COMPUTATIONAL MODELING TO REDUCE IMPACT OF HEAT STRESS IN LACTATING COWS

by

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DEDICATION

A Elisabet & Brenda, para que las ilusiones no mueran.

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ABSTRACT

Climatic conditions inside the dairy barn do not concern dairy farmers until those conditions begin to affect productivity and, consequently, profits. As heat and humidity increase beyond the cow's comfort levels, milk production declines, as does fertility and the welfare of the cow in general. To reinforce the cooling mechanisms currently used, this work proposes an alternative system for reducing the risk of heat stress. This innovative conductive cooling system does not depend on current weather conditions, and it does not require significant modifications when it is installed or during its operation. Also, the system circulates water that can be reused. Given that a review of the literature found very few related studies, it is suggested that each freestall be equipped with a viable prototype in the form of a waterbed able to exchange heat. Such a prototype has been simulated using Computational Fluid Dynamics (CFD) and later verified by a set of experiments designed to confirm its cooling capacity. Furthermore, this investigation sets the foundation for modeling temperature in a water supply system linked to the waterbeds. EPANET, a software program developed by the Environmental Protection Agency, simulates the hydraulic model. Its Water Quality Solver has been modified according to an analogy in the governing equation that compares mass to heat transfer and serves to simulate water temperature as the water is transported from its source to the point of delivery and then as it returns to the same source.

Keywords: dairies, heat stress, heat exchanger, heat conduction, cooling/heating system.

1. INTRODUCTION

1.1 Heat stress in milk cows

To produce milk and its byproducts in USA's rapidly growing southern regions, from Arizona, to Texas to Florida, dairy barns must be properly constructed and outfitted with equipment specifically designed to reduce the risk of heat stress. Inadequate dairy-barn design and operation can shift climate conditions as important as feeding regimes and management practices. Commonly used equipment, such as fans, soakers, misters, and wet pads, will alleviate heat stress during dry weather; however, these types of equipment do not operate effectively in geographical regions that are regularly too humid or periodic seasons such as the monsoon in Arizona. On the other hand, heat stress, which alters a cow's physiology, can lead to adaptive responses that adversely affect productivity and also the cow's overall well being; this stress can be measured in terms of heat intensity, time of exposure and frequency.

According to Armstrong (1994) heat stress sets in when temperatures rise above 23°C or 31°C for 100% or 0% levels of relative humidity, respectively. The intensity of the stress can be estimated using the Temperature-Humidity Index (THI), which considers the effects of convection, radiation and evaporation. An early investigation by Igono et al. (1992) found a correlation between THI and the consequences of heat stress (a decrease in milk production and a lowered fertility rate in summer, which will offset the cow's gestation cycle over the rest of the year).

Interest in this topic over the past 30 years has produced a series of works contending that milk production and gestation are affected (Yousef, 1985; Hansen and Arechiga, 1999; Barash et al., 2001; Bouraoui et al., 2002; West, 2003; Bohmanova et al., 2007; Piccioli et al., 2009; Thatcher et al., 2010). These investigations showed that heat stress becomes a multi-factor problem; even such parameters as feeding and genetic selection are influenced by the intensity of the heat-related stress. Furthermore, heat stress arises from a combination of variables having to do with the animal's age, weight and also seasonal weather variations and how the animal is managed. A summary of research on the effects of heat stress in lactating dairy cows was presented by Kadzere et al. (2002), and the relation between heat stress and metabolic responses is discussed in Baumgard and Rhoads (2011).

Regardless of breed, age, size, feeding and the health of dairy cows; heat stress will affect two main productivity criteria: milk production and fertility rate. Additionally, heat stress will also undermine the cow's physical state. To prevent profit loss and protect the cow's health, this study proposes alternative equipment and techniques for keeping the cow cooling enough to avoid heat stress. The systems presently used in barns to alleviate heat stress—systems that typically involve mechanical ventilation and fogging apparatus—can be reinforced with alternative methods such as removing heat by conduction.

1.2 Literature review

In addition to experimental investigations that consider several of the factors that cause or exacerbate heat stress, there is a need to study the subject as a heat transfer problem. It considers not only the transfer of heat occurring between the cow and its surroundings but also the transfer occurring between the climate inside and the climate outside of the barn.

The common paths by which that transfer occurs in livestock were first investigated by Wiersma and Nelson (1967). Their pioneering study used a replica of a cow to quantify the effects of cross ventilation. Another study by Armstrong, (1994), also found that providing ventilation and reducing radiation could reduce the impact of heat stress.

Evaporation, which is a phase-change phenomenon, can be a highly effective means of removing heat. Therefore several studies have attempted to quantify evaporative heat loss (Gebremedhin and Wu, 2001; Gebremedhin and Wu, 2002; Jiang et al., 2005; Maia et al., 2005; Gebremedhin et al., 2008; Berman, 2009; Berman, 2010; Gebremedhin et al., 2010; Gomes da Silva et al., 2011).

Further investigation of the heat stress problem has sought to evaluate scenarios that were designed to quantify heat release from cow body. Models capable of simulating heat transfer by considering various physiological aspects such as sweating, vaso-dilation, panting, respiratory heat loss, heat storage and the rates at which body temperature changes have been proposed (McGovern and Bruce, 2000; Berman, 2005;

Zahid, et al., 2006; Keren and Olson, 2007). Investigations about heat production have also been conducted (CIGR, 2002 and CIGR, 2006).

Recently, too, a number of researchers have used Computational Fluid Dynamics (CFD), which is a method for finding numerical solutions, to evaluate convection, radiation and conduction simultaneously and thereby analyze the cow-environment interaction, (Rojano et al., 2011; Mondaca et al., 2012). Also, cooling systems have been investigated in order to determine their efficiency. An early effort by Avendaño (1995) presents an analysis of the cost-benefits associated with a system involving fans and sprinklers. Another study by Gottardo et al. (2005) showed that ventilation and sprinklers will increase milk production by about 5%. Exhaustive studies by Norton et al. (2007, 2009, and 2010) and Gebremedhin and Wu, (2005) suggest that a CFD model is capable of analyzing convection and radiation heat transfer modes. Subsequent investigation based on modeling has been proposed as a way to study temperature and humidity in a barn and also how evaporation and the heat generated by animals affect those conditions; and the model adapted to a non-linear control design (Daskalov et al., 2006).

Even so, most of this investigation focuses on radiation, convection and evaporation since a lactating cow remains standing while eating and being milked as well as during other activities, despite the fact that a cow's natural behavior indicates, it prefers to recline for long periods of time every day (Osterman and Redbo, 2001; Manninen et al., 2002; Veissier et al., 2004; Cook et al., 2008; Provolo and Riva, 2008; Tolkamp et al., 2010). This behavior influences the effectiveness of cooling mechanisms primarily by convection, radiation, evaporation. Conduction, another important heat transfer mechanism, will only be effective when the bedding surface beneath the animal is cooler than the animal's skin. For this reason, several recent studies have proposed methods for cooling the bedding. One such study involves a cooled pad placed under the bedding and working as a heat exchanger (Shaffer et al. 2001). This device contributes to the animal's ability to slough off excess heat in order to avoid heat-related discomfort.

The challenge of cooling by conduction has to do mainly with the percentage of the animal's skin that will be exposed to the bedding surface most of the time. Normally it will be no more than 20%, according to Bastian et al. (2003). The effectiveness of the system also depends on the thermal properties of the bedding material and its depth, the capacity of the heat exchanger to remove heat, and the heat fluxes allowed by the cow. However, an adequate design could offer a promising solution, if it takes into account such factors as energy and water consumption and their associated costs.

The current evaporative cooling mechanisms that are particularly popular in arid and semi-arid lands such as Arizona are based on the temperature and humidity of the ambient air. These mechanisms are efficient and effective when water vapor can still be added to the air. However, few regions of the globe can count on these conditions most of the time. Even more, they also require intensive energy and water use.

In exploring the possible use of a conductive cooling system, this work considers key technical issues. The first involves the way that a dairy cow interacts with its environment, and an analysis of that interaction should take into account for the experimentation done by previous researchers as well as that conducted in current studies. Also, the variables that have to do with maximizing efficiency should be clearly defined. And because the proposed system is untested under the field conditions, the present investigation should involve elements that can be used in comparison with other current cooling techniques. Moreover, evaluating this proposed system in conjunction with current cooling techniques should lead to a search for higher efficiency and also further innovation.

1.3 Objectives

The proposed conductive cooling system is intended to serve as a way to augment existing systems that are designed to reduce the danger of heat stress. Since no such system is currently used on a commercial scale, the present work must propose a feasible design that will meet the following general and specific objectives:

General objectives

- Design a system that will help to decrease the threat of heat stress in dairy cows
- Verify that the system will, at minimum, help to maintain milk-production and fertility rates
- Design an alternative cooling mechanism that will use less energy and water

Specific objectives

- Design a bed that can serve as a heat exchanger (Appendix A)

- Verify the efficacy of the prototype by means of computer simulation and direct experimentation (Appendix A)
- Simulate a hydraulic and thermal model of the pipe network that will supply water to each heat exchanger (Appendix B)

2. PRESENT STUDY

2.1 Overall summary

The present work was conducted in order to design and successfully operate a conductive cooling system that could serve as an alternative means of preventing heat stress in dairy cows. Regardless of whether a dairy provides its animals with free-stalls or tie stalls, it can use this system because it involves a heat exchanger placed under the bedding on the floor of the stall. These heat exchangers are implemented in a water supply system that pumps chilled water through the waterbed and transports heat away from the bed so that the temperature of the bed remains at a level that is lower than the cow's skin temperature. Then, the system can be regulated to meet specific cooling requirements.

The waterbed's dimensions and geometry follow standards recommended by the ASABE (2010). Materials meet the standards regarding comfort and heat transfer parameters. The design operates in such a way that cooling needs can be met. Because this is a new device, its ability to fulfill heat transfer and comfort requirements is demonstrated. The proposed waterbed is located under a layer of sand, which is the most commonly used bedding material because it provides comfort effectively and inexpensively.

Both previous and current studies have aided in the development of an adequate design that can operate as intended. These studies deal with heat transfer performance and evaluate the heat exchanger. They use Computational Fluid Dynamics (CFD), a numerical method that relies on the equations governing heat and mass transfer. Results obtained from CFD helped us to determine the characteristics that will yield the best design and the best way to operate it.

This design, as a conceptual proposal, has been tested. The waterbed specifications mentioned in Appendix A were followed during its fabrication. Then the prototype was tested using an experimental setup (also described in Appendix A). To determine the heat fluxes passing through the operational parameters (flow rates, the temperature of the water supplied to the waterbed, and the thickness of the sand layer and its moisture content) were taken into account. This parametric study determined the cooling capacity and will serve as a baseline for further studies related to economic, environmental and energy issues.

The second part of this investigation proposed a large-scale setup that simulated the entire chilled-water supply system with a set of heat exchangers. The heat exchangers' performance could thus be determined separately using the method described in Appendix A. The water supply system requires predicting water temperature when water runs through an insulated pipe network. To make this prediction, we used a methodology (described in Appendix B) that computes heat transfer rates for each pipe based on the pipe's thermal boundary conditions. This approach enabled us to calculate the water's temperature at the time that the water was delivered to the heat exchanger. The water temperature and flow rate conditions in each heat exchanger determine the system's cooling capacity. If it is possible to know the water temperature in the whole system, then it is possible to know the capacity and efficiency of the system. Furthermore, including hydraulic parameters for pumping makes it possible to estimate the amount of energy spent to operate the system versus the amount of energy transferred by heat exchangers.

Appendix B provides an example of the chilled water supply system linked to a set of heat exchangers. The pipe network design takes care of such parameters as the locations of the heat exchangers, the number of heat exchangers, the flow rate demanded, and the topography of the area. The same example also shows how the heat exchangers are linked to the water supply system. The heat exchangers' performance should be based on the heat transfer rates obtained from a particular water temperature and flow rate. Additionally, the pressure drops occurring in the heat exchangers need to be included. Then the system's efficiency can be determined.

EPANET can simulate a large-scale water supply system using its hydraulic model. The program can check the parameters related to flow rate and corresponding water velocity throughout the pipe system. The Water Quality Solver within EPANET, a module that has been modified by following the heat and mass transfer analogy, simulates water temperature. This water quality module within EPANET was originally intended to simulate such mass transfer phenomena as chlorine content and its decay as a function of time and/or distance; however, the module has been adapted so that it can be used to estimate heat transfer rates and thus also be used to benchmark any pipe network, adapt any heat exchanger and simulate a specific operation, taking into account thermal boundary conditions.

