MODIFICATION TO THE CLINKER COOLER IN A CEMENT PLANT
by
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Abstract
The clinker cooler in a cement plant serves two functions: (i) the rapid cooling of clinker leaving the rotary kiln, and (ii) preheating of the air supplied to the rotary kiln by recovering heat from the clinker. When the production capacity of a cement kiln is increased it becomes necessary to modify either the operating conditions or the construction of the clinker cooler. If such modifications fail to meet the performance requirements the clinker cooler should be replaced.

The present work concerns modifications to the clinker coolers in cement kilns at the Puttalam Cement Works where the production capacity of the kilns was to be increased from 650 t/d to 1000 t/d in each kiln. This paper presents the theoretical analysis, the proposed modifications and the plant performance after modifications. The performance of a cement kiln at the Kankesanthurai Cement Works where similar modifications were effected is also commented on.

Introduction
The study of the cement plant at the Puttalam Cement Works (PCW) by Jayatilleke et al. 1 revealed that the production capacity of the two cement kilns at PCW could be raised from 650 t/d to over 900 t/d each by effecting simple modifications to the plant; and to 1000 t/d by introducing a precalciner. 2 These estimates were considerably higher than the production potential estimated by an earlier investigation, 3 and demanded a careful review of the individual thermal processes at every stage of the production process.

In most parts of the plant relatively simple modifications such as the speeding up of a drive or the replacement of a motor or the addition of a new element were adequate to meet the increased load. It was the cyclone preheater system, too expensive to modify or to replace, which restricted maximum production to 1000 t/d per kiln. The clinker cooler was one component where the study was not as simple and straightforward as in other units. The plant layout did not permit the lengthening of the cooler, which would otherwise have been, in the long run, the most economic proposition. Widening the cooler was another possibility, but it amounted to replacing the existing cooler by a new one and was subject to space restrictions. It was therefore necessary to explore the possibility of maintaining the existing cooler and modifying the operating conditions in order to achieve the desired clinker cooling capacity and the prescribed heat recovery for the kiln air supply.

This paper describes how the desired performance was achieved by making inexpensive modifications to the existing cooler. The functioning of the clinker cooler is briefly described in the next section, and the theoretical analysis is given in the section which follows. The next section contains the recommended modifications, and is followed by a discussion of the performance of the plant after the modifications were effected and comments on the performance of one of the cement kilns at the Kankesanthurai Cement Works (KCW) where modifications were undertaken on the basis of the present work, in order to overcome some problems in clinker cooling. The final section is a brief summary of the main conclusions.

The Clinker Cooler
The location of the clinker cooler in relation to the other thermal process units of the cement plant is shown in Figure 1. The operation of the cooler is illustrated in Figure 2. The cooler is intended to cool the clinker leaving the rotary kiln at around 1370°C to 70°C, and the “initial part” of the cooling is to be done rapidly. The cooler is also intended for preheating to 850°C 0.9 m³ (at STP) of air per kilogram of clinker by recovering heat from the clinker.

The initial section of the clinker bed is sloping and is considerably thicker than the clinker bed further down. The need for rapid cooling in the initial section of the cooler makes it necessary to have a larger air flow rate in the initial section than in the rest of the cooler and the high temperature in the initial section contributes to increased resistance to air flow, and, as a result, the under-grate air pressure is subject to a large variation along the length of the bed. In all clinker coolers these variations in air flow rate and under-grate pressure are accommodated by compartmentalising the air supply. The clinker cooler shown in Figure 3 is rated for 800 t/d and has six air supply compartments. In local cement kilns only three or four compartments are provided in order to save on capital investment.
Increase in cement production implies a proportionate increase in the load on the clinker cooler and, in turn, demands an increase in air flow rate. If the existing systems were taken as optimised, then the matching air flow rate can ideally be expected to be proportional to the new rate of production. This does not necessarily imply that the requirements of clinker cooling and heat recovery cannot be met under any other flow condition. Two possible extreme operating conditions will serve to illustrate the point:

(1) The cooling may be achieved by maintaining the clinker bed depth the same as in the existing set-up. This is achieved by increasing the grate speed. The inadequate bed depth demands an increase in air flow rate to meet the cooling requirements and a redistribution of under-grate air flow to meet the demands of adequate heat recovery from the clinker. This operating condition effectively implies an excessive air flow rate.

(2) The cooling may also be achieved with the bulk mean velocity of the clinker remaining the same. Here the grate speed remains the same and the bed thickens in proportion to the rate of production of clinker. The main implication of this is that the resistance to air flow is increased and that the fans should overcome a larger frictional head than necessary under ideal conditions. Another minor, but nevertheless significant, implication is the reduction of the clearance available between the clinker bed and the hood of the cooler. Excessive thickening of the bed, especially during periods of surge, can restrict air flow to the kiln and it is therefore preferable to keep the bed thickness as small as possible.

