THE USE OF TRANSPIRATION IN A PRECISION TEMPERATURE-CONTROLLED ENCLOSURE

by

RICHARD THOMAS MASTANDUNO

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Abstract

Transpiration is a method of forcing a flow of gas (transpirant) through a porous material to promote higher heat transfer than that afforded by conduction and convection only. The resulting temperature distribution inside the porous material has been shown to be exponential, with a characteristic length dependent on material and transpirant properties and flow rate, in tests done on porous metals.

The exponential temperature distribution was expected to be very useful in attenuating changes in outside temperatures for constant temperature enclosures. In addition, the shape of the distribution implies that walls of a certain thickness could behave as near perfect insulators.

A simple theoretical model of a porous material during transpiration was developed, and an apparatus was designed and built to test its validity. Specifically, the goals were to confirm the expected exponential temperature distribution within a wall made of insulating material and its subsequent attenuation of step changes in outside temperature occuring outside a constant temperature enclosure.

The data from the tests done appear to confirm the exponential nature of the wall temperature distribution, but the numerical value of the characteristic length thus obtained was not found, due to the fact that the permeability of the samples was not measured. Recommendations for an improved apparatus based upon what was learned from the present design are made.

Thesis Supervisor: Dr. Michael Cleary

Title: Associate Professor of Mechanical Engineering

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Introduction

Maintaining a constant, uniform temperature in an air-filled enclosure is often necessary, since many devices and processes are affected by temperature. For example, measurements in precision machining processes must be done at constant temperatures, since metals (and all other materials) change dimensions with changes in temperature. The exact value of the temperature at which the inside is kept may be critical in some (e.g., biological) applications, but often the variation about a set point is all that is important. Here, commercial systems (constant temperature rooms) are available to maintain most sized enclosures at a set point ±2.5°C[1]. These systems generally consist of heating and/or cooling coils, a system for mixing air in the enclosure, and a control circuit to regulate the amount of heating or cooling. More precise systems are available to regulate water baths (accuracy and uniformity better than ±0.1°C)[2], but this is a much easier task: since the heat capacity of a given volume of a substance is proportional to density, the loss of a certain amount of energy has a thousand times smaller effect on temperature if the enclosure is filled with water than if it is air-filled.

A good criterion for evaluating constant temperature enclosures is the ability of a given system to compensate for

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[1] "top of the line" models (courtesy of Fisher Scientific Co.) of ovens and incubators claim accuracy and uniformity values of better than ±1.5°C.

[2] courtesy of Blue M Co., Blue Island, Illinois.

different types of thermal disturbances (see Figure 1):

1. heat generated within the enclosure. This may be a result of motors, electronic circuitry, or any biological, chemical, or physical process occuring within the enclosure. This may be a constant heat input (as in a motor mounted inside to provide air circulation), or a step input (as in heat generated during an experiment carried out inside).
2. outside air or objects entering the enclosure. This may occur constantly, as in the case of leaks in a sealed enclosure, or at certain times, if the enclosure is opened to manipulate or insert material.
3. changes in outside air temperature or velocity. This may change the heat transfer occuring at the outside surface, so that if heat transfer from the enclosure to the environment is being used as a means of providing heat or cold in response to a temperature controller signal, altered external heat transfer will change controller effectiveness (see below for more discussion of this). Even in a sealed room, room temperature varies cyclically over the course of a day, with a larger cycle occuring over the course of a year.

A variety of methods may be devised to maintain a constant, uniform temperature in the face of the above disturbances. Figure 2 lists a selection of methods varying in complexity, cost, and ability to respond to different types of disturbances.

