

EXPERIMENTAL CHARACTERISATION OF THE THERMAL PERFORMANCE OF A FINNED-TUBE HEAT EXCHANGER

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ABSTRACT

Finned-tube heat exchangers are frequently used in industry for gas-to-liquid heat transfer. For ORC power cycles, they are employed when heat is recovered from exhaust gas streams. The heat exchanger can either be used to directly evaporate the working fluid or to transfer the heat to an intermediate extraction loop.

While the literature on heat exchanger design is comprehensive, there is limited information on how the performance of an existing unit can be determined. Performance information is needed to investigate the overall system and to design heat exchangers of a similar type.

This paper presents how the thermal performance of an existing finned-tube heat exchanger is determined experimentally. In the experiment, heat is transferred from an exhaust gas stream to a thermal oil. The experimental setup is discussed including the thermal fluid selection process, the components, instrumentation and resultant measurement uncertainty. The heat transfer is calculated individually on either side of the heat exchanger.

The purpose of the article is to present a procedure for measuring the performance of a heat exchanger and determining the individual heat transfer coefficients for both sides given limited information from the experiment.

1. INTRODUCTION

This paper describes how the thermal performance of an existing finned-tube heat exchanger is determined experimentally. A procedure is described to calculate the overall heat transfer coefficient and individual convection coefficients. Once the measurements are conducted, the procedure will be applied to compare the results to existing correlations. The aim is to provide a foundation for the development of a heat exchanger model that can be used for sizing and optimization purposes.

In this experiment the assumptions of common heat exchanger analysis are not valid so that the use of existing thermal performance expressions would lead to incorrect results. While only a numerical model could consider all of those effects, the subject of this paper is to discuss what information can be gained with the present experimental setup. As the resultant improvement, it is shown how the heat transfer to ambience can be considered to a certain extent.

2. BACKGROUND

In heat exchanger theory, the thermal characteristics for a given flow arrangement are usually described by seven parameters (Shah and Sekulic, 2007):

$$T_{h,out}, T_{c,out} \text{ or } \dot{Q} = f(T_{h,in}, T_{c,in}, \dot{C}_h, \dot{C}_c, U, A, \text{flow conf.}) \quad (1)$$

h denotes the hot and *c* the cold side.

The same relationship can also be described by three non-dimensional parameters, for which several methods exist.

2.1 P-NTU-Method

A non-dimensional approach is the P-NTU method, where the heat exchanger is described by the variables *P*, *NTU* and *R*. The temperature effectiveness *P* is defined as

$$P_1 = \frac{T_{1,out} - T_{1,in}}{T_{2,in} - T_{1,in}} \quad (2)$$

for fluid 1 (hot or cold side) and

$$P_2 = \frac{T_{2,in} - T_{2,out}}{T_{2,in} - T_{1,in}} = P_1 R_1 \quad (3)$$

for fluid 2. The heat capacity ratio *R* is defined as

$$R_1 = \frac{\dot{C}_1}{\dot{C}_2} \quad (4)$$

$$R_2 = \frac{1}{R_1} \quad (5)$$

\dot{C}_1 is the heat capacity stream:

$$\dot{C}_1 = \dot{m}_1 c_{p,1} \quad (6)$$

And \dot{C}_2 is defined respectively. The number of transfer units, *NTU*₁ and *NTU*₂ are

$$NTU_1 = \frac{UA}{\dot{C}_1} \quad (7)$$

$$NTU_2 = \frac{UA}{\dot{C}_2} = NTU_1 R_1 \quad (8)$$

The equations show that the thermal characteristics for a given flow arrangement can be described if any three of those six variables are known.

The mean temperature difference between the hot and the cold fluid is defined in relation to the logarithmic-mean temperature difference (LMTD) for a counterflow heat exchanger:

$$\Delta T_m = F \Delta T_{lm} \quad (9)$$

with

$$\Delta T_{lm} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln\left(\frac{T_{h,in} - T_{c,out}}{T_{h,out} - T_{c,in}}\right)} \quad (10)$$

and

$$F = f(R_1, P_1, \text{flow config.}) \quad (11)$$

The present heat exchanger can be considered as a mixed-unmixed crossflow unit. The expression for F is

$$F = \frac{\ln(1 - R_1 P_1) / (1 - P_1)}{\left(1 - \frac{1}{R_1}\right) \ln[1 + r_1 \ln(1 - P_1)]} \quad (12)$$

where fluid 1 is mixed (exhaust gas in this experiment) and fluid 2 unmixed (thermal oil).

