 4.2 are decoupled by taking the second derivative:

\[
\frac{d^2T_g}{dx^2} + \left( 1 + \frac{\dot{w}_g}{\dot{w}_s} \right) \cdot \frac{K_{\text{eff}} A_{s,g}}{\dot{w}_g L} \cdot \frac{dT_g}{dx} = 0 \tag{4.3}
\]

\[
\frac{d^2T_s}{dx^2} + \left( 1 + \frac{\dot{w}_s}{\dot{w}_g} \right) \cdot \frac{K_{\text{eff}} A_{s,g}}{\dot{w}_s L} \cdot \frac{dT_s}{dx} = 0 \tag{4.4}
\]

By using the following boundary conditions

at \( x = 0 \), \( \bar{T}_g = \bar{T}_g(0) \)
\[
\bar{T}_s = \bar{T}_s(0)
\]

and re-expressing equation 4.3 as

\[
\frac{dT_g}{dx^2} + A_1 \frac{dT_g}{dx} = 0 \tag{4.5}
\]

where

\[
A_1 = \left[ 1 + \frac{\dot{w}_g}{\dot{w}_s} \right] \cdot \frac{K_{\text{eff}} A_{s,g}}{\dot{w}_g L}
\]

The general solution to equation 4.5 is simply

\[
\bar{T}_g = A_2 e^{-A_1x} + A_3 \tag{4.6}
\]
A$_2$ and A$_3$ are integration constants which are evaluated from the boundary conditions in conjunction with equation 4.1, i.e.

\[
A_2 = \frac{\bar{T}_g(0) - \bar{T}_s(0)}{1 + \bar{\dot{W}}_g/\bar{\dot{W}}_s}
\]

\[
A_3 = \bar{T}_g(0) + \frac{\bar{T}_s(0) - \bar{T}_g(0)}{1 + \bar{\dot{W}}_g/\bar{\dot{W}}_s}
\]

Substitution of $A_2$ and $A_3$ into equation 4.6 and rearranging gives the dimensionless temperature distribution in the gas phase:

\[
\frac{\bar{T}_g(x) - \bar{T}_g(0)}{\bar{T}_s(0) - \bar{T}_g(0)} = \frac{1}{1 + \bar{\dot{W}}_g/\bar{\dot{W}}_s} \left[ 1 - \exp \left( - 1 + \frac{\bar{\dot{W}}_g}{\bar{\dot{W}}_s} \frac{K_{\text{eff}} A_{s,g}}{\bar{\dot{W}}_g L} \right) \right]
\]

Similarly for the solid phase:

\[
\frac{\bar{T}_s(x) - \bar{T}_s(0)}{\bar{T}_g(0) - \bar{T}_s(0)} = \frac{1}{1 + \bar{\dot{W}}_s/\bar{\dot{W}}_g} \left[ 1 - \exp \left( - 1 + \frac{\bar{\dot{W}}_s}{\bar{\dot{W}}_g} \frac{K_{\text{eff}} A_{s,g}}{\bar{\dot{W}}_g L} \right) \right]
\]

4.3 Circumferential Temperature of a Thin Rotating Wall

In establishing the temperature distribution of a thin rotating wall, conduction in the circumferential direction is neglected. Shell losses and the possibility of wall radiation to the charge's upper surface are considered.
FIGURE 4.3 Notation employed for thin-walled analysis

For a unit length of kiln the net heat exchange, \( dq \), between the kiln wall element 1 and the gas, charge and ambient is (see figure 4.3):

\[
dq = h_{gw} (T_g - T_u) R_1 \, d\theta - \frac{h_{ws} B_s}{(2\pi - \theta_o) R_1} (T_u - T_s) R_1 \, d\theta - h_a (T_u - T_a) R_1 \, d\theta
\]

where

\( T_u \) - temperature of the upper (exposed) portion of the wall

\( h \) - heat transfer coefficients

Equation 4.11 can be re-written as

\[
dq = \bar{h}_g (\bar{T}_g - T_u) R_1 \, d\theta
\]

by using the following substitutions

\[
\bar{T}_g = \frac{T_g h_{gw} + T_s h_w + T_a h_a}{h_{gw} + h_w + h_a}
\]

\[
\bar{h}_g = h_{gw} + h_w + h_a
\]
When the wall reaches an equilibrium, the quantity of heat, $dQ$, must also leave the element 1 by transport. The transported heat $dQ$ is

$$dQ = (R_o - R_i) V_w \rho_w C_w \int \frac{dT_u}{d\theta} d\theta$$

where

- $V_w$ - velocity of the wall
- $\rho_w$ - density of the wall
- $C_w$ - specific heat of the wall

Equating equations 4.12 and 4.16 yields the differential relation satisfied by every element of the uncovered portion of the wall:

$$\frac{1}{B} \frac{dT_u}{d\theta} = \frac{h_g (T_g - T_u)}{R_i}$$

where

$$\frac{1}{B} = \left[ \frac{R_o - R_i}{R_i} \right] V_w C_p \rho_w$$

Similarly a heat balance on the element 2 in the covered section of the wall yields:

$$dQ = \left[ h_{wc} (T_g - T_s) + h_a (\bar{T}_f - T_a) \right] R_i d\theta$$

Equating 4.19 to the energy transported by the rotating wall:

$$dQ = (R_o - R_i) V_w \rho_w C_p \left[ -\frac{dT_f}{d\theta} \right] d\theta$$

results in:

$$-\frac{1}{B} \frac{dT_f}{d\theta} = \frac{h_{wc} (T_g - T_s)}{R_i}$$

where

$$\bar{T}_s = \frac{T_s h_{wc} + T_a h_a}{h_{wc} + h_a}$$
\[ h_{wc} = h_{bc} + h_a \]  

Integrating 4.17 and 4.21, respectively

\[ \begin{aligned} 
\int_{T_u'}^{T_u} \frac{d T_u}{(T_g - T_u)} &= \frac{\theta}{\theta_o} \\
\int_{T_g'}^{T_g} \frac{d T_g}{(T_g - T_s)} &= -\frac{B h_g}{h_{bc}} d\theta 
\end{aligned} \]

\[ \frac{T_g - T_u(\theta)}{T_g - T_u(\theta_o)} = \exp \left[ -B h_g (\theta - \theta_o) \right] \]

\[ \frac{T_g(\theta) - T_{\tilde{g}}}{T_g(\theta) - T_{\tilde{s}}} = \exp \left[ -B h_{bc} \theta \right] \]

Equating the temperatures at \( \theta = 0 \) and \( \theta = \theta_o \), according to the boundary conditions 4.28 and 4.29 gives the required relationship for the temperature distribution of the lower (l) and upper (u) parts of the wall.

\[ T_l(0) = T_u(2\pi) \]

\[ T_l(\theta_o) = T_u(\theta_o) \]

By using condition 4.28 into equation 4.25 results in:

\[ \frac{T_g - T_l(0)}{T_g - T_u(\theta_o)} = \exp \left[ -B h_g (2\pi - \theta_o) \right] \]

rearranging equation 4.30:

\[ T_l(0) - T_s = T_g - T_s - \left[ T_g - T_u(\theta_o) \right] \exp \left[ -B h_g (2\pi - \theta_o) \right] \]

By using condition 4.29 into equation 4.27 results in:

\[ T_u(\theta_o) = T_s + \left[ T_l(0) - T_s \right] \exp \left[ -B h_{bc} \theta_o \right] \]
rearranging equation 4.32:

\[-(\bar{T}_g - T_u(\theta_o)) = \left(\bar{T}_s - \bar{T}_g\right) + \left[T_f(0) - \bar{T}_s\right] \exp - B \bar{h}_{wc} \theta_o\]  

4.33

Substitution of equation 4.33 into 4.31

\[T_f(0) - \bar{T}_s = (\bar{T}_g - \bar{T}_s) + \left[(\bar{T}_s - \bar{T}_g) + (T_f(0) - \bar{T}_s)\right] \exp - B \bar{h}_{wc} \theta_o]\]  

4.34

final rearrangement gives:

\[T_f(0) - \bar{T}_s = \frac{(\bar{T}_g - \bar{T}_s) (1 - \exp - B \bar{h}_{wc} (2\pi - \theta_o))}{1 - \exp - B (\bar{h}_{wc} \theta_o + \bar{h}_g (2\pi - \theta_o))}\]  

4.35

Substitution of equation 4.35 into 4.27 and rearranging gives the temperature distribution of the covered part of the wall:

\[T_f(\theta) = \bar{T}_s + \frac{(\bar{T}_g - \bar{T}_s) (1 - \exp - B \bar{h}_{wc} (2\pi - \theta_o)) \exp - B \bar{h}_{wc} \theta_o}{1 - \exp - B (\bar{h}_{wc} \theta_o + \bar{h}_g (2\pi - \theta_o))}\]  

4.36

By using condition 4.29 into equation 4.27 results in:

\[\frac{T_u(\theta_o) - \bar{T}_s}{T_f(0) - \bar{T}_s} = \exp - B \bar{h}_{wc} \theta_o\]  

4.37

re-expressing equation 4.37

\[-(\bar{T}_g - T_u(\theta_o)) = \left(\bar{T}_s - \bar{T}_g\right) + \left[T_f(0) - \bar{T}_s\right] \exp - B \bar{h}_{wc} \theta_o\]  

4.38

Recalling equation 4.31 and substitution for \([T_f(0) - \bar{T}_s]\) into equation 4.38:

\[-(\bar{T}_g - T_u(\theta_o)) = \left(\bar{T}_s - \bar{T}_g\right) + \left[\left(\bar{T}_g - \bar{T}_s\right) - (\bar{T}_g - T_u(\theta_o))\right] \times \exp - B \bar{h}_g (2\pi - \theta_o) \exp - B \bar{h}_{wc} \theta_o\]  

