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A study of the transient state of transpiration cooling

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A STUDY OF THE TRANSIENT STATE OF TRANSPIRATION COOLING

BY

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A

THESIS

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I. ABSTRACT

This thesis is a successful attempt to show a new approach to the transpiration cooling problem. Transpiration cooling has long been known as an effective way of removing heat from porous material.

It is shown here that transpiration cooling from a porous material, assuming a temperature differnce between the coolant and the porous material, is less effective than where the coolant within the pores is assumed to be at the same temperature as the material adjacent to the pore. Previous investigators made the latter assumption.

The specific problem considered here is that of a uniform, porous plate with constant physical properties, receiving on one side a constant heat flux and on the other a coolant of constant physical properties flowing counter to the heat flux at a constant rate. It is further assumed that there exists a finite temperature difference between the coolant in the pores and the adjacent solid of the plate. Trnasient conditions are considered and they are found to be of very short duration. However, they are longer than in the case where the temperature of the solid and the coolant are assumed to be the same at a pore.

The mass flow rate, the heat flux, and the specific heat of the coolant control the maximum temperature at the surface during steady state conditions. Thus, with these constants at hand there is an effective check on the final results. However, there is no equivalent method available for checking the rate of change of the temperature curve.

The constants used in this problem are the same as those used by Herbert S. B rahinsky in a thesis in which he considered a similar problem (12). This was done in order to have available a method of comparison in final results.

Although the thesis, when compared with other work on the same problem, shows that there is a somewhat slower cooling process than was previously believed, the difference in time is of a matter of seconds and then only in the hotter portion of the porous plate. Therefore, this would undoubtedly be an effective cooling process.

It should be emphasized at this point that although this is a theoretical thesis with no particular application mentioned, the author's interest in the problem stems from the fact that he is preparing for the field of nuclear engineering and it is his belief that a satisfactory knowledge of effective cooling methods for the reactor field will be of prime importance in bringing about the developement of reactors which will be able to compete with conventional power supplies.

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II. ACKNOWLEDGEMENT

Upon finishing his first full scale research undertaking, it is the considered opinion of the author that the value of such a short thesis lies not so much in the amount of knowledge one gains about a certain subject but rather in the lessons learned in the areas of diligence and logical thinking.

Because of the nature of the problem here involved, and because of a lack of previous experience in the mathematics of cooling problems, the author finds himself more than ordinarily in debt to many people for their advice and patience during the work. Therefore, the author takes this occasion to express his gratitude to Dr. A.J. Miles, for his encouragement and advice, to H. Bhattacharya and Charles Dana, for their patience, and to John F. Curtin, for his very real help in the programming of the above work.

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III. INTRODUCTION

Many modern engineering structures, components, and accessories require shielding from extreme heat fluxes. Among these are reactors, electrical components, and fire walls. Heat fluxes of over a million BTU/hr ft^2 are not uncommon. The porous plate with a coolant flowing counter to the heat flux is one of the most effective deterents to heat flow known to the author. This method of shielding has a short transient temperature build up when subjected to a sudden heat flux. Likewise, when subjected to a cyclic heat flux, the cooling period is of short duration. After the steady state has been reached its effectiveness as a shield remains so long as the coolant continues to flow. It is in this regard that the porous shields have advantages not found in such types of cooling as ablation shielding. The latter is used on the mercury capsule for the man in space project. The protective coating placed on the capsule is burned to destruction when the capsule re-enters the atmosphere and encounters the heat barrier. This is a one shot affair and of a very short duration, where as the porous shield with coolant is of a long duration and may be used repeatedly.

In previous investigations of the porous plate with coolant (12), it was assumed that the temperature of the coolant within a pore of the porous plate was the same as that of the material surrounding the pore. This may be the case in the limit as the size of the pore is reduced to zero diameter and/or the flow of the coolant is reduced to zero. If these temperatures are different, the effectiveness of the porous plate with coolant as a shield may be other than that which was predicted by the previous assumption of earlier investigators.

Therefore, the problem selected for investigation in this thesis is that of a porous plate receiving a constant heat flux on one surface, while coolant fluid is forced in through the plate counter to the heat flux. The temperature of the plate is initially uniform and at the temperature of the coolant. The plate is exposed to the heat flux and begins heating up. This is the transient heat problem which will be studied. Heat conduction will be considered through both the solid and the fluid. There is also a change in internal energy due to the rise in temperature which is also considered in both the solid and the fluid. Finally, heat convection is considered as it will be taking place between the solid and the coolant.

The assumptions which must be made in order that a practical solution to the problem can be found must now be considered. These are as few and reasonable as possible and are those commonly made in engineering applications of heat transfer problems.

The assumptions involved in this problem are:

1) The thermal conductivities and specific heats of the solid and coolant are constant.
2) There is a known convection heat transfer coefficient existing between the solid and the coolant.
3) In the porous plate the difference in coolant temperature over a given time interval and the difference in plate temperature over the same time interval at the same node are approximately equal.

The problem is solved in the following manner. First, the transient equations are written for the temperature inside the porous plate. Then they are converted into a more useful form and solved on a computer.

A general program has been written for this problem so that the data fed into the computer can be readily changed in order that other conditions can be studied. It is assumed that any approximations that might be made by the computer itself are well within the limits necessary for engineering methods.

NOTATION

P	Porosity	$\frac{\text{Pore Volume}}{\text{Total Volume}}$
t	temperature of metal (^{o}F)	
t'	temperature of metal after some time increment (^{o}F)	
T	temperature of coolant (^{o}F)	
T'	temperature of coolant after some time increment (^{o}F)	
G	mass flow rate (lbs/hr ft^2)	
V	volume of node considered (ft^3)	
A	area of node considered (ft^2)	
δ	Distance between nodes (ft)	
K_f	thermal conductivity of coolant (BTU/ft^2 ^{o}F/ft)	
K	thermal conductivity of solid (BTU/ft^2 ^{o}F/ft)	
Q	heat flux (BTU/hr ft^2)	
C_p	specific heat of coolant (BTU/lb ^{o}F)	
C	specific heat of solid (BTU/lb ^{o}F)	
$\Delta\theta$	time increment (hr)	
ρ	specific weight of solid (lbs/ft^3)	
ρ_f	specific weight of coolant (lbs/ft^3)	
h	heat convection film coefficient (BTU/hr ft^2 ^{o}F)	
S	average surface of the particles/Vol.	

