Heat Transfer and Pressure Drop for Nitrogen Flowing in Tubes Containing Twisted Tapes

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Heat transfer and pressure-drop studies were made on a series of electrically heated tubes containing full-diameter, full-length twisted tapes. The A nickel tubes had a nominal inside diameter of 0.4 in. (10.1 mm.) and a heated length of 12 in. (30.5 mm.). Nitrogen at 200 lb./sq.in. gauge (14.6 atm.) was the working fluid, and the tests covered the Reynolds number range from 20,000 to 200,000. Twist ratios, defined as the ratio of the tube length per 180 deg. twist to the tube diameter, were varied from 2.5 to 14. The effect of tube wall-to-gas temperature ratio was studied for values of up to 1.8 and the heat flux was varied from 10^4 to 10^5 B.t.u./hr. · sq.ft. (3×10^4 to 3×10^5 w./sq.m.).

The results were compared with the results of previous heat transfer and pressure-drop studies and were generally found to be in good agreement. An empirical correlation was developed for the heat transfer results that accounts for the effects of twist ratio, wall-to-gas temperature ratio, and tube length and includes most of the previous single-phase results.

As the operating temperatures of nuclear reactors and rocket nozzles increase and their heat fluxes go higher, the problem of getting the most heat out of the system with the least amount of pressure drop becomes increasingly important. The enhancement of heat transfer from tubes, rods, plates, etc., by means of various promoters has been studied for a number of years at Oak Ridge National Laboratory in connection with its various reactor development programs. The study discussed here is related to the gas-cooled reactor program; its purpose was to investigate the use of full-length, full-diameter twisted tapes inside tubes as a means of increasing heat transfer in gaseous systems.

Although twisted tapes have been used for over forty years, the nature of their effect on heat transfer and pressure drop is still not completely understood. They have been used in both single and two-phase systems and for heating and cooling. The increase in heat transfer they produce apparently is due to a combination of effects, the most significant of which are an increased path length and an increase in the amount of mixing due to centrifugal forces and/or turbulence level.

In order to establish the limits for this investigation, it was assumed that future high-performance gas-cooled reactors would be likely to have heat fluxes of the order of 10^6 B.t.u./hr.·sq.ft. (3×10^6 w./sq.m.), would operate at high pressures (2,000 lb./sq.in.abs., 136 atm.), and would use helium as the coolant. It was also reasoned that the flow would be inside tubular passages, since this eliminates the problems of vibration, bowing, etc., associated with fuel pins under these conditions. An initial analysis showed that these conditions could be simulated by using 200 lb./sq.in.gauge (14.6 atm.) nitrogen at a heat flux of 10^5 B.t.u./hr.·sq.ft. (3×10^5 w./sq.m.). In addition, an upper value for the Reynolds number of 200,000 was selected so that the maximum Mach number in a helium system would be < 0.1. With these criteria as a basis, a nitrogen system was designed and operated in which four twisted-tape geometries and an empty tube (which served as a control) were studied.

PREVIOUS WORK

One of the earliest experimental studies of twisted tapes was made by Colburn and King (*1*), who investigated heat transfer from air to a cooled surface. They reported increases of ~ 50% in the heat transfer coefficient at constant mass velocity but found that this was accompanied by a fourfold increase in pressure drop. Apparently they did not detect much effect due to twist ratio (Y),* since they made no mention of it. Most of their results were for Y values near 3, and for Reynolds numbers < 50,000.

Later Koch (*2*) extended the Reynolds number range to 80,000 in a study of heat transfer to air. He examined tubes with Y values of 2½, 4-1/3, and 11. Again he found a factor of 4 in the pressure-drop increase but was able to get a ratio of the heat transfer coefficient with a tape to that without a tape of 2.5. This improvement in heat transfer was greatest for the smallest twist ratio and occurred at low Reynolds numbers (4,000); however, the performance diminished as flow increased.

Kreith and Margolis (*3*) in 1959 made measurements with both air and water and found remarkable increases in heat transfer with twisted tapes for water as opposed to air in both heating and cooling. Again, improvement in

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* The twist ratio (Y) is defined as the ratio of tube length per 180 deg. tape twist to the tube diameter.

heat transfer was accompanied by large increases in pressure drop.

