A NEW COMPACT, VERTICAL MASSECUITE REHEATER

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Abstract

A new design of massecuite reheater has been developed. The equipment has a number of significant advantages over traditional designs. Features described in the paper include:

- Compact vertical configuration, requiring less floorspace.
- Vertically finned tubes, having higher fin efficiency than conventional transverse fins.
- Considerably higher heating surface / volume ratio than traditional designs.
- An even massecuite flow distribution, minimising the possibility of any short-circuiting or dead pockets.

A key feature of the design is the unique configuration of the fins, which ensures close proximity of every part of the massecuite to a heated surface. Various considerations regarding heat transfer coefficients and resistance to flow are discussed. Preliminary results from the first installation are given. These indicate a higher than expected heat transfer coefficient. International patents on the design are pending.

Keywords: massecuite heater, curing, reheater, finned tubes, heat transfer, heat exchanger

Introduction

Maximising exhaustion (i.e. sugar recovery) of final massecuites usually involves retaining the massecuite in a stirred crystalliser over a protracted period (20-45 hours) and slowly cooling (to 42-48^{o}C), to deposit as much as possible of the sucrose from the mother liquor onto the crystals. This often results in a massecuite that is too viscous to cure. The massecuite is therefore usually reheated to about 50-56^{o}C to lower its viscosity, in a manner designed to minimise any dissolution of sucrose from the crystals.

The requirements for such reheating are that all parts of the massecuite remain below or near the "saturation temperature". For this:

- heating must be achieved with as small a ΔT as possible, preferably less than 2^{o}C above the saturation temperature (Honig, 1959).
- retention time at the raised temperature must be as short as possible, and
- heating must be evenly distributed through the massecuite, without any "hotspots" (no channelling or dead areas).

It is also desirable that the reheater be of low capital cost, require little maintenance and occupy as little space as possible, since they are often located in a congested part of the factory.

Traditional reheating equipment

In June 2001, Bosch Projects was called upon to design and supply a massecuite reheater. As a first step, it was decided to review past practice and evaluate the merits of various existing designs.

Most reheaters use hot water as the heating medium, transferring heat through straight or finned tubes. Initially, this was done by passing hot water through the final elements of conventional cooling crystallisers. Such plant was thermally inefficient, required long retention times and resulted in considerable sucrose dissolution.

Because of these deficiencies, attempts were made to heat massecuites by conducting an electrical current through the massecuite. Renton (1969) described one of these installations at Darnall. However, the devices were costly, required high electrical power, proved difficult to control and were unreliable (e.g. easily shorted by tramp metal such as welding electrodes in the massecuite). The concept of electric resistance heaters was therefore eventually abandoned in the South African industry.

Another idea that was tried (at Maidstone) in an attempt to improve heat transfer efficiency and speed of heating, was a scraped-surface reheater. This device was abandoned because of high mechanical needs.

Most reheaters now in use are purpose-designed heaters, with heating supplied to the massecuite by passing it between closely packed tubes that carry hot water.

One Australian design uses layers of plain tubes, each layer oriented alternately at right angles to that below. The objective of this design is to maximise the turbulence of the massecuite and its exposure to heated surfaces, but it results in a complex mechanical construction with tube ends on all four sides of the heater. It presumably encourages a fairly even flow distribution across the massecuite path, but uneven contact distance of massecuite from the heating surfaces, with numerous pockets of massecuite relatively far from any tube surface. The main inefficiency of the design, however, lies in its use of plain (unfinned) tubes.

Because of boundary layer conditions, the heat transfer from water into the tube metal is far higher (>2,000 W/m^2K) than that from the metal surface into viscous massecuite (c.40 W/m^2K). For this reason, finned tubes provide a much better balance between the rate of heat transfer per unit length of tube on the water and massecuite surfaces. This results in a smaller heating unit, a reduced residence time and less crystal dissolution (White *et al.*, 1982). Most modern reheaters therefore use finned tubes.

The fins are usually square and transversely mounted on the water tubes. It is essential to have good heat transfer between the tube and fin. For this reason, the fins are cast integrally with the tube, welded to the tube, or pressed onto the tube with a tight-fitting extended inner lip.

Whatever type of transversely finned tube is used, the massecuite flow is across the tube. To minimise the possibility of channelling, the cross-section of the massecuite flow path should be as small as possible. However, this requires a multiplicity of short tubes, which is costly and mechanically problematic (due to large flat surfaces under internal pressure, and to multiple welds or tube expansions).

