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Experimental investigation of parallel-walled and divergent-walled open thermosyphons

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Abstract—Natural convection experiments are performed for plane-walled, slot-like open thermosyphons having either parallel principal walls or divergent principal walls. The fluid in the thermosyphon and its surroundings is water ($Pr \simeq 5$). When the two principal walls are heated and maintained at the same uniform temperature which exceeds that of the surroundings, unstable operation is encountered. The instability is believed due to inherent obstructions to the transport of cool fluid from the surroundings to the thermosyphon, such transport being needed to make up for the hot fluid discharged from the thermosyphon to the surroundings. Steady-state Nusselt number results are obtained for the one-sided-heated thermosyphon. For the parallel-walled case, a universal Nusselt number correlation is achieved as the 0.25 power of the product of the inverse aspect ratio and the Rayleigh number. This correlation yields heat transfer coefficients that are independent of the interwall spacing. Furthermore, aside from a 7% difference in a multiplicative constant, the correlation is identical to that for natural convection in a one-sided-heated vertical channel open at both the top and bottom. Divergence of the principal walls of the one-sided-heated thermosyphon has only a small effect on the Nusselt number, with departures of no more than 5% from those for the parallel-walled case.

INTRODUCTION

THIS PAPER describes an experimental study of natural convection in plane-walled open thermosyphons. Two classes of such thermosyphons will be considered, as illustrated schematically in the side view in Fig. 1. In one class, the principal walls are parallel (left-hand diagram), while in the other class the principal walls are divergent (right-hand diagram). The experiments were performed with the thermosyphon assembly situated in a large test chamber filled with distilled water. The free surface of the water lay above the opening of the thermosyphon at a distance approximately equal to the depth of the thermosyphon cavity.

Two different heating conditions were employed during the course of the experiments. In the first, both principal walls were heated and maintained at the same uniform temperature which exceeded the temperature of the water in the test chamber. In the second (that designated in the figure), only one of the principal walls was heated to a uniform elevated temperature, while the other principal wall was unheated and took on a temperature very nearly equal to the water temperature. In both cases, the base and the side walls of the thermosyphon cavity were adiabatic.

The two-sided-heating experiments did not yield steady-state operation, thereby precluding the collection of quantitative heat transfer data. The non-attainment of steady operation in the two-sided-heating case can be made plausible by arguments which provide insights into the possible patterns of fluid flow. If steady operation had occurred, there would be a buoyant stream continuously rising along each of the two principal (heated) walls. Upon exiting the thermosyphon, each of the aforementioned streams of hot fluid would rise to the free surface of the water, forming a curtain-like sheet. If each of the twin sheets were continuous, it would be difficult for cool fluid from the surroundings (i.e. from the test chamber proper) to enter the inter-sheet space. Such an infusion of cool fluid is essential for the steady operation of the thermosyphon. Cool fluid, flowing downward, must enter the opening of the thermosyphon between the wall-adjacent upflows and penetrate into the cavity.

The stifling of the supply of cool fluid as described in the foregoing tends to preclude steady operation. In unsteady operation, the buoyant flow exiting the thermosyphon and rising to the free surface does not form a continuous sheet, thereby permitting cool fluid from the surroundings to have access to the opening of the thermosyphon.

It appears that the unsteady operation encountered here has not been addressed in the papers reviewed in the encompassing survey article by Japikse [1]. It may be conjectured that the instability was avoided by the installation of a cooling device just above the opening of the thermosyphon. The cooling device served to convert the hot stream(s) rising from the wall(s) of the thermosyphon into cool fluid which re-entered the opening of the thermosyphon and penetrated downward into the cavity. Because of this, there was no need for cool fluid to be transported from the surroundings to feed the thermosyphon opening, so that the shrouding effect encountered here was not an issue.

