**Heat transfer and Pressure Drop Comparison for Corrugated Tube and Different Numbers of Copper Foam Cylindrical Inserts in laminar flow**

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**Abstract:**

The chief objective of the present investigation is to study experimentally the hydraulic and thermal performance of a corrugated tube with copper foam cylindrical inserts (MFCI) containing a new types of inserts with the purpose to improve the procedure of heat transfer, which gives an elevated thermal enhancement factor (PEF) more than that of smooth tube beneath the similar operating circumstances.Also, inthis paper,the study of the effect on the number of cylindrical insert was made from metal foam (10 PPI and porosity 0.9) on theNusselt no. and Nusselt no. ratio as a double heat exchanger furnished with the suggested MFCI with various values of Reynolds numbers ranges from (1600) to (4000) via using water as the test fluid.The investigational outcomesrevealed an enhancement inheat transfer as well as thermal performance of the corrugated tube with MFCI which being significantly augmented in comparison to those for smooth tube. Additionally, the average rise into the value of heat transfer is within (74%) and (90%) at the test range, relying upon the number of cylindrical insertsas well as Reynolds no., whereas the max. thermal performance is obtained to be around (2.4) for utilizing the corrugated tube having (4) cylindrical inserts at low Reynolds no. Furthermore, the outcome of the loss of pressure manifested that the corrugated tube's mean friction factor is within 96% and 97% more than the smooth tube.

**Keywords:** Heat exchanger, Corrugated tube, Copper foam, thermal‐hydraulic performance, cylindrical inserts

1. **Introduction:**

Double tubes heat exchanger are utilized as a heat transfer method which is broadly employed in industrial processes, such as solar energy field, chemical industry, nuclear power in the last few decades, and many technologies have been studied to improve heat transfer and increase the thermal performance in order to reduce the energy consumption and the operational cost[1–5].

Several techniques have been used for improving the rate of heat transfer via decreasing the thickness of boundary layer as well as the introduction of better fluid mixing, which is conducted by: 1- Heat exchanger surface methods, 2- Working fluid methods and, 3- Combination of the two (working fluid + surface methods).

In heat exchangers, the modification of surfaces (corrugated) and inserting fluid tabulators are utilized upon the tube surface side that gets into contact with a fluid of low convective heat transfer for decreasing the thickness of boundary layer and induce chaotic blending of fluid blending. And, the main techniques for reducing the thickness of boundary layer are done viathe augmented velocity of stream and the swirl blending.As well, secondary vortex flows can more improve the flows of heat transfer from the wall core for reducing the boundary layer thickness as well as the secondary flows from the wall to the core encourage blending[6].

A combination method of two or more of the existing technologies can be used simultaneously to provide an improvement higher than that achieved by only one technology alone. This technology is known as composite enhancement[6].

Composite enhancement contributes to increasing the heat transfer coefficients' values in comparison with to the values sum achieved from the separate methods.

There are few researchers have concerned with corrugated tube with inserts. Experimental and theoretical studies on composite of heat transfer enhancement technology have been very encouraging. Al-Fahed et al.[7], evaluated experimentally the heat transfer coefficient and pressure drop of both microfin and twisted tape inserts for fully developed laminar flow conditions.

V. Zimparov[8–11], conducted an investigational work of the coefficient of heat transfer and the drop of pressure for a corrugated tube with twisted tape tubes and compared the experimental results with smooth tubes in the range of Reynolds number (4000-60000). Considerably, the heat transfer coefficient and pressure drops are significantly higher than the smooth tubes under the same operating conditions.

K. Wongcharee and S. Eiamsa [12], tested the effect of the CuO/water nanofluid in a corrugated tube with a twisted tape on the heat transfer and friction factor. The results showed that the max. thermal performance factor was found to be (1.57) with the nanofluid utilization at a (0.7%) concentration.Heat transfer rate and friction factorrise with raising the nanofluid CuO/water concentration as well as the twisted ratio of twisted tap.

Wan et al.[13], manifested the nanofluids' heat transfer enhancement mechanism via investigational method using a corrugated tube packed with the copper foam (40 PPI) and in comparison with that in test tubes without using copper foam.And, the outcomes depicted that the core-improvment zone for the heat transfer utilizing investigational tubes packed with the copper foam is remarkably unalike from that of the tubes without using copper foam

Ding et al.[14] ,conducted a numerical heat transfer investigation, and the heat transfer as well as the flow features of TiO2-H2O nanofluids with different mass fractions of TiO2 in smooth and corrugated double-tube heat exchangers were compared using numerical simulations. The results revealed that the heat exchange rate slowly augmented with increasing the mass fraction of TiO2. In general, the nanofluids and corrugated tubes utilization greatly improves the heat exchange rate and thus the heat transfer efficiency of the heat exchanger.