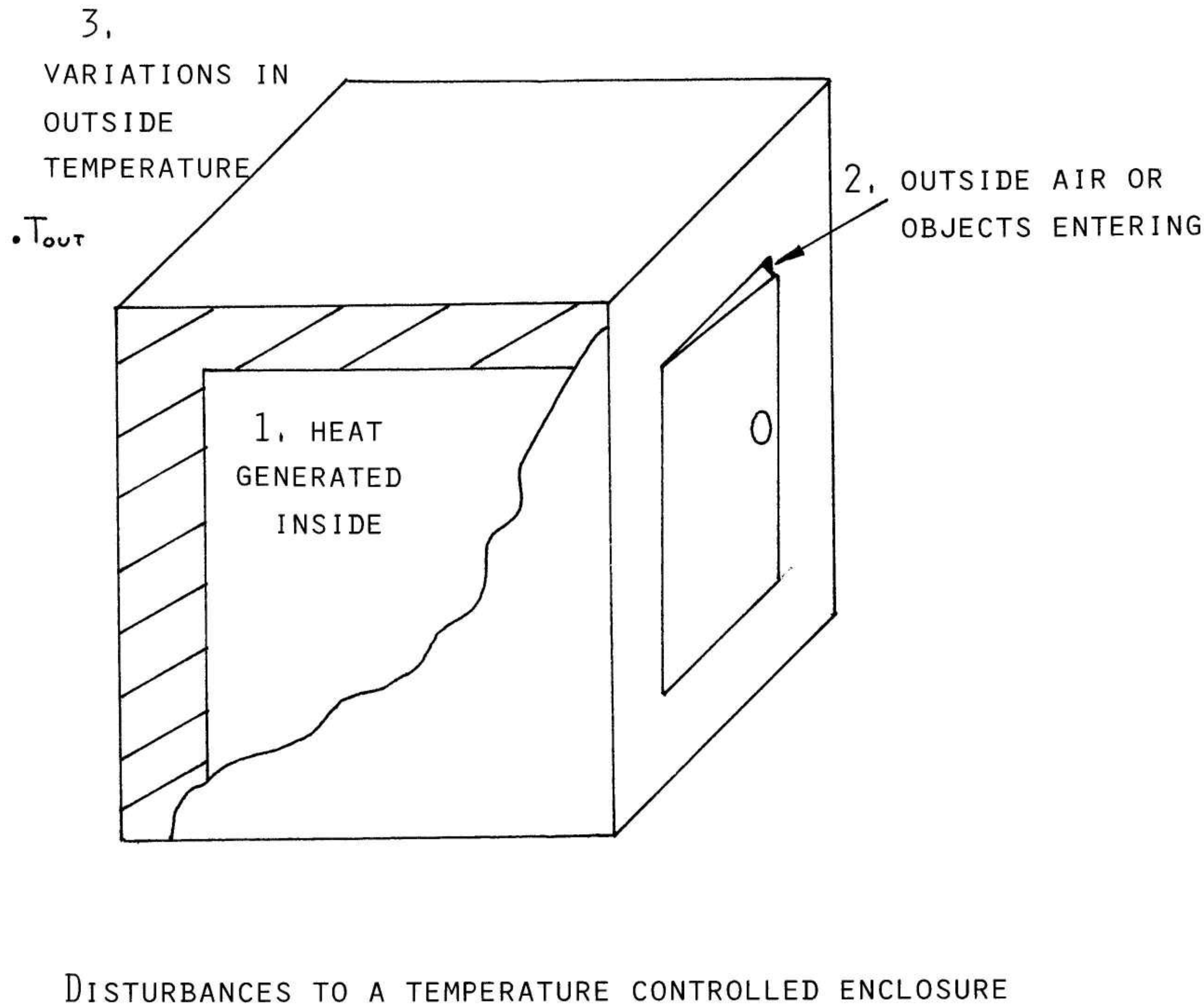


Figure 1: Disturbances to a temperature controlled enclosure

METHODS OF TEMPERATURE CONTROL

1. NO CONTROL, HEAVY INSULATION
   - HEAVY INSULATION DAMPS OUT SHORT TERM DISTURBANCES
   - LONG TERM DISTURBANCES HAVE AN EFFECT
2. CONTROL OF BOTH HEATING AND COOLING
   - COMPLEX
   - FAST RESPONSE
   - LOW POWER POSSIBLE
3. CONTROLLED HEAT FLOW IN ONE DIRECTION ONLY WITH
   - A. WELL-INSULATED ENCLOSURES
     - SLOW RESPONSE TO CONTROLLER OVERSHOOTS
   - B. STABLE COOLING LOAD THROUGH CONDUCTION
     - NONUNIFORMITY CAUSED BY BUILDUP OF BOUNDARY LAYER NEAR WALLS
     - HIGHER POWER REQUIRED
   - C. STABLE COOLING LOAD THROUGH UNCONTROLLED COOLING COILS
     - HIGHER POWER NEEDED
     - HIGH COOLING LOAD MAY CAUSE CONTROLLER OSCILLATIONS
   - D. STABLE COOLING LOAD THROUGH INCOMING STREAM OF AIR EXITING THROUGH AN OUTLET PORT
     - NONUNIFORMITY NEAR OUTLET
     - NONUNIFORM LEAKS MAY CAUSE TEMPERATURE NONUNIFORMITY
   - E. STABLE COOLING LOAD THROUGH INCOMING STREAM OF AIR EXITING THROUGH POROUS WALLS
     - NEAR PERFECT INSULATION
     - ALL DISTURBANCES QUICKLY SWEPT AWAY
     - LOW POWER REQUIRED
     - UNIFORM EXIT OF AIR PROMOTES UNIFORM TEMPERATURE INSIDE

Figure 2: Selection of temperature control strategies.

The first method, an uncontrolled, heavily insulated enclosure, has the advantage of extreme simplicity, good attenuation of short term external disturbances, and no energy cost for operation at ambient temperature. In addition, the use of external uncontrolled (constant power) heaters or cooling coils can allow its use at other temperatures. The major problem with this approach is very slow correction for internal disturbances (like 1 and 2, above), since the heavy insulation which attenuates short time external disturbances also prevents rapid heat transfer needed to allow these disturbances to decay quickly. Other problems with this approach are variations in internal temperature with long-term changes in outside temperature, and the bulkiness of providing enough insulation to attenuate outside disturbances.

The second method, control of heat loss and gain, has the advantage of quick response to internal disturbances, since both heating and cooling are available whenever the controller senses a disturbance. The major problems with this approach are the complexity of hardware required, especially in providing an easily controlled cooling source, and the susceptibility to external disturbances if insulation is not heavy enough at all points.

The third method attempts to reduce some of the complexity of the above approach by providing heat controlled heat flow in one direction only. In the first case (3a), the enclosure is well-insulated to provide attenuation of short-term external disturbances. This has the same disadvantage as the well-insulated enclosure discussed above — internal disturbances take a long time to decay, since they must do so by conduction through well-insulated walls. This can be important if the heater controller overshoots the set point, as is the case if an underdamped control circuit is used.

To allow quick response to internal disturbances, different methods of heat transfer may be used. The following discussion will be restricted to providing a stable cooling load for a controlled heater, but the same principles could be applied if the cooling source were controlled.

One way to provide a fairly stable cooling load is to allow conduction through moderately insulated walls (3b). This allows fairly quick heat loss when needed, but there is an increase in steady-state power required, as is the case for all the methods of providing a constant cooling load discussed below. The major problem, though, is that although the control point of the enclosure may be kept very stable, variations in the air flow patterns within the enclosure, especially as related to the buildup of a boundary layer near the walls, can cause nonuniformity which the controller cannot affect (since it only senses the temperature at its control point). Very good mixing is the only way to prevent this, but its implementation is very difficult for an enclosure with an arbitrary amount of objects inside.

Since good mixing is hard to achieve, it appears that a better way to promote uniformity is to provide effective insulation so that the cooling source is more localized and therefore easier to keep mixed uniformly. Of course, more effective insulation brings the same slow response if the enclosure relies on conduction.

To provide fast response and uniformity, uncontrolled cooling coils may be used with effective insulation(3c). This has the advantage of better localized mixing, but adds the expense and complexity of cooling coils. Also, a high power cooling coil (needed if very fast cooling is desired) may cause large oscillations in heater power as the control circuit tries to maintain constant temperature at its control point. This may cause nonuniformity if the controller and mixing system are not properly designed.

Instead of using cooling coils, outside air may be used as a cooling means if the enclosure set point is always above ambient temperature. Outside air can be admitted to the enclosure at a constant rate, heated, and mixed with the air already inside. This allows cooling by merely reducing heater power (3d and 3e).

One method for providing the discharge of air is to have an outlet port or a certain amount of leaks in an otherwise sealed enclosure. These methods have the disadvantage that they are somewhat susceptable to changes in outside temperature, depending on outside insulation, so large amounts of insulation to provide precise control. The major advantage of this approach, however, is that since there is a steady flow of air through the enclosure, any internal disturbance will be quickly convected away.

To take advantage of the quick removal of internal disturbances afforded by having a continuous air flow through an enclosure, and to try to reduce the amount of insulation required, a constant stream of outside air may be admitted and allowed to escape through porous walls. This approach is called transpiration, and is expected to have the advantage of near perfect insulation of the walls, caused by the "blanket" of exiting warm air, and the exponential temperature distribution caused by the heat exchange going on between the wall material and the outflow of air. Other advantages are expected to be no stagnant regions, since air is exiting at all points, and low power requirements if a careful choice of materials and flow rate can be made.

The purpose of this work will be to investigate the use of transpiration for precision temperature control. Specifically, the expected attenuation of step changes in outside temperature is to be investigated experimentally. The need for this precise (±0.01°C) control arises from a need to maintain an experimental apparatus made of a temperature-sensitive rubber at a constant temperature[7]. The device requires an enclosure of at least 0.2 m^3 volume, so that maintaining uniformity inside is a problem. Also, the room in which it is located is subject to daily temperature excursions of at least 5°C at certain times of the year. This makes rejection of outside temperature disturbances a major priority.

Analysis and advantages of transpiration

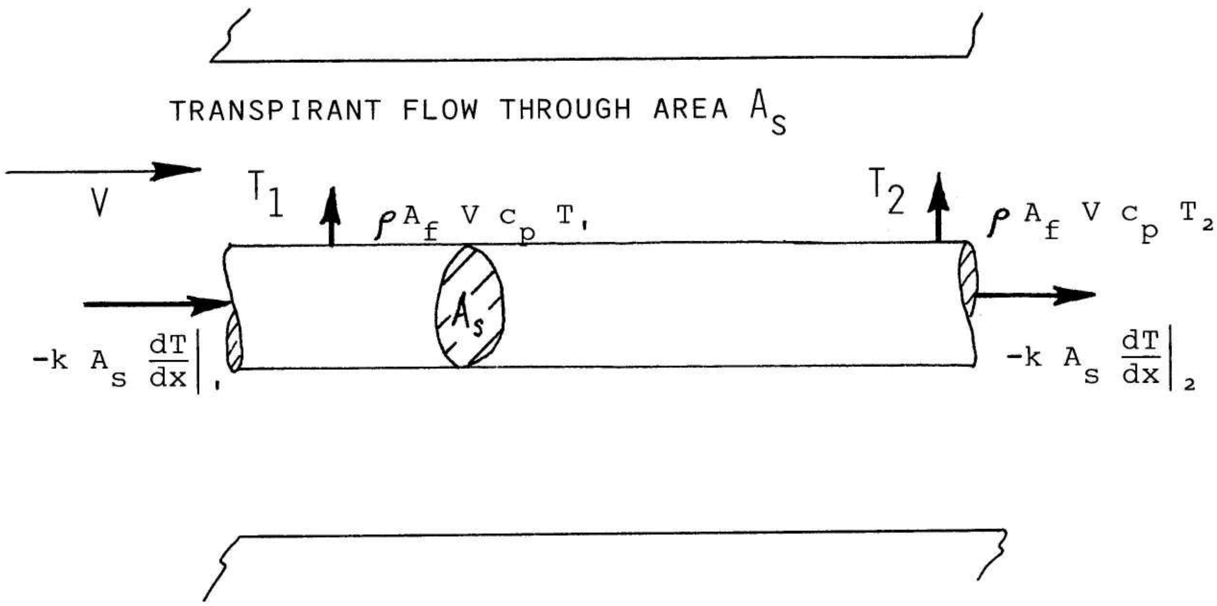
Transpiration cooling is the use of positive pressure to force a fluid through a porous material to carry away extra energy from its surface quickly. The technique has found its largest use in the aircraft industry, where cooling air is forced through specially designed blades which have channels to allow the high velocity air to exit from the leading edge surfaces. This technique has allowed operation of turbines at higher temperatures, since the "blanket" of transpirant around the blades can keep their temperature several hundred degrees below that of the exhaust gases surrounding them.