2.1 Heat transfer in heat exchangers

The heat transfer in a heat exchanger can be described as

$$\dot{Q} = U_m A \cdot \Delta T_m \quad (13)$$

U_m is the overall heat transfer coefficient:

$$\frac{1}{U_m A} = \frac{1}{h_1 A_1} + \frac{t}{k_w A_w} + \frac{1}{h_2 A_2} \quad (14)$$

h is the convection coefficient of the respective surface and k_w the thermal conductivity of the wall.

3. EXPERIMENTAL SETUP

The setup is shown in Figure 1: The thermal fluid enters the heat exchanger in the lower header drum. The inlet and outlet temperatures of both streams are measured. The inlet temperature of the thermal fluid ($T_{tf,in}$) can be controlled by the mass flow rate of the cooling water. To account for density changes, an expansion tank is installed.

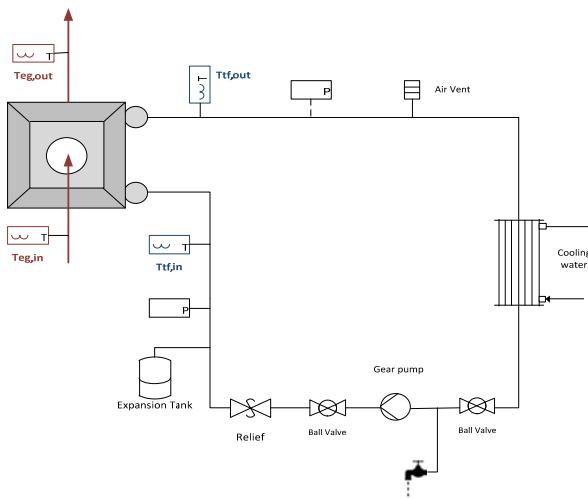


Figure 1: Layout of the experimental setup

3.1 Selection of thermal fluid

The design and sizing of the cycle components is based on the physical properties of the thermal fluid, so it was chosen in at an early stage of the experimental design. Pressurized water, glycol and thermal oils have been considered as potential thermal fluids. They were evaluated based on the following physical and practical criteria:

- Fluid properties: A low viscosity and high density, heat capacity and conduction coefficient are desired
- Vapor pressure: A low pressure, liquid-only system is desired because it is inexpensive and easy to handle. This means that the vapor pressure must not exceed ambient pressure for the maximum system temperature. The maximum system temperature is the film temperature that is reached at the tube walls. It is assumed to be the maximum exhaust gas temperature of $T=280^\circ\text{C}$.
- Thermal degradation over time
- Handling and safety requirements
- Costs and ease of supply in New Zealand

After evaluating 10 different fluids, the thermal oil "Purity FG" was chosen for its physical properties (Petro-Canada, 2012) and because it is available from New Zealand.

3.2 Components and instrumentation

3.2.1 Heat exchanger

The heat exchanger to be investigated is a finned-tube one-pass crossflow unit. It transfers heat from a hot exhaust gas stream, flowing over the finned-tube surfaces, to a thermal oil inside. The thermal fluid is distributed into 6 parallel tubes in staggered alignment. The design is shown in Figure 2.

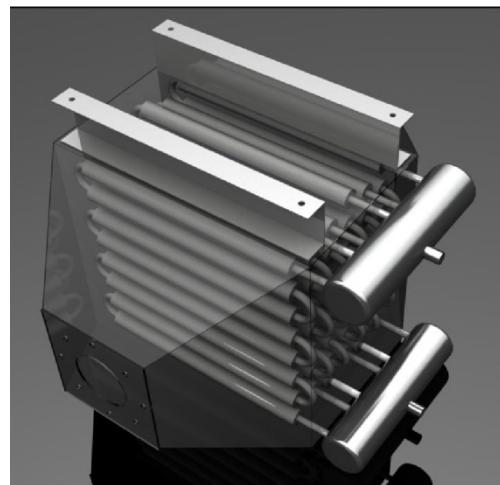


Figure 2: Model of the finned-tube heat exchanger to be investigated (Meyer et al. 2012)

3.2.2 Exhaust gas generator

For exhaust gas generation, a micro gas turbine (Capstone C30) is used. The parameter that can be controlled is the electrical power production. The exhaust gas mass flow rate and temperature is coupled and primarily a function of the electrical power generation, but also depends on the ambient

conditions. For ISO Standard conditions ($T=15^\circ\text{C}$, $p=1.013\text{bar}$) they are given in the turbine specifications (Capstone, 2006).

