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POT GAS HEAT RECOVERY AND EMISSION CONTROL

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Abstract

Substantial quantities of heat is released to the ambient through pot exhaust, and present pot gas temperature of 150-180ºC also affect the operation of the Gas Treatment Centres (GTC). Standard polyester filter bags used in the GTC can only sustain gas temperatures of 135ºC. A sharp rise in fluoride emissions (HF) is seen as pot gas temperatures exceed 100ºC. Dilution of the pot gas with ambient air is used to achieve acceptable GTC gas temperatures (110-115ºC) and emission levels. This results in a need for substantial increase in the filtration capacity of the GTC.

A heat exchanger has been developed to combine heat recovery and cost efficient cooling of pot gas. The technology has been tested on pot gas in a pilot plant. Promising stable heat exchange and pressure drop, and minimum fouling deposits over longer test periods have encouraged Alstom to continue the development into a commercial product.

Introduction

The gas from the electrolytic production of aluminium (pot gas) contains high concentration of HF gas that typically is recovered in the dry scrubbing process that takes place in the gas treatment centers (GTC's). The valuable and potentially toxic HF gas is in the scrubbing process absorbed on the fresh alumina and thereby recycled to the pots.

The HF emission level after the GTC is normally very low, and <0.5 mg/Nm^3 HF is a typical smelter requirement around the world. Elevated gas temperature does however affect the absorption process in a negative way. It is known that many plants struggle to meet the emission levels during the high temperature season, and in some cases doubling of the emissions during the summer period has been measured.

The standard solution to the high pot gas temperature problem is to add dilution air into the gas duct upstream the GTC's. This does however require that more fan capacity and filter compartments are added to the GTC's since more gas volume must be treated. The added volume of gas increases the power consumption of the fans, and the required additional filterbags will also increase the maintenance costs significantly.

Another solution that is used is to add water sprays into the gas. This does however require large blowers to atomize the water, and still there will always be some concern regarding clogging of the nozzles, hydrolysis of the filter bags, increased corrosion, and increased water content in the enriched alumina. Also lots of water is required, and this is a costly and a sparse commodity in many locations in the world.

It is therefore a better solution to apply indirect heat transfer in a heat exchanger (HEX). In this case there will be no added volume of air or water to the pot gas, and the recovered heat can be put to productive use if a suitable application is found locally. Even if application of the recovered heat is not feasible, it is calculated that the use of a HEX is in most cases less expensive compared to the standard dilution air solution, and at the same time significantly less power is required for operation of the GTC fans.

The HEX will facilitate an almost constant air flow to the GTC independent of the ambient temperatures. This improves the stability of the system over time, and makes operation conditions easier to optimize.

The main challenge for the HEX is that the pot gas is contaminated with alumina, fluorides, and other trace contaminants that can deposit on the heat transfer surfaces as scale or other types of fouling dust layers. Deposits that are formed by impact are many times referred to as scale, and often grow opposite to the flow direction after impact on some sharp edged object.

Scales are typically dark grey in color, very dense, and high gas velocities can increase the scale formation. Other deposits are often more light grey in color, more porous, and may cover the entire surfaces.

In general scales can cause blockages to many parts of the alumina handling systems. N. Dando [1] suggests that highly exothermic re-hydration of alumina is the key energy source for the scale formation. Adding water either as humid dilution air or direct water injection may therefore increase the scaling problems.

Alumina can give uncontrolled wear and erosion of the steel surfaces. Both fouling and wear can be detrimental to the heat transfer and equipment durability. Pilot size tests over sufficient time with pot gas is necessary to predict the performance of full size installations.

Several attempts on energy recovery from dirty pot gas have previously been attempted [2,3]. Typically heavy scaling has prevented operation to continue for more than some weeks in these previous tests. Therefore, a HEX design must be robust, with minimum risk for scaling and easy access for cleaning if needed.