2.2 Overall conclusions and recommendations

This work was, overall, an analysis of heat transfer as it relates to dairy cows and of how the suggested system can reduce the threat of heat stress. The investigation shows that the proposed system, which uses a heat exchanger to augment the cooling effects produced by chilled water, can significantly help to meet the normal body temperature of an average cow during hot, humid weather. Since the system is likely to work simultaneously with others, its cooling contribution will be affected by temperature and flow rate of the water supplied to the system, as well as how the system is operated and the climatic conditions of the barn. Additionally, the study considers the material qualities of sand and its moisture content, since both affect heat fluxes, but variations of these parameters do not change the energy required to operate the heat exchanger.

The EPANET software program can evaluate a series of heat exchangers within a pipe network. The program determines the hydraulic parameters and defines the water supply system's key components, such as pipe diameters, valves, junctions, and so forth. A particular pipe setup will require a specific total head and flow rates. Additionally, a modified version of the Water Quality Solver, which is a module of EPANET, can serve as a Heat Transfer Solver for estimating variations in water temperature as the water moves through the pipe network. By combining hydraulic and thermal parameters, we were able to evaluate the transfer of heat occurring at the heat exchanger and throughout the pipe network. This approach provided the information we needed to compute the efficiency of the system. Also, such an approach can be used to evaluate efficiency, taking into account heat transfer rates occurring at a given flow rate and energy needs for pumping.

The exemplary, large-scale, chilled-water supply system described in this study was designed to operate in regions where weather causes heat stress. The proposed system could also serve as a heating mechanism without further modification other than changing the temperature of the water supplied to the system.

The methodology suggested herein should serve as a basis for an evaluation of any pipe network linked to a particular design of heat exchanger and can also aid in determining system size, cooling needs, flow rate and thermal boundary conditions. In this study we analyzed the fluid flow and heat transfer of the entire system, and also the heat exchanger's design (along with its experimental verification) when the exchanger was placed underneath the bedding material. It should, however, be noted that we did not verify the accuracy of our estimations of the heat transfer occurring in the large-scale pipe network. Only after this is done can the design and the efficiency of its operation be certified. Also, future experiments and/or operations involving this system may raise new challenges having to do with insulation, maintenance, fouling, lifetime expectancies and other factors involved in the operation.

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APPENDIX A

DESIGN AND EVALUATION OF A HEAT EXCHANGER FOR CONDUCTIVE COOLING IN DAIRY COWS

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Abstract

This study proposes an individual waterbed that can act as a conductive cooling system for dairy cows. Installed in dairy-barn freestalls, the device would help to decrease the danger of heat stress whenever the barn's main cooling equipment fails to produce the desired climate conditions. Placed under the stall's bedding material, the waterbed would work as a heat exchanger. The bedding material (typically sand) would conduct the body heat produced by a reclining cow to the waterbed. Dimensions, materials and operation are suggested. An analysis of the heat exchanger was performed using Computational Fluid Dynamics (CFD) to quantify heat fluxes based on the sand's moisture content, the coolant temperature, the coolant flow rate and the thickness of the sand layer. Moisture content and the thickness of the sand were investigated because they influence the heat flux rates but do not affect the amount of energy needed to operate the device. A set of experiments confirmed the consistency of CFD outcomes; the CFD model was in turn used as a baseline for evaluating the heat exchanger's performance under a set of possible scenarios. The outcomes suggest that heat fluxes are equally affected by sand thickness and moisture content; a moisture content of up to 5% (based on weight) can increase significantly the sand's thermal conductivity, and when combined with a sand layer less than 15.2cm thick, the proposed system can achieve cooling rates of more than 150W/m^2.

Keywords: waterbed, heat conduction, heat flux, Computational Fluid Dynamics.

Introduction

Productivity on a daily basis requires dairy farmers to make strategic decisions having to do with feeding, sanitation, milking schedules and microclimatic conditions. When cows become susceptible to heat stress, the persistence of high temperature and humidity within the dairy barn become an important issue. Investigations have demonstrated that, along with adversely affecting the welfare of the cow, heat stress can significantly decrease milk production and fertility rate (Yousef, 1985; Hansen and Arechiga, 1999; Barash et al., 2001; Bouraoui et al., 2002; West, 2003; Bohmanova et al., 2007; Piccioli et al., 2009; Thatcher et al., 2010).

In contrast to sanitation, regular feeding and milking, all of which can be provided unvaryingly during the year, both the temperature and the humidity inside the dairy barn are subject to seasonal cycles and the constantly fluctuating outside weather; hence, controlling these variables is key to regulating production and determining profits. Rather than attempting to improve the cooling systems currently in use by the dairy industry, to address this problem, the present work focuses on improving the cow's ability to slough excess body heat during periods of high temperature and humidity, when the dairy barn's cooling system cannot maintain conditions below tolerable levels. The proposed system should effectively reduce the effects of hot, humid weather and enable dairy farmers to predict production levels with greater confidence and thereby increase the profitability of their operations. Farmers will also be able to better manage the health and welfare of their herds and most likely gain additional years of productivity from their cows.

As many studies have shown, during hot, humid weather, a dairy cow risks suffering from heat stress, a physiological response that reduces the cow's overall welfare and productive capacity. Investigations of heat stress, such as Armstrong, (1994), have found that stress induced by excessive heat begins to set in when the ambient temperature rises above 23°C or 31°C in conjunction with 100% and 0% relative humidity, respectively. Armstrong (1994) and Thatcher et al. (2010) in their previous investigations of the climatic conditions within dairy barns—that is, the temperature and humidity of the air in the barn—identified a thermo neutral zone and developed a heat-stress scale derived from observations. An index combining temperature and humidity (THI) was also developed and serves to specify heat stress intensity. Even though the differences associated with various breeds, as well as an animal's age, management, feed type and feeding routine, will all affect the THI's estimations (West, 2003), this scale does provide a reliable means of gauging a cow's welfare based on climate and thus can help farmers to know when their cows are at risk of suffering heat stress.

Under typical conditions, unless a cow finds relief, heat stress is unavoidable when temperatures rise close to or go above the animal's core body temperature (38°C) because the cow begins to lose its capacity to release heat effectively. The intensity of the heat stress and the length of time the cow suffers will determine the severity of physiology modification (Armstrong, 1994); such frequency and variability will have repercussions over the short and long term. The cow's initial reactions depend on its own cooling mechanisms; the cow will sweat and also send more blood flowing to its skin, carrying with it body heat to be released via the perspiration (Collier and Collier, 2012).

A heat-stressed cow will also pant, thereby expelling heat through water vapor with each breath. And in order to increase the amount of skin surface to the air, a cow will stand up for longer periods when it is suffering heat stress. All of these actions require metabolic energy, and to supply it the cow will reduce the amount of metabolic energy it would normally spend on producing milk (Barash, et al. 2001), and it will also eat less because digestion requires metabolic energy as well. Needless to say, over long periods of heat stress, the cow's health will deteriorate and it will also become less fertile (Thatcher, et al., 2010, Hansen and Arechiga, 1999). Often during periods of high heat and humidity, impregnating a heat-stressed cow (by means of artificial insemination) will require multiple attempts costing time and money. And once an insemination is successful, to preserve the fetus and provide the best conditions under which a fetus can develop into healthy calf, the cow must be relieved of its heat stress.

Achieving the present investigation's stated aims requires an analysis of heat stress as a heat and mass transfer problem. A pioneering work by Wiersma and Nelson (1967) investigated the effects produced by ventilating dairy barns. This investigation subsequently produced interest because the cooling equipment typically used in dairies began to cost more due to rising energy prices. Several ensuing projects involved experiments, simulations and proposed models aimed at solving heat-stress problems and reducing costs. These included studies by McGovern and Bruce (2000), Gebremedhin and Wu (2001), Gebremedhin and Wu (2002), Gebremedhin and Wu (2005), Jiang, et al. (2005), Maia, et al. (2005), Norton et al. (2007), Norton et al. (2009), Gebremedhin et al. (2008), Berman (2009), Berman (2010), Gebremedhin et al. (2010), Norton et al. (2010)

and Gomes da Silva et al. (2011). These efforts considered the cooling capacity of the equipment; a cow's physiological reaction once it is exposed to a particular climate; barn climate modeling; and how a cow exchanges heat with its surroundings.

A number of researchers have used Computational Fluid Dynamics (CFD) to investigate climate and a cow's corresponding heat fluxes. This tool has the capability to combine the effects of heat and mass transfer. Even more, it can include user defined functions that enable the user to represent specific phenomena, such as sweating. The work of Mondaca et al. (2012) is an example of this application. To include all the cooling mechanisms, a version of CFD was used that included user defined functions; thus, it was possible to study convection, evaporation, radiation and sweating simultaneously.

Although all of these cooling mechanisms produce a significant overall effect, if a cow lies down, the benefits gained will be reduced, and observations have shown that a typical cow will spend up to 50% each day lying down (Tolkamp et al., 2010). Furthermore, an investigation carried out by Bastian et al. (2003), found that the cooling to be gained via conduction can be as great as 19% of the total cooling experienced by a cow. In a subsequent investigation, Rojano et al. (2011) used CFD to demonstrate that the cooling achieved via conduction can actually be as high as 90% of the total cooling effect. Mondaca et al. (2012) produced an improved model that included sweating and humidity and was thus able to determine that the contribution made by the conductive cooling device can be as much as 70%. That study also analyzed a cow's interactions with its environment, taking into account the four paths by which heat could be transferred: convection, radiation, evaporation and conduction.

In accordance with our stated motivation for proposing a conductive cooling system, we considered the cooling requirements of dairy cows, and we used CFD to evaluate the proposed device. We also conducted a set of experiments to verify the CFD outcomes. Furthermore, this work introduces an alternative cooling method that addresses the key variables involved and provides an effective way to cope with heat stress.

Problem description

Cows will often rest for up to 12h during a day, and this is considered natural behavior. Depending on the freestall conditions and dairy management, a cow will have five or more resting occasions each day (Osterman and Redbo, 2001; Manninen et al., 2002; Veissier et al., 2004; Cook et al., 2008; Provolo and Riva, 2008; Tolkamp et al., 2010). We can expect cows with comfortable bedding material to spend more time resting, and although such time is constrained since cows should stand and move during the day, resting does provide the opportunity to operate a conductive cooling system by creating an interface between the portion of the cow's skin that is in contact with the bedding. Based on observations, about 20% of a resting cow's total skin surface will be in contact with the floor (Bastian et al. 2003) although that percent may at times be less because a cow does not always recline fully.

The proposed conductive cooling system is preferable for two reasons: it can reduce the threat of heat stress, and it will work in locations where evaporation is constrained due to prevalent high humidity. The proposed device was simulated and then tested using water as a coolant kept at constant temperature and flow rate. The heat fluxes found demonstrate the feasibility of the waterbed when sand is used as the bedding material.

Design of the conductive cooling system

The dimensions of the device are based on those of a typical cow weighing approximately 600 kg and having a total skin surface of approximately 5.6 m^2 (McGovern and Bruce, 2000). Given that 20% of the total skin will be exposed to the floor when the cow reclines, the system should be designed to cool an area of 1.12 m^2. Consequently, the device we designed is 80cm in wide and 140cm in length, and it is in agreement with the standard dimensions of a cow and a freestall, as indicated in ASABE Standards (2010). Since the skin surface contour of a cow does not have a rectangular shape once the cow reclines, we can expect the percentage of the animal's surface area in touch with the waterbed to be approximately 87.7% of the area of the specified rectangle (assuming a circular contour at the corners with a radius equal to 40cm).

Table 1 lists the materials used to construct the waterbed prototype as described. Thermal properties, such as the specific heat of the sand, are based on measured values obtained by Mampaey (1990), Papadikis et al. (2009), Shusaku and Matsubayashi (2009), Papadikis et al. (2009b) and Papadikis et al. (2010). Density was determined by measuring the sand used in the prototype. The sand's thermal conductivity was found by experimenting with the sand used in this investigation (because particle size, moisture content and composition affect thermal conductivity). Nevertheless, the thermal conductivity of the sand used matched outcomes obtained in previous investigations by Smits, et al. (2010) and Haigh, (2012). Thermal conductivity was determined using:

$$q = k\frac{\Delta T}{\Delta z} \tag{1}$$

where q is heat flux (in W/m$^{2)}$, ΔT is temperature difference between the two surfaces (in °C), and Δz is the distance between the two surfaces (in m). Equation 1 provided an estimate of the heat flow possible between the two surfaces. An average of five measurements found the thermal conductivity to be, respectively, 0.9, 2, 2.6 and 3.05 W/m-°C for 0.5, 5, 10 and 15% of the moisture content based on weight. Moisture content was found according to the standard ASTM D2216-10 (2010).

The dimensions and locations of the heat flux sensors and the thermocouples (including specifications and operating conditions) used to find the sand's thermal conductivity are described in the experimental setup section. The waterbed has three inlets on its left side, distanced 20cm apart, and an outlet at the center on its right side (see figure 1). The diameter of each inlet and the outlet pipe is 1.6cm. The material used

Table 1. Material parameters.