In the application here, transpiration will be used to provide an insulating "blanket" of air on the outside surface of a temperature-controlled enclosure. To understand why it is expected to be so useful in rejecting outside temperature disturbances, analysis of a simple model of the porous wall during transpiration was done. Figure 3 shows a small element of the material. The major assumptions made in this model are simple one-dimensional conduction through the thickness of the wall, and that the solid material of the wall and the transpirant are at the same temperature at each point within the cross section.

Equating the rate of energy increase of the solid, given by Fourier's law of conduction

$$Q_2 - Q_1 = -k\ A_s \left.\frac{dT}{dx}\right|_2 - -k\ A_s \left.\frac{dT}{dx}\right|_1 \qquad (1)$$

with the rate of energy decrease of the transpirant, given by



ASSUMPTIONS:

- AIR AN IDEAL GAS WITH CONSTANT SPECIFIC HEAT
- ONE-DIMENSIONAL HEAT FLOW ONLY
- CONDUCTION DOMINANT MODE OF HEAT TRANSFER
- TRANSPIRANT AND SOLID AT SAME TEMPERATURE AT EACH LOCATION

Figure 3: Element of a porous wall during transpiration

$$Q_2 - Q_1 = \rho A_f V c_p T\Big|_2 - \rho A_f V c_p T\Big|_1 , \qquad (2)$$

where Q_1 and Q_2 are the heat flows at locations 1 and 2, k is the thermal conductivity of the solid portion of the wall, A_s is the area of the solid portion of the wall, $\frac{dT}{dx}$ is the rate of change of temperature with respect to distance along the wall, A_f is the area of the void portion of the wall, V is the velocity of the transpirant within the wall, c_p is the specific heat of the transpirant (this assumes a constant specific heat for these temperatures), and T is the temperature at each location along the wall, gives

$$-k A_s \frac{dT}{dx}\Big|_2 + k A_s \frac{dT}{dx}\Big|_1 = \rho A_f V c_p T\Big|_2 - \rho A_f V c_p T\Big|_1 \qquad (3)$$

as the energy balance. If locations 1 and 2 are very close together, by the definition of the derivative, equation 3 may be written as

$$-k A_s \frac{d^2T}{dx^2} = \rho A_f V c_p \frac{dT}{dx} \qquad (4)$$

or

$$\frac{d^2T}{dx^2} + \frac{\rho A_f V c_p}{k A_s} \frac{dT}{dx} = 0 \qquad (5)$$

the solution to which can be shown to be

$$T - T_i = T_o e^{-\frac{x}{\lambda}} \qquad (6)$$

where T is the temperature at location x, T_i is the temperature at the inside surface of the wall, T_o is the temperature at the outside of the wall, and λ is the characteristic length of temperature decay for the wall, which is equal to $\frac{k A_s}{\rho A_f V c_p}$ and has units of length.

From equation 6, it can be seen that beyond a certain length, the temperature inside the wall is essentially at the final (inside) temperature, so inside temperatures are almost totally unaffected by the value of the outside temperature. In this sense, the walls act as almost perfect insulators. The wall thickness required to do this depends upon the characteristic length, which is a function of wall material properties, transpirant properties, and the velocity of flow through the wall. The limitations of this is that the velocity of the air (transpirant) flowing through the wall must be large enough to overcome the cold conduction through the wall, i.e. the assumption of the solid temperature being equal to the transpirant temperature breaks down at low flow rates. The other limitation is that the internal pressure (inside the box) must be at least as large as the typical external dynamic pressure. This is to ensure that the transpirant be forced out of the enclosure. These limitations put certain limitations on the required flow and temperature supplies for a transpiration controlled enclosure.

Another advantage of a transpiration controlled enclosure is that there are no stagnant regions within the enclosure, since internal air is constantly being forced out from all areas of the wall due to the positive pressure inside. This allows comparison to a well mixed, well insulated enclosure. If a step temperature change is applied to a this type of enclosure, the response can be given by equating the heat flow into the enclosure with the rate of change of energy within the enclosure. Using the same relations as for the temperature distribution within the wall,

$$-k\ A_{wall}\ \frac{\Delta T}{t} = m_{tot}\ c_p\ \frac{dT}{dt} \tag{7}$$

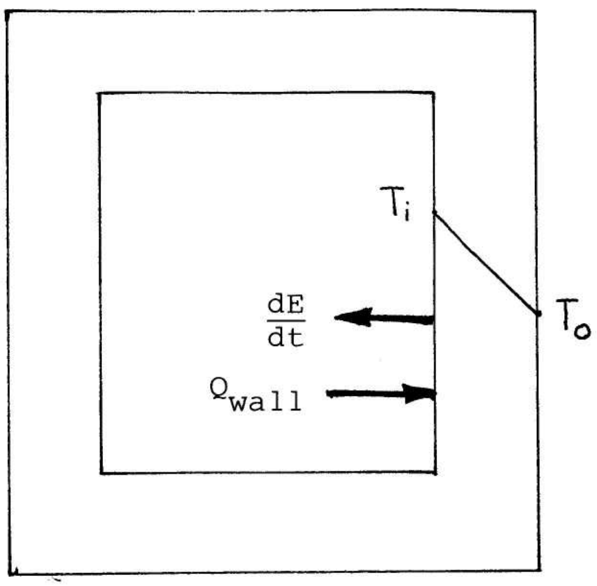
where k is the thermal conductivity of the wall, A_{wall} is the total surface area of the enclosure, ΔT is the temperature difference across the walls of the enclosure, t is the insulation thickness, m_{tot} is the total mass of air in the enclosure, c_p is the specific heat of air, and $\frac{dT}{dt}$ is the rate of change of temperature with respect to time. This energy balance is shown in figure 4, along with a comparison of the temperature distributions arising from steady conduction as compared to those occuring during transpiration.

The solution to equation 7 has exponential character, with a characteristic time equal to $\frac{t\ m_{tot}\ c_p}{k\ A_{wall}}$. This describes the time necessary for temperature equilibration. after a step change in outside temperature. For a transpiration controlled enclosure, the time for any disturbance to be carried away is about the time required for one complete air change in the enclosure, if the velocities are small and the flow is steady enough to be considered laminar. This is the "air curtain" principle used in clean room equipment to keep contaminants out of a work station using outward laminar flow. The time required for a complete air change is

$$t_{change} = \frac{Vol}{V\ A_{wall}} \tag{8}$$

where t_{change} is the time for an air change, Vol is the total volume of the enclosure, V is the exit air velocity, and A_{wall} is the total surface area of the enclosure. Equating this to the time constant to the conduction controlled enclosure,

$$t_{change} = \frac{Vol}{V\ A_{wall}} \ll \frac{t\ \rho\ Vol\ c_p}{k\ A_{wall}} \tag{9}$$



NET HEAT FLOW THROUGH WALL

$$Q_{wall} = -k\, A_{wall} \frac{\Delta T}{t}$$

(FOURIER'S LAW)