A compromise is therefore usually adopted, with a relatively large massecuite flowpath cross-section. This unfortunately results in significant channelling, with "dead pockets" in some areas while most of the massecuite short-circuits through a narrow pathway (Figure 8). Because of this inefficient flow distribution, higher temperature heating water has to be used, causing some re-solution of crystal (purity rise) across the heater.

Bosch Projects massecuite reheater

For their new reheater, Bosch Projects started out with the following objectives:

- Reduced capital cost
- Reduced retention time
- Higher fin efficiency
- Reduced space requirement.

The design was completed in August 2001 and during the second half of the year, two reheaters were manufactured and installed in Jamaica and the Philippines to meet specifications set by the clients. The first reheater was commissioned early in 2002 with excellent overall system results (pan–crystalliser–reheater), but a faulty massecuite temperature transmitter has thus far prevented analysis of the separate unit performances of the crystalliser and reheater. The Philippines reheater will be commissioned in September 2002.

It was intended in this paper to review actual performance against the design objectives, with the presentation and analysis of actual operating data recorded in Jamaica. Although delayed by the faulty temperature transmitter, it is hoped that this data will be available for presentation at Congress.

Heat transfer coefficients and fin efficiency

Reason for finned tubes in the massecuite reheater

When heat has to be transferred to or from a fluid, the heat flow rate is:

$$Q = U*A*\Delta t,$$

where

Q	=	Rate of heat transfer, Watts
U	=	Heat transfer coefficient (HTC), $W/(m^2K)$
A	=	Area for heat transfer, m^2
Δt	=	Temperature difference, K

When the HTC U involved is very low, it is obvious from the above that the rate of heat transfer will be low. Usually there is nothing that can be done about the temperature difference Δt, which leaves the area A to be increased. One way is by giving the surface fins, ribs or spikes. Common examples involve heat transfer to or from air, which has a low value of U, and in car radiators, in motor cycle engines and in air conditioners, fins are used to counter this.

Using the basic heat transfer equation, a complex computer program was developed to characterise the heat transfer from the heat source at the root of the fin, into the massecuite. This programme was used to optimise the fin configuration, including fin shape, thickness, area and spacing.

Some important findings from the program were the optimum fin length and the most effective fin configuration.

Optimum fin length

The metal temperature of the fin will decrease as distance from the heat sink (tube surface) increases, at a rate dependent on the thermal conductivity of the material of the fin, the Δt between fin and massecuite, the HTC between the fin metal and massecuite, the fin thickness and the fin shape. The heat transfer rate will therefore also decrease with distance away from the heat sink.

