ANALYSIS OF PIPELINE TRANSMISSION LOSSES OF THERMAL PACKETS IN AN URBAN THERMAL GRID

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Abstract: An urban thermal energy grid will allow buildings to meet thermal energy demand by taking advantage of more optimal heat sources installed in surrounding buildings such as waste heat or unused heat generation capacity. Energy can be transferred on demand in the form of “thermal packets” of hot or cold water through a grid of pipelines. Short-term transient heat losses as the pipes are heated or cooled and diffusion along the packet length must be accounted for in control and evaluation of the system.
The efficiency of transferring thermal energy in insulated water pipelines of various materials is evaluated through simulation and experiment. The temperature of the packet as a whole degrades as it travels through the pipes. Initial results indicate the degradation of the packets can occur at the front and tail ends as “blunting” or the packet may retain a “sharper” shape but the temperature of the entire packet including its “core” tending to drop. Small scale experiments over 25m pipe lengths reveal that heat transfer with system components can be significant. Large scale experiments are set to begin in summer 2014.

Key Words: thermal packet, district heating and cooling, transient loss, pipe flow

1 INTRODUCTION

An urban thermal energy grid would be used to allow buildings to meet thermal energy demand by taking advantage of more optimal heat sources installed in surrounding buildings; such as waste heat, stored thermal energy, or unused heat generation capacity from more efficient heat sources. The energy can be transferred as needed in the form of hot or cold water through a pipeline in the form of “thermal packets”.

We use the term “thermal packet” both to emphasize the transient nature and the allusion to telecommunications data packets. Hot or cold water (relative to a base temperature, such as the temperature of the city water supply) is stored in tanks at buildings linked to the network. An experimental “thermal loop” is under construction in Osaka (first section to be completed and begin testing in summer 2014) connecting buildings at distances on the scale of hundreds of meters with water pipelines. If successful the system could be expanded into a “thermal grid”.

Water at base temperature is kept in the thermal loop. When heat energy is to be transferred, valves at the sending tank open, and a set amount of hot or cold water is pumped into the loop as the water in the loop is set in motion. The source valve closes to end the packet, a loop valve opens and the packet continues along the loop, driven by base temperature water behind it. When it arrives at the destination, the destination storage tank valve opens and takes in some or all of the packet. If some of the packet remains, it can be taken in by other buildings. The water in the loop continues moving, with buildings along the loop giving or taking hot or cold packets. Unused packets can be stored in multi-temperature, multi-level holding tanks and used when needed.

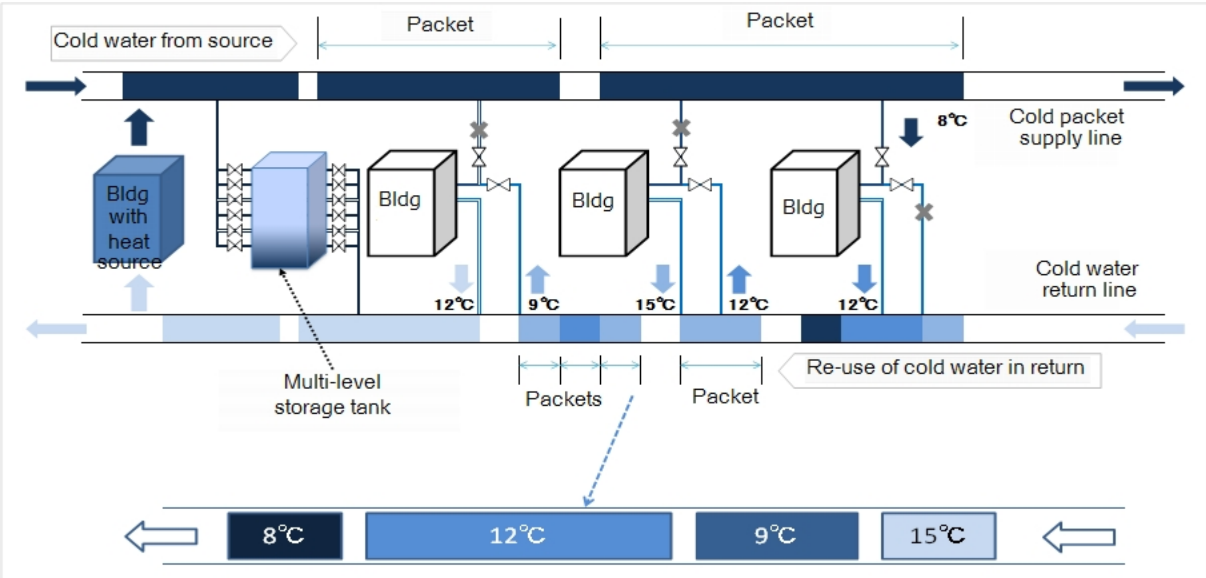

Figure 1: Concept diagram of thermal loop with 4 buildings and 1 storage tank (second box from left). Thermal packets of different temperatures are added to or taken from the loop for use in buildings or for storage.

