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[Continued on next page]

(54) Title: MICROCHANNEL FIN HEAT EXCHANGER

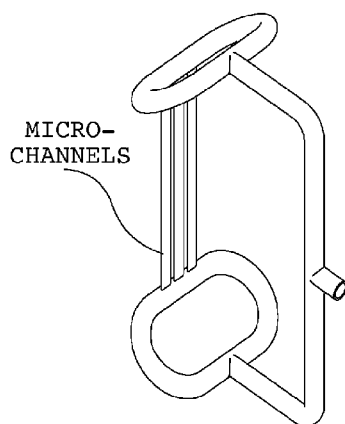


FIG-1A

(57) Abstract: An efficient and compact heat exchanger, such as a thermosyphon, having fins with microchannels formed therethrough. The array of microchannels enhances the heat transfer between the fin and the fluid flowing over the fins. The gaseous phase of the working fluid, such as propane, condenses on the condenser sidewalls as heat is removed, and the gaseous phase extends through the center of essentially the entire length of the microchannels. This results in an isothermal fin, which enhances heat transfer due to maximization of the temperature difference between the working fluid and the fluid flowing over the fins.



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TITLE:

MICROCHANNEL FIN HEAT EXCHANGER

BACKGROUND OF THE INVENTION

5 [0001] The invention relates to a heat exchanger using microchannel fins as a thermosyphon system.

[0002] Heat exchangers are thermal devices that are used to transport heat between moving fluids at different temperatures. Heat exchangers are widely used in many different applications and industries worldwide, such as aerospace, energy, heating
10 and air conditioning, and transportation. A solid fin on a computer processor, a radiator for cooling a car or truck engine, and a condenser for an air conditioner are typical heat exchangers.

[0003] Traditional solid fins are thin conductive metal extensions that are joined to an object (the source) that is releasing or receiving heat. The fins provide enhanced
15 heat transfer to or from the ambient air or other coolants due to an effective increase in the surface area of the object. However, this increase comes at a cost of efficiency due to conduction resistance within the fin. The finite thermal conductivity of the fin material is a limiting factor in the heat transfer because of the thermal resistance and temperature drop across the length of the fins. The heat transfer of a fin is proportional to the surface
20 area, and also proportional to the temperature difference between the fin surface and the surrounding fluid (i.e., a gas or a liquid). In a heat exchanger where fins are used to cool a hot source, the best performance is obtained where the fins are at the highest possible temperature: i.e. the hot source temperature. This maximizes the temperature difference between the fin and the surrounding cooler fluid flow (typically air). Therefore, the ideal
25 system should have fins of very high thermal conductivity (ideally infinite conductivity), resulting in isothermal fins and 100% fin efficiency.

[0004] The performance of actual solid fins, however, is limited by the thermal conductivity of the fin material, which produces a temperature drop along the length of the fin. This reduces the fin efficiency, and limits the fin lengths that can be used
30 effectively. The ideal fin can be thought of as one with thermal conductivity high enough so as to be isothermal throughout a desired or specified length. The efficiency of an actual

fin can be expressed as the ratio of heat actually transferred to the heat that would be transferred with an ideal fin.

[0005] In a known study of plate fin heat sinks, it was shown that the effect of finite thermal conductivity resulted in lower values of fin efficiency and a proportional reduction in the Nusselt number and the total heat transfer. The reductions were in the range of about 15% to 25%.

[0006] Conduction resistance reduces the temperature of a fin so that much of the extended surface of the traditional solid fin always has a lower temperature than the heat source at the base of the fin. The typical efficiency for a solid aluminum fin is about 60%. This means that, for every 100 Btu of heat transferred from a classical fin, there is the potential for an additional 67 Btu to be transferred from the same fin. Many heat exchangers use solid fins to enhance heat transfer, and they are all limited by the fin efficiency factor.

[0007] The fin efficiency can be increased by making the temperature of the whole surface of the fin to be the highest possible value, i.e. the same temperature as the hot source (typically the base temperature of the fin) to which the fin is attached to conduct heat away. In other words, an ideal, high efficiency fin should be isothermal all along its length with a large surface area; and the whole surface should be the highest possible temperature (the heat source temperature). In case of cooling a system, the fin should still be isothermal, but have the lowest possible temperature.

[0008] Finned heat exchangers are also used in the transport industry to enhance heat transfer; the automobile radiator is an example. As in the radiator, the cooling fluid is made to flow past the fin, either by natural or forced convection. Such fins can also be used in heat sinks for cooling of electronics and other heat sources that require efficient cooling methods.

[0009] The cooling requirement for automotive engines often requires a large frontal area for sufficient airflow to cool the radiator fluid. This increases the friction on the vehicle, especially at high speeds, and reduces fuel efficiency. A disadvantage of conventional radiator designs is that heat transfer coefficient and heat transfer area are limited by the materials and configuration used. The most commonly used number for measuring the performance of the automobile radiator is the ratio of heat transfer to the frontal area divided by the difference of the inlet temperatures of coolant and air ($Q/(A_f \times$

ΔT)). This number is typically 0.31 W/K-cm^2 for the state of the practice radiator. In the current design, it is not possible to increase this “performance” parameter cost-effectively.

[0010] Microchannel tubes have been used in heat exchangers to pump a hot or cold fluid through the microchannel tubes. The advantage of this approach is the simplicity of the design, and the increased heat transfer due to high surface to volume ratio in the microchannels. The disadvantage is that the channels can be blocked due to scaling or due to particles in the fluid. In addition, the pumping power necessary to pump sufficient fluid through small channels can be quite high due to the fact that smaller cross section passages have a greater pressure drop. The pumping process itself can also introduce particulates into the microchannel flow.

[0011] Thermosyphons and heat pipes can carry high heat fluxes with a small temperature difference between the condenser and evaporator, and have been studied for thermal management of electronics.

15 BRIEF SUMMARY OF THE INVENTION

[0012] The purpose of the invention is to produce efficient and compact heat exchangers by using highly efficient fins with microchannels formed therein that work as heat pipes or thermosyphons. The overall configuration could also be used as a high efficiency heat exchanger that can replace radiators, evaporators, condensers, plate-fin heat exchangers, or heat sinks in a variety of applications.

[0013] The invention consists of a heat exchanger using fins that contain an array of microchannels operating as heat pipes or thermosyphons that make the fins isothermal and enhance the heat transfer between the fin and the surrounding fluid. Efficient heat exchangers using these fins can be used to either heat or cool a system. Applicants have completed the design, model and test of a more efficient, small heat exchanger working on the thermosyphon principle with microchannel tubes embedded into fins.

[0014] To take advantage of the very high apparent conductivity of thermosyphons, a loop, two-phase thermosyphon system was designed to keep a set of fins close to isothermal conditions. In the experimental system, the condenser of the thermosyphon loop was located within copper fins in a set of micro-channel tubes with propane as the working fluid. Experimental results showed that this integrated arrangement of thermosyphon and fins results in reasonably isothermal fins, producing

high fin efficiency. The specific volume of the working fluid was determined and used to characterize the system performance and stability.

[0015] The invention thus uses internal flow paths of a micro-channel tube as the condenser of a loop thermosyphon, while the external surface is used as a fin to be cooled by a flow of air. Fins must be very thin structures, and therefore Applicants have noted that microchannels are most appropriate for flow inside the fins. Microchannels provide greater heat transfer due to greater heat transfer area per unit volume. Flows in microchannels have greater heat transfer coefficient. Therefore, the condensing fluid inside the microchannels is very effective in keeping the fin at the same temperature as the heat source. The microchannel fin heat exchanger is very compact. Therefore, a radiator using such fins will reduce the frontal area of the radiator, thereby allowing more design latitude to reduce the drag coefficient of automobiles and trucks. This would increase their fuel efficiency.

[0016] The thermosyphon design uses a passive, such as gravity, flow that eliminates the need to actively pump fluid through the microchannels, such as with a mechanical or other pump device. Heat pipes likewise use capillary action, which is another mode of passive circulation. The invention eliminates or reduces the pumping requirement for the fluid. Since the pressure drop in microchannels is quite high, the high pumping power for microchannel flow is eliminated. The thermosyphon is a sealed unit, and therefore the possibility of flow obstruction in the microchannel due to scaling or particulates in the flow can be eliminated.

[0017] The thermosyphon effect produces a uniform temperature in the microchannel fins. This increases the fin efficiency to almost 100%.

[0018] Microchannel heat pipes inside a fin can significantly increase the heat transfer of the fin, because heat pipes and thermosyphons transfer heat by phase change, unlike conventional radiators. In the phase change, working fluid that is in vapor form condenses on the sidewalls of fluid flow passages, thereby changing to a liquid form. Vaporized working fluid transports heat rapidly without the need for a pump. This process thereby maintains the microchannel sidewalls at high temperature, because the condensed liquid on the sidewalls is close to the temperature of the vaporized working fluid. The micro-channels typically have a central portion that is vaporized fluid surrounding by condensed working fluid on the sidewalls of the micro-channels, but this is not necessarily so only in the micro-channels nor is it always present in the micro-

channels. Therefore, with the invention the fin surface can be close to isothermal and higher heat fluxes are realized.

[0019] Another advantage of the invention is that pumping in the microchannels is eliminated. The issue of flow blockage can also be eliminated due to the sealed working
5 fluid – the fluid at the exterior of the sidewalls of the passages containing the working fluid does not communicate with the fluid inside the heat exchanger, and therefore no contaminants can block the flow. Additional metal fins (the traditional solid, heat exchanger type) can be brazed or bonded to the microchannel fins to increase the heat transfer area. A disadvantage of heat pipes is that the fins, including the heat pipes and
10 wick inside, involve a more complex design than thermosyphons.

[0020] The microchannel arrays can be thermosyphons, which have a simpler design than heat pipes. Microchannels can be operated as thermosyphons with no modification of the channels. The disadvantage is that thermosyphons need to be vertical, or have sufficient vertical orientation to use gravity for driving the fluid flow. In
15 situations where the vertical orientation is possible, the thermosyphon can be a preferred configuration, because it allows the heat exchanger to operate at very high efficiency.

[0021] In order to test the invention, a heat exchanger with flow across microchannel fins was built. The fins were made of “microchannel tubes”, which are metal ribbon or tape-like structures containing an array of microchannels running through
20 them. The channels are preferably in the range of about 50 to about 1,000 microns in hydraulic diameter for the particular working fluid used.

[0022] In the invention, the microchannels are used to contain the thermosyphon or heat pipe fluid which is sealed under high pressure. The ribbon shape makes it feasible to use the tubes as fins over which air or any other fluid is directed, while the
25 microchannel tubes inside these fins are used as heat pipes or thermosyphons. In a loop thermosyphon, the microchannels can be operated as a condenser or an evaporator. The desired end result is that the “microchannel tubes” perform similarly to fins, but with very high efficiency.