Based on the evaluation of previous experience with heat recovery, and existing knowledge on gas handling in the aluminium industry, a new HEX design has been designed (patent pending [4]). In principle the chosen design is a counter flow vertical HEX of fire tube design containing several parallel pipes with water on the outside, and gas in the inside. The hot dirty pot gas enters the top of the HEX, and the water enters the bottom.

For this type of geometry many semi empirical equations can be applied provided the surfaces are clean (no fouling).

The main uncertainty is the fouling rate. If too frequent cleaning of the tubes is required, then the operation of the HEX may either require some automatic cleaning device, or the operation may be too cumbersome.

Several tube sizes have been tested in pilot size with real life dirty pot gas for more than 18 month without any cleaning. Further tests are ongoing to optimize the operation parameters.

Theory

In this section the main equations for evaluation of the measured data are presented. Of special interest is the development of the heat transfer and the friction factor on the dirty gas side over time. These are calculated based on the gas flow measurements, the pressure drop and the gas temperatures drop across the HEX.

For thin walled tubes with thin layers of deposit, the overall heat transfer coefficient, U *(W/m²ºC)*, can be written:

$$\frac{1}{U} = \left(\frac{1}{h_g} + \frac{1}{h_w} + \frac{l_{steel}}{k_{steel}} + \frac{l_{fouling}}{k_{fouling}} \right) \tag{1}$$

The gas side heat transfer coefficient h_g *(W/m²ºC)* is given by:

$$h_g = \frac{\overline{Nu_D} k}{d_h} \tag{2}$$

Typically the Nusselts number is given as a function of the Reynolds number, and the Prantl number, and some constants to be determined empirically. Several such semi empirical relations for the average Nusselts number $\overline{Nu_D}$ can be found in the literature [5,6,7], and they may be considered similar. k *(W/mºC)* is the gas conductivity, and d_h is the hydraulic diameter in meters.

For our case the heat transfer coefficient on the water side, h_w *(W/m²ºC)* is much higher than for the gas side. Also h_w may be considered relatively constant over time, and has been neglected since the main focus is to evaluate the development of fouling over time.

Similarly the heat conduction of steel, k_{steel} *(W/mºC)*, is very high in comparison to the other terms, and thus l_{steel}/ k_{steel} is neglected, where l_{steel} is the thickness of the thin walled steel tube in meters. The final term is the heat resistance in the deposit layer with a thickness $l_{fouling}$ *(m)*, and a heat conductivity, $k_{fouling}$ *(W/mºC)*. When the tubes are clean this term is zero, thus equation 1 is then reduced to:

$$U_{th} \approx h_g \tag{3}$$

Where U_{th} *(W/m²ºC)* is the predicted or theoretical overall heat transfer coefficient. The overall heat transfer coefficient is by definition equal to:

$$U_m = \frac{q_g}{A \Delta t_m} \tag{4}$$

Where U_m*(W/m²ºC)* is the measured overall heat transfer coefficient including the insulating effect of a fouling deposit layer, A *(m²)* is the heat transfer area of the tubes , q_g is the heat transferred from the gas in Watts, and Δt_m is the overall logarithmic mean temperature difference in ºC, i.e.:

$$q_g = m_g \, c_{pg} (t_{gin} - t_{gout}) \tag{5}$$

and

$$\Delta t_m = \frac{\Delta t_{in} - \Delta t_{out}}{\ln\left(\Delta t_{in} / \Delta t_{out} \right)} \tag{6}$$

Where m_g is the gas flow in *kg/sec*, c_{pg} *(J/kgºC)* is the specific heat of the gas, t_{gin} *(ºC)* is the inlet gas temperature, t_{gout} *(ºC)* is the outlet gas temperature, Δt_{in} *(ºC)* is the temperature difference between the gas and water at the inlet, and Δt_{out}*(ºC)* is the temperature difference between the gas and water at the outlet.

Finally a heat transfer ratio, *HR (%)*, is defined as follows:

$$HR \equiv 100 \frac{U_m}{U_{th}} \tag{7}$$

In the calculations of the *HR* ratio, all the parameters in the U_m term are based on independent measurements of gas temperatures and gas flows. U_{th} or h_g is for a large part determined from semi empirical relations for the calculations of the Nusselts number.

