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Nanoporous membrane tube condensing heat transfer enhancement study

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ABSTRACT

A Transport Membrane Condenser (TMC), made from nanoporous membrane tube bundles, was developed to recover the water vapor and its significant amount of latent heat from boiler flue gases to improve boiler efficiency and save water. Experiments have been carried out to study the phenomena for both a nanoporous membrane tube bundle and an impermeable stainless steel tube bundle with the same characteristic dimensions. Flue gas streams with water vapor mass fraction 11.3%, temperature range from 65 °C to 95 °C are used for the experimental study, which covers the typical TMC waste heat recovery application parameter ranges. Results show convection Nusselt numbers of the membrane tube bundle are 50–80% higher than that of the impermeable stainless steel tube bundle at typical condensation heat transfer conditions. The parametric study has been done by varying cooling water inlet flow rate, water inlet temperature, flue gas inlet flow rate, inlet temperature, and inlet dew point. The condensing heat transfer enhancement effect gives a good perspective for using nanoporous membrane surface to design high efficiency condensing heat exchangers to recover both water vapor and its latent heat from high moisture content low grade waste heat streams.

1. Introduction

Energy efficiency of current gas-fired boiler is typically in the range of 75–85% and further improvement is mainly limited by stack loss. In gas-fired boiler application, there is about 18% in volume of water vapor in the flue gas, which usually exhausts with its substantial latent heat from the stack [1]. Fig. 1 shows the theoretical relationship of boiler efficiency based on natural gas higher heating value (HHV) versus exit flue gas temperature. The 85% boiler efficiency corresponds to an exit flue gas temperature of 160 °C. If the exit flue gas temperature were reduced to 50 °C from 160 °C, about 7% additional boiler efficiency could be gained. From 1970s, condensing devices have been developed for boiler applications to reduce stack loss. As a result, it is of great significance for energy conservation to study condensing device heat transfer [2].

In recent years, Gas Technology Institute (GTI) has developed a new technology based on a nanoporous ceramic separation membrane to extract a portion of the water vapor and its latent heat from flue gases and return the recovered water and heat to the boiler steam cycle [3,4]. Water vapor condenses inside the membrane pores and passes to the permeate side which is in direct contact with a low-temperature water stream. Contaminants of flue gases such as CO_2, O_2, NO_x, and SO_2 are inhibited from passing through the membrane by the membrane high selectivity [5–7]. The recovered water is of high quality, therefore can be used as supplemental makeup water for almost all industrial processes. Fig. 2 shows the photomicrograph of ceramic porous layer coated on the porous membrane tube surface. It consists of a top layer with a pore size of 60 to 80 Å (about 2 to 4 μm thick), a 500 Å pore size intermediate layer (typically 20 to 50 μm thick), and a 0.4 μm pore size substrate (about 1 mm thick). This structure is used for both polymeric and ceramic nanoporous separation membranes to achieve high separation ratio with minimal resistance to flux of the permeating species.

Single-phase convective heat transfer over a bare tube bundle has been investigated long time ago. Zukauskas [8,9] developed correlation equations for predicting mean convection Nusselt number between a horizontal tube bundle and a fluid flowing over the bundle exterior surface. The correlation equation is of the form

$$Nu_{conv} = CRe^m Pr^{0.36} (Pr/Pr_w)^{0.25} \tag{1}$$

where $Nu_{conv} = hd_o/\lambda$ is convection Nusselt number, $Re = ud_o/\nu$ is Reynolds number, and $Pr = \nu/a$ is Prandtl number. For the investigation of heat transfer over a tube bundle, Reynolds number is

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Nomenclature

A	bundle surface area (m^2)
A_c	cooling water flow cross section area (m^2)
B_l	longitudinal pitch of the tubes (m)
B_t	transverse pitch of the tubes (m)
C_p	specific heat (J/kg K)
d	tube diameter (m)
g_m	mass transfer coefficient (kg/m^2 s)
h	heat transfer coefficient (W/m^2 s)
L	tube length (m)
L_w	latent heat of condensation (kJ/kg)
MTD	mean temperature difference
M	mass flow rate (kg/s or kg/min)
n	total tube number of a bundle
Nu	Nusselt number hd/λ
P	pressure (Pa)
Pr	Prandtl number
Q	water volumetric flow rate (m^3/s)
R	heat resistance (m^2s/W)
Re	Reynolds number ud/ν
T	temperature (°C)
W	mass fraction
F	heat flux (W/m^2)
γ	porosity
λ	thermal conductivity (W/mK)
ν	kinematic viscosity (m^2/s)
ΔT_m	MTD, mean temperature difference (°C)

