REFEREED PAPER

SUPERHEATER PERFORMANCE IN DUAL FIRED INDUSTRIAL WATERTUBE BOILERS WITH A COMPARISON OF TWO ATTEMPERATOR TECHNOLOGIES

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Abstract

In an era where cogeneration is becoming increasingly attractive, the overall efficiency of the steam power cycle is of primary concern for sugar mills. For turbine alternator sets to operate at their highest efficiency, the inlet steam should be at maximum possible temperature across the load range.

This paper begins by investigating the performance of superheaters in industrial watertube boilers for bagasse and coal firing. The relative contributions of the three primary modes of heat transfer in the superheater are then presented for each fuel across a range of boiler loads. It is shown that for an uncontrolled unscreened superheater, the steam temperature at the outlet varies with the boiler load and that the temperature achieved during coal firing is lower than for bagasse firing.

It is beneficial to use a controlled superheater to provide steam at a constant temperature across a range of loads to maximise the power generated from turbine alternator sets. Different attemperator technologies are suggested to control the steam temperature, and two typical systems are described in detail: the mud drum type and the spray type with integral condenser. These two attemperators are compared in the areas of performance, cost and maintenance.

Keywords: watertube boilers, attemperation, superheater, spray type attemperator, mud drum attemperator, desuperheater

Introduction

Watertube boilers are integral to the sugar manufacturing process. In the southern African (SA) sugar industry, boilers are often designed to burn both bagasse and coal and are consequently called dual fired boilers.

As a fibrous fuel, bagasse has different combustion characteristics to coal, which is a fossil fuel. These differences affect the superheater performance and, in the case of an uncontrolled superheater, result in lower steam temperatures when burning coal compared to bagasse.

This paper compares the superheater operating characteristics for bagasse and coal firing by investigating the relative contribution of the different modes of heat transfer for a range of boiler loads. Thereafter, two attemperation technologies are presented and compared to show how a constant steam temperature can be achieved with a controlled superheater.

Comparison of boiler operation under bagasse and coal firing

Furnace design parameters

Bagasse is usually the primary fuel in dual fired boilers in the SA sugar industry, while coal is used as a support fuel or for power generation during off-crop. A moving grate is required to burn the coal because peas are generally the preferred size grading.

Certain design parameters are used to determine the furnace dimensions required to achieve complete combustion. As a fibrous fuel with a large proportion of fine particles, bagasse is burned primarily in suspension in modern spreader fired boilers. Two combustion parameters therefore apply to bagasse firing: (i) volumetric heat release rate (VHRR) and (ii) grate heat release rate (GHRR) (Magasiner and de Kock, 1987). The VHRR and GHRR are measures of the heat released by the fuel per unit volume of the furnace (typically expressed in kW/m^3), and per unit grate effective area (MW/m^2) respectively. The furnace release rate (FRR) is sometimes used as a third parameter for suspension firing (Singer, 1981). This is a measure of the heat released by the fuel per unit area of effective projected radiant surface (MW/m^2). Where the requirements for VHRR and GHRR are met, the limitations for FRR are usually satisfied.

In contrast to bagasse firing, the majority of the coal burns on the moving grate due to the relatively high proportion of larger particles. Consequently, the principal combustion parameter for coal is the grate rating, which is the measure of the heat released per unit area of the grate (Magasiner *et al.*, 2001). Each boiler supplier has its own preferred basis for measuring the grate rating. John Thompson determines the mass of coal burned per unit area of the grate per hour ($kg/h.m^2$). Expressing the rate on a mass basis rather than a heat basis intrinsically accounts for variations in coal quality.

Experience at SA sugar mills burning SA coals has shown that the maximum continuous rating (MCR) for coal firing is typically 80% of the MCR for bagasse firing.

Combustion characteristics influencing superheater performance

Heat is transferred to the steam in superheater tubes by a number of modes depending on whether the superheater is screened or unscreened. Screened superheaters are shielded from the luminous radiation from the furnace by screen tubes, which are typically rear-wall tubes that are extended to the roof of the furnace.

Since the screen greatly reduces the influence of luminous radiation, heat is transferred to screened superheater tubes by two primary modes:

1. Convection from the hot flue gasses passing across the tubes, and
2. Non-luminous radiation from the hot flue gases surrounding the tubes.

Unscreened superheaters can be classified as platen type and combination type. Platen superheaters are positioned at the top of the furnace directly above the flame, and are typically of the pendant type as shown in Figure 1. Combination type superheaters are positioned above the nose of the furnace where heat is transferred by a combination of radiation and convection (Rathore, 2010). These superheaters and can be of the pendant or drainable type.

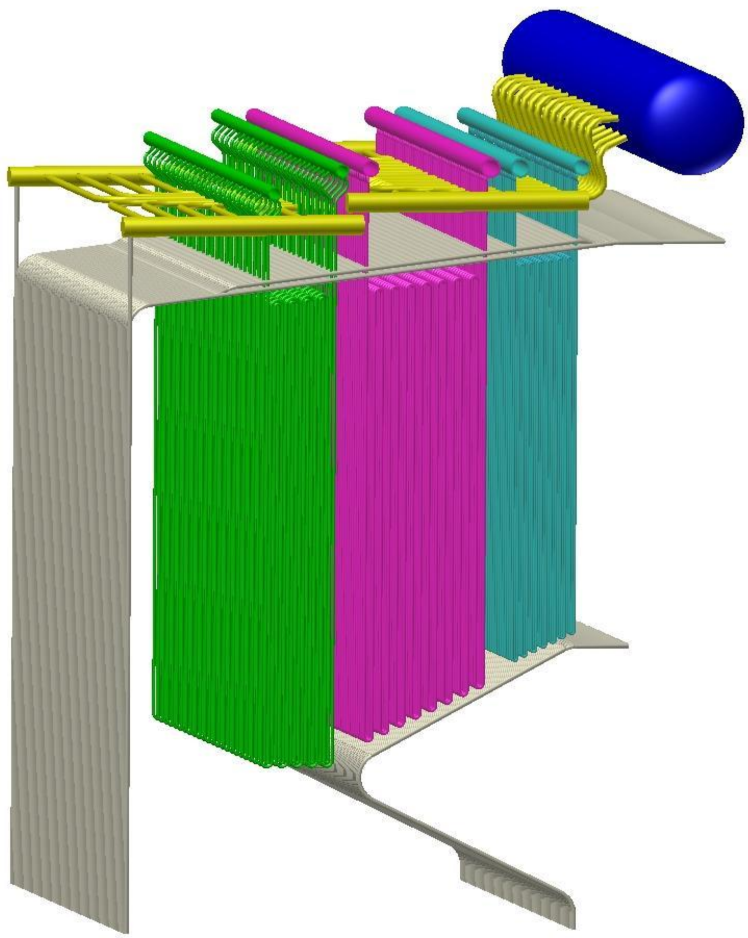

Figure 1. The top of a furnace showing the platen superheater directly above the furnace and two combination superheater banks above the furnace nose.