To control the system, the amount of heat lost in transit and the temperature of the packet on arrival must be predicted. The issue addressed in this paper is simulation and initial small-scale experiments to determine the transient thermal losses from the pipes used to transfer heat between buildings. Influences of heat loss and gain via the pumps, valves and fittings are also being examined. Interaction with systems in the buildings such as heat pumps and chillers is not examined here.

2 EXPERIMENT APPARATUS

Testing of the thermal packet losses is proceeding on two scales. Small scale experiments are done in a laboratory space with lengths of 10A copper pipe linking water tanks at lengths from 25m up to 75m. Here, we discuss the results of the 25m pipe length. Packets are controlled by a system of automated valves termed a “thermal router”. A diagram is shown in Figure 2. Properties of the pipe and insulation are given in Table 2.
Large scale trials will be done in 2014-2015 at a site in Osaka City linking 4 facilities, a large convention center, a hotel, a shopping mall and a train station. Some pipes will be installed along existing rail lines. Before selecting pipes, two pipe types were simulated, one is PE plastic with rigid foam insulation and a PE plastic outer cover. The other is a stainless steel pipe with similar insulation and cover. Properties are listed in Table 2.

In the small scale experiment, sheath thermocouples are installed at several positions along the pipes. In the 25m case examined here, they are installed 2.5m, 12.5m, and 22.5m along the pipe from the source tank. A pump that draws 400W of electricity is used to drive the flow. Its mass is 6kg. Valves are automated and operated through a control program. However, some flow rate regulation valves are set manually. To fit the lab space, 10 elbows and 2 tees are incorporated into the initial segment of piping including the pump and control valves in the region from x=0m at the tank outlet to about x=1.5m, after which the piping is smooth but in a coil of about 1m diameter, with no elbows or tees. There was some difficulty in maintaining a uniform speed when switching from tank-to-tank packet transmission to loop-to-tank at the end of the packet. Further, velocity and temperature at the inlet had a delay of about 7 seconds in response. This is discussed in the comparison with simulation results.

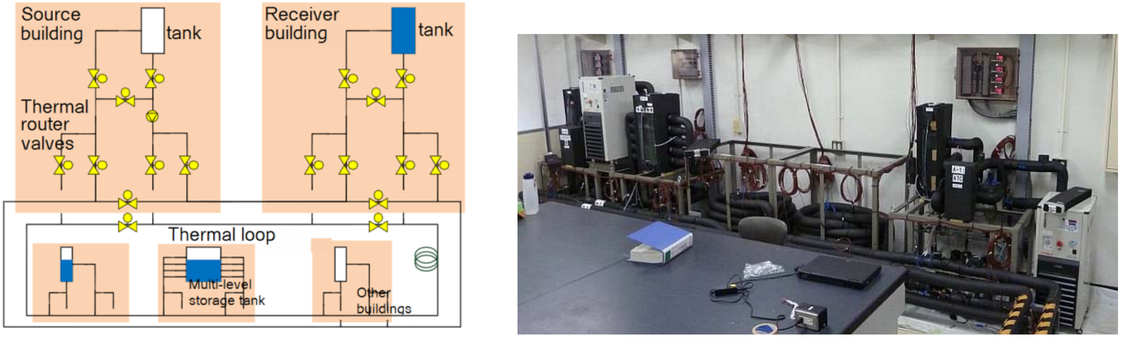

Figure 2: Left: Diagram of thermal loop small scale model of 4 “buildings” and 1 storage tank. Right: Photo of two “buildings” with tanks, a multi-level storage tank and piping.

