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Design and testing of a liquid cooled garment for hot environments

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ABSTRACT

Liquid cooled garments (LCGs) are considered a viable method to protect individuals from hyperthermia and heat-related illness when working in thermally stressful environments. While the concept of LCGs was proposed over 50 years ago, the design and testing of these systems is undeveloped and stands in need of further study. In this study, a detailed heat transfer model of LCG in a hot environment was built to analyze the effects of different factors on the LCG performance, and to identify the main limitations to achieve maximum performance. An LCG prototype was designed and fabricated. Series of tests were done by a modified thermal manikin method to validate the heat transfer model and to evaluate the thermal properties. Both experimental and predicted results show that the heat flux components match the heat balance equation with an error of less than 10% at different flowrate. Thermal resistance analysis also manifests that the thermal resistance between the cooling water and the ambient (R_2) is more sensitive to the flowrate than to the one between the skin surface and the cooling water (R_1). When the flowrate increased from 225 to 544 mL/min, R_2 decreased from 0.5 to 0.3 °C m^2/W while R_1 almost remained constant. A specific duration time was proposed to assess the durability and an optimized value of 1.68 h/kg was found according to the heat transfer model. The present heat transfer model and specific duration time concept could be used to optimize and evaluate this kind of LCG respectively.

1. Introduction

A stable core body temperature is essential for maintaining optimal functions of the human body. However, when working in extreme hot and stressful environments, it is often arduous to achieve this without extra cooling (Angelo, 2009). Examples of such extreme situations can be found in many working places, such as mines, foundries, construction sites, boiler factories and inside military vehicles. In these environments, excessive heat can lead to failure of the human thermoregulatory system, resulting in heat-related illness such as heat cramps, heat syncope, heat exhaustion and heat stroke (Koppe et al., 2004; Altman et al., 2012).

To reduce the health risk caused by these hazard heat environments, microclimate cooling technologies have been developed to enhance heat exchange between the human body and the environment. Cooling garments have been proved to be one of the most promising technologies. Since the first prototype proposed for crewman by Burton and Collier at the Royal Aircraft Establishment in 1960s (Burton and Collier, 1964), liquid cooled garments (LCG) have been widely adopted as an effective cooling technology with advantages of higher cooling efficiency, reliability and adjustable capacity, compared to other microclimate cooling techniques, such as air cooling and phase change cooling (Nunneley, 1970).

Prototypes of LCG have been built for a variety of applications. McLellan and Selkirk (2006) gave detailed description of an LCG used for firefighters to manage the heat stress. 70% of the cooling was achieved within the first 10 min, even though the prototype could not last for more than 20 min in practical usage. Beenakker et al. (2001) used LCGs to treat multiple sclerosis patients. Experimental results showed that the active cooling garment did relieve fatigue and improve muscle strength and postural stability. LCG technology is crucial in the field of aerospace medicine (Webbon et al., 1981; White and Roth, 1979; Chambers, 1970; Nunneley et al., 1971; Light and Norman, 1980). LCGs are still the main cooling technology to remove heat stress for astronauts (Miller et al., 2011; Nyberg et al., 2001; Perez et al., 2003; Qiu et al., 2001; Campbell et al., 1998). However, it is notable that the specific design of LCG for spacesuit is often not practical in other fields.

Based on the review of the abovementioned researches, we conclude that the design of LCG is still immature and most prototypes are made based on experimental experience, even though the concept of LCG was proposed half a century ago. The design of an LCG involves principles of physiology, bio-medical, engineering and ergonomics. It is a challenge to design a compact and portable LCG which has high cooling efficiency to maintain the thermal

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comfort while minimizing the cost of coolant and power requirements. Since many factors need to be considered in the design, a theoretical model that takes all of the factors into account to analyze the performance of LCG is desirable.

In this paper, a detailed theoretical model of heat transfer from the human skin to the environment through the LCG was built. A prototype of a water cooled garment that was optimized based on the model was fabricated. A series of performance tests were conducted using a modified thermal manikin method to validate the heat transfer model. It was found that the errors between the experimental results and the calculated results were less than 10%. A maximum work duration time of 3.36 h for the LCG was obtained at 45 °C ambient temperature when the flowrate was 224.5 mL/min and the max cooling rate was 243.2 W/m^2.

2. Principles of LCG

Fig. 1 shows the schematic of a liquid cooled garment system. The basic working mechanism of this LCG is that the cold liquid, driven by the micropump, flows through the tubing network embedded in basic garment and takes away the heat. After dissipating the extra heat produced by the human body or from the hazard heat environment, the cold liquid turns warm and is then circulated to the liquid reservoir and re-cooled by the ice pack. The process repeats until the ice pack melts and the whole liquid system becomes warm. The basic garment is usually made of cotton fabrics with high elasticity, thus the tubes can firmly contact with the human body.