Platen superheaters usually form part of a multiple-stage superheater system where high steam temperatures are required. They are not common in boilers in the SA sugar industry because the relatively low main steam temperatures do not necessitate their use. Furthermore, they require very careful monitoring during start-up to prevent overheating, and are avoided where combination superheaters can be used instead. Platen superheaters will consequently not be considered in this paper.

A combination type unscreened superheater is shown in Figure 2, where the bottom of the superheater bank is protected from the flame in the furnace by the rearwall nose, and the front of the bank is exposed to the flame in the furnace.

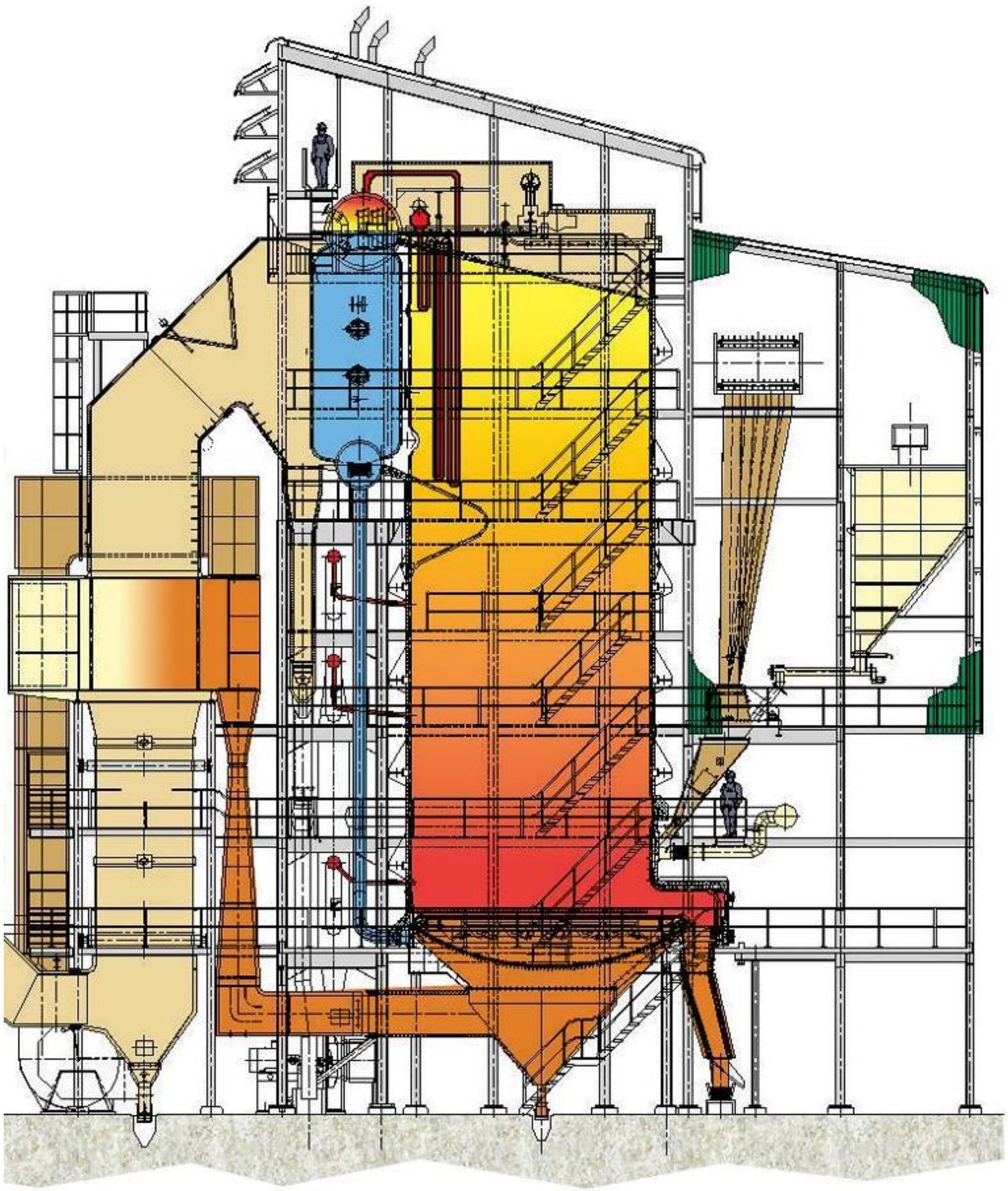

Figure 2. Boiler with a combination type unscreened superheater.

Heat is transferred to unscreened superheaters by direct luminous radiation from the flame in the furnace as well as the convection and non-luminous modes described above. This type of superheater is common in the SA sugar industry.

A number of factors influence the heat transferred by the three modes described above. The following sections show that the overall superheater performance depends largely on the boiler load and the fuel that is being burned.

Modes of heat transfer in an unscreened superheater at various loads

The relative contributions of the three modes of heat transfer in an unscreened superheater are given in Figures 3 and 4 for bagasse and coal firing respectively. The data have been determined for a boiler with maximum continuous ratings of 100 and 80 tons of steam per hour (tph) for bagasse and coal respectively.