Table 1: Pipe specifications

	Small scale experiment		PE plastic 50A pipe for large scale trial			Stainless 50A pipe for large scale trial		
Layer	pipe	insul.	Pipe	insul.	Cover	pipe	insul.	Cover
Material	Cu	Foam rubber	PE	Rigid PU foam	Low-den PE	SUS 303	Rigid PU foam	Low-den PE
Inner radius (mm)	5.7	6.35	25.5	30.5	67.5	28.6	30.3	67.3
Thickness (mm)	0.65	33	5.0	37	3.0	1.7	37	3
Density (kg/m^3)	8920	48	945	80	933	7800	80	933
Thermal cond-uctivity (W/mK)	397	0.037	0.0232	0.0263	0.026	16.3	0.0263	0.026
Specific heat (kJ/kgK)	379	1050	2050	1627	2300	460	1627	2300
Water/pipe therm. mass ratio	141		31			53		
Therm. diffusivity of pipe layer(m^2/s)	1.16×10^{-4}		1.19×10^{-8}			4.54×10^{-6}		

3 THEORY

To effectively use relatively small pulses of hot or cold water, the transient losses must be evaluated. Three 3 main sources of loss of useful energy from the thermal packet as it travels through the pipe are expected:

- Heat transfer with the pipeline as the pipe temperature is changed
- Heat transfer out of the pipeline, through any insulation to the environment
- Degradation of the packet due to non-uniform velocity profile and interaction with the base-temperature water at each end of the packet

A long history of research on flow in pipes has shown that the velocity profile of turbulent flow is fairly uniform, while laminar yields a parabolic velocity profile (Streeter & Wylie, 1985). Laminar flow would lead to quick degradation of the packet. Even if heat energy were not lost from the system, but the packet would “stretch” itself, effectively diffusing into base temperature water at each end. Turbulent flow will help the packet keep its shape. However, turbulent flow yields much higher heat transfer with the pipe wall as per correlations such as by Gnielinski (below). If the thermal mass of the pipe is relatively small compared to the thermal mass of the water inside and the pipe is insulated, the loss to the pipe should be relatively small.

For comparison of packet length and its effect on losses, we define the "unit thermal packet". A pipe of length L at base temperature T_b is full of water also at base temperature. A pulse of hot (or cold) water at pulse temperature T_p at average velocity v is sent through the inlet. The pulse continues until the elapsed time of the pulse t_p reaches the time at which the front edge of an ideal lossless pulse should arrive at the destination, when

$$t_p = L/v \tag{1}$$

we term this a "unit pulse" of 1U. A pulse of twice the duration as 2U, half this duration as 0.5U, and so on. The pulse is ended at the source by switching the inlet water temperature back to the base temperature, but continuing the flow of water at average velocity v. It is expected that packets of much greater duration than 1U will approach steady-state behavior, such that the only remaining thermal losses would be through the pipe insulation to the environment. Single pulses or fractional pulses should be dominated by transient heat losses.

3.1 Heat transfer in pipe flow

Fluid flowing through a pipe of different temperature exchanges heat at the pipe inner surface. Heat transfer continues through the pipe wall, to any insulation and eventually to the surrounding air or soil. There are empirical relationships for the Nusselt number, Nu of pipe flow which are used to find the heat transfer coefficient, h for specific pipes of diameter D with a flowing fluid of thermal conductivity k using Eq. 2. The overall heat transfer coefficient, β for a pipe determines the rate of radial heat flow from the fluid to the surrounding environment (Lienhard, 2011).

$$Nu = hD/k \tag{2}$$

$$\beta = \frac{1}{\frac{1}{h_{in}} + \frac{1}{h_{pipe}} + \frac{1}{h_{out}}} \tag{3}$$

An empirical relationship for Nu under turbulent pipe flow developed by Gnielinski (1976) is given in Eq.4 which requires the Reynolds number, Re and friction factor, f found with Eq. 5.

$$Nu = \frac{(f/2)(Re - 1000)Pr}{1 + 12.7\sqrt{\frac{f}{2}}\left(Pr^{2/3} - 1\right)} \tag{4}$$

$$f = 0.079\,Re^{-0.25} \quad for\, Re < 10^5 \tag{5}$$

The Prandtl number, Pr is a function of viscosity, thermal conductivity and specific heat. In water it is temperature dependent. In the case of evaluating flow in heat exchangers pipes, the average of the fluid inlet and outlet temperatures is often used (Sucec,1985) to determine their values. Here, we use the average of the pulse and base temperatures.

3.1.1 Iterative 1-D model for thermal packet flow

Models of "plug flow" have been used by Dahm (2001) and Nobe (1989) as a basis for computer programs to quickly evaluate transient pipe losses in district heating systems and building air conditioning systems respectively. They treat the water flowing in the pipe as a