IV. REVIEW OF THE LITERATURE

The assumptions made in the solution of this problem come from several sources. Weinbaum and Wheller (1) are the authorities from which the first and third assumptions came. The all important assumption concerning the existance of a film coefficient was first brought to the authors attention by A.J.Miles (13).

L. Green (10) was P. J. Schneider (2) are probably the best sources readily available for discussion of the theory pertaining to transpiration cooling. Green is especially well versed in the writing of differential equations which cover the steady state situation. Also, Wylie, (5) has some excellent material on the boundary conditions for the solutions of the above mentioned steady state differential equations.

Gebhart (6) and Kern (7) are the references used in this thesis on the heat transfer problems as well as being a handy reference for physical constants. Kern has some good sample problems in heat transfer as well. All the rest of the constants used in this thesis are taken either from the Mechanical Engineers Handbook (8) or from the Handbook of Chemistry and Physics (9).

In the programming of the problem, the sole references used was Hildebrand (3) and Lee (4). By the very nature of the program, which is large, there is probably no single well known computer method which exactly duplicates it. The program is very straightforward in method and the techniques are those taught by Prof. Lee in his Numerical Analysis course at the School of Mines.

Kern (7) is the reference from which the bulk of the discussion on the film coefficient comes. The actual value used in the data for this film coefficient also comes from this text.

Miles (13) is the authority quoted on the use of the term S, which will be discussed later, and which has to do with the nature of the porous material used.

It must be stated in conclusion that great use is made of certain work done by Herbert S. Brahinsky in his thesis on this same problem (12). He gives very explicit explanations of many of the terms and equations which are used for this problem.

V. DISCUSSION

The problem considered here is that of a thin uniform porous plate, the thickness of which is small compared with the other two dimensions. The physical properties are independent of the temperature. The plate is exposed on one side to a uniform heat flux and simultaneously receives a uniform coolant on the other surface which flows through the plate counter to the heat flux in such a way as to remove all or part of the heat. The extent of the plate is such that the problem is a one dimensional situation in regard to the coolant flow and to the heat flow. This can be seen in figure 1.

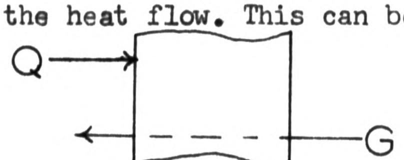


figure 1.

As is usual in these problems, the plate is divided into three full nodes and two half nodes, the half nodes being at the edge of the plate as shown in figure 2. so that the surface temperatures can be considered.

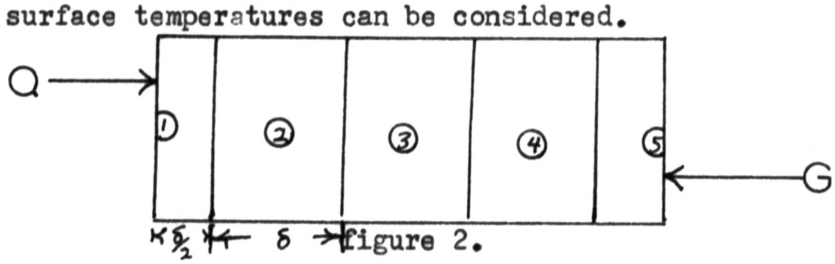


figure 2.

The macroscopic picture of the internal structure of the porous plate can be seen in figure 3. The plate, being porous in nature, is made up of very small irregularly shaped solids, with open spaces between them. The small solids of the plate are very much larger than atomic dimensions. Each of them possesses an individual film coefficient although numerically they are considered to be at the same value.

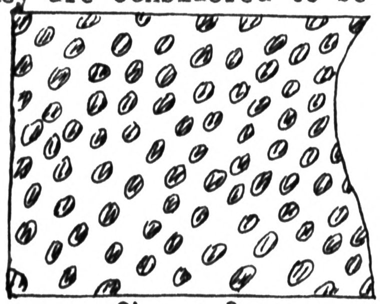


figure 3.

From the above discussion, it is obvious that the heat flow, while always possessing a temperature gradient in a single direction, will essentially be divided into three parts. There will be flow of heat through the porous material, the flow of heat through the fluid in a direction counter to the flow of the coolant, and the flow of the heat from the material to the coolant because of the temperature difference between the two.

The writing of the heat balance equation for node 1., the first half node nearest the point of entry of the heat flux, can now be undertaken.

The heat balance for the first node is shown below. All of these are based on the principle that the heat into the system must equal the heat going out of the system in the steady state.

$$A\,Q\,\Delta\theta = \frac{K\,A\,(1-P)\,\Delta\theta\,(t_1-t_2)}{\delta} + \frac{K_f\,A\,P\,(T_1-T_2)\,\Delta\theta}{\delta} + G\,C_p\,A\,\Delta\theta\,(T_1-T_2)$$

$$+\,\rho\,C\,A\,\frac{V}{2}\,(1-P)(t_1'-t_1) + \rho_f\,C_p\,A\,\frac{V}{2}\,P\,(T_1'-T_1)$$

The terms in the equation are defined as follows: the heat flux entering the small surface area during the time increment

$$A\,Q\,\Delta\theta$$

The amount of heat given up to the coolant by the solid

$$G\,C_p\,A\,(T_1-T_2)\,\Delta\theta$$

Conduction away from point 1 through the solid

$$\frac{K\,A\,(1-P)\,\Delta\theta\,(t_1-t_2)}{\delta}$$

Conduction away from point 1 through the coolant

$$\frac{K_f\,A\,P\,\Delta\theta\,(T_1-T_2)}{\delta}$$

The increase of internal energy of the solid

$$\rho\,C\,A\,\frac{V}{2}\,(1-P)(t_1'-t_1)$$

The increase in internal energy of the coolant

$$\rho_f\,C_p\,A\,\frac{V}{2}\,P\,(T_1'-T_1)$$

The amount of heat given up to the coolant by the solid by convection

$$h A \frac{\delta}{2} s (t_1 - T_1) \Delta\theta$$

Because there are three unknown temperatures, two more relationships must be found before the problem can be solved. Therefore, the assumption is used that the change in temperature of the solid and the coolant are the same over the same time interval at the same point.

$$(t_1' - t_1) = (T_1' - T_1)$$

Then, it is shown that the heat given up by convection is equal to the heat given up by the other term concerning the passing of heat from the solid to the coolant.

$$G C_p A (T_1 - T_2) \Delta\theta = h A \frac{\delta}{2} s (t_1 - T_1) \Delta\theta$$

The above term, which is dependent upon the rate of flow of the coolant, shows that the temperature difference is that difference between the temperature of the liquid at node one and the temperature of the coolant at node two in the case of the mass flow term. In the convection term, the temperature difference is between the solid and the coolant temperatures at the same node. Half volumes are used in the above equations because the first node is only a half node.