In a survey made in 1962, Gambill and Bundy (*4*) collated available data and illustrated the wide diversity in experimental measurements. They suggested that microscopic surface roughness has a marked effect in twisted-tape systems and may explain some of the scatter reported, especially with regard to the friction data.

The effect of tape length was studied by Seymour (*5*) over a wide range of twist ratios in an atmospheric air system. He found in one case that a short section of twisted tape (20% of the tube length) produced 87% of the increased heat transfer attributable to a full-length tape. Smithberg and Landis (*6*) made measurements in air and water for Reynolds numbers up to ~60,000 and were able to develop good correlations for their data over the range of parameters studied. Two recent studies, one using water by Lopina and Bergles (*7*) and the other using air by Thorsen and Landis (*8*), indicate that the wall to fluid temperature difference is a significant parameter in swirl flow heat transfer. Examination of these studies makes one conclude that the heat-transfer and pressure-drop characteristics of tubes containing twisted tapes are still fertile fields for research. Little mention was made in these works of the effects of the heat flux, except in connection with boiling (*9, 10*), or wall-to-bulk temperature ratio; and it appears that these parameters were not studied directly until quite recently. Since they are often significant in high thermal performance systems, it was felt that their effects should be evaluated for moderate values initially and eventually for heat fluxes up to 10^6 B.t.u./hr.·sq.ft. (3×10^6 w./sq.m.).

EXPERIMENTAL APPARATUS

The experimental setup is shown schematically in Figure 1. The nitrogen was supplied from a tank truck containing 2,000 lb.$_m$ (900 kg.) at 1,800 lb./sq.in.gauge (123 atm.) and was certified to be 99.997% nitrogen, with a maximum of 5 ppm. water vapor. The pressure in the inlet to the test section was maintained at ~200 lb./sq.in.gauge (14.6 atm.) by means of a pressure regulator. Since the gas supply was located outside of the building, the inlet gas temperature varied from 60 to 90°F. (16 to 32°C.), depending on the season. The flow rate was measured with three rotameters in parallel. These were checked against a positive-displacement metering system and found to be accurate and precise to better than ±1%.

The inlet pressure to the system was measured on a 0 to 500 lb./sq.in.gauge Heise gauge. The pressure drop across the test section was measured on either a water-filled U-tube manometer or a well type of manometer filled with 2.95 specific gravity Meriam oil, depending on the magnitude of the drop. Manometer fluid densities were corrected for temperature variations.

System temperatures were obtained from thermocouples whose outputs were displayed on a 12-point Brush recorder. The inlet gas temperature was obtained by means of a sheathed Chromel-Alumel thermocouple located in the inlet line. The outlet temperature was measured downstream of a mixing section to insure good mixed-mean temperatures. Outside wall temperatures were determined by nine couples spot welded at

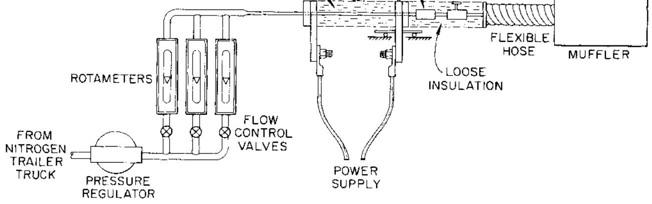

Fig. 1. Schematic diagram of experimental apparatus.

equal intervals along the outside of the test section. Power was supplied from a 440v. 60 cycle/sec. a.c. line stepped down through a saturable reactor and a multitap transformer to the appropriate voltage level. The voltage across the test section was measured on a Polyranger voltmeter accurate to ¾%, and the current was read on a precision Weston ammeter (¾%) connected to a 1,000:5 current transformer.