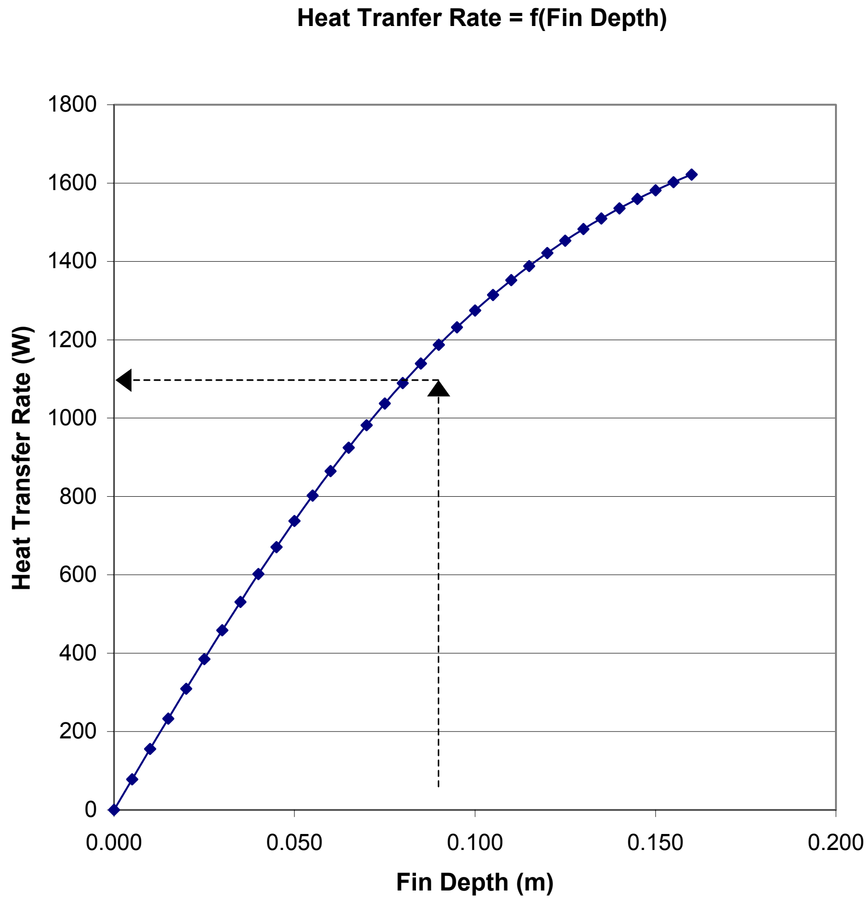

Figure 1. Variation of heat flow Q with fin depth L.

From this information, an optimum fin length of between 80 and 100 mm was selected.

Fin configuration - transverse vs longitudinal fins
The graph in Figure 1 also clearly indicates the benefit of maximising the area of the fin closest to the heat source. A comparison was done between transverse and longitudinal fins in order to find the most efficient solution.

This clearly showed the superior fin efficiency of longitudinal fins (efficiency 85%) over transverse fins (73% efficiency). The reason for this is demonstrated by the following simplified analysis. In Figure 2 below the fin areas for transverse and longitudinal fins are divided into thirds based on distance away from the heating source.

The table shows why a longitudinal fin has higher fin efficiency and is more cost effective than a transverse fin of equivalent dimensions. In this case, the longitudinal fin transfers nearly 20% more heat per unit area of fin.

In a practical example, White *et al.* (1982) in evaluating one Australian reheater design calculated the fin efficiency of its (thin) transverse fins at only 65%. Bosch Projects use 81% efficiency for their design calculations – 25% higher.

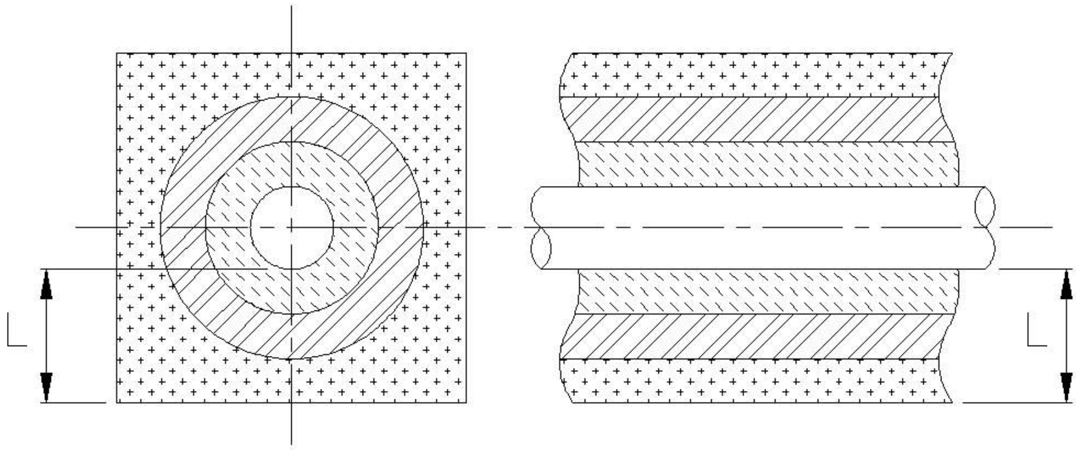

Figure 2. Transverse and longitudinal fin areas.

The areas are presented in Table 1 below.

Table 1. Longitudinal and transverse fin areas.

	Longitudinal fin	Transverse fin
Area less than $^{1}/_{3}$ L away from tube	33.3%	14 %
Area between $^{1}/_{3}$ L and $^{2}/_{3}$ L away from tube	33.3%	26 %
Area greater than $^{2}/_{3}$ L away from tube	33.3 %	60 %

The reheater design

Once the longitudinal fin had been selected, both the fin and reheater design had to be finalised. Aspects of the design to be discussed here are:

- Fin element design
- Heat transfer coefficients
- Massecuite flow resistance
- Reheater shell and heating water design.

