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Changing the World's Energy Future

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INTRODUCTION

Microreactors are currently of great interest in the nuclear industry due to their distinct advantages, such as their relatively small capital cost, ability to be fully assembled at a factory, transported by truck, train, ship, or plane, and operate semiautonomously. One category of these microreactor designs utilizes heat pipes as the primary mechanism for transferring heat from the core to the secondary side. Heat pipes are attractive passive heat transfer devices featuring no moving parts and an extremely high thermal efficiency.

Due to the interest in heat-pipe-cooled nuclear microreactors, the engineering-scale heat pipe application Sockeye is under active development [1], and its heat pipe models require validation. To provide high-resolution data for heat pipe performance analysis and code validation, the Michigan Single Sodium Heat Pipe (MISOH1) facility was constructed at the Experimental and Computational Multiphase Flow (ECMF) Laboratory at the University of Michigan [2, 3, 4]. Numerous tests have been conducted at this facility, featuring the heat pipe in various gravitational orientations and different heating and cooling configurations. In this work, TEST-018 is modeled with Sockeye and compared to experimental measurements as a means of code validation.

EXPERIMENT DESCRIPTION

Figure 1 shows the schematic diagram and a photograph of the MISOH1 facility. The MISOH1 facility is composed of a single sodium heat pipe, heated by a silicon carbide heater and cooled by a multi-channel heat exchanger. To minimize heat loss, the heater and adiabatic length are covered in mineral wool insulation. The multi-channel heat exchanger uses air in the inner channel and water in the outer channel. The inner channel features two passes, while the outer channel features only one. The gap between the heat pipe surface and the inner surface of the heat exchanger was filled with copper wire to enhance the heat transfer capacity. The mass flow rate of each coolant was measured using a Coriolis flow meter. Thermocouples were bound using stainless-steel wires on the heat pipe surface, as well as the insulation surface. Two additional thermocouples were attached to the heater's surface. The inlet and outlet temperatures of both coolants were also measured to estimate the heat pipe heat transfer rate. All the data was recorded through a National Instruments (NI) module on the LabVIEW interface.

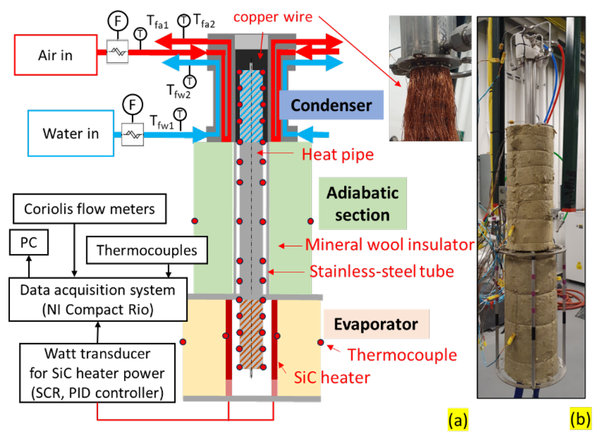

Fig. 1. MISOH1 test facility (a) schematic diagram; (b) photograph.

MODEL DESCRIPTION

System Model

TEST-018 featured the heat pipe in a horizontal orientation and started immediately following the end of TEST-017, thus starting the heat pipe in a hot state. The event table for TEST-018 is given in Table I. Note that the last time does not feature any event other than the end of the experiment.

The model in this effort is a two-dimensional (2D), transient heat conduction model featuring various subdomains coupled with various interface conditions. Figure 2 shows the model domain approximation. Axially, the domain is divided into three sections: the evaporator section, the adiabatic section, and the condenser section, respectively, from left to right. The evaporator section features a heater region, separated from the heat pipe by a gap and covered by insulation. The unheated length of the heater is excluded from the model. The adiabatic section features a steel support tube offset from the heat pipe by a gap and wrapped in insulation. Finally, the condenser section features a filler region in direct contact with the heat pipe; on the filler's outer surface, it comes in direct contact with the heat exchanger, which is not modeled explicitly. The heat pipe itself is divided into three radial regions: the vapor core, the wick plus liquid gap, and the heat pipe wall. The heater and heat pipe support structures are neglected from the model.

TABLE I. TEST-018 event timing for changes to heater power and mass flow rates of the air and water in the heat exchanger. A "..." denotes a continuation of the previous value.

Time [s]	Power [W]	Air [g/s]	Water [g/s]
0	0	3	50
130	...	...	32
181	500	...	...
320	750	...	...
900	1,000	...	...
1,266	1,200	...	...
1,623	...	5.9	...
1,772	1,400	...	...
1,986	...	7.4	...
2,132	1,300	...	...
2,423	...	...	...

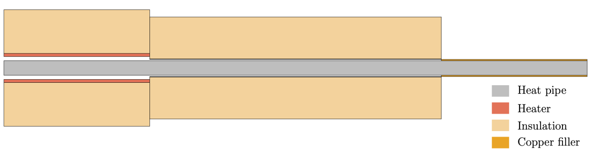

Fig. 2. Model domain approximation, drawn to scale.