Variables		**Value**
Aluminum	Density (kg/m^3)	2719
	Specific heat (J/kg-°C)	871
	Thermal conductivity (W/m-°C)	202.4
Sand	Density (kg/m^3)	1668
	Specific heat (J/kg-°C)	835
Water	Density (kg/m^3)	998.2
	Specific heat (J/kg-°C)	4182
	Thermal conductivity (W/m-°C)	0.6
	Viscosity (kg/m-s)	0.001003

for the heat exchanger is aluminum alloy 5052 with a 0.2cm thick. We designed this prototype in such a way that it could demonstrate cooling capacity; we did not predicate the design on finding the optimal material to serve as the heat exchanger. Also, any minor change that may be required to reinforce the waterbed so as to enable it to adequately support a standing a cow must be considered in the subsequent works.

The waterbed's effectiveness as a heat exchanger was analyzed using CFD. Equation 2 provides numerical solutions and deals with the steady-state convection-conduction problem.

$$\frac{\partial u_j \phi}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_j}\right) + S_\phi \qquad (2)$$

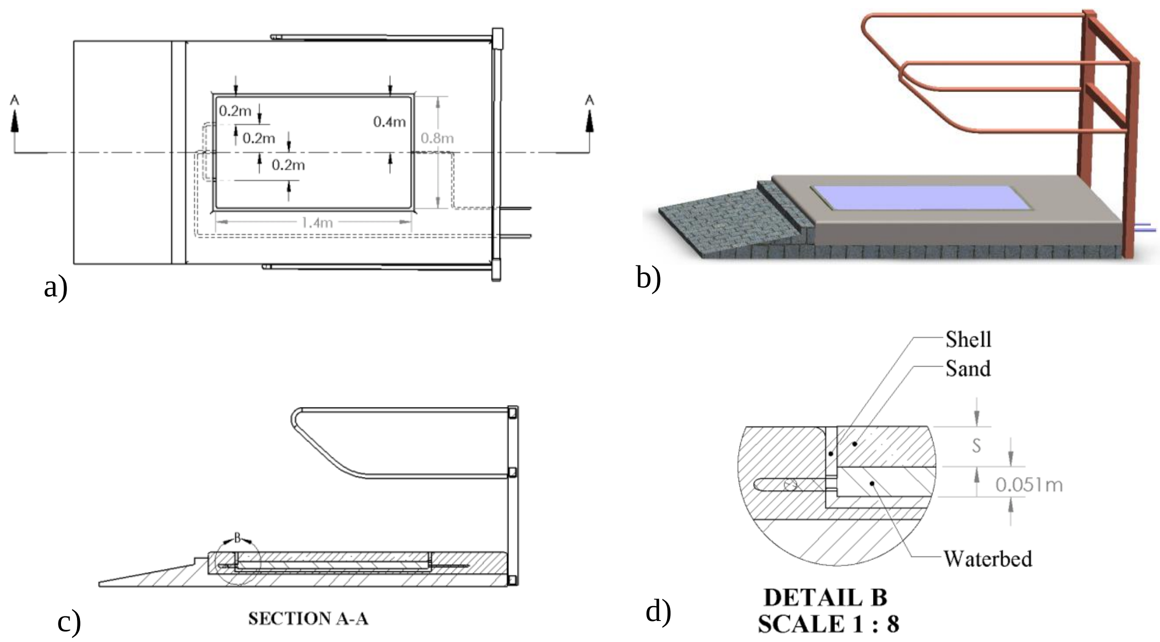

Figure 1. Waterbed specifications.

where x is the component for the u velocity in the direction j, ϕ is the variable of interest (either velocity (m/s) or temperature (°C)), Γ_{ϕ} is the diffusion coefficient, and S_{ϕ} is the source term (heat generation).

The sand used in this investigation was sieved in order to determine its particle size and then distributed following the standard set by ASTM D6913–04 (2009) (see figure 2). Based on these experiments, and according to the ASTM D2487-11 (2011) guidelines for soil classification, the type of sand used (considering a cumulative particle-size distribution curve) corresponds to the type referred to as "poorly graded sand" (SP).

Variables in the conductive cooling system

To determine the heat fluxes inherent to a conductive system, we assumed that there would be homogenous thermal properties, adiabatic conditions and fixed dimensions. The problem also involves a convection section with a laminar flow within

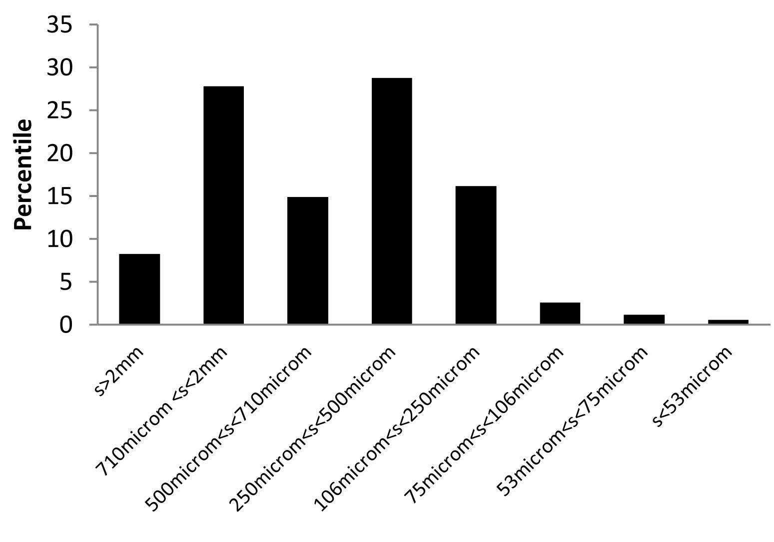

Figure 2. Sand particle size.

the heat exchanger and a conduction section for heat transfer within the bedding material. For both sections, the more important variables to consider when determining heat fluxes are the coolant water's temperature and its flow rate, the sand layer's thickness and its moisture content. How the sand layer affected heat fluxes was investigated in order to establish the most appropriate depth at which to place the waterbed. How the temperature of the coolant water and its flow rate affected heat fluxes was determined using simulations.

Numerical solution

For the evaluation of the waterbed and sand, we constructed a three-dimensional CFD model. A mesh for the geometry was generated as shown in figure 1. The waterbed geometry aspect ratio is ($L/h \cong 27$). To generate the mesh, we used the Meshing component of Ansys (2012) and the view A-A of figure 1 is shown in figure 3 with tetrahedral cells. Figure 3 shows a refinement at the interface between the fluid and the solid with the size growing at a rate of 1.05 for 20 and 25 layers for the fluid and solid sections, respectively; the section below the arrow corresponds to the waterbed and above the arrow to the bedding material section. The thickness of the first layer was set at 0.15cm. The mesh, corresponding to sand of 7.6cm, required approximately 900,000 cells with a skewness of 0.2 and maximum aspect ratio of 47.

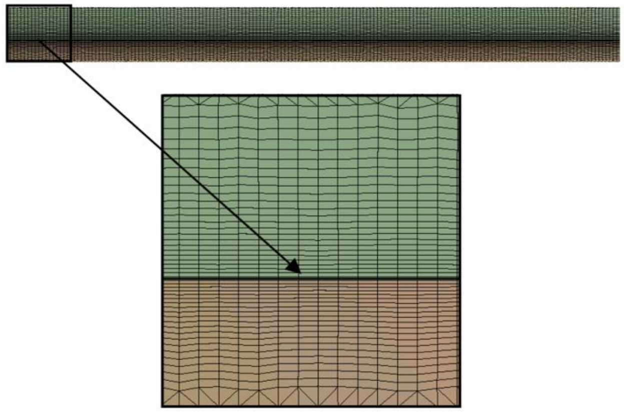

Figure 3. Waterbed and sand mesh structure for the A-A view of figure 1.

Water running through the waterbed had undeveloped laminar conditions (due to the geometry) and a low Reynolds number (between 39 and 272 for 1L/min and 7L/min, respectively). The side and bottom walls were adiabatic, and the interface between the waterbed and the sand was set with a combined wall thickness of 2mm according to the specifications for the aluminum sheet. We used a 3D solver (Fluent from Ansys) with a first-order upwind for momentum and energy. The pressure-velocity coupling algorithm follows the SIMPLE scheme to obtain the fluid dynamics field. The material properties used for simulations are mentioned in table 1. A solution was found at double precision in steady state conditions through an iterative solution. Energy balance after simulations obtained residuals on the order of 10^{-3}.

To simulate a cow's skin temperature (35°C), a constant temperature was maintained at the upper surface of the sand. The three inlets supplied water to the waterbed evenly, at constant temperature and flow rate.

Figure 4a shows the temperature contours of the CFD simulation of the sand-layer thickness of 7.6cm, the moisture content of 0.53%, and the water provided to the heat exchanger at 1L/min and 15°C. The jets of water at the inlets (see figure 4a) produce a known effect, and each jet's path afterward influences the heat fluxes contours (as described in figure 4b). Assuming that a cow will recline in the center of its stall, a higher efficiency was expected (since cooling capacity at the border was lower). However, a cow might lie down anywhere within the freestall, and that unpredictability makes it more difficult to suggest the best heat flux contours.

Model Sensitivity

Apart from the thickness of sand and its moisture content, the heat fluxes of the proposed waterbed can also change when a coolant is supplied at a different temperature (T_i), or supplied at a different flow rate. An evaluation—which considered as its baseline the following model characteristics: 5% of sand moisture content for a sand layer of 7.6cm (3in)—indicated how much heat flux can, on average, vary. The effects of supplying water at different temperatures and the flow rates can be coupled into a single equation (3) dealing with dimensionless temperature (θ).

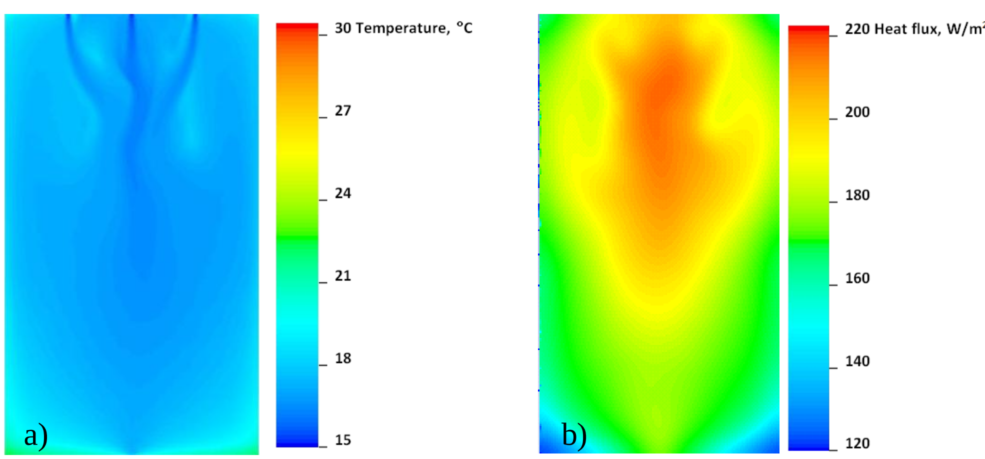

Figure 4. Top view of (a) temperature at the level of water inlets and (b) heat flux contours at the sand surface; case: 7.6cm (3in) sand thickness and 0.5% of moisture content.

$$\theta = \frac{T_b - T_i}{\frac{q'' D_h}{k_f}} \tag{3}$$

where T_b is the cow's skin temperature (in °C); T_i is the inlet water temperature at the heat exchanger (in °C); q'' is the heat flux between the heat exchanger and the sand layer (in W/m2); D_h is the hydraulic diameter of the heat exchanger, (in m); and k_f is water thermal conductivity (in W/m-°C). Heat fluxes can be computed using equation 3, with different flow rates (see figure 5).

Experimental validation

Experimental setup

The waterbed was placed within a carbon steel box 82.5cm in width, 142.5cm in length, and 18cm in height. The gap between the waterbed and box on each side was filled with foam, and the bottom was covered with foam 5.1cm thick. Additionally, the external side surfaces of the box were covered with foam 3.81cm thick. To retain the

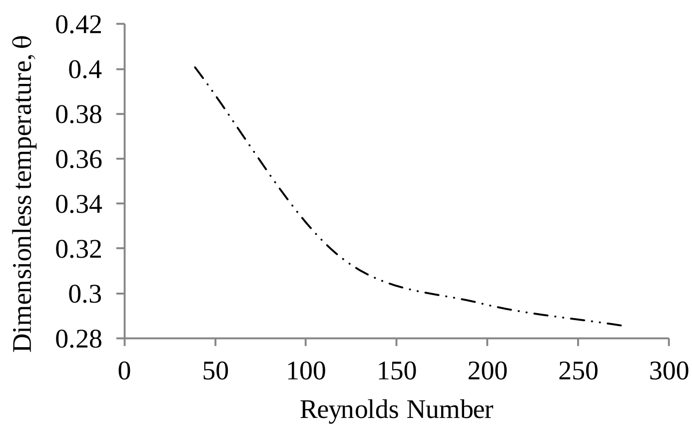

Figure 5. Dimensionless temperature under different Reynolds numbers; case: sand layer of 7.6cm (3in) and 5% of moisture content.

layer of sand, a carbon steel frame 80cm in width, 140cm in length and 16cm in height was placed on top of the waterbed. That frame was also covered with foam 5.1cm thick. The degree of insulation corresponds to 10R-value units that imply a heat-loss rate of 6.8W, computed based on a temperature difference between the environment and the waterbed of 9°C.

