Thermodynamic Modelling and Testing of Marine Growth Mitigations

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Abstract

Passive subsea coolers are used in subsea oil and gas processing as they create a simple means of cooling the process gas with minimal moving parts. The cooling of the gas at the seabed can provide numerous benefits, including lower pipeline costs and sometimes more straightforward operation. Unlike conventional coolers there is no active control on the output temperature of the process gas therefore marine growth and changing seabed currents are all capable of affecting the heat transfer. One part of this project looks at various marine growth mitigation strategies and comes to the conclusion that anti-biofouling coatings could be effective. However these coatings create an extra layer of resistance to heat transfer so their thermal conductivities are examined. The second part of this project creates a computational fluid dynamics model which predicts the output temperature of a gas flowing through a horizontal cylinder with a periodic cross flow of water. This model is designed to simulate a passive heat exchanger being exposed to periodic seabed currents and is currently being verified by experimental results obtained using the mini O-tube facility at the University of Western Australia.

1. Introduction

The recovery of reserves from remote subsea reservoirs is a current challenge for the oil and gas industry. One solution is to process the fluids, which includes pressure boosting the multiphase fluids and may include compression and separation of liquids from gases, at the seabed. The use of a heat exchanger allows lower material requirements, reduced compressor duty and increased water removal. One of the preferred methods of cooling the well fluid is to use a passive heat exchanger whose simplified design consists of a bundle of pipes being cooled by the surrounding seawater. Previous work by Kimberly Chieng highlighted the effects marine growth can have on passive coolers by creating an extra layer of heat transfer resistance and suggested that changing seabed currents could affect performance (Chieng, 2013). Understanding how these issues affect performance is important, as if the fluid is cooled too much it could enter the hydrate and wax formation regions, whereas if it is insufficiently cooled higher specification pipeline materials are required.

1.1 Thermal Conductivity of Anti-Biofouling Coatings

The growth of organisms on submerged structures is termed biofouling and is an issue in many areas including shipping, power generation and offshore structures. In shipping, coatings are applied to hulls which can be segregated into two commercially available categories; foul-release and anti-foul (AF). Foul release coatings have a low surface energy and are generally silicone or fluoropolymer based, making it difficult for fouling organisms to adhere to the coating. The benefits of this mechanism include its small environmental footprint due to no toxic leaching and the ease of cleaning, whereas drawbacks include lower effectiveness in still water conditions (Chambers et al., 2006).

AF paints have a long history and leach toxins into the sea which kills sea organisms. Following the worldwide ban of tributyltin based paints in 2008 due to environmental concerns, the active biocide in AF paints is generally copper in combination with another "booster" biocide (Yebra et al., 2004). These biocides continue to come under regulatory scrutiny with limits placed on copper leaching rates in several countries and the use of several booster biocides, such as Irgarol, being restricted (Wendt et al., 2013). These paints require a thicker coating to last longer periods, creating more resistance to heat transfer in the passive cooler, and will leech faster due to the raised temperature of the cooler surface.

Both of these coating systems create an extra layer of resistance to heat transfer and their thermal conductivities must be found if they are to be used on a passive cooler. The standard way of measuring the thermal conductivity of a solid is to place it between two temperature sensor plates with heat flow through the system and measure the temperature drop through the solid (Slifka, 2000). This method allows the thermal conductivity to be determined at various temperatures very accurately but was unavailable for this project so another method was used which is outlined in section 2.1. A coating manufacturer has made one of their AF and foul release coating systems available for testing.

1.2 Fluctuating Seabed Currents

As the seawater current velocity changes over a passive cooler, the heat transfer will change as well. This is due to changes in the seawater convection coefficient which is a function of the seawater current velocity. When the water velocity is high, forced convection will dominate whereas at low velocities natural convection will dominate. These changes in the convection coefficient will cause changes in the outlet temperature of the cooler, so the magnitude of these changes was modelled.

Water currents are not constrained to one direction of flow in the open ocean, and designs for passive heat exchangers are varied including orientations with bundles of vertical or horizontal pipes. By modelling the problem as a horizontal pipe with the current velocity perpendicular to the pipe it was possible to test the accuracy of the model using the mini O-tube facility at the University of Western Australia. The equations used in the model to estimate convection coefficients are functions of dimensionless numbers so if the model is accurate under testing in the mini O-tube it will be accurate when scaled up to actual flow conditions in a passive subsea cooler.

2. Process

This project involved three distinct tasks which included the thermal conductivity experiments, transient passive cooler model and testing of the model.