[0023] Significant improvements result when isothermal fins are used, with heat
30 transfer increased due to higher fin efficiency, and fins operating very effectively at longer than typical fin lengths. The invention has several advantages for a heat sink. The thermosyphon produces a uniform temperature in the micro-channels due to the high apparent conductivity. Since each fin contains a thermosyphon condenser, it is essentially

isothermal. Vapor can move through the channels at essentially one temperature, and condenses against the sidewalls at a precise condensation temperature. The condenser is cooled by the external flow of common coolants (such as air) over the outside surface of the fin, which is heated isothermally by the gaseous refrigerant and/or condensing liquid inside. Thermosyphons also eliminate the need to pump the working fluid through the micro-channels due to gravity flow resulting from the phase change. If there is need for flexibility with respect to orientation of the fins, the thermosyphon can be replaced by heat pipes within the fins.

[0024] The invention integrates thermosyphons and fins by embedding the condenser of a thermosyphon loop within the fin. This can be achieved by using a product similar to the commercial micro-channel tubes used in the HVAC industry. These micro-channel tubes, made of aluminum alloys, are typically flat, ribbon-like in shape, and usually contain about 5 to 10 micro-channels per tube. They are advantageous in HVAC applications since they require less storage space, and are highly efficient. Although aluminum alloy is the typical material used for these tubes, copper presents some advantages over aluminum. The main advantage of copper for heat transfer applications is its thermal conductivity (approximately 400 W/mK), which is almost double the thermal conductivity of aluminum (approximately 235 W/mK). Other advantages of copper are that it has anti-microbial properties and it is easier to join and repair than aluminum.

[0025] An important characteristic of micro-channel tubes is that they increase both the heat exchange and efficiency of the heat exchanger designs. This characteristic allows the micro-channel heat exchanger to be smaller and yet have the same performance as a regular heat exchanger, or to get improved performance in the same volume as a conventional heat exchanger.

[0026] These micro-channels have the additional advantage that they have the geometry of fins which are used extensively in heat sinks and heat exchangers. The thermosyphon system was designed by using copper micro-channel tubes that are similar to aluminum micro-channel tubes. Copper micro-channels were selected because of ease of joining and potential for improvement in heat transfer.

[0027] Fig. 1 shows a cross section of a copper micro-channel tube. There are nine individual micro-channels in this sample, with each channel approximately 1 mm² in area. These are typical dimensions for commercial micro-channels. However, these channels may be more accurately described as “meso-channels”. Nevertheless, the term

“micro-channel tube” is used commonly by the HVAC industry for these flow tubes, and will also be used herein.

[0028] Fins can be used for both heating and cooling the fluid that is flowing across their surfaces, depending on the condensation or boiling temperature of the working fluid inside the thermosyphon or heat pipe. In a loop thermosyphon, the heating of the external fluid flow takes place when the micro-channels inside the fin are used as condensers, and the cooling of the outside flow is achieved by using the micro-channels as evaporators.

10 BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWINGS

[0029] Fig. 1A is a view in perspective illustrating an embodiment of the preferred microchannel fin heat exchanger.

[0030] Fig. 1B is a side view illustrating the embodiment of Fig. 1A.

[0031] Fig. 2 is a table that includes properties of various common working fluids for heat pipes and thermosyphons.

[0032] Fig. 3 is a pressure-enthalpy (P-h) diagram for propane.

[0033] Fig. 4 is a table illustrating the temperature of the top two positions on microchannel fins versus the bottom position temperature. The temperatures at the top, middle and bottom are almost perfectly correlated, thereby indicating an isothermal system.

[0034] Fig. 5 is a view in section illustrating a copper microchannel fin of the type used in the present invention.

[0035] Fig. 6 is a view in perspective illustrating an embodiment of the preferred copper microchannel fin heat exchanger on a loop thermosyphon prototype.

[0036] Fig. 7 is a view in perspective illustrating a prototype embodying the invention.

[0037] Fig. 8 is a side view illustrating the experimental set-up used in the experiments described herein.

[0038] Fig. 9 is a schematic illustration of a computer simulation showing the contours of velocity in the flow across the 3 copper micro-channel fins.

[0039] Fig. 10 is a schematic illustration of a computer simulation showing temperatures in the airflow across the 3 copper micro-channel fins.

[0040] Fig. 11 is a graphical illustration of the data obtained for an experiment using a prototype without propane (with air velocity=0.5 m/s). The fin temperatures are given by nine thermocouples showing temperatures in the back, mid and front fins, at top, middle and bottom locations. The reading number corresponds to time and shows the transient and steady state behavior.

[0041] Fig. 12 is a graphical illustration of the experimental fin temperature data obtained for three microchannel locations.

[0042] Fig. 13 is a graphical illustration of the data obtained for an experiment using a prototype with propane (0.007 m³/kg, evaporator at ~50°C, 0.5 m/s). The fin temperatures are given by nine thermocouples showing temperatures in the back, middle and front fins, at top, middle and bottom locations. The reading number corresponds to time. The graph shows the transient and steady state behavior of the apparatus.

[0043] Fig. 14 is a graphical illustration of the data obtained for an experiment using a prototype with propane (0.007 m³/kg, evaporator at ~50°C, 1.0 m/s). The fin temperatures are given by nine thermocouples showing temperatures in the back, middle and front fins, at top, middle and bottom locations. The reading number corresponds to time. The graph shows the transient and steady state behavior of the apparatus.

[0044] Fig. 15 is a table listing the data obtained for an experiment using a prototype with propane at 0.007 m³/kg.

[0045] Fig. 16 is a table listing the data obtained for an experiment using a prototype with propane at 0.009 m³/kg.

[0046] Fig. 17 is a table listing the data obtained for an experiment using a prototype with propane at 0.011 m³/kg.

[0047] Fig. 18 is a table listing energy balance results showing heat transfer to the air at two air flow velocities comparing the invention to the conventional (classical) solid fin.

[0048] Fig. 19 is a view in perspective illustrating an embodiment of the automotive radiator using heat pipes. The heat pipes are shown approximately horizontal, whereas thermosyphons would be operated in a vertical or "close to vertical" configuration.

[0049] Fig. 20 is a view in perspective illustrating an end of an embodiment of a fin showing the array of micro-channels inside the copper fin.

[0050] Fig. 21 is a view in perspective illustrating the thermosyphon with thermocouples mounted thereon. The fins are shown exposed in the wind tunnel, while the rest of the thermosyphon loop is insulated.

[0051] Fig. 22 is a graphical illustration of the data obtained for an experiment, in particular the temperature history of the thermocouples during experimentation of the copper micro-channel thermosyphon system. The top set of data (prototype) includes all nine readings from the micro-channel tube fins. The two lines shown below represent the temperature readings of the air flow at the inlet and the outlet of the experimental wind tunnel.

[0052] Fig. 23 is a table listing the data obtained for an experiment using propane at $0.007 \text{ m}^3/\text{kg}$ for two different air flow velocities.

[0053] Fig. 24 is a table listing the data obtained for an experiment using propane at $0.007 \text{ m}^3/\text{kg}$ for air flow at 0.5 m/s , particularly listing the temperatures at the three locations (front, middle and back along the direction of air flow) of each of the three fins.

[0054] Fig. 25 is a table listing the data obtained for an experiment using propane at $0.011 \text{ m}^3/\text{kg}$ for air flow at 0.5 m/s , particularly listing the temperatures at three locations (front, middle and rear along the direction of air flow) for each of the three fins.

[0055] Fig. 26 is a table listing the data obtained for an experiment using propane at $0.011 \text{ m}^3/\text{kg}$ at two air flow velocities.

[0056] Fig. 27 is a graphical illustration of the data obtained for an experiment, in particular the temperature history of the thermocouples during an experiment of the copper micro-channel system without the working fluid (propane). The temperatures reach a steady state, but the micro-channel tube fins are not isothermal.

[0057] Fig. 28 is a table listing the data obtained for the experiments and comparing respective heat transfer for the thermosyphon and the simple fin apparatus.

[0058] Fig. 29 is a schematic view illustrating an alternative embodiment of the present invention.

[0059] In describing the preferred embodiment of the invention which is illustrated in the drawings, specific terminology will be resorted to for the sake of clarity.

However, it is not intended that the invention be limited to the specific term so selected and it is to be understood that each specific term includes all technical equivalents which operate in a similar manner to accomplish a similar purpose. For example, the word connected or terms similar thereto are often used. They are not limited to direct

connection, but include connection through other elements where such connection is recognized as being equivalent by those skilled in the art.

DETAILED DESCRIPTION OF THE INVENTION

5 **[0060]** U.S. Provisional Application No. 61/257,513 filed November 3, 2009, the above claimed priority application, is incorporated in this application by reference.

[0061] In order to verify that the working fluid within the loop thermosyphon would circulate and produce isothermal microchannel fins with high heat transfer, a simple initial prototype was designed with microchannel and built (Figs. 1A and 1B).
10 Two embodiments are discussed below. First, a heat sink based on microchannel fins was fabricated as a prototype heat exchanger. Test results are discussed below. Second, an automotive radiator using thermosyphons and/or heat pipes is illustrated and discussed. The microchannel fins can be used to enhance the design further.

[0062] The inventors designed, fabricated and measured the performance of a heat
15 sink using a microchannel fin heat exchanger. For additional details of the validation of the microchannel fin concept, a discussion is given below about the experimental design and results from a heat exchanger made from a set of three microchannel fins made from copper microchannel tubes. It was tested as a heat sink by using an electrical heater as the heat source.

20 **[0063]** The microchannel fin heat exchanger shown in Figs. 1A and 1B is based on a loop thermosyphon design that is to be operated over the range of ambient temperature of -30 to 50°C. This permits the heat exchanger to be cooled by air flow in most environments. In most cases, the thermosyphon fluid must be sealed under pressure so that the thermodynamic cycle is in the two phase range. For the case of microchannels,
25 the fluid viscosity should be low. Even though the operating range of carbon dioxide is not suitable for this particular application (due to micro-channel size), carbon dioxide can be a preferred thermosyphon fluid in some applications because of lower viscosity. Successful experiments have been carried out by the applicants using carbon dioxide in a thermosyphon.

30 **[0064]** The thermosyphon fluid for the operating range of -30 to 50°C was selected on the basis of the Fig. 2 data for many thermosyphon fluids. Based on Fig. 2, ammonia seemed to be a good candidate, but because of its corrosiveness and toxicity it was not used in the apparatus. This is not meant to be an exclusion of ammonia for all

purposes, but for the apparatus described herein. Therefore, for other conditions, such as temperature range, channel size, environment and/or material type of the thermosyphon sidewalls, ammonia or other fluids could be selected as a working fluid.

[0065] Although propane is not a common working fluid for thermosyphons, it was chosen for this project because (i) it can be efficient in the operating temperature range of -30°C to $+90^{\circ}\text{C}$, (ii) has a relatively low global warming potential, (iii) it is readily available, and (iv) it is not toxic. The thermodynamic chart of propane is shown in Fig. 3. The charging point was chosen based on the specific volume so that the system operates on a large enthalpy (h) range for maximum heat transfer. The charging and operating ranges are included in Fig. 3. Another possible working fluid with acceptable operating ranges is ammonia which can be used in the range of -60°C to 100°C . However, ammonia must be handled with care because of toxicity.

