

CONVERGING-DIVERGING DUCT HEAT TRANSFER
AND HYDRODYNAMIC RESISTANCE FOR TURBULENT
WATER FLOW

انتقال الحرارة والمقاومة الهيدروليكية في حالة
السيان المضطرب للماء في قنوات لامة مفرقة

By

AWAD, H. M., SHALABY, H. A., and EL-SHAZLY, S. A.

Mechanical Engineering Department, Faculty
of Engineering, El-Mansoura University,
El-Mansoura, Egypt.

الخلاصة:

يتضمن موضوع هذا البحث دراسة عملية لانتقال الحرارة بالحمل المبرى وكذلك لمقاومة الاحتكاك في المبادلات اللوحية ذات القنوات اللامة المفرقة بهدف زيادة معامل انتقال الحرارة نتيجة لحفير القطع على سطح القنوات مع زيادة ملحوظة في مقاومات الاحتكاك. وقد صممت هذه الدراسة تجريبيا بمعامل كلية الهندسة جامعة المنصورة وذلك بانفا، وتصميم جهاز حثيين تيار من الماء خلال مروره في مبادل حراري لوض من النوع ذي القنوات اللامة المفرقة وقد استخدم في هذا البحث أربعة ألواح من النحاس الأصفر ابعاد كل منها ١٠٠ × ١٠٠ × ١.٠، وقد شكل اثنان من الألواح بالتشعير بحيث انه عند وضع اللوحين متوازيين تعطى لناة مفرقة لامة بنسبة ١:٥ وزاوية انحراف ١٢° وطول الجزء المنفرح ٢٥ سم وطول الجزء اللام هم هذا بالإضافة الى عكس اتجاه سريان الماء ليعطى لناة مفرقة لامة بنسبة ١:٥ وزاوية انحراف ٢٧° في حين خلقت المسافة بين اللوحين ٢٠ سم وذلك باستخدام لوهين من الزجاج المطاطي، وقد اجتمعت الألواح اللامة اشكال مختلفة من القنوات، وتم حثيين ألواح النحاس باستخدام بخار الماء الجاف المعدب بدرجة حثيين من ٢:٢م وتشعير سرعة الماء لتعطى رقم رينولد من ١٥٠٠ الى ١٠٠٠٠، ورقم براندتل من ٣,٣ : ٧,٣ .
ولقد تم عمل دراسة مقارنة بهدف تحديد المثل لناة حثلي اعلى معدل انتقال حراري والى نسبة زيادة مقاومات الاحتكاك مع الاحتفاظ بالى مساحة سطح ولد الظهت النتائج ان معدل انتقال حرارة بالمبادلة باللناة العاديكية المسحوية الا سطح كان ١,٤١٥ الى حين كان معامل لدره السطح لم تتجاوز ٢,٠ .

ABSTRACT- This paper presents an experimental study of hydrodynamic resistance in unconventional ducts in case of water flow. These ducts are made of two opposite vertical copper plates in the 10 mm and has been shaped by milling. The duct is an open loop through the ducts of 20 mm. The first has two plain slides, the second has two symmetric shaped slides and the third has two symmetric shaped slides and a central plate. The results are also obtained when the duct is used as a heat exchanger. The data are presented for Reynolds number and Prandtl number range between 3.3 and 7.3. The frictional pressure. A comparative evaluation is made of heat transfer as compared to the straight duct. The pumping power has a factor of about 0.2 and 0.175.

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INTRODUCTION

Intensification of convective heat transfer, in plate heat exchanger, can be achieved by different methods. One of the most effective methods can be obtained by creating an alternating pressure gradient and consequently alternating momentum forces in the flow stream. Means and ways of powering turbulent heat transfer in ducts have been vigorously explored in the recent past with a view toward increasing the efficiency of heat exchangers [1-4].

For flow in a duct of constant cross section (conventional duct), the velocity distribution becomes independent of the streamwise coordinate at sufficiently large distances from the duct inlet. Such an unchanging velocity distribution is said to be fully developed. However, for the temperature field, the fully developed regime is not as easily characterized as that for the velocity. Aside from some certain special cases, the temperature distribution does not become independent of the streamwise coordinate and the usual definition of the thermally developed regime is that the shapes of the temperature distributions at successive streamwise locations are the same, so that the successive distributions can be brought together by a suitable scaling. In such a thermally developed flow, the heat transfer is independent of the streamwise coordinate in the light of the aforementioned properties, the reported fully developed heat transfer coefficients (for unconventional ducts) are much higher than those for conventional duct flows and show a remarkable dependence on Reynolds number.