The assembly of the parts is illustrated in figure 6a, where views A-A, B-B and C-C are detailed sections of sand to indicate where sensors were placed to measure the sand's thermal conductivity. Section D-D, specified in figure 6c, is at the bottom of the upper tank, where the temperature representing the cow skin was monitored. Section E-E, shown in figure 6d (in conjunction with figure 6b) indicates the locations of the heat-flux sensors and thermocouples within the sand layer. Section F-F is the waterbed; figure 6e indicates the thermocouples, flow meters, pressure gauge and valves required.

Sensors, placement equipment and operation

Data coming from the heat-flux sensors and thermocouples was sent to a CR23X data logger, and from there data was transmitted to a computer. Data coming from the flow meters was sent to a CR3000 data logger, after which the experimental data was stored in a computer. Data generated by all the sensors was logged every two minutes.

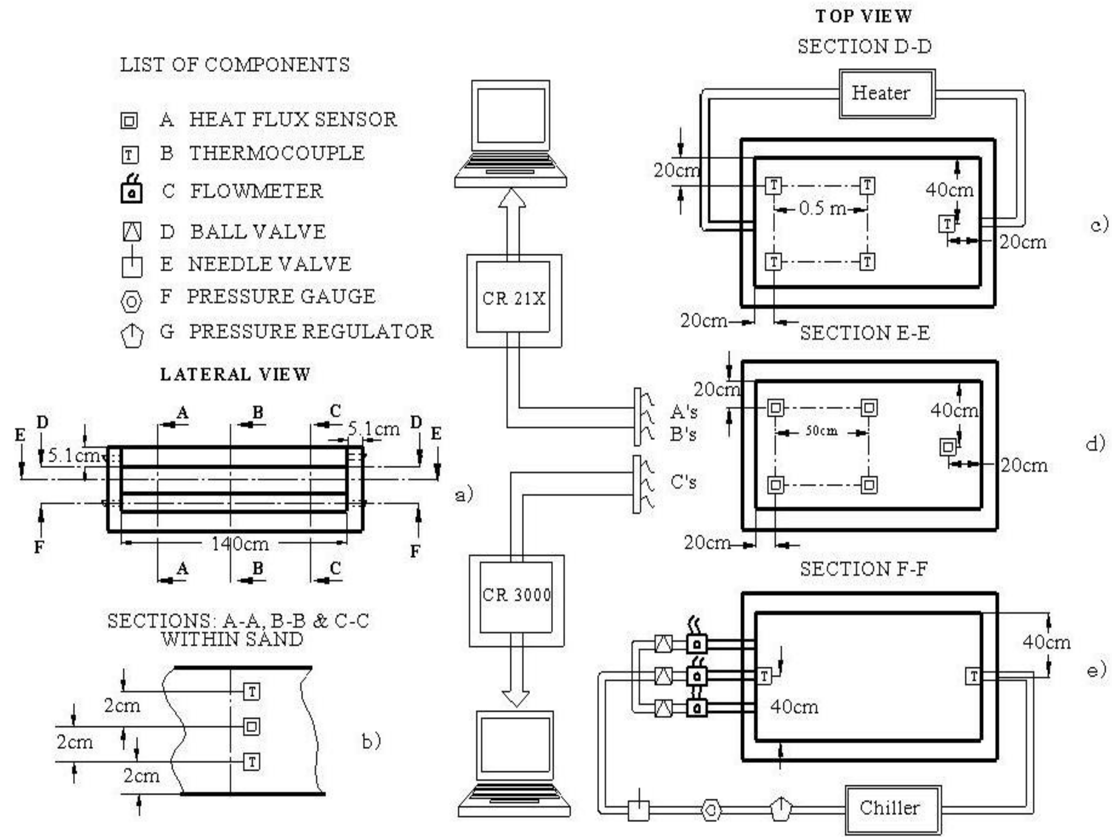

Figure 6. Experimental setup.

The water supplied to the waterbed came from a chiller set at 15°C, passing through a pressure valve regulator and a pressure gauge to assure a low enough pressure (less than 5psi). After that, the water ran to a needle valve that set a flow rate of 1L/min, and the pipe was split into three channels leading to inlets (see figure 6e). We placed a ball valve at each of the three inlets and then a flow meter to maintain an even flow-rate distribution. A FTB600B, ultra-low flow meter sensor from Omega Engineering, Inc., was used. One thermocouple type K was placed in the center of the waterbed inlet; another was placed at the waterbed's outlet in order to find water temperature increment.

Water exiting from the waterbed was sent back to the chiller. Details of the set up are shown in Section F-F. A flow rate of 1L/min will cause undeveloped laminar conditions due to periodic oscillations from flow instability.

Five thin-film heat-flux gauges (HFS-4 from Omega Engineering, Inc) were placed within the sand, and two thermocouples were placed on top and below the heat-flux sensor in order to measure the thermal conductivity of sand; locations are indicated in figures 6b and 6d. The five heat-flux gauges have the following characteristics: sensitivity is 1.8μV/W-m^2; thermal resistance is 0.004°C/Wm^2; the thermal capacitance is 1000W-sec/m^2-°C; and the time response is 0.7s. Thermocouples used were type K.

The cow's skin was simulated by setting the water temperature at the surface of the sand to 35°C. A plastic sheet 178μm thick, placed on top of the sand, retained the water to 5.1cm in depth. That water moved in recirculation to and from a heater at a rate of 6L/min in order to maintain a constant temperature. To verify the 35°C temperature and its homogeneity at the sand top surface, we placed five type K thermocouples; these locations are shown in figure 6c. We covered the thermocouples and heat-flux gauges with thermal grease (Omegatherm, from Omega Engineering, Inc.) in order to maintain consistent readings without air gaps. Omegatherm has a thermal resistance of 0.71W/m-°C.

Computational fluid dynamics and experiments

Additional numerical solutions were computed in order to include the effects of the sand's thermal conductivity. The 3D simulation considered three different cases (5, 10 and 15% moisture content), and figure 7 shows the coolant temperature gradients at the level of the inlets and also the corresponding heat-flux contours, in accordance with the features described in the numerical solution, which match those specified in the experimental setup (figures 1 and 6). The material's thermal properties were taken from table 1. A dry sand case (0.5% of moisture content) was modeled as described (see figure 4).

Heat flux values were taken at the five locations indicated in figure 6d. On the other hand, the experimental results required obtaining heat flux values from the same locations. Three repetitions were executed in each case, and the maximum and minimum

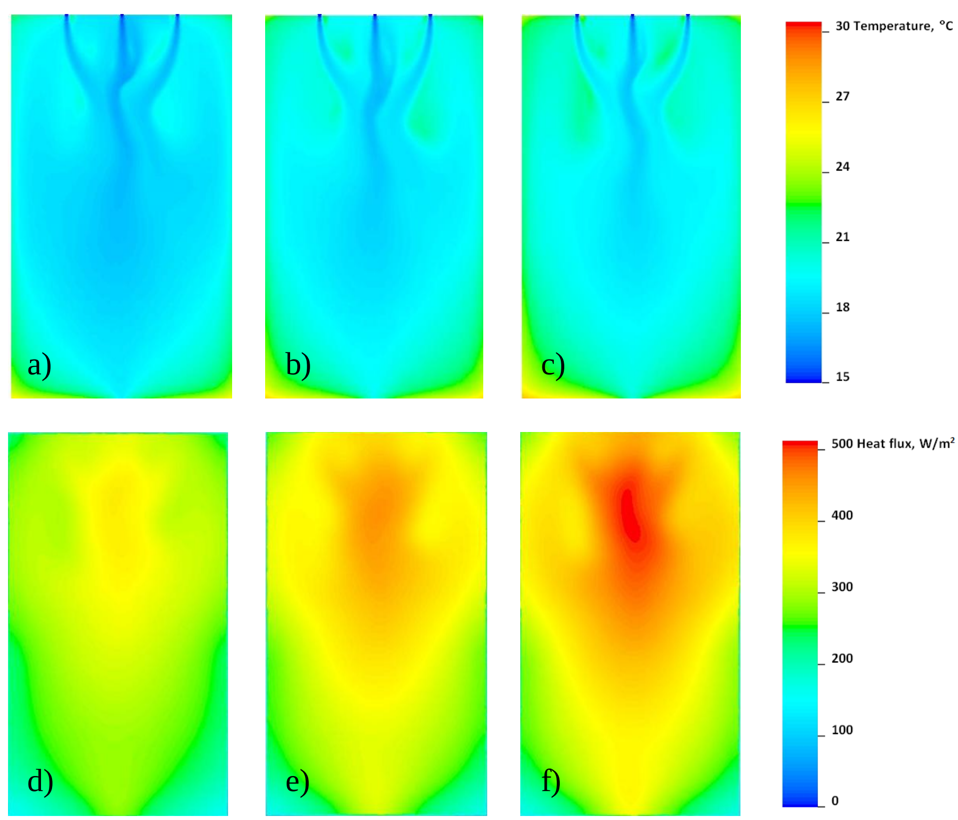

Figure 7. a) - c) Coolant temperature at the level of inlets, d) – f) heat flux contours at surface of sand, for 5, 10 and 15% of moisture content, respectively, in 3in of sand.

values are shown in figure 8 (including those of the CFD outcomes). Experimental data obtained under steady-state conditions were found when heat-flux sensors and thermocouples, placed in the bedding material (figure 6b and 6d), stabilized the measurements.

We chose the five locations specified in figure 6d in order to achieve a convergence between the average in heat flux for the entire surface of sand and the values obtained at the five locations using data obtained from CFD. The variation among the five locations was found to be less than 6% when compared to the average of the heat flux over the entire surface.

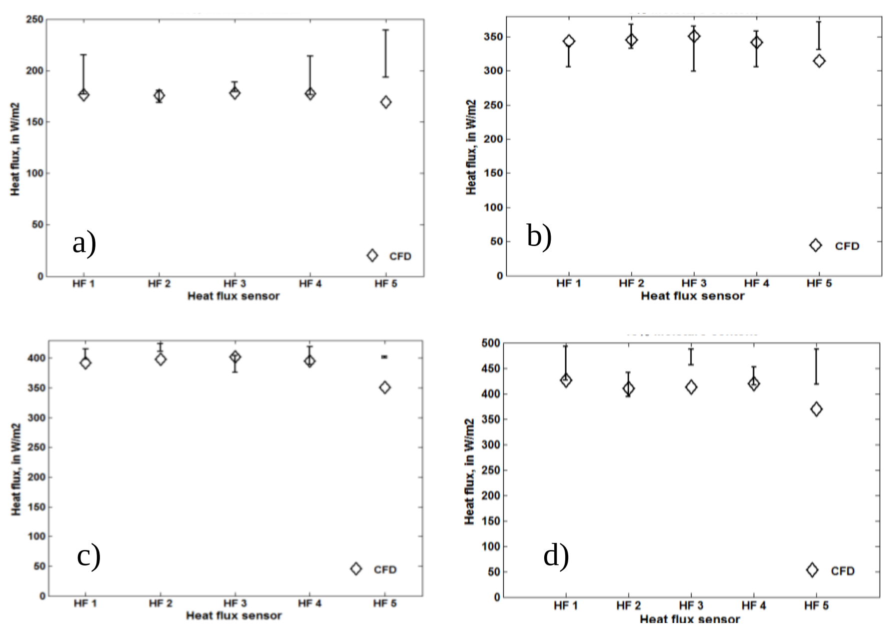

Figure 8. Experiments vs. CFD; a) – d) for moisture contents of 0.5, 5, 10 and 15% in 3in of sand.

The heat-flux sensor HF5's measurements were higher than the values obtained from CFD, perhaps because of the difficulty inherent to representing, in CFD, the mixture of the three water jets under undeveloped laminar conditions. All the data obtained from the experiments was compared with data from the simulations (see figure 9), whenever there was an R^2-value of 0.9144.

We also compared the CFD and experimental values with respect to changes in the water's temperature once it reached the outlet of the waterbed. Figure 10 takes into account four cases in which the outcomes from CFD and the experiments match. Water temperature at the outlet allowed us to calculate energy balance, finding an error of less than 10%.