Fin element design
Numerous alternatives were considered. All the simple-fin options initially tried presented problems of wide "easy flow" gaps once the tubes were arranged in a bank. The concept was close to being abandoned when the unique fin design and orientation illustrated in Figures 3 and 4 was "discovered".

As can be seen, this configuration provides a number of beneficial features:

- Efficient space coverage, with no massecuite more than 40 mm from a heating surface
- Cost effective construction, as standard pipe and angle iron sections are used
- High fin area to weld ratio.

The uniqueness of the design has enabled a patent to be registered.

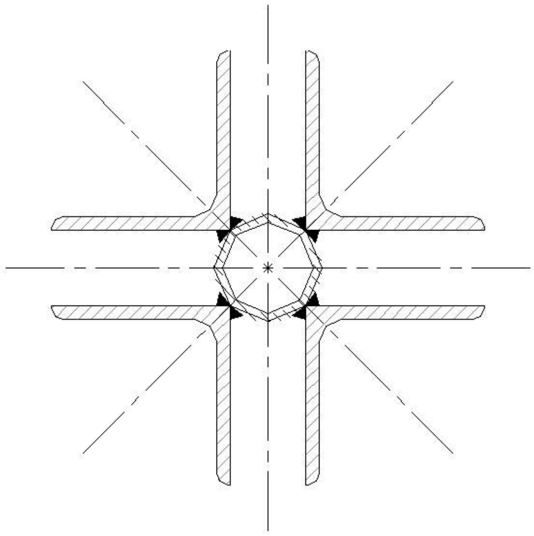

Figure 3. Cross section of reheater fin element.

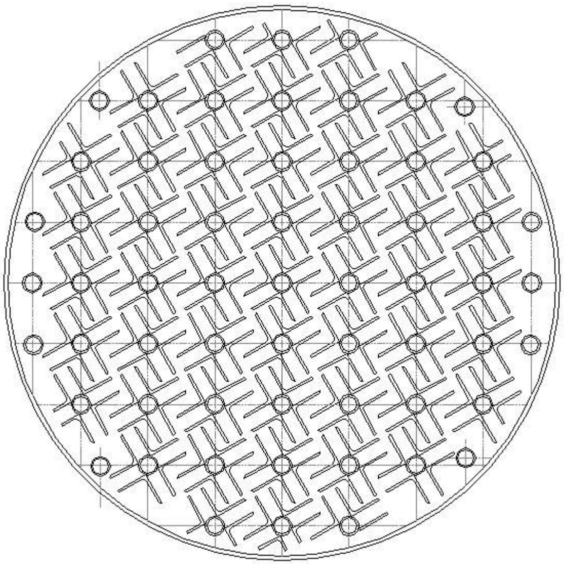

Figure 4. Cross section of reheater showing the fin arrangement.

Heat transfer coefficients

An aspect of the design which involved extensive research and much debate, was the applicable heat transfer coefficient (HTC) between massecuite and hotter (or colder) metal surface.

A wide range of sources was consulted and these provided a wide range of heat transfer coefficients. Some of the data gathered is shown in Table 2.

Table 2. Values of metal to massecuite HTCs in the literature.

Source of data	HTC (W/m²K)
Hugot (1986)	14 – 29
Honig, Vol II (1959)	6 – 46
Honig, Vol III (1959)	17 – 31
Rouillard (1976)	5 – 30
Kirby *et al.* (1976)	25 – 33
Ness (1981)	10 – 40

HTCs were also derived from various specifications available at Bosch Projects from equipment designs of BMA, Tongaat-Hulett Sugar, Illovo Sugar, Fletcher Smith and Fives Cail-Babcock. The HTCs calculated from these fell mainly within the range 20-30 W/m²K.

Some of the data was for cooling crystallisers, which operate over a range of higher average temperatures (from pan boiling temperature to cooled massecuite) than would a reheater (from cooled massecuite to curing temperature). Because of the lower viscosities at higher temperatures and the shearing action in crystallisers, higher HTCs may be expected in this equipment.

Faced with such varied information, Bosch Projects designed their first reheaters on the assumption of an average HTC of 11 W/m²K. From this, fin temperature profiles and hence the heat flux rates were calculated.

It is expected that measured results from actual installations will prove higher HTCs. These can then be used for more accurate designs in future.