Next, the second node is considered. Here the heat balance equation for this node is shown below. The heat flux term is not present but two additional conduction terms are.

$$\frac{K A \Delta\theta (1-P)(t_1 - t_2)}{\delta} + \frac{K_f A \Delta\theta P (T_1 - T_2)}{\delta}$$

$$= G C_p A \Delta\theta (T_2 - T_3) + \frac{K A \Delta\theta (1-P)(t_2 - t_3)}{\delta}$$

$$+ \frac{K_f A \Delta\theta P (T_2 - T_3)}{\delta} + \rho V A C (1-P)(t_2' - t_2) + \rho_f V A C_p P (T_2' - T_2)$$

The heat entering point 2 by conduction through the solid is

$$\frac{K A \Delta\theta (1-P)(t_1 - t_2)}{\delta}$$

The heat entering point 2 by conduction through the liquid is

$$\frac{K_f A \Delta\theta P (T_1 - T_2)}{\delta}$$

The rest of the terms are similar to those used in the previous heat balance equation except that a full volume is used as a full node is now under consideration.

The heat leaving point 2 by conduction through the solid is

$$\frac{K A \Delta\theta (1-P)(t_2 - t_3)}{\delta}$$

The heat leaving point 2 by conduction through the coolant is

$$\frac{K_f A \Delta\theta P (T_2 - T_3)}{\delta}$$

The heat given up to the coolant by the solid is

$$G C_p A \Delta\theta (T_2 - T_3)$$

The increase in internal energy of the solid is

$$\rho V(1-P)C(t_2'-t_2)A$$

The increase in internal energy of the coolant is

$$\rho_f V C_p (T_2'-T_2) PA$$

Finally, the convection term is given by

$$hA\delta S(t_2-T_2)\Delta\theta$$

As before, the approximation is made that the change in temperature of the solid and the coolant are the same over the same time interval at the same point.

$$(t_2'-t_2) = (T_2'-T_2)$$

Then the term describing the heat given up by convection is equal to the term describing the heat given up to the coolant by the solid.

$$hA\delta S(t_2-T_2)\Delta\theta = G C_p A \Delta\theta (T_2-T_3)$$

This is the set of equations governing the second node. The third and fourth nodes are handled in the same way. Their equations follow.

These are the equations for nodes three and four.

$$\frac{K A \Delta\theta (1-P)(t_2-t_3)}{\delta} + \frac{K_f A \Delta\theta P (T_2-T_3)}{\delta}$$

$$= G C_p A \Delta\theta (T_3-T_4) + \frac{K A \Delta\theta (1-P)(t_3-t_4)}{\delta}$$

$$+ \frac{K_f A \Delta\theta P (T_3-T_4)}{\delta} + \rho V c (1-P)(t_3'-t_3) A$$

$$+ \rho_f V C_p P (T_3'-T_3) A$$

$$(t_3'-t_3) = (T_3'-T_3)$$

$$G C_p A \Delta\theta (T_3 - T_4) = h A \delta S (t_3 - T_3) \Delta\theta$$

and

$$\frac{K A \Delta\theta (1-P)(t_3-t_4)}{\delta} + \frac{K_f A \Delta\theta P (T_3-T_4)}{\delta}$$

$$= G C_p A \Delta\theta (T_4-T_5) + \frac{K A \Delta\theta (1-P)(t_4-t_5)}{\delta}$$

$$+ \frac{K_f A \Delta\theta P (T_4-T_5)}{\delta} + \rho_f V C_p P (T_4'-T_4) A$$

$$+ \rho V c (1-P)(t_4'-t_4) A$$

$$(t_4'-t_4) = (T_4'-T_4)$$

$$h A \delta S (t_4-T_4) \Delta\theta = G C_p A \Delta\theta (T_4-T_5)$$

Finally, the fifth node is considered. Again, this will require that half volumes be used in order to again obtain a surface temperature as was done in the first node. The heat balance for the fifth node does not contain a term for conduction away through the solid as it is a surface point. Using the methods described before, the equation for the fifth node is written readily.

$$\frac{K_f A \Delta\theta P (T_4 - T_5)}{\delta} + \frac{K A \Delta\theta (1-P)(t_4 - t_5)}{\delta}$$

$$= \frac{K_f A \Delta\theta P (T_5 - T_6)}{\delta} + G C_p A \Delta\theta (T_5 - T_6)$$

$$(t_5' - t_5) = (T_5' - T_5)$$

$$h A \tfrac{\delta}{2} S (t_5 - T_5) = G C_p A \Delta\theta (T_5 - T_6)$$

The following constants are defined so that the equations can be written in more manageable form.

The constants are as follows:

$$C_1 = \rho A \frac{V}{2} (1-P) c$$

$$C_2 = \rho_f A \frac{V}{2} P c_p$$

$$C_3 = \frac{K A (1-P) \Delta\theta}{\delta}$$

$$C_4 = \frac{K_f A P \Delta\theta}{\delta}$$

$$C_5 = G c_p A \Delta\theta$$

$$C_6 = h A \delta S \Delta\theta$$

Next, the values of the temperatures of the coolant are solved for at the various nodal points. These are gotten from the third relationship given for each node.

$$T_1 = \frac{\frac{C_6 t_1}{2} + C_5 T_2}{C_5 + \frac{C_6}{2}}$$

$$T_2 = \frac{C_6 t_2 + C_5 T_3}{C_5 + C_6}$$

$$T_3 = \frac{C_6 t_3 + C_5 T_4}{C_5 + C_6}$$

$$T_4 = \frac{C_6 t_4 + T_5 C_5}{C_5 + C_6}$$

$$T_5 = \frac{\frac{C_6 t_5}{2} + C_5 T_6}{C_5 + C_6/2}$$

The values of the constants can be computed and as the initial temperatures of the nodal points and also the initial temperature of the coolant, T, will be known and so the initial temperatures of the solid at the various nodal points can be computed. Actually, all of this is done as part of the computer program.