RATE OF CHANGE OF AIR'S INTERNAL ENERGY

$$\frac{dE}{dt} = m_{air}\, c_p \frac{dT}{dt}$$

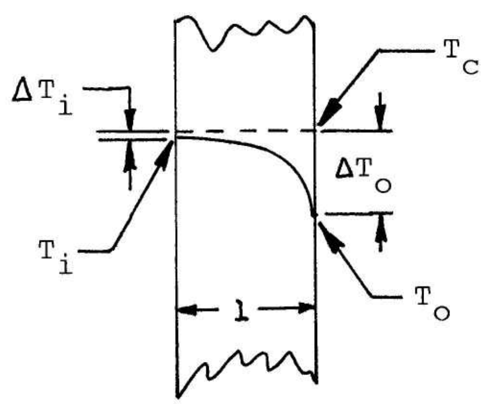
(IDEAL GAS LAW)

ASSUME

- AIR WELL MIXED (UNIFORM TEMPERATURE)
- NEGLECT HEAT CAPACITY OF WALL

CONDUCTION THROUGH WALL:

LINEAR TEMPERATURE DISTRIBUTION



$$\frac{\Delta T_i}{\Delta T_o} = e^{-\frac{l}{\lambda}}$$

TRANSPIRATION THROUGH WALL:

EXPONENTIAL TEMPERATURE DISTRIBUTION

Figure 4: Transient conduction in an insulated box and comparison of conduction and transpiration temperature profiles.

where the total mass of the enclosure has been replaced by the density times the total volume of the enclosure. This means that if

$$V \gg \frac{k}{t\,\rho\,c_p} \tag{10}$$

transpiration will be significantly faster than conduction only. Using typical values for the conductivity of styrofoam, the air flow necessary to satisfy equation 10 comes out to be 3.6×10^{-4} m/s for 0.1 m(4 in.) thick walls, corresponding to a Reynold's number inside the box of about 10(based on a 1 m long box). Of course, the velocity must be kept low and mixing avoided for the air change time of equation 9 to be valid. If this is done, any disturbance inside the box, no matter how large, will be completely swept away in a time of approximately t_{change}.

Using the velocity criterion in equation 10 in the equation for the characteristic length given in equation 5, the expected characteristic lengths for a transpiration enclosure may be calculated. Also, the minimum heater power required may also be found. The characteristic length,

$$\lambda = \frac{\rho\,k\,A_s}{A_f\,V\,c_p} \tag{11}$$

turns out to be 0.04 m(1.6 in.) maximum, decreasing with increasing velocity. The minimum heater power required,

$$P_{min} = \dot{m}\,c_p\,\Delta T_{in} \tag{12}$$

where $\dot{m}$ is the inlet mass flow rate per unit inside surface area (density of air times velocity), comes out to be 4.25 W/m^2 inside surface area. Equations 11 and 12 are based on styrofoam walls with a set point 10 C° above room temperature.

The power requirement in equation 12 can be compared to the power required by an insulated enclosure,

$$P_{min} = k \frac{\Delta T_{in}}{t} \tag{13}$$

where k is the wall thermal conductivity and t is the wall thickness. The power required turns out to be 7.2 W/m^2 for 0.05 m(2.0 in.) thick styrofoam walls.

Based upon these anticipated benefits of the use of transpiration for temperature control (near perfect insulation, no stagnant regions, quick and complete response to all types of disturbances, and low power required), a test enclosure was designed and built to investigate the actual benefits of transpiration.

Experimental

To confirm that the simple model discussed in the previous section accurately predicts the behavior of a transpiration-walled enclosure, one could instrument a cross-section of a wall with a transpirant flowing through it to determine the actual temperature distribution. Researchers have already done this — tests were done where the solid and transpirant temperatures along a cross-section of a transpiration cooled piece of sintered porous metal were accurately measured [4]. What was not done, however, was to measure the steady-state response of such a system when a step temperature change was applied. Also, since the specimen was a metal with high conductivity, the model developed may not be valid for insulating materials which have a much lower thermal conductivity.

The purpose of this investigation is to determine the steady-state temperature error in an enclosure when a porous wall is subject to a step temperature on one side. To test this, one would like to apply a step temperature change to one side of a porous material which was being supplied with a steady flow of constant temperature air and measure the steady-state temperature change on the other side. Testing different thicknesses and flow rates would allow the exponential temperature distribution and estimated power required to be confirmed.

A test apparatus to do this requires a source of controlled temperature air at a known, preferably selectable, flow rate, a means to hold different samples of material in the controlled flow for transpiration, and instrumentation to measure the steady-state temperatures. A block diagram of such a system is shown in figure 5.

Figure 6 shows a schematic of the actual test apparatus used. It consists of four major sections: an intake section, a fan/heater section, a test enclosure, and a cooling coil. The intake section is used to meter the flow entering the enclosure by measuring the pressure drop caused by passing the inlet flow through a pipe containing a diameter change. The details of this construction, and those of all the other sections, are contained in the Appendix.

The fan/heater section contains a fan to provide the necessary inside pressure for transpiration. It also contains the electronically controlled heater which attempts to maintain the incoming air at a constant temperature.

The test enclosure was built in two pieces (details in the Appendix) to facilitate emplacement of the thermistors needed for measuring the inside wall and transpirant temperatures. Its walls were constructed of 0.1 m(4 in.) thick Styrofoam (foamed polystyrene), so that most of the heat loss from the enclosure would be through the specimen under test. The heat loss through these walls was calculated to be approximately 0.6 W. The test enclosure houses the material under test in a cut out hole.

The cooling coil is the means used to apply a step temperature change to the outside of the material under test. It consists of a copper tube soldered to a copper screen. An ice-

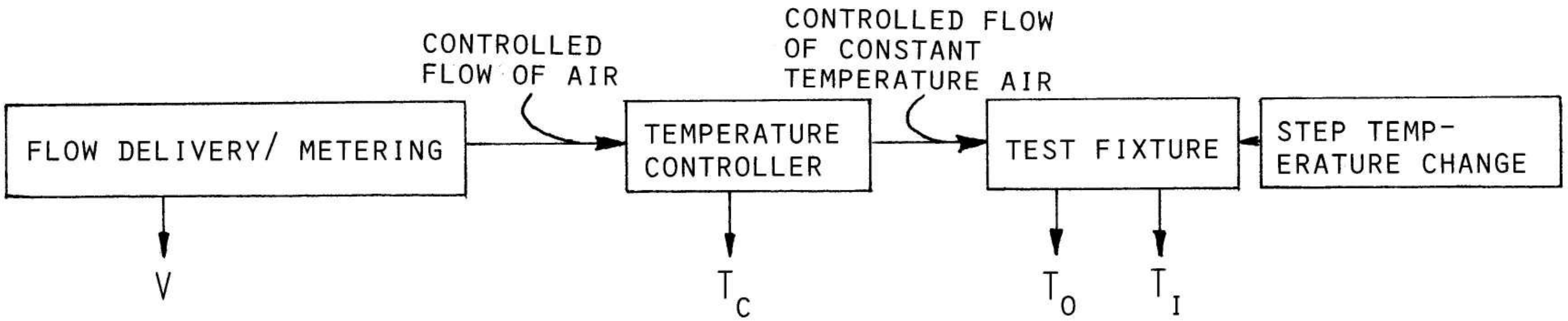


Figure 5: block diagram of a system to measure the steady-state temperature response of a sample of porous material