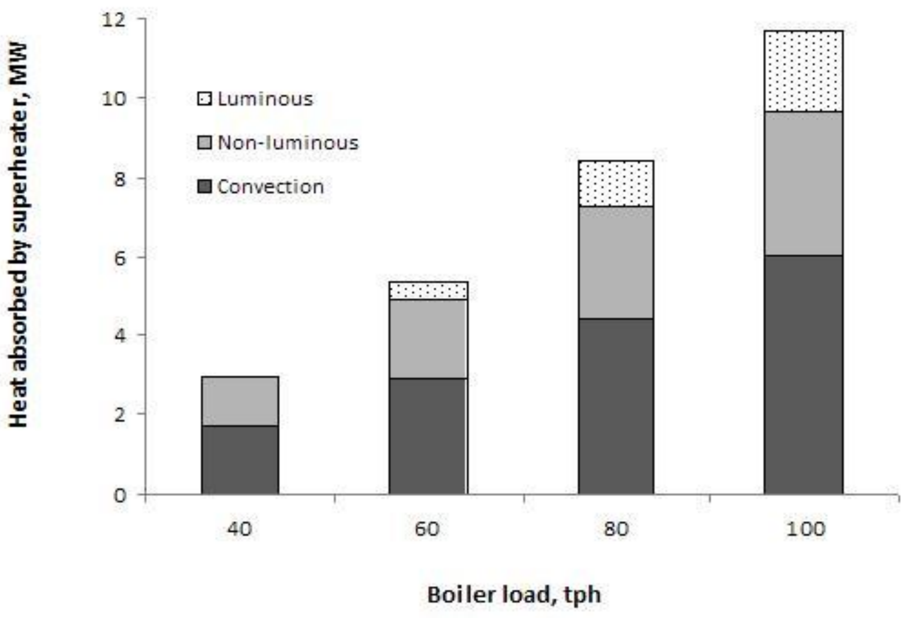

Figure 3. Different modes of heat transfer for bagasse firing.

The data presented in Figure 3 were determined for bagasse with total moisture and ash contents of 50% and 4.5% respectively. This shows that the total heat absorbed by the superheater increases as the boiler load is increased and that the relative contribution of heat transfer from luminous radiation increases from less than 1% at 40 tph up to 17% at 100 tph.

The data presented in Figure 4 were determined for coal with total moisture and ash contents of 6% and 14% respectively. As for bagasse firing, the total heat absorbed by the superheater increases steadily with increasing boiler load, and the relative contribution of heat transfer from luminous radiation increases from 2% at 40 tph up to 14% at 80 tph.

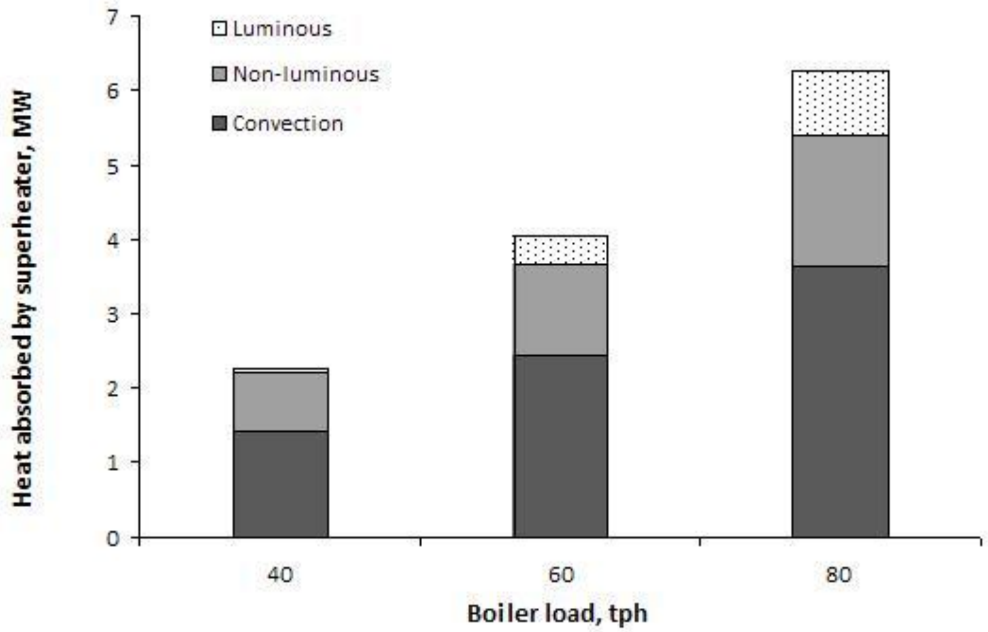

Figure 4. Different modes of heat transfer for coal firing.

The relevant data are summarised in Table 1 to compare the differences between bagasse and coal firing.

Table 1. Performance comparison of an unscreened superheater for bagasse firing vs coal firing.

Boiler load, tph	**40**	**60**	**80**	**100**
Main steam temperature, bagasse (°C)	332	356	380	399
Main steam temperature, coal (°C)	307	323	339	
Total heat to superheater, bagasse (kW)	2941	5373	8413	11673
Total heat to superheater, coal (kW)	2249	4044	6261	
Percentage luminous radiation, bagasse	0%	8%	13%	17%
Percentage luminous radiation, coal	2%	9%	14%	
Flue gas flow rate, bagasse (tph)	79.9	112.7	148.2	182.1
Flue gas flow rate, coal (tph)	71.9	100.1	131.9	

It can be seen that the final steam temperature increases steadily with increasing load for both fuels. In addition, the steam temperatures achieved while firing bagasse are consistently higher than for coal firing. Table 1 shows that the total heat transferred to the superheater is consistently higher for bagasse than for coal, yet the relative contribution of luminous radiation is similar for the two fuels at each flow rate.

The heat transferred by convection is higher during bagasse firing for two main reasons. Firstly, the flue gas mass flow rate is higher than for coal firing. Secondly, during coal firing the external surfaces of the superheater tubes tend to become fouled by accumulated fly ash deposits. This inhibits heat transfer to the steam and hence reduces the steam temperature at the superheater outlet. The extent of the fouling varies, depending on the ash characteristics of the coal (Magasiner *et al*., 2001). Sootblowers are usually used to clean the superheater tubes in boilers that burn a significant amount of coal. Fouling is not usually a problem during bagasse firing because the superheater tubes tend to be cleaned by sand in the fly ash, which is entrained in the flue gases.