NOMENCLATURE

- A surface area of heated wall
- c_p specific heat
- g acceleration of gravity
- h average heat transfer coefficient, $Q/A(T_w - T_\infty)$
- k thermal conductivity
- L length of plate
- Nu_ξ Nusselt number, $h\xi/k$
- Pr Prandtl number
- Q rate of heat transfer
- Ra_ξ Rayleigh number, $[g\beta(T_w - T_\infty)\xi^3/\nu^2]Pr$
- S spacing between principal walls
- S_{max} maximum interwall spacing
- S_{min} minimum interwall spacing
- T_w temperature of heated wall
- T_w^* temperature of unheated wall
- T_∞ temperature of fluid surroundings.

Greek symbols

- β coefficient of thermal expansion
- θ half-angle of divergence
- μ viscosity
- ν kinematic viscosity
- ξ characteristic dimension
- ρ density.

When the experiments were performed with heating at only one of the principal walls, steady operation of the thermosyphon was achieved. This is plausible because the absence of a buoyant upflow from the unheated principal wall provided an obstruction-free path for cool fluid from the surroundings to reach and enter the opening of the thermosyphon.

The steady operation of the one-sided-heated thermosyphon enabled quantitative heat transfer data to be collected. The remainder of the paper will be focused on reporting the results for the one-sided-heating case, both for the parallel-walled and divergent-walled configurations of Fig. 1.

The relevant literature will now be briefly considered. As documented in ref. [1], the overwhelming majority of the published work on open thermosyphons has dealt with cavities of circular cross section. To the best knowledge of the authors, research on open thermosyphons of non-circular cross section is confined to refs. [2, 3]. In ref. [2], a thermosyphon of rectangular cross section was investigated, with emphasis on flow visualization. The reported thermal data involved the temperature along the symmetry line of the cavity—a quantity which would usually not be known to prospective users.

Thermosyphons with both triangular and rectangular cross sections were studied in ref. [3]. The heat transfer correlations given there for water are for the turbulent regime, whereas the present experiments fall in the laminar regime. It is also noteworthy that all bounding surfaces of the rectangular thermosyphons of refs. [2, 3] were heated, in contrast to the one-sided heating employed here.

EXPERIMENTAL APPARATUS

The description of the experimental apparatus is facilitated by reference to Fig. 1. The same apparatus

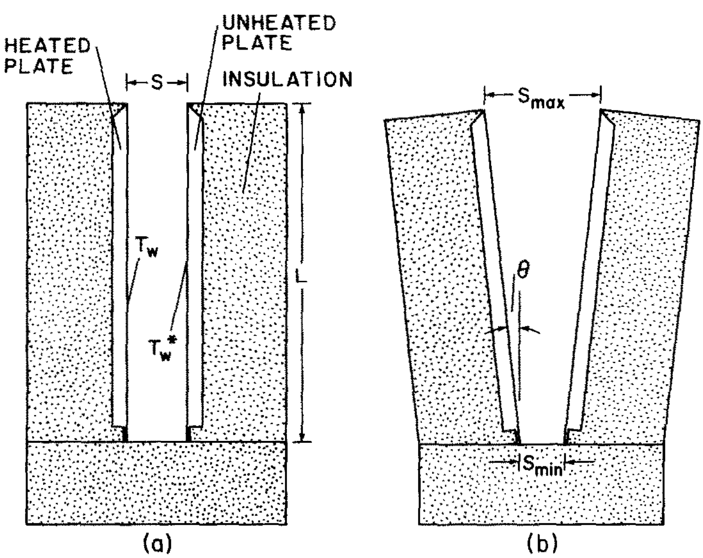

FIG. 1. Schematic diagrams of the investigated plane-walled thermosyphons: (a) parallel principal walls; (b) divergent principal walls.

was used for the parallel-walled and divergent-walled cases, the only difference being in the parallelism or non-parallelism of the principal walls. Each principal wall was made up of a copper plate insulated from behind by a block of water-tolerant, closed-pore insulation (Styrofoam). Each plate had a length L of 14.52 cm, a width W (normal to the plane of the figure) of 9.67 cm, and a thickness of 0.635 cm. The upper and lower edges of the plate were contoured to avoid extraneous heat losses at those locations. The exposed surface of each plate (i.e. the bounding surface of the cavity) was maintained free of tarnish (oxidation) throughout the experiments.