As evinced from literature, there's no data upon pressure drop and heat transfer in corrugated tube with copper foam cylindrical inserts.The present investigation is to study the importance of utilizing corrugated tube with a various numbers of copper foam cylindrical inserts (MFCI) and comparing its thermo-hydraulic performance factor with smooth tube for filling such gap of knowledge via an investigational effort founded upon the PEF.Also, the aim of this research into using innovatively designed MFCI is for increasing the rate of heat transfer between the surface of the corrugated tube and the fluid, which gives a better PEF than those for a smooth tube with the similar working circumstances as well as power of pumping. Additionally, the innovative design is able to open a broad door for adapting, designing, production, and fabricating the heat exchangers in the manufacturing.

**2. Experimental Procedure:**

Figure 1 represents the schematic of the experimental work. The experiment was conducted in a double-tube heat exchanger with and without cylindrical inserts. The specifications of the heat exchanger possessed a (di=26 mm) internal diameter with a (1 mm) wall thickness and a (500 mm) length, whereas the Perspex tube has a (D=66 mm) diameter with a wall thickness of 1 mm and a length of 700 mm. The outer wall surface of the Perspex tube was completely insulated with glass wool to reduce the heat exchange with surrounding.

The cooled and heated water were pumped from a cold water storage tank as well as a hot water storage tank, respectively. And**,** the hotwater was pumped into the external tube and the cold water into the internal tube. Also, the cold and hot water closed loops comprising 0.75 m3 storage tanks are kept constant at their specified temperature using a cooling cycle, andan electric heater governed via the voltage controller, respectively.Thecycleof coolingcontains a compressor, evaporator (a cooling coil submerged in a storage tank), condenser, refrigerant, and expansion.Few valves as well as (2) flow meters were employed for controlling and measuring the rate of mass flow through the test section with an accuracy of 2% of a full scale.

Twelve K type thermocouples were installed in the sections upon the corrugated tube's external surface depicted in fig. (2A), previously calibrated in the test section as well as dispersed evenly. Also, the cold and hot water temperatures at the exit and inlet of test section were measured. And, the temperatures reading were connected to a data logger system having a capacity of 12 channels. The drop of pressure across the interior tube was registered via utilizing (2) pressure transducers, (1) at every test section end.

A fresh kind of a cylindrical shape inserts was inserted into a corrugated tube. Three cases of insert were used; each case has a different number of cylindrical (2, 3 and 4 cylindrical inserts). The cylindrical inserts were made from copper foam (10 PPI and porosity 0.9) with a diameter of 14 mm and a length of 28 mm[15], as shown in fig (2B). The corrugated tube was made from copper with a (1 mm) wall thickness, a (26 mm) interior envelope diameter, a (1.5 mm) depth (e) of helical corrugation, and a (11 mm) pitch () of corrugation. And, in every test, the cylindrical inserts direction was oriented parallel to the spiral direction of corrugated tube (denoted as the parallel arrangements), as shown in fig. (2A).

Experiments were carried out with different mass flow rate values and for the whole circumstances. The hot water (Perspex tube) mass flow rate value and the inlet temperature were maintained at 2 lit/min and 70 correspondingly. And, the cold water flow varied between the inlets (from 2 to 4 lit/min), and the inlet temperature was around 22, measured at the steady-state condition. The experimental work was then repeated for without and with cylindrical inserts using various numbers of copper foam cylindrical.

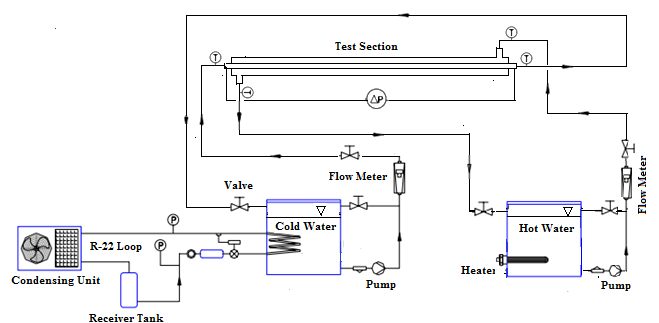
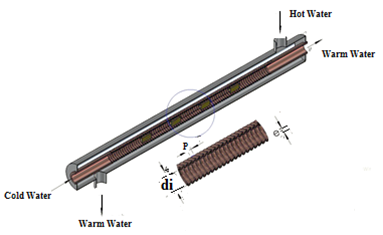


Figure 1: Schematic diagram of the experimental setup



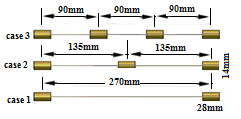


Fig. 2(A): Corrugated tube equipped with copper foam cylindrical inserts. (B) Geometrical arrangement of the copper foam cylindrical inserts.