The increasing contribution of luminous radiation with increasing boiler load for both fuels indicates that the superheater performance is influenced by the height of the flame in the furnace.

Controlling steam temperature with attemperators

Discussion of the need for attemperators

The electrical efficiency of a turbine alternator (TA) set is proportional *inter alia* to the steam temperature at the inlet (Energy and Environmental Analysis Inc, 2007). If steam from an uncontrolled superheater is used to drive a TA set, the variation in steam temperature will adversely affect the turbine performance. However, if a controlled superheater is used, the boiler will be able to supply steam at a constant temperature, which will allow the TA set to operate at its maximum efficiency across the range of loads.

There are a number of different options for controlling the boiler main steam temperature, depending on the load range and steam temperature required. These all involve a combination of superheater banks and attemperator (also known as 'desuperheater') technologies. In the simplest system, a single superheater bank operates in conjunction with a single attemperator. Additional design flexibility is achieved with increasing complexity through the addition of superheater banks and attemperators.

The modern trend in the SA sugar industry is towards cogeneration, which involves increasing the operating system pressures and temperatures to achieve higher system efficiencies for power generation. To achieve the high steam temperatures required for operation at pressures in excess of 45 bar, a three-stage superheater with two interstage attemperators is required as shown in Figure 5. This will be the most complex superheater system envisioned for the SA sugar industry in the foreseeable future.

Monodrum boilers such as that shown in Figure 5 are normally selected in preference to the traditional single pass bi-drum boilers for operating pressures in excess of 45 bar. The higher cost of the bi-drum pressure parts is one of the many reasons for this selection.

Considering that the majority of mills in the SA sugar industry operate at main steam pressures of 31 bar or lower, a two-stage attemperator with inter-stage attemperation has been the preferred technology for controlled superheaters to date.

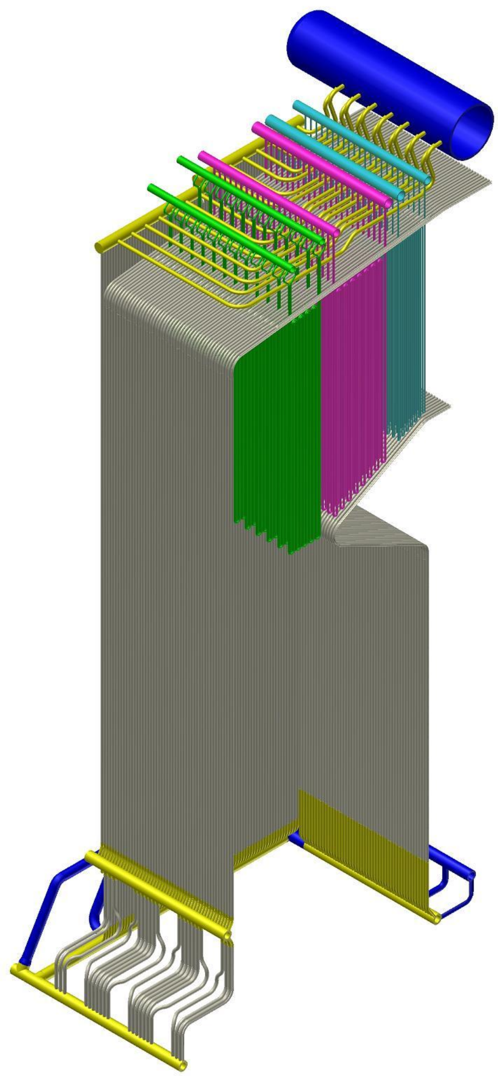

Figure 5. A three-stage superheater in a monodrum boiler.

Design of a two-stage superheater system

When designing a two-stage superheater for a dual fired boiler, the boiler design engineer will start by sizing the superheater banks to achieve the specified steam temperature on coal at the lowest control load. This will result in a superheater design that is oversized for bagasse firing. The design engineer will then select an attemperator that will be capable of controlling the final steam temperature to the specified value at MCR on bagasse.

Experience has shown that two types of attemperator systems are feasible for use at operating pressures up to 45 bar in industrial watertube boilers at SA sugar mills: the surface heat exchanger type and the spray type. The surface heat exchanger is commonly called a mud drum attemperator because it is usually installed in the boiler mud drum. These two systems are discussed in detail and compared in the sections that follow.

This discussion is not applicable to monodrum boilers because they do not have a second drum to accommodate a mud drum attemperator. In most conventional watertube boilers steam separation equipment is installed in the steam drum, leaving insufficient space for a heat exchanger. Moreover, monodrum boilers generally require multiple attemperators because they are usually supplied with three-stage superheater systems to achieve the high steam temperatures required for operating pressures in excess of 45 bar. The following comparison between a mud drum attemperator and a spray type attemperator is therefore applicable to boilers operating at pressures up to 45 bar.

Attemperator operating philosophies

Mud drum attemperator

A mud drum attemperator is classified as a surface heat exchanger because there is an impenetrable barrier between the steam and the cooling medium. The steam is contained within tubes that are arranged in bundles and submerged in the boiler mud drum. Heat is transferred from the steam to the cooler boiler water through the tube wall. The arrangement of a typical mud drum attemperator system is shown schematically in Figure 6.

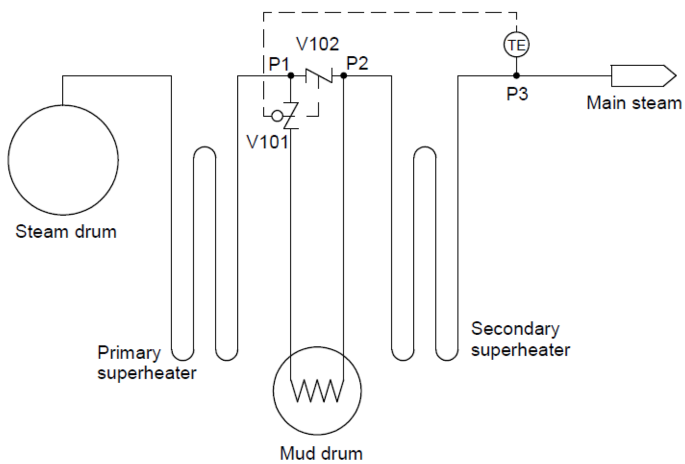

Figure 6. Mud drum attemperator system.

