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STEADY AND TRANSIENT NATURAL CONVECTION IN ENCLOSURES BETWEEN HORIZONTAL CIRCULAR CYLINDERS (CONSTANT HEAT FLUX)

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Abstract—Steady state and transient heat-transport processes through a stagnant water layer between two horizontal circular cylinders, have been investigated experimentally. Concentric as well as eccentric configurations have been realized in an experimental set-up. Heat generated by electric current in the inner cylinder creates a constant heat flux condition. The heat is transported by free convective motion. Correlations fitting the data for the steady state have been obtained. Most of the transient results can be described with the aid of simplified mathematical models.

NOMENCLATURE

a, thermal diffusivity of water [m^2/s];
c_p, specific heat of water [J/kg°C];
D_i, diameter of inner tube [m];
D_0, inner diameter of outer tube [m];
g, acceleration due to gravity [m/s^2];
h_1, centreline distance of inner tube to top of water layer [m];
h_2, centreline distance of inner tube to bottom of water layer [m];
Nu, Nusselt number;
Q, heat removed by water jacket [W/m];
Q_i, initial heat flux before step change [W/m];
Q_f, final heat flux after step change [W/m];
ΔQ, step change in heat flux [W/m];
Ra, Rayleigh number;
R_{t1}, inner radius of outer tube [m];
R_{t2}, outer radius of outer tube [m];
t, time after step change [s];
T, temperature [°C];
T_i, mean wall temperature of inner tube [°C];
T_0, mean wall temperature of outer tube [°C];
T_1, temperature of ideally mixed volume part 1 [°C];
T_2, temperature of ideally mixed volume part 2 [°C];
T_c, temperature of cooling water jacket [°C];
ΔT_{loc}, local temperature drop [°C];
ΔT_{mean}, mean temperature drop [°C];
V, water volume per unit length [m^3/m];
V_1, ideally mixed volume 1 [m^3/m];
V_2, ideally mixed volume 2 [m^3/m];
V_a, apparent enlarged volume of water layer [m^3/m].

Greek symbols

β, thermal expansion coefficient [1/°C];
δ, gap width $\left(=\frac{D_0-D_i}{2}\right)$ [m];
λ, thermal conductivity of water [W/m°C];
λ_t, thermal conductivity of PVC [W/m°C];
ν, kinematic viscosity [m^2/s];
ρ, density of water [kg/m^3];
ϕ, exchanging water stream per unit length [$m^3/m.s$].

INTRODUCTION

HEAT dissipation in conductor and insulation limits the transmission capacity of buried cable systems. To increase the current rating, various cooling techniques have been proposed and tested [1, 2]. A possible method is surface cooling where the primary coolant (water) is in direct contact with the cable outside surface.

The possible failure of forced surface cooling in practical installations enhances the need for more knowledge of both heat transfer through and heat storage capacity of a stagnant enclosed water layer. The interests especially concern the transport processes by free convection with constant heat flux conditions during transient as well as steady state conditions.

"Snaking" of the cable inside the water pipe due to thermal expansion changes continuously the radial position of the cable relative to the pipe wall. The geometries involved are therefore not restricted to the annular configuration but the eccentricity of the inner cylinder must also be taken into account. The purpose of our work was to investigate the heat-transfer characteristics due to free convection between two horizontal circular cylinders with constant heat flux conditions under situations comparable with operating surface-cooled cable systems. As far as we know no correlations exist in literature which describe these processes and for that reason an experimental set-up was carried out.

LITERATURE

Some decades ago the first experimental results about the steady-state heat transfer by free convection between isothermal horizontal concentric cylinders were published [3, 4].

Both liquid and gas as the intermediate medium were considered and correlations were established between Nusselt-, Grashof- and Prandtl Numbers. Thirty years later Liu *et al.* [5] reported results correlated differently. During the Third International Heat Transfer Conference (1966) two papers, both dealing with gas-filled annuli, were presented [6, 7].

All the experimental results cited above were rearranged by Itoh *et al.* [8]. One single relation remained in which the Nusselt number is only dependent on a modified Rayleigh number. A more complicated mathematical expression based on boundary-layer concepts was recently derived by Kuehn and Goldstein [9]. They also studied the effect of eccentricity of the inner cylinder. An enhancement of the free convective motion occurs by positioning the inner cylinder below the centre of the outer cylinder (negative eccentricity). Experiments for positive eccentricities were not reported. In their equations the sign of the eccentricity was irrelevant. Similar experiments with enclosed spheres evidently showed an effect [10]. A decreasing heat transport for positive eccentricities was obtained.

The literature cited describes the heat-transfer characteristics with isothermal boundary conditions. Theoretically, a substantial deviation between isothermal and constant heat flux conditions may be expected. This is supported by experimental results of Kim *et al.* [11] studying the heat transfer from a single cylinder in an unbounded medium with both boundary conditions.

Some experiments of natural convection in enclosures with various changes in wall temperatures have been reported [12]. Changes in the constant flux heating have recently been described by Bar-Cohen [13]. The investigation however was restricted to vertical enclosures.