Cokhman and Kirpikov [1] studied experimentally the turbulent motion of air flow along the converging-diverging duct. They reported that intensification of heat transfer was high and there was also moderate increase in the pumping power.

Araid and Awad [2] studied heat transfer and hydrodynamic resistance in air plate heat exchangers type converging-diverging sides. They were studying the spacing effect between the two slotted sides on the heat transfer as well as on the pumping power for channel of 8 mm length for the diverging part and of 4 mm length of the converging part. In the work, the heat transfer intensification was about a factor of 1.30, and the pumping power was also about a factor of 0.40. Araid et al. [3] studied, on the same test section, the same problem using transformer oil instead of air. In this case they found that the heat transfer intensification was about a factor of 1.55 and the pumping power factor was equal to 0.35.

Awad et al [4] studied experimentally heat transfer and hydrodynamic resistance for air flow through a periodically converging-diverging duct. The converging part was of 5 mm length and the diverging one was of 25 mm length, and the slotted sides gap was equal to 20 mm. The results show that the heat transfer intensification is a factor of about 1.50 and the hydrodynamic resistance is a factor of about 0.238.

The present work is devoted to study experimentally, on the bases of the above literature review the enhancement of convective heat transfer as well as the pumping power effect in case of water flow through a periodically converging-diverging duct. The duct was formed by two slotted sides of converging to diverging ratio equal to 1:5. The two vertical sides of the duct were heated by a steam at atmospheric pressure and superheated by 2 to 3 °C. The two vertical sides are made of copper plates 480 x 100 x 10. mm, and are shaped by milling, in which have a conical angle 2°. The channel top and bottom walls that fix the spacing 20 mm between the vertical sides of the duct are made of perspex [6].

EXPERIMENTAL APPARATUS AND PROCEDURE

The experiments for determining heat transfer coefficients and coefficient of friction for water flowing in a periodically converging-diverging duct utilized the

open-loop system shown in Fig. 1. City water first enters the system through flowmeter (1) which is ducted to the top of a large upstream plenum tank (3). The slotted test section (7) spans between the upstream plenum tank (3) and a downstream plenum tank (17). The purpose of the plenum tanks is to provide a well-defined abrupt inlet and exit for the test section. The water leaves the downstream plenum tank from the top into a drain. The mass flow rate of water is sensed by the flowmeters (1) with the help of the control valve (18). The pressure drop along the test section channel (7) is measured by a differential pressure water manometer (14), to an accuracy of 5%. Two air vents (16) are located on the top of the test channel to make sure that the duct is completely air empty. Slightly superheated steam (2 - 3 °C) at atmospheric pressure is used as a heating medium. The steam is generated in the electric boiler (25) and superheated by means of the electric heater (27). Electric power is supplied by a welding rectifier unit, type MCRA 900, of maximum power 58 KW, maximum current 900 amps, and maximum voltage 65 volts. Through the control valve (18), the steam flows from the boiler to the two side insulated boxes (8). These two boxes are setted to keep the heating medium in contact with the channel slotted sides. The condensate is collected in the condensate tank (21). A glass wool (29) of thickness 50-100 mm is used as insulation material for the hot line. The temperature of the water flow at the test section inlet and outlet is measured by two copper-constantan thermocouples (10,12) of 0.15 mm diameter. The water flows from the upstream plenum tank (3) to the stabilizing section (5), of length 100 mm, through the conical connection (4). Then the water enters the working duct (7) and discharges to the drain through the downstream plenum tank (17). The heated test channel (7) is insulated by two insulation flanges (6 and 9). The temperature of the inner surface of the slotted sides of the duct is measured by thermocouples at four locations (11) placed on the mid height of the heated plate. All the used thermocouples are connected to a 24-point self switching temperature recorder (13), having a full scale of 200°C. For sensing pressure and temperature along the steam line of the test loop three calibrated pressure gauges (22) and three mercury thermometers (23), with scale divisions of 0.1 °C, are used.

Fig. 2 shows the assembled relative positions of the two steam receivers which are connected with the test channel. The mean spacing between the two slotted sides is equal to 20 mm. In each slotted plate there are sixteen convergent-divergent cycles, each of them has a length of 30 mm. Out of this length, 5 mm is the length of diverging part and the remaining length is for the converging part.