By rearranging the first relationship for each nodal point, there can now be written equations for t' in terms of known quantities. These are then printed out by the computer and then the new temperatures are loaded back into the program until the problem goes into equilibrium. The principle equations are shown below.

$$t_1' = \frac{(c_1+c_2)t_1 + AQ\Delta\theta - c_3(t_1-t_2)}{(c_1+c_2)}$$

$$\frac{-c_4(T_1-T_2) - c_5(T_1-T_2)}{(c_1+c_2)}$$

$$t_2' = \frac{c_3(t_1-2t_2+t_3)+c_4(T_1-2T_2+T_3)}{2(c_1+c_2)}$$

$$-\frac{c_5(T_2-T_3)}{2(c_1+c_2)} + t_2$$

$$t_3' = \frac{C_3(t_2 - 2t_3 + t_4) + C_4(T_2 - 2T_3 + T_4) - C_5(T_3 - T_4)}{2(C_1 + C_2)} + t_3$$

$$t_4' = \frac{C_3(t_3 - 2t_4 + t_5) + C_4(T_3 - 2T_4 + T_5) - C_5(T_4 - T_5)}{2(C_1 + C_2)} + t_4$$

$$t_5' = \frac{C_4(T_4 - 2T_5 + T_4) + C_3(t_4 - t_5) - C_5(T_5 - T_6)}{2(C_1 + C_2)} + t_5$$

To illustrate the above solutions, two sample problems have been worked out on the computer. The numerical and graphical results follow. Heat fluxes of 50,000 BTU/hr ft^2 and 1,500,000 BTU/hr ft^2 were used primarily because these are the problems used by H. Brahinsky in his work in this field and his work provides a comparative study of the final results. A mass flow G, of 2000 lbs/hr ft^2, is used for the coolant and a one inch plate is used with δ then becoming one quarter of an inch. A constant cross-sectional area and a cubic shaped volume are used in defining the geometry of the problem. Therefore, the area is equal to $\delta \cdot \delta$ and in the same way, the volume is equal to $\delta \cdot \delta \cdot \delta$. Although the words coolant and liquid are used interchangeably through out the work, the cooling fluid used is air. The plate is a high alumina refractory.

The following physical properties of these materials are to be found in Marks Mechanical Engineering Handbook, in the Handbook of Chemistry and Physics, and in Process Heat Transfer, by Kern.

$$C_p = .25 \quad BTU/lb\ ^\circ F$$

$$C = .23 \quad BTU/lb\ ^\circ F$$

$$K = 1 \quad BTU/ft^2\ hr\ ^\circ F$$

$$K_f = .033 \quad BTU/ft^2\ hr\ ^\circ F$$

$$P = .25$$

$$\rho = 128 \quad lbs/ft^3$$

$$\rho_f = .0753 \quad lbs/ft^3$$

The values of the initial metal temperatures and the area, volume relationships are listed below. Also, the time increment, $\Delta\theta$, is listed below. More will be said on the choosing of this later.

$$\Delta\theta = 1 \text{ sec.} = \frac{1}{3600} \text{ hr.}$$

$$\delta = \frac{1}{4}'' = \frac{1}{48} \text{ ft.}$$

$$A = \frac{1}{48} \cdot \frac{1}{48} \text{ ft}^2$$

$$V = \frac{1}{48} \cdot \frac{1}{48} \cdot \frac{1}{48} \text{ ft}^3$$

$$t_1 = t_2 = t_3 = t_4 = t_5 = 100^\circ F$$

$$h = 3$$

$$t_f = t_6 = 92^\circ F$$

Then the S term used in the previous equations must be considered. This is defined earlier in the thesis and its numerical value is gotten from the literature of A.J. Miles (13) which is listed in the bibliography. S is the average surface of the particles per unit volume. The dimensions would be in reciprocal feet. It is given by the following equations.

$$\frac{T}{S} = \frac{\delta}{6} \quad \therefore \quad S = \frac{T \cdot 6}{\delta}$$

T is the average volume of the particles and is given by,

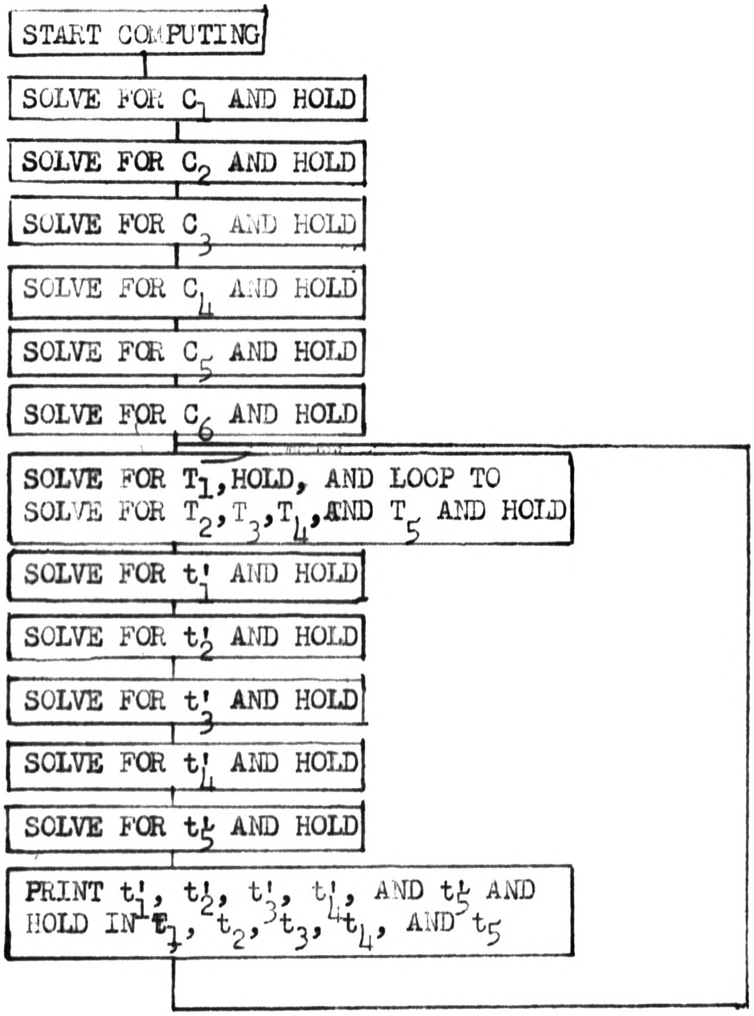
$$T = \left(\frac{1}{48}\right)^3 \cdot \frac{3}{4}$$

$$\therefore S = \frac{\left(\frac{1}{48}\right)^3 \cdot \frac{3}{4} \cdot 6}{\frac{1}{48}} = .00151909 \frac{1}{ft.}$$

The increment of time, $\Delta\theta$, was arbitrarily chosen to be one second and then a later check of the Fourier modulus has shown that this is well within the limits of this problem. The calculation for the Fourier modulus is given in Schneider (2).