For the boundary conditions involved in this study the information available in the literature is not adequate to predict the transport processes in horizontal enclosed circular cylinders. Both transient and steady state situations with constant heat flux conditions have to be investigated experimentally.

EXPERIMENTAL SET-UP

In Fig. 1 a schematic view of the test section is given. A fixed outer tube (PVC), 299 mm I.D. and a thickness of 8 mm is cooled by a waterjacket manifolded to obtain a homogeneous flow circulation. A thermo-regulated mixing tank provided an adjustable temperature level of the jacket within 0.1°C.

A stainless steel inner tube is directly connected to an electric power supply. A uniform circumferential heat flux is then obtained. Two inner cylinders are used having diameters of 50 and 129 mm respectively. The position of the inner tube relative to the outer one is determined by the geometry of the end plates. For each diameter two sets of exchangeable end plates are available.

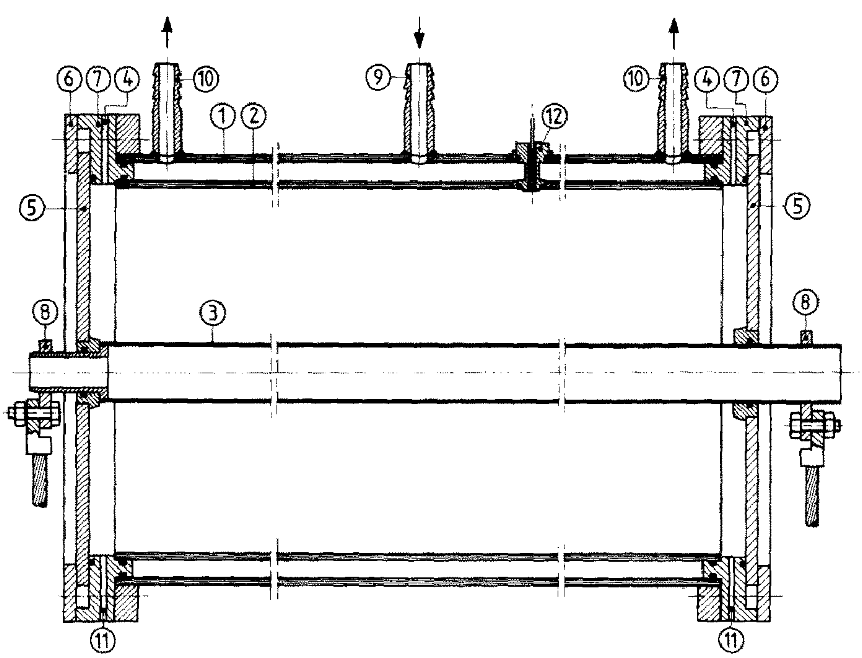

FIG. 1. Experimental set-up: 1. outside of water jacket; 2. outer cylinder; 3. inner cylinder; 4. connection to expansion vessel; 5. end plates; 6. attachment ring; 7. flanges; 8. connection to power supply; 9. inlet cooling water; 10. outlet cooling water; 11. drain of testsection; 12. thermocouple connection.

Four positions of the inner tube are considered. Apart from the concentric configuration the centre line of the inner tube is placed above, below or adjacent the centre line of the outer tube. The distance between the centre lines of both tubes is 115 mm for the 50 mm and 75 mm for the 129 mm inner pipe. The complete test section (2 m long) is insulated with glasswool.

With the aid of chromel–alumel thermocouples the temperatures of the inner and outer cylinder are measured at various axial and circumferential positions as indicated in Fig. 2. Every 2 h the data are collected by a computer controlled data-logger (Solarton). At each time interval the results of a heat balance, together with the data, are printed on a teletype.

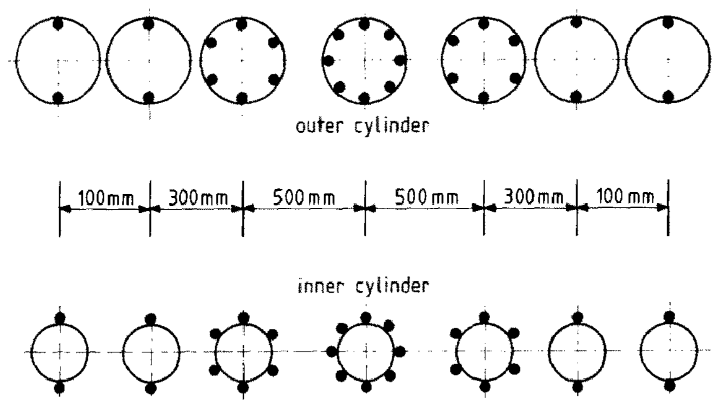

FIG. 2. Thermocouple location.

The experimental apparatus was automatically controlled by the logger. As soon as the heat balance indicated steady-state, the heat flux was changed stepwise. The auxiliary units are a motor driven variac and a power transformer. The outline of the circuit is given in Fig. 3.

For each geometry the heat flux was varied from 10 to 140 W/m. Step values in the flux (negative or positive) of maximal 120 W/m were performed.