The correlation between the two will therefore indicate the ability to predict the future heat transfer rates. In addition the development of the *HR* over time is an important indication on the fouling rate. Since the HEX is started with clean tubes the *HR* should then be equal to 100%. Over time it should be expected that the *HR* is reduced if there is significant deposit growth on the surfaces.

Another important indicator of fouling is the development of the friction factor, f, over time. f is calculated from the measured pressure drop across the HEX, ΔP*(Pa)*, and the dynamic gas pressure, P_d *(Pa)*, i.e. as follows:

$$f = \Delta P \frac{d_h}{l(1+c)P_d} \tag{8}$$

Where c is the sum of the inlet and outlet pressure loss factors, and l is the length of the tubes. The friction factor depends on the relative roughness of the tubes and the Reynolds number as typically presented in the well known Moody diagram. For turbulent flow Haaland [8] provide the following equation for the friction factor:

$$1 \;/\; f^{1/2} \approx -1.8 \log\left(\frac{6.9}{\text{Re}} + \left[\frac{k/d}{3.7} \right]^{1.11} \right) \tag{9}$$

Where d is the tube diameter in meter, and k is the roughness in m. A deposit layer in the range of 1 mm means that the friction factor can more than double compared to that of clean steel tubes. Since the deposit layer may grow in a relatively similar manner independent of the tube diameter, the friction factor for small tubes may be greater than for large tubes.

Experimental

As shown in Figure 1, pot gas is collected in a split stream at approximately 110-120ºC isokinetically through an insert into the main gas duct. The hot gas flows through a venturi gas flow tube for measurements of gas flow before it enters the top of the HEX. The gas is cooled to roughly 90-100ºC, and the heat is exchanged with water that feeds countercurrent into the bottom of the HEX.

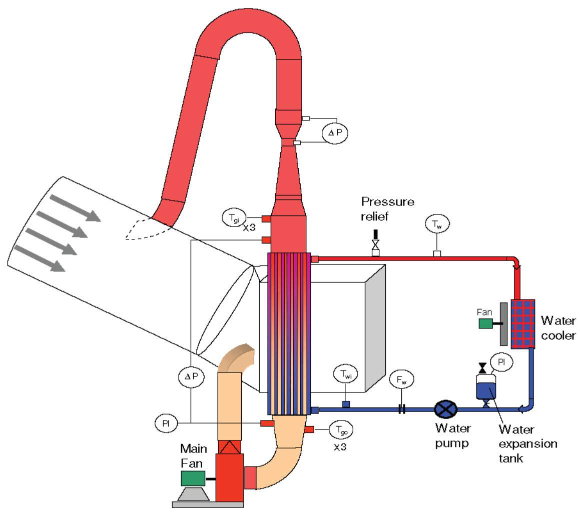

Figure 1: HEX pilot test showing real life pot gas in a slip stream arrangement.

The hot water, roughly 80ºC, is cooled in a water cooler with ambient air to approximately 60ºC, before the water flows back into the HEX in a closed loop. All temperatures, pressure drops and flows are monitored in an automatic data collection system. From the raw data heat balances, heat transfer coefficients, and pressure loss coefficients are calculated and compared to theoretical values.

Two main designs have been tested, large tubes, and small tubes where the small tube diameter is approximately 50% size of the larger tubes. In total 8 tests have been performed for different operation parameters, and durations. The main parameter that has been varied is the gas velocity. Tests 1 and 5 started with clean tubes (large and small tubes), and no cleaning of the tubes has been performed between the tests.

Accuracy of the measurements

High quality, standard commercial instruments for temperature, pressure and flow rates have been implemented in the measurements such as PT 100 temperature instruments, and for Δp measurements, type PDM75. Standard methods for commissioning the HEX including maximum, and minimum range testing have been performed.

The calculated results have been tested for sensitivity of the measurements. By imposing a small error on each of the measurements one at the time, the corresponding error in the result can be calculated. By this method it can be found that one of the most critical measurements is the gas temperatures in and out of the HEX. One reason for this is the relatively small temperature difference across the HEX, typically 20ºC.