[0066] In this invention, the fin has an array of small channels which are used as heat pipes or thermosyphons. The channels can be as small as 50 to 100 microns, or as large as 1,000 microns approximate hydraulic diameter. (“Hydraulic diameter” is a conventional term for a passage that is non-circular that is defined as the equivalent circular diameter of the passage after taking into consideration the effect of the non-circular passage on a fluid flowing through it.) The result is a fin structure that operates at isothermal conditions throughout its length (from top to bottom). This produces almost 100% fin efficiency over a range of heat transfer rates, and it improves the efficiency of the heat exchanger. A “fin” is defined herein as a structure that is substantially thinner than it is wide. Typical fins are an order of magnitude (or more) wider than their thickness. For example, the fins of the prototype are about 1-3 mm thick and about 10-15 mm wide.

[0067] Tests were carried out on the prototype shown in Figs. 1A, 1B and 6, and the results are shown in Fig. 4, in which the temperature variation of the fin top section (measured at the top and the middle) is plotted as a function of the temperature at the bottom location. It can be observed that the temperatures at the top and middle locations are very close to each other, and the three temperatures (top, middle and bottom) were very closely correlated by a linear relationship with a slope of 1.005, which is a variation of only 0.5% from perfect match. This is a validation of the thermosyphon principle. The temperatures remained steady at the desired power input, showing that there was no flooding or dry out in the system. This shows that the microchannel fin structure is

operating as a thermosyphon loop, thus producing nearly isothermal fins. Additional checks were carried out to verify that the microchannel fins are operating as highly efficient heat transfer surfaces.

[0068] The copper micro-channel tubes are shown in section in Fig. 5. The prototype heat exchanger was designed by using three copper micro-channel tubes as the fins and additional copper connecting tubes as shown in Figs. 1 and 6. The prototype heat exchanger was then fabricated with a charging tube connected as shown in Fig. 7. The fins in the prototype are aligned parallel to the direction of the flow of air, and coplanar to one another. Alternatively, the fins could be arranged in arrays to intersect the flow of air or to be parallel to the flow of air, but not coplanar.

[0069] In order to test the performance of the prototype, a special, small-scale wind tunnel was fabricated. Fig. 8 shows the experimental set-up for the project. The wind tunnel has a flow meter in series to measure the air flow entering the experiment. Computer simulations were performed to determine the heat transfer characteristics and Figs. 9 and 10 show the top view of the velocity and the temperature as the air flows across the three copper micro-channel fins.

[0070] A total of fourteen type-K thermocouples were calibrated and used to collect data during the experiment. Experiments were run with the prototype charged at three different specific volumes (0.007, 0.009 and 0.011 m³/kg), two different air flow speeds (0.5 and 1.0 m/s), and a temperature of approximately 50°C at the evaporator, for a total of six experiments. The same experiments were repeated with the evaporator temperature at approximately 70°C. The main goals of these experiments were to test the design and check for dry out conditions in the prototype.

[0071] This corresponds to the classical fin experiment. The experiments without propane were run with air flow at 0.5 and 1.0 m/s until the temperature at the base (heat source) level of the fins reached approximately 50°C. Fig. 11 shows the data for the experimental run with no propane, and an air flow velocity of 0.5 m/s. It also shows that the fins are not isothermal; i.e. they are operating in the classical fin mode. Typically, the fins were 20 to 30 degrees colder than the heat source (which reduces the heat transfer from the fins in a traditional heat exchanger). The heat transfer to the air is obtained from the air inlet and outlet temperatures. At steady state conditions, these were 21.86°C and 23.77°C, respectively when the evaporator was approximately 50°C with air flow at 0.5 m/s. Without the propane in the micro-channels, the fins appear to behave as classical

fins and this is confirmed by comparison of the classical fin temperature curve and the experimental data. This is shown in Fig. 12 for air flowing at 1 m/s.

[0072] Experiments were also performed with the microchannel fin loop thermosyphon prototype charged with propane at specific volumes of 0.007 m³/kg, 0.009 m³/kg, and 0.011 m³/kg. The heat source for the heat exchanger was an electrical heater. Therefore, this was a heat sink configuration. The temperature history for a typical run is shown in Figs. 13 and 14. It can be seen that the nine temperature measurements on the fin are isothermal. This confirms that the micro-channels tubes are operating as thermosyphons, leading to ideal fin behavior.

[0073] Figs. 15, 16 and 17 present summaries of the results for the experiments with the prototype charged with specific volumes of 0.007 m³/kg, 0.009 m³/kg, and 0.011 m³/kg, respectively.

[0074] An energy balance was carried out in order to quantify the system performance, and the results are shown in Fig. 18. It can be seen that, at the lower flow velocity, the heat transfer with the thermosyphon effect is significantly higher by as much as 63%. Therefore, the micro-channel fins produced significant improvement to the classical fin performance and the heat exchanger performance improved by up to 63%. The loop thermosyphon configuration is especially applicable to the heat sink system where the evaporator of the loop thermosyphon can be placed on the heat source and the fins can be used to dissipate the heat in the air.

[0075] The thermosyphon loop with the fins is shown in Figs. 1 and 6. The complete loop is 292 mm high, and the three copper fins (with micro-channels inside) are required to be vertically oriented for proper functioning of the thermosyphon. The Fig. 1 drawing forms the basis of the fabricated thermosyphon loop shown in Fig. 7.

[0076] The three fins comprise the condenser section of the loop thermosyphon. Each fin is 190 mm high and 13 mm wide with nine flow channels inside. When the loop is operated in the thermosyphon mode, the fins are expected to be approximately isothermal, and the heat transfer will be enhanced.

[0077] The rest of the loop (shown in Figs. 1, 6 and 7) is made of copper tubing, including the evaporator section at the bottom. The evaporator is heated by a heating coil. In an actual heat sink application, the evaporator geometry should be modified so that it can be an integral part of the heat spreader, such as a computer processor. For example, the evaporator can be thermally linked to a computer process, such as by adhesive,

soldering, or by forming channels through the computer processor. Thus, the term “thermally linking” means a connection that permits thermal energy to pass from a warmer object to a cooler object.

[0078] The highest pressure used for charging was 4.2 MPa, which gave a specific
5 volume of 0.007 m³/kg. When the thermosyphon is operated at about 50°C with this specific volume, the pressure is about 1.7 MPa, and the system has an overall quality of 20%, i.e., 80% of the mass is liquid. However, the liquid density is 14.5 times the vapor density under these conditions, so the volume of the liquid is 22% of the total loop volume. This is a rather large liquid volume fraction, so dry out of the evaporator is not
10 expected to take place. However, partial flooding of the condenser (accompanied by single phase convection) is possible, which would reduce the heat transfer.

[0079] In order to test the performance of the thermosyphon-fin combination, the wind tunnel shown in Fig. 8 was used. The cross section for the wind tunnel is 25.4 mm x 222 mm. This cross sectional area provides approximately 12 mm on each side of the
15 three micro-channel tubes. A flow meter was attached to the wind tunnel and the air flow to the wind tunnel is provided from a pressurized air line through the flow meter.

[0080] A flow mixer was placed after the thermosyphon so that an average temperature of the air flow can be measured at the outlet of the wind tunnel. The heat loss from the flow to the walls of the wind tunnel was estimated and found to be negligible,
20 particularly because of the small increase in the temperature of the air flow.

[0081] The measurements of primary interest during the experiment are the temperatures along the length of the three micro-channel tube fins and the air inlet and outlet temperatures. Nine K-Type thermocouples were used to measure the temperature along the length of the copper micro-channel fins; three thermocouples were placed at the
25 bottom, middle and top of each tube. The thermocouple locations are shown in Fig. 21. These thermocouples were used to monitor and record the temperatures as a function of time until a steady state was reached.

[0082] The temperature at the bottom of the three micro-channel tube fins can be taken to be the evaporator temperature since they were placed next to the junction of the
30 micro-channel tube and the top of the evaporator section. During the experiments, it was observed that the bottom thermocouples on the downstream side were slightly cooler, probably due the interaction of the airflow and the system geometry. However, all three fins were close to isothermal when the thermosyphon reached steady state; so the full

lengths of three micro-channel tubes were approximately at the temperature of the evaporator.

[0083] A set of matched thermocouples was carefully selected and a comparative calibration check was performed over the temperature range of 25°C to 75°C. Special emphasis was given to the nine thermocouples that measure the temperature of the micro-channel tubes, and the thermocouples that measure the inlet and outlet air temperatures. It was determined that over the calibration range of 25°C to 75°C, the thermocouples that were to be used at the air inlet and outlet differed in their readings by no more than 0.06°C; and in the expected range of operation (~25°C), they differed by approximately 0.01°C.

[0084] The thermocouples selected for the micro-channel tubes compared well in the calibration, with their differences having a standard deviation of 0.2°C in the range of 25°C to 75°C; and only 0.1°C when the upper limit was 50°C. Therefore these thermocouples were almost identical in their output values, and they provided a good evaluation of the temperature variations along the length of the micro-channels. Because the micro-channels were nearly isothermal, the matched sets of thermocouples showed almost identical values.

[0085] Detailed experimental results are presented herein for runs with the system charged at two different specific volumes (0.007 and 0.011 m³/kg), two different air flow speeds (0.5 and 1.0 m/s), and temperature of approximately 50°C at the evaporator (the actual temperatures of the micro-channels are shown in tables of results).

[0086] Additional experiments were also carried out at several specific volumes and at higher evaporator temperatures (65°C to 74°C). The main aim of the additional experiments was to test the design for performance stability, and to check for dry out conditions which can occur at higher temperatures. Because of the high liquid fraction at 0.007 m³/kg, tests did not show any dry out behavior at this specific volume. In fact, the results are indicative of flooding of the condenser section. However, some of the tests appeared to show that, within the established operating range, some flow blockage (and reduced heat transfer) can sometimes occur. These were probably due to bubbles that occasionally get entrenched at the junction where the micro-channel tubes are soldered to the copper tubes. Therefore the micro-channel thermosyphons should be fabricated with special focus on the quality of the joints.

[0087] A typical temperature history is shown in Fig. 22, which is obtained with 0.007 m³/kg of propane and air flow of 0.5 m/s. In the steady state, the nine thermocouples that measured the copper micro-channel temperatures (marked “prototype”) showed an average temperature of 53.7°C. The air inlet temperature was 22°C, and the outlet temperature was 24.8°C.

[0088] The thermosyphon behavior is clearly observed. The thermocouples attached to the micro-channel tube fins indicated a nearly uniform temperature throughout the heat up period and steady state, with a standard deviation of 0.2°C at the steady state (~53.7 °C). The evaporator temperature is not well defined because of the heater attached to this section. However, as discussed earlier, the average temperature of the nine thermocouples can be considered to be representative of the evaporator temperature, which is at the bottom of the fin, since the fins were almost isothermal. The steady state values are shown in Fig. 23. With the evaporator temperature at about 50°C, the micro-channels seem to behave more isothermally at the lower flow rate, but the standard deviations are less than 1% of the evaporator temperature for both flow rates. With the evaporator above 70°C, the standard deviations are higher, but dry out was not detected even at this higher temperature.