4.39
re-arranging equation 4.39

\[ \bar{T}_g - T_u (\theta_o) = \frac{(\bar{T}_g - \bar{T}_s)(1 - \exp - B \bar{h}_{\text{wc}} \theta_o)A}{1 - \exp - B(\bar{h}_g (2\pi - \theta_o) + \bar{h}_{\text{wc}} \theta_o)} \quad 4.40 \]

Substitution of equation 4.40 into 4.25 and rearranging gives the temperature distribution of the exposed part of the wall.

\[ T_u (\theta) = \bar{T}_g - \frac{(\bar{T}_g - \bar{T}_s)(1 - \exp - B \bar{h}_{\text{wc}} \theta_o) \exp - B \bar{h}_g (\theta - \theta_o)}{1 - \exp - B(\bar{h}_g (2\pi - \theta_o) + \bar{h}_{\text{wc}} \theta_o)} \quad 4.41 \]

With reference to figure 4.4 equations 4.36 and 4.41 can be expressed more concisely as equations 4.42 and 4.43 respectively:

\[ T_{w_s} (\theta_2) = \bar{T}_s + (\bar{T}_g - \bar{T}_s) \left[ \frac{1 - E_g}{1 - E_{gs}} \right] \exp - (B \bar{h}_{\text{wc}} \theta_2) \quad 4.42 \]

\[ T_{w,q} (\theta_1) = \bar{T}_q + (\bar{T}_s - \bar{T}_g) \left[ \frac{1 - E_s}{1 - E_{gs}} \right] \exp - (B \bar{h}_{\text{gw}} \theta_1) \quad 4.43 \]
4.4 Evaluation of $K_{\text{eff}}$

where:

- $\theta_s = \theta_o$
- $\theta_q = 2\pi - \theta_o$
- $E_q = \exp \left(-B h_{GW} \theta_q \right)$
- $E_s = \exp \left(-B h_{WC} \theta_s \right)$
- $E_{gs} = E_q \cdot E_s$
- $T_{w,s}(\theta_2) = T_l(\theta)$
- $T_{w,g}(\theta_1) = T_u(\theta)$

From figure 4.5 the heat flow $dQ_{gs}$ is calculated from:

$$dQ_{gs} = h_{gs,\text{eff}} \left( T_g - T_s \right) \frac{A_{s,g}}{L} \ dx \quad 4.44$$

where $A_{s,g}$ can be evaluated from figure 4.4:

$$\frac{A_{s,g}}{L} \ dx = 2 R_i \sin \left( \frac{\theta_s}{2} \right) \ dx \quad 4.45$$
The gas to wall term $dQ_{gw}$ can be expressed by

$$dQ_{gw} = h_{gw} R_1 \int_{0}^{\theta_g} \left[ \bar{T}_g - \bar{T}_{w,g}(\theta_1) \right] d\theta_1 \quad 4.46$$

and for the wall in contact with the solids, $dQ_{ws}$, by

$$dQ_{ws} = h_{ws} R_1 \int_{0}^{\theta_s} \left[ \bar{T}_{w,s}(\theta_2) - \bar{T}_s \right] d\theta_2 \quad 4.47$$

Because there is a distribution of temperature in the wall, a mean wall temperature must be defined in order to evaluate a heat balance. A mean wall temperature can be defined with reference to a heat balance for the wall in contact with the gas and solid, equations 4.48 and 4.49 respectively (see figure 4.6).

![Diagram of Temperatures of the Wall in Contact with Gas and Solid](after Gardeik and Jeschar (1979)).

$$\theta_g \cdot \bar{T}_{w,g} = \int_{0}^{\theta_g} T_{w,g}(\theta_1) d\theta_1 \quad 4.48$$
Integrating equation 4.48 with 4.43:

\[
\theta_s \cdot \bar{T}_{w,s} = \int_0^{\theta_s} \bar{T}_{w,s} (\theta_z) \, d\theta_z
\]

4.49

\[
\theta_g \cdot \bar{T}_{w,g} = \int_0^{\theta_g} \bar{T}_g + (\bar{T}_s - \bar{T}_g) \frac{1 - E_{gs}}{1 - E_{gs}} \exp \left( - \frac{E h_{gw}}{E_{gs}} \right) \, d\theta_1
\]

4.50

Yields

\[
\bar{T}_{w,g} = \bar{T}_g + \frac{(\bar{T}_s - \bar{T}_g)}{E h_{gw} \theta_g} \exp \left( - \frac{E h_{gw}}{E_{gs}} \right)
\]

4.51

Where

\[
E = \frac{(1 - E_g)(1 - E_s)}{1 - E_{gs}}
\]

Similarly, integration of 4.49 with 4.42 results in equation 4.52:

\[
\bar{T}_{w,s} = \bar{T}_s + \frac{(\bar{T}_g - \bar{T}_s)}{E h_{wc} \theta_s} \exp \left( - \frac{E h_{gw}}{E_{gs}} \right)
\]

4.52

Performing an energy balance on the gas stream:

\[
\dot{\omega}_g \frac{d \bar{T}_g}{dx} + \frac{d Q_{gs}}{dx} + \frac{d Q_{gw}}{dx} = 0
\]

4.53

\[
\dot{\omega}_g \frac{d \bar{T}_g}{dx} + h_{gs,eff} (\bar{T}_g - \bar{T}_s) 2 R_i \sin \left[ \frac{\theta_s}{2} \right] + h_{gw} R_i
\]

\[
\int_0^{\theta_g} \frac{\theta_g (\bar{T}_g - \bar{T}_s)}{E h_{gw} \theta_g} \, d\theta_1 = 0
\]

4.54

Resulting in

\[
\dot{\omega}_g \frac{d \bar{T}_g}{dx} + \left[ 2 h_{gs,eff} R_i \sin \left( \frac{\theta_s}{2} \right) + \frac{h_{gw} R_i E}{B h_{gw}} \right] (\bar{T}_g - \bar{T}_s) = 0
\]

4.55
re-calling that \( B = \left[ \frac{R_1}{R_o - R_1} \right] \left[ \frac{1}{V_w \rho_w} \right] \) equation 4.55 becomes:

\[
\frac{d T_g}{\omega_g} + \left[ 2 h_{gs, eff} R_i \sin \left( \frac{\Theta S}{2} \right) + V_w C_w \rho_w (R_o - R_i) E \right]
\]

\[
(\bar{T}_g - \bar{T}_s) = 0 \tag{4.56}
\]

Similarly an energy balance on the solids stream gives

\[
\frac{d T_s}{\omega_s} + \frac{d Q_{gs}}{\omega_s} - \frac{d Q_{ws}}{\omega_s} = 0 \tag{4.57}
\]

\[
\frac{d T_s}{\omega_s} - 2 h_{gs, eff} R_i \sin \left( \frac{\Theta S}{2} \right) (\bar{T}_g - \bar{T}_s) - h_{wc} R_i \]

\[
\int_0^{\Theta_S} \frac{(\bar{T}_g - \bar{T}_s)}{h_{wc} \Theta_s} E \, d\Theta_s = 0 \tag{4.58}
\]

resulting in

\[
\frac{d T_s}{\omega_s} + \left[ 2 h_{gs, eff} R_i \sin \left( \frac{\Theta S}{2} \right) + V_w C_w \rho_w (R_o - R_i) E \right]
\]

\[
(\bar{T}_s - \bar{T}_g) = 0 \tag{4.59}
\]

By comparing the coefficients of equations 4.58 and 4.1, and 4.59 and 4.2, yields the relationship for \( K_{eff} \):

\[
K_{eff} = h_{gs, eff} + \frac{V_w C_w \rho_w (R_o - R_i) E}{2 R_i \sin \left( \frac{\Theta S}{2} \right)} \tag{4.60}
\]

With the aid of equation 4.51, \( K_{eff} \) can be rearranged to give:

\[
K_{eff} = h_{gs, eff} + \frac{h_{gw} \Theta g}{2 \sin \left( \frac{\Theta S}{2} \right)} \left[ \frac{T_{w,g} - \bar{T}_g}{\bar{T}_s - \bar{T}_g} \right] \tag{4.61}
\]
Keff comprises two terms: the first term characterises the direct heat transfer between the gas and solid, the second term takes account of the indirect transfer of heat.
5 DESCRIPTION OF APPARATUS AND EXPERIMENTAL TECHNIQUES

5.1 Kiln Chain Model

5.1.1 The Kiln Tube

The kiln chain model \((L = 4 \text{ ft}, D = 1 \text{ ft})\) is a \(\frac{1}{8}\)th scale model of the chained section of Chinnor No. 1 kiln \((L = 32.25 \text{ ft}, D = 8.875 \text{ ft})\) of the Rugby Portland Cement p.l.c. For modelling conditions to be satisfied, the voidage in the model must be equal to that of the real system and hence the chained density must be equal in both systems (see section 3.3.1). For the model, the scale chain chosen (Carter, 1981) from commercially available sizes was a mild steel, oval link type \((1.9 \text{ mm} \times 18 \text{ mm})\). After construction the model chainbank (see section 5.1.2) weighed 10.85 kg, with a chaining density of 3.45 kg/ft\(^3\). The real system has a chaining density of 3.59 kg/ft\(^3\) (Carter), hence the chaining density of model and industrial counterpart are virtually equal. The flow diagram of the experimental rig is set out in figure 5.1.