RESULTS AND DISCUSSION

The main objective of these experiments is the determination of fully developed Nusselt numbers and the coefficient of friction for water flowing in a periodically converging-diverging duct. The determination of fully developed Nusselt number from the experimental data began with the calculation of the overall bulk temperature rise based on an energy balance of the flowing water. The heat gained by the working fluid is calculated from the energy change of the water flow through the test channel. The corresponding fully-developed Nusselt number is given by

$$Nu = h D_h / k \quad \dots (1)$$

The coefficient of friction is evaluated using the pressure drop (Δp) along the heated duct and is given by

$$f = \frac{2 D_h \cdot \Delta p}{\rho_i w^2 L} \quad \dots (2)$$

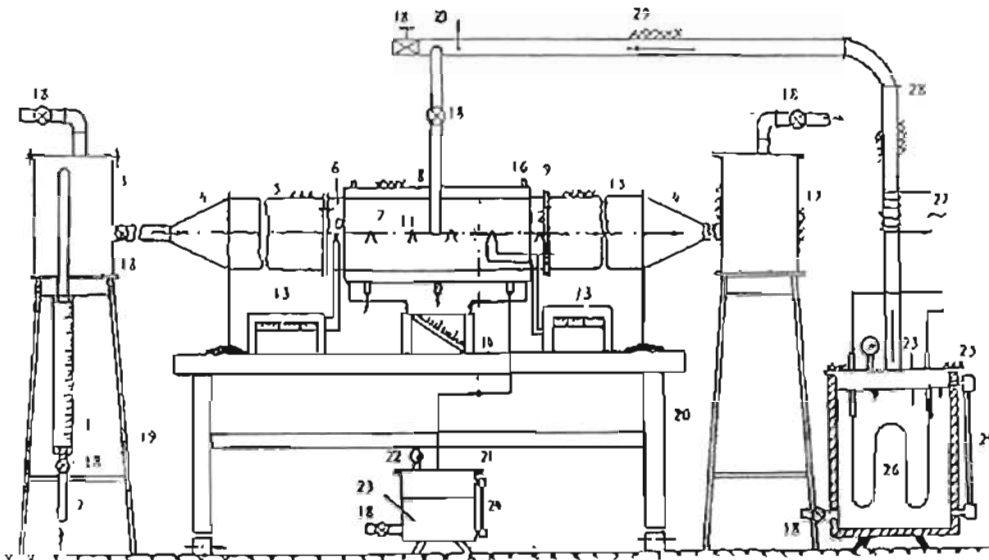


Fig. (1) Schematic layout diagram of the experimental test rig for case of water.
 1. flow meter, 2. water inlet, 3. upstream plenum tank, 4. conical connections, 5. stabilizing section, 6 & 9. insulation flanges, 7. test section, 8. insulated boxes, 10, 11, 12. copper-constantan thermocouples, 13. temperature recorder, 14. inclined manometer, 15. connection part, 16. air vent, 17. downstream plenum tank, 18. control valve, 19. supports, 20. movable base, 21. condenser tank, 22. pressure gauges, 23. thermometers, 24. indicator glass level, 25. electric boiler, 26. electric main heater, 27. secondary electric heater, 28. steam pipe supply, 29. insulation.

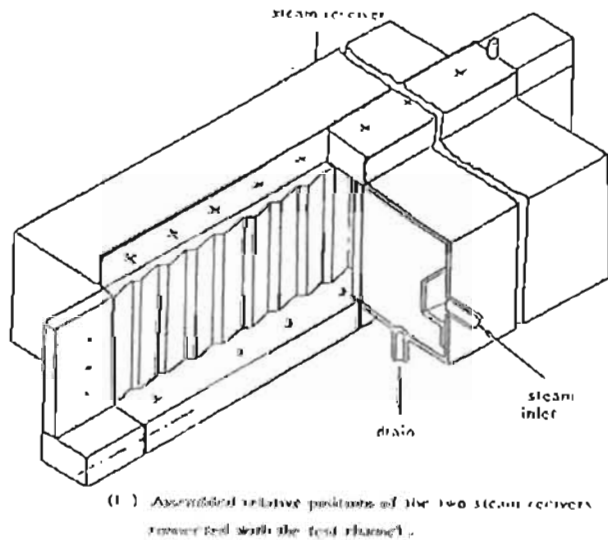
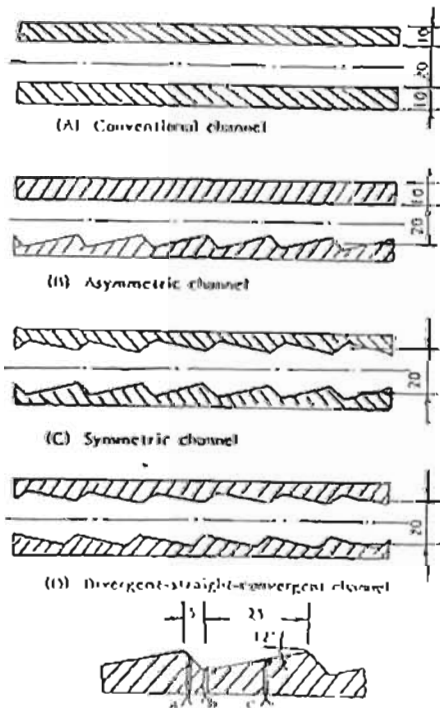