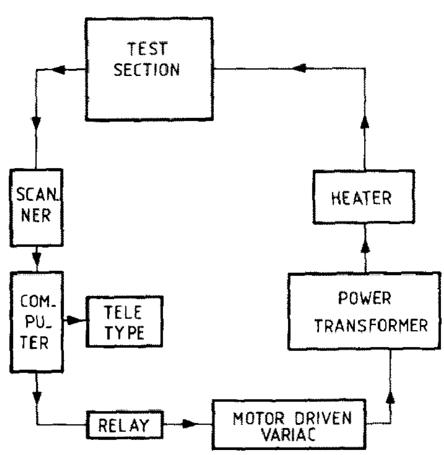

FIG. 3. Computer controlled power supply.

EXPERIMENTAL RESULTS (STEADY STATE)

In axial direction only a temperature gradient exists near the end plates. The convective motion in the inner part of the experimental set-up is therefore supposed to be two-dimensional and only data from thermocouples located in the latter area are used for the calculations.

Wall temperature gradients in circumferential direction of a few degrees centrigrades, especially for the outer tube, are obtained during the experiments. With such large gradients relative to the temperature drop across the water layer, a lot of information is lost by calculating merely an arithmetic mean of the wall temperature. For that reason also local temperature drops across the waterlayer are considered.

Dimensional analysis of the parameters governing free convection in concentric enclosures leads to familiar equations in which the Nusselt number is related to the Prandtl- and Grashof number. Very often the product of the two latter numbers is used (Rayleigh number).

The most simple expression is then:

$$Nu = f(Ra). \tag{1}$$

The dimensionless groups can be defined as:

$$Nu = \frac{Q}{2\pi\lambda\,(T_i - T_0)} \ln \frac{D_0}{D_i}, \tag{2}$$

and

$$Ra = \frac{\delta^3 g\,\beta(T_i - T_0)}{\nu a}, \tag{3}$$

in which T_0 and T_i are the mean wall temperature of outer- and inner cylinder, respectively. The physical properties of water are evaluated at the arithmetic mean of T_i and T_0.

Definitions of Ra or Nu purely based upon heat fluxes complicate the mathematical expressions considerably due to the configurations involved. Therefore the representation given above is preferred.

By eccentric positioning of the cylinders the convective transport processes decrease or increase

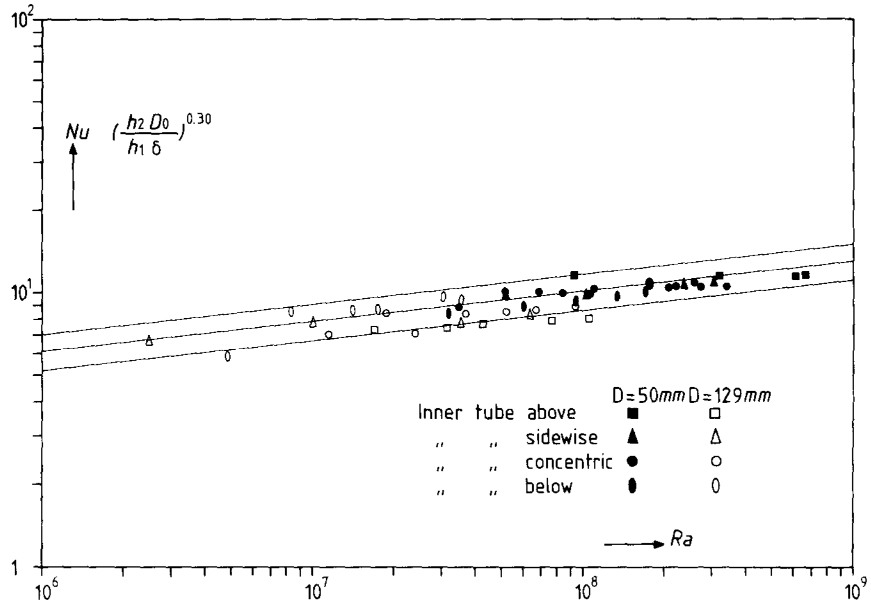

FIG. 4. Steady state heat-transfer results.

dependent on whether the inner pipe is above or below the centre line of the outer cylinder. An additional dimensionless group has to be introduced to describe the effect of the eccentricity. The experiments show that a sidewise displacement of the inner cylinder does not affect the heat-transfer characteristics. The distances from the inner cylinder wall to both bottom and top of the waterlayer appear to be significant. By combining the ratio of these distances together with the relative gap width the following dimensionless eccentricity group is suggested as extension of equation (1):

$$Nu = f\left(Ra, \frac{h_1 \delta}{h_2 D_0}\right). \tag{4}$$

Correlation of the data reveals that a simple expression could be achieved. Regression analysis results in a relation satisfying the data within an accuracy of 15%:

$$Nu = 1.34 Ra^{0.11}\left(\frac{h_1\delta}{h_2 D_0}\right)^{0.30} \tag{5}$$

All the measurements are plotted in Fig. 4. Due to the established flow patterns the temperature at the bottom of the waterlayer is rather insensitive to the value of the heat flux and mainly determined by the temperature of the jacket.