The problem is solved by writing a floating point program for the equations. These problems take about forty five minutes to run to equilibrium.

FLOW CHART FOR COMPUTER



PROGRAM FOR LGP ROYAL MCBEE COMPUTER

```
;0004800'
/0000000'

r6300'u0400'i5200'1c0005'1i0002'1e0000'b5212's5210'
m5202'm5204'm5246'm5222'd5208'h5300'b5224'm5204'
m5246'm5210'm5222'd5208'h5302'b5212's5210'm5204'
m5214'm5216'd5206'h5304'b5218'm5204'm5210'm5216'
d5206'h5306'b5220'm5222'm5204'm5216'h5308'b5200'
m5204'm5206'm5226'm5216'h5310'a5308'h5312'b5310'
1m5230'h5314'b5308'm5240'd5312'1h5320'h5240'1z4847'
b5300'a5302'h5330'b5328's5326'h5332'b5308'a5306'

m5332'h5332'b5238's5236'm5304'h5334'b5242'm5204'
m5216's5334's5332'h5332'b5330'm5238'a5332'd5330'
h5340'b5326's5324'm5308'h5332'b5326'm5244'm5208'
a5324'a5328'm5306'h5334'b5236'm5244'm5208'a5238'
a5234'm5304'a5334's5332'd5330'd5208'a5236'h5342'
b5324's5322'm5308'h5332'b5324'm5208'm5244'a5326'
a5322'm5306'h5334'b5234'm5244'm5208'a5236'a5232'
m5304'a5334's5332'd5330'd5208'a5234'h5344'b5322'

s5320'm5308'h5332'b5322'm5208'm5244'a5324'a5320'
m5306'h5334'b5232'm5208'm5244'a5234'a5230'm5304'
a5334's5332'd5330'd5208'a5232'h5346'b5320's5228'
m5308'h5332'b5320'm5208'm5244'a5322'a5228'm5306'
h5334'b5232's5230'm5304'a5334's5332'd5330'd5208'
a5230'h5348'm0000'b5340'p0000'd0000'b5342'p0000'
d0000'b5344'p0000'd0000'b5346'p0000'd0000'b5348'
p0000'm0000'b5348'h5230'b5346'h5232'b5344'h5234'

b5342'h5236'b5238'h5246'b5340'h5238'b5246's5238'
u4804'

.0004800'
```

TEMPERATURE DISTRIBUTION IN A POROUS PLATE

(50,000 BTU/hr ft^2)