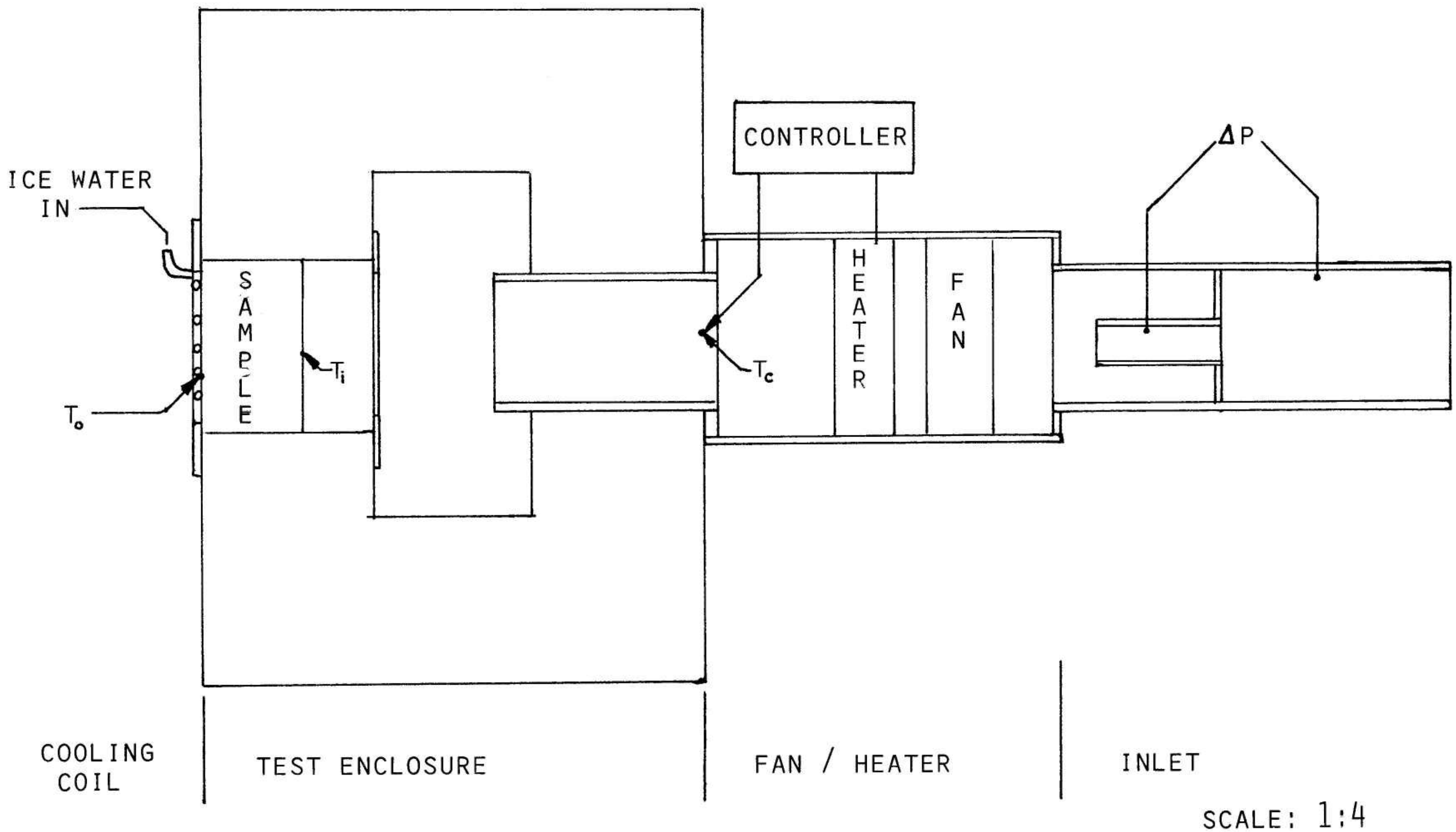


Figure 6: Schematic of the test apparatus

water mixture was poured through the tube (through a funnel) to provide a long-term step temperature change to the outside wall.

Tests were done by measuring the steady-state temperature change on the inside surface of different thicknesses of foam samples. The signal measured was the difference between the output of a thermistor mounted in the inlet flow near the control thermistor and the output of a thermistor mounted on the inside surface of the foam sample under test. This was to partly compensate for drift in the heater control circuit.

From the expected exponential temperature distribution predicted by the simple model discussed above, the change in inside temperature should be

$$T_i - T_c = (T_o - T_c)\, e^{-\frac{l}{\lambda}} \tag{14}$$

which can be written as

$$\frac{\Delta T_i}{\Delta T_o} = e^{-\frac{l}{\lambda}} \tag{15}$$

where T_i is the temperature on the inside of the wall, T_c is the inlet (controlled) temperature, l is the sample thickness, and λ is the characteristic length of the sample under test. ΔT_i is defined as the difference between the inside wall temperature and the controlled temperature, and ΔT_o is the difference between the outside wall temperature and the controlled temperature. Taking the logarithm of equation 15,

$$\ln \frac{\Delta T_i}{\Delta T_o} = -\frac{l}{\lambda} \tag{16}$$

so a plot of $-\ln \frac{\Delta T_i}{\Delta T_o}$ versus l should be a straight line with slope $\frac{1}{\lambda}$.

Since the flow rate varied from sample to sample (the inlet section served more as a constant pressure source than a constant flow source), the data should be plotted with respect to the flow rate through them, as characteristic length is inversely proportional to flow rate. Also, since there were found to be some leaks in the enclosure, the pressure at the inside surface of each sample was measured. Relating the flow through the sample to the pressure drop across it and its thickness,

$$Q = K \frac{\Delta P}{l} \tag{17}$$

where K is a permeability of the sample and Q is the flow through it. Combining this with equation 16 gives

$$-\ln \frac{\Delta T_i}{\Delta T_o} = C\ \Delta P$$

where C is a constant containing material properties of the solid and transpirant only, equal to $\frac{\rho c_p K}{k A_s}$, so a plot of $-\ln \frac{\Delta T_i}{\Delta T_o}$ versus ΔP should be a straight line with slope C. Since the permeability of the sample was not measured (its need was not anticipated at the outset of experiments), the test data are not quantitatively useful for determining λ yet, but they may still fulfill the original purpose of these tests: to investigate the steady-state temperature response of a transpiration controlled enclosure, and to verify the exponential temperature distribution for low conductivity materials.

Figure 7 is a plot of equation 18 for the samples tested. It seems to be linear, suggesting that the temperature response to step changes is in fact exponential in character.