The kiln model comprises a mild steel seamless cylinder through which furnace flue gas flows counter-currently across a DRY bed of raw cement meal nodules. Plate 5.1 is a general view of the chainbank rig, showing control panel and instrumentation.

The kiln tube has a wall thickness of 19 mm, an inside diameter of .304 m and is 1.2 m in length. The kiln is angled at approximately three degrees to the horizontal, the lower end, being the solids outlet is fitted with a one inch high dam.

Two steel tyres are welded to the kiln circumference. These tyres are supported on brass rollers and locating grooves in the rollers preventing axial movement of the kiln body (Plate 5.2). The kiln assembly is driven by a variable speed D.C. electric motor, via a gear ring bolted around the model shell.
Figure 5.1: Chain Rig Flowsheet
5.1.2 The Chainbank

The chainbank (Plate 5.3) is built to a scale of approximately 1/8th and is based on the chained section of Chinnor No. 1 kiln of the Rugby Portland Cement Group.

A three spiral start is modelled using a cylindrical cage to support the three spiral hangers. Each hanger is formed from 3.2 mm square section rod and is silvered soldered to the cage at each contact point so that the spirals are held rigidly. The spiral direction is clockwise when viewed from the solids feed end; the kiln is driven in the same direction. The spiral starts are mutually spaced at 120° and the spiral pitch is 1 inside kiln diameter.

Each chain top link is silvered soldered to the inside face of the spiral hanger and the other end allowed to hang freely. The gap between each successive chain is 1/2 inch, and there are approximately 240 chains in each spiral, the total number of chains being 720 in the chainbank. The chainbank is a close fit inside the kiln and is easily installed or removed from the kiln.

5.2 The Vortex Combustor

5.2.1 Mark I Combustor

The Mark I design (Plate 5.4) is an improved version of that used by Tate (1982) for shaping glass beads, this combustor being unlined. This lack of refractory lining led to several problems when trying to achieve the desired inlet temperature (400°C) to the model kiln. These difficulties were:

(a) the appearance of localized hot spots on the combustor chamber and in the elbow of its flue.

(b) poor flame stabilization.

(c) only low flowrates of air/natural gas were possible; any increase resulted in (b).

(d) an uncontrollable exhaust temperature.
5.2.2 Mark II Combustor

The Mark I combustor was subsequently refractory lined. The combustor body was lined with ram refractory as was the lid, base and inlet arm. To the base was added a refractory moulded internal chimney (wall thickness 2 cm., i.d. 12 cm, height 10 cm). This internal chimney prevents short circuiting and increases the recirculation of the gaseous mixture within the chamber.

The combustor's flue which carried the hot gas from the combustor to the inlet of the model kiln was completely lined with H.T. bricks. To the flue, at its elbow, was added an air dilution arm (see Plate 5.5 and Figure 5.2).

The refractory lining of the combustor body and flue increased the range of operating conditions of the combustor, and aided the elimination of any hot spots.

The air dilution arm increased the gas flowrate through the kiln four fold. Any further increase of air flowrate through the air dilution arm resulted in vortex instability.

By a series of test runs for a given air to fuel ratio to the vortex combustor the air dilution flowrate was varied till the desired inlet temperature of 400°C to the kiln model was achieved, steady-state conditions being obtained in 60 - 70 minutes.

5.2.3 Vortex Combustor Instrumentation and Burner Control

A Teckni BAL burner control unit is the heart of the instrumentation (figure 5.3). At start up, both the main burner air valve and gas valve are closed. A signal from the pressure switch is fed back to the unit BAL, the primary air fan being on.

When current is supplied to the unit BAL and the 'start' button is pressed, the pilot solenoid valve and the spark generator are simultaneously energised. With the pilot established, releasing the
Vortex combustor and flue.

Combustor chamber: 16swg mild steel construction lined with 3cm ram refractory.

Internal chimney: 12cm i.d. x 2cm cast refractory.

Flue lined with h.t bricks.

Principal dimensions, shown all in cm.
Figure 5.3
FLOW DIAGRAM OF BURNER CONTROL

AIR
pressure switch
micromanometer
main burner air valve
vortex combustor
pilot
rotameter
flame trap
regulator
main solenoid valve
non-return valve
gas valve
main burner gas valve
main solenoid valve
flame safeguard control BA1
flame detector
interlocks
neon illuminator
start
stop
ignition transformer
'start' button energises the main gas solenoid valve. If a flame is present at the start of the cycle, the control will go to lockout and indicating neon light will be illuminated. In this state the main gas valve is opened along with the main air valve till the desired firing conditions are obtained, also the dilution air fan is turned on.

An annubar pitot-rake in the air line is connected to a micromanometer, indicating the air flow. A rotameter indicates the natural gas flow. A pitot tube placed in the centre of the dilution air line and also connected to a micromanometer indicates the dilution air flow.

If the pilot failed to ignite within three seconds the main gas valve is closed and main air valve is fully opened and manual purging commences for ten seconds. After this period the main air valve is fully closed and the lighting up procedure repeated.

5.3 Solids Handling System

5.3.1 Bin-Hopper System

This simple bin-hopper system comprises a standard oil drum with its base joined to a conical hopper. It was found that our simple bin-hopper used to feed the kiln was inadequate, the solids flow from the system being erratic. This irregular flow being attributed to dynamic arch formation and visual observations of rat-holing, which are characteristics of a funnel-flow behaviour.

5.3.2 Bin-Hopper

For arching to be prevented and flow to be kept continuous, the strength of any incipient arch must be less than the forces tending to break it. These forces can commonly be considered to be gravitational, although a differential gas pressure across an arch is a possible additional force.

Fixed rigid inserts are generally used inside a bunker to help improve vessel performance during both filling and discharge of material.
Johanson (1967) suggests that inserts of an appropriately chosen shape when placed at a certain critical height above the hopper outlet may reduce the size of stagnant zones observed in a funnel-flow bunker considerably, thus enhancing uniform flow behaviour.

Richards (1966) reports the use of a fixed rigid insert. It consisted of two cones, one with its apex up and one with its apex down, which were joined at their bases.

5.3.3 Design of a Flow Regulator

For the design of the flow regulator careful attention must be paid to the geometry of the hopper and the depth at which arching occurred in the hopper. The outlet aperture of hopper was 16.2 cm and the included angle 62°. It was found that arching generally occurred at about 23 cm below the hopper level. Although the arch was covered by material above it, during observations if a pipe was inserted into the hopper, the material above the arch, on prodding with the pipe would give way, revealing the depth of the arch in the first instance.

For simplicity the slopes of the cones were chosen to be the same as the hopper angle (62°).

The base diameter of the double cone was set at 16 cm, which is virtually the same as outlet aperture diameter (16.2 cm) of the hopper.

5.3.4 Application of Differential Air Pressure Across the Material

The apex of both cones were to be left open to allow the passage of 1/2" B.S.P. pipe. The end of the pipe would end in a cross, each arm of the cross having a length of 16.2 cm. Each arm was drilled in the horizontal plane with seven holes (dia. = 1/8") spaced 2 cm apart. The cone was then slotted on to the pipework, the lower apex resting on the cross configuration and the upper apex held in position by a collar. Figure 5.4 illustrates the bin-hopper system with solids flow regulator. Low pressure air was used, and fluidization was avoided.
Figure 5.4  Bin-hopper system with solids flow regulator
5.3.5 Solids Feeding

A variable speed screw feeder (Simon-Solitec) was used to feed the nodules to the kiln tube via a chute, a graduated hand wheel dial controlling the speed of the screw feeder. The screw feeder was calibrated against the settings on the hand wheel dial, by a 'bucket and stopwatch' method. The screw feeder provided an accuracy of ± 4.5% for feed rates at 0.3 kg/min and ±2.5% for feed rates set at 1.0 kg/min.

5.4 Flowrate Measurement and Control

The total gas flow to the chain model comprises three separate gas flow lines:

(a) primary air flow to vortex combustor

(b) natural gas flow to vortex combustor

(c) dilution air.

5.4.1 Primary Air Flow

This is the air required for combustion inside the vortex combustor, the flowrate being measured by a stainless steel Annubar pitotstatic flow element. The differential pressure from the Annubar is measured by a portable pressure and flow analyser. This micromanometer has a range of 0 to 500 mm Wg. Calibration of the Annubar pitotstatic flow rake was made against a standard pitot tube, in accordance with BS 1042 Part 2A (1973). The air flowrate was controlled by altering damper "A" (figure 5.1).

5.4.2 Natural Gas Flow

The natural gas flow to the vortex combustor was through a series 35X (metric) rotameter. The manufacturer's calibration chart was checked with a Dresser meter; a negligible discrepancy was found.
5.4.3 Dilution Air Flow

The dilution air is required to control the temperature of the flue gas from the vortex combustor and to boost up the mass flow of gas through the chain model. A ten point pitotstatic transverse was made across the air dilution line, the average differential pressure reading to the mid-point reading being virtually unity. In accordance with BS 1042 a standard pitot tube was placed in the centre of the air dilution line, and held in position by a vernier mounting clamp. The dilution air flowrate was controlled by altering damper "B" (figure 5.1). The differential pressure from the standard pitot tube was measured on a micromanometer.

5.5 Temperature Measurement

5.5.1 Chain, Solids and Gas Temperature

Ten thermocouples, 10 cm apart are mounted at top dead centre. Each thermocouple is sealed by glands screwed through the kiln shell. The thermocouples are 3 mm o.d. stainless steel sheathed and have been proven to have the same thermal inertia as a hanging chain (Patterson, 1980). Each thermocouple tip extends approximately 25 mm inside the kiln and is mounted at right angles to the inside kiln wall.

