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EXPERIMENTS ON NATURAL CONVECTION HEAT TRANSFER ON THE FINS OF A FINNED HORIZONTAL TUBE

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Abstract—Experiments were performed to determine the natural convection heat flux distribution on the faces of isothermal circumferential fins affixed to a horizontal heated (or cooled) tube. The naphthalene sublimation technique was used in the execution of the experiments, and the mass-transfer results were transformed to heat-transfer results by using the analogy between the two processes. At each of several interfin spacings, the tube diameter was increased systematically while the fin dimensions were kept fixed, thereby resulting in a systematic decrease in the fin heat-transfer surface area. At small interfin spacings, the fin heat-transfer rate was unchanged as the transfer surface area was decreased by the increase of the tube diameter. For the larger spacings, there was a modest decrease of the fin heat-transfer rate, but much less than the reduction in surface area. These findings suggest that the highest heat-transfer coefficients occur adjacent to the periphery of the fin while the lowest coefficients are at the inner portion of the fin, adjacent to the tube. This outcome stands in contrast to the conventional model which assumes that the heat-transfer coefficient is uniform across the face of the fin.

NOMENCLATURE

A, plate mass-transfer surface area, Fig. 2 and equation (1);
A_0, full-face surface area of square plate, H^2;
b, interplate spacing, Fig. 1;
D, diameter of teflon disk;
$\mathscr{D}$, naphthalene–air diffusion coefficient;
g, acceleration of gravity;
H, vertical height of plate;
K, mass-transfer coefficient, equation (2);
M_a, molecular weight of air;
M_n, molecular weight of naphthalene;
$\dot{M}$, rate of mass transfer from fin surface area A;
$\dot{M}_0$, rate of mass transfer from fin surface area A_0;
Ra, Rayleigh number, equation (4);
Sc, Schmidt number;
Sh, Sherwood number, equation (2);
W, plate width, equal to H;
ρ, mixture density;
ρ_w, mixture density at plate surface;
ρ_∞, ambient mixture density;
ρ_{nw}, naphthalene vapor density at plate surface;
$\rho_{n\infty}$, ambient naphthalene vapor density;
μ, viscosity.

INTRODUCTION

THE USE of plate-type circumferential fins to enhance natural convection heat transfer from horizontal tubes is standard practice in many heating and cooling applications, most notably for space heating (e.g. baseboard heating). Typically, such heat-transfer configurations consist of a horizontal tube to which is affixed a succession of uniformly spaced fin plates. The fins are either square or circular metal plates positioned perpendicular to the axis of the tube, with a central hole whose diameter corresponds to that of the tube. Thus, the fins defined an array of parallel vertical channels. Each channel is open to the ambient at its lateral edges as well as at the top and bottom but is internally obstructed because of the presence of the tube.

In considering the pattern of fluid flow through such a channel, it is relevant to note that the buoyant forces which drive natural convection airflows are relatively weak, especially for small and moderate temperature differences such as occur in space heating applications. Such weakly driven flows tend to avoid paths where there is high frictional resistance and favor paths of lower resistance. For channels where the lateral edges are open, in addition to the open top and bottom edges, the flow has considerable freedom in finding the path of least resistance.

In light of the foregoing, it may be expected that the fluid flow would prefer paths adjacent to the periphery of the fins rather than paths which penetrate to the central portion of the channel (i.e. adjacent to the tube surface). These expectations should have increasing validity as the interfin spacing decreases, since there would be an increase in the resistance encountered by fluid attempting to flow into the inner reaches of the channel.

These considerations have direct relevance to the heat transfer characteristics of the fins. If more vigorous fluid flows exist adjacent to the periphery of the fins, then the heat-transfer coefficients should be higher in the peripheral regions than adjacent to the tube, where the velocities may be lower. This is at variance with the conventional model which assumes that the heat-transfer coefficient is uniform at all points of the fin surface. That model yields a fin heat flux distribution which has its highest value adjacent to the tube and its lowest value adjacent to the periphery. In view of the foregoing discussion, the actual heat transfer distribution may be just opposite to that of the model, that is, the highest heat fluxes may occur adjacent to the periphery and the lowest fluxes near the tube.