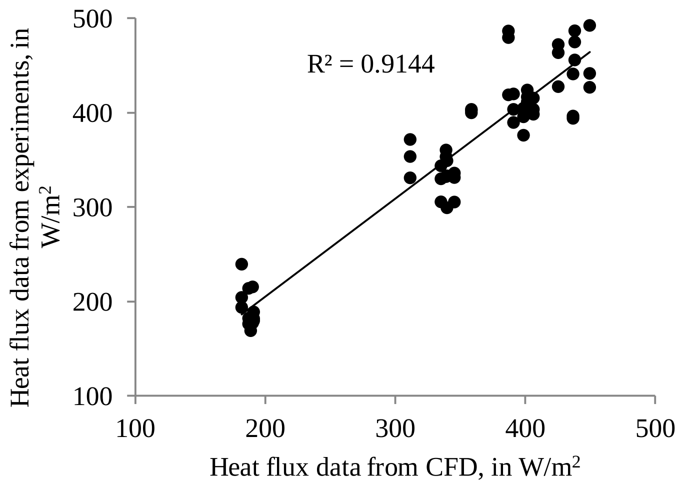

Figure 9. CFD vs. Experimental heat-flux data.

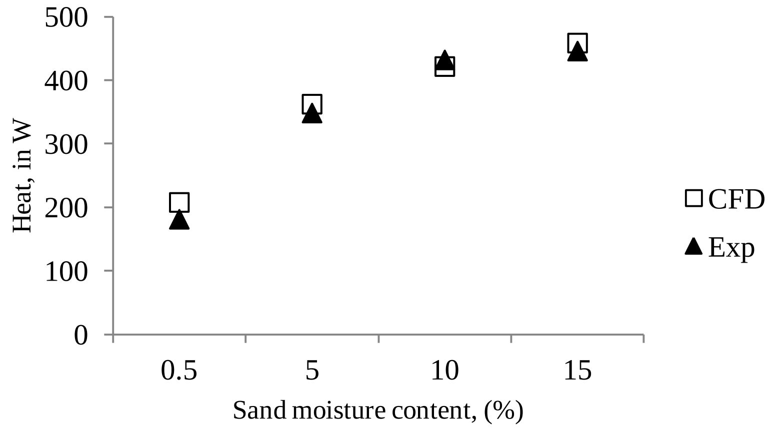

Figure 10. Heat removed by cooling water according to different sand moisture contents.

Simulation of representative scenarios

An objective of this work was to evaluate the performance of the waterbed under a variety of possible scenarios. Given that the experiments and CFD outcomes were deemed satisfactory, the CFD model could be used as a baseline. This section explores the effects produced by the thickness of the sand layer and its thermal conductivity. We especially wanted to know if the heat transfer changed according to the depth at which the exchanger was placed under the waterbed. We assumed that a thicker layer of bedding material would provide better comfort for a cow but would also reduce the system's cooling capacity and, too, that a thinner layer, while not as comfortable, would drastically increase cooling capacity.

For this investigation, the CFD model was modified by varying the thickness of the sand layer. We considered three different thicknesses: 2.54cm (1in), 15.24cm (6in) and 22.86cm (9in). These changes required creating a new mesh for each case, each according to the specifications used to create the original mesh of 7.6cm (3in). Only the number of layers in the solid section varied (12, 36 and 44 for 1in, 6in and 9in, respectively). The meshes obtained 0.013, 0.17 and 0.12 of skewness for the 1in, 6in and 9in, respectively. The maximum aspect ratio was 37.7 (370,000 cells), 65 (~1,500,000 cells) and 62 (~1,900,000 cells), for 1in, 6in and 9in, respectively. To obtain a numerical solution in each case, the CFD solver also followed the same conditions mentioned before, with respect to the 3in case.

Dimensionless temperature variation due to sand-layer thickness and sand thermal conductivity (K_s), under the relation K_s/K_f, is represented in figure 11. Within the ranges considered, the effects of maximum sand thermal conductivity will be more important when the thickness of sand is reduced. In the other hand, as the sand layer becomes thicker, thermal conductivity due to moisture content will have less influence on heat fluxes.

Outcomes obtained are valid for sand-layer thicknesses from 2.5cm to 22.8cm and moisture content levels in the sand from 0.5% to 15%. Figure 12 summarizes the CFD outcomes related to the water's temperature increment once it passed through the waterbed. Clearly, the thickness of the sand layer and its moisture content both affect water temperature. The energy balance of this thermal system implies that higher amounts of heat are related to a higher water temperature increment. Such information is beneficial when designing the appropriate pipe network for a set of waterbeds connected in a serial way.

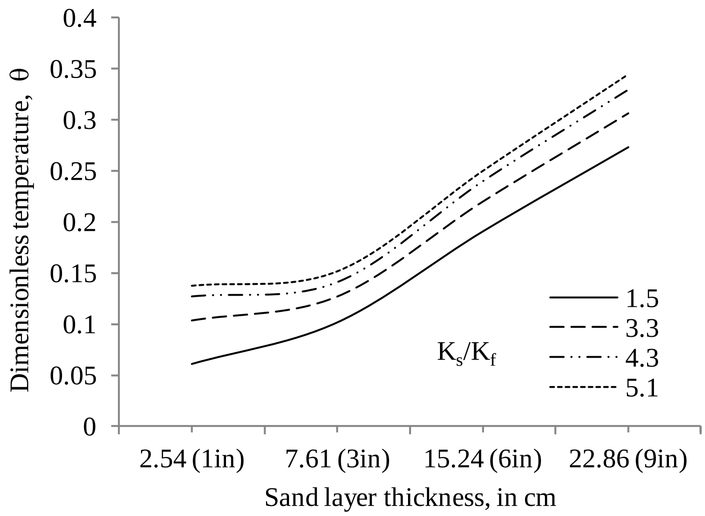

Figure 11. Effects of sand-layer thickness and moisture content in 3D-CFD heat-flux outcomes.

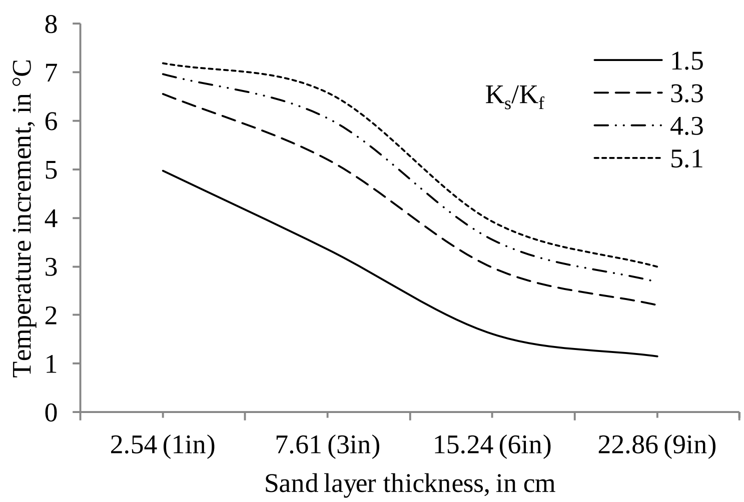

Figure 12. Temperature increase in the coolant for different sand-layer thicknesses and thermal conductivity in 3D-CFD outcomes.

Discussion

Experimental measurements help to validate the reliability and accuracy of the computational models. This work sought to demonstrate a good agreement between the heat-flux results we obtained by experimentation and our computational simulation outcomes of the heat fluxes occurring between a waterbed and the top surface of a layer of sand when it was kept at a constant temperature.

We also had to determine the thermal conductivity of sand (since particle size, moisture content and composition all combine to determine that values). Given the heat fluxes and the thermocouple measurements, it was possible to find an average thermal conductivity using measurements taken at five locations within the boundaries of the surface under study. Earlier works provided values for the thermal conductivity of generic sand, but the study's particular sand's thermal conductivity values could not be strictly compared to the sand used in previous investigations because particle size and composition were most likely different. We also determined the effects that the moisture content would have on thermal conductivity. Low flow rates of 1L/min served as reference for the model, combined with different moisture contents in a sand layer of 7.6cm (3in). Once that is validated, we could create different scenarios in order to obtain reliable outcomes without the need of experimental verification. With respect to the simulated waterbed, the undeveloped laminar conditions in the geometry proposed caused periodic oscillations in flow, and these should be further investigated. However, such oscillations did not affect convergence, and sufficient accuracy was attained for the energy balance of each simulation. Consequently, our research allows us to claim that the heat flux values obtained in CFD simulations can be used to find optimal heat exchanger designs under steady state conditions. The divergence that occurs between the outcomes of the simulations and the results of experimentation are the product of the imperfect insulation in side and bottom walls. We found, however (based on the specified dimensions of the foam used), that the total heat transfer at the insulated walls amounted no more than 5% of the total heat transfer. Thus the effects of this imperfect insulation were deemed negligible. However, further study could include either thermocouples or heat flux gauges to estimate the heat lost through each sidewall and the bottom of the proposed shell.

Water inflow was set as constant at each of the inlets with an even flow rate distribution, thus helping to increase thermal efficiency. Further work may find that an improved manifold leading with the water entrance and exit would increase the efficiency of the heat transfer due to higher water mobility within the waterbed. With these modifications, the volume of water in the waterbed could be reduced, given that within the waterbed there are stagnant zones that become part of the thermal gradient instead of transporting heat to the outlet; however, reducing the thickness of the waterbed and including an optimized manifold in order to achieve a more efficient device could create an optimization problem relating the pressure loss and the amount of material required for the waterbed fabrication.

Another factor affecting thermal conductivity and heat fluxes is pressure. Sand under pressure will have a different thermal conductivity value. This work does not consider the effects of pressure, but the only weight on top of the sand came from the layer of water, which was 5.1cm deep. Further studies accounting for the effects of pressure on thermal conductivity would likely be able to quantify the variation in thermal conductivity.

Conclusions

This investigation explored the cooling capacity of a device that should help to decrease heat stress in dairy cows. Previous investigations of heat flux considered the design and thermal analysis required (Bastian, et al., 2003, Rojano, et al., 2011 and Mondaca, et al., 2012). However, since few studies investigated this particular cooling mechanism, the proposed setup is a conceptual approach for developing a device that would help to cool cows by conduction. Accurate calculations and a corroborated model using CFD helped to determine the cooling capacity and its range of operation.

It goes without saying that experimentation and CFD simulation are key to confirming what the heat exchange rates would be, once particular conditions are set for water temperature, moisture content and sand layer thickness. All these variables combined can be handled in CFD, and various scenarios can be modeled to reduce the amount of experimentation needed; the experimentation that was undertaken as part of this work was sufficient to demonstrate that, apart from matching the CFD outcomes, a waterbed such as the one proposed can perform adequately as a heat exchanger. Earlier works (Bastian, et al., 2003, Rojano, et al., 2011 and Mondaca, et al., 2012) contend that heat fluxes should fall between 100 and 200 W/m^2. In this work, those heat fluxes were achieved. Nonetheless, heat fluxes can also change depending on the manner in which the heat exchanger is installed and operated and also depending on several other external factors that are particular to any given dairy.

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APPENDIX B

DEVELOPMENT AND EVALUATION OF A WATER NETWORK AND HEAT TRANSFER MODEL FOR DAIRY SYSTEMS

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Abstract

A water supply system connecting a series of heat exchangers installed in a barn used to shelter dairy cows could serve as a means of cooling the cows. Much like current equipment currently used to control most indoor environments, this system could both remove and supply heat. Convection and conduction phenomena occurring along the pipe network are primarily considered in this work, which uses water as a means of carrying thermal energy. A hydraulic model created by EPANET simulates the supply system's pipe network. A water supply system example illustrates, in the first place, a feasible hydraulic model; in the second place, it computes the variations in water temperature. The system operates within a feasible range of flow rates (1 LPM < Q < 7 LPM) and velocities (0.05 m/s < v < 2 m/s). This investigation uses the heat and mass transfer analogy to model heat transfer rates and the corresponding temperature distributions throughout the pipe network. To estimate water temperature, the Water Quality Solver module of EPANET (which follows a mass balance model) is modified into an energy balance model, which acts upon the hydraulic model. Then, it is possible to evaluate efficiency of the pipe network based on pumping power and cooling/heating capacity.

Keywords: cooling, heating, distributed systems, water network, heat and mass transfer.

Introduction

Fluids are often used to transport and to distribute heat and thus control a climate or maintain constant heat fluxes in a system or at a specific location. A Heating, Ventilation, and Air Conditioning (HVAC) system, for instance, relies on the thermophysical properties of air and thereby controls the climate in an enclosed or semi-closed space. Many engines, such as automobile motors and the power sources used in refrigerators and air conditioners are cooled by circulating a fluid around the heat-generating component. Typically, water or oil are used because they have a higher heat capacity and are inexpensive and readily available. Because all of these applications require a short time response, using a large-scale distributed system is challenging. Yet some cooling problems require a distributed system with a defined section for transporting heat from one location to another. Water is most often used because it is so inexpensive and easy to obtain; however, in contrast to air, which can be applied directly to a heat generator, water may require a device, such as a heat exchanger (HE), to transfer heat via conduction or thermal radiation so that the fluid will not come into direct contact with the material from which the heat is removed.