The gas temperature thermocouples are the same type as used for the solids measurement, but extend to the axis of the kiln. Five positions are monitored, 20 cm apart to give the gas temperature profile along the kiln section and the thermocouples are mounted at bottom dead centre (Plate 5.6).

5.5.2 Shell Temperatures

Shell temperatures are monitored by using flexible, fibreglass insulated, exposed junction chromel-alumel thermocouples of wire diameter 0.193 mm. A thermocouple is held in position by a copper spring clip which is attached to the shell surface by a self tapping screw.
5.5.3 Thermocouple Monitor Unit

To enable measurements of temperatures from thermocouples placed on a rotating kiln model and keep slip ring contact to a minimum, a system was developed using a Universal Asynchronous Receiver Transmitter (UARTS) type 6402, to transmit serial data from the control and data logging system to a selector system on the kiln.

The transmitter unit allows manual or automatic selection of one of sixteen thermocouples. A thumbwheel switch on the data logger feeds a BCD code to the UART which converts the parallel BCD code to serial data string which is sent repeatedly to the selector unit via the slip rings. An identical UART (Plate 5.7) on the kiln receives the serial data and converts it back into its original BCD code. The UART also checks to see if the data has been received correctly (i.e. parity, framing and overrun errors) and is cleared to receive the next set of data. A one of sixteen decoders type 74154 is used to convert the BCD code to select one thermocouple through a reed relay selector unit to an ANCOM thermocouple and linearization circuit which converts the small emf from the thermocouple to give a proportional signal of 10 mV per °C. This signal is returned via the slip rings to the data logger.

Power to the selector unit is also fed as unregulated DC through the slip rings and is regulated in the unit to give +15 V, -15 V and +5 V DC.

5.5.4 Thermocouple Slip-Ring Assembly

The slip-ring assembly is a precision-built unit of machined copper rings which are gold plated and sandwiched in a perspex holder (Plate 5.8). The brushes are constructed from low resistance carbon, and are spring loaded against the slip-rings, the entire unit being enclosed in a perspex box to minimise dust pick-up.

The assembly is driven by a drive shaft (Plate 5.6), through a gland in one end of the model exit flue. The drive shaft houses the electrical
connections to the slip-rings and is driven by the kiln as it rotates. Compressed air is blown via the slip-ring assembly through the drive shaft and out along one arm of the spider arrangement. This ensures that none of the electrical connections are overheated.

5.5.5 Data Logging System

A 3530 Orion data logging system is used to monitor and store the thermocouple readings. It is programmable and allows the setting up of tasks (subroutines) up to a maximum of eight. Each task can be triggered to call on another task. The trigger function is a microswitch on the slip-ring assembly (Plate 5.8).

Each task can hold eight binary patterns. The thermocouple numbering scheme (figure 5.5) employed starts from the hot end of the kiln, and progresses to the cold end. These Roman numbers are assigned binary values, and these binary numbers are then programmed into two tasks (eight in each). As the kiln rotates the microswitch is struck and triggers the logging procedure (figure 5.6). Each thermocouple is read 12 times in one revolution. After sixteen revolutions the logging procedure is completed.

All thermocouple measurements are stored on a magnetic cartridge. An RS232 interface provides transference of data on the magnetic cartridge to the University main frame computer.
Figure 5.6
Flow diagram of data logging procedure

1. Start
2. Time delay
3. Bit pattern
4. Micro switch
5. Rotation
6. Time delay
7. Scan complete
8. Run

Tasks

Triggers
6. RESULTS

6.1 Treatment of Results

6.1.1 Calculation of Heat Balances

From an enthalpy balance taken over the kiln as illustrated in figure 6.1, the following equations can be derived for the gas and solid phases respectively:

\[
Q_g = Q_{gw} + Q_{gs} \tag{6.1}
\]

\[
Q_s = Q_{gs} + Q_{ws} \tag{6.2}
\]

where

\[
Q_g = \dot{m}_g c_g (\Delta T)_g
\]

\[
Q_s = \dot{m}_s c_s (\Delta T)_s
\]

\[
Q_{gw} = h_{gw} A_{gw} (T_g - T_w)
\]

\[
Q_{gs} = h_{gs} A_{gs} (T_g - T_s)
\]

\[
Q_{ws} = h_{ws} A_{ws} (T_w - T_s)
\]
and

\[(\Delta T)_g\] - temperature drop of the gas stream

\[(\Delta T)_s\] - temperature rise of the solid stream

Substitution of equations 6.3 into 6.1 and 6.2 reveals the fact that there are three unknown variables namely the heat transfer coefficients (h) existing in two independent equations.

As stated in section 1 the emphasis of this work is to maximize the enthalpy loss of the gas stream within the kiln. Because of the intractable nature in trying to evaluate the three individual heat transfer coefficients, instead, an overall heat transfer coefficient is determined (see section 4.4).

6.1.2 Heat Balance Deviation

An overall heat balance (see figure 6.1) on the kiln can be written down quite simply as:

\[Q_g - Q_s = Q_l\]  \hspace{1cm}  6.4

where \(Q_l\) is kiln shell heat losses. A heat balance deviation (error) is defined as:

\[\text{error} = 100 \left( \frac{Q_s + Q_l - Q_g}{Q_g} \right)\]  \hspace{1cm}  6.5

Negative deviations from the heat balance indicate that more heat was calculated to be given up by the gas stream than was gained by the solids and lost through the wall.

6.1.3 Prediction of Kiln Shell Heat Losses

These losses are essentially the heat lost to the atmosphere by conduction through the kiln shell. Providing, that the system is under steady state conditions, the losses by conduction through the shell can be directly related to the radiative \((Q_R)\) and convective \((Q_C)\) heat losses from the shell surface:
\[ Q_{R} = Q_{R} + Q_{C} \]  

where \( Q_{R} \) is given by:

\[ Q_{R} = \sigma e (T_{K}^{4} - T_{A}^{4}) A_{S} \]

\( A_{S} \) - shell surface area  
\( \sigma \) - Stefan-Boltzmann constant  
\( e \) - shell emissivity  
\( T_{K} \) - shell temperature  
\( T_{A} \) - ambient air temperature

The emissivity of heavily dust coated steel shells is taken as 0.88 (Patterson, 1980). The convective component is given by:

\[ Q_{C} = h_{C} A_{S} (T_{K} - T_{A}) \]

The heat transfer coefficient \( (h_{C}) \) for natural convection of air for horizontal pipes is given as (Coulson and Richardson, 1977):

\[ h_{C} = 1.65 (T_{K} - T_{A})^{0.25} \]

6.1.4 Evaluation of the Overall Heat Transfer Coefficient

Recalling equation 4.61:

\[ K_{eff} = h_{gs} + h_{gw} \frac{\theta_{g}}{2 \sin \left( \frac{\theta_{g}}{2} \right)} \left[ \frac{T_{w,g} - T_{g}}{T_{g} - T_{w,g}} \right] \]

It can be seen from the above equation that the terms relating to \( K_{eff} \) are directly measurable. The included angle \( (\theta_{g}) \) of the bed is evaluated from the chord lengths of the bed (see figure 6.2). The chord lengths are measured at the exit and at the feed end of the kiln, the average value is taken to determine \( \theta_{gs} \):

\[ \theta_{gs} = 2 \sin^{-1} \left( \frac{\ell_{ave}/2}{R} \right) \]
6.1.5 Evaluation of Gas to Wall Heat Transfer Coefficient

These tests were conducted without solids feeding in order to isolate and characterise the heat flow rate between the gas and the wall in the kiln.

The gas to wall heat transfer coefficient being evaluated from:

\[ h_{gw} = \frac{Q}{A_{gw} (\Delta T)_{gw}} \]  

where

\[ Q = \dot{m} c_g (\Delta T)_g \] - the enthalpy of the gas stream

\[ \Delta T_{gw} = \bar{T}_g - \bar{T}_{sh} \]

\[ \bar{T}_g = \frac{1}{2} (T_{gas in} - T_{14}) \] - the average gas temperature

\[ \bar{T}_{sh} = \frac{1}{3} (T_0 + T_8 + T_{15}) \] - the average shell temperature

\[ A_{gw} = (2\pi - \theta_s) RL \]
These experiments were performed with the kiln chained (CK), and unchained (EK), see Appendix A.

6.1.6 Evaluation of Gas to Solids Heat Transfer Coefficient

Once the gas to wall heat flow rate is characterised, even though, for an empty kiln, the gas to solids heat flow term can be isolated, using equation 6.1:

\[ Q_{gs} = Q_g - Q_{gw} \]  

6.14

The gas to solids heat transfer coefficient is thus determined:

\[ h_{gs} = \frac{Q_{gs}}{(T_g - T_s) A_{gs}} \]  

6.15

\( T_g \) = Average gas temperature

\( T_s \) = Average solids temperature

\( A_{gs} \) = \( \bar{l} \) ave \( \cdot \) L

6.1.7 Gas to Chain Heat Transfer Coefficient

Patterson (1980) in his study of chains system in wet process kilns, assumed that a chainbank can be represented by a bank of tubes. This assumption was necessary in order to evaluate the radiative exchange areas within the kiln. This section of work aims to clarify Patterson's assumption.

A response technique was used to evaluate the gas to chain heat transfer coefficient. The technique employed is based on the method of Looney (1957).

Looney related the first order time constant to the heat transfer coefficient as follows:
\[ T = \frac{1}{h} \left( \frac{C_w}{\pi D_o} \right) \quad 6.16 \]

\[ C_w = \frac{\pi}{4} \rho C_p D_o^2 \quad 6.17 \]

\( D_o = \text{outside diameter of chain link} \)

Equation 6.16 is a simple rearrangement of the conventional definition of the first order time constant (Benedict, 1977).