[0089] In a specific run, the temperature values on each fin were more uniform than indicated by Fig. 23. This can be observed in Fig. 24, which shows the results for an evaporator temperature of approx. 54°C, and an airflow of 0.5 m/s. The thermocouples on the micro-channel tube fin in the front shows the most deviation. This is probably due to the fact that this fin experiences the full cooling effect of the incoming air. The second and third micro-channel tube fins have a temperature variation of 0.1°C, which is the standard deviation observed in the calibration process.

[0090] The specific volume of 0.007 m³/kg corresponds to high liquid volume fraction. Therefore, additional runs were carried out, with the lowest specific volume at 0.011 m³/kg, which corresponds to liquid fraction of 11% of the total thermosyphon loop volume. Fig. 25 shows the temperatures at the same locations as in Fig. 24. However, the evaporator temperature for this run is lower since the steady state system is determined by multiple factors and the steady state evaporator temperature could not be set a-priori. The variation in the temperature is higher, and the bottom thermocouple (on top of the evaporator) also shows a greater decrease from front to the back of the fin array.

[0091] It should be noted that the variations within a single fin are 1°C or lower. When all the top and middle temperatures of the fins were compared with the bottom temperature, the correlation curve had a slope close to unity, confirming that each fin was very nearly isothermal.

5 [0092] Fig. 26 summarizes results for this lower liquid volume fraction (11%) of propane in the thermosyphon loop. To compare the heat transfer rates of the thermosyphon tube fins with a simple fin, additional experiments were carried out after the propane was removed from the loop, so that the copper micro-channel fins worked like classical fins. Fig. 27 shows the resulting temperature history, and even though a
10 reasonably steady state is attained, the temperature of the fins varies along the length as well as the location of the fin in the air flow path.

[0093] The heat transfer results of the tests are summarized in Fig. 28, where the data are shown for two different air velocities and three different propane charges in the loop: (a) Sp. Vol.=0.007 m³/kg, and (b) Sp. Vol.=0.011 m³/kg (c) no propane in the loop
15 (classical fin situation). The table shows (i) the heat transferred to the air flow, (ii) the coefficient of convective heat transfer (h), and (iii) the improvement in the heat transfer over the simple fin case (no propane in the thermosyphon). The improvement is calculated by taking the heat transfer for each of the two cases with propane (a) and (b), and dividing by the heat transfer result of (c) without propane.

20 [0094] From Fig. 28, it can be seen that the thermosyphon loop had better performance when the propane density (or liquid volume fraction) was lower. But the difference was less at higher air flow rates. In comparison with the case when the micro-channel tubes worked as a simple, classical fin (without propane), the heat transfer increased by 45% to 63%, depending upon the amount of propane used in the loop, and
25 the air flow rates. This should not be taken as an accurate measure of the enhancement over classical fin, since these fins are partly hollow. The improvement with higher specific volume of 0.011m³/kg (lower liquid fraction in the thermosyphon loop) is an indication that the condenser section was probably partially flooded due to higher volume fraction of liquid at the lower specific volume of 0.007 m³/kg. However, dry out
30 phenomena was absent in all the experimental runs.

[0095] The experiment tested an integrated thermosyphon-fin system by embedding the condenser section of the thermosyphon within a set of copper micro-channel tubing used as fins. Propane was used as the working fluid for the thermosyphon,

with the liquid volume fraction quantified by determination of the specific volume. The fins were shown to be nearly isothermal due to the thermosyphon effect, and significant improvement in fin performance was demonstrated. The performance improved when the liquid volume fraction was reduced from 22% to 11% of the thermosyphon loop volume, with condenser flooding as a possible cause for the difference. Since specific volume (or liquid volume fraction) of a thermosyphon can be measured at the superheated state, this approach can be very useful in designing the thermosyphon to operate in an optimum range between the extremes of flooding and dry out.

[0096] The design of an automotive radiator is another embodiment of the thermosyphon/heat pipe. The use of heat pipes or thermosyphons as fins can produce a new design of a radiator. The design can be further improved by using the microchannel fins as discussed below.

[0097] The solution to the problems of the prior art is to use thermosyphons or heat pipes such that the thermosyphons will be immersed in a radiator chamber where the coolant flows across the outside the thermosyphon tubes. The top of the thermosyphon tubes will be exposed to airflow. As air flows over the thermosyphon tubes (Fig. 19), the heat flow from the radiator fluid will be transferred to the air. The thermosyphon tubes should be vertically oriented. If heat pipes are used, the tubes do not have to be vertically oriented. The thermosyphons need not be straight, and can be substituted by heat pipes if required. By using thermosyphons, the invention can increase the performance significantly beyond the 0.31 W/K-cm^2 and the radiator frontal area can be reduced.

[0098] As a special case, the microchannel fins described herein can be used to make the radiator, which conventionally uses a large number of traditional fins. Instead of the engine coolant flowing through the microchannel, the new design (Fig. 29) is thermally linked, such as by physically mounting it, to a thermosyphon that has its own working fluid in the microchannel fins. Such a system can be used to extract heat efficiently from the car radiator or other part of the car cooling system, and can be used in series or parallel with other similar or identical thermosyphons to increase the thermal transfer. There is also the possibility of recovering energy from car exhaust systems for improving energy efficiency.

[0099] This detailed description in connection with the drawings is intended principally as a description of the presently preferred embodiments of the invention, and is not intended to represent the only form in which the present invention may be

constructed or utilized. The description sets forth the designs, functions, means, and methods of implementing the invention in connection with the illustrated embodiments. It is to be understood, however, that the same or equivalent functions and features may be accomplished by different embodiments that are also intended to be encompassed within
5 the spirit and scope of the invention and that various modifications may be adopted without departing from the invention or scope of the following claims.

CLAIMS

- 1 1. A heat exchanger containing a working fluid within sealed sidewalls having internal
2 fluid passages that are not in fluid communication with an exterior of the sidewalls, the
3 sidewalls defining a fluid flow path through which the working fluid circulates passively,
4 the sidewalls including an evaporator portion and a condenser portion over which a
5 second fluid passes to transfer heat between the second fluid and the heat exchanger
6 without the second fluid mixing with the working fluid, comprising:
7 at least one of said portions including at least one fin having a plurality of internal
8 micro-channels formed through the length of said at least one fin's sidewalls,
9 wherein the working fluid flows through the micro-channels in said at least one fin
10 and the working fluid in the micro-channels is in a vapor phase and a liquid phase,
11 thereby resulting in substantially isothermal conditions along substantially the
12 entire fin.
- 1 2. The heat exchanger in accordance with claim 1, wherein said at least one of said
2 portions is the condenser portion.
- 1 3. The heat exchanger in accordance with claim 1, wherein said at least one of said
2 portions is the evaporator portion.
- 1 4. The heat exchanger in accordance with claim 1, wherein said at least one of said
2 portions is the condenser portion and the evaporator portion.
- 1 5. An improved two phase thermosyphon with sealed sidewalls having connected,
2 internal fluid passages that are not in fluid communication with the exterior of the
3 sidewalls, thereby defining a fluid flow loop through which a working fluid circulates
4 passively, the sidewalls including an evaporator thermally linked to a heat source to
5 receive heat and a condenser over which a second fluid passes to transfer heat between
6 the working fluid and the second fluid without the fluids mixing, the improvement
7 comprising:

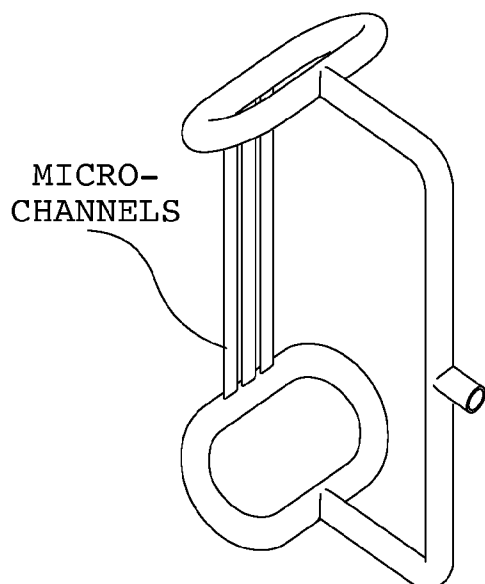
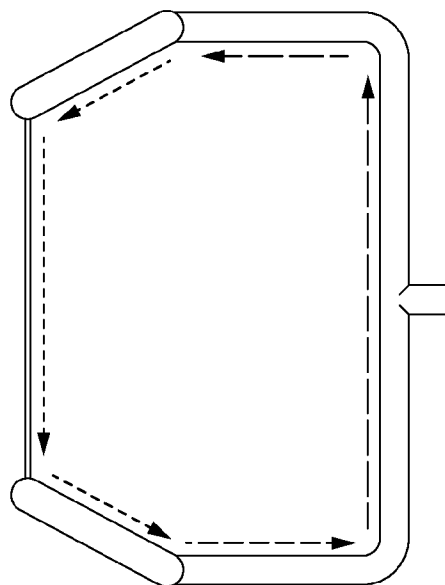
8 the condenser including at least one fin having a plurality of micro-channels
9 formed through the length of said at least one fin's sidewalls; and
10 wherein the fluid flow loop extends through the micro-channels in said at least one
11 fin and the working fluid in the micro-channels is in a vapor phase through a
12 substantial portion of the micro-channels' length and a condensed liquid phase in
13 a portion of the microchannels, the improvement resulting in substantially
14 isothermal conditions along substantially the entire fin.

1 6. The improved thermosyphon in accordance with claim 5, wherein the working fluid is
2 propane.

1 7. The improved thermosyphon in accordance with claim 5, wherein the sidewalls are
2 made of metal.

1 8. The improved thermosyphon in accordance with claim 5, wherein the evaporator also
2 includes at least one fin having a plurality of micro-channels formed through the length of
3 said at least one fin's sidewalls and wherein the fluid flow loop extends through the
4 micro-channels in said at least one fin and the working fluid in the micro-channels is in a
5 vapor phase through a substantial portion of the micro-channels' length and a condensed
6 liquid phase in a portion of the microchannels.