Both plates were designed and fabricated to be heated, but, for the reasons discussed earlier, only one plate was heated during the experiments in which quantitative data were collected. Heating was accomplished by means of an electrical resistance wire embedded in shallow grooves in the rear surface of the plate. The heating wire for each plate was subdivided into three separate and independently controlled circuits fed by a regulated a.c. supply. The control provided by the independent heating circuits in conjunction with the high thermal conductivity of the copper enabled nearly perfect temperature uniformity to be attained at the exposed surface of the heated plate.

Each plate was provided with eight calibrated, 30-gage, chromel–constantan thermocouples. The thermocouples were installed in holes drilled into the rear of the plate, with the junctions positioned no more than 0.05 cm from the exposed face. The junctions were held in place with copper oxide cement which also facilitated good thermal contact. Typically, for the heated plate, the difference between an individual thermocouple reading and the plate mean temperature was about 1% of the temperature difference between the plate and the fluid surroundings. For the unheated plate, temperature uniformity was within 1–2 μV for those experiments for which data are reported here.

The rear surface of each of the copper plates was covered by a block of 3.5-cm-thick Styrofoam insulation which ensured that the heat losses there would be negligible. Furthermore, as seen in Fig. 1, the base of the thermosyphon assembly was also a block of Styrofoam. The face of the Styrofoam which closed the bottom of the cavity was covered with plasticized tape to provide a hydrodynamically smooth, impermeable surface. The side walls of the thermosyphon (not shown) were 1.27-cm-thick plexiglass plates.

Figure 1 also conveys nomenclature relevant to the experiments. The plate length is L, as already noted. The interwall spacing for the parallel-walled case is denoted by S. For the divergent-walled case, the minimum and maximum interwall spacings are S_{min} and S_{max}, respectively, and θ is the half-angle of divergence. The value of θ for the divergent-walled thermosyphon illustrated in Fig. 1 is 6°. The temperatures of the heated and unheated principal walls are T_w and T_w^*, respectively.

As was noted in the Introduction, the experiments were performed with the thermosyphon situated in a large distilled-water-filled test chamber. The test chamber was thermally guarded by means of a tank-in-tank arrangement. In this setup, the test chamber was surrounded by a still larger chamber. Distilled water was also used to fill the inter-chamber space. A temperature control and water circulation unit situated in the inter-chamber space facilitated the thermal guarding of the test chamber. The respective dimensions of the test chamber and the outer chamber were $74 \times 43 \times 44$ cm and $102 \times 66 \times 48$ cm (length $\times$ width $\times$ height). Both chambers were fabricated from plexiglass. The open top of the chamber assembly was covered by a coated sheet of Styrofoam insulation lined with a plastic vapor barrier. The cover suppressed heat and moisture losses from the free surfaces of the water in the test chamber and the inter-chamber space.

The thermosyphon assembly was positioned in the test chamber with its axis vertical. It was centered with respect to the length and width of the chamber and with respect to the height of the water. The opening of the thermosyphon was about 14 cm below the free surface of the water.

Thermocouples were installed in the test chamber to measure the temperature T_∞ of the fluid surroundings of the thermosyphon (needed in the data reduction) and to monitor possible stratification. There were three thermocouples arranged in a vertical array, respectively positioned 7.6, 16.5, and 24.1 cm above the floor of the chamber. The array was situated 12 cm to the side of the thermosyphon. The thermocouples were made from Teflon-coated chromel–constantan wire, chosen both for inertness in water and high thermoelectric sensitivity.

The temperature information provided by the aforementioned thermocouple array was supplemented by that from a pair of 0.1°F ASTM-certified thermometers. One of these was placed in the test chamber and the other in the inter-chamber space, with the placement being such that the two thermometers were very close to each other.