The steam piping between the primary and secondary superheater banks (inter-stage piping) is arranged so that a portion of the steam is diverted through the heat exchanger in the mud drum, while the remainder is bypassed. The cooler steam from the heat exchanger is mixed with the bypassed steam at point P2 to give the desired steam temperature at the inlet to the secondary superheater. Valves are installed in the heat exchanger inlet line (V101) and the bypass line (V102) to control the steam flow rate through the heat exchanger.

The valves are mechanically linked and are controlled by a single signal in a master-slave arrangement. The slave valve closes (and opens) in the proportion that the master valve is opened (and closed). The actuator is typically installed on the inlet valve, making it the master and the bypass valve the slave. The actuator is selected to fail open so that the full steam flow is directed through the heat exchanger in the event of a signal or air failure, to protect the downstream components from overheating. The main steam temperature is the process variable for the controller.

Spray attemperator
A spray attemperator is classified as a direct contact heat exchanger because the cooling medium is mixed with the steam. A suitable nozzle assembly is used to spray cooling water directly into the steam and thereby reduce its temperature. The arrangement of a typical spray attemperator system is shown in Figure 7.

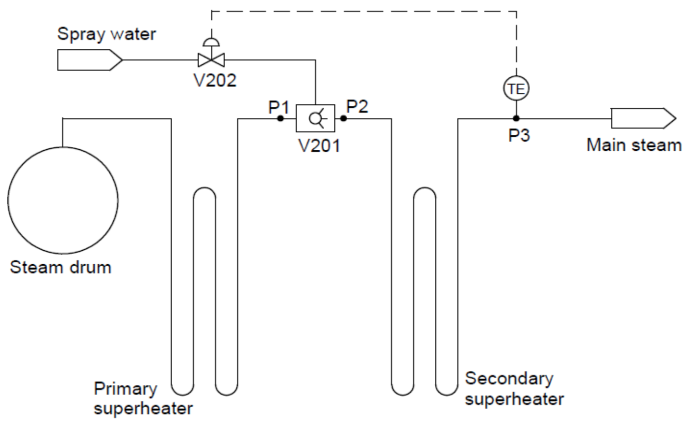

Figure 7. Spray attemperation system.

The attemperator unit (V201) is installed in the inter-stage piping and connected to a spray water supply pipe. The final steam temperature is used as the process variable to the controller, which regulates the spray water flow rate to the attemperator through a control valve (V202). Although V201 and V202 are shown as discrete components in Figure 7, they are sometimes combined in a single unit.

Attemperator units are designed to reduce the size of the spray water droplets as far as possible to encourage rapid evaporation in the steam. This process is called atomisation, and is commonly achieved by passing the water through a set of nozzles.

The spray water should contain minimal suspended or dissolved solids because these will be precipitated out of the water when it evaporates in the steam. The solids will be carried along with the steam and be deposited on the internal surfaces of the piping and superheater tubes. This insulates the tubes, which reduces the superheater performance and can lead to overheating of the tubes. If the solids are carried along to the turbine, they can deposit on the turbine blades and cause the rotor to become unbalanced.

A mud drum attemperator will supply steam of a very high purity because the steam does not mix with the cooling medium. With a mud drum attemperator therefore, the possibility of insulating boiler internal surfaces or damaging downstream equipment is much smaller than for a spray type.

The purity required for attemperator spray water is specified in boiler water treatment codes. Although demineralised water can be used, it usually requires further purification in a polishing plant before it can be sprayed into the steam. Demineralisation plants are not common in sugar factories that operate at steam pressures below 45 bar. With current technology, it is usually not feasible to install a dedicated demineralisation plant with downstream polisher, due to the high capital and running costs.

High purity condensate can also be used as spray water. Some sugar factories have available condensate of adequate purity, but this is generally not used for attemperation due to the perennial danger of contamination from the process. A dedicated condenser can, however, be incorporated into the boiler to generate high purity condensate with minimal risk of contamination. A typical integral condenser system is shown in Figure 8.

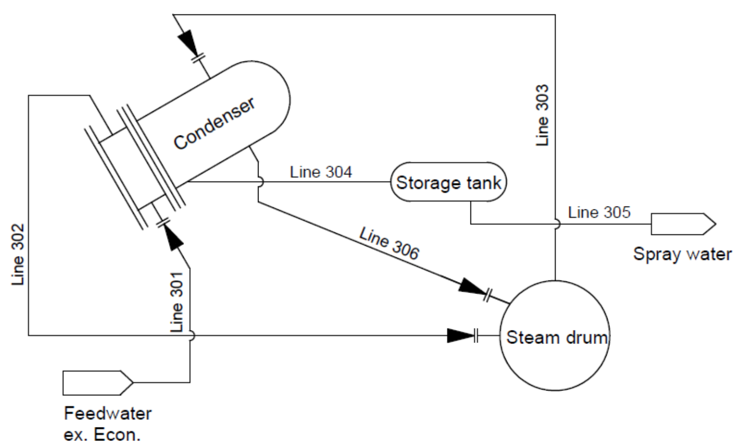

Figure 8. Integral condenser system.

A shell and tube heat exchanger is normally used to condense the saturated steam from the drum (Line 303) with boiler feedwater as the cooling medium (Lines 301 and 302). In some systems, the spray water is drawn directly from the condenser. The system shown in Figure 8 includes a condensate storage tank, which provides a reservoir of condensate to supply spray water under upset conditions. An overflow pipe (Line 306) is included to feed excess condensate back into the drum.