series of "plugs" of length Δx of uniform velocity and temperature. Heat is exchanged with the inner pipe surface as per an appropriate relationship for heat transfer coefficient. Each plug travels along the pipe, exchanging heat with it and changing temperature uniformly. Plugs are modeled one after another to simulate the complete flow. Conduction of heat along the pipe is usually small, due to the small cross-sectional area and the smaller temperature difference compared to the temperature difference with the fluid such that it can be ignored for typical pipe sizes and fluid flow rates in simple plug flow models.
Here, a simple version of the model is adapted to run iteratively in Microsoft Excel® to solely test the effect of heat transfer between the pipe and water, with the pipe outer surface set as adiabatic. The pipe is divided into cylindrical shells of the plug length Δx. A 1m length of pipe is used, with a fluid velocity of 1m/s for a duration of 1s, followed by base temperature water for 4s. Results showed no significant change for reducing Δx further than 1/500 of the pipe length. A unit packet is modeled to start entering the pipe at time zero. Velocity is set at 1m/s. The pipe and water base temperature is set at 5˚C. The packet temperature is set at 20˚C. Each plug is set as 0.002m in length. 500 plugs are sent through the pipe at 20˚C, followed by 2000 plugs at 5˚C to yield the effect of the base temperature water pushing the packet.

Results of the model showing the pipe outlet flow temperature and the average temperature of the entire pipe for a 50A SUS pipe and a 50A PE pipe are in the figures below. Losses are defined in terms of the percentage of exergy loss L_b evaluated at the pipe exit and integrated over time through to 5s, compared to that contained the original packet where exergy, B is defined as (van Wylen et al., 1994).

$$B = Q\left(1 - T_o / T_b\right) \tag{2}$$

and percentage of temperature change lost L_t as maximum temperature reached at the outlet against the temperature of the original packet. Heat losses are net zero due to the adiabatic outer shell condition. Heat taken in by the pipe is recovered in the base temperature water at the tail of the packet. The calculations yielded net heat loss of zero.
Three packets, of lengths 0.5U, 1U, and 2U were evaluated. The shape of the packet is slightly "blunted" at the start in each case, but the packet temperature profile is straight at the edges as no diffusion along the packet length is included here. Each packet has a tail of similar size and shape to the area lost at the front "blunted" end. This is the recovered heat from the pipe.

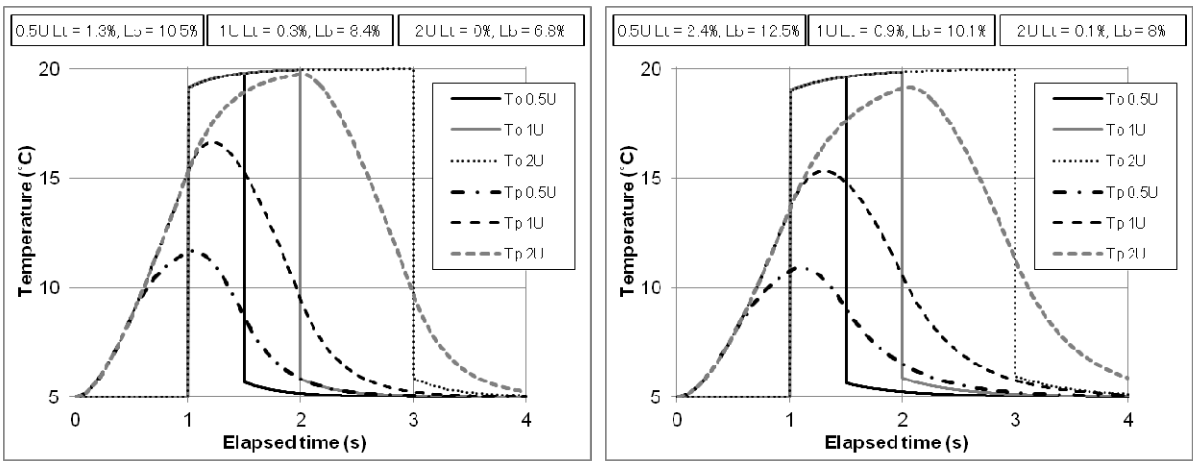

Figure 3: Results of simple 1-D plug model for 1/2 unit pulse, 1 unit pulse, and 2 unit pulse in 50A SUS pipe (left) and 50A PE pipe (right)

The SUS pipe shows that for the 2U case, the pipe temperature reaches the packet temperature, with no temperature loss. This shows the pipe has reached a steady state, longer packets will also achieve steady state. In the 1U case, the temperature loss is only 0.3%, nearly steady-state. The PE pipe does not quite reach the packet temperature even in the 2U case, with slight temperature loss remaining. In all cases, percent temperature and exergy losses increase in inverse proportion to the packet length. It is clear that longer packets are desirable. This model does not account for continuing losses as the packet continues to travel along a longer pipe, where any temperature losses will be compounded.

3.2 CFD simulation of thermal packets

Two sets of models of the thermal loop pipe were created in Ansys FLUENT® 14.0. The first model sets are 2-D axi-symmetric models of straight lengths of pipe. The second set of models are 3-D models of short sections of pipe with elbows or large masses connected to the pipe as a model of the thermal mass of pumps or instruments.