3.2.3 Pump and flow rate measurement

An external hydraulic gear pump with a displacement volume of 11cm^3 is used. A gear pump was preferred over other options such as centrifugal, piston or diaphragm types for the following reasons:

- Gear pumps work well for the thermal oil viscosity range at typical operating temperatures
- They deliver a continuous flow rate with low pulsation
- They deliver approximately the same volume per revolution, independent of pressure and fluid property changes.

The flow rate of the thermal fluid can be determined from the pump speed as it is of positive displacement type. In theory, the flow rate depends only on the displacement volume and the frequency of rotation:

$$\dot{V}_{th} = 11\text{cm}^3 \cdot n$$

In reality, the flow rate is reduced due to back flow:

$$\dot{V}_e = \dot{V}_{th} - \dot{V}_s$$

For small Δp , the leakage rate \dot{V}_s is a linear function of Δp and the dynamic viscosity μ :

$$\dot{V}_s \sim \frac{\Delta p}{\mu(T)}$$

For the pump used in this experiment, the flow rate \dot{V}_e is given for $\Delta p = 2\text{ bar}$ and $\mu = 35\text{ cSt}$:

In the experiment, the viscosity is expected to be higher while the pressure drop is considerably lower than 2 bar. It was therefore concluded that the flow rate is best estimated by calculating the average between the specifications and the theoretical maximum and considering them as the error limit:

$$\dot{V}_e = \frac{\dot{V}_{th} + \dot{V}_{e,spec}}{2} \pm \frac{\dot{V}_{th} - \dot{V}_{e,spec}}{2}$$

For the exhaust gas side, the specifications for ISO conditions provided by the manufacturer (Capstone, 2006) are used as they come very close to the actual working conditions.

3.2.3 Temperature measurement

The inlet and outlet temperatures of both the oil and gas side were measured by thermocouples. For the exhaust gas stream, type K thermocouples were used, while type T was preferred for the thermal fluid for their accuracy.

The thermal fluid thermocouples were placed so that they point directly into the outlet and inlet header drums, ensuring a sufficient immersion depth. Thereby a potential error due to conduction along the probe sheath can be neglected. The resultant error limit is composed of the thermocouple tolerance, the cold junction compensation error and gain and offset errors of the data acquisition device:

$$e_{TC} = e_{TC,Tol} + e_{CJC} + e_{gain} + e_{offset}$$

The error limits were calculated based on the specifications for the data acquisition device (National Instruments, 2009). For an inlet and outlet temperature of 20°C and 100°C respectively the error limit is

$$e_{TC,tf} = (0.5 + 0.8 + 0.1 + 0.1)\text{K} = 1.5\text{K}$$

For the thermal fluid side and

$$e_{TC,eg} = (1.5 + 0.8 + 0.2 + 0.1)\text{K} = 2.6\text{K}$$

for the exhaust gas side.

4. THERMAL CHARACTERIZATION

The thermal characteristics of a heat exchanger are determined by its flow configuration and the overall heat transfer coefficient UA . The flow configuration in the heat exchanger can be considered as a one-pass crossflow, with one mixed fluid (exhaust gas) and one unmixed stream (thermal fluid).

4.1 Overall heat transfer coefficient

As stated in section 2, the thermal characteristics of a heat exchanger can be described by either seven dimensional or three non-dimensional parameters. If the P-NTU method is used, the overall heat transfer coefficient can be determined as a function of P and R.

This implies that the functional relationship between P, NTU and R must be known to calculate UA from the experimental data. Although an expression for the present flow configuration exists, it relies on the validity of basic assumptions (Shah and Sekulic, 2007). This means that certain effects are neglected, although they are relevant in this experiment:

1. Heat is dissipated into ambience
2. Longitudinal heat conduction over the tube walls
3. Overall heat transfer coefficient is constant (and its individual components)
4. Fluid properties are not constant (especially viscosity is changing significantly)

All four effects can only be considered by formulating a differential energy balance that includes parasitic heat flow, longitudinal heat conduction and variable convection coefficients and properties. For variables such as convection coefficients and fluid properties, correlations from literature would have to be used. The use of correlations is necessary, because the temperature (and pressure) profile inside the heat exchanger is not measured and the local heat transfer can only be estimated. After solving the energy balance equation by numerical integration, the model calculations can be compared to the results of the experiment (model validation). This approach is based on modelling rather than experimentation so it is not further discussed here.