Local heat-transfer rates and transfer coefficients for the fins of fin-tube natural convection heat transfer devices have not, apparently, been previously investigated in the published literature. There are, on the other hand, several useful publications dealing with the overall performance of these devices (e.g. [1–4]). The absence of local information is reflective of the formidable difficulties that have to be surmounted in order to make local measurements in a fin array.

The research described here is, seemingly, the first investigation aimed at determining the nature of the local heat-transfer distribution due to natural convection on the fins of a finned horizontal tube. To facilitate the experiments, mass-transfer measurements were made rather than direct heat-transfer measurements, and the experimentally determined mass-transfer coefficients were converted to heat-transfer coefficients by applying the well-known analogy between the two processes. The mass-transfer experiments were performed by employing the naphthalene sublimation technique. The use of this technique completely eliminates the extraneous losses due to radiation and conduction which frequently play a major role in natural convection heat-transfer experiments. Generally, results of higher accuracy are obtainable from the naphthalene technique than from corresponding heat-transfer experiments.

For the experimental model, it was deemed sufficient to deal with only one channel of the array. The fin surfaces bounding the channel were modeled by square plates made of naphthalene. The sublimation of the naphthalene creates density differences which induce a natural convection flow in a manner completely analogous to the thermally induced density differences which drive the natural convection flow in the corresponding heat-transfer problem. The mass transfer boundary condition at the naphthalene surfaces corresponds, by analogy, to isothermal heat-transfer surfaces. Thus, the experimental results to be reported here pertain to heat-transfer fins having an efficiency η equal to unity. This characteristic obviates the need to make corrections for $\eta < 1$, which are normally required in heat-transfer experiments.

To model the tube on which the fins are affixed, a teflon disk was sandwiched between the naphthalene plates, the disk axis being horizontal and the plate surfaces vertical. Disks of different diameters and thicknesses were employed. The disk thickness served to fix the interplate spacing. With a disk in place, sublimation is fully suppressed on the circular area on which the disk contacts the plates with the result that mass transfer occurs only on those portions of the plates that are not covered. Thus, as the disk diameter is increased, the surface area for mass transfer is decreased.

The experiments were performed for five different interplate spacings characterized by ratios of spacing to plate height (plate height = plate width) ranging from 0.042 to 0.167. At each fixed spacing ratio, the disk diameter was systematically increased from zero (no disk) to a value slightly less than the side dimension of the square plates. Correspondingly, the mass-transfer surface area of the plates decreased from a value A_0 (equal to the face area of the plates) to $0.34A_0$.

A primary focus of the work is to determine how, for any given interplate spacing, the rate of mass-transfer changes as the transfer surface area decreases in response to systematic increases of the disk diameter. A marked decrease in the mass-transfer rate would indicate that the plate surface area covered by the disk had been, prior to covering, a region of substantial mass transfer. On the other hand, if the mass transfer were to change only slightly, it could be inferred that the covered area had, when uncovered, made only a small mass-transfer contribution.

The use of a teflon disk rather than a naphthalene disk to model the tube means that the modeled tube does not participate in the mass-transfer process, although it does participate as a barrier to fluid flow. Thus, the present experiment does not conform in all aspects to the actual fin-tube situation. It does, however, model the main features since the tube surface area is, in almost all the cases studied, a very small fraction of the fin area. The decision to use a non-participating tube was based on two considerations. First, it was regarded as a logical initial step in a long-range research plan whose overall goal is to identify the separate roles of the fins and the tube. Second, it led to experiments which are much more tractable than those involving a participating tube—a feature that is attractive in a first study of a complex problem.

THE EXPERIMENTS

The description of the experimental apparatus is facilitated by reference to Fig. 1, which shows a schematic diagram of the main features of the fin-tube model. As seen there, a pair of vertically oriented naphthalene plates are positioned parallel to each other by means of coaxial support rods. The rods are part of a precisely machined support system which includes a spring-loading mechanism for maintaining the interplate spacing, levels and adjustments for attaining the desired vertical orientation of the plate surfaces (as well as horizontal orientation of the upper and lower edges of the plates), and a heavy stand for firmly positioning the fin-tube assembly in the test environment (to be described shortly). Each support rod was fitted with a flange which mated with a corresponding flange affixed to the metal backing of the naphthalene plate.