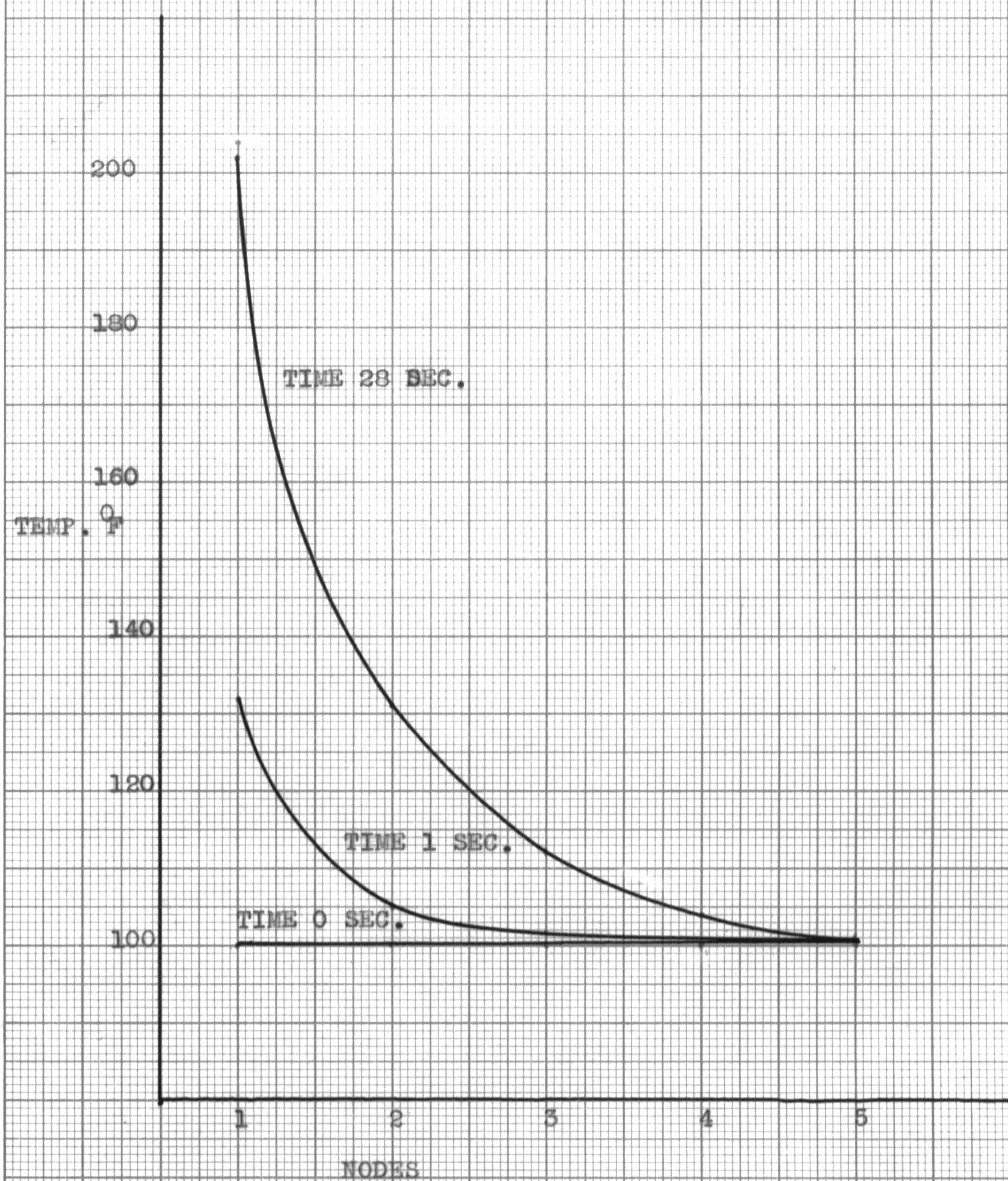
TABLE I

TIME SEC.	TEMPERATURE °F NODE 1	NODE 2	NODE 3	NODE 4	NODE 5
0	100.00000	100.00000	100.00000	100.00000	100.00000
1	132.08576	105.00352	101.78370	100.00000	100.00000
2	155.67354	111.98356	105.28971	103.12895	101.05380
3	168.02145	115.16891	108.01258	103.84850	101.14584
4	179.26715	119.00093	109.19444	103.99126	101.57118
5	186.4870	125.23347	109.31949	104.23859	101.61890
6	193.13711	127.74922	109.84866	104.38127	101.62718
7	197.64143	128.56881	111.13313	194.69006	101.62845
8	199.74910	128.79131	111.16122	104.71832	101.62851
9	201.45052	128.97623	111.58541	104.72951	101.62853
10	202.00231	129.25683	111.78659	104.72999	101.62851
11	202.08764	129.57810	112.00001	104.73156	101.62853
12	202.13159	131.00012	112.27904	104.73178	101.62851
13	202.19132	131.09836	112.35611	104.73179	101.62853
14	202.23229	131.23581	112.36774	104.73179	101.62851
15	202.39721	131.41671	112.36781	104.73179	101.62853
16	202.46513	131.68999	112.36783	104.73179	101.62851
17	202.48823	131.51078	112.36783	104.73179	101.62853
18	202.48859	131.63442	112.36783	104.73179	101.62851
19	202.48901	131.71352	112.36783	104.73179	101.62853
20	202.48923	131.75632	112.36783	104.73179	101.62851
21	202.48949	131.89223	112.36783	104.73179	101.62853
22	202.48973	131.81560	112.36783	104.73179	101.62851
23	202.48991	131.81672	112.36783	104.73179	101.62853
24	202.49135	131.81831	112.36783	104.73179	101.62851
25	202.49139	131.81835	112.36783	104.73179	101.62853
26	202.49142	131.81835	112.36783	104.73179	101.62851
27	202.49143	131.81835	112.36783	104.73179	101.62853
28	202.49143	131.81835	112.36783	104.73179	101.62851

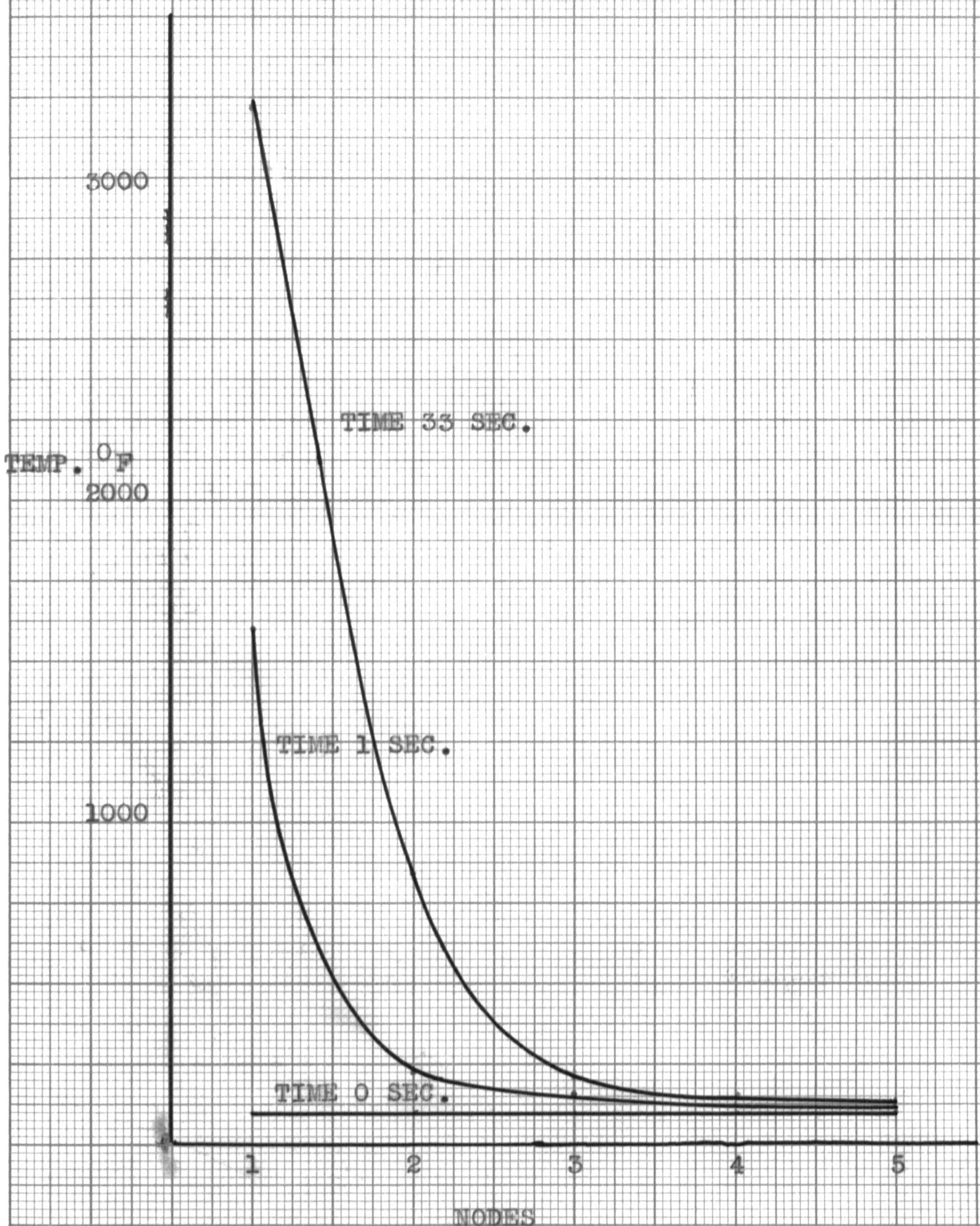
TEMPERATURE DISTRIBUTION IN A POROUS PLATE
(1,500,000 BTU/hr ft^2)

TABLE II.