The test sections were made from commercial grade ½ in. O.D. (12.7 mm.) A nickel tubing that had been annealed and pickled. They were fabricated by placing a piece of twisted Inconel shimstock 0.015 in. (0.381 mm.) thick and 0.400 in. (10.15 mm.) wide inside a section of tube and then swaging the tube down to an internal diameter of 0.395 to 0.398 in. (10.05 to 10.1 mm.). The nominal wall thickness of the tubing was 0.05 in. (1.27 mm.); measurements of several samples of the tubing after swaging showed the wall thickness was approximately 0.045 in. (1.14 mm.) and varied less than 2% over the length of the test sections. The surface roughness on the inside of typical tubes was of the order of 100 micro in. arithmetic average (2.4 μ) and that of the tapes was 12 micro in. (0.3 μ). When the swaging was completed, the test sections were cut to the desired lengths and x-rayed so that accurate values of the as-formed twist length-to-diameter ratio (Y) could be measured. After silver soldering the test sections into copper electrodes, pressure tap holes were drilled ~¼ in. (6.355 mm.) from each end, the thermocouples and voltage taps were spot welded on the outside surface, and the assembly was mounted in the flow system and insulated.

RESULTS AND DISCUSSION

Analysis of the Heat Transfer Data

Methods of reducing and correlating high heat flux-heat transfer data have been widely investigated over the years, and it now appears that there are general techniques that can be used with some confidence. However, it must be recalled that these are based on semi-empirical methods and are thus not completely general.

Humble, Lowdermilk, and Desmon (*11*) looked into this problem for air flowing in smooth tubes in 1951 and found that they could correlate their data by evaluating the physical properties at the film temperatures and by taking the test section length-to-diameter ratio into account. Taylor and Kirchgessner (*12*) followed up on this and proposed that the film temperature be used to characterize the system and that the length-to-diameter ratio be accounted for by the term, $1 + (L/D)^{-0.7}$. In 1962, Ward Smith (*13*) recommended that the bulk temperature be used for all heat transfer correlations; further, he suggested that the Stanton number be adopted as the standard heat transfer parameter and that the definitions of density, area, and velocity be consistent with the continuity equation. More recently, the use of the wall temperature-to-inlet temperature ratio was proposed by Dalle-Donne and Bowditch (*14*).

In studies by Lel'chuk and Elfimov (*15*), Taylor (*16, 17*), Petukhov, et al. (*18*), Petukhov (*19*), and Ter-Oganes'yants and Shorin (*20*), the effects of temperature and/or length-to-diameter ratio were further investigated and various schemes for correlating the results were presented. A recent paper by Perkins and Worsoe-Schmidt (*21*) contains a summary of many of the previous studies and shows some of the similarities and some of the differences in their results. The primary conclusion one reaches is that for smooth tubes there are several acceptable ways in which to handle the temperature and length effects but that the specific values of exponents on some of the terms are still an open question.

In this experiment the problem was, of course, further complicated by the presence of the twisted tapes. A number of possible correlations were tried in the process of finding a satisfactory method for correlating the data. These calculations were facilitated by the use of an IBM 7090 digital computer.

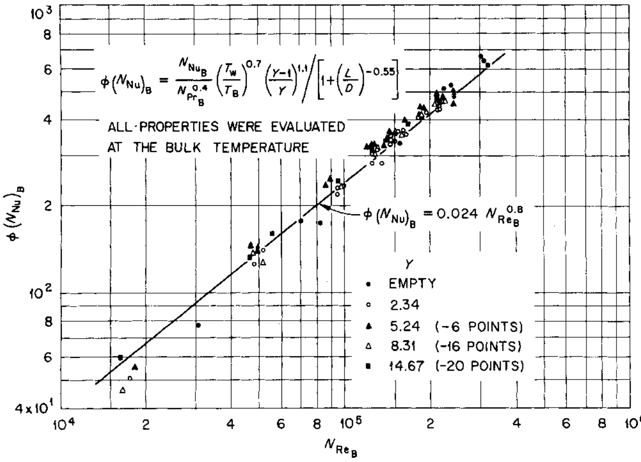

Fig. 2. Heat transfer results.