**3. Uncertainty Analysis**

Uncertainties in the current investigational outcomes rely on the separate measuring apparatuses accurateness and the tubes production accurateness.And, the instrument accurateness is restricted via its min. division (its sensitivity). Uncertainties are assessed depending upon the differential estimate.They are estimated relying upon the differential estimate technique.Also, for a characteristic experimentation, the temperature uncertainties are (0.1°C), the uncertainty of pressure around (±2%), the surface area of flat tubes is (3%), and the rate of flow is (2%), in addition tothose of geometric factors. For the results of the physical factors, the uncertainties of Re no., (*f*), and mean Nu no. are assessed as a max. of 4.1%, 2.8%, and 3.2%, correspondingly.

**4. Data Reduction**

During the experimental test, hot water supplied heat to the cold water that flows in the heat exchanger. And, the heat supplied via hot water and heat absorbed via cold water can be written as [16]:

(1)

(2)

Where, and

From the investigational outcomes, the heat absorbed via the cold water is obtained that it ranged between 5% and 9% lower than the provided heat by hot water at energy balance due to the radiation and convection heat losses from the test rig to the surrounding. The average heat transfer rates, is calculated by using the following equation:

For the flows of fluid in a double tube-heat exchanger, an overall coefficient of heat transfer is evaluated by:

Where,

An inside coefficient of heat transfer) is usually calculated from the overall heat transfer coefficient as follows:

If the latest (3) terms are upon the right-hand side of Eq. (5) are reserved fixed, this equation (5) can be stated as follows:

The coefficient of heat transfer is connected with the Re no. is[17]:

Rearranging and substituting equation (7) into equation (6) yields:

Equation (9) suggests that the plot between (Re-m) and (1/U) is a straight line with the interception at (B) in the Y-axis of (1/U). And, this aids obtaining the values of (A) and (B).

The average Nusselt number based on the cold water (corrugated tube) side surface area is given by:

The Re no. for the cold water corrugated tube inlet is:

Where, μ is the water's dynamic viscosity

The friction factor () is determined from [18]:

Thermal performance factor can be calculated from [19]:

Where, are the friction factors and Nusselt numbers for the smooth tube, respectively.

The effectiveness equation can be calculated with [19]:

(14)

**5. Exergy Analysis**

Pressure drop difference and temperature are the two main causes of losses in heat exchanger. And, in the present study, the study of exergy is founded upon irreversibility of heat transfer. The loss of exergy can be computed that [20, 21]:

In this study, it can be assumed that the amount of heat supplied by hot water is equal the amount of heat absorbed by cold water, and amount of the loss of heat is ignored. Then, the rate of the loss of exergy is as following:

The loss of exergy can be expressed as:

are the loss of exergy for cold water and hot water, respectively, using these following equations:

Entropy modifies can be obtained as below

Substituting Eq. (18) and (19) into Eq. (17) yields:-

Finally, the loss of dimensionless exergy is written as following:

Where, = is the ambient temperature (equals to 25°C).

**6. Experimental Results**

The corrugated tube influences with various numbers of copper foam cylindrical inserts MFCI (N = 2, 3, 4) upon the heat transfer as well as (*f*) being introduced in such section. First, the friction factor and heat transfer determined from the current smooth tube as well as the investigational ability being confirmed with such determined from standard relationships.Second, the cylindrical inserts no.influences upon the friction, heat transfer, and thermal effectiveness features of heat exchangers are illustrated.

**6.1 validation of the system**

For testing and verifying the dependability of the investigational data for the flow, as well as heat transfer performance, have to be studied initially.The empirical equations or relevant investigations about the flow and the heat transfer performance of the fluids of test into the corrugated tube using cylindrical inserts are fairly rare, and upon the else hand, the investigational apparatus of corrugated tube is the similar as circular tube in the present work, thus the circular tube's flow and the heat transfer features are investigated experimentally, chiefly for validating the investigational regime.

