Experimental and modelling evaluation of possible solutions for compact design of producer-gas heat exchangers

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Abstract. An experimental and numerical campaign was carried out to investigate the performances of tubes in shell heat exchanger applied to cool down the producer gas of a small-scale commercial wood biomass gasifier built by All Power Labs. Producer gas is contaminated with soot and tars that condensate under 300 °C, for this reason the heat exchanger was designed in order to have an in-situ cleaning mechanism. The heat exchanger was tested both with standard plain tubes and with a set of metal twisted tapes (TT) with the aim of enhancing the heat transfer between producer gas and water. Temperatures and mass flows analyses shown a maximum increase of thermal power output and overall heat transfer coefficient respectively of 19% and 76% when TT are applied to the standard plain tubes in shell heat exchanger. Numerical simulations shown a consistency in the trends giving an average discrepancy with experimental results of the 12%.

1. Introduction
Small-scale gasification power plants consist in systems where a solid carbonaceous fuel (mainly wood biomass) is converted into a gas then used to fuel an internal combustion engine. Producer gas, also known as syngas, is the gaseous fuel generated in a gasification system and it is mainly composed of H₂, CO, CO₂, H₂O, CH₄ and N₂. Downdraft gasifier fuelled with wood biomass yields a syngas with a calorific value that ranges from 4 to 7 MJ/Nm³ [1].
Gasification reactions take place at high temperatures up to 1100 °C, the overall energy balance is endothermic and the different reaction kinetics drop in speed when temperature falls below 600 °C [2]. For this reason, the large majority of reactor designs produce syngas in a temperature range between 500 and 700 °C depending on the reactor capability to self-recover some of the gas heat for internal use [3, 4, 5]. Hot gas must be cooled down before reaching the power generation unit otherwise it will reduce the engine volumetric efficiency and will increase the risk of knocking [6].
Gas temperature conditioning is also a strategic part of the filtration stage. At different temperatures the pollutants traveling within the gas can be filtered through a variety of systems. The control over the gas temperature allows to selectively work on specific pollutants. Dust and particulate matter are usually filtered at high temperature (300 to 600 °C), while tarry compounds and other condensable compounds are separated at lower temperature [7].
This work focuses on the modeling and the design of a water-gas heat exchanger above water condensation. The challenge is to take advantage of the vast literature about flue gas turbolated heat exchanger [8-10] in biomass boilers and to import it into a gasification systems. Several techniques are available to improve the heat exchange between two different continua in tubes and shell heat exchanger,
such as: coil wire inserts, conical tube inserts, helical screw inserts and twisted tape inserts [11]. The common ground of these devices is to use the energy already available in the system to work, they can thus be named as passive systems. They aim at reduce the convective resistance between the solid and the fluid domains by promoting the recirculation of the flow reducing the thickness of the boundary layer [11, 12].

It is known from literature [13] that the regularly spaced twisted tapes perform better than the full length twisted tapes in terms of pressure drop at constant heat duty. Since, here the aim of this device is not only to promote the heat exchange but also to keep the tubes clean from the tar and particulate matter. In this work, a set of 7 full length (1.5 m long and 2 mm thick) TT were inserted in the 7 tubes of the HX. In this way it was avoided the use of inserts with centred rod that needs welding techniques instead of simple tape twisting. The cleaning mechanism consists in moving up and downward the twisted tapes in order to remove the soot from the tube walls and restore the heat transfer effectiveness. The final system was tested in a 20 kW small-scale commercial gasifier.

2. Materials and methods

2.1. Gasification facility

To evaluate the performance of the gas cooling device a PP30 gasifier by All Power Labs (CA, USA) [14] was used. The gasification system is composed of a gas generating unit, a gas temperature control system, a filtration stage and a spark-ignited internal combustion engine. As shown in Figure 1, the gas making facility consists in a hopper that contains the drying stage, a pyrolyzing stage, named “Pyrocoil” where engine exhaust gases provide the sufficient heat to pyrolyze the biomass, and a gasification stage where tar gases and pyrolyzed biomass are converted into producer gas passing through combustion and reduction zones. The heat exchanger, that was used to cool down the producer-gas (≈300 °C), was placed between the cyclone and the filter stage and, as cold fluid, the water coming from the engine cooling system (≈ 80 °C) was used. Typical biomasses used in these systems range from wood chips [15-17] up to agro industrial waste such as: chipped vine prunings [18], corn cobs [19], corn stover [20,21], wood residues from river maintenance [22,23], solid residues from vegetable oil production [24], solid residues from cotton crops [25], municipal green waste [26] etc. For the tests detailed in this work, vine prunings pellet were used. Table 1 shows a typical producer-gas composition from vine prunings biomass [18] and the resulting higher heating value (HHV) calculated according to Equation 1.