Even if three Pt100 elements arranged 120 degrees apart are used at each measurement cross section an almost 20% deviation in the predicted heat transfer ratio was calculated at the start up of test 5 (first test with small tubes). With a sensitivity analysis it was calculated that this deviation would correspond to a 2.5ºC error in the outlet gas temperature.

Fortunately it is possible to calculate the heat balance by two independent sets of measurements, i.e. the heat transferred from the gas and the heat absorbed by the water. By comparing the two heat flows it was found a corresponding shift in the outlet gas temperature, and it was decided to move the outlet gas temperature location approximately 0.5 m downstream. After this the calculated *HR* and the heat balance re-aligned.

The explanation for the deviation appears to be that the distribution in the gas temperatures at the outlet was different from tests with the large tubes setup compared to the tests with small tubes.

Another error in the measurements was clogging of the pipes for the ΔP pressure sensors. Approximately once a month it was therefore necessary to purge the pipes by high pressure air. It is relatively easy to see when this happens from the measurement curves, and for clarity these data points have been omitted.

The gas flow is measured with a standard shape venturi made according to ISO standard No. 5167, rough welded sheet. This measurement is known to be reliable, provided the internal duct diameters do not grow with deposits, and sufficient length of undisturbed duct length is provided. Deposits inside the venturi were not detected during the tests.

The water flow was measured with a Prosonic flow type 92F25.

In general the distribution of water and gas temperatures inside the HEX is one of the main uncertainties in the evaluation since inlet, and outlet temperatures are measured only. One assumption in the analysis that is related to this is the assumption of log mean temperature difference (equation 6). Errors in the internal distribution will however not change the overall heat balance, and thus the main results.

Results

The development of the heat transfer over time is shown in Figure 2 (large pipes), and in Figure 4 (small pipes). Four tests have been performed for each of the tubes as shown in Table 1.

Table 1. List of experiments and observations

Test	Duration (weeks)	Observations from curves
1	7	Large tubes, *HR* and *f* steady
2	6	Large tubes, *HR* varies slightly, *f* steady
3	8	Large t., *HR* slight reduction , *f* steady
4	2 days	Large tubes, *HR* back to start level.
5	5	Small tubes, *HR* and, *f* steady
6	11	Small tubes, *HR* and, *f* steady
7	24	Small tubes, *HR* varies, *f* increases
8	2	Small tubes, *HR* varies, *f* increases

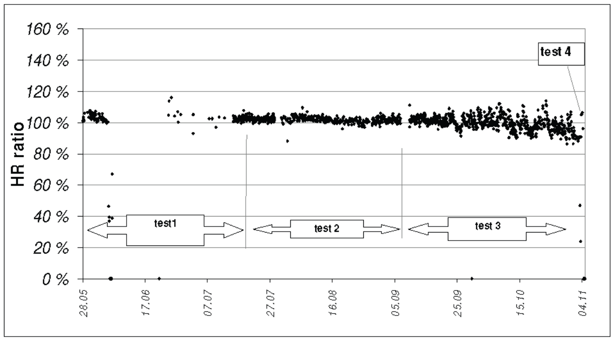

Figure 2: *HR* for large tubes versus real time. Start test 5, 27 May 2007, to test 4, ended November 4. *HR* better than 100% means that measured heat transfer is better than theoretical as seen from equation 7.

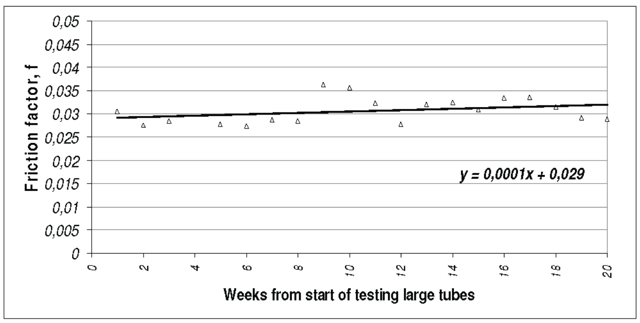

Figure 3: Friction factor for large tubes versus real time. Start test 5, 27 May 2007, to test 4, ended November 4.