Looney extracted \( h \) from:

\[ \frac{h D_o}{K} = C_1 \text{Re}^m \text{Pr}^n \quad 6.18 \]

and substituted \( h \) into equation 6.16

\[ T = \left( \frac{C_w \cdot D_o}{\pi D_o \cdot C_1 K} \right) \frac{1}{\text{Re}^m \cdot \text{Pr}^n} \quad 6.19 \]

Thus a plot of \( T \) against Re should enable the exponent \( m \) to be evaluated.

In order to determine the first order time constant, a bare wire chromel-alumel thermocouple was silver soldered to a chain end link (see Figure 6.3) and a separate thermocouple to the chain top link. The thermocouple output was fed directly into the Orion data logger, the data being read at 100 readings per second. The procedure was to obtain a constant temperature output from the vortex combustor, once this was achieved, the chain was placed inside the kiln and the output from the thermocouples logged. This procedure being repeated at different flowrates through the kiln.
203.2 mm.

ie. 2/3 KILN DIAMETER

THERMOCOUPLPE TYPE:
Exposed junction NiCr/NiAl
Fibreglass insulated
Wire diameter 0.193 mm.
Comark Ltd.

THERMOCOUPLE JUNCTION
SILVER-SOLDERED TO CHAIN END LINK

Figure 6.3 Thermocouple and chaining attachment
(After Patterson (1980))
6.2 Results of the Experimental Investigation

6.2.1 Determination of Steady-State

Before any solids feeding tests were undertaken empty kiln runs were performed to determine the time required for the attainment of steady state. Figure 6.4 is a plot of the vortex outlet (gas in) temperature and the kiln exit (gas out) temperature plus the probe thermocouples set in the kiln (see section 5.5.5). This figure represents the operating conditions of the Mark I combustor, and, as can be seen the desired gas in temperature of $400^\circ\text{C}$ is not attained with the system not reaching steady state. Figure 6.5 shows the performance of the Mark II combustor. The desired temperature being achieved and steady-state running conditions obtained after approximately 70 minutes.

6.2.2 Chain Response

Figures 6.6 to 6.7 represent the chain response curves at different flowrates. Experiments were also performed on 3 mm o.d. chromel-alumel thermocouples so that a qualitative comparison of the chain response curves against a standard thermocouple could be appreciated (see Figure 6.8).

The correlation of the first order time constant ($\tau$) against the Reynolds number yielded:

$$\tau = 70.79 \times Re^{-0.409}$$  \hspace{1cm} 6.20

Using the following values for the chain link material properties (i.e. mild steel):

$$\rho = 7833 \times \text{kg/m}^3$$
$$C_p = 465 \times \text{J/Kg.K}$$
$$D_0 = 1.9 \times \text{mm}$$

and the thermal conductivity of air being taken as:

$$k = 0.0425 \times \text{W/m.K}$$
Figure 6.4  DETERMINATION OF STEADY-STATE (Mk. I)
Figure 6.5  DETERMINATION OF STEADY-STATE(Mk.II)
Figure 6.6  CHAIN RESPONSE
Figure 5.7  CHAIN RESPONSE

- 89 -
Figure 6.8  THERMOCOUPLE RESPONSE
the term in parenthesis of equation 6.19 is evaluated

\[
\frac{C_w D_o}{\pi D_o C_k k C_1} = 77.34
\]

Equating equation 6.19 to 6.20, enables \( C_1 \) to be found:

\[
70.79 \text{Re}^{-0.409} = \frac{77.34}{C_1} \text{Re}^{-m} \text{Pr}^{-n}
\]

Assuming the \((\text{Pr})^{0.33}\) number is virtually unity, then by equating coefficients of \( \text{Re} \), enables \( C_1 \) to be determined:

\[
C_1 = \frac{77.34}{70.79} = 1.09
\]

while, obviously, \( m = 0.409 \)

Substituting these values into equation 6.18, results in the Nusselt number for the gas to chain interaction:

\[
\text{Nu} = 1.09 \text{Re}^{0.409} \text{Pr}^{0.33} \quad 6.21
\]

Taking the \( \text{Pr} \) number to be 0.7 simplifies the above equation to:

\[
\text{Nu} = 0.969 \text{Re}^{0.409} \quad 6.22
\]

6.2.3 Gas to Wall Convective Heat Transfer Coefficient

The relationship between the dimensionless Nusselt and Reynolds groups based on the heat transfer between the kiln wall and the gas flowing through the model is presented in Figure 6.9. Two relationships are shown:
Figure 6.9 \( \log_{10}(\text{Nu}(\text{gas:wall})) \) vs. \( \log_{10}(\text{Re}) \)
(i) the model operating with a chain system (gas: chained wall)

(ii) the kiln running with no chains (gas: wall)

For case (i) this relationship has been correlated to yield an equation of the form

\[ \text{Nu} = 0.00661 \text{ Re}^{0.15} \text{ Pr}^{0.33} \]  \hspace{1cm} 6.23

The individual heat transfer coefficients contained in the Nusselt group of equation 6.23 has also been correlated to yield:

\[ h = 0.000759 \text{ Re}^{0.16} \] \hspace{1cm} 6.24

\( h \) being in the range 30 - 80 W/m².K for the Reynolds number 10,000 - 20,000.

For case (ii) the results are:

\[ \text{Nu} = 0.214 \text{ Re}^{0.772} \text{ Pr}^{0.33} \] \hspace{1cm} 6.25

and

\[ h = 0.0295 \text{ Re}^{0.761} \] \hspace{1cm} 6.26

The range of \( h \) being 33 - 55 W/m².K for the same Reynolds number range as for case (i).

6.2.4 Solid and Gas Temperature Profiles

Figures 6.10 and 6.13 illustrate typical axial temperature distributions found along the kiln and rotating temperature profiles at the mid-point of the kiln. These figures represent steady-state conditions. Steady-state operation was confirmed by monitoring the one inch probe thermocouples (see Plate 5.6) continuously on the Orion data logger. Once the cyclic profiles (figures 6.11 and 6.13) were reproducible then steady-state is ascertained. Attention being paid to the feed, mid-point and exit end of the kiln.
Figure 6.10  AXIAL TEMPERATURE PROFILES (Re=19000)
Figure 6.11  ROTATING TEMPERATURE PROFILES (Re=19000)
Figure 6.12  AXIAL TEMPERATURE PROFILES (Re=15000)
Figure 6.13  ROTATING TEMPERATURE PROFILE (Re=15000)
6.2.5 Shell Losses and Heat Balance

Table 6.1 represents the heat balances performed on the model kiln operating with the chainbank. The percentage error is nominally in the range ± 20%, with an overall average value of ~ 5.6%. The heat flow rates are expressed in kilowatts.

6.2.6 The Overall Heat Transfer Coefficient

The parameters necessary to evaluate the overall heat transfer coefficient, $K_{eff}$ (equation 6.10) are presented in Table 6.2. CK denotes the kiln model equipped with the chainbank and EK the model running with no chains.

Using the data presented in Table 6.2, $K_{eff}$ is computed and is presented, along with the kiln operating conditions in Table 6.3.

The essential results, namely $K_{eff}$ and Re are presented as a log-log plot in figure 6.14. A correlation of $K_{eff}$ against Re gave:

(i) for the model operating with chains

$$K_{eff} = 0.002188 \text{Re}^{1.28}$$

(ii) with no chains

$$K_{eff} = 0.1276 \text{Re}^{0.839}$$

These correlated lines are superimposed on the data of figure 6.14.

6.2.7 Comparison of Gas Enthalpy Loss

Figure 6.15 is a plot of the enthalpy lost by the gas stream against the enthalpy gain of the solids. The data being obtained with the kiln chained and unchained.

The enthalpy is in fact a specific enthalpy and is calculated from the energy change for each stream divided by its mass flowrate.
Table 6.1 Shell Losses and Heat Balance

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Table 6.2 Parameters needed to evaluate $K_{eff}$

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Table 6.3 Results for $K_{eff}$ plus Kiln Operating Conditions

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Figure 6.14  LOG(KeFF) vs. LOG(Re)

+ with chains
v without chains
Figure 6.15  COMPARISON OF CHAINED AND UNCHAINED KILN
6.2.8 Solids Motion: Mean Retention Time

The measured mean retention time (T) is calculated from the holdup (H) and feedrate (F) as follows:

\[ T = \frac{H}{F} \]  \hspace{1cm} 6.29

Table 6.4 is a list of measured mean retention times, and predicted values, calculated from the Sullivan et al (1927) equation:

\[ T = \frac{1.77 L \sqrt{\beta}}{2 R N \alpha} \]  \hspace{1cm} 6.30

where

- \( L = 1.2 \text{ m} \)
- \( R = 0.1524 \text{ m} \)
- \( \alpha = 3^\circ \)
- \( \beta = 37^\circ \)
Table 6.4 Mean Retention Time

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</table>
7. DISCUSSION

7.1 Solid and Gas Temperature Profiles

Figures 6.10 and 6.12 illustrate typical axial temperature distributions found along the kiln. In these typical runs the temperature profiles are not linear. In the gas phase, the flow is non-uniform as a result of the vortex combustor and air dilution injection into the kiln. Within the kiln the spider arrangement (see Plate 5.6) acts as an obstruction to the flow of gas, and recirculation of the hot gases is a possibility. The effect of gas flowrate (Reynolds number) on the temperature profiles is also depicted in these figures. The gas entering the kiln was kept at the same temperature, but the outlet gas temperature is lower for the smaller value of the Reynolds number. Also, with the chain system the outlet gas temperature is lower, and the solids outlet temperature is higher, when compared to the kiln running without the chain bank.