3. Thermal analysis and system design

To maintain a stable body temperature, the heat loss needs to balance the heat production and the heat gain. Otherwise, the heat content of the body will change, which will cause core body temperature fluctuating. Metabolic heat produced inside the body either provides the energy for working or ends as heat. The heat storage can be written as (Havenith, 2002)

$$Q_s = (Q_m - W) - (Q_{conv} + Q_{cond} + Q_r + Q_{eva} + Q_{res}) \tag{1}$$

where Q_s is the heat storage of the body, Q_m is the metabolic heat production, W is the mechanical work power, Q_{conv} is the heat

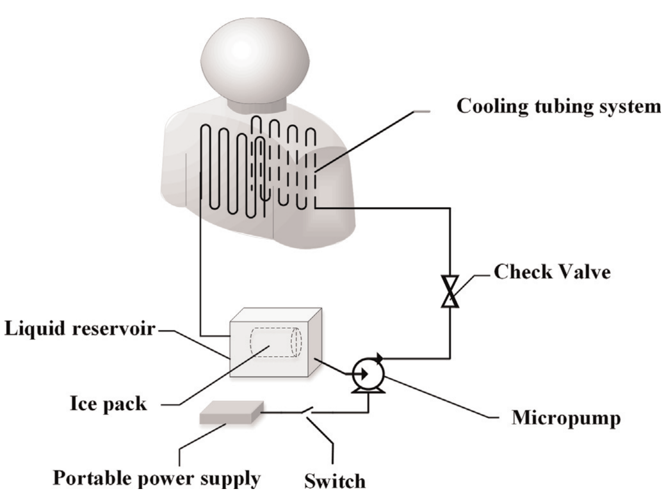

Fig. 1. The schematic of a liquid cooling garment system. The micropump drives the cold water flowing through the tubing system to exchange heat with human body. The water then goes back to the liquid reservoir and gets chilled by the icepack, and that cycle repeats. Switch and valve are used to control the system.

loss due to convection, Q_{cond} is the heat loss due to conduction, Q_r is the heat loss due to radiation, Q_{eva} is the heat loss due to sweat evaporation, and Q_{res} is the heat loss due to respiration.

It should be noted that the heat loss components in Eq. (1) can be neglected, which means that human body actually gains heat. For instance, when the ambient temperature is higher than the skin, the radiation heat flux Q_r from the environment actually enters the human body. This formula can be modified to apply to the situation where the LCG is used. The effective mechanical work power is small and can be neglected in most cases. When wearing an LCG, the clothes is tightly close to the skin, acting as a barrier for the sweat evaporation. Moreover, the sweat production can be decreased or even stopped due to the cooling effect of LCG on the body. In most of the cases, the heat loss by respiration and conduction would be less than 5% and 1%, respectively. Therefore, this part of heat loss is also negligible for engineering analysis (Koppe et al., 2004). Based on the analysis above, following hypotheses were proposed to simplify the heat transfer model:

1. The heat transfer is steady-state.
2. Heat transfer occurring in the cooling garment and on the skin are only along the normal direction.
3. The cooling garment and the human skin are treated as homogeneous plate.
4. The effect of the air layer trapped between the LCG and the human skin is neglected, and the temperature of skin is regarded as the same with the temperature of the inner side of the basic garment.
5. The effects of sweat evaporation and respiration are ignored.
6. The velocity of the ambient air is zero, meaning that the type of convection is natural convection.

Thus the heat balance equation can be written as

$$Q_w = Q_{conv} + Q_r + Q_m \tag{2}$$

where Q_w is the total heat taken away by the cooling water of the LCG. Based on the assumptions that the whole human skin and the cooling garment are homogeneous and the heat transfer is steady-state, the heat flux form of Eq. (2) can be given as

$$q_w = q_{conv} + q_r + q_m \tag{3}$$

Fig. 2 shows the heat transfer processes that occurring among the human skin, LCG and the ambient environment in hot environments. The heat flux q_w, which indicates the heat removal rate of LCG, is calculated from the measurement of inlet and outlet temperature as well as the flow rate:

$$q_w = \dot{m}c_p(T_{out} - T_{in})/A_{cl} \tag{4}$$

where $\dot{m}$ is the mass flow rate of the circulating system in kg/s, c_p is the specific heat capacity of water (4.2 kJ/(kg K)), T_{out} and T_{in} are the outlet and inlet temperature of the tubing network, respectively, A_{cl} is the effective cooling area of the LCG. R_{conv} and R_r are the convective and radiative thermal resistance between the LCG and the ambient environment. The thermal resistance between the human skin and the cooling garment R_s can be calculated as follow:

$$R_s = \frac{T_s - T_{cl}}{q_m} \tag{5}$$

where T_s and T_{cl} are the temperature of the human skin and the clothes, respectively.

The heat flux of the heat loss by the convection q_{conv} is given by

$$q_{conv} = h_c f_{cl}(T_{cl} - T_a) \tag{6}$$

where T_a is the ambient temperature; f_{cl} is the clothing area factor,

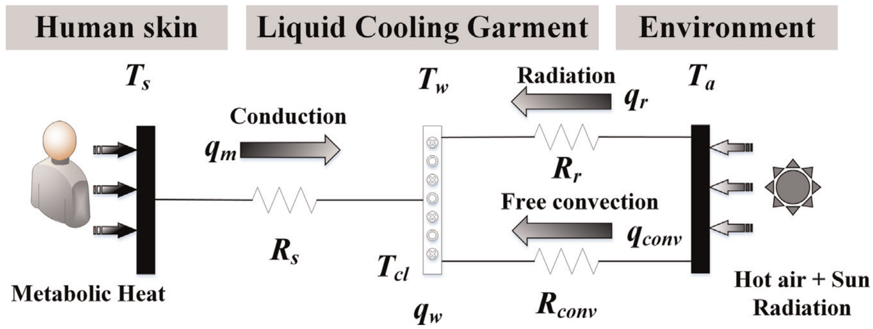

Fig. 2. Heat transfer model of human-liquid cooling garment-environment in hot environment. The ambient temperature T_a and the skin temperature T_s are both higher than the cooling water temperature T_w. The metabolic heat of human body q_m flows to the LCG against the thermal resistance R_s. And the radiant heat q_r and convective heat q_{conv} from the environment also flow to the LCG, overcoming the thermal resistance R_r and R_{conv} separately. T_{cl} is the clothes surface temperature.