Massecuite flow resistance
Rouillard (1975) attempted to analyse the friction loss from flows through finned tube massecuite reheaters. The flow patterns are too complex, designs too varied and massecuite rheological properties too unpredictable to be able to develop a formula to predict the head loss through any new reheater design. However, Rouillard did derive formulae that can be reduced to characterise the main relationships as:

$$\Delta H \quad \alpha \quad L.V.\mu / (e.D^2.\rho.g)$$

where:

ΔH	=	head loss through reheater, m
L	=	length of flow channel, m
V	=	actual average velocity of massecuite, m/s
μ	=	viscosity, kg/(m.s) or Pa.s
e	=	void fraction
D	=	mean hydraulic diameter, m
ρ	=	density, kg/m³
g	=	gravity constant, m/s^2

Of these terms:

- "Void fraction" is the proportion of total volume not occupied by the tubes
- "Mean hydraulic diameter" is 4 x Volume of flow channels / wetted surface
- Density is nearly a constant for C-massecuites.

Two conclusions relevant to this paper can be drawn from this relationship:

1. Compared to traditional transverse fin designs, in the new Bosch Projects design:
   - L, the length of flow channel, is longer
   - V, the velocity of massecuite flow, is much higher
   - e, the void fraction, is much smaller and
   - D, the mean hydraulic diameter, is much smaller.

These differences all contribute to a higher massecuite flow resistance and will therefore improve the uniformity of massecuite flow distribution. For more viscous massecuites the reheater would be made shorter and with a greater diameter. This will result in a lower value of L and V and a higher value of e. The net result would be a lower head loss through the reheater.

2. Once the flow resistance through one reheater has been measured, the relationship can be used to estimate the resistance through other geometrically similar reheaters. Using this, Bosch Projects propose to use the measured heads across their first two reheaters – at Appleton (Jamaica) and La Carlota (Philippines) – to predict the head required through future reheaters.

The relationship also demonstrates the manner in which the high viscosity of South African massecuites will need to be taken into account (resistance being proportional to viscosity).

Reheater shell and heating water design
In order to minimise costs and space utilisation, a vertical reheater arrangement was selected, as illustrated in Figure 5.

The hot water flowrate and number of passes is selected based on the required design parameters of the reheater. The heating water system has the following advantages over most traditional designs:

- Easy to maintain – no gaskets
- Cost effective
- No sealing or leak problems
- No external water header piping.

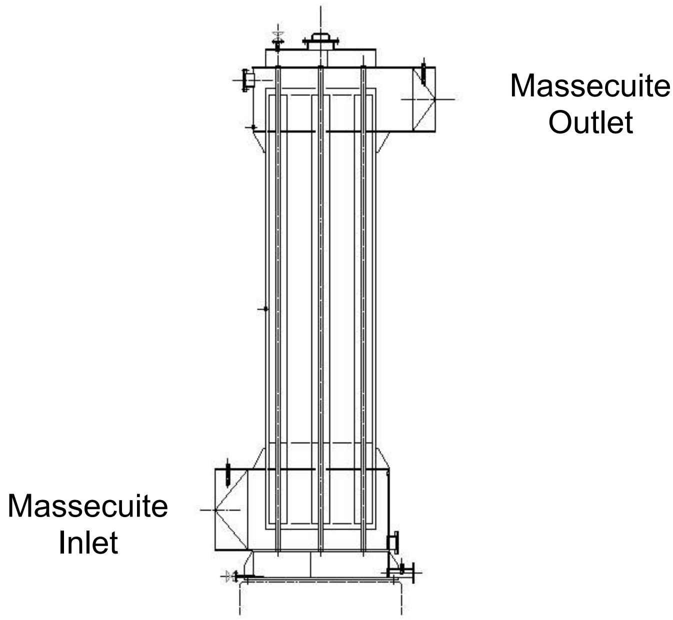

Figure 5. Reheater general arrangement.

It is possible to design the reheater with full counter-current water heating (with water: in top – out bottom; massecuite: in bottom – out top). However this requires very high water flows to maintain sufficient turbulence in the tubes, and results in a temperature drop of < 0.5^0C in the water from inlet to outlet. It was therefore decided to use a 4-pass water circuit (water in and out at the bottom). This resulted in a much smaller total water flow rate, with a negligible change to the log mean temperature difference, since the outlet water is still < 1.5^0C cooler than the inlet water.