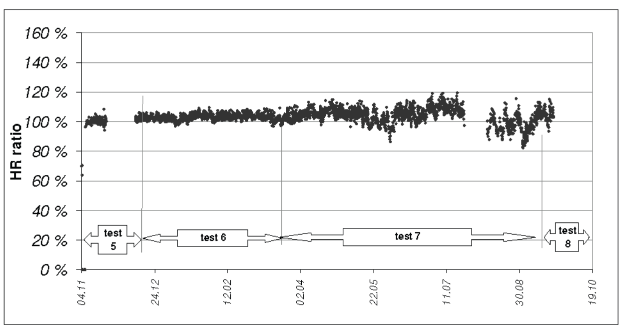

Figure 4: *HR* for small tubes versus real time. Start test 5, 4 November 2007, to test 8, (still ongoing). *HR* better than 100% means that measured heat transfer is better than theoretical as seen from equation 7.

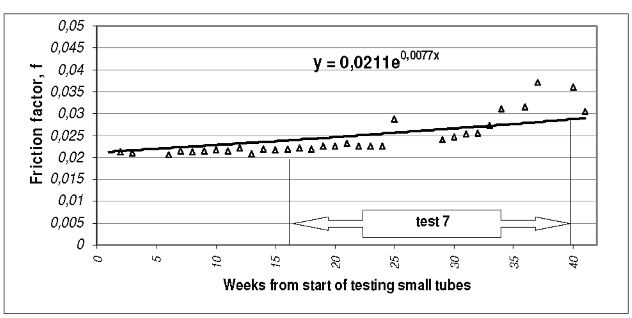

Figure 5: Friction factor for small tubes versus real time. Start test 5, 4 November 2007, to test 8.

The HEX have been stopped and inspected inside several times. A thin layer of dust and a few black spots has been observed on the heat exchanger surfaces, but no major scaling or wear is observed, see Figure 6.

Discussion

As shown the *HR* curves (Figure 2 and 4) the measured heat transfer is predicted well. No significant reduction in heat transfer over time is seen. Some spread in the data, and possibly a slight drop in the *HR* are observed during test number 3. For test 4 the *HR* is back on the start level even without any cleaning of the tubes. For tests 5 through 8 the spread in the data points are similar, and the average HR is close to 100%.

One reason for the variations in the data points is that the water cooler from time to time struggled to maintain a water inlet temperature of 60ºC to the HEX. Also the spread will be affected by seasonal and daily ambient temperature variations, and corresponding changes in the heat sink effects of the system. For the same reason the system will be more sensitive to spread in the data points at low gas velocities.

The large tubes did not show any significant development in friction factor as shown in Figure 3. After approximately 6 month of testing of the small pipes (test 7), an increase in the friction factor was observed. Since the development is very slow, the test was extended to more than 6 months for the given test parameters. Over this time a 30% increase in the friction factor is observed as shown in Figure 5. Further testing will be done to optimize the operation parameters.

Inspection of the tubes reveals that there is a thin layer of deposits on the tube surfaces, and this can explain the increase in the friction factor during test 7. Since the heat transfer does not show a corresponding reduction one can conclude that the deposit exhibit a relatively high thermal conductivity, and that the pressure drop may be the first indicator of fouling in the HEX.

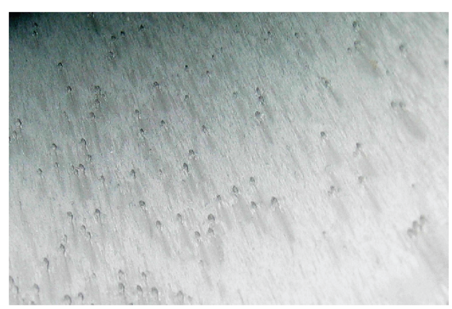

Figure 6: Typical fouling layer with a thickness in the range of 1 mm.