\[ HHV = \frac{12.75H_2 + 12.63CO + 39.72CH_4}{100} [MJ/kg] \]  

Table 1. Higher heating value and producer-gas composition, expressed in molar fraction, used to calculate the properties of the gas mixture.

| CO₂ | CO  | N₂  | CH₄ | H₂  | HHV [MJ Nm⁻³] |
|-----|-----|-----|-----|-----|---------------|
| 11.84 % | 23.99 % | 41.23 % | 1.77 % | 21.18 % | 6.20 |

2.2. Producer-gas/water heat exchanger

2.2.1. Tubes and shell HX

The heat exchanger (HX) prototype was designed to fit in the existing filtration stage and, as shown in Figure 2, the HX needed to be placed between the cyclone and the filtration stage. The other key parameter was to bound the minimum and maximum temperature of the gas to cool down. From one hand it is not advisable to exceed 150°C at the gas outlet to avoid the filter failure, but from the other hand, it is necessary to maintain the producer-gas at the minimum temperature of 50-60 °C to avoid the condensation of light tars and water [27] that cause the premature failure of mechanical components [28,29]. For this reason, the coolant water of the internal combustion engine was used. A mechanical thermostatic valve automatically regulates the minimum temperature of the coolant at 78°C while the maximum temperature (95°C) is kept under control by the cooling system of the engine.
The HX layout was chosen to be multiple tubes in shell: the gas passes through the tubes and the water runs between the outer layer of the tubes and the shell. 5 equispaced \( l_b \) baffles were added to the water path in order to reduce stagnation points and thus increase the heat transfer. The inlet of the HX \( D_{g,in} \) is connected to the cyclone with a 2” flanged joint and its outlet \( D_{g,out} \) is connected to the filtering stage with a 6” sanitary type joint. The net length of the heat exchanging zone \( l_t \) is 1500 mm and a 250 mm long plenum \( l_p \) was added on the top in order to provide better distribution of the gas and to contain the mechanism for the tubes cleaning. In the 6 inches diameter shell, 7 tubes \( D_t \) 1 ¼ inches, regularly spaced, were placed. The water inlet \( D_{w,in} \) and outlet \( D_{w,out} \) are chosen to be 1 inch of diameter. In Figure 3 the HX dimensions are detailed.

\[D_{g,in} \quad D_{g,out} \quad l_b \quad D_{w,in} \quad D_{w,out} \quad l_t \quad l_p\]

\[w=35 \text{ mm}, \quad p=200 \text{ mm} \quad \text{and thus } y=5.88\]

The clearance between the tube and the tape was designed to be 1 mm per part in order to reduce the tendency of the tape to stick to the tube surface, acting on a thicker layer of tar and particulate matter in the cleaning movement. Figure 5 shows the tubes distribution in the HX and the installation of one the mentioned twisted tape.

\[D_{g,in} \quad D_{g,out} \quad l_b \quad D_{w,in} \quad D_{w,out} \quad l_t \quad l_p\]

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Figure 1. Longitudinal section of the HX.

Figure 2. Picture of the top of the HX.

2.2.2. Twisted tape inserts

The two characterizing parameters of the twisted tapes are the pitch \( p \) and the twist ratio \( y \) that is the ratio between the pitch and the width of the tape. Figure 4 illustrates the mentioned parameters that, for this prototype were chosen to \( w=35 \text{ mm}, \quad p=200 \text{ mm} \quad \text{and thus } y=5.88\). The clearance between the tube and the tape was designed to be 1 mm per part in order to reduce the tendency of the tape to stick to the tube surface, acting on a thicker layer of tar and particulate matter in the cleaning movement. Figure 5 shows the tubes distribution in the HX and the installation of one the mentioned twisted tape.

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\[w=35 \text{ mm}, \quad p=200 \text{ mm} \quad \text{and thus } y=5.88\]

Figure 3. Longitudinal section of the HX.

Figure 4. Dimensions of the twisted tape inserts used.