2.1 Thermal Conductivity Experiments

With the standard technique for finding the thermal conductivity of a solid outlined in section 1.1 unavailable, a different method was formulated. A ¼ inch steel tube was placed in a water bath of constant temperature. Methane gas of 40°C flowed through the tube and the entrance and exit temperatures of the gas was recorded once the system was at steady state using flow rates varying from 100 ml/min to 10 ml/min. The tubes diameter was then measured several times along its length with a micrometer and then coated with one of the coating systems. The experiment was then repeated and by assuming that the convection coefficient of the gas and water bath were constant for the coated and uncoated pipe, the thermal conductivity of the coating system was inferred. The experimental rig is shown below in figure 1.

The coating systems include an anticorrosive undercoat and a tie coat to help with adhesion to the substrate. They will always be present with the top coat, so the thermal conductivity of the system as a whole was found, not each layer individually. Each layer of coating in normal use is to be applied at a thickness of ~125μm and to reduce error in the experiment the coatings were applied at a larger thickness. By looking at the critical film thickness for various materials, micro effects will not occur in the recommended coating thicknesses, as although the coatings are thin, they are still an order of magnitude away from the critical film thickness (Flik et al., 1992).

This method is less accurate than the standard method used by Slifka, but makes adjustments to minimise the error in the experiment (Slifka, 2000). The temperature of the coating falls along the length of the tube and thermal conductivities change with temperature, however decreasing the length of the pipe to minimise this error increases the errors in the temperature readings and the uncertainty in the coating thickness measurement. Error could have been reduced further by having the thermocouples closer to where the tube enters and exits the water bath, however this error was partially mitigated by using extensive insulation.

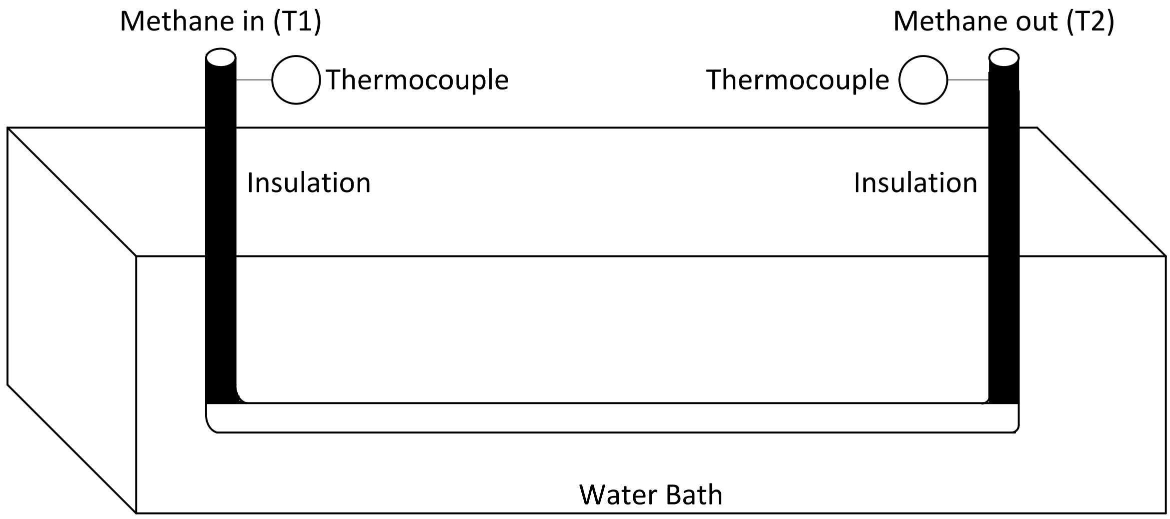

Figure 1 Thermal conductivity experimental rig

2.2 Transient Passive Subsea Cooler Model

The convection coefficient of the water is regarded as variable in this transient model. This is calculated via equation 1.

$$u = \frac{A}{2} sin\left(\frac{2\pi}{B} t\right) + \frac{A}{2} + C \tag{1}$$

Where u is the sea water velocity, A maximum seawater velocity, B is the period of the seawater current and t is time. C can be used to represent a minimum current velocity or for a constant current velocity by setting A to zero. There are four cases for calculating the convection coefficient of the sea water which include forced convection dominating, forced convection and slight natural convection, forced and natural convection of equal magnitude and natural convection dominating (Fand and Keswani, 1973). These regions are governed by two dimensionless numbers, the Reynolds and the Grashof. In the region where forced and natural convection are of the same magnitude, there are no accurate correlations for convection coefficients in a horizontal cylinder as the convection coefficient oscillates with random period between two extreme values (Fand and Keswani, 1973). In this region the correlations for natural and forced convection are simply added together as an approximation.