Also, the obtained investigational outcomes from the present study are depicted in figs. 3 and 4.In addition,the Nusselt no. and friction factor results are compared with the connected correlations as well as preceding studies, and the standard relationships of convective heat transfer that are recommend via the literature [22–25] are revealed below:

Sieder-Tate formula [22]:

Gnielinski formula [23]:

The laminar friction factor () of Petukhov was given analytically as follows[24]:

Using developed formula Nusselt no. and frication factor for a corrugated tube by Vicente et al. [25]:

As can be seen, both results of Nusselt number (Nu) and Darcy friction factor ( development in the Reynolds no. (Re) function are in a virtuous agreement with each another, with the max. deviation (8.5%) of Nusselt number and the max. error (8.1%) of friction factor for the Petukhov [24] relationships, correspondingly.

The experimental work results of corrugated tube have a good agreement with that in previous research. The max. deviation is (4%) of Nusselt numbers. While the max. error of the friction factor is (3%), therefore the heat exchanger is reliable.

Fig. 3: Results data qualification of Nusselt no. for smooth tube and corrugated tube

Fig. 4: Results data qualification of friction factor for smooth and corrugated tube

**6.2 Heat transfer results**

The effect of the numbers of metal foamed cylindrical inserts corrugated tube is going to be tested in such section.And, the development of both Nu no. and the Nu/Nus ratio of the corrugated inner tube heat exchanger at the function of the Reynolds number for the numbers of cylindrical inserts are illustrated in figs. 5 and 6. It's interesting to note that the Nu rises with the rise of corrugation parameters and the lowering of Nu/Nus ratio for all Reynolds numbers. Investigational data evinces that the smooth tube performance is the lowermost and doesn't manifestanimportant rise in spite of raising the Reynolds number value. Whereas, the suggested design MFCI elucidates a noticeable improvement in the Nu no. in comparison with the smooth tube.

Maximum average Nusselt number (Nu) values for smooth tube is about 9 in the all Reynolds no. values. But, such values of Nusselt no. dramatically rise for the MFCI and get four times, and in accordance to fig. 5, the corrugated tubes with MFCI give higher Nu nos. than the smooth tube for the whole Re nos. at about (74% to 90%) relyingupon thenumber of copper foam cylindrical insertsutilized. And, in corrugated tube situation with MFCI, the average rise in Nusselt no. is around (90%) at the corrugated tube of the four cylindrical inserts.

Such heat transfer enhancement value via MFCI created a swirls motion leading to the generated higher turbulence between the MFCI and the heat exchanger's tube wall, which results in the secondary vortex creation in the metal foam at elevated intensity given to the flow of fluid. As well, the rate of heat transfer increase of employing MFCI increases with the cylindrical inserts number increase, but MFCI portrayed the uppermost rate of heat transfer with regard to the smooth tube. And, that's due to that the area of heat transfer between the fluid and solid components of the network of metal foam is greater than that into the smooth tube [26].

The corrugated tube utilization in heat exchanger raises the elevated contact surface as well as better blending between the corrugated wall fluid and the core owing to the turbulence/re-circulation flow between the elements of tube wall**.** Corrugations generatesecondary flows throughout the heat exchanger, and such flows reduce the thickness of thermal boundary layer made upon the interior tube's external surface area [27].

Fig. 5: Relationship between Nu no. and Re no. (Re)

Fig. 6: Relationship between (Nu/Nus) ratio and Re no.

**6.3 Friction Factor**

Effect of the number of the cylindrical inserts on the isothermal pressure drop denoted via the features of friction factor () and () at different Reynolds nos. is depicted in figs. 7and 8.And, the drop of pressure () and () in the corrugated tube slowly reduces with raising Reynolds no. for all cases.Also, at a certain Reynolds no., the pressure drops (friction factors) of interior corrugated tubes being reliably greater than that for smooth tube. It's observed that the corrugation tube geometry with cylindrical inserts induce a harmony and orderly vortices at secondary flow region which helps dissipate the fluid dynamic pressure owing to the elevated losses of viscosity close the wall of tube from the corrugated surfaces, resulting in the fluid particles collisions due to curvatures which causes more friction between the working fluid and the tube wall. The friction factor values of the corrugated tube with MFCI, N = 2, 3, and 4 are found to be 91%, 92% and 93% over the smooth tube, respectively. While, the maximum value are found to be 96%, 96,5% and 97% over the smooth tube.

Fig. 7: Relationship between friction factor and Re no.