A precondition for the initiation of a data run was that the fluid in the test chamber be quiescent. To fulfill this condition, it was required that the thermometers in the test chamber and in the inter-chamber space display the same reading, that the vertical array of thermocouples show no stratification, and that the thermocouples in the plates and in the fluid give the same temperature as that of the thermometers. In addition, at least an hour was allowed between stirring of the water in the test chamber (to promote temperature homogeneity) and the initiation of a data run.

A data run was initiated by supplying power to the heating circuits of one of the principal walls of the thermosyphon. The power settings of the individual circuits had been preset, on the basis of experience, to provide a uniform temperature at the heated plate.

Depending on the power level and on the configuration of the thermosyphon (e.g. interwall spacing and divergence angle), the time required to achieve steady state ranged between 20 and 35 min. For the operating conditions for which quantitative data were collected, the largest temperature stratification in the test chamber was 2% of the temperature difference between the heated plate and the fluid surroundings.

RESULTS AND DISCUSSION

Data reduction

The results to be reported here are average Nusselt numbers for the one-sided heated thermosyphon. The Nusselt numbers were evaluated from the experimental data by means of the defining equations

$$h = Q/A(T_w - T_\infty), \quad Nu_\xi = h\xi/k \tag{1}$$

where ξ is a characteristic dimension to be specified shortly. In the equation for h, the quantities A and T_w are, respectively, the surface area and the temperature of the heated wall of the thermosyphon, while T_∞ is the temperature of the fluid surroundings. Both T_w and T_∞ are averages, over eight plate thermocouples and three fluid thermocouples, respectively. In view of the near uniformity of the respective sets of thermocouple readings, as documented earlier the averaging is entirely acceptable. The heat transfer rate Q is the sum of the electric power inputs to the three circuits of the heated plate.

The Nusselt number results will be presented as a function of the $(\xi/L)Ra_\xi$ group, where the Rayleigh number was evaluated from

$$Ra_\xi = [g\beta(T_w - T_\infty)\xi^3/\nu^2]Pr. \tag{2}$$

The thermophysical properties appearing in equations (1) and (2) were evaluated at T_w, as suggested in ref. [4]. Algebraic representations for k, ρ, μ, and c_p in the temperature range of interest here are presented in ref. [5], and β, ν, and Pr follow directly (β by differentiation of the representation for ρ). The Prandtl number (based on T_w) ranged from 4.75 to 5.20 for the operating conditions of the experiments, so that the Nusselt number results can be characterized as corresponding to a nominal Prandtl number of 5.

Parallel-walled thermosyphons

Attention will first be focused on the Nusselt number results for one-sided-heated, parallel-walled thermosyphons. The results encompass experiments performed using six distinct cavity aspect ratios, namely, $L/S = 2.29$, 2.86, 3.81, 4.57, 5.72, and 7.62. This range was achieved by varying the interwall spacing S at a fixed plate length L. Experiments at larger L/S (i.e. smaller interwall spacing) were also performed, but time-varying wall temperatures were encountered. This unsteady operation for narrow cavities is believed due to the cavity opening being bridged by the hot, buoyant stream rising from the heated wall, thereby making it difficult for cool fluid from the surroundings to enter the opening.

Another specification relating to the Nusselt number results to be presented is that they correspond to operating conditions for which the temperature T_w^* of the unheated wall was very nearly equal to the temperature T_∞ of the fluid surroundings. Specifically, the presented results satisfy the criterion

$$(T_w^* - T_\infty)/(T_w - T_\infty) < 0.02. \tag{3}$$

For a given aspect ratio L/S, the temperature ratio $(T_w^* - T_\infty)/(T_w - T_\infty)$ increased with increasing Rayleigh number, so that criterion (3) served to cap the reported Rayleigh number range for each L/S. Only at the largest reported Rayleigh number for each L/S did the temperature ratio of criterion (3) approach 2%; at lower Rayleigh numbers, the values of the ratio were below 0.5%.