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**FIG-1A****FIG-1B**

Working Fluid	Melting Point (C) (p=1 atm)	Boiling Point (C) (p=1 atm)	Application Range (C)
Ammonia	-77.5	-33.1	-60 to 100
Pentane	-129.9	36.2	-20 to 120
Freon 113	-36.5	47.8	-10 to 100
Acetone	-93	56.4	0 to 120
Ethanol	-111.9	78.1	0 to 130
Methanol	-97.9	64.1	10 to 130
Water	0.1	100.1	30 to 200
Toluene	-94.9	110.1	50 to 200
Propane	-187.6	-42.1	Not Known

FIG-2

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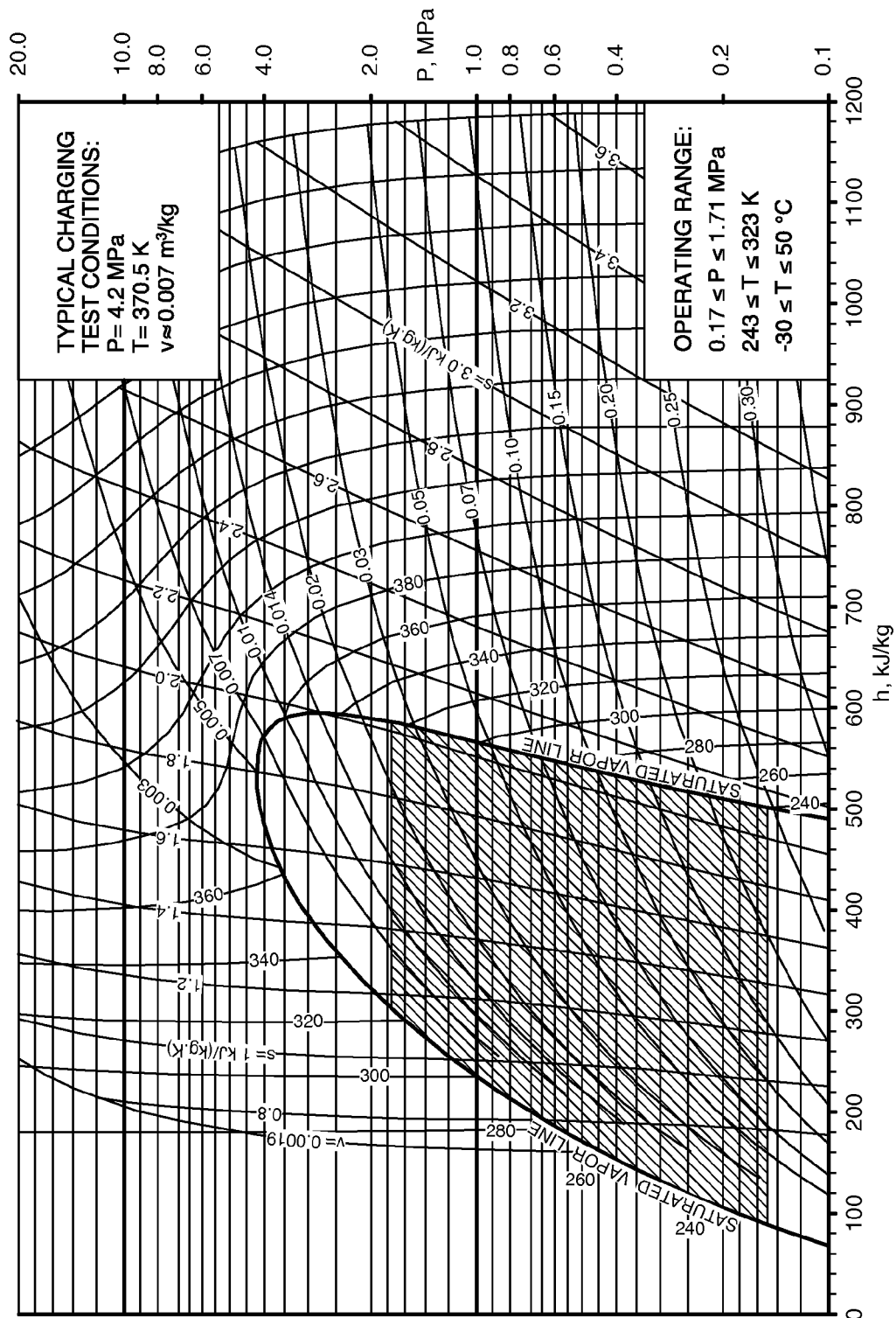
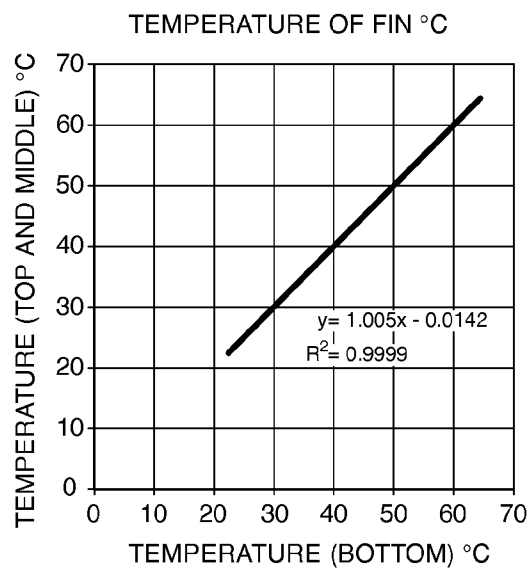
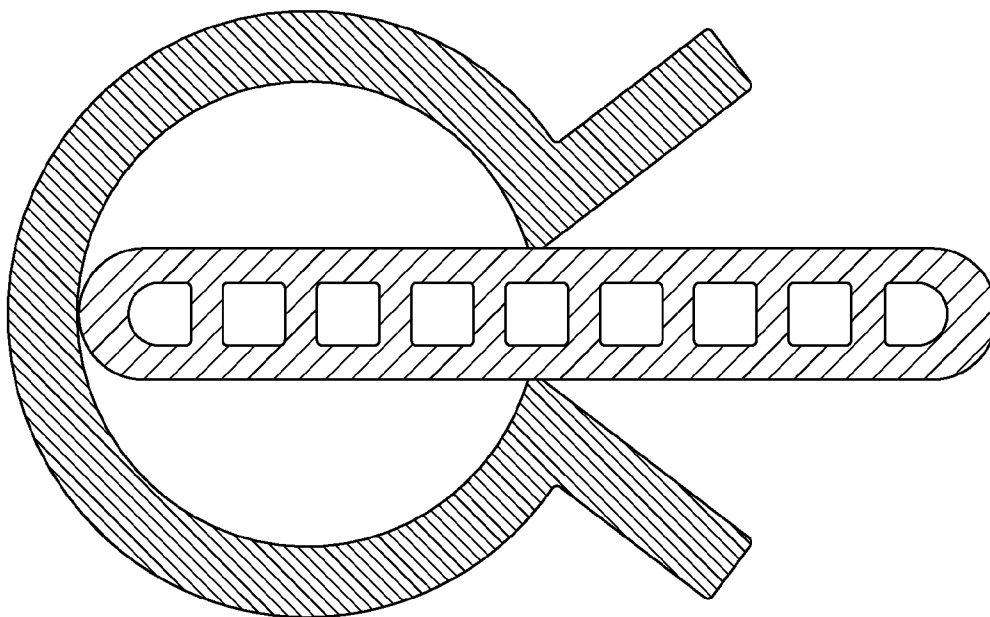


FIG-3

SUBSTITUTE SHEET (RULE 26)

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**FIG-4****FIG-5**

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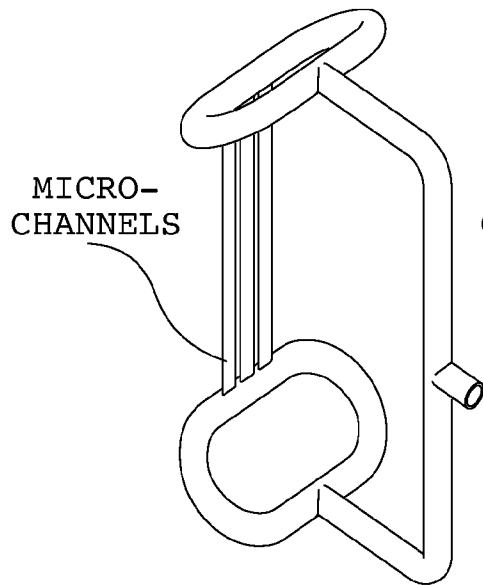