A sensitivity study was done with a 25m length of simulated pipe for number of mesh elements, as well as effect of elevated environment temperature and addition of heat after the inlet, such as from a pump. In these cases a cold packet at 280K is sent through a pipe at initial temperature of 293K. Velocity is 1.22m/s, the same as the experiment setup, with a packet time of 50 seconds. Pipe insulation outer surface was set to $5W/m^2K$ natural convection loss. Simulations were run at time steps on the order of 0.001s yielding a Courant number of near unity until elapsed time of 200 seconds. In Case 0, the air temperature is set to the same 293K with natural convection. Case 1 tests the effect of a higher environment temperature. Case 2 tests the effect of adding 100W of heat after the inlet. Case 3 tests a mesh with more detail in the pipe wall. Cases 4 and 5 test mesh with finer mesh.

The exit temperature is integrated over time to yield the net heat remaining in the packet . Compared to the Case 0, where 1.12% of the cooling energy is lost, there is no great change in the other cases, except the case of added heat. The pump in experiment has a maximum power of 400W, indicating it might affect the experiment results.

Table 2: 2-D axi-symmetric pipe model conditions and results

Case	Temperatures			Mesh elements	Packet velocity (m/s)	Added heat	Cooling Energy lost
	Packet	Base	Env. air				
0	280 K	293 K	293 K	81000	1.22	0	1.12%
1	280 K	293 K	**300 K**	81000	1.22	0	0.98%
2	280 K	293 K	293 K	81000	1.22	**100W**	3.66%
3	280 K	293 K	293 K	**99000**	1.22	0	1.51%
4	280 K	293 K	293 K	**371000**	1.22	0	1.23%
5	280 K	293 K	293 K	**187000**	1.22	0	1.44%

Three 3-D models of the initial 80cm at the inlet were also made in FLUENT. A section with 2 elbows and a section with 2 elbows and a 6kg mass representing the pump are shown below. A straight pipe was also modeled of length such that the pipe volume is precisely the same as the pipe with 2 elbows.

The simulations were run to 4 seconds at the same velocity as the experiment. The outlet temperature of the 2 elbow bend case was not significantly different from the straight case. However, the outlet temperature of the pump mass case was almost 0.2K higher, a loss of about 1% of the 20K temperature difference between base temperature and the cool packet. Pump mass and mass or large valves could be significant.

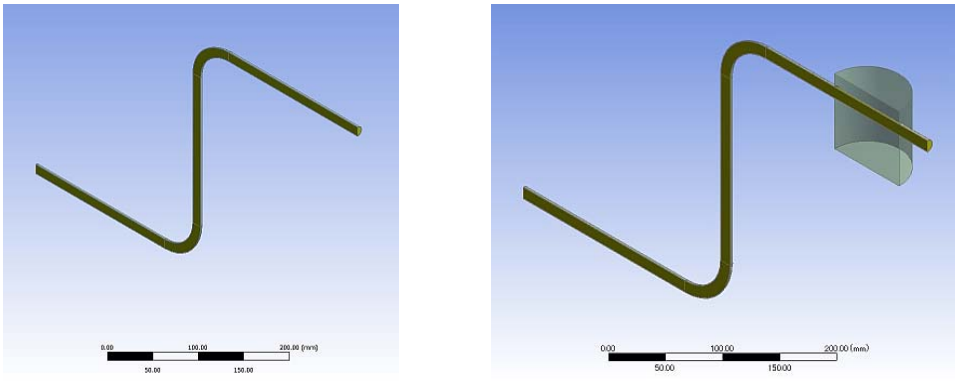

Figure 4: 3-D model of pipe inlet section with elbows (left) and added 6kg pump mass (right)