2.3. Experimental setup and procedures

Four different tests were performed to assess the response of the heat exchanger at three different gas mass flow rates. Since, the gas volumetric flow rate is almost directly proportional to the internal combustion engine power output, tests were carried out at three different electrical power output of the cogenerator: 7.7, 11.6 and 16 kW, both with and without twisted tape inserts. During each test the inlet
and outlet temperatures of the gas were measured using K-type thermocouples (± 2°C), while the water inlet and outlet temperatures were measured using T-type thermocouples (± 1°C). The datalogging system used was a Picotech TC-08 thermocouple datalogger.

Both gas and water flow rate were measured using respectively a calibrated orifice meter and an impeller meter for hot water. The orifice meter was placed downstream the filtering stage of the Power Pallet and, for this reason, the temperature of the gas from the filter outlet was used to calculate the volumetric flow rate from the pressure difference measured across the orifice. The pressure loss generated in the gas stream by the heat exchanger was measured through a water U-tube manometer.

In Figure 6 the experimental setup is shown. During each test, the steady state condition was reached before to start to sample the temperatures and flow rates. The sampling lasted for 300 s for every case and the mean value of each parameter was used to setup the relative boundary conditions in the CFD model. For each test, the thermal power lost by the producer gas ($\dot{Q}_g$) and the overall heat transfer coefficient ($HTC_g$) of the tested HX version were calculated according to Eqs. 2 and 3. The thermal power is referred to the gas mass flow rate ($\dot{m}_g$) and the specific heat capacity of the gas ($c_{p,g}$). The overall HTC calculation is referred to the tubes internal surface area ($S$) and the logarithmic mean temperature difference ($\Delta T_{LM}$) calculated using gas and water inlet/outlet temperature.

\[
\dot{Q}_g = \dot{m}_g c_{p,g} (T_{g,in} - T_{g,out})
\]  
\[
HTC_g = \frac{\dot{Q}_g}{S \Delta T_{LM}}
\]  

Figure 6. Setup scheme of the instrumentation.  
Figure 7. Boundary conditions of the HX.

2.4. CFD setup

Each experimental test was replicated in a computational fluid dynamics environment simulating both the cases with twisted tapes and without those devices. The flows were considered steady, turbulent and incompressible, thermal radiation was neglected. The thermo-physical properties of the fluids were considered temperature independent except for the thermal conductivity of the gas. Instead of considering the producer gas mixture, it was considered a single gas with the properties of the producer gas. The gas composition is represented in Table 1 and it is the average results of gasification tests conducted with vine prunings [18]. The boundary conditions are detailed in Figure 7: velocity inlets and pressure outlets were used both for gas and water. To take into account thermal dispersion at the outer shell, convective boundary condition was applied by setting the ambient temperature $T_{amb}=20$ °C and the estimated convective heat transfer coefficient $h_{amb}=10$ Wm$^{-2}$K$^{-1}$. 
The turbulence model selected was the $k$-$\varepsilon$ that takes into account the transport of two different variables: the turbulent kinetic energy ($k$) and the rate of dissipation of turbulence energy ($\varepsilon$). According to [30], this method gives better results in terms of heat transfer prediction than the $k$-$\omega$ turbulence model. Since the expected Reynolds number at the tubes inlet ranges from 1000 and 2000, for each case both the realizable $k$-$\varepsilon$ [31] and the low Reynolds standard $k$-$\varepsilon$ [32] turbulence models were tested. The viscous sub-layer was solved for each of these methods. Turbulent kinetic energy ($k_t$) and turbulent dissipation rate ($\varepsilon_t$) were calculated (Equations 4, 5) and used to initialize every boundary of the system.

$$k_t = \frac{3}{2} (v I)^2$$

(4)

$$\varepsilon_t = C_\mu \frac{3}{2} k_t l_t^{3}$$

(5)

Where $v$ is the fluid velocity, $I$ is the turbulence intensity, $C_\mu$ is a parameter set to 0.09 [30] and $l_t$ is the turbulent length scale at the selected boundary.

Polyhedral mesh was used for the core volumes, while prism layers were added at the fluid interfaces to properly solve the viscous sublayer ($y^t < 1$). During mesh execution, the outlets of gas and water were extended to 10 diameters beyond the original dimensions in order to obtain a more representative computational domain. The mesh used for the plain tubes cases counts 4 million cells, while the mesh used for the $TT$ cases counts 4.3 million of cells.