Subscripts

conv	convection
cond	condensation
i	inlet or inner
int	interface
FG	flue gas
o	outlet or outer
v	vapor
w	water

based on the tube outer diameter d_o as the characteristic length and the velocity u at the minimum flow area as the characteristic velocity.

Flow and heat transfer with condensation of the tube bundle in moistened gas mixture was also studied in the past, and recent studies include Che et al. [10,11], and Osakabe et al. [12,13]. Che et al. have investigated the convection–condensation heat and mass transfer through a 20-mm-outer-diamater in-line impermeable carbon steel tube bundle, within the range of flue gas Reynolds number of 3900–7300. Osakabe measured the condensation heat transfer on 10.5 mm and 4 mm outer diameter staggered stainless steel tube bundles in actual flue gas, and developed a prediction method to calculate the convection–condensation heat transfer Nusselt number. For impermeable tube bundles, experimental results show convection–condensation heat transfer coefficients are 1–3.5 times higher than those of the forced convection without condensation. And experiments also show tube bundles exterior surface are partially or totally wetted by condensate, and water film thickness and its heat resistance affect heat transfer and condensation.

To enhance the convection–condensation heat transfer between bundles and a fluid, some approaches by modifying the heat transfer surface are used, such as porous medium coated surface. Renken and Raich [14] have presented experimental results for force convection pure steam condensation heat transfer on an impermeable porous-layer coated surface. Results show the porous coating

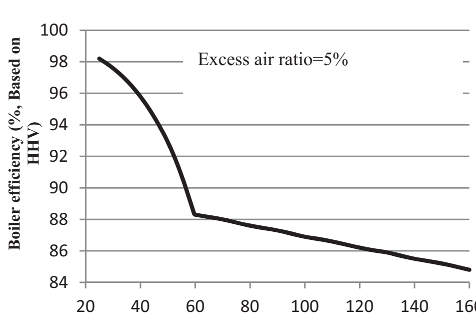

Fig. 1. Relationship of gas-fired boiler efficiency versus exit flue gas temperature.

produces a considerable heat transfer enhancement (250% higher) compared with noncoated surface. To our knowledge, heat transfer between permeable tube bundles and actual flue gases with condensation process has not been investigated.

In this paper, a permeable nanoporous membrane tube bundle was built to recover both water vapor and its significant amount of latent heat from flue gases. Experiments were conducted to compare the convection–condensation heat transfer performance of a porous membrane tube bundle with an impermeable stainless steel tube bundle. Based on the experimental data, convection Nusselt numbers in flue gas side and condensation rates for both bundles are compared. Parametric study was also carried out by varying cooling water flow rate, cooling water inlet temperature, flue gas inlet flow rate, flue gas inlet temperature, and flue gas inlet dew point.

2. Experimental investigation

2.1. Porous membrane tube bundle and impermeable stainless steel tube bundle

A nanoporous ceramic membrane tube bundle and an impermeable stainless steel tube bundle with the same characteristic dimensions were built to study their heat transfer and condensation phenomena. Both bundles have 78 tubes in 12 rows with a staggered arrangement. The longitudinal pitch B_1 is 13.6 mm, the transverse pitch B_t is 8.8 mm, and the tube length L is 432 mm, for both bundles. Fig. 3 shows the staggered arrangement in the tube bundle. For the porous membrane tubes, the average porosity was estimated at 30%. Dimensions for the porous membrane tube bundle and the impermeable stainless steel tube bundle are listed in Table 1. The impermeable tubes are made from type 304 stainless steel.

2.2. Test rig

Fig. 4 is the schematic of the experimental apparatus. As shown in Fig. 4, flue gas from a natural gas combustor flows into the duct, where it is cooled to the desired temperature by the flue gas cooling section. And then the flue gas flows upwards through the test section where test bundle installed. Cooling water flows into the