Conventional attemperator units typically specify a pressure difference of 650 to 900 kPa between the spray water and the steam to achieve effective atomisation. Where spray water is supplied from an external source, the spray water pumps should be able to supply spray water at the required pressure. These pumps tend to be expensive because they deliver a large head at a relatively low flow rate.

When the spray water is drawn from an integral condenser unit there is an added complication in that the temperature of the condensate is only a few degrees below the saturation temperature. Consequently, the system should be designed with care to avoid cavitation at the inlet of the spray water pumps.

Generally, John Thompson uses venturi spray attemperators because they require a much lower pressure differential between the spray water and the steam than a conventional attemperator. If a venturi attemperator is used, a spray water static head of six metres between the condenser and the attemperator is adequate and spray pumps are not required. An additional level is usually required in the boiler support structure to accommodate the condenser at the required elevation above the attemperator.

Attemperator design constraints

In the sections that follow, a mud drum attemperator will be compared against an integral condenser system with venturi attemperator. The performance of the two attemperator systems is limited by certain design constraints, which are described in this section.

Design constraints of a mud drum attemperator

The heat exchanger in the mud drum is the heart of the mud drum attemperator because it determines the load range that can be achieved. The boiler designer must take cognisance of the following variables when designing the heat exchanger:

- Heat transfer coefficient: This is a measure of the heat transfer effectiveness. It is affected by a number of factors, the most important being the velocity of the steam within the tubes.
- Heat transfer area: This refers to the surface area of the heat exchanger tubes. The larger the area, the more heat can be transferred from the steam to the water.
- Log mean temperature difference (LMTD): This is a measure of the temperature difference between the steam within the tubes and the cooling water surrounding the tubes.

The heat transfer coefficient can be increased by increasing the velocity of the steam within the heat exchanger tubes. The steam velocity should nevertheless be increased with caution, because it also increases the friction loss through the heat exchanger. This pressure drop increases the steam drum design pressure and therefore affects the required thickness of the boiler pressure parts, which can have a significant influence on the capital cost of the entire boiler.

Attemperators should therefore be designed to have the lowest possible pressure drop. This is particularly important where an attemperator system is retrofitted into an existing boiler, since the design pressure is fixed. Considering that both the heat transfer coefficient and the pressure drop over the heat exchanger tubes are proportional to steam velocity, the final design will involve a compromise between these two aspects.

Butterfly valves are commonly selected as the inlet and bypass valves because they are able to control the flow rate through the heat exchanger with a relatively low pressure drop. This helps to reduce the total pressure drop over the system and hence reduce the boiler capital cost.

The heat transfer area is limited by the available volume within the mud drum. The design should ensure that there is adequate space around the tube bundle to allow access for inspection and expansion of mainbank tubes. In addition, it is good practice to design the heat exchanger tube bundle so that it can fit through the mud drum manholes for ease of installation and replacement. For a new boiler it is possible to increase the size of the mud drum to accommodate a larger heat exchanger, but this too comes at additional capital cost.

The heat transfer in the mud drum attemperator can also be improved by increasing the LMTD between the steam and the cooling water. Since the temperature of the water is determined by the saturation temperature at the boiler operating pressure, only the steam inlet temperature can be manipulated to increase the LMTD. There is theoretically no upper limit to the steam inlet temperature, although it is usually advisable to limit the steam temperature to a maximum of 410°C so that carbon steel components can be used in the inter-stage piping. Materials with improved strength characteristics at higher temperatures are required when the steam temperature exceeds 410°C. The capital cost of the boiler will be higher where these materials are used.

The heat exchanger should be designed so that 60 to 80% of the steam flow is directed through the heat exchanger at MCR while firing bagasse. Experience has shown that this provides adequate margin for steam temperature control during upset conditions and boiler load ramps. The drum safety valve set pressures should be determined based on the pressure drop resulting from full steam flow through the heat exchanger. This will prevent the unnecessary lifting of safety valves during upset conditions.

The heat exchanger should be designed so that the steam temperature at the outlet is at least 15°C above saturation temperature to avoid condensation in the inter-stage piping, and steam traps should be installed at the lowest points in the piping as an additional measure.

Design constraints of a spray attemperator

The design constraints for spray type attemperators are intended to ensure effective evaporation of the spray water in the steam. If the water droplets do not evaporate properly, they can impinge on the internal surfaces of the pipework and cause damage. Furthermore, attemperator manufacturers usually specify a minimum straight length of piping downstream of the attemperator nozzles to ensure that the water droplets have adequate opportunity to evaporate before the steam reaches valves or bends.

The first constraint involves the maximum amount of spray water with respect to the total steam flow rate. The requirements differ depending on the type of attemperator but, as a rule of thumb, the spray water flow rate should be limited to a maximum of 11% of the mixed steam flow rate at the outlet of the attemperator.

The minimum steam temperature at the outlet of the attemperator should also be limited. As for the mud drum attemperator, there should be an adequate temperature margin to avoid condensation of the steam in the interstage piping. Attemperator manufacturers have their own requirements for this limit, but a minimum temperature margin of 15°C above the saturated temperature is usually acceptable.

Good design practice for spray attemperator systems

It is also advisable to install an annular sleeve in the pipework downstream of the nozzles to increase the velocity of the steam, and thus encourage faster evaporation of the water droplets.

The sleeve also protects the internal surfaces of the pipework from water droplet impingement, and is usually designed to be removable so that it can be replaced when necessary.

As discussed for the mud drum attemperator, it is advisable to limit the steam temperature at the outlet of the primary superheater to 410°C in order to avoid the use of expensive components in the inter-stage pipework.

It is good practice for the spray water control valve to fail open to ensure that spray water is introduced into the steam under all circumstances, thereby avoiding excessively high steam temperatures in the downstream components.

In addition, the inter-stage piping should be designed to avoid flooding the superheater banks should the valve fail when the steam flow rate is low. Where possible, the attemperator should be installed below the level of the superheater manifolds and the inter-stage piping should have a steam trap installed at its lowest point to drain excess water. The steam trap should be sized to drain water at the same rate as it could be introduced through the failed control valve. As an additional safety measure, an actuated valve with a level switch could be installed at the lowest point of the steam piping to ensure that excess water is removed effectively. This will, however, introduce additional cost and complexity.