With the foregoing as background, the Nusselt number results will now be presented in Fig. 2. In the figure, Nu_S is plotted as a function of $(S/L)Ra_S$, which extends over the range from about 2×10^4 to 1.5×10^7. The data for the individual aspect ratios L/S are delineated by separate symbols. In addition to the data, the figure contains a solid line which represents a least-squares fit of the data. The equation of the line is

$$Nu_S = 0.583[(S/L)Ra_S]^{0.252}. \tag{4}$$

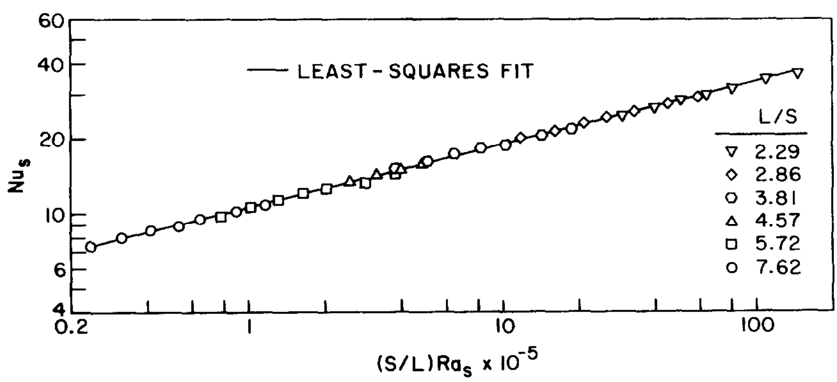

FIG. 2. Nusselt number results for one-sided-heated, parallel-walled thermosyphons.

It is noteworthy that 85% of the data fall within $\pm 2\%$ of the correlating line and that the maximum deviation of the data from the line is 3.6%.

Inspection of Fig. 2 shows that the use of $(S/L)Ra_S$ as the correlating parameter brings together the data for the various L/S; that is, there is no separate dependence of Nu_S on L/S. Also worthy of note is the excellence of the power-law-type (i.e. straight line) representation.

The 0.252 exponent is so close to the standard 0.25 laminar-regime exponent that a correlation using the latter was sought, with the result

$$Nu_S = 0.598[(S/L)Ra_S]^{0.25}. \tag{5}$$

It was found that 87% of the data fall within $\pm 2\%$ of equation (5) and that the maximum deviation of the data from the equation is 3.7%. Furthermore, the deviations between equations (4) and (5) are well within 1% over the $(S/L)Ra_S$ range of Fig. 2. Therefore, these equations may be regarded as being identical for any practical purpose. The 0.25 exponent of equation (5) strongly suggests that the flow was laminar for all the operating conditions of Fig. 2.

Another highly significant feature of equation (5) is that the interwall spacing S cancels out of both sides of the equation (note that the bracketed quantity is proportional to S^4). Thus, the heat transfer coefficient h is independent of the interwall spacing for the operating conditions considered here.

With regard to comparisons of the results of Fig. 2 with the literature, it has already been noted in the Introduction that neither of the published papers [2, 3] on open thermosyphons of rectangular cross section dealt with the one-sided-heated case. Apart from this, comparisons are further precluded because the thermal results of ref. [2] involve temperatures along the symmetry line of the cavity and those of ref. [3] for water are for the turbulent regime.

The fact that the present Nusselt number results are correlated with $(S/L)Ra_S$ suggests a comparison with the results for the one-sided-heated vertical channel open at both the top and bottom, since the same parameter is applicable there. For the channel, from ref. [6]

$$Nu_S = 0.642[(S/L)Ra_S]^{0.25}. \tag{6}$$

Comparison of this equation with equation (5) indicates that the channel Nusselt numbers are about 7% higher than the thermosyphon Nusselt numbers. That the channel Nusselt numbers are higher is quite reasonable since the channel provides a much simpler and more open fluid flow path than does the thermosyphon. Indeed, the fact that the deviation is only 7% is somewhat surprising.