One of the basic assumptions underlying the modelling of a rotary kiln is that bed temperature in the radial direction is uniform, although the surface temperature is known to be higher. This assumption is tested out in figures 6.11 and 6.13. With the chainbank, the temperature profile is virtually flat, whereas, with no chainbank there is a gradient within the bed. These two conditions illustrate the "well-mixed" (with chains) and "non-mixed" (no chains) cases highlighted by the theoretical work of Imber and Paschkis (1960, 1962).

7.2 Solids Motion

Table 6.4 shows quite significantly that the bed velocity with the chain system is approximately twice as great as when compared with the kiln without the chainbank. The measured retention times, again, for the chained system are in good agreement with the formula of Sullivan et al (1927). The chain system provides effective radial mixing of the bed, which can be seen from the uniformity of radial bed temperatures (figures 6.11 and 6.13).
Much of the reported work in the literature concerning solids flow through rotary devices have typically used material such as sand and limestone, of a very narrow particle size range, where the upper particle size rarely exceeded 1 mm (see section 2.2). The feed material used in this work is raw meal cement nodules of a wide size range (see Appendix A) and is highly friable. Cold studies (Hart et al, 1983) with the model kiln running without the chainbank, have revealed visually the existence of a layer of fines (-1.4mm) right next to the wall. A similar segregation/percolation mechanism (Richards, 1966) has been reported by Wachters and Kramers (1964). These visual studies have shown the bed to be in the kilning/sliding mode. The bed is essentially unmixed, this is also shown by the hot studies, where the radial temperature profiles are not uniform (Figures 6.11 and 6.13).

Zablotny (1965) reviewed a number of mean retention formulae for the flow of granular material through rotary devices and highlighted that the basic assumption in the development of these formulae is the requirement that the bed velocity is constant at any point in the kiln, once mass flow steady-state has been achieved. From the axial temperature profiles (figures 6.10 and 6.12) in the solid phase, it can be seen that the profiles, starting from the feed end do not increase in a uniform way, rather there are 'spikes' in these profiles. This can be interpreted as a change in bed behaviour - a change in the degree of mixing giving rise to a partial mixing between the two limits of the well mixed and non-mixed cases, as discussed by Vaillant (1965). This change in bed behaviour will result in a non-uniform bed velocity through the kiln (Henein et al, 1983a, 1983b).

7.3 Heat Transfer Coefficients

7.3.1 Gas to Chain

The chain response technique yielded the gas to chain (equation 6.22) Nusselt correlation:

\[ \text{Nu} = 0.969 \text{Re}^{0.409} \]
One of the earliest, and most cited correlations for the heat transfer from circular cylinders is that of Hilpert (1933). Morgan's (1975) extensive review of the overall convective heat transfer from smooth circular cylinders has revealed a minor error in the data of Hilpert. Morgan comments that the values used for the thermal conductivity of air up to 1955 are 2-3% lower than those now generally accepted. Morgan's corrections to Hilpert's correlations are presented in Table 7.1.

Table 7.1 Corrections to Hilpert's Correlation for Crossflow Forced Convection from Cylinders in Air

<table>
<thead>
<tr>
<th>Hilpert Re</th>
<th>Corrected Re</th>
</tr>
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<tr>
<td></td>
<td>From</td>
</tr>
<tr>
<td>1-4</td>
<td>0.891</td>
</tr>
<tr>
<td>4-40</td>
<td>0.821</td>
</tr>
<tr>
<td>4-10⁴</td>
<td>0.615</td>
</tr>
<tr>
<td>4-10⁴</td>
<td>0.174</td>
</tr>
<tr>
<td>4-10⁵</td>
<td>0.0239</td>
</tr>
</tbody>
</table>

\[ \text{Nu} = D₂ \text{Re}^{n₁} \]
Zukauskas (1972) has also investigated the heat transfer from a circular tube in crossflow with viscous fluids and gases, the correlated heat transfer data is presented in Table 7.2.

Table 7.2 Zukauskas' Correlation

<table>
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<tr>
<th>$Re$</th>
<th>$C$</th>
<th>$m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 - 40</td>
<td>0.75</td>
<td>0.4</td>
</tr>
<tr>
<td>40 - 1x10^3</td>
<td>0.51</td>
<td>0.5</td>
</tr>
<tr>
<td>1x10^3 - 2x10^5</td>
<td>0.26</td>
<td>0.6</td>
</tr>
<tr>
<td>2x10^5 - 1x10^6</td>
<td>0.076</td>
<td>0.7</td>
</tr>
</tbody>
</table>

$$Nu = C Re^m Pr^{0.37}$$

In addition, Zukauskas presents a correlation, in the range of $Re$ from 10 to 100, for the heat transfer from inner tubes in a staggered tube bank arrangement, and is expressed by:

$$Nu = 0.9 Re^{0.4} Pr^{0.37}$$  \[7.2\]

Comparison of tables 7.1 and 7.2 highlights discrepancies in these experimental correlations from different observers. Kestin (1966) attributes the many such discrepancies to the effect which free-stream turbulence has on the rate of heat transfer.

From the laws of similarity the standard functional relation is obtained

$$Nu = f(Re_1, Pr) \text{ with } Re = U_0 l/\nu$$  \[7.3\]

where the flow is uniform and the free stream can be described by a single velocity $U_0$. But, the essential characteristic of a turbulent free stream resides in the fact that $U_0$ is quasi-steady, i.e. the velocity fluctuates in magnitude and direction with respect to time:

$$U_∞ = \bar{U}_∞ + U'_∞$$
where the vector \( \mathbf{\bar{u}}_0 \) is a function of time and position

\[
\mathbf{\bar{u}}_0 \text{ denotes the uniform, constant, average velocity vector}
\]

\( \mathbf{U}'_\infty \) is a vector whose three components oscillate in a stochastic manner.

The assumption is that \( \mathbf{u}_0 \) can be replaced by \( \mathbf{\bar{u}}_0 \) and that the Reynolds number is formed with its magnitude

\[
Re = \frac{\mathbf{\bar{u}}_0}{\gamma}
\]

From the above discussion, Kestin (1966) concludes by noting that two turbulent streams can never be strictly similar, because only limited control can be exercised over the component \( \mathbf{U}'_\infty \). Kestin advocates the replacement of the elementary result (equation 7.3) by a more precise equation:

\[
Nu = f(Re, Pr, Tu) \quad 7.4
\]

where the longitudinal intensity of turbulence \( (Tu) \) is defined by:

\[
Tu = \left( \frac{\left( \mathbf{\bar{u}}_0 \right)^2}{\mathbf{\bar{u}}_0} \right)^{1/2} \quad 7.5
\]

and \( \mathbf{\bar{u}}_0 \) is the x-component of \( \mathbf{U}'_\infty \) and is time averaged.

Morgan estimates that the free stream turbulence intensity of Hilpert's work to be low. Zukauskas (Table 7.2) gives a value of less than 1% for \( Tu \).

The gas to chain Nusselt correlation \((25 < Re < 63)\), equation 7.1, is in good agreement with Zukauskas correlation (equation 7.2) for the
heat transfer from inner tubes in a staggered tubebank arrangement. This close agreement can be attributed to the similarity of the flow fields existing within the chainbank and a staggered tubebank arrangement.

7.3.2 Gas to Wall

The correlations (see section 6.2.3) for the gas to chained wall and gas to bare wall are respectively:

\[ \text{Nu} = 0.00661 \text{Re}^{0.15} \text{Pr}^{0.33} \]  \hspace{1cm} 7.6

and

\[ \text{Nu} = 0.214 \text{Re}^{0.772} \text{Pr}^{0.33} \]  \hspace{1cm} 7.7

The correlations by the author are presented in figure 7.1 and compared with the standard correlation of Dittus and Boelter (1930) for a fully developed flow system:

\[ \text{Nu} = 0.023 \text{Re}^{0.8} \text{Pr}^{4} \]  \hspace{1cm} 7.8

The degree of enhancement of heat transfer between the gas and wall compared to a fully developed non-swirling flow is quite apparent. This augmentation in the heat transfer rate is attributable to a combination of:

(a) swirling flow

(b) combined hydrodynamic and thermal entry length

(c) entrance configuration.

Zaherzadeh and Jagadish (1975) have examined the heat transfer in decaying swirl flows, where the working fluid was tap water with Reynolds numbers ranging from 20 000 to 100 000. Their correlated results yielded:

\[ \text{Nu} = c \text{Re}^{0.0029} \text{Pr}^{4} \]  \hspace{1cm} 7.9
Figure 7.1 COMPARISON OF NUSSELT NUMBERS
where \( c \) is in the range \( .00423 - .00296 \), and was found to be dependent on the geometry of the swirl generator employed by these researchers.

By far, the most extensive experimental study on measurement of local Nusselt numbers in the entry region of a circular tube for various entry configurations is that of Boelter et al (1948). They used air flowing through a steam-heated tube in which the steam jacket was sectioned in such a way that the local heat transfer rate at each section could be determined from a measurement of the condensate rate, a similar technique was used by Zaherzadeh and Jagadish (1975). An abstract of Boelter's et al (1948) results is plotted in figure 7.2. The ratio of local Nusselt number to fully developed Nusselt number is plotted as a function of \( x/D \) for a Reynolds number of 50,000. These curves represent typical heat-exchanger entrances. The very high Nusselt number corresponds to the 90° mitre bend. As can be seen from Plate 5.5, the vortex combustor employed in this thesis has a flue which is a 90° mitre bend. Kawashima et al (1983) have investigated the heat transfer characteristics of a two-dimensional right angle, T-shaped flow junction, this corresponds to the air dilution arm installed in vortex combustor's flue (see Plate 5.5). Kawashima et al report a ratio of the local Nusselt number to the fully developed one (equation 7.8) of up to 7. They found this ratio to be highly dependent on the geometry of the confluence and the flow rate ratios.