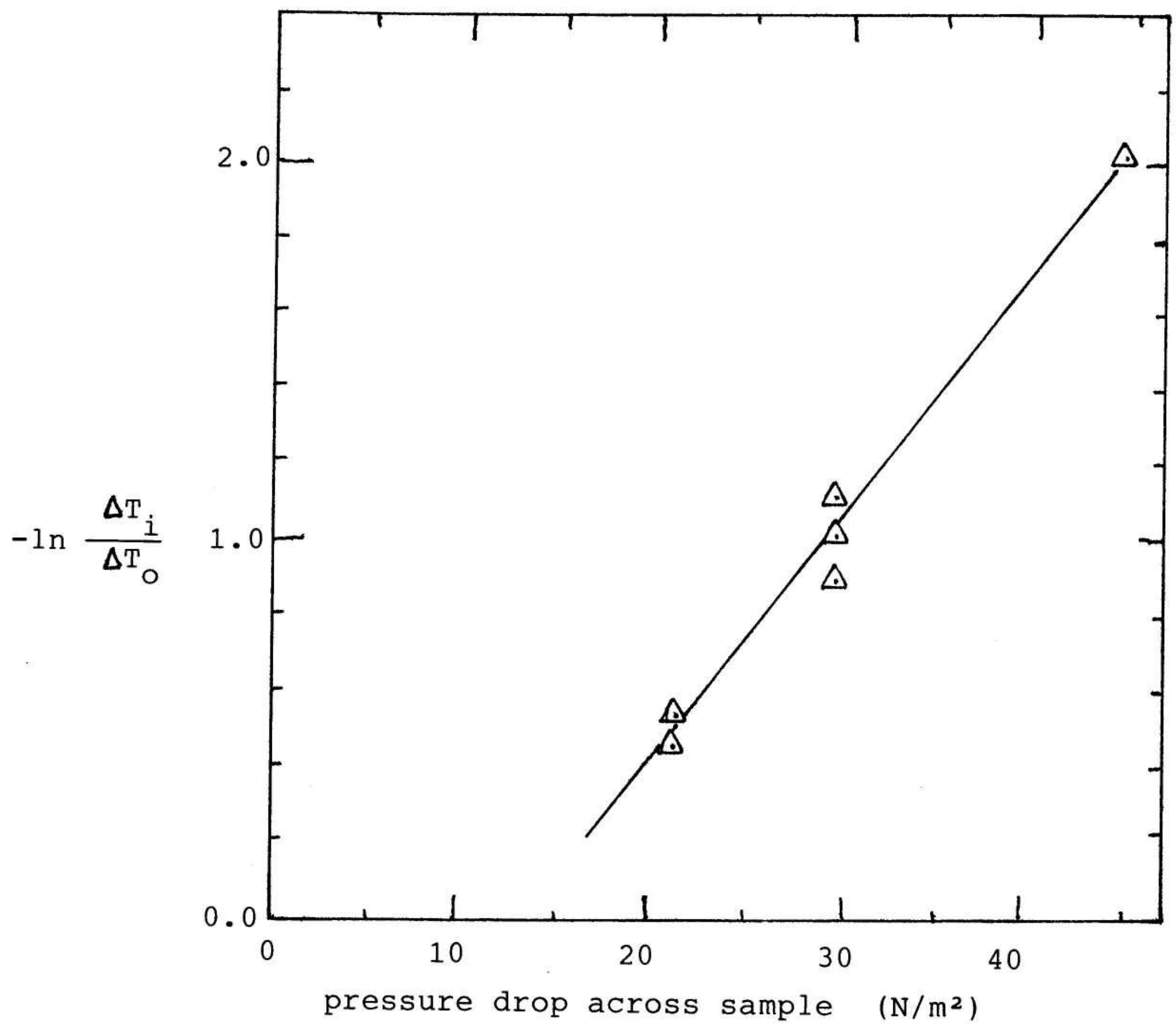


Figure 7: Plot of $-\ln \frac{\Delta T_i}{\Delta T_o}$ versus pressure drop across sample for the samples of polyurethane foam tested

Discussion and Conclusions

From the test data presented, it appears that the expected exponential temperature distribution does exist in the insulated wall tested. Also, the attenuation of step changes in outside temperature was also confirmed.

This implies that transpiration may indeed be useful for the high insulation quality and quick response to all types of thermal disturbances needed for precision temperature controlled enclosures.

Since the data from the tests were not useful quantitatively, the following recommendations are made for the design of a test apparatus suitable for producing quantitative design data;

1. a better method for measuring the flow through the sample is needed. This was the major limitation of the present design, since the low flow rates involved necessitate the use of sensitive pressure-measuring devices if a differential pressure measurement to determine flow rate is used. Also, small leaks in the enclosure can have a large effect on accuracy if only the inlet flow is measured. The suggested way to measure the flow is by using a low range differential pressure transducer to measure the pressure drop through the sample, which can be related linearly to the flow through it by its permeability.
2. a constant, adjustable flow supply would be useful to

allow data to be taken over a wider range. The fan used in the present design was not powerful enough to provide sufficient pressure to allow testing of very thick samples of foam. A suggested way is to use a a high pressure of air behind a small orifice (which could be changed to vary flow rates), to provide a near-constant mass flow rate independent of downstream pressure.

3. a circulation system, along with the heater, should be located inside the test enclosure to allow tests to be done with no flow through the wall. This would allow a quantitative comparison of the power required for transpiration to the power required for insulation relying on conduction only.

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CONTINUITY

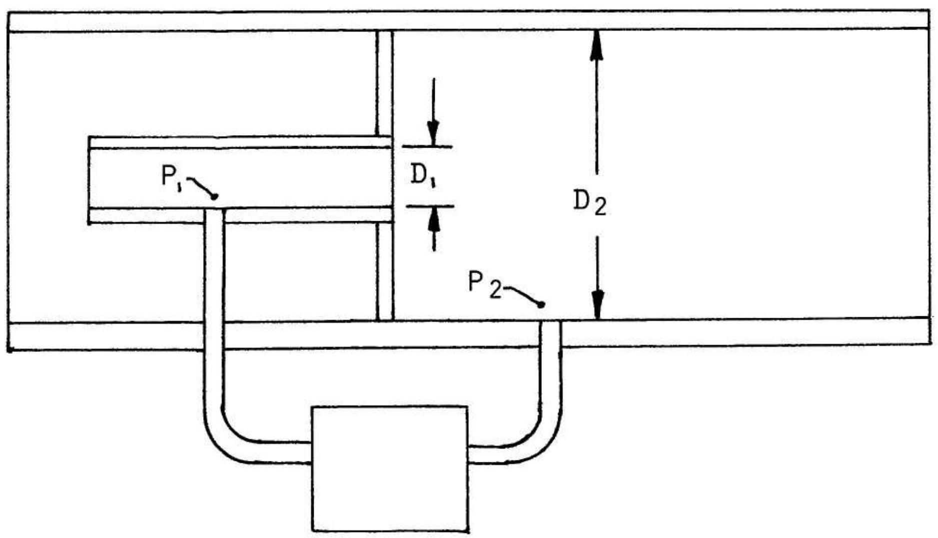
$$A_1 V_1 = A_2 V_2$$

BERNOULLI'S EQUATION

$$P_1 + \frac{1}{2}\rho V_1^2 = P_2 + \frac{1}{2}\rho V_2^2$$

$$D_1 = 0.01294 \text{ M}$$

$$D_2 = 0.06972 \text{ M}$$



DIFFERENTIAL PRESSURE TRANSDUCER

MATERIALS

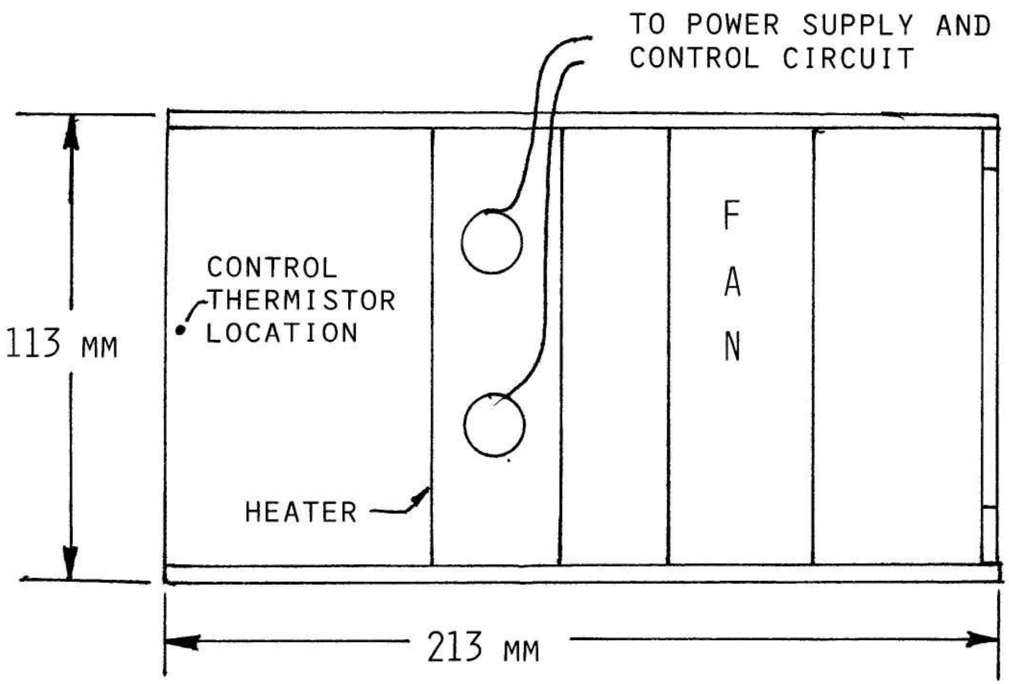
ACRYLIC PIPE
- 76 MM I.D. X 225 MM
- 13 MM I.D. X 76 MM

ACRYLIC SHEET
- 6 MM THICK

POLYETHYLENE TUBING
- 6 MM O.D.

DIFFERENTIAL PRESSURE TRANSDUCER
- 3500 N/M^2 F.S.