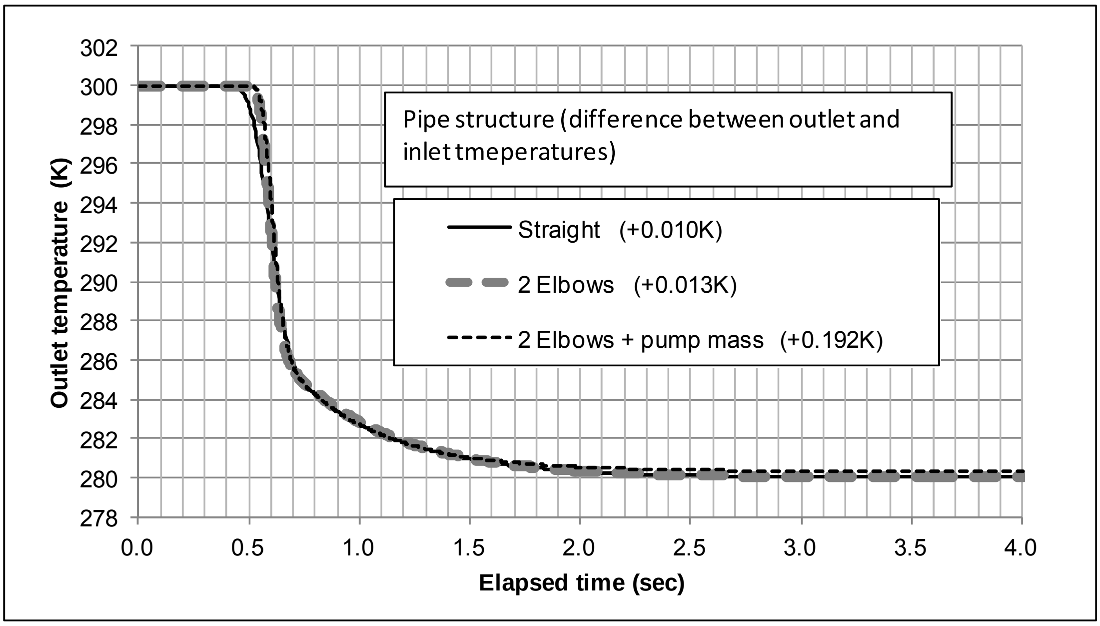

Figure 5: Outlet temperature results of 3-D model of pipe inlet sections

A simulation of the experiment case using the axi-symmetric model was modified to account for the delay in temperature change and velocity change at the inlet. Instead of a clean step function for temperature, an approximation was derived from experiment data, programmed in C++ and input as a user-defined function into the FLUENT inlet boundary condition. The results of the simulation and experiment temperature values at the 3 sensor locations are shown below. Although the shape is generally the same, and the slope of the temperature recovery at the end of the pulse (elapsed time 60 – 80 seconds) is similar, the simulation and experiment do not match well. Experiment temperature drops do not fall to the temperature sent from the inlet. The pulses also are shorter than predicted in simulation.

As the initial section of piping has many elbows, tees, and valves, a second simulation was run only on the smooth section of pipe from x=2.5m to x=22.5m, with the inlet condition modeled on the experimental data for x=2.5m.The shapes of the x = 12.5m curve matches the simulation more closely, indicating the elbows may be a cause of the mismatch. However, the timing of the x=22.5m curve is still off. This may indicate the velocity of the pulse is different than in the simulation. Further, it is possible there are greater heat losses in the insulation layer or to the pipe fittings and supports. The thermal diffusivity of the copper pipe is about 100 times that of steel, and 10,000 times that of plastic. The “blunting” of the packet edges can be expected to be quite strong if more heat is lost to fittings, sensors, and tees. It is clear more investigation is needed.

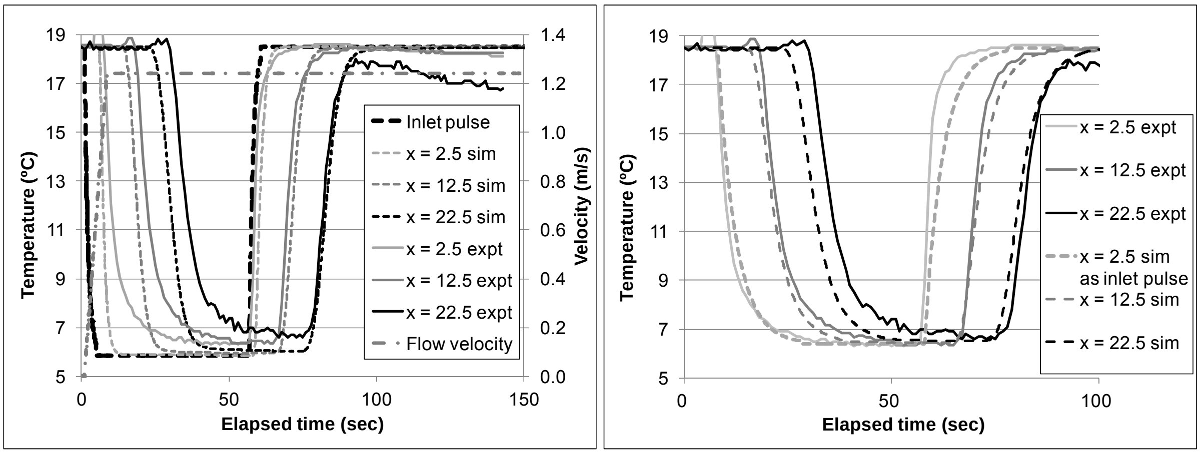

Figure 6: Comparisons of experiment(expt) and simulation(sim) results. Left: Simulation from x=0 inlet does not match well. Right: Simulation with inlet condition modeled after x=2.5m data yields better match, indicating valves and elbows near inlet may distort the packet.