Fig. (2) Test Section Channels

(E) Thermocouple places on one location on the slotted surface.

In this section the presentation will start with the heat transfer results to be followed by the hydrodynamic resistance results and then ended with the comparative evaluation.

HEAT TRANSFER RESULTS

The fully developed Nusselt Numbers are displayed on log-log coordinates in Fig. (3) The figure shows Nusselt number plotted as a function of Reynolds number for four separate test sections. These four test sections make us able to investigate six sets of results. The set number 1 is for a duct that has two plain sides and the set number 2 is for the duct that has one side plain and the other shaped. The shaped plate is designed to consist a converging path with the plain plate, in which the diverging section length is equal to 25 mm and the converging section length is equal to 5 mm. The set number 3 is for the duct of set number 2 but the test section is reversed with respect to flow direction. In case of the duct that has two symmetric shaped sides, there are two sets of results which have been obtained. One of them is for set number 4, for channel of diverging-converging ratio 5:1, and the other one is for set number 5, in which the direction of water flow is reversed. The duct of set number 6 is formed when one of the shaped plates, in the symmetric duct, is rotated in its vertical plane 180° and the other one is kept fixed. The heat transfer results of each set are plotted in the figure along with its corresponding symbol. The straight lines plotted through each set of data in Fig. (3) is determined from a least squares fittings, through all the data points.

In Fig. (3) Nusselt number is plotted as a function of Reynolds number for the six separate test ducts. The heat transfer data for each duct is plotted in the figure along with its corresponding data symbol. Each Nusselt number shown has been corrected to its set reference Prandtl number by a $Pr_f^{0.43} (Pr_f / Pr_s)^{0.25}$ dependence. The enhancement of heat transfer coefficient provided by the unconventional ducts is obviously seen in Fig. (3), by comparison with the heat transfer results for the conventional duct of set number 1.

It is also seen that the enhancement of the heat transfer provided, generally, tends to increase with Reynolds number for the whole sets of data, except the same of set number 3 in which its values start to increase with Reynolds number and then the values come to decrease for $Re > 6000$. This may be due to the decrease in the pressure force along the converging part (25 mm long) for high Reynolds number. The results of the sets number 4 and 5 are shown on the figure top. This may be because of the symmetric flow passage in which the maximum alternating pressure gradient exists along the passage of flow and consequently the effects of momentum force and pressure force are more. The results of the set number 4 are shown slightly higher than the same of the set number 5 by about 7%. This may be due to the difference effect of the compressibility forces on the film layer adhered on the inner surface of the duct 5.

The results obtained for the set number 6 lie below those of the set number 5 by about 10%. The reason may be because of the lengths of the converging and the diverging parts, in which are comparatively short and consequently the alternating pressure gradient effect is less.

In the case of the asymmetric ducts (sets number 2 and 3), figure shows that the data of set number 2 come below the same of the set number 6 by about 6.25%, and the values obtained for the set number 3 are, in average, higher than the same of sets number 5 and 6 and match with those of set number 4 for $Re < 6000$. As such one can conclude that for $Re > 6000$ the other unconventional ducts may consider much better than the set number 3 from heat transfer standpoint.

Overall inspection of Fig. (3) shows that the heat transfer data in the region of low Reynolds number ($Re < 2500$) have some kind of scattering, due to the existence of the transition flow along the duct.