FIG-6

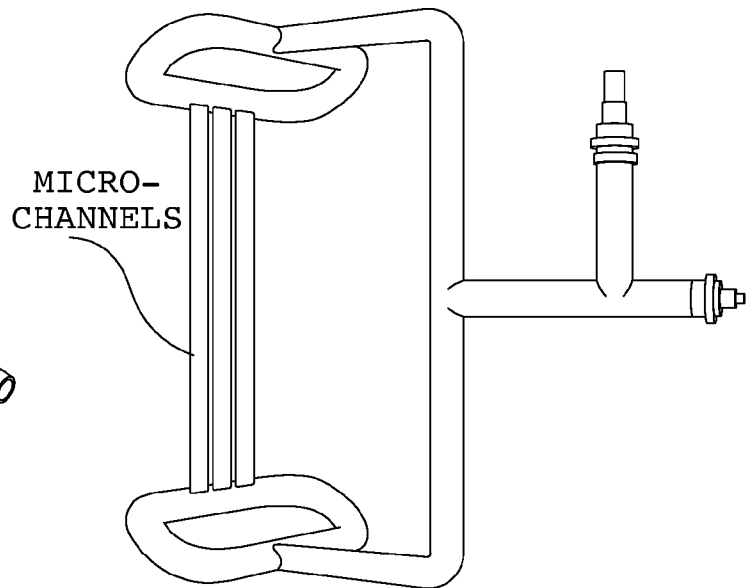


FIG-7

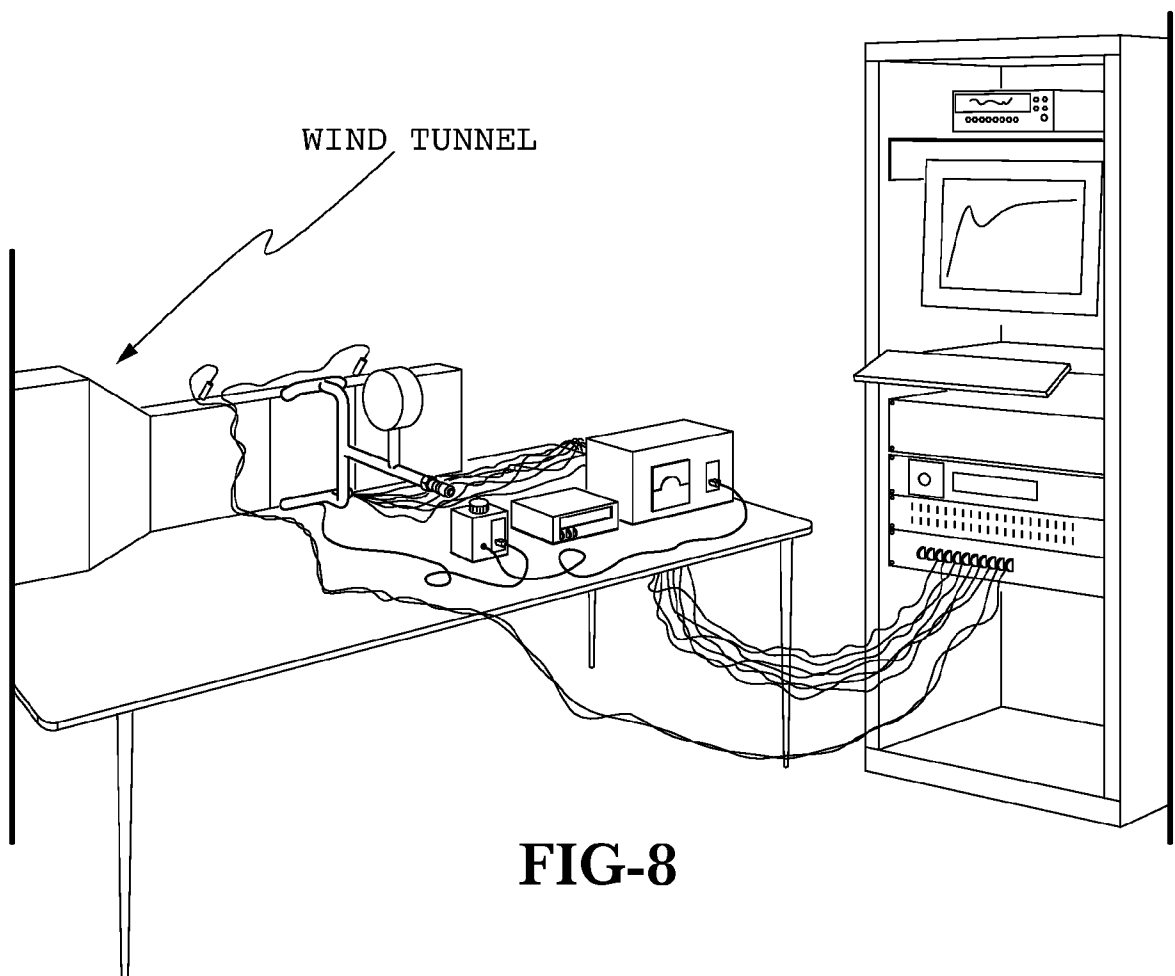


FIG-8

SHEET 5 OF 13

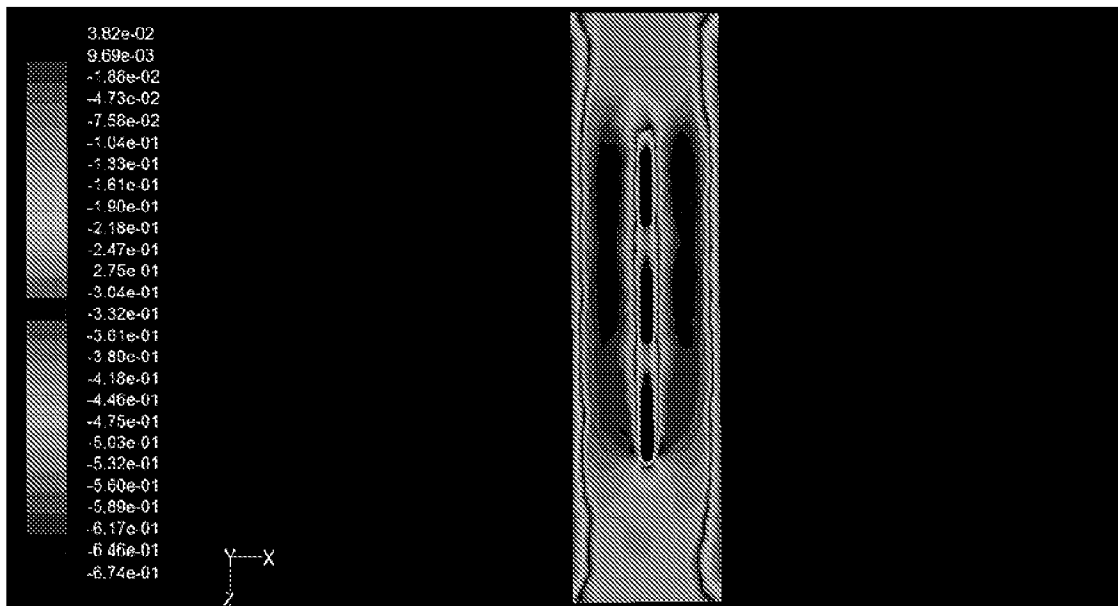


FIG-9

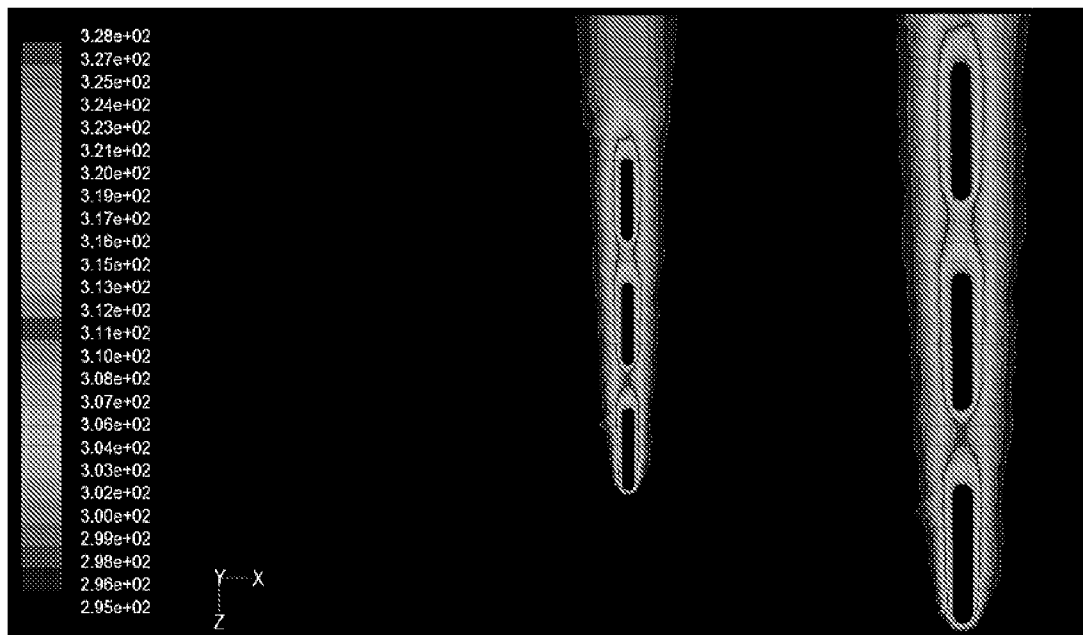


FIG-10

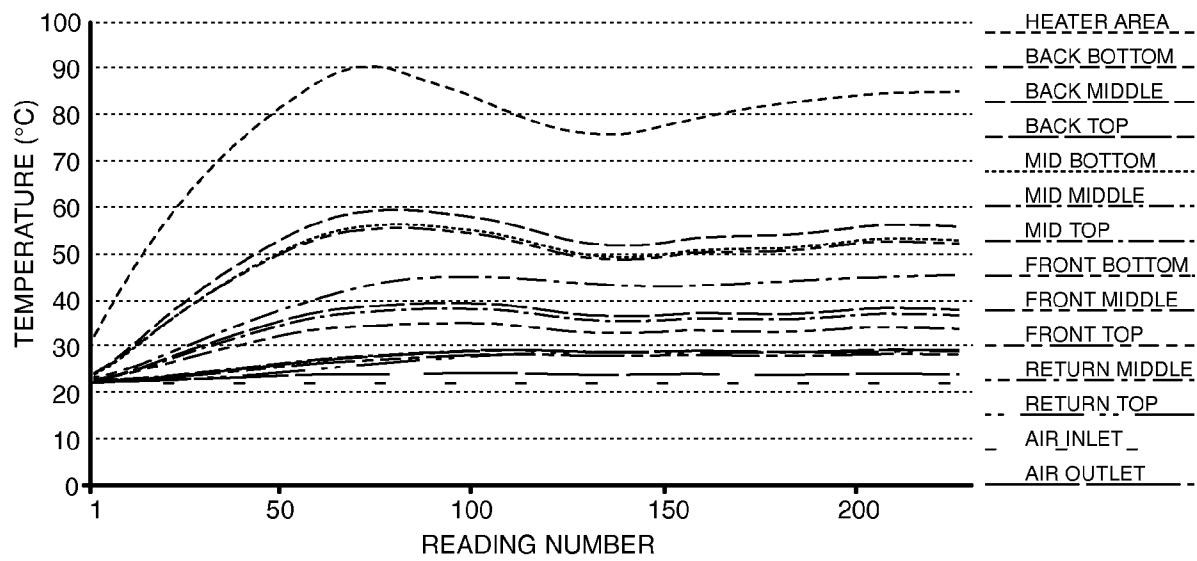


FIG-11

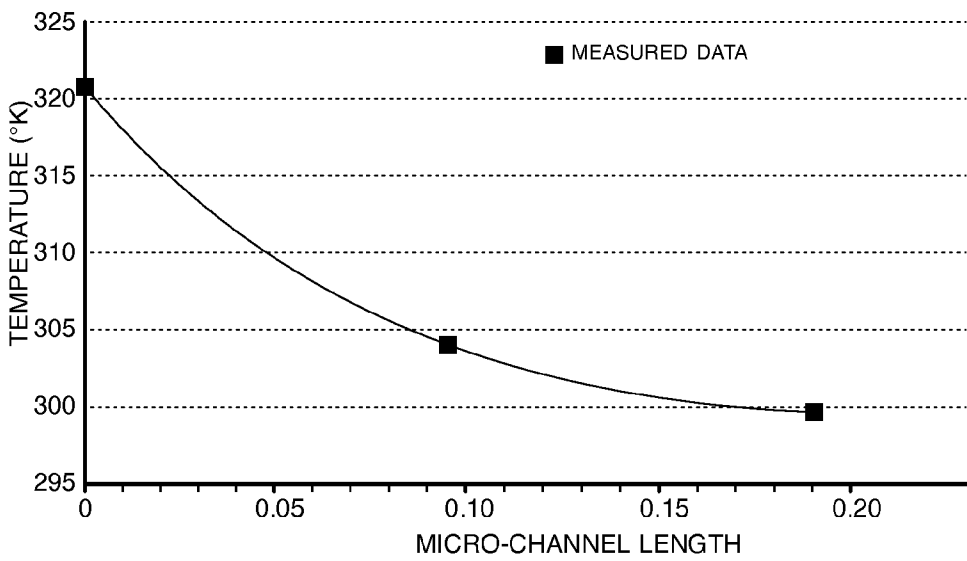


FIG-12

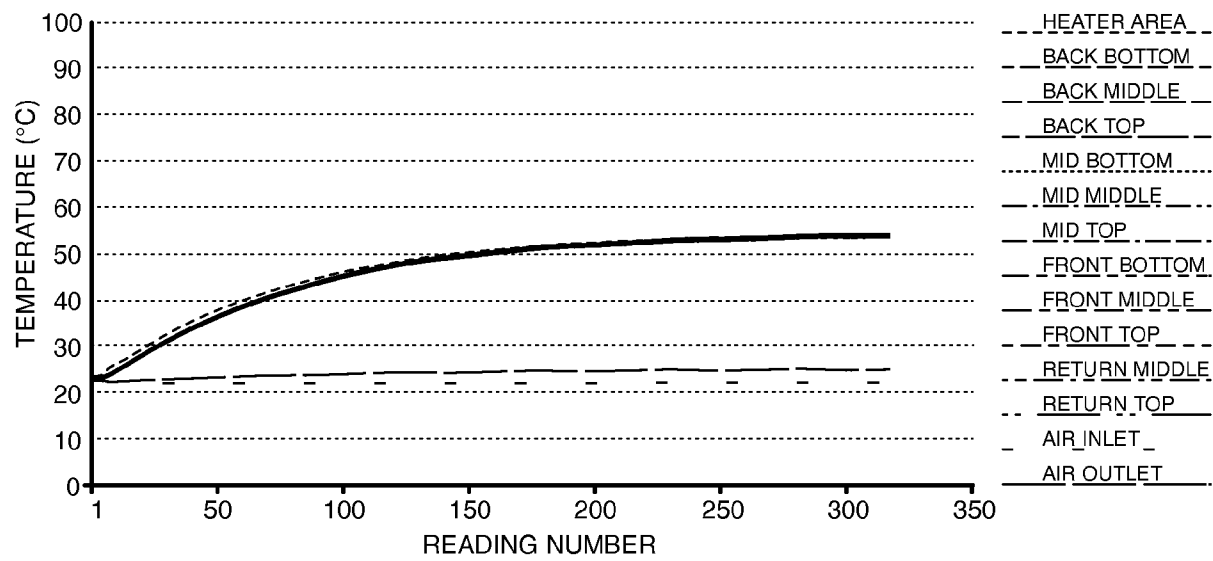


FIG-13

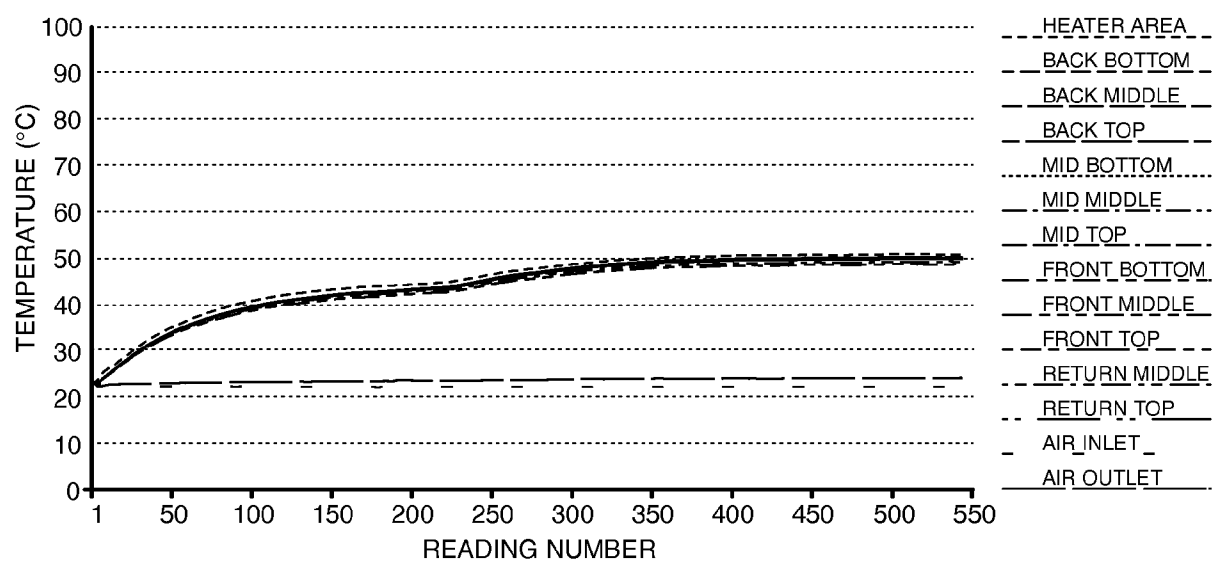


FIG-14

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Experimental Data (0.007 m ³ /kg)	Evaporator at -50°C (0.5 m/s)	Evaporator at -50°C (1.0 m/s)	Evaporator at -70°C (0.5 m/s)	Evaporator at -70°C (1.0 m/s)
Average prototype temperature (°C)	53.73	49.59	73.58	70.14
Standard deviation (°C)	0.17	0.51	1.19	1.27
Average air inlet temperature (°C)	22.00	22.14	22.29	22.03
Standard deviation (°C)	0.01	0.01	0.01	0.01
Average air outlet temperature (°C)	24.76	24.00	28.15	25.36
Standard deviation (°C)	0.09	0.02	0.12	0.05

FIG-15

Experimental Data (0.009 m ³ /kg)	Evaporator at -50°C (0.5 m/s)	Evaporator at -50°C (1.0 m/s)	Evaporator at -70°C (0.5 m/s)	Evaporator at -70°C (1.0 m/s)
Average prototype temperature (°C)	49.87	47.07	70.96	65.51
Standard deviation (°C)	0.98	1.11	2.32	1.94
Average air inlet temperature (°C)	22.52	21.92	22.67	22.02