Instead only the information gained from the experiment are used to calculate the overall heat transfer coefficient U_{mA} . This implies that only the first effect, energy loss to the environment \dot{Q}_{amb} , can be taken into account, as the others rely on an infinitesimal representation.

\dot{Q}_{amb} is calculated by a energy balance over the hot side (exhaust gas) heat exchanger:

$$\dot{Q}_{amb} = \dot{C}_{eg}(T_{eg,in} - T_{eg,out}) - U_m A \cdot \Delta T_m \quad (15)$$

$U_m A$ can be determined from the cold side (thermal fluid) energy balance:

$$U_m A = \frac{\dot{C}_{tf}(T_{tf,out} - T_{tf,in})}{\Delta T_m} \quad (16)$$

ΔT_m is the mean temperature difference. It can be calculated by multiplying the log-mean temperature difference by the correction factor F . For the present mixed-unmixed cross flow heat exchanger, eq. (12) can be used.

If h would be calculated by directly using the measured mass flow rates and temperatures, the resultant overall heat transfer coefficient would be too low because the heat loss to ambience is not considered in the equation. Therefore, the following correction for the mass flow rate is used:

$$\dot{C}_{eg,corr} = \frac{\dot{C}_{tf}(T_{tf,out} - T_{tf,in})}{\dot{C}_{eg}(T_{eg,in} - T_{eg,out})} \dot{C}_{eg} \quad (17)$$

This correction is thought to give a more accurate estimation for the thermal performance, although the general limitations of heat exchanger analysis assumptions stated above still apply.

4.2 Individual convection coefficients

To determine the convection coefficients for each side individually, it is ideal to vary the mass flow rate on one side while both mass flow rate and LMTD on the other side are kept constant. This means that the known side's thermal resistance remains constant and the unknown side's one varies according to the changing mass flow rate and temperatures. The overall thermal resistance consists of two constant and one variable part:

$$R(Re, Pr) = \frac{1}{U_m A} = R_k + R_{wall} + R_u(Re, Pr) \quad (18)$$

with R_u as the variable, unknown side's thermal resistance

$$R_u(Re, Pr) = \frac{1}{\eta_0 \cdot h(Re, Pr) \cdot A} \quad (19)$$

The equation shows that changes in the overall thermal resistance for different mass flow rates \dot{m}_u can be attributed to a change in the Reynolds and Prandtl number which in turn leads to a variation of the convection coefficient $h(Re, Pr)$. Using eq. (18) and (19), h can be expressed as

$$h(Re, Pr) = \eta_0 A \left[\frac{1}{U_m A} - (R_k + R_{wall}) \right] \quad (20)$$

The entire procedure to determine the individual convection coefficients can be summed up as follows:

1. Calculate $U_m A$ for different mass flow rates from the experimental results using eq. (16)
2. Plot $\frac{1}{U_m A}$ over the mean Reynolds number
3. Determine the constant thermal resistance $R_k + R_{wall}$ by extrapolating the results to $Re = 0$. This idea is loosely based to the Wilson technique (Wilson, 1915).
4. Determine $h(Re, Pr)$ from eq. (20)

In the present experimental setup, only the mass flow rate and LMTD of the thermal fluid can be controlled independently so that only the exhaust gas side convection coefficient h_{eg} can be determined according to the method described above. However, if the procedure is repeated for different thermal fluid mass flow rates, the convection coefficient h_{tf} can be estimated from the constant proportion of the thermal resistance. For each thermal fluid mass flow rate, $R_k + R_{wall}$ is calculated according to step 3 and R_k is determined by subtracting

$$R_{wall} = \frac{t}{A \cdot k} \quad (21)$$

t is the tube thickness and k the thermal conductivity of the material. The results can be compared to existing correlations.

5. CONCLUSION

A setup for heat exchanger experimentation was described, including the selection of a thermal fluid, components, instrumentation and error estimation.

It was discussed why the common assumptions for heat exchanger analysis would lead to incorrect results in this experiment. Because the setup only yields limited information, modeling techniques would have to be used to consider all relevant effects. In an attempt to characterize the thermal performance based the experimental results only, a procedure was developed to determine the overall heat transfer coefficient and individual convection coefficients, in which the heat transfer to ambience is considered to a certain extent.

Once the measurements are conducted, the procedure will be applied to the present setup. The results will be compared to existing correlations to provide the foundation for a detailed model that can be used for sizing and optimization purposes.

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