Finally, it is advisable to arrange the spray water supply piping with a U-leg at the attemperator to provide a water seal that will prevent the propagation of steam into the water supply line should there be a loss of spray water.

Comparison of attemperator systems

To select an appropriate attemperator technology for a specific project, a number of aspects should be considered. This section compares the mud drum attemperator and the spray attemperator with respect to their performance characteristics and capital cost.

Performance comparison

The performance of an attemperator system can be evaluated in terms of the range of loads for which the boiler is able to supply a controlled steam temperature. For this aspect of the investigation, operational data from previous projects were used to determine the expected performance of the two attemperator systems installed in a 100 tph dual fired boiler operating at 31 bar and 385°C.

The superheater banks were designed specifically for each system to optimise performance and comply with the respective design parameters. The total heat transfer areas for the two designs are given below.

	Spray type attemperator	**Mud drum attemperator**
Primary superheater HT area, m^2	351	365
Secondary superheater HT area, m^2	156	102

A standard mud drum attemperator with 20 elements was used for this investigation. The standard unit is designed to fit into a standard JT mud drum with adequate clearance to expand the mainbank tubes without disturbing the attemperator tube bundle. The total superheater heat transfer area for the mud drum attemperator system had to be less than that of the spray system to comply with the design constraint of 60 to 80% steam flow through the attemperator at MCR on bagasse.

The performance of the two attemperator systems for bagasse and coal firing are summarised in Tables 2 and 3 below.

Table 2. Comparison of mud drum and spray attemperator performance for a range of boiler loads while firing bagasse.

Boiler load		**[tph]**	**110**		**100**		**40**	
Attemperator type			**MD**	**Spray**	**MD**	**Spray**	**MD**	**Spray**
Steam temperatures	Primary S/H outlet	[°C]	392	396	388	390	342	333
	Secondary S/H inlet	[°C]	300	274	304	278	338	329
	Main steam	[°C]	385	385	385	385	374	385
Steam pressures	Steam drum	[kPa(g)]	3739	3630	3618	3545	3200	3197
	Attemperator dP	[kPa]	145	145	97	120	37	35
Steam flow (MD attemperator)	Through attemperator	[tph]	89.8		73		0	
	Percent through	[%]	82		73		0	
Spray water flow	Flow rate	[tph]		14.8		12.4		0.1
	Percent	[%]		13.5		12.4		0.3

The spray attemperator is able to maintain a final steam temperature of 385°C for the full range of boiler loads while firing bagasse. The standard mud drum attemperator system, however, is only able to achieve a final steam temperature of 385°C for loads greater than 50 tph while firing bagasse.

At 100 tph, 73% of the steam passes through the mud drum attemperator, which leaves an additional margin of 27% for upset conditions. This is within the design limits for this type of attemperator. At MCR with the spray type attemperator, 12.4% spray water is required. This is higher than the generally recommended value of 11%, but is acceptable according to the manufacturer of the venturi attemperator.

Table 3. Comparison of mud drum and spray attemperator performance for a range of boiler loads while firing coal.

Boiler load		**[tph]**	**90**		**80**		**40**	
Attemperator type			**MD**	**Spray**	**MD**	**Spray**	**MD**	**Spray**
Steam temperatures	Primary S/H outlet	[°C]	344	336	339	331	315	307
	Secondary S/H inlet	[°C]	332	313	335	317	310	305
	Main steam	[°C]	385	385	385	385	337	348
Steam pressures	Steam drum	[kPa(g)]	3444	3486	3380	3420	3196	3200
	Attemperator dP	[kPa]	37	95	35	85	37	40
Steam flow (MD attemperator)	Through attemperator	[tph]	9.9		0		0	
	Percent through	[%]	11		0		0	
Spray water flow	Flow rate	[tph]		2.3		1.2		0
	Percent	[%]		2.6		1.5		0

The standard mud drum attemperator system can only achieve 385°C at 80 tph, but not at loads less than 80 tph while firing coal. The spray type attemperator on the other hand can achieve 385°C at loads greater than 70 tph.

It is possible to design a mud drum attemperator system that is able to match the performance of the spray type attemperator by increasing the size of the heat exchanger in the mud drum from 20 to 26 elements. The incorporation of the enlarged heat exchanger makes it possible to increase the total heat transfer area of the primary and secondary superheater banks to 377 m^2 and 123 m^2 respectively.

There are two disadvantages of installing the enlarged heat exchanger into a standard mud drum. Firstly, it will require more time to install because access is reduced. Secondly, there is inadequate clearance between the heat exchanger tubes and the internal surface of the mud drum to expand the mainbank tubes when they need to be replaced. To provide adequate clearance for expansion of the mainbank tubes, the welded connections between the steam pipes and the tube bundle header boxes need to be cut so that the tube bundles can be moved. The header boxes will need to be welded back in place after the tube expansion is completed. In the case of a new boiler, the mud drum diameter can be increased to accommodate the enlarged heat exchanger where necessary. This will, however, increase the cost of the boiler.

Capital cost comparison

The capital cost of installing an attemperator system is often an important consideration when selecting an attemperation technology. The relative costs of supplying and installing the different components of the attemperator systems are given in Table 4.

Table 4. Capital cost comparison of attemperator systems.

Component	Mud drum attemperator	Spray attemperator	Integral condenser
Condenser			588
Additional pressure part work	324		39
Piping	377	208	123
Valves	298	125	42
Instrumentation	1	27	28
Totals	**1000**	**360**	**820**

The costs presented in Table 4 have been adjusted to an arbitrary common base for the purposes of comparison. It can be seen that the relative cost of a spray type attemperator with integral condenser is 1180, which is 18% higher than the standard mud drum attemperator system.

The enlarged mud drum attemperator costs about 20% more than the standard version. Furthermore, the costs of retrofitting a mud drum attemperator into an existing boiler are higher than those for a new boiler (as shown in Table 4) due to the difficulties associated with drilling large bore holes in the mud drum for the steam pipes *in situ*.

Table 4 does not include the cost of the steelwork required to provide an additional level for the condenser.