Results

The first reheater of this design was manufactured, installed and commissioned in Jamaica in February 2002. The design and actual performance data are presented in Table 3 below.

Table 3. Reheater design vs actual performance data.

	Design	Actual
Massecuite flow rate (t/h)	14.2	10
Massecuite temperature in (^{o}C)	42	38
Massecuite temperature out (^{o}C)	52	50
Hot water flow rate (t/h)	50	26 ?
Water temperature in (^{o}C)	65	53
Water temperature out (^{o}C)	64.1	51.5

The "Actual" figures in Table 3 must be treated with caution and are subject to confirmation, as test conditions were not ideal. However, these preliminary figures indicate a heat transfer coefficient of approximately 25 W/m^{2}K, which is more than twice the deliberately conservative figure of 11 W/m^{2}K assumed in the initial design.

Comparison with conventional designs

Based on theoretical results, with the same massecuite-metal and metal-water HTCs, the following comparisons were done against a conventional horizontal reheater using transverse fins.

Cost

The Bosch Projects vertical reheater has a 60 % lower volume for the same effective heating surface area. This is mainly attributed to the way massecuite is fed into the reheater, in an attempt to avoid short-circuiting. The photograph in Figure 6 shows a cross section of a conventional reheater indicating the large volumes above and below the bank of tubes.

Figure 6. Conventional finned reheater.

This lower volume requirement of the vertical design translates into lower total mass (approximately 25% less) and hence a 25-30 % lower cost for an equivalent reheater.

Flow uniformity

Conventional reheaters are normally fed from one end at the bottom and discharge from the other end at the top. However, in most designs the cross-sectional area of the flow path is so large and resistance to flow therefore so small, that channelling is inevitable (see Figure 7).

The Bosch Projects design has a much smaller flow cross-section; so much so that depending on the incoming massecuite characteristics, the first section of pipes have only half sets of fins. This enables the massecuite to distribute evenly without excessive pressure drop. However, massecuite viscosity is approximately halved for every 5ºC temperature rise, and as soon as the massecuite temperature has increased slightly, full fin coverage is introduced.

In this regard, South African massecuites are notoriously viscous, and the transition to full fins would therefore usually be further up the heater than, for example, on the Jamaican reheater.

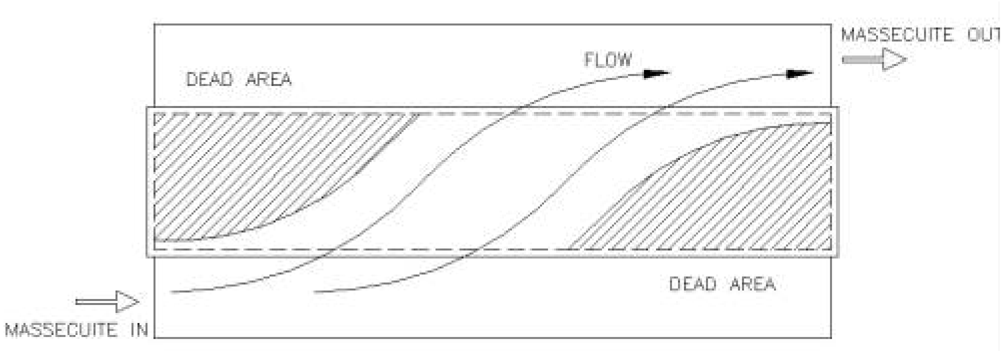

Figure 7. Conventional reheater flow pattern.

It is hoped soon to conduct tracer tests to confirm the expected uniformity of flow (plug flow) of the design.

Floor space utilisation

Many factories face problems of limited available floor space due to expansions and or modifications that have taken place over the years. The Bosch Projects reheater footprint requirement is approximately 80 % less than a conventional reheater and therefore provides greater flexibility for installation.

Figure 8 also illustrates the compact nature of the Appleton reheater relative to the vertical crystalliser that supplies massecuite to the reheater.

Conclusion

The first Bosch Projects reheater has been successfully designed, manufactured, installed and commissioned without problems. The client is pleased with the equipment. As expected, preliminary test results indicate a considerably higher heat transfer rate than was conservatively assumed for the initial design. The new reheater is expected to give significant cost and performance advantages over existing designs.

Figure 8. Appleton Mill (Jamaica) vertical C crystalliser, with matched reheater.

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