Standard deviation (°C)	0.01	0.01	0.01	0.02
Average air outlet temperature (°C)	24.97	23.74	26.94	25.31
Standard deviation (°C)	0.08	0.01	0.08	0.05

FIG-16

Experimental Data (0.011 m ³ /kg)	Evaporator at -50°C (0.5 m/s)	Evaporator at -50°C (1.0 m/s)	Evaporator at -70°C (0.5 m/s)	Evaporator at -70°C (1.0 m/s)
Average prototype temperature (°C)	51.89	46.84	73.15	65.28
Standard deviation (°C)	0.66	0.83	1.12	1.02
Average air inlet temperature (°C)	22.40	21.91	22.41	21.92
Standard deviation (°C)	0.08	0.01	0.01	0.01
Average air outlet temperature (°C)	25.41	23.90	27.64	25.50
Standard deviation (°C)	0.05	0.02	0.08	0.06

FIG-17

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Changing Test Condition (m ³ /kg)	Air Flow Velocity (0.5 m/s)		Air Flow Velocity (1.0 m/s)	
	$\dot{E}_{air}$ (W)	Improvement Over Classical Fin	$\dot{E}_{air}$ (W)	Improvement Over Classical Fin
0.007	9.4	45%	12.7	52%
Standard deviation (°C)	8.4	29%	12.4	49%
Average air outlet temperature (°C)	10.3	62%	13.6	63%
No Propane (No Thermosyphn Effect) Classical Fin Case	6.5		8.3	

FIG-18

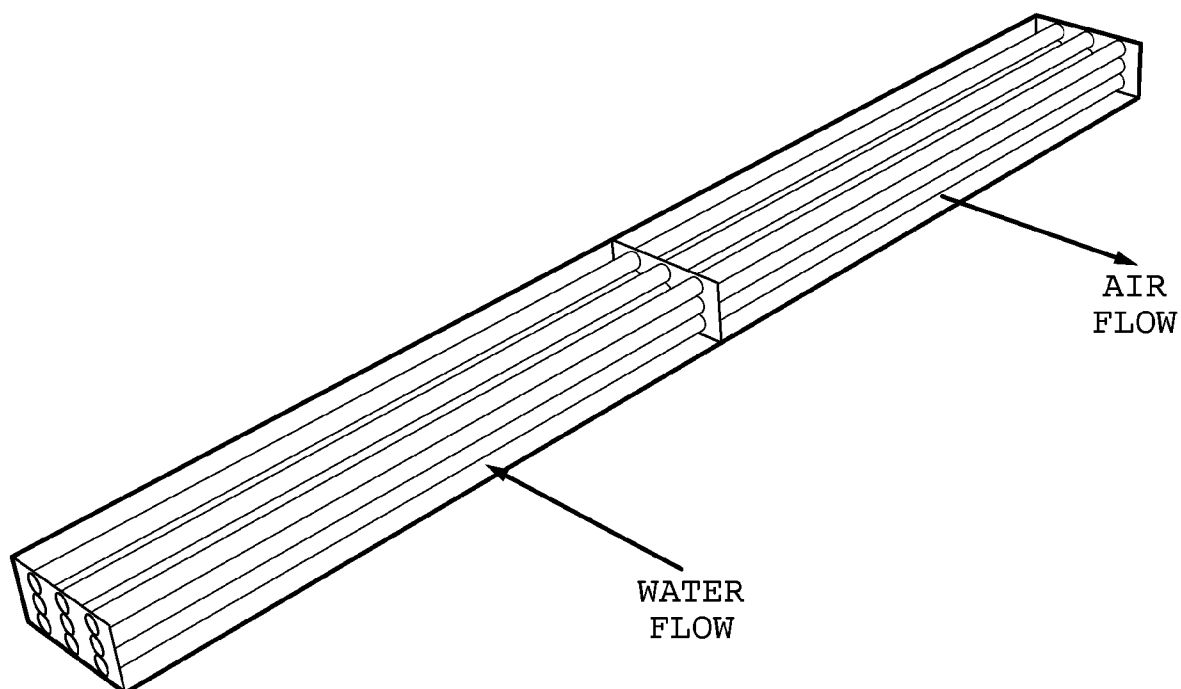


FIG-19

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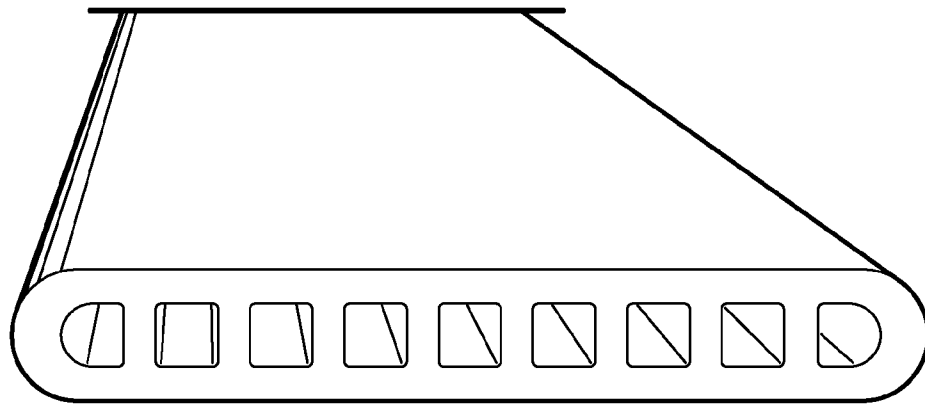