Each test plate was a composite consisting of a layer of naphthalene backed by a stainless steel plate which had been painstakingly ground to a flatness of better than 0.00254 cm (0.001 in), with the final thickness being 0.1092 cm (0.0430 in). The backing plate served as one part of a multi-part mold in which the naphthalene plate was cast. A special feature of the mold was a precisely machined three-point-of-contact spacer arrangement which ensured the uniformity of the casting thickness (0.635 cm, 0.250 in) and the flatness of the surface which participated in the mass-transfer process. Smoothness of the participating surface was assured by using a hand-lapped stainless steel plate as its contacting mold part. Reagent grade naphthalene was used for the casting. For each data run, a new pair of naphthalene plates was prepared using fresh (i.e. previously unused) naphthalene.

The unmolding of the casting exposed one of the square faces of the naphthalene plate as well as its four edges. Since the experiments were concerned only with the mass transfer at the square face, all edges were covered with a pressure-sensitive tape which fully suppressed mass transfer. The precision of the taping operation was facilitated by a specially designed jig.

The final dimensions of the exposed surface of the naphthalene plate are 7.62 × 7.62 cm (3.00 × 3.00 in).

As noted earlier, the tubes of the fin-tube system were modeled by teflon disks. Teflon was selected because of its excellent machining characteristics and because, in contrast to a metal, it would not mar the naphthalene surface. Disk thicknesses of 0.318, 0.635, 0.953, 1.15 and 1.27 cm (0.125, 0.250, 0.375, 0.453 and 0.500 in) were selected. These thicknesses serve to define the interplate spacing b (Fig. 1). The corresponding ratios of interplate spacing to plate height H are 0.042, 0.083, 0.125, 0.151 and 0.167. For each thickness, disks having diameters of 3.81, 5.08, 6.35, and 6.99 cm (1.50, 2.00, 2.50, and 2.75 in) were fabricated. Therefore, all told, twenty different disks were employed. Each disk was drilled along its axis and fitted with a pin which mated with a centrally positioned drill hole in one of the naphthalene plates (see Fig. 1). This arrangement ensured concentric alignment of the disks and the plates.

In addition to the data runs with the disks in place, other runs were made without the disks but at the same interplate spacings b as those established by the disk thicknesses. These spacings were fixed by a special locking arrangement which maintained the separation distance between the plates once it had been set by the insertion of temporary spacers.

A controlled test environment was created in recognition of the facts that: (a) natural convection is highly sensitive to extraneous flows and disturbances in the environment and (b) naphthalene sublimation is quite responsive to temperature level since the vapor pressure of naphthalene increases by $\sim 10\%/^{\circ}C$. To attain constancy of temperature, the experiments were performed in a temperature-controlled laboratory room. However, owing to the forced-draft heating and cooling system, the air currents in the room were judged to be unacceptable for natural convection experiments. To surmount this problem, a 12 × 8 × 8 ft (length × width × height) enclosure was constructed in the laboratory—actually, a room within a room. The temperature-controlled air circulated about the outer walls of the enclosure but did not enter it. The walls and ceiling of the enclosure were made of polystyrene building insulation sheets, and batts of fiberglass were placed on the floor.

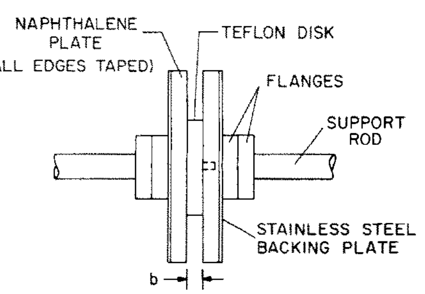

FIG. 1. Schematic of fin-tube model.