representing the ratio of the surface area of the clothed body to the surface area of the nude body, and it can be given by (Holmer et al., 1999)

$$f_{cl} = 1.00 + 1.97R_{cl} \tag{7}$$

where R_{cl} is the basic heat exchange resistance, which can be measured with the hot plate method suggested by The American Society for Testing and Materials (ASTM F1868-14, 2014; ASTM D1518-11a, 2011).

The natural convection heat transfer coefficient h_c [W/(m^2 °C)] increases with the rise of the temperature difference between the human skin and the ambient. The relation can be given by (Nielsen and Pedersen, 1952)

$$h_c = 2.38(T_{cl} - T_a)^{0.25} \tag{8}$$

The heat gain by radiation from the hot ambient environment is expressed as

$$q_r = \varepsilon\sigma f_{eff}\left[(T_{cl} + 273)^4 - (T_{mrt} + 273)^4\right] \tag{9}$$

where ε is the emissivity of the clothes, σ is the Stefan–Boltzmann constant of 5.67×10^{-8} W/(m^2 K^4), f_{eff} is the effective radiation area coefficient, and T_{mrt} is the mean radiant temperature of the ambient environment. Considering that the temperature range of the radiation heat transfer is narrow, the fourth-order temperature difference can be replaced by a linear temperature difference (ISO 7933, 2004):

$$q_r = f_{cl} h_r (T_{cl} - T_{mrt}) \tag{10}$$

where the radiation heat transfer coefficient h_r can be calculated as follow:

$$h_r = 5.67 \times 10^{-8} f_{eff}\,\varepsilon \frac{(T_{cl} + 273)^4 - (T_{mrt} + 273)^4}{T_{cl} - T_{mrt}} \tag{11}$$

According to Fanger's study (Fanger, 1967), a typical value of 4.7 W/(m^2 °C), which fulfill the accuracy requirements for most situations, can be employed for h_r in normal indoor environment.

Substituting Eqs. (4), (6), (9) into Eq. (3), we obtain Eq. (12) as follow:

$$\dot{m}c_p(T_{out} - T_{in})/A_{cl} = h_c f_{cl}(T_{cl} - T_a) + f_{cl} h_r (T_{cl} - T_{mrt}) + q_m \tag{12}$$

To validate the heat transfer model developed above, a prototype of LCG was built and tested. Fig. 3 shows the prototype on the manikin. The prototype consists of 17 m tubing with 2/4 mm of internal and external diameters respectively. The basic garment was made of flexible spandex and cotton mixture fabric. The effective cooling area A_{cl} of the prototype was 0.568 m^2. The water was recooled by an icepack with a volume of 600 ml. The assumptions for the theoretical model were also applied to experiments. The total weight of the LCG at work is about 1.5 kg.

Fig. 3. The photo of the prototype of LCG on the manikin. Cooling tubings were sewed to the lightweight basic garment and a zipper was used to make the LCG easy to take on and off.

4. Performance test

4.1. Test method

A thermal manikin method was proposed to test the thermal insulation of clothes by the American Society for Testing and Materials (ASTM F1291-10, 2010). The thermal manikin is electrically heated and divided into several individually controlled segments and can simulate the sweating mechanism of human body. Garment to be tested is dressed on the manikin and put into a temperature-controlled chamber. The total thermal resistance can be calculated by the following equation:

$$R_t = (T_s - T_a)A/H_c \tag{13}$$

where R_t is the total thermal resistance of the clothing ensemble and surface air layer (°C m^2/W), A is the area of the manikin surface (m^2), T_s is the temperature at the manikin surface (°C), T_a is the air temperature (°C), and H_c is the power input to heat the manikin (W).

However, for an LCG in hot environment, the heat is only taken away by the cooling water, which means that no heat dissipates directly through the garment. On the contrary, heat fluxes from the environment and the body both go into the cooling water. Thus this method is not suitable for LCG thermal property test.

Cao et al. (2005) proposed a test method to evaluate the cooling capability of the LCG by using a sweating guarded hotplate. A piece of liquid-cooled textile system was put on the hotplate with the top side thermally insulated from the environment by a thick fabric. When the hot plate and the environment temperature reached the set values, the liquid cooling system was triggered to start working. A similar thermal resistance is defined:

$$R = (T_H - T_C)A/H \tag{14}$$

Here, R is the thermal resistance between the hotplate and the cold liquid inside the tubing (°C m^2/W), A is the surface area of the test object (m^2), T_H is the hot plate temperature (°C), T_c is the cold water temperature (°C), and H is the power input to heat the hot plate (W).

This hotplate test method is simple and easy to conduct. The limitations of this method are that it does not take the heat exchange between the environment and the liquid cooled garment into account, and only the thermal properties of a small sample of the liquid cooled garment system, rather than the complete system, can be measured.