Heating and cooling systems that work by transporting heat have been used to solve a number of specific problems both large and small. Phetteplace (1995) studied a large-scale heat distribution system and recommended that such system may lose significantly heat as fluid circulated through the system. Bobenhausen (1994), Castro et al. (2000), Sugarman (2000) and Chandan (2010) all present results from studies of small-scale HVAC systems that use water to transport heat. Other study cases dealing especially with a distributed cooling/ heating system that uses water have been done by Maguire et al. (2011), Jinhai et al. (2008), Flores et al. (2010), Ponce-Ortega et al. (2010), Cortinovis et al. (2009), Sanaye et al. (2012) and Picon-Nunez et al. (2011). These studies took into account issues such as heat loss, pressure drop, system capacity and system optimization. In special cases, the phenomena involved in heat transfer create a complexity that is difficult to analyze; in such cases, heat transfer rates can be determined using alternative methods such as Artificial Neural Networks (ANNs). Hosoz et al. (2007), Barteczko-Hibbert et al. (2009) and Malinowski et al. (2011) used ANNs to include all the phenomena associated with a heat-transfer system. However, obtaining generic solutions for systems that involve complex pipe network configurations still requires reliable and accurate estimations.

To solve these types of problem addressed in this research, we used EPANET, a software program developed by the Environmental Protection Agency, to simulate the pipe network and consider the heat-transfer rates, The EPANET version we used had been modified so that it could calculate heat-transfer rates. Given that every application involves a particular pipe network design and operation, this work was aimed at setting a foundation for a generalized solution, one that could be used to produce a viable hydraulic model capable of analyzing energy transfer throughout the network as well.

Problem description

We wanted the present investigation to achieve outcomes that would allow us to suggest a methodology for handling heat transfer in water distribution systems. In order to do this, we split the entire system in two sections: the devices assigned for heat transfer

(HEs) and the Water Supply System (WSS). We used EPANET to analyze the WSS as a hydraulic model in order to determine the most feasible characteristics of the pipe network and the best operating conditions. In addition, we employed a module of EPANET (the Water Quality Solver, or WQS) to modify the mass balance model and turn it into an energy balance model. This enabled us to compute heat transfer rates.

EPANET has been used to simulate the WSS that transport water used for drinking, fire protection, and irrigation, as well as other applications. Male and Walski (1990), Walski (1992), Rossman (2000), Mays (2000), and Walski et al. (2001), used the program to compute the hydraulic parameters that determine the characteristics of a system's key components such as the pipes, the various fittings, the reservoirs, the water sources, the valves and the pumps. The accurate findings obtained in these studies has aided other researchers investigating other issues such as chlorine decay, water age and water discoloration. Among such efforts are those of Rossman (2000), Mays (2000), Rossman et al. (1994), Geldreich et al. (1996), Volk et al. (1999), Rodriguez and Serodes (2001), Boxall and Saul (2005), DiChristo and Leopardi (2008), Zechman (2011) and Husband and Boxall (2011). In addition to the applications mentioned, it is also possible to use EPANET to calculate heat transfer rates.

Through a comprehensive estimation of the heat-transfer rates occurring throughout an entire system, a simulation can deal with rates happening in any pipe at any time. Outcomes obtained can be coupled with rates happening within each HE. As stated, whether the system is gaining or losing heat, the system's operator must consider the heat lost and the energy required to run the system's pumps. Parameters such as system size, pipe network layout, flow rate demanded by each HE, the spatial distribution of HEs and the cooling needs would set the system's hydraulic performance. Determining thermal performance requires the knowledge of additional parameters of the system, such as pipe material, insulation and thermal boundary conditions.

Inherent to a distributed system, the distances that water must travel affects the energy required for pumping and also the cooling/heating time response in each HE. For applications not constrained by time response, an investigation can be focused on estimating the total heat lost and the pumping power needed. However, a cooling/heating system with a long-distance distribution system must attend to variables that minimize energy losses.

The pipe network layout, the size of the system, the number of HEs, the flow rate demanded by each HE, and such pipe characteristics as length and diameter will determine the pumping power for the WSS. EPANET will indicate the feasible range of operation for the hydraulic model under study. Once the WSS is set, then the system can be used to compute heat-transfer rates.

A WSS will have a range of operation determined by the hydraulic parameters that combined with the thermal parameters of the pipe network's characteristics (such as pipe material, inside pipe wall roughness, the thermal conductivity of the material involved, and temperature at the boundaries) will determine the thermal performance. The set of solutions found can be evaluated to determine what hydraulic operating parameters are convenient plus the thermal parameters that are related to the pipe and insulation materials used to minimize heat loss.

Figure 1 shows the way to proceed if a particular pipe network has been proposed and is able to supply water to each HE. In the first place, it is necessary to set the pipe network layout and its characteristics in order to find the hydraulic parameters within the achievable range of operation. This data determines the network's pumping requirements and the parameters for heat-transfer rates. In the second place, it is also necessary to include the settings for the pipe's thermal boundary conditions that, combined with the dimensionless parameters for heat transfer, will compute changes in water temperature and, consequently, the system's cooling capacity.

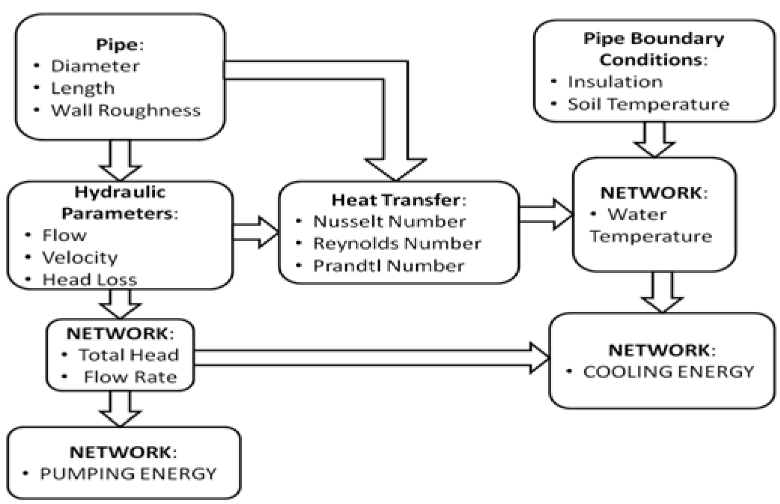

Figure 1. Heat transfer in hydraulic models.

Since the system will serve as a cooling/heating system, the thermal properties of water must be defined within the liquid phase. Then, the values for the thermo-physical properties of water, such as viscosity, density, and the Prandtl number, can be considered fixed, which means heat-transfer estimations will be accurate within a range of temperature.

Hydraulic model

The layout of the pipe network determines the hydraulic model. For the parameters such as flow rates, velocities and pressure drops, which are determined for each pipe, EPANET requires data pertaining to pipe characteristics, including overall length, diameter, elevation, flow-rate demand and pipe wall roughness. Then, it is necessary to verify that those values simulate a hydraulic model within a feasible range of operation. One recommendation for water distribution systems (AWWA, 2004) indicates that water velocities should be below 3m/s. Additionally, to make sure that water will run through the entire loop, EPANET computes the total head and flow rate needed. The simulation must include information regarding local minor losses based on what types of junctions are used; the coefficients required can be taken from Rossman, 2000. For these applications, the flow rate is defined by the HE demand. If either a constant or a variable flow is needed, the flow rate can be set by a control valve installed within the pipe network.

For this study, the layout followed was split into three sections: 1) pipes delivering water to HEs; 2) elements representing the HEs, and 3) pipes returning water to the original point. The delivering and returning sections may follow a similar path. The HEs connects both sections via pipes that are equivalent in length and water volume.

Heat transfer solver

Heat-transfer rates can be considered as either rates that impose a constant heat flux or rates that impose a constant temperature at the external pipe wall. Both cases assume a uniformity of conditions in a single pipe. For this work, the approach followed is based on constant temperature.

To quantify energy flow changes, the heat transfer solver runs according to the hydraulic model. Heat transfer rates, primarily estimated by analytical equations describing convection and conduction, are going to be governed by how the system is operated, the pipe and insulation thermal properties, flow regime, thermal boundary layer, and the temperatures involved.

In the convection section, the water flow regime may follow either laminar, transitional or turbulent conditions, each of which influences heat-transfer rates differently. The conduction section, which is determined by thermal conductivity, depends on the materials used for the pipes and the insulation. Both sections with their corresponding temperatures are confined to an inside pipe wall with a hydraulic and thermal boundary determining heat-transfer rates, taking into account the roughness of inside pipe wall.

The energy carried by water running through the pipe is described by the advection equation (1) finding simultaneously heat-transfer rates as the water moves and exchanges heat with the pipe wall. Water movement is simulated by the hydraulic model already included in EPANET, and it also finds the parameters required for thermal analysis (such as flow rates and velocities). A combination of the hydraulic and thermal parameters will define thermal performance.

Since many applications imply setting non-uniform thermal boundary conditions, the heat-transfer solver is capable of dealing with particular settings in each pipe and indicating the outside pipe-wall temperature and insulation. The thickness of the pipe-wall and insulation (along with their corresponding thermal conductivity values) are key to defining the thermal boundary conditions, and these parameters are included in the conduction section. A constant temperature can be assumed on the external walls of pipes or insulation exposed to convection. A summary of the parameters needed is shown in figure 2.

Advection

The advection equation (1) replicates how water moves along a pipe. Essentially, we adopted the Lagrangian approach in order to estimate temperature variation.

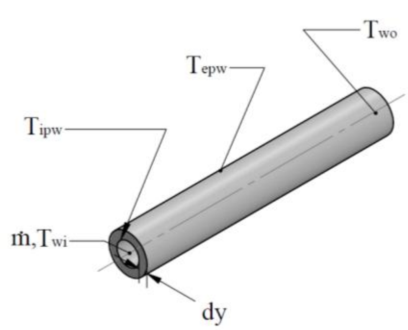

Figure 2. Variables involved in heat exchange associated with a pipe.

$$\frac{\partial T_i}{\partial t} = -u_i \frac{\partial T_i}{\partial x} + S_i \qquad (1)$$

where T_i is the temperature (in °C) in pipe i, t is time (in s), x is the location (in m) within the pipe, u_i is the axial velocity (in m/s) in pipe i, and S_i is the temperature change per second (in °C) due to wall heat transfer at the pipe i.

Temperature at junctions

Pipe junctions cause flows to mix and affect the corresponding water's temperature. At any junction, some pipes supply water and other pipes conduct water on into the rest of the network. Equation 2 calculates temperature at any junction, assuming there is perfect and instantaneous mixing.

$$T_{i|x=0} = \frac{\sum_{j \epsilon I_k} Q_j T_{j|x=L_j}}{\sum_{j \epsilon I_k} Q_j} \qquad (2)$$

where i denotes the pipe leaving the junction k, I_k is the set of pipes that supply flow into junction k, L_j is the length of pipe j and Q_j is the flow (in m^3/s) in pipe j. The notation $T_{i|x=0}$ represents the temperature at the beginning of pipe i (in °C), while $T_{j|x=L_j}$ is the temperature at the end of the pipe.

Constant temperature approach

A practical formulation that relates to real conditions happening at the thermal boundary of each pipe involves setting a constant temperature for the external pipe wall insulator. Given that the insulation is imperfect, a constant temperature will affect the interior wall temperature, which in turn will affect the water temperature; knowing this makes it possible to determine the difference between the water temperature at the beginning of each pipe and the temperature at the end (equation 3).

$$\delta T_i = (T_{i,wi} - T_{i,wo}) \tag{3}$$

where $T_{i,wi}$ is the water temperature (in °C) at the beginning of the pipe, and $T_{i,wo}$ is the water temperature (in °C) at the end of the pipe i. Equation 4 finds the temperature at the end of pipe, $T_{i,wo}$

$$(T_{i,ipw} - T_{i,wo}) = (T_{i,ipw} - T_{i,wi})e^{\left(-\frac{P_i L_i h_i}{\dot{m}_i Cp}\right)} \tag{4}$$

where $T_{i,pw}$ is the pipe-wall interior temperature (in °C), P is the pipe perimeter (in m), L is the pipe length (in m), h is the heat transfer coefficient (in W/m^2-°C), $\dot{m}_i$ is the flow for the pipe i (in kg/s), and Cp is the specific heat (for water is 4182 J/kg-°C). The Gnielinski equation (Gnielinski, 1976) is used to calculate the heat-transfer coefficient (h) with a water flow regime under transitional or turbulent conditions:

$$h_i = \left(\frac{k_f}{D_i}\right)\frac{(f_i/8)(Re_i - 1000)Pr}{1 + 12.7(f_i/8)^{1/2}(Pr^{2/3} - 1)} \tag{5}$$

where k_f is the water thermal conductivity (in W/m-°C), Re_i is the Reynolds number for pipe i, Pr is the Prandtl number (equal to 7.01), and f_i is the friction factor. According to Gnielinski (1976), heat transfer (equation 5) estimations are acceptable under the transitional and turbulent conditions, with the following conditions: $0.6 \leq \mathrm{Pr} \leq 2000$ and $2300 \leq \mathrm{Re} \leq 10^6$. However, error increases within a transitional flow regime, and it becomes higher once it is closed to laminar flow regimes.