The results of these calculations gave the correlation shown in Figure 2. The best correlation was obtained with all physical properties of the nitrogen evaluated at the bulk gas temperature.† In this analysis the average Nusselt number for the entire heat section was defined as:

$$N_{Nu,B} = \frac{hD_e}{k_B} \tag{1}$$

where the average heat transfer coefficient, h, was calculated from the heat picked up by the gas stream, the area of the tube (the area of the strip was not considered as part of the heat transfer surface since the heat generation in it was only of the order of 3 to 5% of the total), and the difference between the mean wall temperature and the bulk gas temperature. Since the equivalent diameter, D_e, arises naturally in the definition of the friction factor, it was used as a matter of convenience and to avoid confusion, as the characteristic dimension in defining the Nusselt and Reynolds numbers as well. It was defined as $\pi D/(\pi + 2)$ for the tubes containing tapes; this is the so-called "zero-tape-thickness" definition. A heat balance (electrical power input minus losses divided by heat content increase of the gas stream) was made for each run; only those runs in which the balance was within ±5% were considered satisfactory. Values of the mean inside wall temperatures were computed by numerically integrating the inside surface temperatures that had been calculated from the readings of the nine outside wall thermocouples. The Reynolds number was similarly defined as:

$$N_{Re,B} = \frac{D_e G}{\mu_B} \tag{2}$$

In these experiments the inside wall heat flux was varied from 10^4 to 10^5 B.t.u./hr.·sq.ft. (3×10^4 to 3×10^5 w./sq.m.); this gave wall-to-bulk temperature ratios up to 1.8. The maximum Mach number was of the order of 0.2, so it was not necessary to take compressibility effects into account in the data reduction.

The data of Figure 2 are correlated by the expression

$$N_{Nu,B} = 0.024\ N_{Re,B}^{0.8}\ N_{Pr,B}^{0.4} \left(\frac{T_w}{T_B}\right)^{-0.7} \left[1 + \left(\frac{L}{D}\right)^{-0.55}\right] \left(\frac{Y}{Y-1}\right)^{1.1} \tag{3}$$

† The physical properties were all taken from National Bureau of Standards 564 (22). Special subroutines were written for the data reduction program to use the National Bureau of Standards data and perform the required interpolation in the tables.

This equation predicts the average (or bulk) Nusselt number for a tube of length L and diameter D containing a tape with a twist ratio Y, when the physical properties in the Reynolds and Prandtl numbers are evaluated at the bulk gas temperature. It also takes into account the average wall temperature for the entire channel length and the bulk gas temperature. The correlating equation is only applicable to the calculation of average Nusselt numbers, not local values. The equation reduces to the typical Dittus-Boelter form for empty tubes or tubes containing flat tapes (where Y is essentially infinite) that have very large L/D ratios.

In analyzing the data, the exponent on the Reynolds number term was taken as 0.8 and the exponent on the Prandtl number as 0.4. Several combinations of inlet, wall, bulk, and film temperature ratios were examined in order to reduce the data to a single curve, and it was eventually found that the best overall results were obtained when the wall-to-bulk temperature ratio to the 0.7 power was used. Since the test sections were short (approximately 30 diam.), the term containing the heated length to tube diameter ratio had to be included. In seeking a term to account for the twist ratio, Y, that was reasonably simple and would approach unity as Y became infinite, linear, exponential, and polynomial factors were considered. Ultimately, the expression $[Y/(Y-1)]^{1.1}$ proved to be the most satisfactory.

Examination of Equation (3) shows that, in essence, the effect of the twisted tape is contained in the term $[Y/(Y-1)]^{1.1}$. Most previous investigations were conducted in such a manner that the empty tubes and/or tubes containing straight tapes were of the same diameter and length as those with the twisted tapes. Also, most of the Nusselt number vs. Reynolds number data in these investigations fall on lines with a slope of ~0.8. Thus, the ratio of the Nusselt number for a tube with a twisted tape to that without can readily be obtained from them. These ratios (*1*, *2*, *5* to *8*) are compared with the results of this study in Figure 3. In a few cases the Reynolds number and Nusselt number were defined in terms of the tube diameter instead of the equivalent diameter, and this was taken into account in producing this plot. As can be seen, the agreement is good especially for twist ratios greater than 2.

When one tries to fabricate a tape with a twist ratio less than approximately 2, the tape has a tendency to buckle and undergoes a transition from a smooth, even, helical form to a more uneven and nonuniform pattern; the exact details of the pattern appears to depend on the relative width and thickness of the tape being twisted. This may or may not be the reason that the data of Figure 3 has more scatter for Y less than 2; however, since this phenomenon does exist, the correlation should not be used in cases where Y is less than 2. This is not a serious restriction since the pressure drop associated with such a tight twist is usually too large for practical systems.