To properly evaluate the thermal properties of our LCG, a modified approach was employed based on the heat transfer model and the above two methods. The thermal resistance of the LCG is redefined and divided into two parts: one from the skin to the cooling water (R_1), the other from the ambient environment to the cooling water (R_2).

Fig. 4 demonstrates the heat transfer paths and the measurement for LCG. The heat flux from the manikin surface to the cooling water (q_1) was measured by heat flux sensor 1, which was inserted between the skin and the LCG, while the heat flux from the ambient environment to the cooling water (q_2) was measured by heat flux sensor 2, which was attached to the outside surface of the LCG. T_{cl} and T_s were the average temperatures of the outside surface of the clothes and the manikin surface. The temperature of the cooling water (T_w) is considered to be ($T_{in}+T_{out}$)/2. Then the thermal resistances can be evaluated as follow:

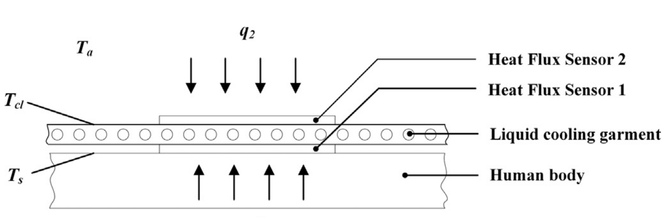

Fig. 4. Details of heat flux and temperature measurements for LCG system. q_1 and q_2 are the heat fluxes measured from metabolism and the ambient environment respectively. T_a is the ambient temperature, T_s is the skin temperature and T_{cl} is the clothes surface temperature.

$$R_1 = \left|\frac{T_s - T_w}{q_1}\right| \tag{15}$$

$$R_2 = \left|\frac{T_a - T_w}{q_2}\right| \tag{16}$$

Thus the total thermal resistance of the LCG is given by

$$R_t = R_1 + R_2 \tag{17}$$

The work duration is also one of the critical evaluation indexes for LCG. It was defined as the time range from the startup of the cooling system to the moment when the temperature of the cooling water reaches the target manikin surface temperature.

4.2. Performance test apparatus

The schematic of the performance test setup is shown in Fig. 5. A half manikin was used to build the performance test setup (Fig. 6). The half manikin was wound around by heating wires and then covered by conductive aluminum foil to improve the heating uniformity. Fig. 7 displays the photo of the overall experimental apparatus. The manikin was put in the heat chamber to get a steady temperature environment. The micropump was driven by a pulse-width modulation (PWM) driver, and powered by a portable power supply. The blue loop represents the liquid cycle-cooling circuit, while the red one represents the data acquisition and transmission paths. Twenty-four thermocouples deployed in pairs were located in the front and the back separately. The inlet and outlet temperature of the tubing network were also recorded.

4.3. Procedure

(1) Dress the manikin with the LCG, and put it in the center of the heat chamber.
(2) The ambient temperature, T_a, i.e., the air temperature inside the heat chamber, is set to target temperature and maintained within a variation less than ± 0.5 °C throughout the whole test to mimic the hot hazard environment. The manikin surface temperature is set to 35 ± 0.2 °C, as suggested by the thermal manikin method mentioned above (ASTM F1291-10, 2010).
(3) After the ensembles reaching the steady state, we start the micropump and activate the LCG. The skin temperature will decrease and the ensembles would reach another equilibrium conditions. All data are continuously recorded. The time for each measurement lasts at least 30 min.
(4) Continue to record the data until the temperature of the cooling water reaches the manikin surface temperate, which would be 35 °C.
(5) Calculate the thermal properties and the work duration time of the LCG.

5. Results and discussion

Comparing the heat transfer path in Fig. 4 with the Eq. (3), we can conclude that the measured heat flux from the manikin surface to the cooling water q_1 is the same as the heat power of the manikin q_m, and the measured heat flux from the ambient environment to the cooling water q_2 should be equal to the sum of the convective heat flux q_{conv} and the radiation heat flux q_r. The ideal relation can be expressed by