The friction factor f_i can be found by the Colebrook-White equation:

$$\frac{1}{\sqrt{f_i}} = -2.0 * log\left[\frac{e/D_i}{3.7} + \frac{2.51}{Re_i\sqrt{f_i}}\right] \tag{6a}$$

where e is the roughness height (in m), and D_i is the diameter of the pipe i (in m). This equation applies to turbulent conditions (Re > 4000). f_i in flow regimes within laminar conditions can be computed using equation 6b (Re <2300):

$$f_i = \frac{64}{Re} \tag{6b}$$

Equations 6a and 6b cover laminar and turbulent conditions; however, the equation for finding f_i for transitional flow regimes is not well defined; EPANET finds f_i by using a cubic interpolation taken from the Moody Diagram (Rossman, 2000). That interpolation requires a Reynolds number (equation 7):

$$\mathrm{Re} = \frac{\rho v D}{\mu} \tag{7}$$

where ρ is water density (1000 kg/m^3), v is water velocity (in m/s), D is the pipe diameter (in m), and μ is water viscosity (0.00103 Pa·s, viscosity is taken as constant for 20 °C).

With respect to heat transfer, if the flow regime is within laminar conditions, then the heat-transfer coefficient can be calculated using equation 8:

$$h_i = \frac{3.66 * K_f}{D_i} \tag{8}$$

Because the pipes are partially insulated, the external wall temperature will cause a heat exchange with a rate defined by the pipe wall's thermal properties. The internal pipe-wall temperature, $T_{i,ipw}$, can be found by solving for the equilibrium between convection and conduction:

$$q_{cond} + q_{conv} = 0 \tag{9}$$

where q_{cond} is heat by conduction (in W) :

$$q_{cond} = \frac{k_p A_i}{dy_i}\left(T_{i,epw} - T_{i,ipw}\right) \tag{10}$$

where k_p is thermal conductivity of the pipe (in W/m-°C), A_i is the pipe-wall surface area (in m^2), dy_i is the pipe-wall thickness (in m), T_{pwe} is external pipe-wall temperature (in °C), and q_{conv} is the heat flow by convection (in W):

$$q_{conv} = \dot{m}Cp\left[\left(T_{i,ipw} - T_{i,wi}\right) - \left(T_{i,ipw} - T_{i,wo}\right)\right] \tag{11}$$

where $\dot{m}$ is the mass flow (in kg/s). Substituting equations 10 and 11 into the equation 9, we found $\left(T_{i,ipw} - T_{i,wo}\right)$. Then, it can be substituted into equation 4 to solve it for $T_{i,ipw}$ and then from the same equation to find the water temperature at the end of the pipe ($T_{i,wo}$).

Heat exchanger

Depending on its application, an HE can be configured in a variety of ways. It can be compact (Shah et al.,2001); it can be in the form of a shell and/or a tube (Lui et al., 2000); it can be a microchannel (Steinke et al., 2005); or it can be simply a rectangular duct (Haji-Sheikh and Beck, 2008; Rojano, et al., 2012). Alternatively, heat transfer can also be determined using equations 1 through 11 if the geometry is similar to a pipe.

Regardless of their application, the characteristics of the HEs used must be adapted to the WSS, especially by including information about the heat fluxes associated with water temperature and flow rates. Also pressure drop is important when determining pumping demand, since various designs not only enhance heat transfer rates but also cause an incremental pressure drop. The volume of water in the pipe representing the HE must be equal to that in the HE in order to maintain mass and energy balance.

Exemplary problem

Defining the pipe network

The following example involves a system with a set of 1000 HEs uniformly distributed across a floor area of 8,600 m^2. The design has two parts: a) a set of 20 clusters each one of them contained 50 HEs (see figure 3), and b) the main line (see figure 4), which provided water to each one of the 20 clusters. The total surface on which each cluster was located has 38m in length and 5.3m in width. Each cluster was arranged in two lines of 25 devices spaced 2.5m apart and rows 1.5m apart. Figure 3 shows the pipe network used, and figure 3b describes its dimensions and components. The HEs in that cluster were connected in parallel and then in a serial configuration; a repeated grouping of eight interconnected HEs is shown in figure 3b, where a set of four were connected serially on one side and another set of four were in parallel on the opposite side. At the end of each set was a flow control valve for setting the flow rate required. The path delivering water was thus similar to the path returning it to the original point. Since this system was used for cooling, water running through the fourth HE had to meet the specified cooling requirements.

Figure 4 shows the main line that distributed water to each of the 20 clusters; dimensions and pipe connections are described. Similar to a cluster, there is a parallel path to return water to its origin, a water reservoir located next to the system, which supplied and received water to and from the main line.

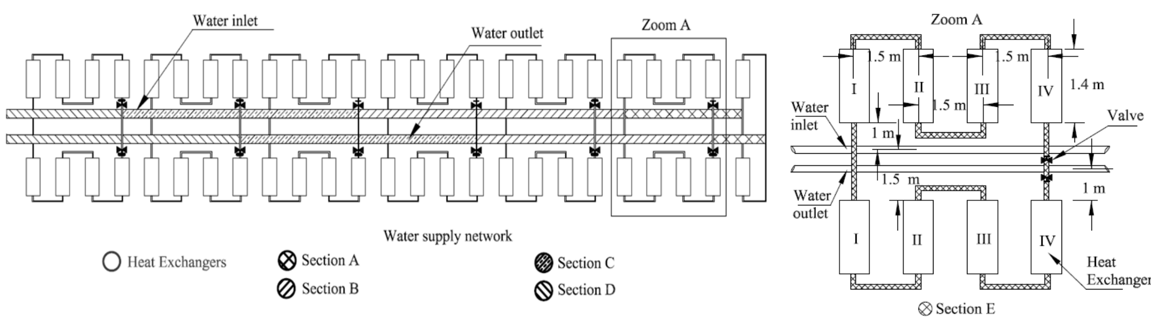

Figure 3. Pipe network for a cluster of 50 heat exchangers.

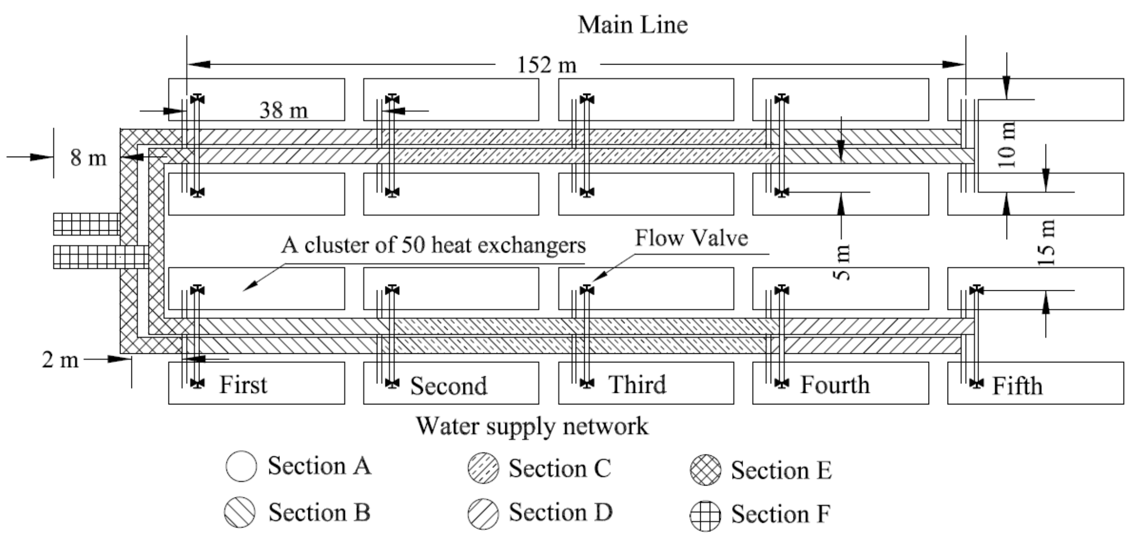

Figure 4. Main line for 20 clusters.

Clusters and the main-line pipe specifications are listed on table 3; the diameters indicated correspond to velocities between 0.05m/s and 2m/s covering the flow rates demanded in each HE, which ranged from 1LPM to 7LPM. Pipe-wall thermal conductivity equaled 0.19 W/m-°C. A pipe-wall roughness of 0.5×10^{-5}m was used to compute local heat losses based on the Darcy-Weisbach equation (Rossman, 2000). Energy losses in each pipe and minor local losses occurring at junctions were included in order to find the total amount of energy required to pump a supply of water through the whole loop. The feasible range of operation will involve a range of pumping needs.

The pipes in the setup shown in figures 3 and 4 were assumed to have a constant external pipe-wall temperature of 26.7°C, (corresponding to the case of the maximum monthly temperature for soil at a depth of 20in in Tucson, AZ, according to AZMet, 2012). In order to show that this approach could deal with either heating or cooling systems, the initial water temperature was set at 15°C. Then a section of the network transported water of a temperature that was lower than the temperature of the external pipe wall, thus causing a heat gain. Once the water temperature became higher than the temperature of the external pipe wall, the heat exchange reversed.

For this simulation, the HE performance was added and governed by equations 1 to 11. Also, it was assumed that all the HEs were rectangular ducts of a length equal to 1.4m, a width equal to 0.8m, and a height equal to 0.051m. Parameters were changed: dy was set equal to 0.02m, K_p set equal to 2W/m-°C, S set equal to 1.12m^2, and P set equal to 0.096m. In equation 7, since flow in each HE was within the laminar flow regime, the equation coefficient of 3.66 was changed to 4.86 in order to calculate the heat-transfer coefficient. It was assumed that surface S would be able to maintain a constant temperature $T_{i,epw}$ equal to 35°C. Pressure drops occurring at the HE were equivalent to those that would be experienced by a pipe dealing with a water volume equal to 57L.

Table 3. Pipe specifications.

Pipe	Diameter, in mm	Pipe wall thickness, in mm
Cluster:		
Section A	15.8	2.8
Section B	21	2.9
Section C	27	3.4
Section D	35	3.6
Section E	15.8	2.8
Main Line:		
Section A	52.1	3.9
Section B	78	5.5
Section C	102.3	6
Section D	154	7.1
Section E	254.5	9.3
Section F	303.2	10.3

Results

Outcomes obtained from the heat-transfer solver are included in this work rather than the outcomes obtained from the hydraulic model provided by EPANET (i.e. water velocity, flow rates and pressure drop in each pipe). The heat transfer rates computed should help to illustrate temperature variations over the entire system (see figure 5) and also variations among the clusters located in the fifth column (see figure 6). The critical cluster at the end of the main line represents the maximum temperature, or in this case, the minimum cooling capacity. The exemplary problem reaches a maximum water

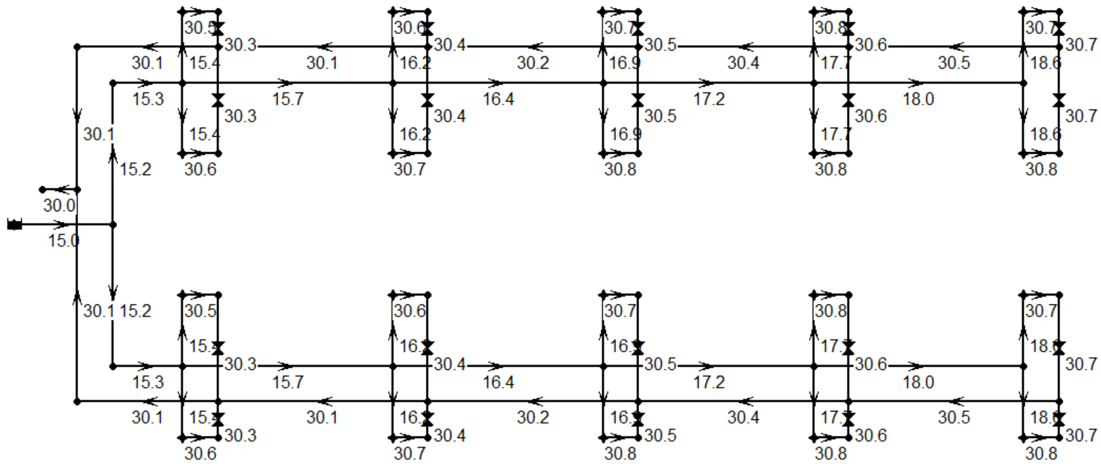

Figure 5. Temperature variation for a flow rate of 1LPM in the main line water supply.