HYDRODYNAMIC RESISTANCE

The values of the coefficient of friction (ξ) are displayed on log-log coordinate as shown in Fig. (4). Each friction factor value has been corrected to its set reference, the ratio of the dynamic viscosity by $(\mu_s / \mu_1)^n$ dependence where

$$\xi = C (P_e D_h / L)^m (\mu_s / \mu_1)^{0.062}$$

for $(P_e D_h / L) \leq 1500$ $C = 2.3, m = 0.1$

$(P_e D_h / L) > 1500$ $C = 0.55, m = 0.1$

Fig. (4) shows that the results obtained for the four ducts of sets number 1 to 6 are displayed. One may observe that the results of the symmetric ducts (sets number 4 and 5) are settled on the top of the figure, and the values displayed for the set number 5 are higher than the same for the set number 4 by about 25%. This may be due to existence of the negative pressure gradient along the diverging parts, in which they represent the major length of the set number 4. Also, the friction factor results for the set number 6 lie below those for the set number 5, specially for low Reynolds number ($Re < 4000$). However, the difference between the results of the two sets gradually vanishes as Reynolds number increases. In fact trend of results is expected because the set number 6 seems to be a semi converging-diverging duct. One may also observe that the results of the asymmetric ducts (sets number 2 and 3) come below the same of the above three sets as expected. The friction coefficients of the set number 3 are higher than the same of the set number 2 by about 60%. This is also expected because the converging part along the flow passage of set number 3 represents the major length of the duct. The lowest values of the hydrodynamic resistance are displayed for the conventional duct of set number 1, in which there is no change in the pressure force and the momentum force along the duct.

Overall inspection of Fig. (4) indicates that the hydrodynamic resistance of whole sets of data increases with Reynolds number. One may also notice that the chosen of the best duct is not that much easy from heat transfer stand point (Fig. 3) as well as pumping power point of view (Fig. 4). One has to do some kind of optimization to define the best channel as shown in the following section.

COMPARATIVE EVALUATION

The performance comparisons between the unconventional ducts and the straight one were carried out for three sets of constrains, as follows:

- 1- Equal transfer surface area (F) and equal pumping power (H). This indicates the variation of KQ with Reynolds number, where KQ represents the ratio of the heat transfer across an unconventional duct to that across the wall of a straight one. This set of constrain shows that the duct of the set number 3 is the best from heat transfer standpoint, in which KQ ranges between 1.48 and 1.37 with Reynolds number range between 1500 and 10,000.
- 2- Equal rate of heat flux (Q) and equal transfer surface area (F). This set of constrain shows the variation of KN represents the ratio of the pumping power needed to

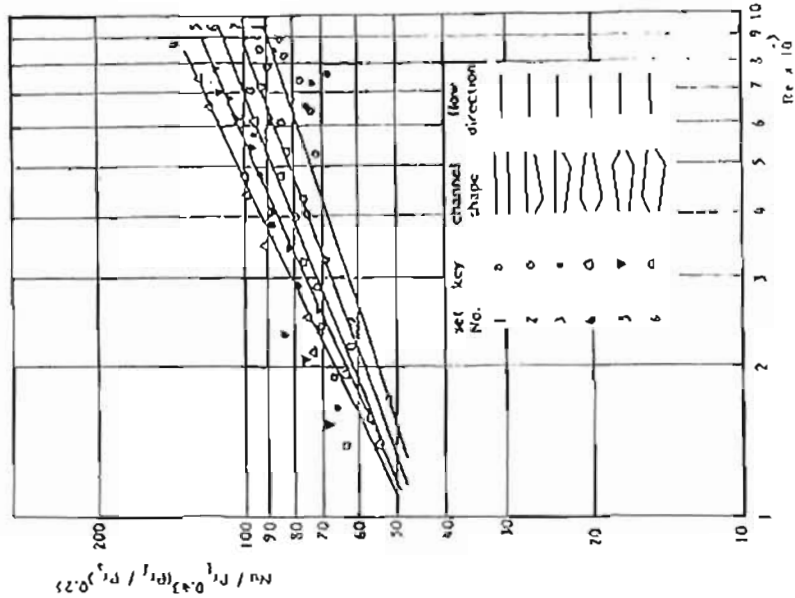


Fig. (4) Comparison of heat transfer results of unconventional channels with those of straight duct.