$$q_1 = q_m \tag{18}$$

$$q_2 = q_{conv} + q_r \tag{19}$$

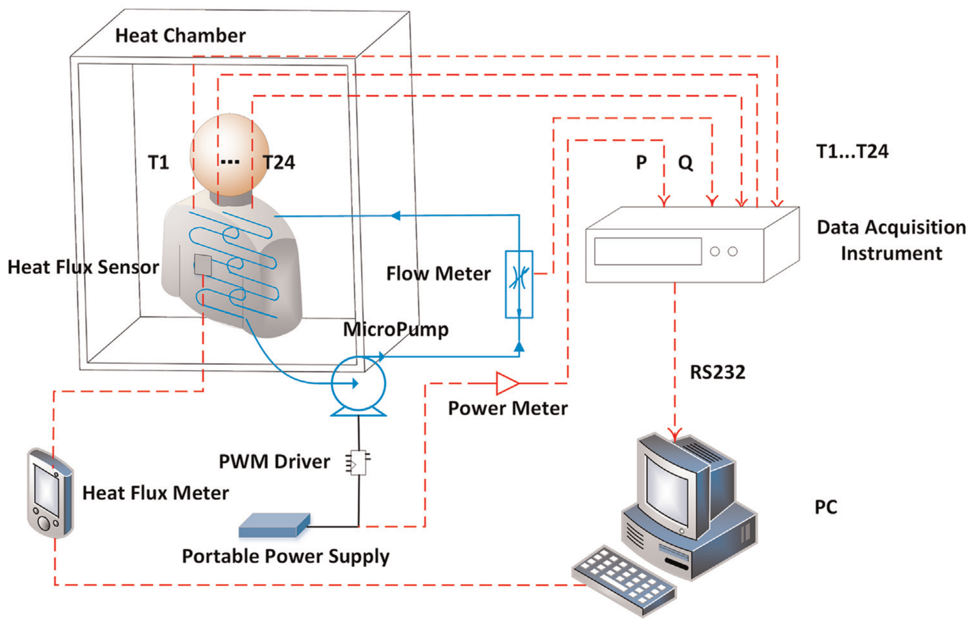

Fig. 5. Schematic of the performance test setup for LCG. The solid blue loop represents the liquid cycle-cooling circuit, while the dotted red one represents the data acquisition and transmission paths. T1 to T24 represents the measured temperature data, while P and Q are the recorded power consumption and flowrate. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

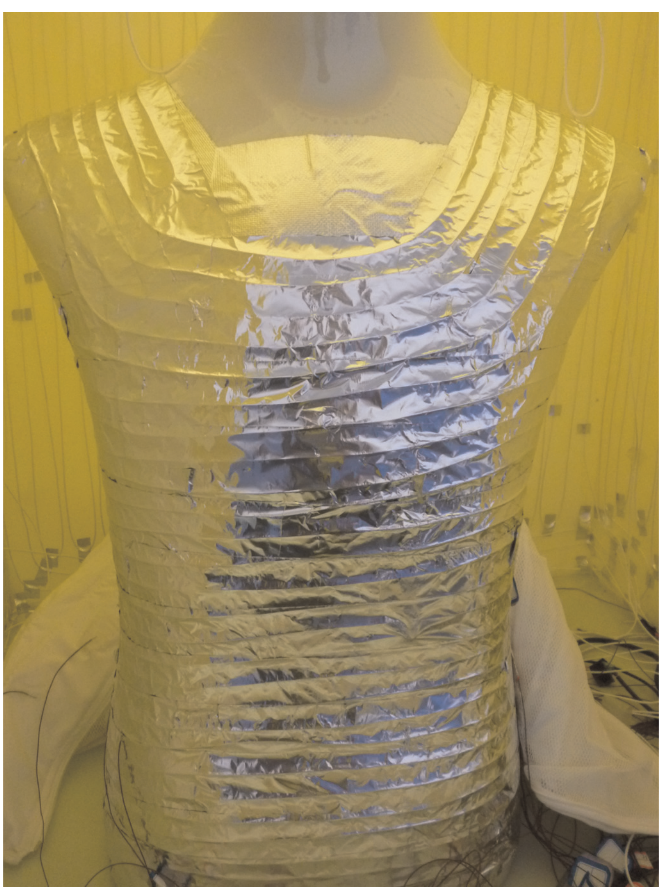

Fig. 6. A simplified thermal manikin used in the test. It consists of a half manikin made by thermal insulation material, heating wires, and a layer of aluminum foil to improve the heating uniformity.

A series of tests with different flow rates were done as well. Table 1 shows the results of the heating power q_m, the convective heat flux q_{conv}, the radiation heat flux q_r, and the measured q_1 and q_2. Here, q_{conv} and q_r are calculated by the above heat transfer model. q_m, q_1 and q_2 are experimentally tested results. E_{r1} represents the percent error between q_1 and q_m while E_{r2} is the error between q_2 and the sum of q_{conv} and q_r. All the measurements were the average values over a period of at least 30 min for each test. It can be seen that both E_{r1} and E_{r2} are under 15%.

Table 2 shows the thermal equilibrium comparison between experimental and predicted results at different flow rates. Here the heat removal rate of LCG q_w is the experimental result. E_{r3} and E_{r4} indicate the thermal equilibrium errors of the experimental and predicted results respectively. According to the heat transfer model and Eq. (3), we obtain the following equations theoretically, $q_w = q_1 + q_2 = q_{conv} + q_r + q_m$. Results in Table 2 demonstrate that errors are less than 10%. Both the two tables prove that the heat transfer analysis is applicable and can be used for engineering practice.

Fig. 8 shows the temperature difference variations of the inlet and outlet water with the flow rate. It can be seen that the temperature differences only decreases slightly as the flowrate increases. Small temperature difference means that the wearer will not suffer thermal discomfort caused by local overcooling or overheating.

Fig. 9 presents the approximate linear relation between the cooling capacity (q_w) and the water flowrate, with a coefficient of determination, denoted R^2 of 0.97. It is easy to estimate the flowrate for a specific cooling power demand by using the linear fit curve.