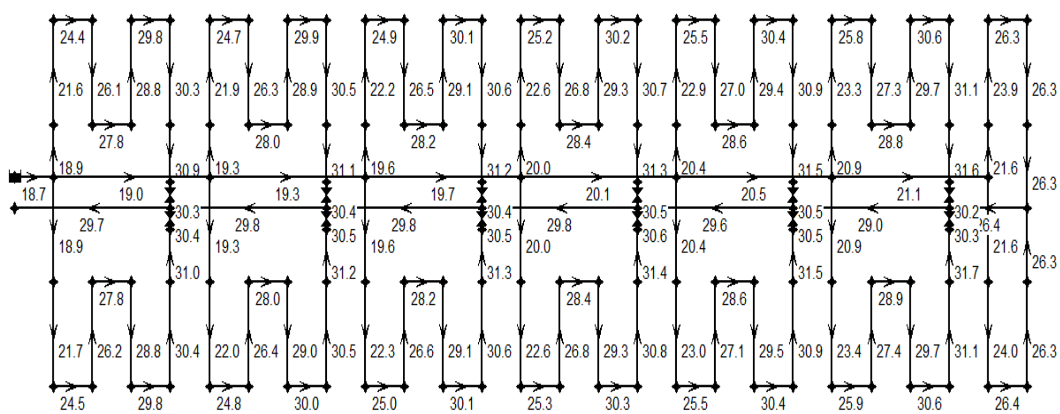

Figure 6. Temperature variation for a flow rate of 1LPM in a cluster.

temperature before the water returns to its point of origin ($T_{water} > T_{ipw}$), causing a reverse effect in heat fluxes.

Even though temperature changes indicate thermal performance, the heat exchange happening in each HE may vary due to its location within the cluster. For the critical clusters located at the end of the main line, the heat transfer average is computed based on the location of the HE: type I, II, III and IV connected in serial way (see figure 3b) once the water supplied changes from 1LPM to 5LPM. The same group is repeated for the entire cluster (figure 3a), except for the last two HEs, which are type I. Water temperature entering these clusters is at 18.7°C (as shown in figure 5) and heat flow variations due to HE location are shown in figure 7. With a flow rate of 1 LPM, a relatively low cooling capacity was found at the HE type IV, representing about 20% of the maximum found in HE type I. However, heat transfer became more uniform under flow rates of 3LPM and 5LPM and also achieved a reduction of approximately 50% between the maximum and the minimum amount of cooling. According to the example provided, higher flow rates will improve the cooling uniformity even though higher flow rates require more energy for pumping. Cooling capacity ranges must be observed since several applications demand specific limits of heat flow.

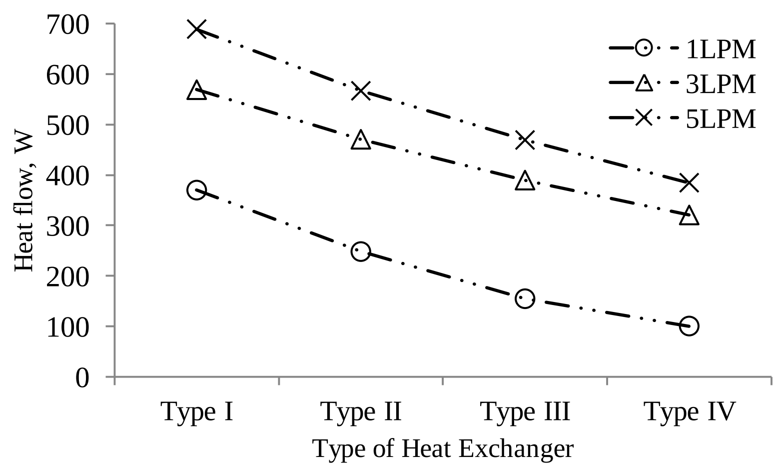

Figure 7. Cooling capacity average achieved by heat exchangers based on order: first (I), second (II), third (III) and fourth (IV) place in a series connection.

Previous outcomes help to find the lowest and highest water temperature, implying the maximal and minimal cooling capacity possible under the effects of water flow rates supplied to the HE. We assumed that all the HEs would receive the same flow rate but water of a different temperature because each HE was in a different location. Thus, flow rate not only affected cooling capacity but also the amount of energy required for pumping. Figure 8 shows the outcomes associated with the energy rates required to maintain a flow-rate range from 1LPM to 7LPM. Pumping needs can be found by using the equation taken from Mays (2000) with the efficiency of the pump set equal to 1; cooling capacity refers to the performance of the HEs, and total heat lost can be computed by measuring the heat transfer that occurs during water transportation.

Given all the energy expenses required to operate the system, the average amount of heat lost in any cluster and also throughout the entire system (see figure 9) becomes the main concern. The system's efficiency diminished when the main line suffered significant heat loss (when water was only distributed to the 20 clusters). And when

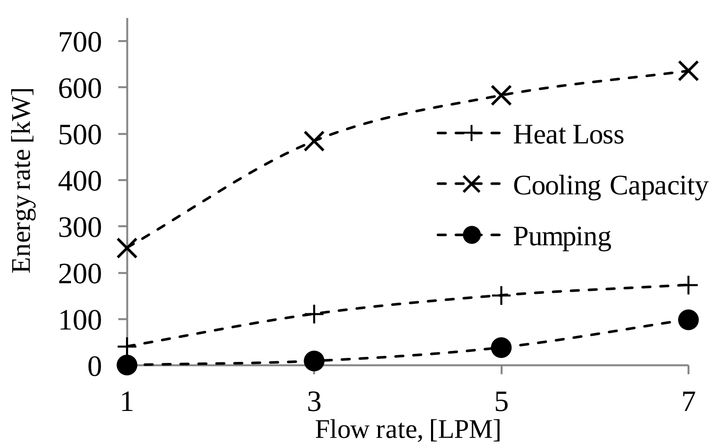

Figure 8. Energy required for the system operation.

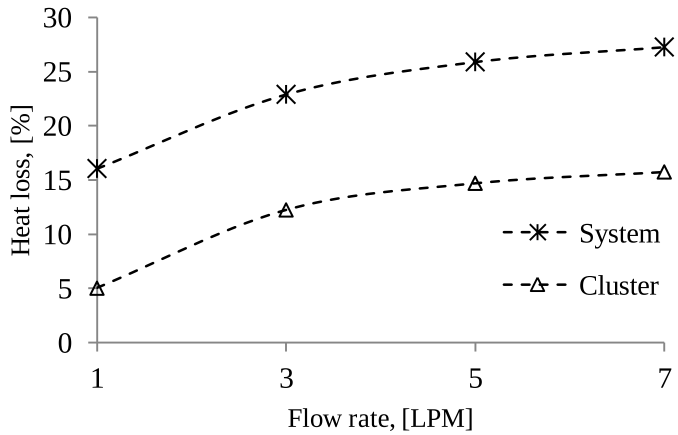

Figure 9. Heat loss estimated in clusters and in the entire system.

water must be transported long distances, the amount of insulation will determine how efficient the system can be. Previous outcomes correspond to those obtained using un-insulated pipes; however, the system's performance could be enhanced if the thermal resistance of the pipes were increased.

A summary of efficiency is shown in figure 10. The energy rate required for pumping, heat loss and the cooling capacity of the system determines the overall efficiency, following equation 12. Our findings indicate that during low-flow rates the system's efficiency could be incrementally improved even though low-flow rates decrease cooling capacity significantly; this was found to be the case when the flow rate was reduced from 7LPM to 1LPM.

$$E = 100 * \frac{\text{Heat loss} + \text{Pumping}}{\text{Heat loss} + \text{Pumping} + \text{Cooling capacity}} \tag{12}$$

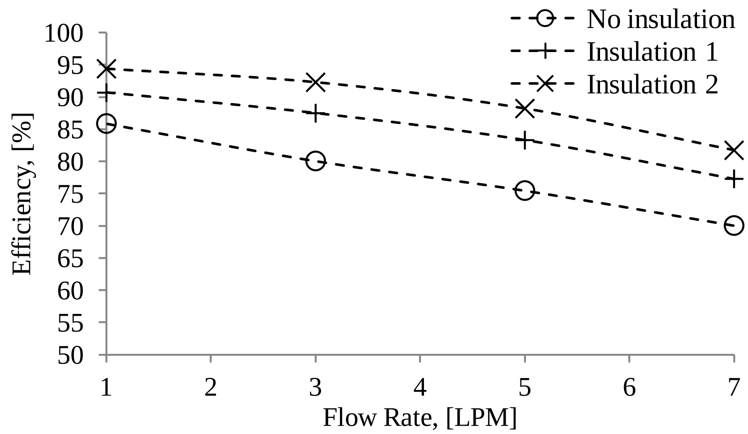

Figure 10. Insulation to reduce heat loss: 'No insulation' corresponds to pipe-wall thermal conductivity, 'Insulation 1' and 'Insulation 2' corresponds to reductions of 50% and 75%, respectively, in the thermal conductivity at the pipe wall.

Insulation should improve the system's efficiency. Figure 10 shows the effects of thermal conductivity in terms of the system's overall efficiency. Thermal conductivity for Insulations 1 and 2 are 0.095W/m-°C and 0.0475W/m-°C, respectively; and efficiency is increased by 7% and 12%, respectively, for a flow rate of 7LPM. When the flow rate is 1LPM, efficiency increases by 5% and 8.4% for Insulations 1 and 2, respectively. These results clearly show that insulation will significantly enhance the system's efficiency, and it should therefore be considered when the external pipe-wall temperature or pipe lengths are significantly affecting the thermal performance.

Conclusions

A pipe network layout that meets hydraulic requirements is the baseline for determining the energy needed to efficiently operate a system. Such a hydraulic model can accurately simulate the transportation of water and the heat exchange occurring as the water moves through the system, regardless of whether the system is used for cooling or heating. Combined hydraulic and thermophysical parameters enable the model system to estimate the key operational issues such as pumping needs and heat-transfer performance and to determine the efficiency of the network. EPANET can find the required hydraulic parameters for the setup and simulate the pipe network. In addition, the heat transfer solver incorporated into EPANET can facilitate an analysis that will determine the cooling/heating capacity and efficiency of the system.

According to the findings of this investigation, a low flow rate will reduce cooling/heating capacity but increase efficiency; therefore, any similar design should be operated at a flow rate such that the HEs in the system can achieve the targeted cooling/heating requirements and maximize energy savings. This criterion applies in cases where time response is not a constraint.

Efficiency diminishes when the water must travel long distances, even if the system is adequately insulated; thus, the size of a system depends on how much energy can be lost without impairing the system's operation. Further investigation could help to determine more precisely the consequences of a system size and to what extent insulation can help to reduce heat leaks and assure that water will be supplied to the HEs at the desired temperature. The system may need modifications in order to provide adequate answers to these considerations.

Our investigation did not consider fouling; however, it should be taken into account for subsequent works focused on assessing lifespan and maintenance. The operation of a WSS used for cooling or for heating involves different rates of corrosion, which can affect the lifespan of a system. In addition, external moisture may affect the insulation's effectiveness. When the system is used to transport hot water, external moisture will not enter the insulation surrounding the pipe, but when the system transports cold water, and as the water temperature reaches the dew point, moisture will accumulate. Future studies should examine such conditions and search for the most convenient actions that can be taken to solve the associated problems.

When HEs are placed at different elevations within a WSS, they will affect the energy rates required for pumping; furthermore, a pressure drop in any HE can significantly affect pumping demands. Because different applications will require networks of significantly different size or layout, and some systems may include insulation or involve different system's components, such as HEs and valves, outcomes obtained by studying other applications might vary significantly from those obtained from this work. However, the methodology presented herein should still apply since the governing equations used to determine heat-transfer values are generic. Additional modifications will have to be made to the EPANET software when it is used for applications that do not involve circular pipes and/or water.

Both the size and the layout of a pipe network affect system's efficiency in conjunction with the arrangement of the HEs (be they connected serially or in parallel); A parallel connection produces a more uniform cooling effect than does a serial arrangement. However, HEs laid out in a series require less water, fewer flow control valves, fewer fittings and less piping. Consequently, it is convenient to increase the number of HEs connected serially.

Regardless of the network layout, adjusting flow valves at different flow rates within the system can also serve to maintain uniform cooling; then the HEs at the end of the cluster can be expected to have flow rates that are higher than those closer to the water source. Nonetheless, differentiated cooling within the system can also be accomplished by adjusting flow rates as desired; the number of valves in the system and the flow-rate range will determine the ability to adjust cooling.

In spite of adequate design and operation, a WSS could fail due to leaks, depleted insulation or thermal boundary conditions, any of which can cause significant heat fluxes. However, a preventive maintenance program can help for a satisfactory system performance.

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