Figure A-1: Detail of inlet section (one-half size)



MATERIALS

FAN

- BOXER TYPE, FOR EQUIPMENT COOLING

HEATER

- 2Ω COILED CHROMEL WIRE ON CERAMIC STANDOFFS
- SUPPORTS MADE OF 6 MM THICK ACRYLIC SHEET

ENCLOSURE WALLS

- 6 MM THICK ACRYLIC SHEET
- 0.8 MM THICK SHEET STEEL STEEL AT CORNERS, FASTENED WITH 6-32 MACHINE SCREWS
- TOP SEALED WITH CLOSED CELL FOAM WEATHER-STRIPPING

Figure A-2: Detail of fan/heater section (one half size)

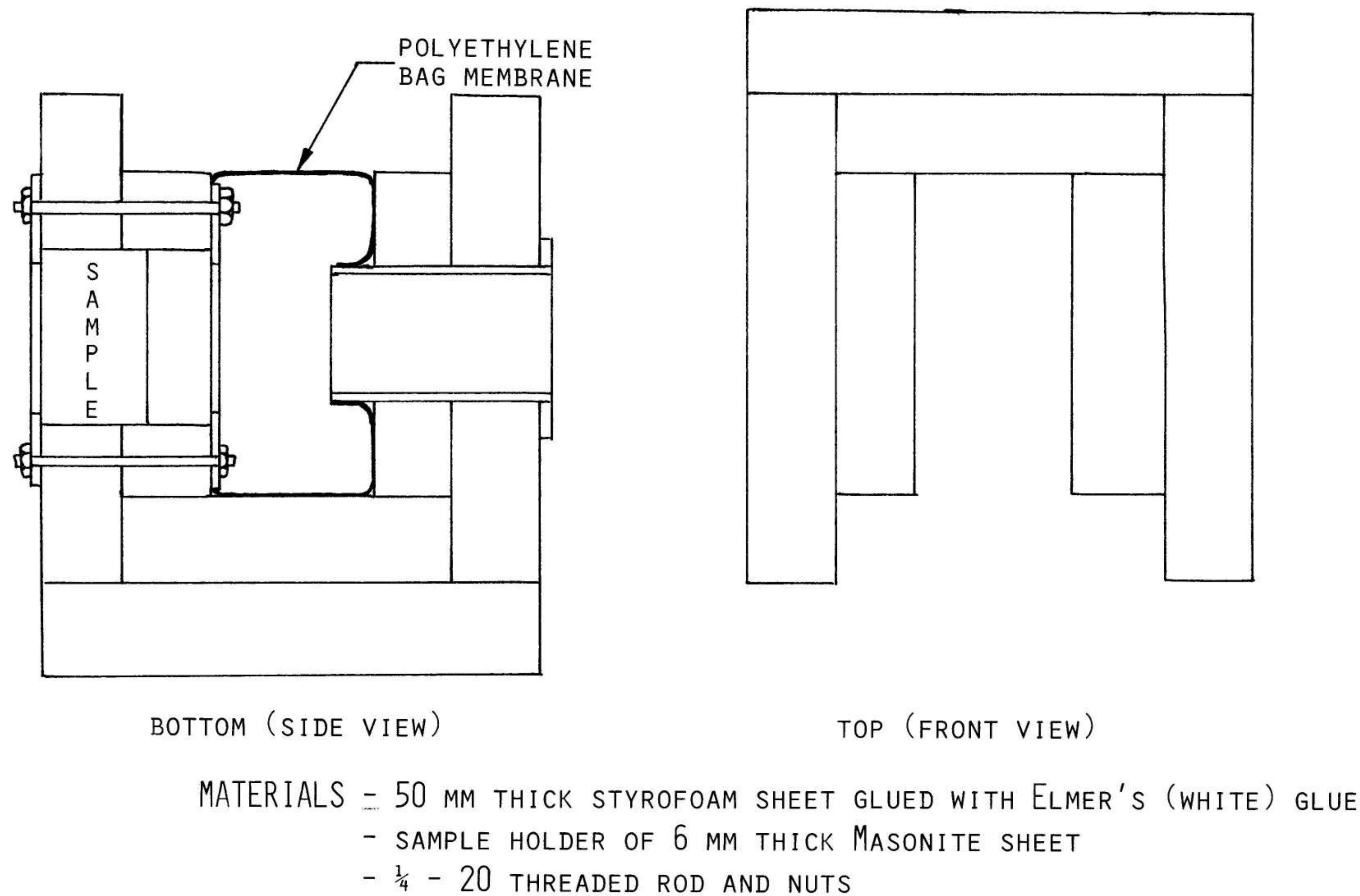
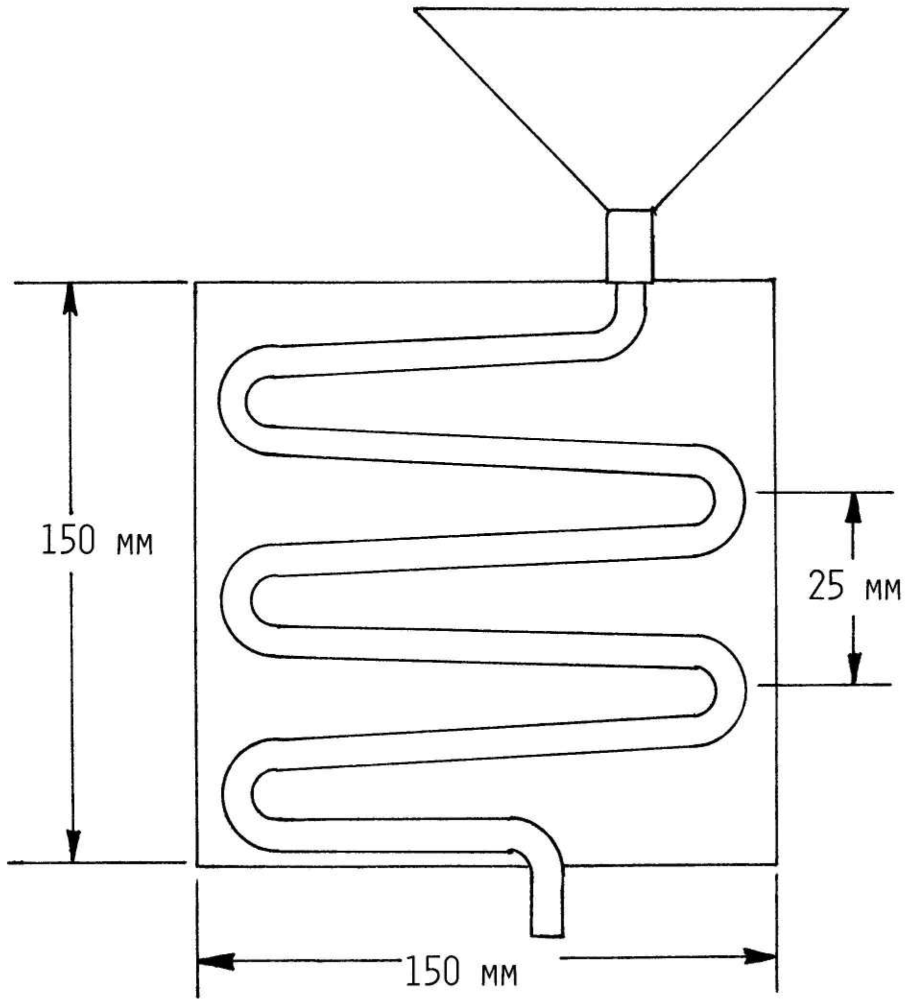


Figure A-3: Detail of test enclosure (one-quarter size)