To summarize, the correlation, given by Equation (3), should predict the average Nusselt number of the single-

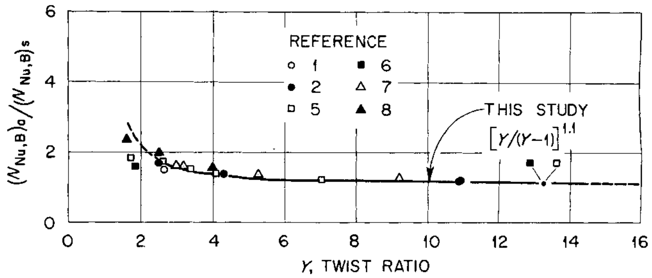

Fig. 3. Effect of twist ratio on heat transfer.

phase flow of gases and liquids for wall-to-bulk temperature ratios up to 2.5 and for twist ratios greater than 2. Such a simple relationship can be of considerable value in estimating the amount of heat transfer increase that can be expected from any given twisted tape in a given tube. The wall-to-bulk temperature ratio (T_w/T_B) was found to give good results and has much to recommend it for the designer's point of view, since these temperatures are common parameters in design work.

Analysis of the Friction Factor Data

The results of that portion of the study dealing with the pressure-drop characteristics of tubes containing twisted tapes are shown in Figure 4. The friction factor was defined by the equation

$$f = f_w \left(\frac{T_w}{T_B} \right)^{-0.6} \tag{4}$$

where f_w is calculated from

$$\Delta P = f_w \frac{L}{D_e} \frac{G^2}{2\ g_c\ \rho_w} \tag{5}$$

and is shown as a function of the wall Reynolds number as given by

$$N_{Re,w} = \frac{D_e G}{\mu_w} \tag{6}$$

Once again the selection of these parameters was based on the results of previous studies (*9* to *15*) and a good bit of trial and error. The inclusion of the wall-to-bulk temperature ratio term reduced the data of the heated and unheated runs to a common curve.

An examination of Figure 4 shows that the data for $Y = 14$ seem to be anomalous; that is, the friction factor for this tape twist ratio is higher than for $Y = 8$ and is about equal to the values for $Y = 5$. To insure that this was not due to an experimental error, the pressure taps were redrilled, cleaned, and pressure checked. The experiment was rerun and data identical to the original set were obtained. As a further check, the tubes for all Y values were tested in another setup in which the pressure taps were located in specially modified Swagelok fittings. Again, the same results were obtained. Thus, it appears that this effect is real; it may be due to a secondary flow that exists for this geometry in this range of velocities. While there is little practical interest in this twist ratio, since it does not increase the heat transfer significantly, it would be of interest to examine this region using flow visualization techniques to find out just what is causing this effect.

A comparison of the increase in friction factor as a function of twist ratio is presented in Figure 5, along with heat transfer results. Since the friction factor vs. Reynolds number curves for tubes containing tapes are not parallel to the empty-tube curves, one actually gets a family of curves that depend on the Reynolds number. The points shown are for a Reynolds number of 10^5; at higher values of N_{Re}, the ratio increases, and vice versa. As was expected, the relative increase in pressure drop was greater than the corresponding increase in heat transfer.

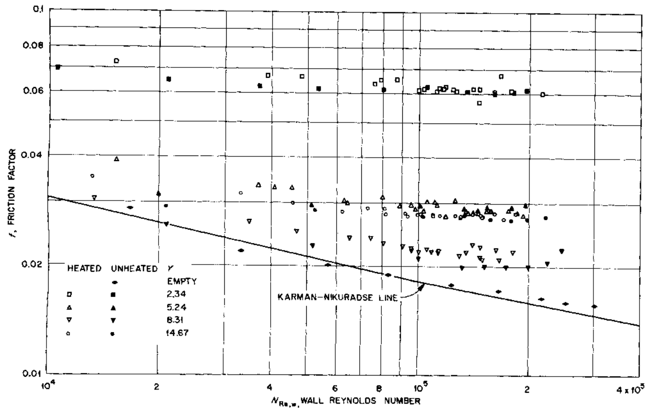

Fig. 4. Pressure-drop results.