To suppress possible extraneous currents within the enclosure, a system of baffles and screens was erected in the neighborhood of the test plate. The baffles and screens were arranged to shield the apparatus from extraneous currents while not inhibiting the natural convection flow induced by naphthalene sublimation.

The test enclosure was kept sealed during the course of a data run, and thermocouples within the enclosure were read remotely by a datalogger situated outside the enclosure.

Prior to each data run, the test plates, wrapped in impermeable plastic, were placed in the enclosure for at least 8 h to attain thermal equilibrium. Then, immediately before the initiation of the run, the plates were weighed with an analytical balance having a smallest scale division of 10^{-4} g. After the run, the weighing was repeated. Subsequent to that and without any delay, the apparatus was set up and then immediately disassembled, and a third weighing was performed. The purpose of the latter procedure was to determine a correction for any mass transfer that may have taken place during the setup and disassembly periods that respectively precede and follow a data run.

Once all measurements for a given run had been made, the enclosure was purged of naphthalene vapor by a blower that discharged at the roof of the building.

Additional information about the apparatus and experimental procedure is available in [5].

RESULTS AND DISCUSSION

Before reporting on the main results of the research, mention will be made of relevant auxiliary experiments. To demonstrate the validity of the experimental technique and of the heat-mass transfer analogy, naphthalene sublimation experiments were performed with a single vertical plate in order to enable comparisons with natural convection heat-transfer results in the literature. A total of five data runs were made. In two of these, side baffles (thin cardboard strips) were fitted to the vertical edges of the plate, while in the other three, no baffles were employed. The measured

Sherwood number* results differed by about 10% from those given by the recent Churchill–Chu heat-transfer correlation [6]. Since the data on which the correlation is based show scatter and deviations from the correlating line that exceeds 10%, it can be concluded that the present vertical plate results are in agreement with the literature.

As a second verification of the experimental technique, data runs of different duration were made for otherwise identical operating conditions. The motivation for these runs was to demonstrate that any accumulation of naphthalene vapor in the test enclosure during a data run was too small to affect the rate of mass transfer. From a comparison of the measured mass-transfer rates for the different duration runs, agreement within 2% was found to prevail. This finding confirms that vapor accumulation did not materially affect the results.

Attention will now be turned to the mass-transfer results for the fin-tube system. In this connection, it may be noted that the sequence in which the experiments were performed had been selected to facilitate the presentation of results. For a specific interfin spacing characterized by the ratio b/H, successive experiments were carried out in which the diameter of the teflon disk (i.e. the modeled tube) was increased systematically, starting with the case in which there was no disk. Since the dimensions H and W $(=H)$ of the plate were fixed, the increasing disk diameter brought about a corresponding decrease in the mass-transfer surface area on the plate (see Fig. 2). The operating conditions for all the runs in such a sequence were maintained identical, within the limits of practicality. This was done to ensure that the driving force for mass transfer would be the same for all runs. Corrections to the data to account for slight departures from a common operating condition were, at most, 2% [5].

At a given b/H, let the measured mass-transfer rate for the no-disk case be denoted by $\dot{M}_0$, and let A_0 denote the surface area of the full face of the naphthalene plate, i.e. $A_0 = HW = H^2$ (see Fig. 2). With a disk in place, the mass transfer rate is $\dot{M}$ and the participating naphthalene surface area is A, where

$$A = H^2 - (\pi D^2/4) \qquad (1)$$

where D is the disk diameter. Note that both $\dot{M}$ and $\dot{M}_0$ are rates of mass transfer, not rates of mass transfer per unit surface area. It may also be noted that the measured mass-transfer rates at the two disks were essentially identical, so that $\dot{M}$ and $\dot{M}_0$ represent their average values.

From the aforementioned experiments for a fixed spacing ratio b/H, the $\dot{M}/\dot{M}_0$ ratios were evaluated and are plotted as a function of A/A_0 in Figs. 3 and 4. Figure 3 shows results for b/H values of 0.042, 0.083, and 0.125, while Fig. 4 is for b/H of 0.151 and 0.167.

* The Sherwood number is the mass transfer counterpart of the Nusselt number.