Higher temperatures exist at the opposite side of the layer. The temperatures here are strongly influenced by the convective motion. Local temperature drops at the top of the waterlayer relative to the mean temperature drop are related with the same dimensionless groups as used before i.e. the Rayleigh number and the eccentricity group.

Regression analysis leads now to the equation:

$$\frac{\Delta T_{\text{loc}}}{\Delta T_{\text{mean}}} = 2.94 Ra^{-0.06}\left(\frac{h_1\delta}{h_2 D_0}\right)^{0.20} \tag{6}$$

The data are shown in Fig. 5. The low exponent of the Rayleigh number indicates the temperature ratio is mainly dependent upon the position of the inner tube and almost independent upon the heat flux. Also here the same eccentricity group is adequate to correlate the results of the measurements which supports correctness of the used parameters.

TRANSIENT NATURAL CONVECTIVE HEAT TRANSFER

Transient transport phenomena in natural convective flows have hardly been investigated so comparison with similar studies could not be made. Also a theoretical background is not available. Analytical description of the transport processes is impossible due to the complexity of the equations of motion. In the scope of this study a numerical solution is not suitable. Therefore attempts were made to formulate the heat transport processes in simplified mathematical models.

As a first approximation a simple model has been worked out based upon two assumptions: an ideally mixed waterlayer and a momentarily change of the convective motion after a step change in the heat flux.

A heat balance leads to:

$$\rho c_p V \frac{dT}{dt} = Q_i - \frac{2\pi\lambda_t}{\ln\left(\frac{R_{t2}}{R_{t1}}\right)}(T - T_c). \tag{7}$$

If the heat flux changes stepwise:

$$Q_f = Q_i + \Delta Q \tag{8}$$

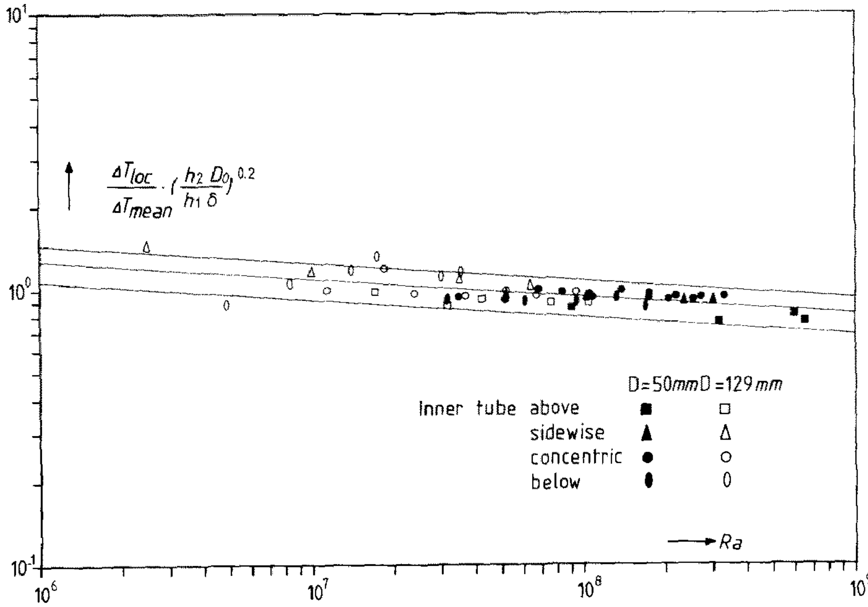

FIG. 5. Local temperature drop at upper side of waterlayer.

straight forward calculation gives:

$$\frac{Q-Q_i}{\Delta Q} = 1-\exp\left(-2\pi\lambda_t.t/\rho c_p V \ln\frac{R_{t2}}{R_{t1}}\right). \quad (9)$$

Comparison with the data (Fig. 7) shows that equation (9) does not adequately describe the transient response of the waterlayer.

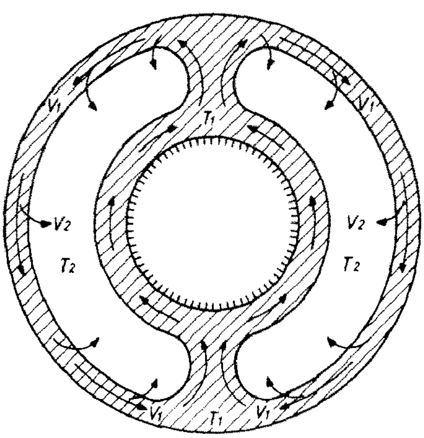

FIG. 6. Schematic view of convective motion.

A more sophisticated model has been set up. An obvious extension of the first approximation is a model with two ideally mixed volume parts. Examination of the flow patterns leads to a model which is schematically given in Fig. 6. The calculations are based upon the following assumptions:

two distinct volumes with different mean temperatures T_1 and T_2 exist;

the mixing time in each volume is small compared with the characteristic time of the transient response and therefore the volumes are considered to behave as ideally mixed;

only an exchanging waterstream ϕ transfers heat between the two volumes;

heat from the inner tube is transported to the outer tube exclusively through volume V_1;

volume V_2 is completely isolated from the walls;

V_1 and V_2 each represent about half the volume of the waterlayer.