The model uses finite differences to solve the circular cylindrical heat diffusion equation shown below in equation 2 to solve the future wall temperatures and then solves the future gas temperatures in the pipe by calculating the heat flow from the gas to the inner cylinder wall via equation 3. To make equation 3 valid, the time step must be a function of the mean gas velocity, u, and the spatial step, z, shown in equation 4. These calculations use physical data from Mulitflash and are calculated in Microsoft Excel using Visual Basic for Applications.

$$\rho c_p \frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(rk\frac{\partial T}{\partial r}\right) \tag{2}$$

$$T_{g,z,t+1} = T_{g,z-1,t} - \frac{2(\Delta z)\pi r_1 h_{g,t}\left(T_{g,z,t} - T_{1,z,t}\right)}{\dot{m}C_p}, \qquad T_{g,0,t} = user\ defined \tag{3}$$

$$\Delta t = \frac{\Delta z}{u} \tag{4}$$

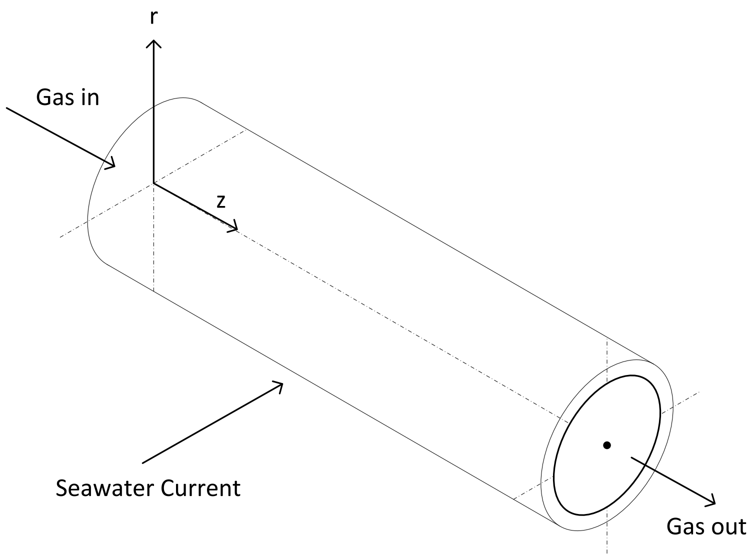

Figure 2 Isometric view of transient model

2.3 Testing Transient Passive Subsea Cooler Model

The testing of the model is being conducted in the mini O-tube facility at the University of Western Australia. This facility allows oscillatory water current over a horizontal tube which has nitrogen gas flowing through it. From comparing the change in temperature of the gas entering and exiting the system to what is predicted from the model, the accuracy of the model can be ascertained. Figure 3 shows a diagram of the mini O-tube.