Simulations of the larger scale system using PE plastic pipe or stainless steel pipe to transmit a warm packet at 2m/s for 25 seconds over a 100m length with no loss to the environment reveal a possible difference in packet behavior. Virtual sensors are modeled at 25m intervals.

Results (below) show the stainless steel pipe will quickly change temperature, absorbing the energy from the packet, causing a “blunting” of the edges of the packet, but the “core” of the packet will remain at the maximum temperature. The PE pipe absorbs the heat more slowly, such that the packet stays “sharp” at the edges, but the core of the packet is also affected, slowly dropping in temperature as it travels. After traveling 100m, the peak temperature is already more than 5% lower than an ideal packet. The trend seems linear, so there would be significant loss if transmitted for lengths of hundreds of meters.

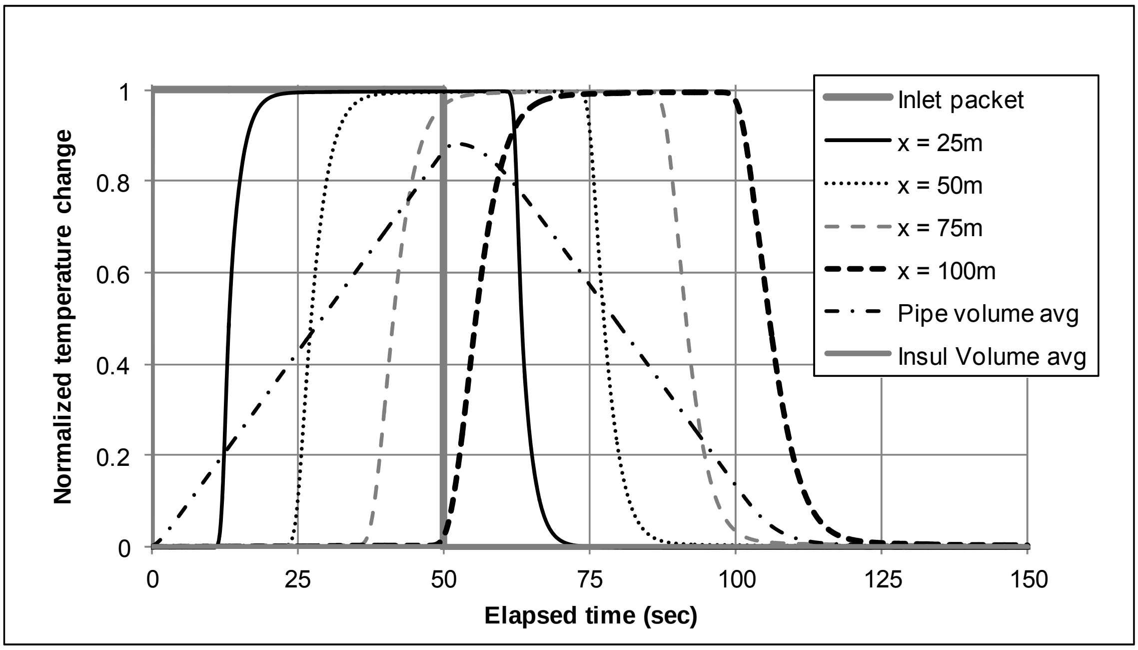

Figure 7: Simulation results for 50A SUS pipe. Packets are “blunted”. “Core” not affected.