This model is worked out in the appendix. The resulting equation is:

$$\frac{Q-Q_i}{\Delta Q} = 1-\exp\left[-2\pi\lambda_t.t\Big/\left(\rho c_p V \ln\frac{R_{t2}}{R_{t1}} + \frac{\pi\lambda_t V}{\phi}\right)\right] \quad (10)$$

Bij assuming an apparent enlarged volume of:

$$V_a = V + \pi\lambda_t V/\rho c_p \phi \ln\frac{R_{t2}}{R_{t1}} \quad (11)$$

equation (10) reduces to a form similar with equation (9):

$$\frac{Q-Q_i}{\Delta Q} = 1-\exp\left(-2\pi\lambda_t.t/\rho c_p V_a \ln\frac{R_{t2}}{R_{t1}}\right) \quad (12)$$

The apparent volume V_a is dependent upon the ratio V/ϕ. Various tube combinations will probably result in different values of the apparent volume. However, the experiments showed that for the tubes investigated only a slight tendency of geometry dependence exists. The additional part of the volume does not seem to be affected by the tube dimensions used in our experiments. A possible explanation is that both V_2 and ϕ are functions of the tube dimensions. In the ratio of these terms this dependency will disappear.

Various step changes in the heat flux have been performed as can be seen in Fig. 7–12. The solid line in the graphs represents equation (12). An apparent enlargement of the water volume of 30% gives the

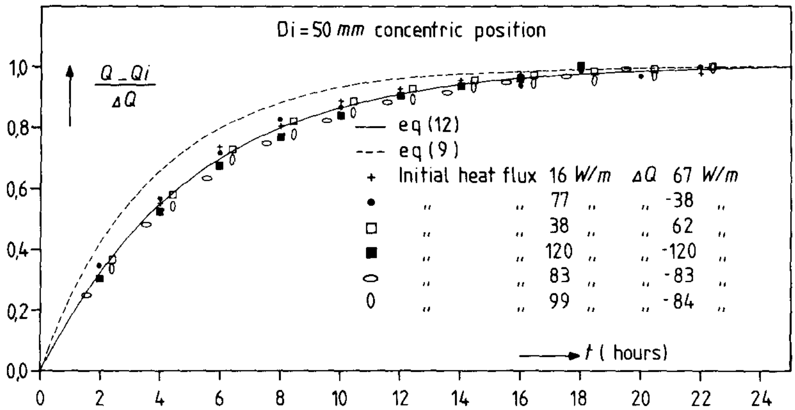

FIG. 7. Stepchange in heat flux.

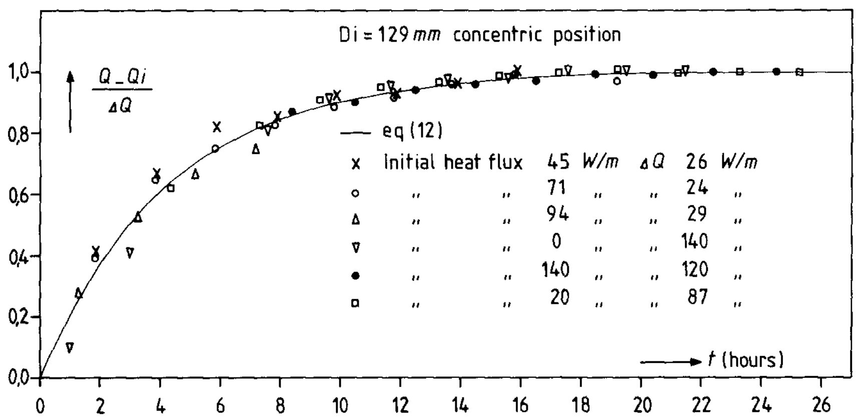

FIG. 8. Stepchange in heat flux.

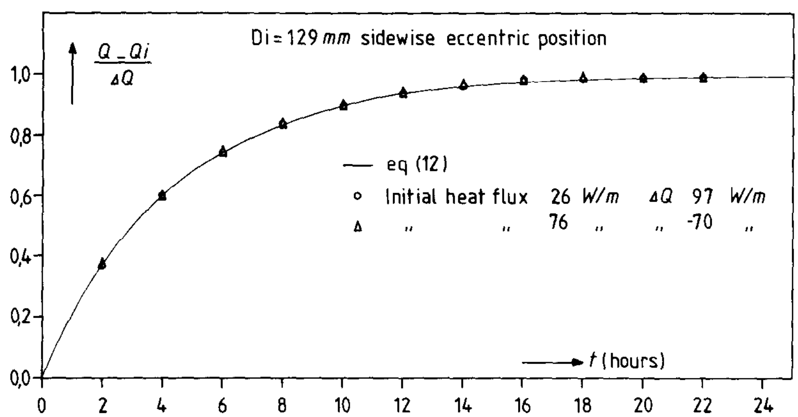

FIG. 9. Stepchange in heat flux.

best fit for all the data. The condition

$$\frac{\pi \lambda_t V}{\rho c_p \phi \ln \dfrac{R_{t2}}{R_{t1}}} < 0.4V$$

used in the derivation of equation (12) (see appendix) has been satisfied.