Divergent-walled thermosyphons

Attention will now be turned to one-sided-heated, divergent-walled thermosyphons. All of the Nusselt number results to be reported here correspond to $L/S_{\min} = 7.62$. Smaller $S_{\min}$ values were considered,

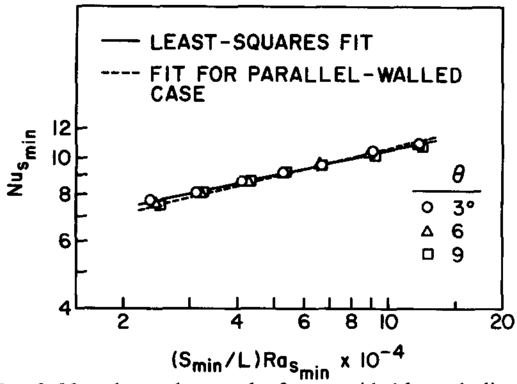

FIG. 3. Nusselt number results for one-sided-heated, divergent-walled thermosyphons.

but the temperature condition given by equation (3) was not fulfilled. Larger $S_{\min}$ could not be employed because of constraints imposed by the support structure. (The actual limitation was related to the size of $S_{\max}$.)

The experiments were performed for half-angles of divergence $\theta = 3$, 6, and 9°. The illustrated θ value in Fig. 1 is 6°. Although 6° may, at first, seem to be a small divergence, the $S_{\max}/S_{\min}$ ratio for the illustrated case is 2.59, which corresponds to a major variation in the cross-sectional dimensions of the thermosyphon.

The Nusselt number results are presented in Fig. 3. As seen there, $S_{\min}$ is used as the characteristic dimension for both the Nusselt number and the abscissa variable which, except for the replacement of S by $S_{\min}$, is the same as that of Fig. 2. Because $S_{\min}$ is common to all the data in Fig. 3, its use as the characteristic dimension facilitates an unambiguous appraisal of the effect of the divergence half-angle θ, which is used to parameterize the data. In addition to the data, the figure contains two lines. The solid line is a least-squares fit of the data for the divergent-walled case, while the dashed line represents the least-squares fit for the parallel-walled case, equation (4).

Inspection of Fig. 3 indicates that the divergence of the thermosyphon walls has a minor effect on the heat transfer results. The majority of the data fall within 2% of the correlation for the parallel-walled thermosyphon, with a greatest deviation of just under 5%. Furthermore, the data points for $\theta = 3$, 6, and 9° are quite insensitive to θ. The greatest apparent sensitivity, about 5%, may well be scatter related, since in adjacent clusters the data are independent of θ.

The least-squares and 0.25-exponent fits for the data for the divergent-walled thermosyphon are given by

$$Nu_{S_{\min}} = 0.811[(S_{\min}/L)Ra_{S_{\min}}]^{0.222} \tag{7}$$

$$Nu_{S_{\min}} = 0.598[(S_{\min}/L)Ra_{S_{\min}}]^{0.25}. \tag{8}$$

Equation (7) is an excellent representation of the data, as witnessed by the fact that 95% of the data fall within $\pm 2\%$ of it, and the greatest deviation is 3.3%. The representation provided by equation (8) is almost as good: 81% of the data fall within $\pm 2\%$ of the equation, with a greatest deviation of 4.2%.

It is remarkable that the 0.25-exponent fit for the divergent-walled thermosyphon, equation (8), is identical to the 0.25-exponent fit for the parallel-walled thermosyphon, equation (5). This is further evidence of the insensitivity of the heat transfer results to the divergence. Note that the interwall spacing cancels out in both equations (5) and (8).

CONCLUDING REMARKS

Plane-walled (i.e. slot-like) open thermosyphons were investigated experimentally here for two heating conditions:

(1) both principal walls were heated and maintained at the same uniform temperature which exceeded the temperature of the fluid surroundings;

(2) one principal wall was heated to a uniform elevated temperature while the other principal wall was unheated.