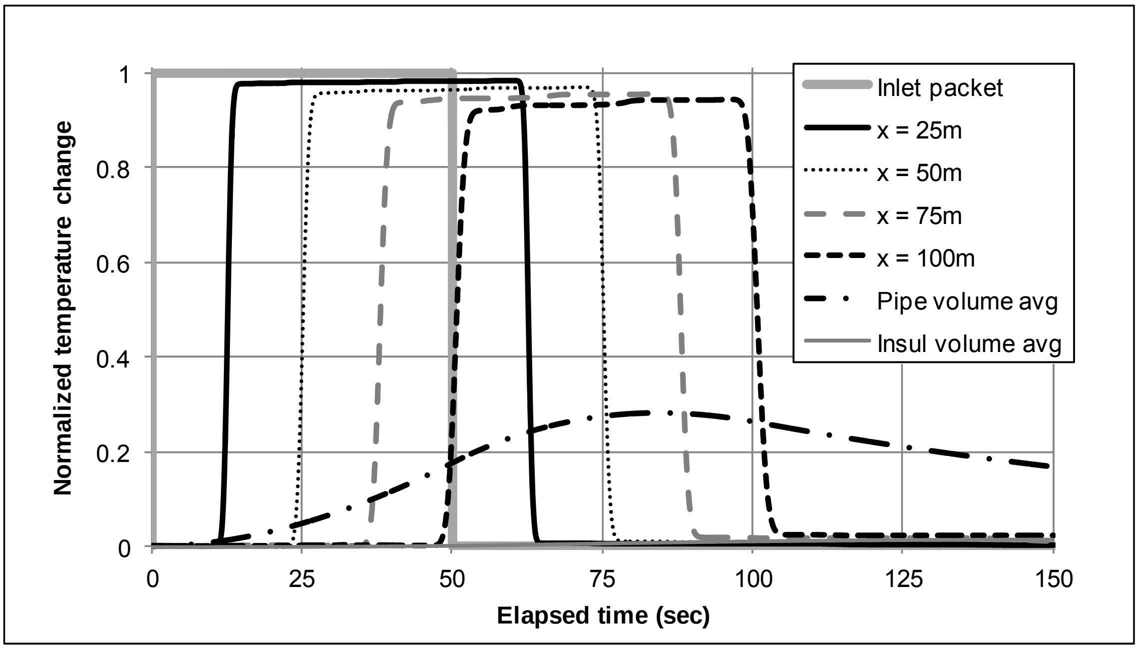

Figure 8: Simulation results for 50A PE pipe. Packets are "sharp". The "core" is affected.

4 CONCLUSIONS AND DISCUSSION

Thermal packets can be transmitted over distance while losing little of their energy. The change in the total energy in the packet was expected to change due to the following factors.

- Heat transfer with the pipeline as the pipe temperature is changed.

The pipe material and thickness can strongly affect the thermal packet. Materials with high thermal diffusivity can quickly absorb the heat from the packet, causing the leading edge of the packet to be strongly affected, "blunting" the shape of the temperature curve over time. However, the quick interaction with the leading edge of the packet, resulting in the pipe quickly reaching the packet temperature, helps leave the "core" of the packet unaffected. The maximum temperature difference of the packet is maintained. Materials with low thermal diffusivity rob heat more slowly from the packet, such that the leading edge is less affected. The packet maintains a "sharper" temperature curve, but when the "core" of the packet travels through the pipe, there is still a temperature difference with the pipe. Heat is transferred from the core of the packet. The temperature difference in the pulse does not stay at the maximum. This would compound as the packet travels in longer pipes. The ratio of the thermal mass of the water inside the pipe to the thermal mass of the pipe itself determines the maximum possible energy loss from the water. It should be kept as high as possible.

- Heat transfer out of the pipeline, through any insulation to the environment.

With proper insulation these losses are smaller than the other effects and can likely be ignored.

- Degradation of the packet due to non-uniform velocity profile and interaction with the base-temperature water at each end of the packet.

Further experiments and simulations are needed at various flow velocities to study this influence on the packet.

- A further influence affecting the packets significantly may be heat transfer with components of the system itself, increased heat transfer due to pipe elbows and fittings, or incorrect physical properties in the model.

Simulations of single pipe elbows showed little difference in average outlet temperature and velocity from straight pipe. Simulations of heat transfer from a large mass attached to the pipe similar to the pump used in the experiment showed a small effect on the order of changing the packet maximum by 1% in that case. Simulation of adding heat to the pipe after the inlet, such as from a pump waste heat, showed that thermal gain could significantly affect the packet. Further experiments are needed to isolate the effect of the pump and its waste heat on the system. Further simulations of the effect on the shape of the packet due to large numbers of elbows and tees in succession are being conducted. The physical properties of the materials, especially the insulation, may differ from specifications. This will be investigated.

Simulations and experiments will continue in an effort to isolate and characterize each of the above factors. Further, the balance between the “blunting” effect and the “core loss” effect must be accurately modeled to facilitate pipe material selection and system design. This will depend on the nature of the demand on the user end. In the case of warm packets, is it better to have a blunted pulse or a sharp pulse? Is it better that the core of the pulse remain at the maximum packet temperature difference, or that the front end of the pulse arrives at a relatively high temperature at the cost of the core not reaching the maximum temperature? What is the shortest packet that is worth sending? Most importantly, how much energy savings does the entire system yield? These questions will be investigated as the project continues.

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