All in all, the result is a breakthrough that means that full scale HEX in dirty pot gas is now possible. Cleaning the tubes at estimated 12 month intervals may be required.

Scale up challenges

The gas flow through the pilot HEX is roughly equivalent to the gas flow from one pot, and the design heat transfer is up to 50 kW. Full scale HEX tubes have been tested.

The remaining challenge is to ensure an even distribution of gas and water across the HEX. CFD simulations combined with Alstoms expertise in this field have resulted in a compact HEX integrated into the GTC main ducts. One such design is shown in Figure 7.

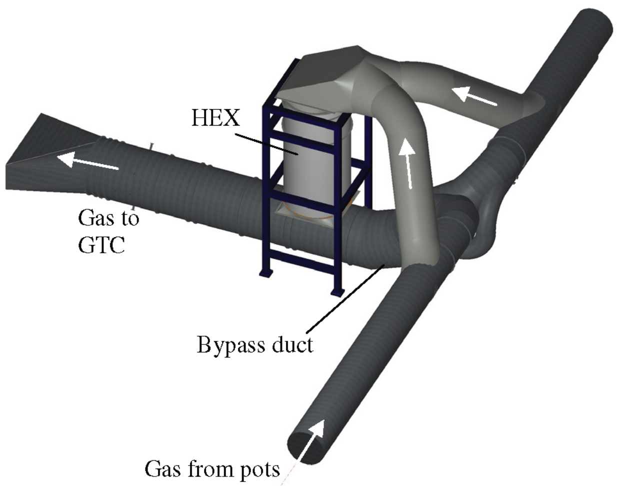

Figure 7: Typical HEX arrangement with guidevanes, and online access for cleaning and inspection.

A system of guidevanes and aerodynamically designed ducts are used to distribute the gas.

It is important that no part of the HEX is left with dead pockets or recirculating flows of hot water. This is one reason for the chosen vertical design of the test unit. In the vertical design, the distribution of the water flowing into the bottom will be aided by the natural buoyancy of the hotter water.

Cleaning of deposits on the tube surfaces

As discussed 12 month or even longer cleaning intervals may be required. It is remarkable the HEX may operate satisfactory for such long periods without any cleaning. The data indicate that at least for some operating conditions, a stable situation can be achieved where a balance between fouling and erosion of the dust layers may be obtained. Variations in the gas velocity and the alumina content will be further explored in future optimization of the design.

It is recommended that provisions are made for inspecting and cleaning the tubes. This is a relatively easy task with the straight tubes in the fire tube HEX design. Several cleaning devices for this type of design are available commercially. One device is a rotating steel brush driven by pressurized air that can be lowered down the individual tubes one by one.

Several automatic or semi-automatic cleaning methods are also ready to be investigated, including horns, mechanical shocks, bullet cleaning, etc, but since the frequency of the cleaning is in the range of one year, manual cleaning may be the acceptable solution.

Pressure drop and Energy consumption

The main fans located after the GTC's must be capable of handling additional pressure drops across the HEX, and in the ducts transporting the gas to and from the HEX. The pressure drop across the HEX is a function of how much the gas is to be cooled, and the hot water temperature requirements, and represents about 95% of the energy consumption for the HEX operation.

The remaining 5% is estimated for running the recirculation pumps. In addition some energy is required to run the water cooling fans if the heat is to be dumped to the atmosphere.

A typical HEX case that has been investigated is to cool pot gas from approximately 150ºC to approximately 120ºC. The water temperatures may be as low as 30ºC if the heat is to be dumped, or up to 120ºC pressurized water if a consumer of hot water is considered.

In these cases the HEX pressure drop has been calculated to 800-900 Pa including the inlet and outlet transitions (see e.g. Figure 7). With a green field installation this may be the total additional pressure drop requirements for the HEX, while in other cases there has to be installed additional ducts to and from the HEX which may increase the total pressure drop by additional 300-400 Pa.