The silicon carbide heater has a spiral shape, but the model approximates it as a solid, cylindrical tube; this approximation was designed to offset the underestimation of heat capacity that was incurred by neglecting the unheated length. The heating process is modeled with a uniform, volumetric heat source in this region, matching the instantaneous, measured heater power from the experiment.

To avoid increasing the model complexity, the heat exchanger is not explicitly modeled; instead, the total measured heat removal rate is applied uniformly along the condenser length, on the outer surface of the filler region:

$$q_{\text{HX}}(t) = -\frac{Q_{\text{air}}(t) + Q_{\text{water}}(t)}{\pi D_{\text{filler}} L_{\text{cond}}} , \quad (1)$$

where D_{filler} is the filler region outer diameter, L_{cond} is the condenser length, and $Q_{\text{air}}(t)$ and $Q_{\text{water}}(t)$ are the heat removal rates of the air and water in the heat exchanger, respectively:

$$Q_{\text{air}}(t) = \dot{m}_{\text{air}}(t) c_{p,\text{air}} (T_{\text{air,out}}(t) - T_{\text{air,in}}(t)) , \quad (2)$$

$$Q_{\text{water}}(t) = \dot{m}_{\text{water}}(t) c_{p,\text{water}} (T_{\text{water,out}}(t) - T_{\text{water,in}}(t)) , \quad (3)$$

where $\dot{m}_i(t)$ denotes a measured mass flow rate, $c_{p,i}$ denotes specific heat capacity, and $T_i(t)$ denotes a measured temperature at the inlet or outlet of a heat exchanger channel.

Boundary conditions on the outer radial surface of the evaporator and adiabatic insulation are taken to be a combination of convection and radiation:

$$q_\infty(\mathbf{x}, t) = h_\infty (T_\infty - T(\mathbf{x}, t)) + \sigma \epsilon_{\text{ins}} \left(T_\infty^4 - T(\mathbf{x}, t)^4 \right) , \quad (4)$$

where h_∞ is the environment heat transfer coefficient due to natural circulation, T_∞ is the environment temperature, σ is the Stefan-Boltzmann constant, and ϵ_{ins} is the emissivity of the insulation surface.

The heater and its insulation are assumed to have perfect thermal contact, as well as the adiabatic section support tube and its insulation. The gaps between the heater and heat pipe and between the adiabatic section support tube and heat pipe are treated with convection plus radiation; on the heat pipe wall side of the gap, the heat flux is computed as:

$$q(\mathbf{x}, t) = h_{\text{gap}} (T_j(\mathbf{x}, t) - T(\mathbf{x}, t)) + \frac{\sigma}{\mathcal{R}_{\text{hp},j}} \left(T_j(\mathbf{x}, t)^4 - T(\mathbf{x}, t)^4 \right) , \quad (5)$$

$$\mathcal{R}_{\text{hp},j} = \frac{1 - \epsilon_{\text{hp}}}{\epsilon_{\text{hp}}} + \frac{1 - \epsilon_j}{\epsilon_j} \frac{D_{\text{hp}}}{D_j} , \quad (6)$$

where h_{gap} is the gap heat transfer coefficient, T_j is the axially adjacent temperature on the surface across the gap, ϵ_{hp} is the heat pipe surface emissivity, ϵ_j is the other surface emissivity, D_{hp} is the outer heat pipe diameter, and D_j is the diameter of the adjacent surface.

The axial boundaries of the system are assumed to be perfectly insulated, and the domain layers external to the heat pipe are assumed to be insulated at their internal axial boundaries. For example, it is approximated that no heat is transferred directly from the heater to the adiabatic section insulation.

Since TEST-018 was run immediately after TEST-017, the initial conditions were taken to be the steady solution at a heater power of 1,000 W, the final power in TEST-017.

Heat Pipe Model

The heat pipe is modeled with Sockeye's "Conduction Model," which approximates the heat pipe with 2D heat conduction and a dynamic definition for the thermal conductivity of the vapor core region to mimic heat pipe performance [1, 5]. In this model, the thermal conductivity of the vapor core is controlled to mimic heat pipe performance according to analytic heat pipe limits; the conductivity value adjusts throughout the transient to attempt to have the heat throughput match the limit at the current operating temperature.

RESULTS

MISOH1 was instrumented with thermocouples at numerous axial positions along the wall of the heat pipe, and each axial position featured one or more positions around the pipe circumference. Additionally, thermocouples were placed on the heater surface, on the insulation surfaces in the evaporator and adiabatic sections, and at the inlets and outlets of the heat exchanger channels.