Fig. 10 shows the trend of the thermal resistances of the LCG based on the measurements. A lower value of the total thermal resistance for LCG means higher heat removal rate. From Fig. 10, it is noticed that unlike normal clothes with a constant thermal resistance value, the total heat resistance of LCG varied a lot when the flowrate changed. The thermal resistance between the skin and water R_1 fluctuates slightly, less than 1% of the average value. This result matches the experimental results of Cao et al. (2005), who demonstrated that R_1 is constant. The reason for the constant thermal resistance is that the heat transfer from the manikin surface to the cooling water is dominated by heat conduction, for which the thermal resistance is determined by the material properties. The resistance between the ambient and the cooling

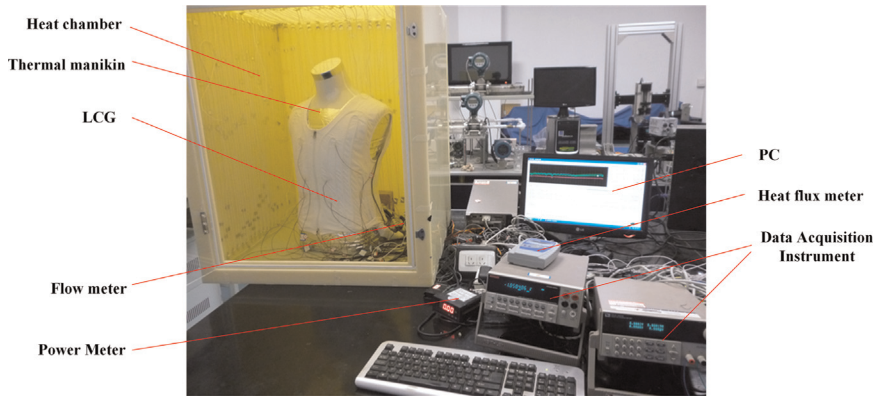

Fig. 7. A photo of the overall performance experimental setup for LCG. Connections and components are consistent with the schematic in Fig. 5.

Table 1
Comparison of the calculated heat flux and measured heat flux at different flow rate.

Flowrate/(mL/min)	q_1/(W/m^2)	q_2/(W/m^2)	q_m/(W/m^2)	q_{conv}/(W/m^2)	q_r/(W/m^2)	$E_{r1}\times 100\%$	$E_{r2}\times 100\%$
224.5	62.3	66.1	67.5	16.3	46.2	7.57	5.38
333.5	85.3	98.1	86.0	31.8	57.8	1.25	8.55
425.9	87.1	128.5	92.1	45.4	79.9	5.62	2.49
544.2	88.5	166.1	97.9	57.2	88.1	9.71	12.54

Table 2
Thermal equilibrium comparison between experimental and predicted results at different flow rates.

Flowrate/(mL/min)	q_w/(W/m^2) (experimental test)	q_1+q_2/(W/m^2) (Experimental test)	$q_{conv}+q_r+q_m$ /(W/m^2) (Model calculation)	$E_{r3}\times 100\%$	$E_{r4}\times 100\%$
224.5	118.3	128.4	129.6	7.86	9.51
333.5	174.5	183.4	176.0	4.82	0.87
425.9	206.6	215.6	217.3	4.17	5.17
544.2	243.2	254.6	242.4	4.46	0.36

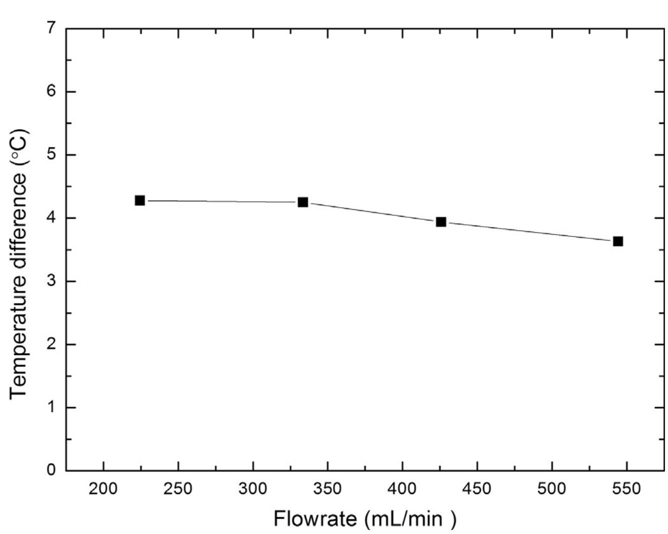

Fig. 8. Variation of temperature difference between the inlet and outlet water at different flowrate.

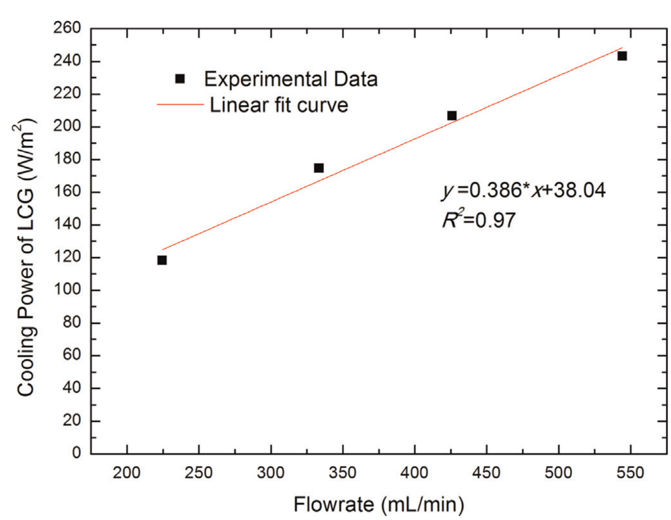

Fig. 9. The effect of flowrate on the cooling power rate of LCG. Experimental data of cooling power were the average value of recorded data for at least 30 min after reaching steady state.

water R_2 is almost cut down into half (from 0.463 to 0.256 °C m^2/W) when the flowrate increased from 224.5 to 544.2 mL/min. The total thermal resistance R_t varied a lot at different flowrate as well. It proves that the environment has great effect on the thermal property of the LCG and it is not appropriate to use thermal resistance R_1 to represent the total thermal resistance of LCG.