In developing the necessary scaling criteria for a chain system Patterson (1980) likened a chainbank to a series of converging-diverging baffles (see section 3.2). Recently Souza Mendes and Sparrow (1984) reported mass transfer measurements at the entrance region in periodically converging-diverging tubes. By employing a naphthalene sublimation technique they found that the ratio of measured mass transfer coefficients in the entrance region of the periodically converging-diverging tubes to that of a fully developed flow in a straight tube were in the range 1.5 - 4.0. In conjunction with the analogy between heat and mass transfer, their findings indicate that converging-diverging tubes possess favourable enhancement characteristics.
Figure 7.2 EFFECT OF ENTRANCE CONFIGURATION ON LOCAL NUSSELT NUMBER
7.3.3 The Overall Heat Transfer Coefficient, $K_{\text{eff}}$

From Table 6.3, for the chained kiln $K_{\text{eff}}$ is in the range 277 - 711 W/m$^2$.K, and for the kiln with no chains 272 - 530 W/m$^2$.K. These ranges are typical of kilns using industrial feed material reported in the literature (see section 2).

An interesting trend can also be seen from Table 6.3. $K_{\text{eff}}$ appears to be proportional to the holdup till an optimum value is reached and then decreases. This trend has been reported by Kelly (1969) while examining the operating procedures in rotary driers and coolers, and also by Tscheng and Watkinson (1979). This effect can be explained by the equation cited by Bowers and Read (1965), (equation 2.1):

$$\theta = \frac{F_0}{r^2 \rho c}$$

and is a measurement of the ability of the bed to absorb heat from the heat sources within the kiln, i.e. the hot kiln gases and wall. From equation 7.8 it can be seen that an increase in the bed height (i.e. a relative measurement of bed radius, r) increases the heating time $\theta$, causing a decrease in bed thermal acceptance. Thus an increase in holdup decreases the kiln bed thermal acceptance. However, when the holdup is increased, the chord length of the bed is also increased, exposing a larger surface of the material to the heat. This increases the bed thermal acceptance, tending to negate the effect of bed depth. This suggests that the ratio of bed volume to bed surface is a factor in the heat transfer process, (Tscheng and Watkinson).

The Fourier modulus $F_0$, (the relative time ratio), which can be determined from Gurney Lurie (1923) charts, decreases as the temperature of the charge increases. Thus, if additional heat is supplied to the bed, i.e. by the use of chains, will decrease $F_0$ and, therefore, $\theta$, and increases the thermal receptivity of the bed. This can be seen in figure 6.15, it is quite apparent that a chained kiln has a higher thermal receptivity than an unchained kiln.
8. CONCLUSIONS AND RECOMMENDATIONS

8.1 Conclusions

(a) The gas-wall heat transfer coefficients both for the chained system and unchained are considerably higher than for fully developed linear flow (see figure 7.1). When compared to an unchained kiln, a chain system considerably increases the heat flow to the bed (see figure 6.15).

(b) An overall heat transfer coefficient has been derived; essentially based on the analysis of Gardeik and Jeschar, and has been correlated against the tube Reynolds number for the chained system to yield the equation (see figure 6.14):

\[ K_{eff} = 0.002188 \text{ Re}^{0.28} \]

The overall heat transfer coefficient appears to be proportional to the holdup up to an optimum value. This qualitative result has also been reported by Kelly.

(c) The gas-wall heat transfer coefficients are of the same order as those quoted by literature for heat transfer in swirling flows and have been correlated for the chained system to yield the equation:

\[ Nu = 0.00061 \text{ Re}^{0.15} \text{ Pr}^{0.33} \]

(d) The use of a vortex combustor to provide low momentum hot flue gas and the particular inlet geometry to the model kiln, which has been an operational compromise, has resulted in the high values of the gas-wall heat transfer coefficient (see figure 7.2 and section 5.2.2).

(e) The particular chaining arrangement used in this model, i.e. forward screw spiral with forward motion, can be seen to increase the velocity of the feed material through the kiln and hence decrease the residence time and holdup.
(f) The use of a chain system ensures that the bed is in the well mixed condition; this can clearly be seen from the temperature plots and residence time calculations (see figures 6.11 and 6.13). This is the sought after situation resulting in enhancement of heat transfer to the bed.

(g) The residence time calculations agree favourably with the empirical equation of Sullivan et al; this is due to the chain system which ensures that the bed is well mixed.

(h) The difficulties inherent in accurate and noise free measurements from rotating kilns has been overcome by the use of a minimum number of gold slip-rings and a programmable data recorder. The continuous monitoring, gathering, storage and retrieval of temperature data has been considerably simplified.

8.2 Recommendations

Introduction of a chain system in a temperature zone not exceeding 400°C, so as to extend chain life, will ensure that the bed is well mixed which will result in enhancement of heat transfer to the bed. By artificially inducing swirl in rotary coolers (and driers) by the use of vanes will further enhance the overall heat transfer to the bed and will result (for rotary cement coolers) in:

(i) increase in the gas-wall heat transfer
(ii) increase in the secondary air temperature
(iii) decrease in clinker outlet temperature
(iv) reduction in cooler shell losses

The geometry of the vane needed to induce swirl in a rotary device can be found through the technique of acid-alkali modelling which will provide a quick and cheap alternative to costly full scale plant trials.
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APPENDIX A : EXPERIMENTAL RESULTS
Run: CK1

Gas Flow (kg/hr) = 492  
Solids flow (kg/min) = .56  
RPM = 1.0  
Holdup (kg) = 6.924  
Chord length at exit = 18  
Bed height at exit = 2.5  
Chord length at feed = 22  
Bed height at feed = 4.6

<table>
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<th>Time interval (secs) between (T_{min} - T_{max})</th>
</tr>
</thead>
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<td>218</td>
<td>239</td>
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<td>1</td>
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<td>312</td>
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<tr>
<td>15</td>
<td>157</td>
<td>-</td>
</tr>
</tbody>
</table>

Stationary thermocouple

T_{gas in} = 408
T_{solids in} = 34
Run: CK2

Gas Flow (kg/hr) = 496  \quad \text{Re} = 18871
Solids flow (kg/min) = 0.73
RPM = 1.5  \quad \text{Time included between reading} = 3.4
Holdup (kg) = 6.044  \quad \text{Holdup (\%)} = 5.9
Chord length at exit = 19
Bed height at exit = 2.5
Chord length at feed = 21
Bed height at feed = 4.2

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<th>Time interval (secs) between (T_{\text{Min}} - T_{\text{Max}})</th>
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Stationary thermocouple

\begin{align*}
T_{\text{gas in}} & = 411 \\
T_{\text{solids in}} & = 30
\end{align*}
Run: CK3

Gas Flow (kg/hr) = 495 \quad Re = 18883
Solids flow (kg/min) = 95
RPM = 2 \quad Time included between reading = 2.5
Holdup (kg) = 4.924 \quad Holdup (%) = 4.8
Chord length at exit = 20
Bed height at exit = 2.5
Chord length at feed = 18
Bed height at feed = 2.5

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</table>

Stationary thermocouple

\begin{align*}
T_{\text{gas in}} &= 402 \\
T_{\text{solids in}} &= 34
\end{align*}
Run: CK4

| Gas Flow (kg/hr) | 496 | Re = 18865 |
| Solids flow (kg/min) | 1.14 |
| RPM | 2 | Time included between reading = 2.5 |
| Holdup (kg) | 8.694 | Holdup (%) = 8.5 |
| Chord length at exit | 20 |
| Bed height at exit | 2.5 |
| Chord length at feed | 23 |
| Bed height at feed | 6 |

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Stationary thermocouple

\text{\text{\text{Gas in}} = 406}
\text{\text{\text{Solids in}} = 31}
Run: CK5

Gas flow (kg/hr) = 439  \hspace{1cm}  Re = 16323
Solids flow (kg/min) = .56
RPM = 1.0  \hspace{1cm}  Time included between reading = 5
Holdup (kg) = 6.284  \hspace{1cm}  Holdup (%) = 6.1
Chord length at exit = 19
Bed height at exit = 2.5
Chord length at feed = 21
Bed height at feed = 4.5

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Stationary thermocouple:

\begin{align*}
T_{gas \text{ in}} &= 421 \\
T_{solids \text{ in}} &= 37
\end{align*}
Run: CK5

Gas Flow (kg/hr) = 440
Solids flow (kg/min) = .83
RPM = 1.5  Time included between reading = 3.4
Holdup (kg) = 7.464  Holdup (%) = 7.3
Chord length at exit = 19
Bed height at exit = 2.5
Chord length at feed = 23
Bed height at feed = 5.5

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Stationary thermocouple

T_{\text{gas in}} = 418
T_{\text{solids in}} = 32
Run: CK7

Gas Flow (kg/hr) = 443  \hspace{1cm} Re = 16719
Solids flow (kg/min) = 1.13
RPM = 2.0 Time included between reading = 2.5
Holdup (kg) = 7.024  \hspace{1cm} Holdup (%) = 6.9
Chord length at exit = 20
Bed height at exit = 2.5
Chord length at feed = 22
Bed height at feed = 5

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Stationary thermocouple:
\begin{align*}
T_{gas \text{ in}} &= 435 \\
T_{solids \text{ in}} &= 33
\end{align*}
Run: CK8