Often the pressure drop requirement for the GTC main fans is in the range of 5 kPa. The additional HEX pressure drop of 0.8 to 1.2 kPa increases the total pressure drop, but at the same time the total flow is lower to the main fans since no dilution air is required.

The power requirements for the fans are:

$$P = \frac{Q\Delta P_{fan}}{\eta} \tag{10}$$

Where η is the fan efficiency, Δp_{fan} is the fan pressure increase, and Q is the fan actual flow. Assuming a constant fan efficiency the energy saving with a HEX can be estimated as follows:

$$\frac{P_{dill}}{P_{HEX}} = \left(\frac{T_{pot} - T_{dill}}{T - T_{dill}}\right)\frac{\Delta P_{fandill}}{\Delta P_{fanHEX}} \tag{11}$$

This relation underestimates the power saving with a HEX since constant specific heats, and perfect mixing are assumed. Still, for many cases equation 11 will be a good first approximation. If the dilution air temperature, T_{dill}, is 20ºC, the pot temperature, T_{pot}, 150 ºC, and GTC temperature, T, 110 ºC, a ratio: P_{dill}/P_{HEX} of 1.2 is calculated. This means that even if the pressure drop increases with up to of 20%, the power required for the fans is reduced with more than 20%.

In tropical countries 50 ºC dilution temperature may be the case. Then the corresponding value is almost 40%, and as the pot temperatures increase, 50% power savings for the main fans are possible.

Pot amperage creep is a typical retrofit case where the gas temperature is increased by 15-20ºC corresponding to the increase in the pot amperage. For this situation a very compact HEX has been developed for easy retrofit and minimal disturbance and modification of the GTC including the use of existing fans.

As shown in equation 10 the power requirements increases proportionally with the pressure drop. This is one of the main advantages of the fire tube design, the pressure drop is very low, maybe about half the values than can be expected from other types of HEX designs.

Applications of the recovered energy

The use of water to transfer the heat from the gas is very efficient. In addition the water coolers for dumping the heat in the water are very compact. Therefore, even if there is no use for the heat in the water, a gas/water type HEX is most likely the most efficient way to reduce the gas temperature to the GTC.

Obviously a large benefit can be obtained if the recovered energy can be used for a practical purpose. Depending on the gas temperatures, roughly 10% of the electrical energy input to the pots may be recovered from the pot gas as heat. This indicates an energy recovery potential for many smelters up to 100-300 MW that can be used.

Many possible applications can be considered such as heating or drying of raw materials, or heating of buildings. In hot climates an absorption cycle may be applied to convert hot to cold water for cooling, or as energy source for desalination plants [4]. The recovered energy may also be converted to electrical energy through a conventional Organic Rankine Cycle.

Conclusion

A fire tube heat exchanger on pot gas has been developed and tested. Heat transfer and fouling have been monitored over time. In general the predictions of the heat transfer are consistent with observed measurements, and the friction factor is in line with expected values. In contrast to previous designs where severe fouling prevented continued operation, the present design is relatively clean even after more than 1.5 years of total operation time.

This breakthrough in the HEX technology can reduce the total investment cost of the GTC, and at the same time reduce the power consumption significantly. Large quantities of energy can be recovered which represents a significant cost saving potential. With steadily increasing energy prices, and increasing pot gas temperatures, energy recovery will be more and more attractive.

The fire tube design is simple and robust and gives very low pressure drop. In many cases the HEX can be installed on existing GTCs without any changes to the existing main fans or filter systems.

To sum up; the HEX will for many cases be the optimum solution to reduce the GTC gas temperature and the corresponding HF emissions with:

- Lower investments
- Lower power consumption
- Lower maintenance costs (reduced number of filterbags to be replaced over time)
- Stable gas flow independent of ambient temperatures
- Robust technical solution and easy cleaning access
- Easy retrofit solution also for future pot amperage increase

Finally the HEX gives added opportunities for cost beneficial and environmentally sound application of recovered energy such as heating or cooling of buildings. Other opportunities should also be explored such as energy source for desalination, power plant utility or raw material preheating.

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