The base and the side walls of the thermosyphon cavity were adiabatic. The experiments were performed with the thermosyphon situated in a large, thermally guarded test chamber filled with distilled water.

Steady-state operation was not achieved for the two-sided-heated case. This behavior is believed due to the blockage of the flow of cool fluid from the test chamber proper to the opening of the thermosyphon. It is conjectured that the blockage was caused by a pair of twin, curtain-like sheets which extended upward from the opening-adjacent edges of the principal walls to the free surface of the water. Each sheet was a plume created by the buoyant upflow which rises along each of the heated principal walls. In unsteady operation, the sheets are not continuous, so that cool fluid from the test chamber proper can intermittently enter the thermosyphon.

The aforementioned unsteady operation appears not to have been previously addressed in the literature on open thermosyphons. It is conjectured that the instability was averted by the installation of a cooling device just above the opening of the thermosyphon. This device converted the hot fluid exiting the thermosyphon into cool fluid, which then re-entered the thermosyphon. This recycling obviated the need for fluid from the surroundings to be delivered to the thermosyphon and, thereby, rendered irrelevant any possible blockage of such a flow. An opening-adjacent cooling device was not used in the present experiments, so that cool fluid had to be delivered from the surroundings to sustain the operation of the thermosyphon.

The foregoing discussion underscores the fact that the operation of an open thermosyphon is not wholly determined by events which occur within the thermosyphon itself. Rather, consideration must also be given to fluid flow and thermal processes in the space beyond the thermosyphon opening.

For the case in which only one of the principal walls was heated, steady-state operation was achieved, and average Nusselt numbers were evaluated from the experimental data. The reported Nusselt numbers correspond to operating conditions for which the temperature of the unheated principal wall was virtually equal to the temperature of the fluid surroundings. These results were obtained both for thermosyphons in which the principal walls are parallel and in which the principal walls are divergent.

For the parallel-walled case, the aspect ratio L/S was varied parametrically. The Nusselt numbers Nu_S were very tightly correlated as a function of the single dimensionless group $(S/L)Ra_S$, without a separate dependence on the aspect ratio. The correlation was of the power-law form, with a 0.25 exponent for $(S/L)Ra_S$. This exponent is indicative of the laminar regime. Also, for this exponent, the interwall spacing S cancels out of the correlation, so that the heat transfer coefficient is independent of S.

The Nusselt number correlation for the one-sided-heated, parallel-walled thermosyphon is, aside from a multiplicative constant, identical to that for natural convection in a one-sided-heated vertical channel open at both the top and bottom. The multiplicative constant for the channel is about 7% greater than that for the thermosyphon.

For the divergent-walled case, the half-angle of divergence was varied parametrically. It was found that divergence had a minor effect on the heat transfer results. In the main, the Nusselt numbers for the divergent case were within 2% of those for the parallel-walled case, with the maximum deviation being just under 5%.

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ETUDE EXPERIMENTALE DE THERMOSIPHONS A PAROIS PARALLELES OU DIVERGENTES

Résumé—Des expériences de convection naturelle sont conduites sur des thermosiphons ouverts, rectangulaires, à parois principales planes soit parallèles, soit divergentes. Le fluide circulant est de l'eau ($Pr \cong 5$). Lorsque les deux parois principales sont maintenues à la même température uniforme, supérieure à l'environnement, on constate un fonctionnement instable. L'instabilité est attribuée à des obstructions inhérentes au transport de fluide froid depuis l'environnement dans le thermosiphon, un tel transport étant nécessaire pour permettre la décharge du fluide chaud du thermosiphon vers l'environnement. On donne des résultats de nombre de Nusselt obtenus pour un thermosiphon chauffé d'un seul côté. Pour le cas des parois parallèles, une formule du nombre de Nusselt est construite sur la puissance 0,25 du produit de l'inverse du rapport de forme par le nombre de Rayleigh. Cette formule donne les coefficients de transfert thermique qui sont indépendants de l'espacement. Avec une différence moindre que 7% sur une constante multiplicative, la formule est identique à celle de la convection naturelle dans un canal vertical ouvert aux deux extrémités et chauffé sur un seul côté. La divergence des parois principales du thermosiphon chauffé sur un seul côté n'a qu'un faible effet sur le nombre de Nusselt, avec des écarts qui ne dépassent pas 5% par rapport au cas des parois parallèles.