Gas Flow (kg/hr) = 361  
Solids flow (kg/min) = .56  
RPM = 1.0  
Holdup (kg) = 7.374  
Chord length at exit = 20  
Bed height at exit = 2.5  
Chord length at feed = 22.5  
Bed height at feed = 5.3  

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Stationary thermocouple

T_{gas\ in} = 408
T_{solids\ in} = 32
Run: CK9

Gas Flow (kg/hr) = 392
Solids flow (kg/min) = .83

Re = 14806

RPM = 1.5 Time included between reading = 3.4
Holdup (kg) = 6.344
Holdup (%) = 6.2

Chord length at exit = 20
Bed height at exit = 2.5
Chord length at feed = 22
Bed height at feed = 4.2

<table>
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Stationary thermocouple

T_{gas in} = 414
T_{solids in} = 35
Run: CK10

Gas Flow (kg/hr) = 392  \quad Re = 14832
Solids flow (kg/min) = 1.13
RPM = 2  \quad Time included between reading = 2.5
Holdup (kg) = 7.074  \quad Holdup (%) = 6.9
Chord length at exit = 205
Bed height at exit = 2.5
Chord length at feed = 22
Bed height at feed = 5.2

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Stationary thermocouple

\( T_{gas \ in} = 399 \)
\( T_{solids \ in} = 32 \)
Run: CK11

Gas flow (kg/hr) = 330
Solids flow (kg/min) = .56
RPM = 1  Time included between reading = 5
Holdup (kg) = 7.154  Holdup (%) = 7.0
Chord length at exit = 21
Bed height at exit = 2.5
Chord length at feed = 22
Bed height at feed = 6.0

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stationary thermocouple
Gas in = 384
Solids in = 27
Run: CK12

Gas Flow (kg/hr) = 332
Solids flow (kg/min) = .83
RPM = 1.5 Time included between reading = 3.4
Holdup (kg) = 6.984 Holdup (%) = 6.8
Chord length at exit = 20
Bed height at exit = 2.5
Chord length at feed = 22
Bed height at feed = 6

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Stationary thermocouple
T_gas in = 399
T_solids in = 34
Run: CK13

| Gas Flow (kg/hr) | = 332 | Re = 12697 |
| Solids flow (kg/min) | = 1.13 |
| RPM | = 2.0 | Time included between reading = 2.5 |
| Holdup (kg) | = 7.234 | Holdup (%) = 7.1 |
| Chord length at exit | = 19.5 |
| Bed height at exit | = 2.5 |
| Chord length at feed | = 21 |
| Bed height at feed | = 5.8 |

<table>
<thead>
<tr>
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Stationary thermocouple

T_{gas \text{ in}} = 393

T_{solids \text{ in}} = 35
Run: CK14

Gas Flow (kg/hr) = 265  
Solids flow (kg/min) = 0.56  
RPM = 1.0  
Holdup (kg) = 7.424  
Chord length at exit = 19  
Bed height at exit = 2.5  
Chord length at feed = 22  
Bed height at feed = 5.7

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Stationary thermocouple  
T_gas in = 401  
T_solids in = 36
Run: CK15

| Gas Flow (kg/hr) | = 266 | Re = 10180 |
| Solids flow (kg/min) | = 0.83 |
| RPM | = 1.5 | Time included between reading = 3.4 |
| Holdup (kg) | = 7.324 | Holdup (%) = 7.2 |
| Chord length at exit | = 20 |
| Bed height at exit | = 2.5 |
| Chord length at feed | = 22.5 |
| Bed height at feed | = 6.1 |

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Stationary thermocouple
T_{gas in} = 389
T_{solids in} = 36
Run: EK1

Gas Flow (kg/hr) = 273
Solids flow (kg/min) = .43
RPM = 1.0  Time included between reading = 5
Holdup (kg) = 9.714  Holdup (%) = 9.5
Chord length at exit = 20
Bed height at exit = 2.5
Chord length at feed = 24
Bed height at feed = 5.8

<table>
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Stationary thermocouple

Tgas in = 387
Tsolids in = 30
Run: EK2

Gas Flow (kg/hr) = 274  Re = 10732
Solids flow (kg/min) = .43
RPM = 1.5 Time included between reading = 3.4
Holdup (kg) = 8.844  Holdup (%) = 8.6
Chord length at exit = 19
Bed height at exit = 2.5
Chord length at feed = 22
Bed height at feed = 5

<table>
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Stationary thermocouple  
Tgas in = 383
Tsolids in = 23
Run: EK3

Gas Flow (kg/hr) = 333
Solids flow (kg/min) = .43
RPM = 2
Holdup (kg) = 7.804
Chord length at exit = 19
Bed height at exit = 2.5
Chord length at feed = 20.5
Bed height at feed = 4.6

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Stationary thermocouple

Tgas in = 404
Tsolids in = 34
Run: EK4

Gas Flow (kg/hr) = 333
Solids flow (kg/min) = .33
RPM = 1.5 Time included between reading = 3.4
Holdup (kg) = 8.214 Holdup (%) = 8.0
Chord length at exit = 19
Bed height at exit = 2.5
Chord length at feed = 21
Bed height at feed = 5

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Stationary thermocouple

\[ T_{\text{gas in}} = 391 \]
\[ T_{\text{solids in}} = 34 \]
Run: EK5

Gas Flow (kg/hr) = 395
Solids flow (kg/min) = 0.33
RPM = 1
Time included between readings = 5
Holdup (kg) = 9.524
Holdup (%) = 9.3
Chord length at exit = 19
Bed height at exit = 2.5
Chord length at feed = 22
Bed height at feed = 5.3

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</table>

Stationary thermocouple
T_{gas in} = 399
T_{solids in} = 37
Run: EK6

<table>
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<th>Gas Flow (kg/hr)</th>
<th>Solids flow (kg/min)</th>
<th>RPM</th>
<th>Holdup (kg)</th>
<th>Chord length at exit</th>
<th>Bed height at exit</th>
<th>Chord length at feed</th>
<th>Bed height at feed</th>
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</table>

Re = 15101

Time included between reading = 3.4

Holdup (%) = 8.0

Rotated Thermocouple No. | Temperature (°C) | Time interval (secs) between (Tmin - Tmax)
-------------------------|------------------|-----------------------------------|
|                        | Tmin  | Tmax |                                |
| 0                       | 191   | 194  | 6.8                              |
| 1                       | 269   | 292  | 10.2                             |
| 10                      | 331   | -    | -                                |
| 2                       | 254   | 283  | 10.2                             |
| 3                       | 249   | 277  | 10.2                             |
| 11                      | 321   | -    | -                                |
| 4                       | 249   | 278  | 10.2                             |
| 5                       | 237   | 274  | 10.2                             |
| 12                      | 302   | -    | -                                |
| 6                       | 223   | 264  | 10.2                             |
| 7                       | 227   | 266  | 10.2                             |
| 13                      | 305   | -    | -                                |
| 8                        | 152   | -    | -                                |
| 9                        | 208   | 258  | 10.2                             |
| 14                      | 293   | -    | -                                |
| 15                      | 134   | -    | -                                |

Stationary thermocouple

T_gas in = 396

T_solids in = 31
Run: EK7

Gas Flow (kg/hr) = 440
Solids flow (kg/min) = 0.33
RPM = 1.5 Time included between reading = 3.4
Holdup (kg) = 7.954 Holdup (%) = 7.8
Chord length at exit = 17.5
Bed height at exit = 2.5
Chord length at feed = 21
Bed height at feed = 4.8

<table>
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<th>Time interval (secs) between (T\text{min} - T\text{max})</th>
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<td>-</td>
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<td>2</td>
<td>257   286</td>
<td>10.2</td>
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<td>10.2</td>
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<td>-</td>
</tr>
<tr>
<td>15</td>
<td>135   -</td>
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</tr>
</tbody>
</table>

Stationary thermocouple

T_{gas in} = 374
T_{solids in} = 33
Run: EK8

Gas Flow (kg/hr) = 441
Solids flow (kg/min) = 0.43
RPM = 2
Holdup (kg) = 7.354
Chord length at exit = 18.5
Bed height at exit = 2.5
Chord length at feed = 20.5
Bed height at feed = 4.8

<table>
<thead>
<tr>
<th>Rotated Thermocouple No.</th>
<th>Temperature (°C)</th>
<th>Time interval (secs) between (Tmin - Tmax)</th>
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<tbody>
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</table>

Stationary thermocouple

Tgas in = 376
Tsolids in = 37
Run: EK9

Gas Flow (kg/hr) = 501  \quad \text{Re} = 19305
Solids flow (kg/min) = 0.33
RPM = 1.0  \quad \text{Time included between reading} = 5
Holdup (kg) = 9.104  \quad \text{Holdup (\%)} = 8.9
Chord length at exit = 18.5
Bed height at exit = 2.5
Chord length at feed = 22.5
Bed height at feed = 5.2

<table>
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<tr>
<th>Rotated Thermocouple No.</th>
<th>Temperature (°C)</th>
<th>Time interval (secs) between (T_{min} - T_{max})</th>
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Stationary thermocouple

T_{gas in} = 389
T_{solids in} = 40
Run: EK10

Gas Flow (kg/hr) = 503
Solids flow (kg/min) = 33
RPM = 1.5  Time included between reading = 3.4
Holdup (kg) = 7.854  Holdup (%) = 7.7
Chord length at exit = 18
Bed height at exit = 2.5
Chord length at feed = 21.5
Bed height at feed = 4.9

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Stationary thermocouple

Tgas in = 380
Tsolids in = 40
Size Distribution of Raw Meal Nodules

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100.01

<table>
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<td>- 11.2 + 8.0</td>
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100.0