The evaluation of the effect of the ambient temperature and the flowrate on the work duration time of the LCG was measured under two conditions. One was under the condition at different ambient temperature with the same flowrate of 475 mL/min, and another one was keeping the temperature at 45 °C while using different flowrate. Fig. 11 shows the experimental results of work duration tests, which was measured at 45 °C when the flowrate was 475 mL/min. The ambient temperature T_a remained 45 °C with a variation less than ±0.3 °C. The startup time is the time when the LCG was activated, and the end time is chosen as the

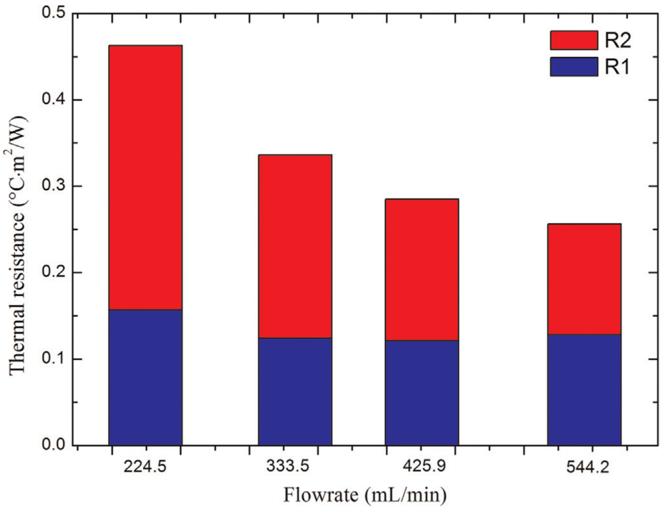

Fig. 10. Comparisons of thermal resistance R_1 and R_2 at different flowrate. R_1 is the thermal resistance from the skin surface to the cooling water, R_2 is the one from the ambient to the cooling water.

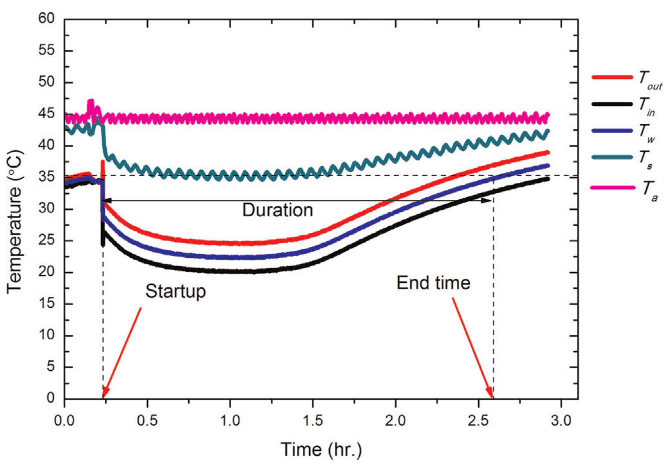

Fig. 11. A typical work duration test at 45 °C ambient temperature with a flowrate about 475 mL/min. This kind of test was repeated in the study of effects of ambient temperature and flowrate on the work duration time. The minor fluctuations of ambient temperature T_a and the manikin surface temperature T_s were due to the flaw of the imperfect temperature control.

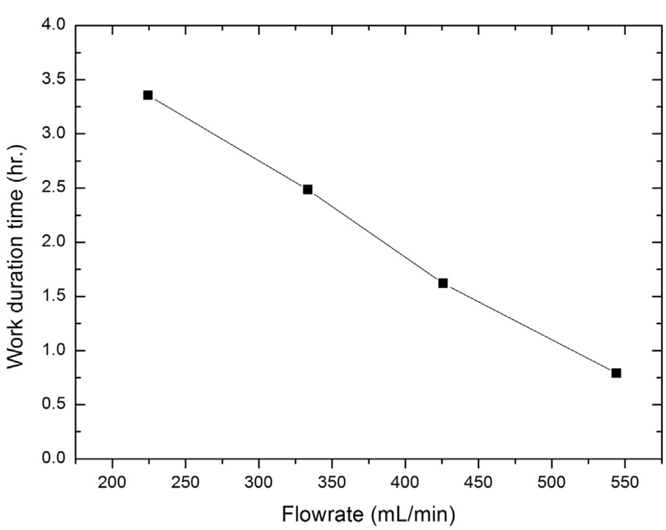

Fig. 12. The effect of flowrate on the work duration time at 45 °C ambient temperature.