TIME SEC.	TEMPERATURE °F				
	NODE 1	NODE 2	NODE 3	NODE 4	NODE 5
0	100.00000	100.00000	100.00000	100.00000	100.00000
1	1623.2086	235.23699	158.46579	116.57483	103.11368
2	2341.5901	367.59538	171.26458	121.59768	105.72380
3	2674.7911	482.47756	189.60089	124.22286	108.48611
4	2731.2328	593.29375	199.92574	126.57849	110.25340
5	2799.6224	720.54739	203.53758	127.35478	111.85034
6	2846.8921	795.02831	205.27589	127.36008	111.46578
7	2869.5555	804.51178	206.61819	127.36188	111.63591
8	2905.0184	819.67581	207.47892	127.36347	111.78321
9	2948.3972	821.92745	207.67324	127.36563	111.87743
10	2999.1512	821.98760	208.45698	127.36789	112.11699
11	3156.5064	821.99994	208.58970	127.36810	112.37259
12	3189.9852	822.00372	208.61893	127.36817	112.43611
13	3207.1257	822.16546	208.67549	127.36820	112.44975
14	3212.1329	822.20898	208.69811	127.36824	112.45003
15	3218.8745	822.21641	208.73967	127.36826	112.45061
16	3219.1561	822.21781	208.77352	127.36828	112.45061
17	3219.1948	822.21794	208.81906	127.36829	112.45061
18	3219.2226	822.21796	208.83541	127.36829	112.45061
19	3219.2942	822.21798	208.85760	127.36829	112.45061
20	3219.3288	822.21798	208.87910	127.36829	112.45061
21	3219.3352	822.21798	208.89564	127.36829	112.45061
22	3219.3681	822.21798	208.89111	127.36829	112.45061
23	3219.4008	822.21798	208.89117	127.36829	112.45061
24	3219.4079	822.21798	208.89117	127.36829	112.45061
25	3219.4167	822.21798	208.89117	127.36829	112.45061
26	3219.4283	822.21798	208.89117	127.36829	112.45061
27	3219.4300	822.21798	208.89117	127.36829	112.45061
28	3219.4353	822.21798	208.89117	127.36829	112.45061
29	3219. 4368	822.21698	208.89117	127.36829	112.45061
30	3219.4374	822.21798	208.89117	127.36829	112.45061
31	3219.4376	822.21798	208.89117	127.36829	112.45061
32	3219.4377	822.21798	208.89117	127.36829	112.45061
33	3219.4377	822.21798	208.89117	127.36829	112.45061

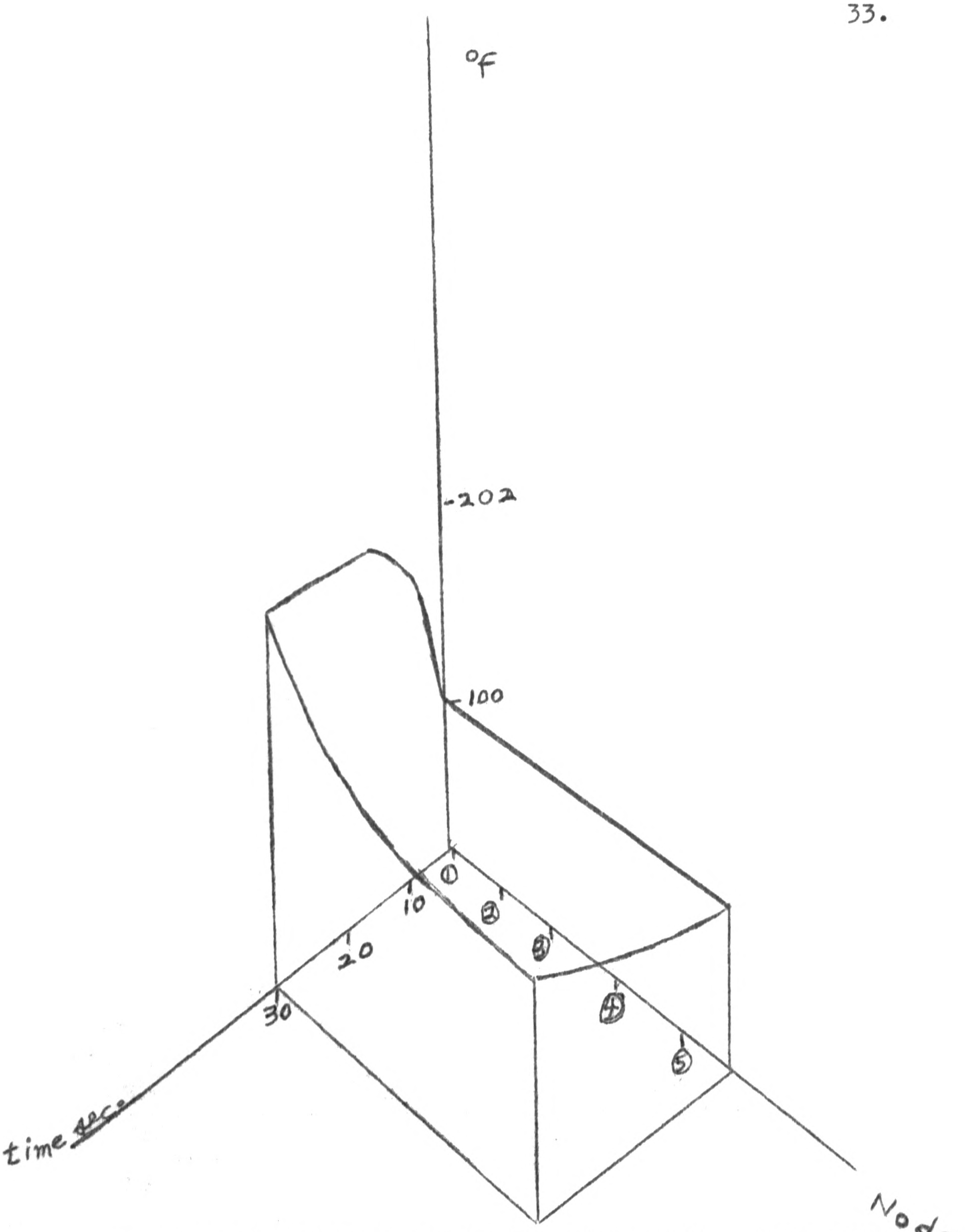


GRAPH NO. 1 - TEMPERATURE OF POROUS MEDIA
(50,000 BTU/hr. ft.^{2})