Figure 3 shows the comparison of the model solution to the experimental data at the near-steady state at the end of TEST-018. The numerical solution matches the data well, but the shape of the temperature distribution along the heat pipe notably differs. In particular, the experimental data shows a temperature difference between the evaporator and adiabatic section of roughly 50 K, despite the theoretical near-isothermal temperature profile in the vapor. This discrepancy is believed to be a measurement error due to the thermal resistance of the thermocouples themselves; those thermocouples in the adiabatic section are more trusted since there should be a much smaller rate of heat going out the boundary of the adiabatic section than that going into the evaporator section.

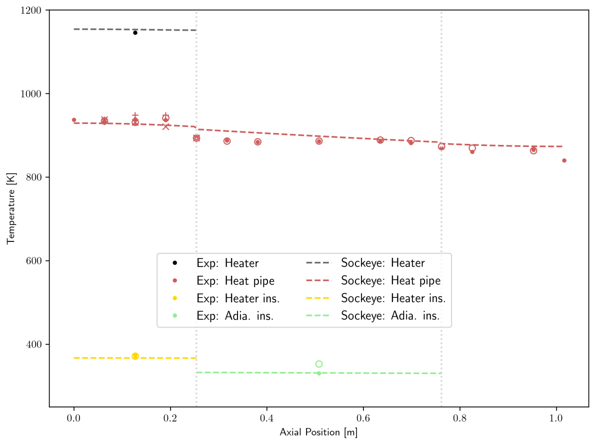

Fig. 3. Steady temperature solution compared to experimental data. The vertical, dotted lines denote the boundaries between the evaporator, adiabatic, and condenser sections. Different symbols of the same color denote different azimuthal positions.

Figure 4 shows the comparison of the model solution to the experimental data throughout the TEST-018 transient. In the very beginning of the transient, there is a short transient in the heat pipe temperatures due to the assembly being reoriented from the vertical orientation used in TEST-017. There is a significant discrepancy in the initial heater temperature, as well as its rate of decay following the power shutoff at the beginning of the transient. After this, the heater temperature matches the measured values very well. The heat pipe temperatures follow the trend shown in the measured data and stay within their bounds, but they lack some features. The measured data indicates a significant temperature difference across the heat pipe until roughly t = 800 seconds. This indicates the sonic limit being active, which is plausible given the initial dropping of temperatures and sudden return of power. The sonic limit remains active until the heat pipe temperature sufficiently increases. The Sockeye model also triggers the sonic limit, but not to the extent shown in the measured data; the temperature drop is not nearly as severe, and it recovers more quickly as well. This discrepancy is due to two factors: the model approximation of the conduction heat pipe model, which treats the heat pipe dynamics using heat conduction instead of advection, and the lack of knowledge of specific heat pipe internals, since these were proprietary information of the manufacturer. The sonic limit is a function of the working fluid equation of state and the cross-sectional area for the vapor in the heat pipe, so if the internal dimensions are unknown, Sockeye's prediction of the sonic limit may have significant errors.

CONCLUSIONS

A 2D heat conduction model of TEST-018 at MISOH1 was developed using Sockeye's conduction heat pipe model

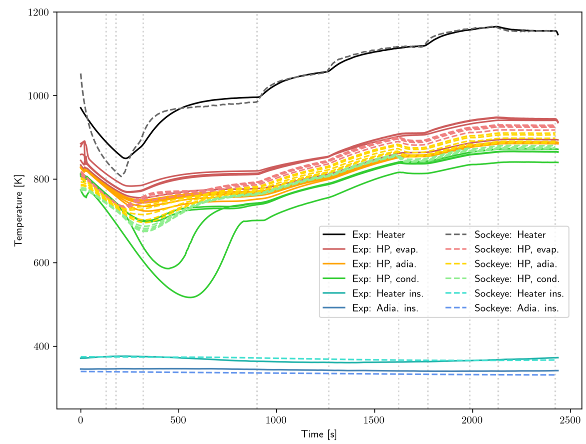

Fig. 4. Transient temperature solution compared to experimental data. The vertical, dotted lines denote various transient events, including heater power changes and changes to the heat exchanger air or water mass flow rates.

and compared to measured data as a means of code validation. Reasonable agreement between the code and experimental data was achieved despite using the simplified heat pipe model. The main discrepancy is in the predicted sonic limit and its impact on the temperature drop across the heat pipe. It is believed that a higher fidelity heat pipe model can help to produce a more realistic transient, as well as having knowledge of the internal dimensions of the heat pipe.

Future planned work includes the substitution of a more mechanistic heat pipe model based on fluid equations rather than heat conduction. Additional potential improvements to this model could include improving the thermal properties of the various materials, an explicit heat exchanger model so that uniformity does not need to be assumed, and the inclusion of more bodies into the model, such as structural elements and the unheated length of the heater. Depending on the mesh approximation of these elements, the problem may need to be a three-dimensional problem rather than a two-dimensional problem. To enhance the experimental data accuracy, we will explore a more precise temperature measurement method to minimize the thermal resistance at the evaporator and condenser sections. Additionally, we consider using uniform filler materials, such as silicon carbide powder, in the gap between the heat exchanger and heat pipe to ensure an accurate account of thermal conductivity.

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