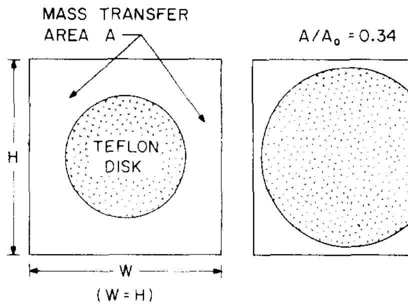

FIG. 2. Mass-transfer surface area.

Inspection of these figures indicates that for the smaller spacing ratios, the mass-transfer rate is altogether unaffected as more and more mass-transfer surface in the central region of the plate is deactivated by the blocking action of the teflon disk. Thus, for example, an area $A = 0.34A_0$ distributed adjacent to the periphery of the plate (see Fig. 2, right-hand diagram) transfers mass at the very same rate as does the entire area A_0 when there is no blockage. This finding has to be regarded as remarkable.

As the interplate spacing increases (Fig. 4), there is a tendency toward a moderate reduction in the mass-transfer rate as the transfer surface area decreases, but this decrease is by no means proportional to the reduction in area. To illustrate this, a dashed line representing the relation $\dot{M}/\dot{M}_0 = A/A_0$ is plotted in the upper graph of Fig. 4, where it is seen to fall well below the data. Since $\dot{M} \sim A$ corresponds to a mass-transfer coefficient which is uniform at all points on the plate surface, it is evident that the uniform-coefficient model is negated by the present data.

The aforementioned results strongly suggest that the active zones of natural convection heat (mass) transfer in a fin-tube system are confined to the peripheral regions of the fin surface, and that the inner portions of the fin do not contribute significantly to the heat transfer. To make this behavior plausible, a small-spacing flow pattern may be envisioned whereby the natural convection flow is confined to the peripheral regions of the fins because it is there that the paths of lowest resistance are to be found. There is little or no tendency for the flow to penetrate into the more interior regions of the interfin space and, as a consequence, the flow pattern is essentially the same whether or not a tube is present.

At the larger spacings investigated here, it appears that there is some flow penetration into the inner reaches of the interfin space, but the near-periphery flow continues to dominate. Most of the mass transfer continues to occur adjacent to the periphery.

As was noted in the Introduction, there does not appear to be prior published work directed at identifying the distribution of natural convection heat transfer on finned surfaces. Generally, only average heat-transfer coefficients have been measured for specific geometrical configurations. Nevertheless, the literature was carefully examined with the view of identify-

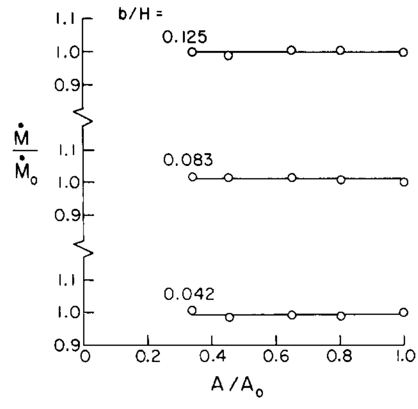

FIG. 3. Comparison of the mass-transfer rate for plate-surface area A with that for the entire face of the plate, area A_0. b/H = 0.042, 0.083, and 0.125.

ing trends which might lend qualitative support to the present findings. In [3], heat-transfer coefficients were measured for three fin–tube systems with respective ratios of fin diameter to tube diameter of 1.97, 2.97 and 5.14. The coefficients included both fin and tube heat-transfer contributions. In appraising these results, attention was focused on copper fins, since they are presumably isothermal and thereby correspond to the boundary condition of the present study. By study of the results of [3], it is clear that the heat-transfer rate did not increase in proportion to the surface area, as would be the case had the heat-transfer coefficient been uniform. Rather, it appears that the heat-transfer rate increased more or less in proportion to the outer perimeter of the fin, which supports the present view that the maximum transfer occurs adjacent to the periphery.