The mathematical model also describes the transient response of eccentric placed inner tubes (side wise and below the centre line). The assumptions might still be valid as could be expected from the nature of the established flow patterns.

For positive eccentricities the model does not agree with the experiments (Figs. 13 and 14). At the bottom of the waterlayer convective motion is hardly present. Buoyancy forces create flows only in the vicinity of the inner tube. The effective water volume contributing to the heat-transport process has decreased. The model will overestimate the time to reach a steady state situation.

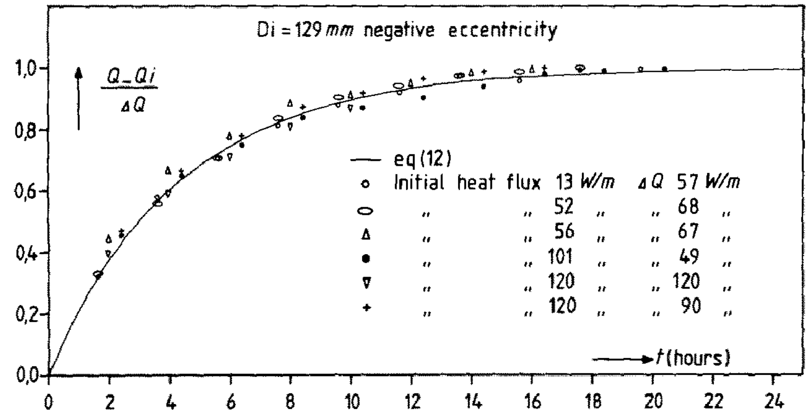

FIG. 10. Stepchange in heat flux.

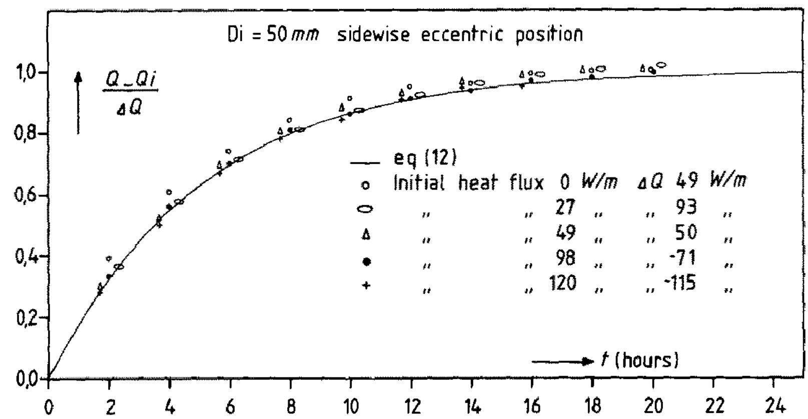

FIG. 11. Stepchange in heat flux.

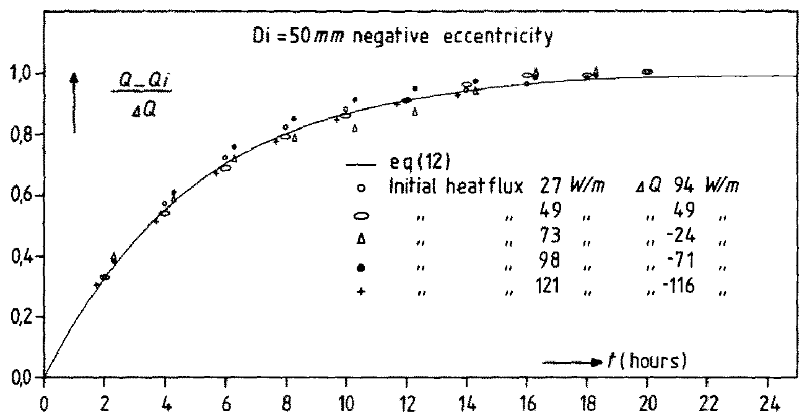

FIG. 12. Stepchange in heat flux.

From these experimental results it may be concluded that for most cases a rather simple mathematical model describes the transient heat transport between two enclosed circular cylinders. The model is based upon an ideally mixed waterlayer with an assumed enlarged water volume of 30%. For positive eccentricities the assumptions for the calculations become invalid. In this case the prediction of the heat transport is too conservative.

CONCLUDING REMARKS

From the experimental results overall steady-state as well as transient heat-transfer relations are obtained which can be used to estimate the cooling of buried cable systems surrounded by a water layer.

The eccentric position of the inner cylinder will effect the transport processes and in the steady state correlation a dimensionless group has been introduced describing this influence.