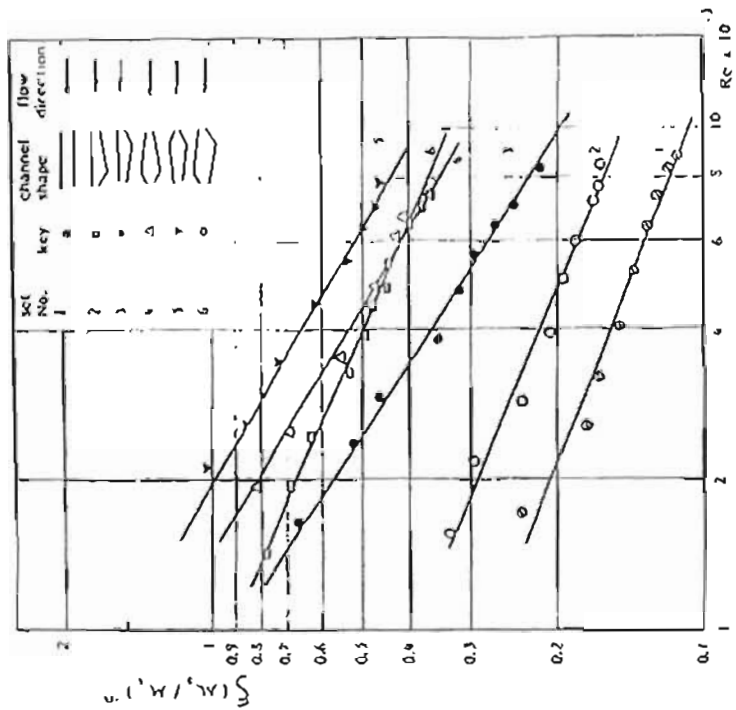


Fig. (3) Comparison of friction results of unconventional channels with those of straight duct.

move water through the unconventional ducts, to that needed in the case of the straight one. The set of constrain also, shows that the duct number 3 gives the less power factor, in which KN ranges between 0.414 and 0.05.

- 3- Equal rate of heat flux (Q) and equal pumping power (H). This set of constrain indicates the variation of KF with Reynolds number, where KF represents the inner surface area of an unconventional duct to that for a straight one. Also, the set shows that the most compact duct is the duct of set number 3, in which KF ranges between 0.214 and 0.152, For more details see reference [6].

CONCLUDING REMARKS

The performance analysis of water flow data shows that the results of unconventional ducts are compared with those for the straight one, for three sets of constrains, out of this analysis, one can conclude that.

- 1- The heat transfer enhancement is associated with the unconventional ducts.
- 2- The comparative evaluation shows that the duct of set number (3) enhanced the heat flux by a factor of about 1.415 and the pumping power by a factor of about 0.20 and the surface area by a factor of about 0.571. This indicates that the surface area density increases about two times .
- 3- This projects the unconventional channels as a strong candidate for high heat flux and compact engineering applications.
- 4- The investigation also shows that the assymmetric channels with one side plate is plain and the other side is slotted, give a remarkable decrease in the pumping power factor.

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NOMENCLATURE

A	cross-sectional area of the channel (mm^2)
c_p	specific heat at constant pressure ($\text{kJ/kg} \cdot \text{K}$)
D_h	hydraulic diameter of the channel, ($4A/P$), (mm)
F	inner surface area of the unconventional duct (mm^2)
h	heat transfer coefficient. ($\text{W/m}^2 \cdot \text{K}$)
K	coefficient of thermal conductivity. ($\text{W} / \text{m} \cdot \text{K}$)
KF	surface area factor, (F / F_0)
KN	pumping power factor, (N / N_0)
KQ	heat transfer factor, (Q / Q_0)
L	Length of the test channel, (mm)
N	pumping power used to move air inside an
Nu	average Nusselt number, ($h \cdot D_h / K$)
\dot{m}	mass flow rate, (kg / sec)
P	Pressure, (N / m^2) and the perimeter bounds A , (mm)
Pe	Peclet number, ($Re \cdot Pr$)

Pr	Prandtl number, ($\mu \cdot cp / K$)
Q	rate of heat transfer flux in an unconventional channel, (W / m^2)
Re	Reynolds number, ($w Dh / \nu$)
w	mean velocity, (m / sec)
μ	dynamic viscosity, ($N \cdot sec / m^2$)
ν	Kinematic viscosity, m^2 / sec
ρ	density, (Kg / m^3)
ξ	coefficient of friction

SUBSCRIPTS

i	properties at inlet condition
o	related to straight tube
s	properties at surface temperature.

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