The results given in Figs. 3 and 4 are in ratio form, referenced to the no-disk case. Therefore, to provide more complete information, quantitative no-disk results will now be presented. To this end, the mass-transfer coefficient K and the Sherwood number Sh are introduced as follows

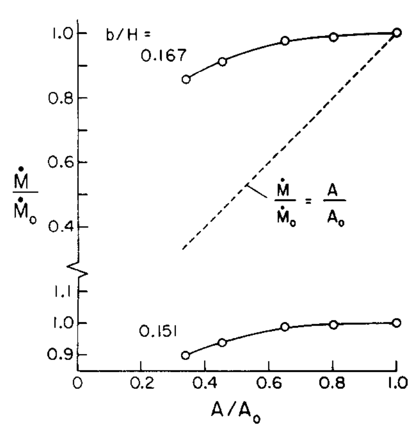

FIG. 4. Comparison of the mass-transfer rate for plate-surface area A with that for the entire face of the plate, area A_0. b/H = 0.151 and 0.167.

$$K = (\dot{M}_0/A_0)/(\rho_{nw} - \rho_{n\infty}), \quad Sh = Kb/\mathscr{D}. \tag{2}$$

In these equations, only the quantities ρ_{nw}, $\rho_{n\infty}$, and $\mathscr{D}$ have not yet been defined. Both ρ_{nw} and $\rho_{n\infty}$ are naphthalene vapor densities, respectively at the plate surface and in the ambient. The former was evaluated from the Sogin vapor pressure–temperature relation [7] in conjunction with the perfect gas law, while the latter is essentially zero for the present experiments. The quantity $\mathscr{D}$ is the naphthalene–air binary diffusion coefficient which was obtained from the relation $\mathscr{D} = \nu/Sc$, where ν is the kinematic viscosity of air and Sc, the Schmidt number, is 2.5 [7].

Following Elenbaas [8], the Sherwood number results will be reported as a function of the group

$$(b/H)Ra \tag{3}$$

where

$$Ra = [g\rho|\rho_w - \rho_\infty|b^3/\mu^2]Sc \tag{4}$$

in which ρ_w and ρ_∞ denote the density of the naphthalene–air mixture at the plate surface and in the ambient, while ρ denotes a mean mixture density. By employing Dalton's Law of partial pressures, it follows that

$$\rho_w - \rho_\infty = \rho_{nw}(M_n - M_a)/M_n \tag{5}$$

where M_n and M_a are the molecular weights of naphthalene and air. The introduction of (5) into (4) yields a Rayleigh number definition which contains quantities that are readily evaluated for the conditions of the experiments.

A listing of the Sherwood number results for the no-disk case is presented in Table 1 as a function of b/H and $(b/H)Ra$. It is seen from the table that both $(b/H)Ra$ and Sh increase as b/H increases.

CONCLUDING REMARKS

The present experiments have provided results which strongly suggest that the largest contributions to the natural convection heat transfer on the fins of a horizontal finned tube occur at the peripheral regions of the fin. These results negate the conventional heat-transfer model whereby it is assumed that the heat-transfer coefficient is uniform at all points on the surface of a fin. Such a model predicts that the lowest heat fluxes occur in the peripheral regions of the fin, while the highest heat fluxes occur at the inner portion of the fin, adjacent to the tube.

The work reported here appears to be the first experimental study concerned with local natural con-

Table 1. Results for $A/A_0 = 1$ (no-disk)

b/H	$(b/H)Ra$	Sh
0.042	0.041	0.076
0.083	0.648	0.188
0.125	3.28	0.349
0.151	7.04	0.498
0.167	10.3	0.615

vection heat transfer on a fin. To supplement and extend the present work, additional experiments can be suggested. In one set of experiments, a tube of fixed diameter should be employed in conjunction with square fins having a range of side dimensions. In another set, the teflon disks presently used to model the tube should be replaced by naphthalene disks which participate in the mass transfer process.