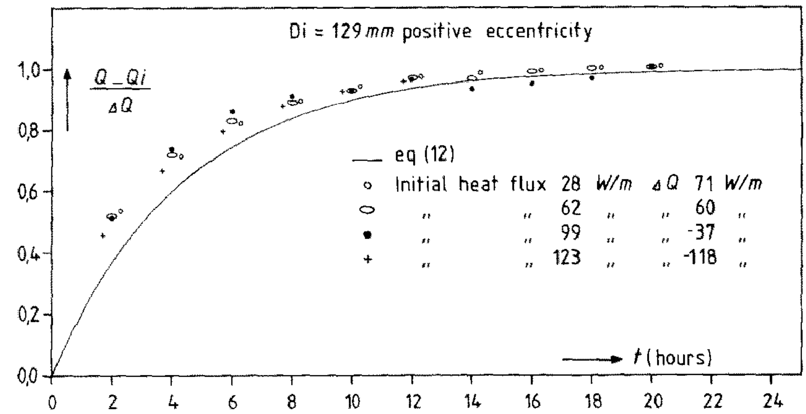

FIG. 13. Stepchange in heat flux.

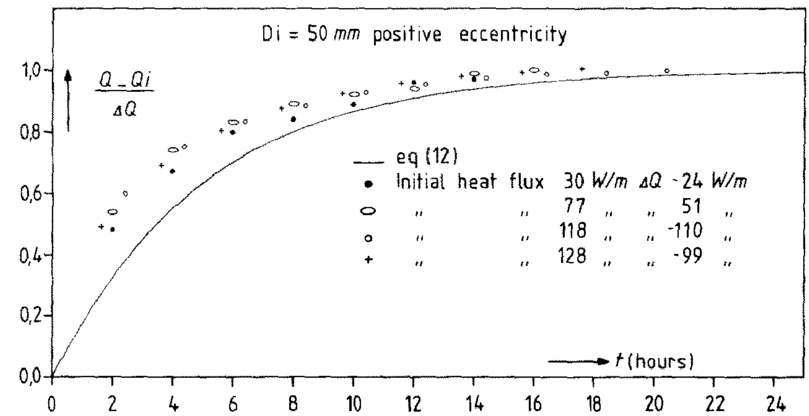

FIG. 14. Stepchange in heat flux.

The mathematical model used for the transient response of the waterlayer, based upon an apparent enlarged (30%) water volume is insensitive for inner tube positions at or below the centreline of the outer tube. Positive values of the eccentricity give too conservative results due to the changed contribution of the effective water volume. Considerable temperature gradients at the tube walls illustrate the difference between the two boundary conditions (isothermal and constant heat flux) commonly used in natural convection experiments.

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APPENDIX

Transient free convective heat transfer

A mathematical model based upon the existence of two separate ideally mixed volume parts of the water layer is worked out below.

According to the assumptions proposed in the text, a block diagram, representing the heat fluxes and exchanging streams, is shown in Fig. A1. A reduced heat balance over volume 1 leads to:

$$\rho c_p V_1 \frac{\mathrm{d}T_1}{\mathrm{d}t} = \Delta Q - \ln \frac{2\pi\lambda_t}{\frac{R_{t2}}{R_{t1}}} T_1 - \rho c_p \phi (T_1 - T_2). \tag{A1}$$

T_1 represents the variable part of the temperature and ΔQ the flux change.

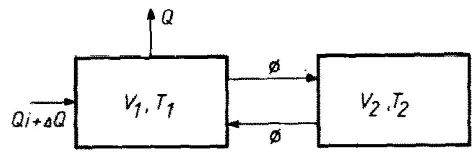

FIG. A1. Block diagram of mathematical model of two ideally mixed volume parts.

A similar balance over volume 2 gives:

$$\rho c_p V_2 \frac{\mathrm{d}T_2}{\mathrm{d}t} = \phi \rho c_p (T_1 - T_2). \tag{A2}$$

Application of Laplace Transforms to the simultaneous equations (A1) and (A2) and rearranging:

$$T_1^1(s) = \frac{\Delta Q \ln \frac{R_{t2}}{R_{t1}} (V_2 s + \phi)}{s\left(\rho c_p V_1 V_2 \ln \frac{R_{t2}}{R_{t1}} s^2 + s\left(\rho c_p \phi V \ln \frac{R_{t2}}{R_{t1}} + 2\pi\lambda_t V_2\right) + 2\pi\lambda_t \phi\right)}. \tag{A3}$$

By partial fractions:

$$T_1^1(s) = \frac{\Delta Q\, V_2}{\rho c_p V_1 V_2 (s+a)(s+b-a)} + \frac{\Delta Q \cdot \phi}{\rho c_p V_1 V_2 s (s+a)(s+b-a)}, \tag{A4}$$

with

$$a = \frac{2\pi\lambda_t \phi}{\rho c_p \phi \ln \frac{R_{t2}}{R_{t1}} \cdot V + 2\pi\lambda_t V_2} \tag{A5}$$

and

$$b = \frac{\rho c_p \phi V \ln \frac{R_{t2}}{R_{t1}} + 2\pi\lambda_t V_2}{\rho c_p V_1 V_2 \ln \frac{R_{t2}}{R_{t1}}}. \tag{A6}$$