3. Results

3.1. Experimental results

The engine power outputs, selected to characterize the performances of the tubes and shell heat exchanger led to three different gas mass flow rates: 0.0071, 0.0086 and 0.0103 kg s$^{-1}$. Each mass flow rate was averaged on a period of 300 s after steady state condition was reached. The gas inlet and outlet temperatures are reported in Table 2 while the temperature of the coolant ranged between 77.5 and 82.4 °C according to the engine power output selected.

| Table 2. Experimental results from the test campaign: gas mass flow and temperatures at inlet and outlet of the heat exchanger. |
|---------------------------------------------------------------|
| TWISTED TAPES       | PLAIN TUBES       |
| $m_{\text{gas}}$ [kg s$^{-1}$] | $T_{\text{gas,in}}$ [°C] | $T_{\text{gas,out}}$ [°C] | $T_{\text{gas,in}}$ [°C] | $T_{\text{gas,out}}$ [°C] |
| $(7.12\pm0.4)\cdot10^{-3}$ | 234±2 | 83±2 | 238±2 | 103±2 |
| $(8.56\pm0.4)\cdot10^{-3}$ | 271±2 | 94±2 | 264±2 | 114±2 |
| $(1.03\pm0.04)\cdot10^{-2}$ | 306±2 | 101±2 | 301±2 | 129±2 |

In Figure 8 and Figure 9 the results on the overall $HTC$ and thermal power output of the $HX$ are shown. The uncertainty on the overall $HTC$ calculation was estimated to be 5% maximum. The thermal power output was calculated based on the gas stream temperature difference, the temperature-corrected specific heat capacity of the gas and the mass flow rate. It was noticed that the pressure drop across the heat exchanger was under the instrument sensitivity in the case of plain tubes and it was assessed to be 20 Pa during the test with $TT$ at the maximum tested mass flow rate of producer-gas.
3.2. CFD results
The comparison between the numerical and the experimental outputs is shown in the figures below. In Figure 10 and Figure 11 a comparison of the gas outlet temperature between the two different turbulence model and the experimental data is shown. The best-fitting model was then selected to compare in Figure 12 and Figure 13 the overall HTC and the thermal power output of numerical and experimental campaigns.
Figure 12. Comparison between numerical and experimental results on the overall HTC calculation. Dashed and solid lines represent the linear regressions.

Figure 13. Comparison between numerical and experimental results on the thermal power lost from the gas stream. Dashed and solid lines represent the linear regressions.

Figure 14. Temperature contours of the gas continuum in the test at $7.12 \times 10^{-3}$ kg s$^{-1}$ with plain tubes.

The realizable $k$-$\varepsilon$ model gave better results in the cases with the twisted tape showing an error between 3-19% for the HTC and 0-3% for the thermal power output calculations if compared to the experimental results. In particular, the model tends to overestimate the overall HTC and to underestimate the thermal power output: this difference is given by the definition of overall HTC that derives from the calculation of the logarithmic mean temperature difference, while, the thermal power output is calculated from the simple gas temperature difference. For the case of plain tubes, the model that gave better results was the standard $k$-$\varepsilon$ low Re applications. In these cases, the error on the calculation of overall HTC was in the range 8-28% and 12-19% on the thermal power output if compared to the experimental results. In both cases the model overestimates the results. According to [30] the comparison between and experimental results seems to be consistent with the literature. As qualitative result from the numerical simulations, it was also noted that the central tube represent a preferential path for the gas stream (Figure 14). Moreover, the pressure drop of the heat exchanger was noticed to be negligible if compared to the total pressure drop of the gasification and filtering stage of the power plant (attested to 5-10 kPa).

4. Conclusions
In this work, twisted tape inserts were applied to a water-gas heat exchanger used to cool down the producer gas generated from a small-scale wood gasification power plant. Results shown an increase of the overall HTC between 57 and 76% and of the thermal power output from 12 to 19%. It was also noted that the central tube represents a preferential path for the gas stream. This non-uniformity is usually associated with a poor heat exchanger performance that can be improved by adding a diffusing conical baffle at the gas inlet. Further tests are needed in order to assess the long-term performances of the system and the influence of pipes fouling on heat transfer. The $TT$ applied to the $HX$ were not increasing the pressure drop of the device in a meaningful way. For this reason the $HX$, in this configuration, does not affect engine performance in terms of volumetric efficiency and, therefore, in terms of maximum electrical power output.

5. References

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