The ideal situation is one where the prescribed clinker cooling and heat recovery are achieved with the minimum necessary air flow and fan power input.
Figure 2: Clinker Cooler System

(C1 - C6, undergrate compartments; DC, dust collector; F1 - F6, fans; K, kiln; firing hood)

Theoretical Analysis

The clinker cooler is essentially a steady flow device. The directions of movement of the clinker and the air are more or less at right angles. The velocity of movement of the clinker is of the order of 0.01 m/s compared to air velocities within the bed ranging from 1 m/s to 10 m/s. The effect of clinker velocity on heat transfer characteristics within the clinker bed is insignificant except by way of determining the heat removal rate. This permits the cooling system to be treated as a packed bed with an internal heat source (of strength varying with position) maintaining steady state operation.

The bed resistance to airflow is large and the depth of the bed (ranging from 0.25 to 0.5 m) is small compared to its length (over 10 m), and the pressure gradient across the bed depth is an order of magnitude larger than that along the bed under ideal conditions. This makes it possible to treat the airflow as unidirectional (i.e., normal to the clinker layer). It should be mentioned here that even where the mean airflow is not quite unidirectional, approximate similarity of flow can be assumed, without serious error, between beds of different, but comparable, depths, provided that the depths remain small relative to the length and the boundary conditions relating to similarity are satisfied in their essence. This simplified treatment, it should be emphasised here, does not introduce uncertainties comparable in magnitude to those relating to the unsteadiness observed in clinker movement.

For the rectangular element shown in Figure 4, the heat generation is given by

\[ \left( c_c \frac{\dot{M}_c}{\gamma} \right) \left( \frac{\partial T_c}{\partial x} \right) \, dx \, dy \]

For steady state operation, the heat generated should be removed by the coolant air, and hence,

\[ \left( c_c \frac{\rho_c}{\gamma} \right) \left( \frac{\partial T_c}{\partial x} \right) \, dx \, dy = \rho_c \gamma \, c_a \, h \left( T_c - T_a \right) \, dx \, dy \, \ldots (a) \]

The heat removal rate is also determined by the heat transfer coefficient \( h \), which is a function of local flow conditions.

\[ \left( c_c \frac{\rho_c}{\gamma} \right) \left( \frac{\partial T_c}{\partial x} \right) \, dx \, dy = c_a \, h \left( T_c - T_a \right) \, dx \, dy \, \ldots (a) \]

For a packed bed of the present type, \( h \) may be expressed in the form

\[ h = \text{constant} \times n \, T_c \]

for the range of flow Reynolds numbers in the present case.

Integrating equations (1) and (2) across the bed,

\[ c_c \frac{\dot{M}_c}{\gamma} \left( \frac{\partial T_c}{\partial x} \right) = \left( \tau_a, x, y \right) = c_c \left( \frac{\partial \tau_c}{\partial x} \right) \mid_{y=0}^y \left( T_c - T_a \right) dy \]

\[ c_c \frac{\dot{M}_c}{\gamma} \left( \frac{\partial T_c}{\partial x} \right) = c_a \, h \left( T_c - T_a \right) dy \, \ldots (4) \]
Equations (5) and (6) imply that an increase in production by a factor of 1.5 (50%) would require an increase in air flow rate by 50% and bed depth by a factor of 1.5^{0.5} (or around 18%), and, correspondingly, the bulk mean velocity of clinker by around 27%.

The Proposed Modifications

The proposed modifications were based on the prescribed air flow distribution for the conventional cooling systems and on the basis of the existing arrangement at PCW. (See figure 5). It was found that the compartmentalisation in the cooler for kiln 2 was certainly inadequate and that the fan and motor capacities were considerably in excess of what was necessary for cooling 650 t/d. This view was supported by the satisfactory performance of the cooler in
Kiln 2 even after an increase in production to 750 t/d. The cooling in kiln 1 did cause problems during periods of surge after increase in production to 750 t/d.

With increased production adequate compartmentalisation is necessary to ensure economy as well as to make sure that there is no excess pressure under the hood. The spacing of the partitions was so arranged as to ensure minimum interference with the existing arrangement and maximum utilization of existing equipment, while ensuring that the air distribution was as close as possible to the ideal. The motor and fan specifications are given in Table 1 and the recommended layout of air distribution is shown in Figure 5. It may be seen that three of the available motors and three of the fans (one after increasing speed by 5% and subject to stresses not exceeding prescribed limits) could be used in the modified set-up.

Discussion of Performance

The proposed modifications were made in the cooler air supply system and the grate speed was increased by 25%. The production capacity of each kiln was increased to a little over 900 t/d without installing the precalciner. The prescribed standards of clinker cooling and air heat recovery were achieved in both kilns and the motors and fans are capable of delivering the necessary air flow when production is raised to 1000 t/d. In fact, the air supply system has adequate provision to meet the demands of higher pressure during intervals of surge. The modified system has been in operation for over two years and the operation troublefree.

Clinker cooling in the KCW kiln 2 with a production capacity of 500 t/d was at times not satisfactory and this was worsened when production was increased to 700 t/d. The cooling requirements could not be met by the mere regulation of air flow. Subsequent to the modifications undertaken in the PCW clinker coolers, the KCW cooler was also modified on the basis of the present analysis and the problem of cooling has been satisfactorily overcome.