EXPERIMENTELLE UNTERSUCHUNG VON OFFENEN THERMOSYPHONS MIT PARALLELEN UND DIVERGIERENDEN WÄNDEN

Zusammenfassung—Experimente zur natürlichen Konvektion bei spaltähnlichen, ebenwandigen Thermosyphons mit entweder parallelen oder divergierenden Hauptwandungen werden vorgestellt. Das Fluid in den Thermosyphons und in deren Umgebung ist Wasser ($Pr \simeq 5$). Wenn die zwei Hauptwände beheizt und auf gleicher Temperatur oberhalb der Umgebungstemperatur gehalten werden, wird ein instabiles Verhalten festgestellt. Als Ursache für die Instabilität wird eine inhärente Behinderung des Transports kalter Flüssigkeit von der Umgebung zum Thermosyphon vermutet. Dieser Transport wird benötigt, um die heiße Flüssigkeit, welche vom Thermosyphon an die Umgebung abgegeben wird, zu ersetzen. Für den stationären Zustand des einseitig beheizten Thermosyphons werden Nusselt-Beziehungen ermittelt. Für den parallelwandigen Fall wird eine allgemeingültige Nusselt-Korrelation als vierte Wurzel des Produkts aus dem inversen Seitenverhältnis und der Rayleigh-Zahl ermittelt. Diese Korrelation führt zu Wärmeübergangskoeffizienten, die unabhängig vom Abstand zwischen den Wänden sind. Weiterhin ist die Korrelation, außer einer 7%igen Abweichung bei der multiplikativen Konstanten, identisch mit der Korrelation für natürliche Konvektion in einem einseitig beheizten, vertikalen, oben und unten offenen Kanal. Die Divergenz der Hauptwände des einseitig beheizten Thermosyphons hat nur einen geringen Einfluß auf die Nusselt-Zahlen, welche nicht mehr als 5% von denen für den parallelwandigen Fall abweichen.

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ОТКРЫТЫХ ТЕРМОСИФОНВ С ПАРАЛЛЕЛЬНЫМИ РАСХОДЯЩИМИСЯ СТЕНКАМИ

Аннотация—Проведены эксперименты по естественной конвекции в щелевых открнтых термосифонах с плоскими стенками для случаев параллельных и расходящихся основных стенок. В качестве жидкости, заполняющей термосифон и окружающей его, использовалась вода ($Pr \cong 5$). В случае нагрева обеих основных стенок, которые поддерживались при одинаковой постоянной температуре, превышающей темцературу окружающей среды, наблюдается нестабильный режим работы термосифона. Нестабильность объясняется, по-видимому, внутренними препятствиями потоку охлажденной жидкости иэ окружающей среды в термосифон, который должен компенсировать потери горячей жидкости, вытекающей иэ термосифона в окружающую среду. Данные для стационарного числа Нуссельта получены для случая с односторонним нагревом термосифона. Для термосифона с параллельными стенками выведено универсальное соотношение, содержащее проиэведение обратной величины отношения геометрических размеров на число Нуссельта в степени 0,25. Иэ этого соотношения получаются коэффициенты теплопереноса, которые не зависят от расстояния между стенками. Эа исключением 7%-ного отличия в коэффициенте, данное соотношение идентично зависимости для естественниой конвекции в вертикальном канале, нагретом се одной стороны, с открытыми обоими торцами. Влияние непараллельности основных стенок термосифона, нагретого с одной стороны, на число Нуссельта неэначительно, причем отклонения не превышают 5% по сравнению с термосифоном с параллельными стенками.