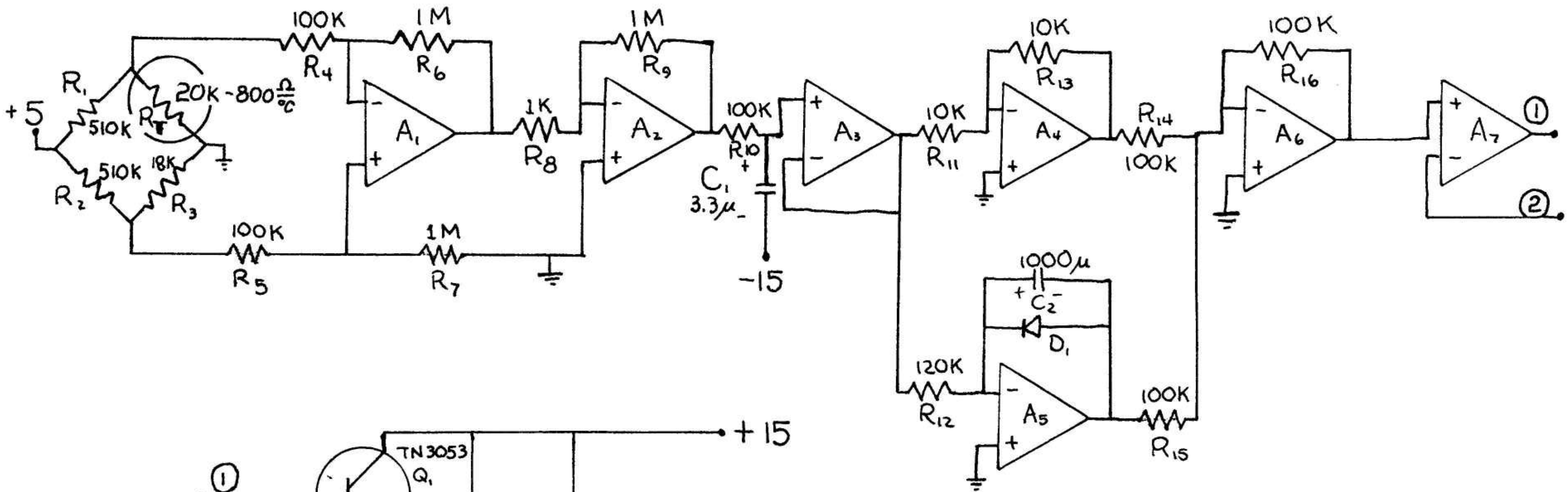


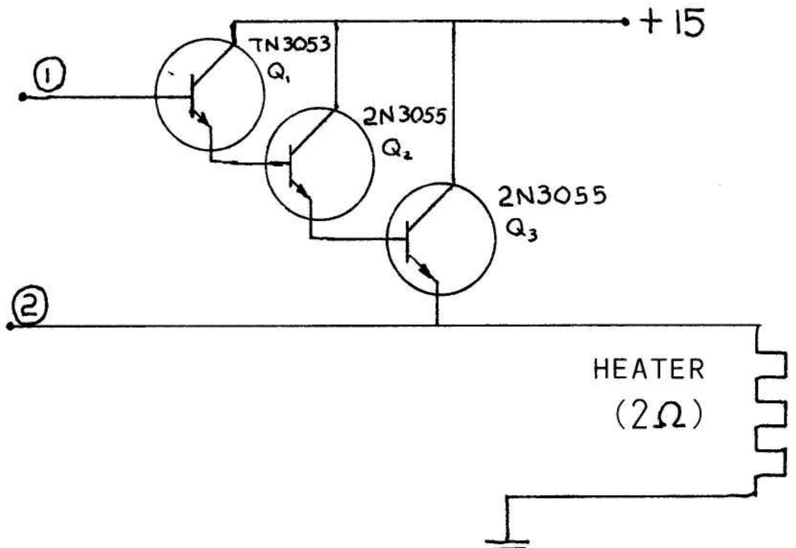
MATERIALS

- 40 MESH COPPER GAUZE
- 5 MM O.D. COPPER TUBING
- SOLDER
- 200 MM TOP DIA. FUNNEL

Note: Funnel not to scale.

Figure A-4: Detail of cooling coil (one-half size)





NOTES

- R_T IS A THERMISTOR IN AIR
- ALL OP-AMPS TYPE LM741CN
- ± 15V SUPPLIES NOT SHOWN
- Q_{1-3} MOUNTED ON A HEAT SINK
- +15V FROM 75W LAMBDA POWER SUPPLY

Figure A-5: Schematic of controller circuit

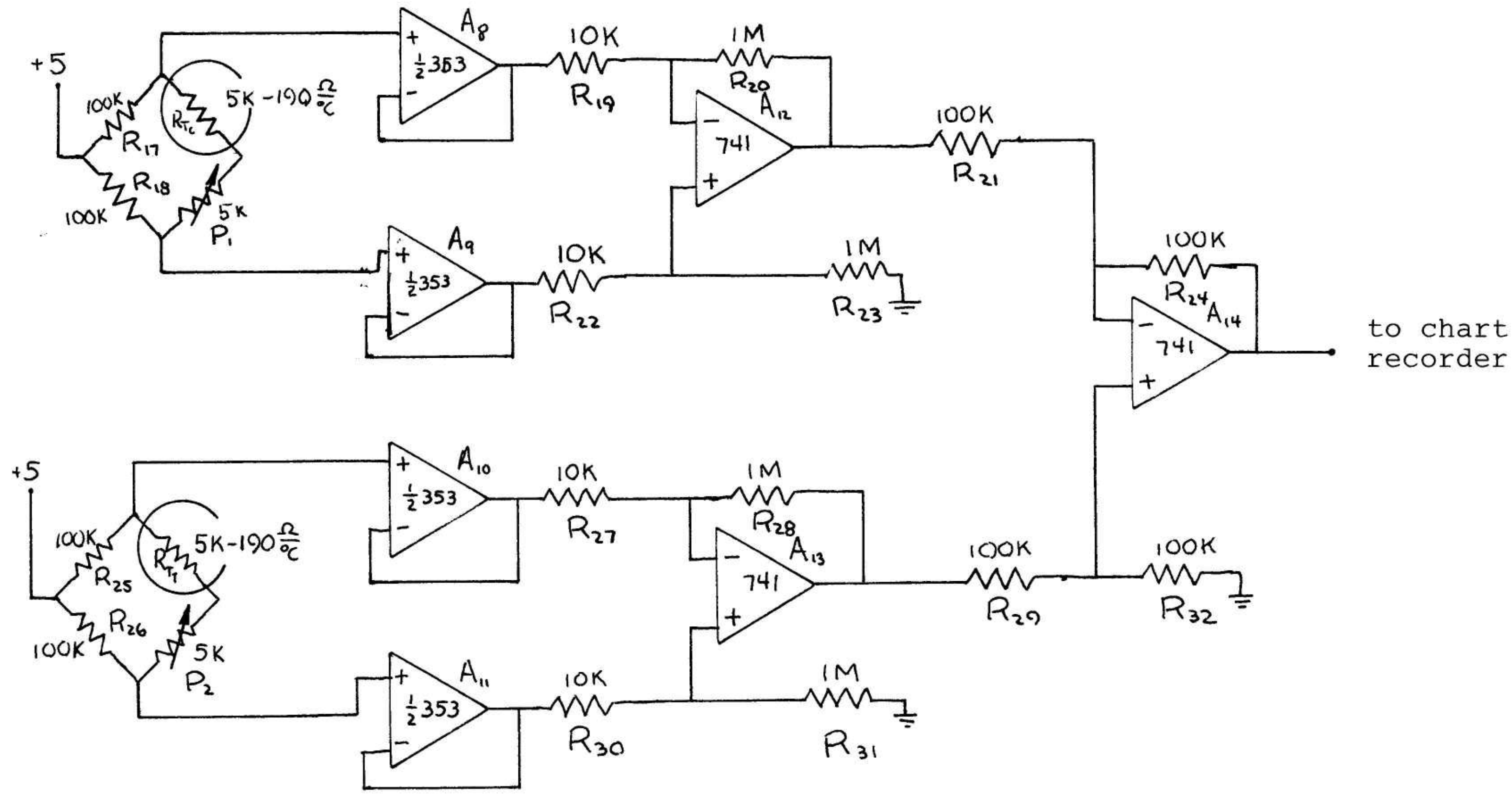


Figure A-6: Schematic of differential temperature circuit