FIG-20

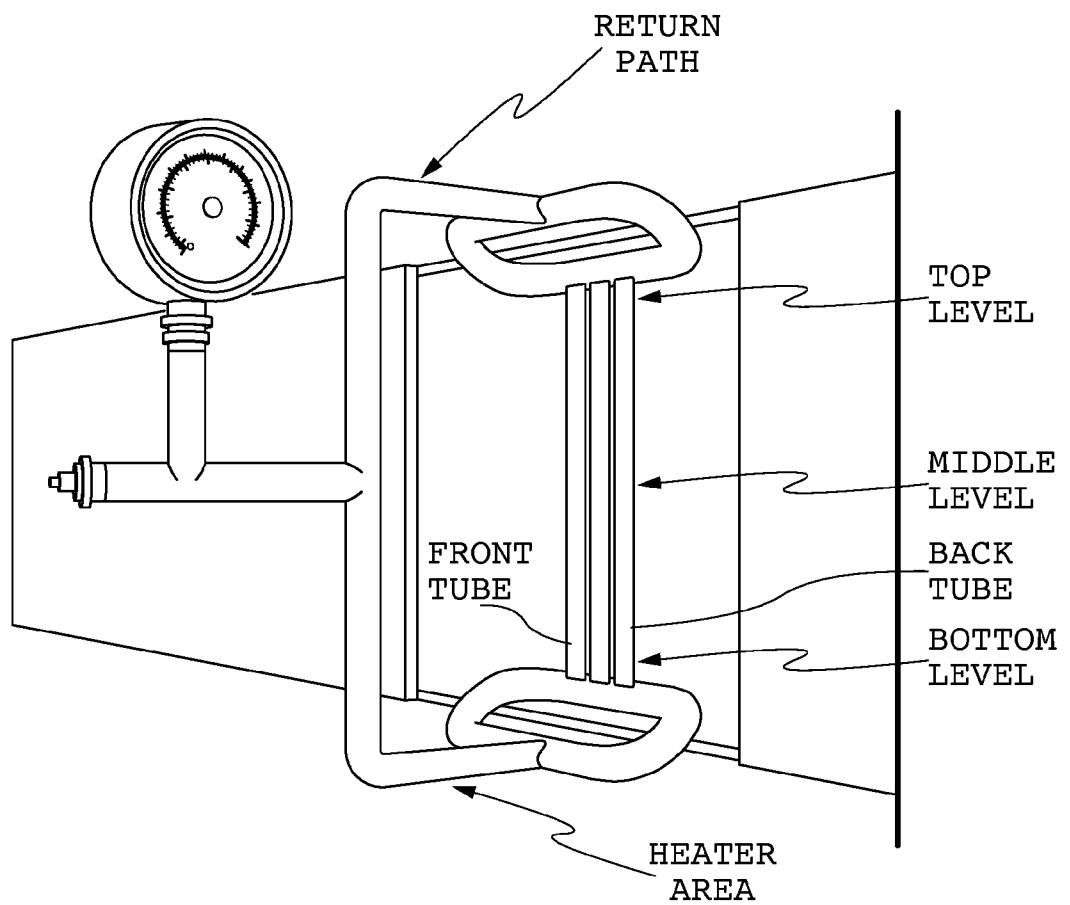


FIG-21

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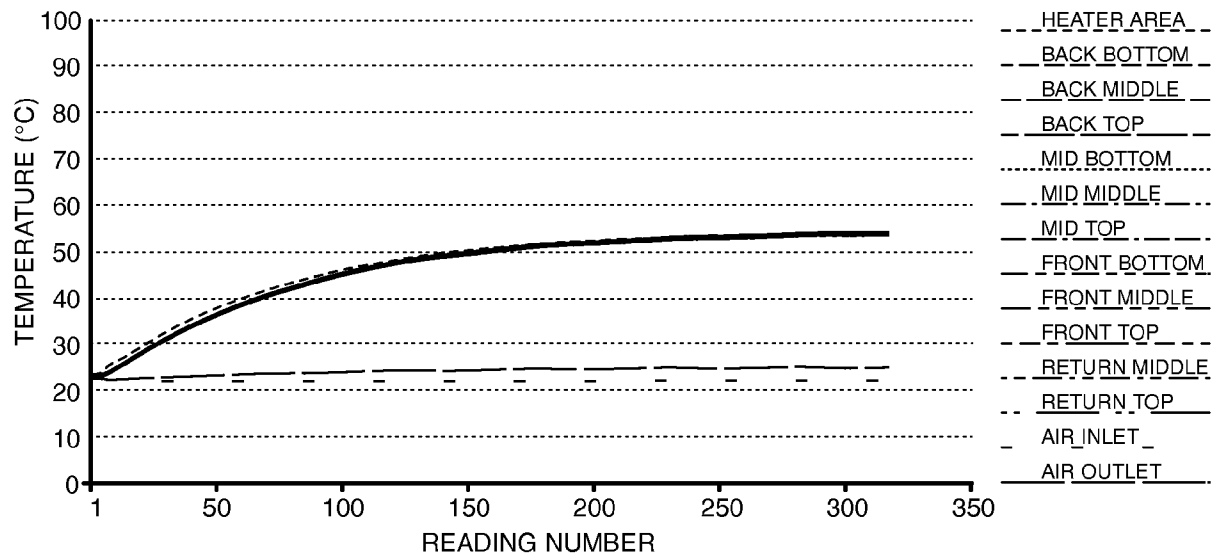


FIG-22

Sp. Volume of Propane 0.007m ³ /kg	Evaporator at ~50°C		Evaporator at ~70°C	
	Air Velocity 0.5 m/s	Air Velocity 1.0 m/s	Air Velocity 0.5 m/s	Air Velocity 1.0 m/s
Average Temperature of 9 Microchannels (°C)	53.7	49.6	73.6	70.1
Standard Deviation (°C)	0.2	0.5	1.2	1.3
Average Air Inlet Temperature (°C)	22.0	22.1	22.3	22.0
Average Air Outlet Temperature (°C)	24.8	24.0	28.2	25.4

FIG-23

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Location of Thermocouple on Micro-Channel Fin Tube	Micro-Channel Tube Location Along Air Flow Direction		
	Front	Center	Back
Top (°C)	53.6	53.7	53.8
Middle (°C)	53.6	53.7	53.7
Bottom (°C)	54.0	53.8	53.7

FIG-24

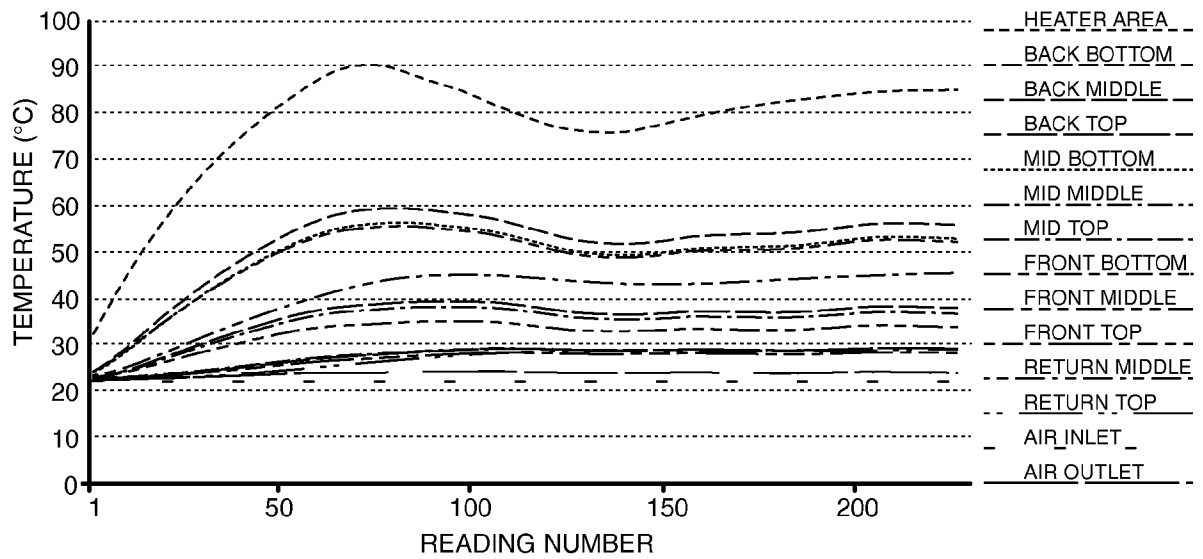
Location of Thermocouple on Micro-Channel Fin Tube	Micro-Channel Tube Location Along Air Flow Direction		
	Front	Center	Back
Top (°C)	51.8	51.1	51.4
Middle (°C)	51.8	50.7	52.5
Bottom (°C)	52.1	51.8	51.5

FIG-25

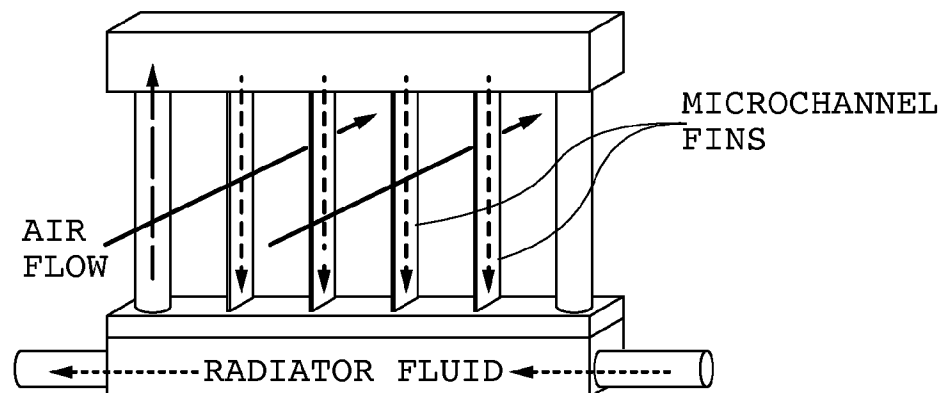
Sp. Volume of Propane 0.011m ³ /kg	Evaporator at ~50°C		Evaporator at ~70°C	
	Air Velocity 0.5 m/s	Air Velocity 1.0 m/s	Air Velocity 0.5 m/s	Air Velocity 1.0 m/s
Average Temperature of 9 Microchannels (°C)	51.9	46.8	73.2	65.3
Standard Deviation (°C)	0.7	0.8	1.1	1.0
Average Air Inlet Temperature (°C)	22.4	21.9	22.4	21.9
Average Air Outlet Temperature (°C)	25.4	23.9	27.6	25.5

FIG-26

SHEET 13 OF 13

**FIG-27**

Propane Sp. Volume (m ³ /kg)	Air Flow Velocity 0.5 m/s			Air Flow Velocity 1.0 m/s		
	Heat Transfer W	h W/m ² K	Improve- ment	Heat Transfer W	h W/m ² K	Improve- ment
(a) 0.007	9.4	20.7	45 %	12.7	32.0	52 %
(b) 0.011	10.3	24.5	62 %	13.6	37.8	63 %
(c) No Propane (simple fin)	6.5	n/a	n/a	8.3	n/a	n/a

FIG-28**FIG-29**

INTERNATIONAL SEARCH REPORT

International application No.

PCT/US2010/055285

A. CLASSIFICATION OF SUBJECT MATTER

IPC(8) - H05K 7/20 (2010.01)

USPC - 361/701

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

IPC(8) - H01L 23/427; H05K 7/20 (2010.01)

USPC - 165/80.3, 80.4; 257/715, E23.088; 361/699, 700, 701, 702, 703, 704

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

PatBase

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	US 2009/0056916 A1 (YESIN et al) 05 March 2009 (05.03.2009) entire document	1-8
Y	US 2002/0056908 A1 (BROWNELL et al) 16 May 2002 (16.05.2002) entire document	1-8

☐ Further documents are listed in the continuation of Box C. ☐

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