The present modifications certainly are not economical in terms of energy consumption. The power input, corresponding to a 50% increase in production, is 300% in excess of the earlier value. The actual increase in power consumption will be in the region of 300 kW at 2000 t/d from both kilns at PCW. This increase is small compared to the increases in input to other units such as the waste gas fan, dust collector fan etc. put together. Also these modifications are the most economical within the existing set-up where financial and other considerations rule out other alternatives. The increase in specific energy input, perhaps, is the price one has to pay for the inflexibility of the existing system and the need to meet sudden increase in demand for industrial products. An important lesson from this exercise is that the system designer should bear in mind the possibility of future expansion and permit reasonable flexibility within the system to meet varying conditions of demand.

Conclusions

1. The cooling capacity of the two clinker coolers in the PCW has been increased from 650 t/d to 1000 t/d by effecting simple modifications.

2. The cooler performance at 900 t/d was found satisfactory and indicates capability of meeting 1000 t/d per kiln.
Figure 5: Existing and Proposed Under-Grate Air Supply System (See Table 1)
Nomenclature

\(c\) specific heat capacity

\(h\) heat transfer coefficient

\(M\) mass flow rate of clinker across unit width of bed

\(T\) temperature

\(V_a\) air velocity

\(x\) distance measured along the bed

\(y\) distance measured normal to the bed

\(Y\) bed depth

\(\rho\) bulk density of clinker

\(\sigma\) specific surface of clinker

Subscripts and superscripts

\(a\) air

\(c\) clinker

\(x, y, Y\) as above

\(\bar{\cdot}\) mean value

\(\text{Grate level}\)

Acknowledgements

Thanks are due to Mr. Asoka Somaratne, Chairman, Sri Lanka Cement Corporation for making the present study possible, to Prof. B. L. Panditharatte, Vice Chancellor, University of Peradeniya for his kind permission, to Messrs M. D. J. F. Jayamanne, S. Jayakumar and M. H. Wickremasinghe of the Sri Lanka Cement Corporation for their valuable assistance and to other members of the technical staff of the Puttalam Cement Works for their kind cooperation.

Table 1: Fan and Motor Specifications for Air Supply to Cooler

<table>
<thead>
<tr>
<th></th>
<th>Existing system</th>
<th>Proposed system</th>
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<tbody>
<tr>
<td></td>
<td>Kiln 1</td>
<td>Kiln 2</td>
</tr>
<tr>
<td>Fan (air flow/m²/min, head/mmH₂O)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>515 300</td>
<td>240 450</td>
</tr>
<tr>
<td></td>
<td>43 1470</td>
<td>29 60</td>
</tr>
<tr>
<td>Motor (power/kW, speed/(rev/min))</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>1285 150</td>
<td>515 300</td>
</tr>
<tr>
<td></td>
<td>60 975</td>
<td>43 1470</td>
</tr>
<tr>
<td>Fan</td>
<td>+ 1150 200</td>
<td>515 300</td>
</tr>
<tr>
<td>3</td>
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<td>53 1470</td>
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<tr>
<td>Motor</td>
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<tr>
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<td>1060 225</td>
<td></td>
</tr>
<tr>
<td></td>
<td>78 975</td>
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</table>

Note * Item usable in proposed system too. + The fan speed is to be raised by 5%.

$ The fans are belt-driven and the motor speed could be matched to any fan speed.

References


\[(c_c \dot{M}_c / Y) (\partial T_c / \partial x) \, dx \, dy\]

For steady state operation, the heat generated should be removed by the coolant air, and hence

\[(c_c \dot{M}_c / Y) (\partial T_c / \partial x) \, dx \, dy = (\partial T_a / \partial y) \, c_a \, \bar{v}_a \, dx \, dy \quad (1)\]

The heat removal rate is determined by the heat transfer coefficient \(h\), which is a function of local flow conditions.

\[(c_c \dot{M}_c / Y) (\partial T_c / \partial x) \, dx \, dy = \rho_c \sigma \, h \, (T_c - T_a) \, dx \, dy \quad (2)\]

For a packed bed of the present type, \(h\) may be expressed in the form, \(h = \text{constant} \, x \, \bar{v}_a \). The value of \(n\) is taken as 0.59 for the range of Reynolds numbers in the present case.\(^8\)

Integrating equations (1) and (2) across the bed,

\[c_c \dot{M}_c \, (\partial T_c / \partial x) = (T_{a,x,y} - T_{a,x,0}) \, c_a \, \bar{v}_a \quad (3)\]

\[c_c \dot{M}_c \, (\partial T_c / \partial x) = \rho_c \sigma \int_0^y h \, (T_c - T_a) \, dy \quad (4)\]

For two packed beds with flow similarity and identical temperature boundary conditions, at any section, \(x\),

\[\dot{M}_c = \bar{v}_a \quad (5)\]

and

\[\dot{M}_c = \bar{y} \, \bar{v}_a \quad (6)\]

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