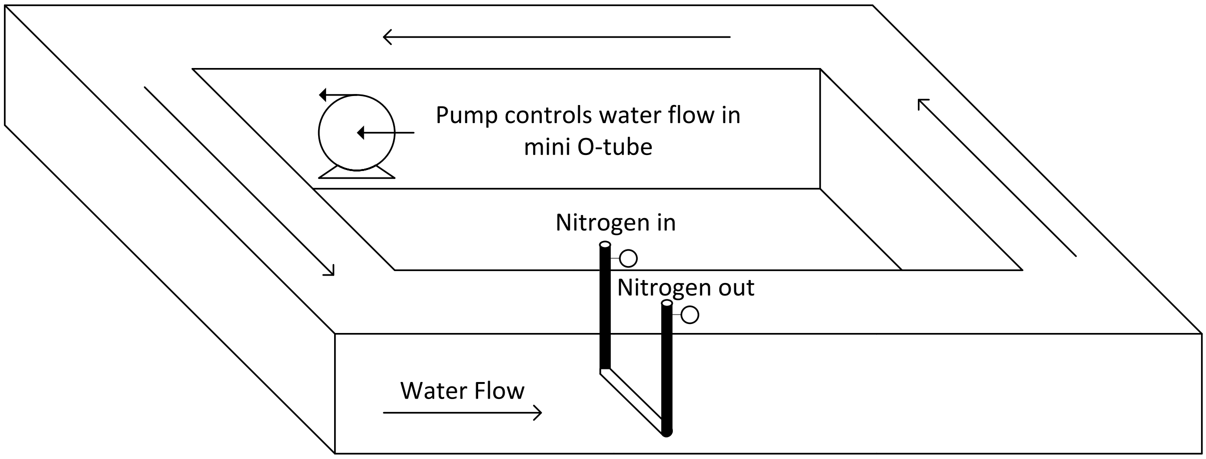

Figure 3 Experimental rig for testing transient model

3. Results and Discussion

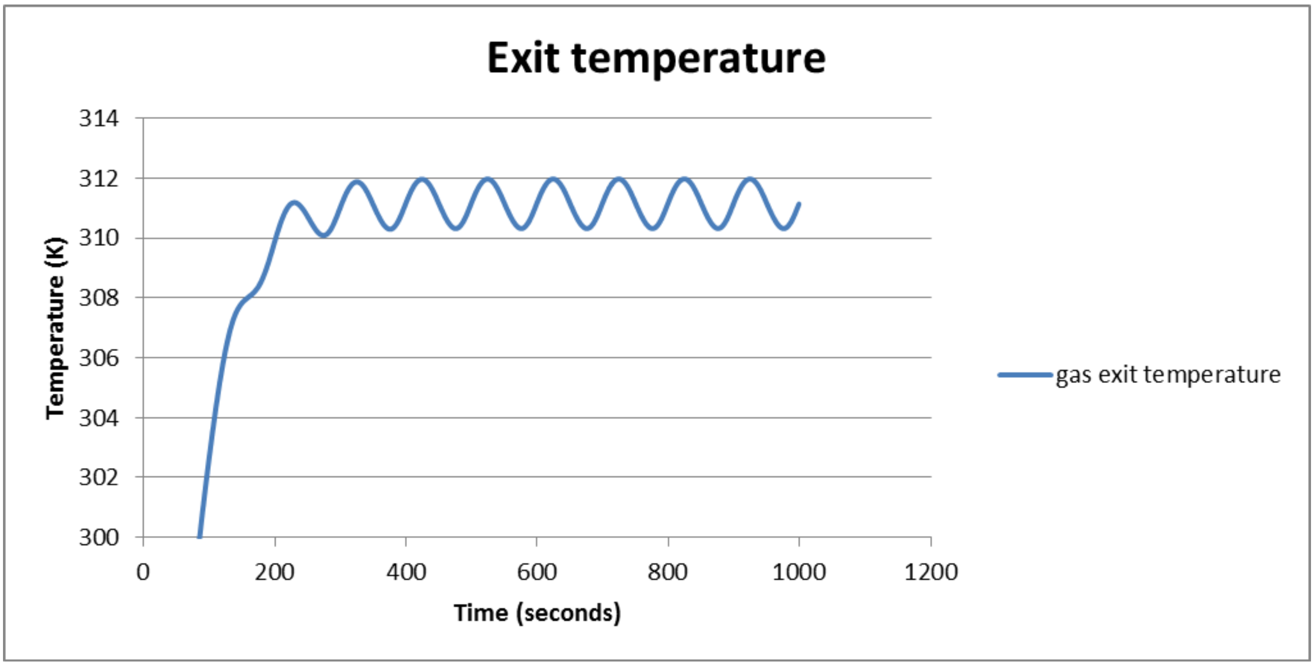

Figure 4 An example output from the transient model

Figure 4 shows the transient response to a passive cooler designed by Kimberly Chieng with an oscillatory water current of period 100 seconds, maximum current velocity of 1.5 m/s, gas residence time of 50 seconds, and 100% methane feed (Chieng, 2013). The initial lower gas exit temperature is caused by the initial condition that the pipe wall is the same temperature as the surrounding seawater at the model start time. Other outputs given by the model include the inner and outer wall temperatures at the pipe exit, the water convection coefficient, the water convection region and the water velocity all against time.

4. Conclusions and Future Work

Preliminary results from the transient model indicate that the magnitude of oscillations of the gas exit temperature are maximised when the mean residence time of the gas is much smaller than the water current period. Further work will investigate the differences in the output of a passive cooler when a flow is at a constant maximum value, when there is no water current and when there is an oscillatory current.

The transient model has also given the benefit of predicting the outer pipe wall temperature along the pipe length which could be used to predict which parts of the pipe will be more susceptible to marine growth. The thermal conductivity experiments are currently ongoing and the mini O-tube experiment will begin in late September.

5. Acknowledgements

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6. References

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