In the derivation of the values of a and b the following approximation is applied:

$$(1-x)^{1/2} \simeq 1 - \frac{x}{2}. \tag{A7}$$

An accuracy of a few percent ($<4\%$) is obtained with the condition:

$$\frac{2\pi\lambda V_2}{\rho c_p \phi V \ln \frac{R_{t2}}{R_{t1}}} \simeq \frac{\pi\lambda V}{\rho c_p \phi V \ln \frac{R_{t2}}{R_{t1}}} < 0.4. \tag{A8}$$

The value of b is according to condition (A8) at least 10 times the value of a. This implies the inverse Laplace transform of equation (A4) simplifies considerably.

$$\frac{1}{(s+a)(s+b-a)} \xrightarrow{L^{-1}} \frac{\exp(-at) - \exp[(-b+a)t]}{b-2a}$$

$$\frac{1}{s(s+a)(s+b-a)} \xrightarrow{L^{-1}} \frac{\exp(-at)}{(2a-b)a} - \frac{\exp[(-b+at]}{(b-a)(2a-b)} + \frac{1}{a(b-a)}.$$

For not too small values of the time the inverse transformation reduces to:

$$\frac{1}{(s+a)(s+b-a)} \xrightarrow{L^{-1}} \frac{\exp(-at)}{b-2a}$$

$$\frac{1}{s(s+a)(s+b-a)} \xrightarrow{L^{-1}} \frac{\exp(-at)}{(2a-b)a} + \frac{1}{a(b-a)}.$$

Straight forward calculation with the assumption $V_1 \simeq V_2 = V/2$ and condition (A8) leads to the following equation for the variable part of temperature T_1.

$$T_1 = \frac{\Delta Q \ln \frac{R_{t2}}{R_{t1}}}{2\pi\lambda_t}\left[1-\exp\left(\frac{-2\pi\lambda_t \cdot t}{\rho c_p V \ln \frac{R_{t2}}{R_{t1}} + \frac{\pi\lambda_t V}{\phi}}\right)\right]. \tag{A13}$$

Superposition with the steady state solution yields the final result based upon heat fluxes:

$$\frac{Q-Q_i}{\Delta Q} = 1-\exp\left(\frac{-2\pi\lambda_t \cdot t}{\rho c_p V \ln \frac{R_{t2}}{R_{t1}} + \frac{\pi\lambda_t \cdot V}{\phi}}\right). \tag{A14}$$

CONVECTION NATURELLE STATIONNAIRE OU TRANSITOIRE DANS L'ESPACE ENTRE CYLINDRES CIRCULAIRES HORIZONTAUX (FLUX DE CHALEUR CONSTANT)

Résumé—On étudie expérimentalement le transfert thermique permanent ou transitoire à travers une couche d'eau entre deux cylindres circulaires horizontaux. Dans un montage expérimental, on a réalisé des configurations concentriques aussi bien qu'excentriques. La chaleur est apportée par un courant électrique dans le cylindre intérieur à flux constant. Cette chaleur est emportée par un mouvement de convection naturelle. On obtient des formules représentant les résultats. La plupart des résultats transitoires peut être décrite à l'aide de modèles mathématiques simplifiés.

STATIONÄRE UND INSTATIONÄRE FREIE KONVEKTION IN HOHLRÄUMEN ZWISCHEN HORIZONTALEN KREISZYLINDERN (KONSTANTE WÄRMESTROMDICHTE)

Zusammenfassung—Es wurden stationäre und instationäre Wärmetransportvorgänge in einer ruhenden Wasserschicht zwischen zwei horizontalen Kreiszylindern experimentell untersucht. Sowohl konzentrische wie exzentrische Anordnungen wurden beim experimentellen Aufbau verwirklicht. Die elektrische Beheizung im inneren Zylinder erzeugt einen Zustand konstanter Wärmestromdichte. Die Wärme wird durch freie konvektive Bewegung transportiert. Es wurden Korrelationen gewonnen, welche die Daten für den stationären Zustand wiedergeben. Die meisten instationären Ergebnisse können mit Hilfe eines vereinfachten mathematischen Modells beschrieben werden.

СТАЦИОНАРНАЯ И НЕСТАЦИОНАРНАЯ ЕСТЕСТВЕННАЯ КОНВЕКЦИЯ В ЗАМКНУТЫХ ПОЛОСТЯХ МЕЖДУ ГОРИЗОНТАЛЬНЫМИ КРУГЛЫМИ ЦИЛИНДРАМИ (СЛУЧАЙ ТЕПЛОВОГО ПОТОКА)

Аннотация — Проведено экспериментальное исследование процессов стационарного и нестационарного переноса тепла через неподвижный слой воды между горизонтальными круглыми цилиндрами. Исследовались как концентрические, так и эксцентрические конфигурации. Внутренний цилиндр нагревался электрическим током, так что создавалось условие постоянства теплового потока. Тепло переносилось свободной конвекцией. На основе экспериментальных данных по стационарному состоянию получены обобщенные соотношения. Большая часть результатов по нестационарному состоянию может быть описана на основе упрощенных математических моделей.