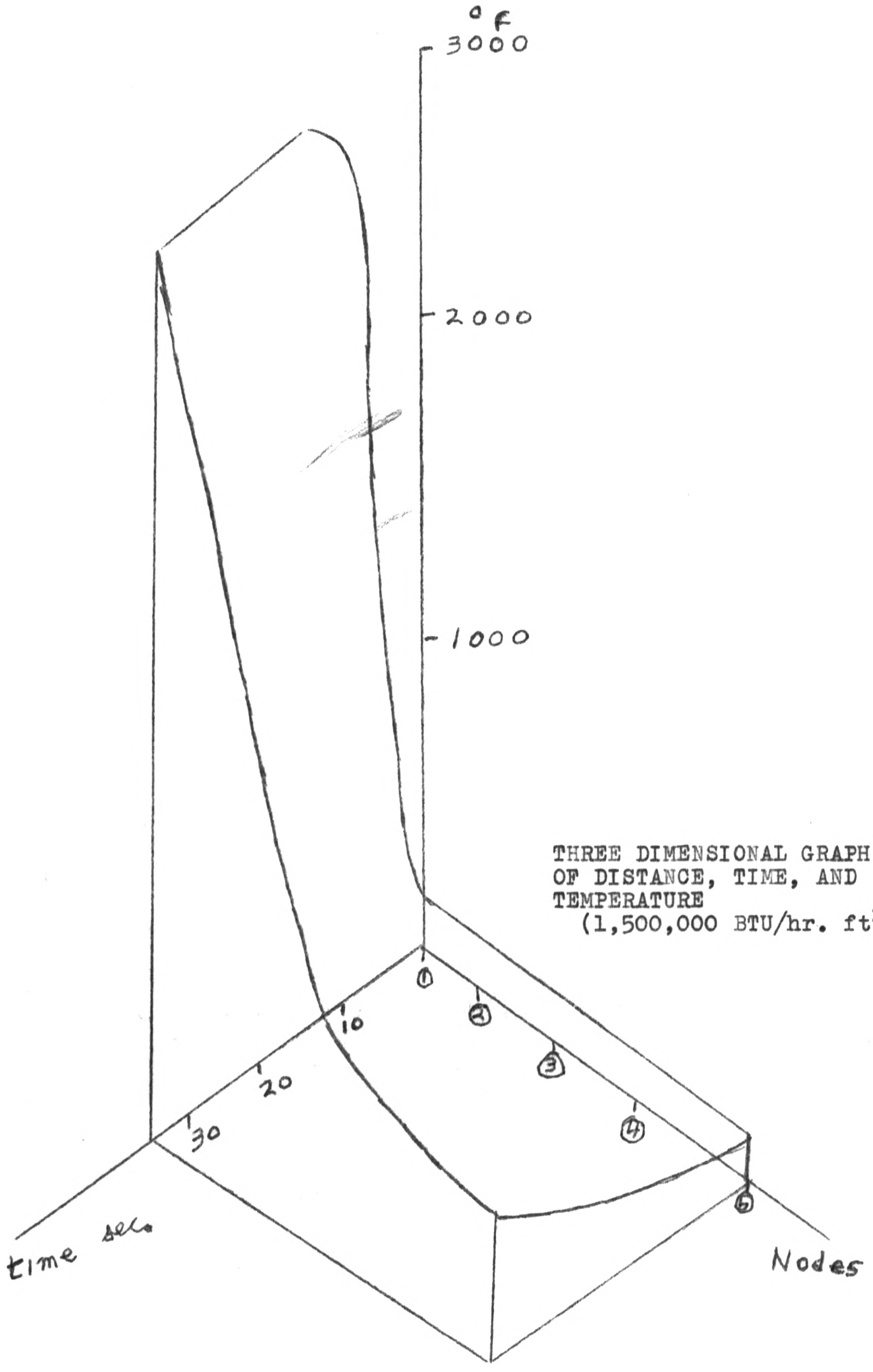


GRAPH NO.2 - TEMPERATURE OF POROUS MEDIA

(1,500,000 BTU/hr. ft^2.)



THREE DIMENSIONAL GRAPH OF DISTANCE, TIME, AND TEMP.
(50,000 BTU/hr. ft.^{2})



THREE DIMENSIONAL GRAPH OF DISTANCE, TIME, AND TEMPERATURE
(1,500,000 BTU/hr. ft.^{2})

It should be noted that although these programs take almost forty five minutes or better to reach equilibrium, one gets a quick idea of the answer in a much shorter time. B ecause of the nature of the program, it differs from the flow chart shown in this thesis to a small extent. When the program was first written, the commands for setting the index registers were put in front of the commands which compute the constants. This made necessary the use of a much larger loop in the program than was entirely necessary. This was never corrected because the author felt that any thing gained by this correction would not justify the use of more computer time.

The three dimensional pictures of the results are not by any means to scale and are only drawn to give the reader some idea of the surface upon which the temperatures lie. It is designed to give one a quick idea of the comparison of the coordinates.

VI. CONCLUSIONS.

The data of this thesis compares very closely with the data from the thesis done on this same problem by H. Brahinsky. This work predicts a somewhat slower rate of decline of temperatures from node to node then previous theoretical work but this is more in line with experimental evidence.

Much of the literature to be found on this subject is concerned with the discussion of the many uses for transpiration cooling, and therefore, this will not be covered here.

There has been no attempt made here to solve for optimum heat fluxes, mass flow rates, or volumes. This was only an attempt to come up with a more exact set of equations to cover a transient state of porous cooling. Probably the next step along the line would be to work on the various heat shield shapes to be used, the amount of mass flow which would be most efficient, and of course, to find an upper limit for the heat flux which would be covered by these equations.

Another area that could stand considerable attention is the third assumption used in this thesis. It is the author's conjecture that a proper weighting factor could be found such that it would make this a better relationship.

It probably would be of interest to attempt to write a program for a computer using these results which would, by a trial and error method, chose the correct factors of design to be used in the designing of a heat shield for the structural material in a reactor. As this is the author's primary field, he can see some immediate future for this method of cooling in the reactor field.

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VITA

William C. Wolkenhauer was born November 21, 1939, in Sault Ste Marie, Michigan. H e received his elementary and high school education in Escanaba, Michigan, where his parents moved when he reached the age of two.

He enrolled in Carroll College of Waukesha, Wisconsin on September 15, 1957 and finished his undergraduate work in January of 1961. In June of 1961 he received a Bachelor of Science degree in mathematics and physics from this school.

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