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ETUDE EXPERIMENTALE DE LA CONVECTION THERMIQUE NATURELLE SUR LES AILETTES D'UN TUBE HORIZONTAL

Résumé—Des expériences sont conduites pour déterminer la distribution du flux thermique par convection naturelle sur les faces des ailettes circonférentielles isothermes fixées à un tube chauffé (ou refroidi) horizontal. On utilise la technique de sublimation du naphtalène et les résultats du transfert massique sont transformés en transfert thermique en exploitant l'analogie entre les deux mécanismes. Pour chaque espacement des ailettes, le diamètre du tube est augmenté systématiquement tandis que les dimensions de l'ailette sont gardées constantes, si bien qu'il en résulte une décroissance systématique de la surface d'échange de l'ailette. Pour les petits espacements, le taux de transfert de l'ailette n'est pas modifié quand la surface d'échange décroit du fait de l'accroissement du diamètre du tube. Pour les grands espacements, il y a une faible diminution du transfert, mais beaucoup plus faible que la réduction de la surface. Ceci suggère que le coefficient de transfert le plus élevé est relatif à la périphérie de l'ailette tandis que le coefficient le plus faible correspond à la partie interne de l'ailette et adjacente au tube. Ceci est en contradiction avec le modèle conventionnel qui suppose que le coefficient de convection est uniforme sur toute la surface de l'ailette.

EXPERIMENTE ZUM WÄRMEÜBERGANG BEI FREIER KONVEKTION AN DEN RIPPEN EINES WAAGERECHTEN RIPPENROHRES

Zusammenfassung — Es wurden Versuche zur Bestimmung der Wärmestromverteilung bei freier Konvektion an den Flanken von isothermen Kreisrippen auf einem waagerechten beheizten (oder gekühlten) Rohr durchgeführt. Für die Versuche wurde die Naphthalin-Sublimationstechnik benutzt, und die Ergebnisse des Stoffübergangs wurden in Wärmeübergangswerte unter Benutzung der Analogie beider Prozesse umgerechnet. Bei jeder von mehreren Rippenteilungen wurde der Rohrdurchmesser systematisch vergrößert, während die Rippenabmessungen konstant gehalten wurden, so daß die wärmeübertragende Fläche der Rippen hierbei systematisch abnahm. Bei kleinen lichten Rippenteilungen blieb der durch die Rippe übertragene Wärmestrom unverändert, obgleich die Übertragungsfläche durch die Vergrößerung des Rohrdurchmessers kleiner wurde. Für größere Teilungen ergab sich eine mäßige Abnahme des an der Rippe übertragenen Wärmestroms, die jedoch wesentlich kleiner war als die Verringerung der Oberfläche. Diese Ergebnisse wiesen darauf hin, daß die größten Wärmeübergangskoeffizienten am äußeren Rippenrand und die kleinsten am inneren Teil der Rippenfläche, dem Rippenfuß, auftreten. Dieses Resultat steht im Gegensatz zum konventionellen Modell, bei dem angenommen wird, daß der Wärmeübergangskoeffizient über die ganze Rippenflanke konstant ist.

ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ СВОБОДНОЙ КОНВЕКЦИИ ТЕПЛА ОТ ОРЕБРЕННОЙ ГОРИЗОНТАЛЬНОЙ ТРУБЫ

Аннотация — Для свободноконвективного теплового потока от изотермических кольцевых ребер на горизонтальной нагреваемой (или охлаждаемой) трубе выполнены опыты с использованием метода сублимации нафталина с последующим пересчетом данных на основании аналогии между процессами массо- и теплопереноса. Через каждые несколько промежутков между ребрами диаметр трубы возрастал, тогда как размер ребер оставался постоянным, что уменьшало площадь теплообменной поверхности ребер. При небольших расстояниях между ребрами интенсивность теплопереноса по мере уменьшения площади поверхности теплообмена оставалась постоянной за счет увеличения диаметра трубы. При больших промежутках наблюдалось некоторое снижение интенсивности теплообмена, но гораздо менее значительное, чем уменьшение площади поверхности. Полученные результаты свидетельствуют о том, что максимальные значения коэффициента теплообмена имеют место у края ребра, а минимальные — внутри ребра у его основания. Этот вывод противоречит результатам, полученным на модели, когда коэффициент теплообмена принимается постоянным поперек ребра.