A coal fired boiler usually requires a feedwater heater to increase the temperature of the boiler feedwater to above 130°C before entering the economiser and hence prevent condensation of volatised sulphuric acid on the gas side of the economiser tubes. Since it is common for a feedwater heater to be installed in the mud drum of the boiler, an external feedwater heater will be required if a mud drum attemperator is selected.

Maintenance and operational considerations

To conduct a holistic comparison of the two attemperator systems, the maintenance and operational characteristics need to be considered. In this section, observations are made on the complexity of each system in turn to give an impression of the maintenance requirements.

Mud drum attemperator system

The mud drum attemperator system is relatively simple and therefore requires little maintenance. The only components with moving parts are the two butterfly valves and their associated actuation equipment. Even so, neither valve requires a tight shut-off and would therefore seldom need to be removed for inspection or repair. After removal, the valves would need to be replaced very carefully to ensure that the tandem linkages are installed correctly.

Should a leak occur in the heat exchanger tubes, it is possible that water could infiltrate the steam system. This water should drain through the steam traps installed in the steam piping, provided the traps are operating properly. Although the risk of damage to equipment is therefore low, the operation of the steam traps needs to be checked regularly. The integrity of the attemperator tubes can be checked with an annual pressure test. If a larger heat exchanger unit is installed in a standard mud drum, then the removal and installation of mainbank tubes would require the steam supply pipe to be cut to afford access to the tubes.

The mud drum attemperator steam piping is relatively simple and compact, with flanges only at the control valves and at the connection to the mud drum. There is consequently a low risk of steam leaks.

The only instrumentation required for the mud drum attemperator is a pair of thermocouples for measuring the steam temperatures at the inlet and outlet of the heat exchanger. These instruments require minimal maintenance.

Spray attemperator system

The spray water system includes one spray water control valve, the attemperator unit, and various manual isolation valves. The spray system is therefore more complex than the mud drum attemperator system and more maintenance-intensive.

As mentioned previously, the steam trap in the inter-stage piping is vital to ensure that the superheater banks do not become flooded in the event of a spray water control valve failure. The steam trap therefore needs to be checked regularly to ensure that it is operating correctly.

Since the steam piping for the spray type attemperator has only two flanges at the spray nozzle unit, the risk of steam leaks is relatively low. There are nevertheless a number of valves in the spray water piping, which increases the chance of water leaks.

As for the mud drum attemperator, only two thermocouples are required for the steam line. The spray water line, however, includes a thermocouple, pressure transmitter and a conductivity probe to monitor the solids content of the spray water. This additional instrumentation increases the cost of maintenance.

Integral condenser system

The only valves in the condenser system are for instrumentation isolation and draining the vessel. Since these are not involved in the general operation of the system, they require minimal maintenance.

There is a danger of contamination of the condensate with the boiler feedwater if there is a leak in the tube bundle. Since the boiler feedwater contains chemicals, contamination of the spray water poses a risk to the attemperator spray system. If the contamination is not detected, the chemicals could block the spray nozzles or precipitate out on the internal surfaces of the boiler pressure parts or on the turbine blades. A conductivity probe is included in the spray water system to alert the operator immediately of the presence of solids in the spray water. It is therefore imperative that the operation of the probe be checked on a regular basis.

The condenser system includes saturated steam pipework, spray water supply pipework, overflow pipework and various drains. This relatively complex pipework system has flanged connections on the steam drum and condenser, which increase the risk of leaks. In addition, the spray water storage vessel is fitted with a remote level indicator, which could require maintenance.

Summary

The maintenance characteristics of the systems are summarised in Table 5.

Table 5. Summary of maintenance characteristics of the attemperator systems.

Component	Mud drum attemperator	Spray attemperator	Integral condenser system
Valves	Few valves. Linked butterfly valves. Low maintenance.	Various valves. Moderate maintenance.	Few valves. Minimal maintenance.
Heat exchanger and risk to downstream equipment if damaged	Mud drum heat exchanger. Low risk of contamination.		Contamination of spray water with feedwater. High risk of damage to downstream equipment if not detected.
Steam traps	Require regular checking.	Require regular checking to minimise risk of flooding superheater banks.	Not applicable.
Piping	Simple arrangement. Few flanges for steam leaks.	Simple arrangement. Few flanges for steam leaks. Higher potential for water leaks in spray water piping.	Various pipelines. Potential for leaks.
Instrumentation	Low maintenance.	Instrumentation for spray water line. Moderate maintenance.	Low maintenance.

The information in Table 5 shows that the mud drum attemperator system is simpler than the spray type attemperator system and consequently requires less maintenance.

Conclusions

In a boiler designed for coal and bagasse firing with a single stage superheater, the final steam temperature for coal firing is consistently lower than that achieved while firing bagasse.

Attemperators with multiple superheater stages can be used to control the steam temperature for a range of boiler loads and to achieve higher steam temperatures while firing coal.

A mud drum attemperator can supply steam of a higher purity than a spray attemperator because it is not mixed with the cooling medium.

Mud drum attemperators are generally suited to boilers operating at pressures up to 45 bar because boilers operating at higher pressures are usually of the monodrum design and require multiple attemperator stages.

A two stage superheater with spray type attemperator is able to control the steam temperature effectively from 40% to 110% MCR while firing bagasse, and from 70% to 90% while firing coal.

A standard mud drum attemperator system is able to control the steam temperature effectively from 50% to 110% MCR while firing bagasse and can achieve the required steam temperature at 80% MCR while firing coal.

A larger mud drum attemperator can achieve the same performance as the spray attemperator, but leaves insufficient clearance for the expansion of mainbank tubes in a standard mud drum.

A spray type attemperator with integral condenser system costs about 18% more than a standard mud drum attemperator.

The spray type attemperator system with integral condenser is more complex than a mud drum attemperator and is therefore more maintenance intensive.

A standard mud drum attemperator system is simpler and more cost effective than a spray type system with an integral condenser.

An enlarged mud drum attemperator can be designed to give the same performance as the spray type system, but at a cost that is approximately equal to a spray type system with condenser.

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