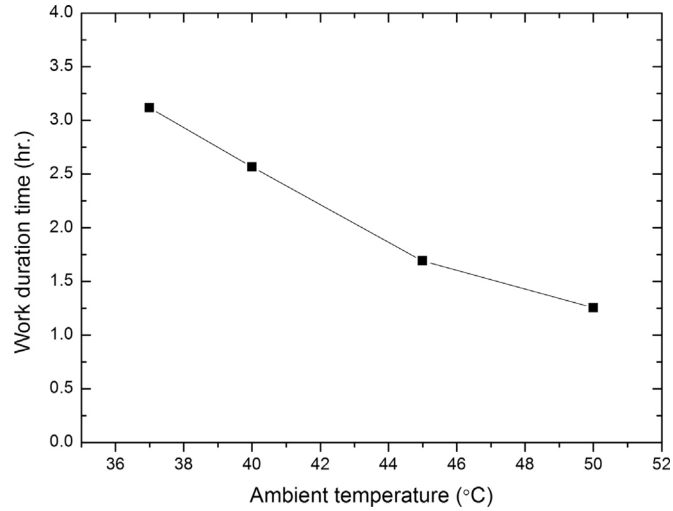

Fig. 13. The effect of ambient temperature on the work duration time at about 475 mL/min.

cooling water temperature T_w reached 35 °C. T_w is the average value of the inlet (T_{in}) and outlet temperature (T_{out}). From Fig. 11, it can be observed that a maximum work duration time of 1.7 h can be obtained at this condition.

Fig. 12 and Fig. 13 show the influences of the flowrate and ambient temperature on the work duration time, respectively. From Fig. 12, the work duration time reduces from 3.36 h to 0.79 h when the flowrate increases from 224.5 mL/min to 554.2 mL/min. Although large flowrate can provide higher cooling rate, the work duration at such condition is much shorter than the one when the flowrate is smaller. As a consequence, the flowrate has to compromise the demands of heat removal rate and the wok duration when designing the LCG.

Fig. 13 illustrates that the ambient temperature greatly affects the work duration time of LCG. In a normally hot ambient of 40 °C, the LCG can work as long as 2.57 h, while only 1.25 h in the extremely hot ambient of 50 °C. It should be noted that it is not suitable to decrease the flowrate to increase the work duration time in such extreme hot environment. The low heat removal rate at low flowrate cannot match the high heat flux gained from the ambient. Therefore, the only way to improve the effective work duration time in the extremely hot environment is to increase the volume of ice pack.

The experimental results of these two tests provide guidelines for designers to optimize the design to fulfill both the demand of work duration and cooling capacity in different ambient temperature.

Different prototypes of cooling garment have different weight because of the different volume of icepack and the material used. Although more icepack can increase the work duration, Chauhan (1988) concluded that the increment of the garment weight would significantly increase the energy expenditure of the wearer and therefore reduce the overall effectiveness of the garment. Here we propose a parameter to describe the cost performance of LCG that balances the requirements of long work duration time and light weight. At the field of absorption cooling, a specific cooling power (SCP) defined as the ratio of the cooling power to the total system mass is normally used to assess the cooling performance (Critoph and Metcalf, 2004; Yong and Sumathy, 2005; Wang et al., 2009). For a nominal cooling load, higher SCP reflects better compactness of the system, namely lower system weight burden cost. Inspired by this, a specific duration time, defined as the ratio of the effective work duration time to the mass of the LCG, was proposed to represent the weight cost for the durability of an LCG, and is expressed as follow:

$$t = t_d/m \tag{20}$$

Table 3
Comparison of specific duration time of different prototypes under similar condition.

Works	Maximum work duration time t_d/(h)	Weight of cooling garment W/(kg)	Specific duration time t (h/kg)	Ambient temperature (°C)
DeRosa and Stein (1976)	2	4.5	0.44	45
Yang et al. (2012)	1	3.4	0.29	40
Ernst and Garimella (2013)	5.7	5.31	1.07	47.5
By our group	3.36	2.0	1.68	45

where t is the specific work time, t_d is the maximum work duration, and m is the mass of the cooling garment. A higher specific duration time indicates a longer work time with the same mass, or in other words, a lighter system with a same cooling time. One matter to note here is that comparison of different prototypes should be conducted under the same conditions of the ambient temperature, the initial icepack temperature and the inlet flowrate. And the specific duration time can also be applied to other kind of cooling garments.

Table 3 show the comparison of specific work time of the present prototype with others'. It could be seen that much longer specific work time can be obtained by the present prototype. This can be explained as follows. When designing an LCG, many parameters usually need to be taken into account. In the present design, all the detailed parameters were included in the former heat transfer model, such as the thermal conductivity and density of the clothes and the tubings, the thickness of the garment and the flowrate of the cooling water. Specific duration time can be estimated in the design process and associated with those detailed parameters. Thus an optimal combination of those detailed parameters was achieved.

6. Conclusion

This study presented a heat transfer model of LCG. In this model, different heat flux terms were studied independently, and combined to be an integrated heat balance equation. Experiments were conducted to validate the heat transfer model and measure the thermal properties of the LCG. The results show that heat flux terms meet the heat balance equation with an error less than 10% between the experimental and calculation results. Thermal resistance analysis demonstrates that the flowrate has greater effect on the thermal resistance between the cooling water and the ambient than the one between the skin surface and the cooling water. Experiments also indicate that both flowrate and the ambient temperature greatly affect the work duration time of the LCG. Specific duration time was proposed to balance the light weight and long work time requirement.

Development of a liquid cooled garment is a multi-disciplinary and complex task. More efforts need to be done to get a more detailed and accurate heat transfer model. The human's thermal comfort also needs to be considered in further researches.

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