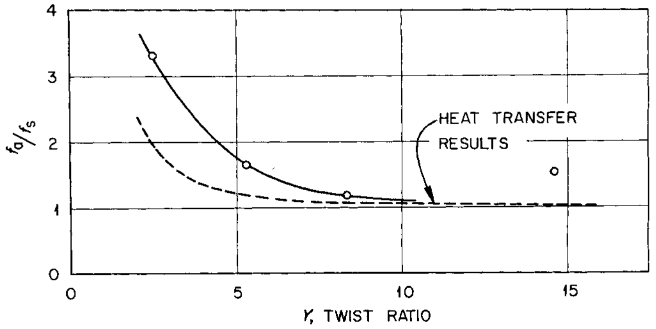

Fig. 5. Effect of twist ratio on friction factor at a Reynolds number of 50,000.

CONCLUSIONS

The heat transfer characteristics of tubes containing twisted tapes were satisfactorily correlated by Equation (3) using bulk fluid properties for Reynolds numbers from 2×10^4 to 2×10^5, heat fluxes up to 10^5 B.t.u./hr·sq.ft. (3×10^5 w./sq.m.), wall-to-bulk temperature ratios up to 1.8, and twist ratios between 2.5 and 14.

The inclusion of the wall-to-bulk temperature ratio allows the data for both heated and unheated tubes with tapes of any given twist ratio to be plotted on a single friction factor vs. Reynolds number curve. However, friction factors for tubes containing twisted tapes remain puzzling. The studies summarized by Gambill (*4*) and the anomalous result of this study for $Y = 14$ indicate the need for more definitive experiments.

ACKNOWLEDGMENT

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NOTATION

A = area
D = diameter
f = Blasius friction factor (= 4 × Fanning friction factor)
G = mass velocity
g_c = gravitational constant
h = heat transfer coefficient
k = thermal conductivity
L = length
P = absolute pressure
T = absolute temperature
Y = twist ratio (number of tube diameters per 180 deg. twist)
μ = viscosity
ρ = density
ϕ = function of Nusselt number

Subscripts

a = augmented
B = bulk
e = equivalent
s = smooth
w = wall

Dimensionless Groups

N_{Nu} = Nusselt number

N_{Pr} = Prandtl number
N_{Re} = Reynolds number

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A Study of the Effects of Internal Rib and Channel Geometry in Rectangular Channels

Part I. Pressure Drop

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Plexiglas models of rectangular channels were fabricated with various rib shapes to determine the effects of rib and channel geometry on the pressure drop. Pressure drops in twenty-four individual variations of channel geometry were investigated using plexiglas models.

From this investigation, an empirical correlation for the pressure drop across the ribbed section of the channel was determined as a function of the linear fluid velocity and the geometric characteristics of the channels. This empirical correlation involves functions of seven geometric parameters of rib pattern and channel geometry.

Ribbed section pressure drops for commercial plate heat exchanger channels were predicted using the geometric characteristics of the commercial plates with the empirical correlations developed from the plexiglas channel studies. The total pressure drop for commercial channels was predicted by adding an average entrance and exit pressure drop to the predicted ribbed section pressure drop.

The correlations developed in this work allow one to determine the pressure drop in a ribbed rectangular channel from the geometric characteristics of the ribs and the channel in question.

Rectangular channels with internal geometries for turbulence promotion have been used for many years in the form of plate and frame heat transfer equipment. Little information has been presented in the literature regarding pressure drop in such channels. The effects of these internal geometries which give ribbed channels quite unique heat transfer characteristics were experimentally investigated. This paper presents the results of these studies in the form of empirical correlations of pressure drop as functions of the rib and channel geometry.

The literature on plate heat exchangers is quite extensive but mainly confined to reporting the technical aspects of particular commercial units. General descriptions of the history and use of plate heat exchangers are presented by Troupe, et al. (*1*) and Seligman and Dummett (*2*). For pressure drop in plate heat exchangers, Baranovskii (*3*) presents a comprehensive work based on the results of many investigations. He presents correlations in the form of $N_{Eu} = A \cdot N_{Re}^{-N}$ for many types of plate heat exchanger channels. This type of correlation, however, is useful only for the particular plate channel for which it was obtained. Watson, et al. (*4*) present similar correlations obtained from experimental work with channels composed of a flat plastic plate and a ribbed metal plate.