Fig. 8: Relationship between friction factor ratio and Re no.

**6.4 Thermal performance factor ()**

The **()** of corrugated tube having various cylindrical inserts nos. is demonstrated in fig. 9.And, (2) remarks can be obtained in such figure: (i) raises with the cylindrical inserts number increase, and (ii) have a tendency to reduce as Re no. raises.Also, it proposes that the corrugated tubes being viable as a function of the energy saving at the lower Re nos. Experimental results were determined for PEF in the situation of employing the suggested MFCI, while the max. and min. thermal performance factor was about 1.9 and 1.37, 2.1 and 1.4, 2.5 and 1.55 with the cylindrical inserts N = 2, 3, and 4, respectively.

And, that's due to the high performance in the procedure of heat transfer as well as Nu in comparison to the frictional losses rise and because of the greater area of heat transfer between the operating fluid and MFCI, in addition to the corrugated wall and the fluid flow inside the heat exchanger pipe.

Throughout the obtained experimental results above, it's enhanced when the highest thermal performance of the suggested MFCI having four cylindrical inserts being advised. In the meantime, if the isothermal of the drops of pressure play an important role, the MFCI with a high number of cylindrical inserts can be advised, as the uppermost gained PEF in such model being around (2.5) at (Re=1820).Such MFCI variation is similarly gained for the designers whose main objective being the thermal-hydraulic performance. And, the PEF increases with increasing the number of copper foam cylindrical inserts for lower Re and slowly drops at greater Re if utilizing heat responsibility as a heat exchanger performance indicator. This is due to that the larger number of cylindrical inserts number can make sturdier secondary flow causing a higher rate of convective heat transfer.

Fig. 9: Relationship between thermal performance factor and Re no.

**6.5** **Effectiveness**

Figure 10 presents the variation of heat transfer effectiveness in terms of the Re number. It's obvious, as displayed in the fig. 10 that the effectiveness lasts to dramatically raise as the rate of Re no. raises, for the whole experimental cases. Additionally, it may obviously see that the effectiveness development is obvious for the corrugated tube with cylindrical inserts in comparison with smooth tube. From the other side, it was observed that the average effectiveness determined via the MFCI employment in the utilized Re range is obtained to be around 82%, 92%, and 93% higher than smooth tube. The reason is that vortices occur around the corrugated surface, and then high and developed vortices occur after the corrugated surface. In addition, these vortices increase as the number of metal foam cylinders increases. And then MFCI with a high number of cylindrical inserts (N=4), is more effective for interruption of development of the thermal/hydrodynamic boundary layer of the fluid flow and increase the degree of turbulence intensity than that with a number of cylinders (N=3 and 2).

Fig. 10: Relationship between efficiency and Re no.

**Loss of Exergy**

From fig. 11, it's noticed that the corrugated having cylindrical inserts raise the loss of exergy in the whole cases. Generally, the causes are that the corrugated tube as well as the cylindrical inserts produced from metal foam generates secondary flow that raises the level of turbulent in the flow of fluid and reduces the boundary layer alongside heat exchanger. The investigational outcomes exhibit the loss of exergy in the state of utilizing the suggested MFCI, while the max. loss of exergy is around 0.22 ,0.44 and 0.45 with cylindrical inserts N = 2, 3, and 4, correspondingly.

Fig. 11: Relationship between loss of exergy and Re nos.

**7. Performance Comparison with Other Previous Works**

Many previous research results have been summarized by the use of inserts in a corrugated tube or Non-corrugated with different geometric shapes as shown in Table 1. Generally, all cases are modified in the heat transfer process and the friction loss is increased. The PEF is an indicator of the extent of system improvement. In general, the PEF decreases with increasing Re number. It is clear that the present modified tapes has a good and acceptable improvement, due to two main factors, one of which is use of a corrugated tube that the corrugations generate secondary flows throughout the heat exchanger, and such flows reduce the thickness of thermal boundary layer . While the other factor, is the use of copper foam inserts that generate secondary vortex, which increases heat exchanger.

TABLE 1 Performance comparison with previously published works

|  |  |  |  |
| --- | --- | --- | --- |
| Author | type | Re rang | PEF |
| Qi et.al [28] | Nanofluids with corrugated tube | 600-2400 | 1-1.13 |
| Wan et. al[13] | corrugated tube with Nanofluids, copper foam | 600-4000 | 1.7-2 |
| P. Sivashanmugam and P.K. Nagarajan[29] | Circular tube fitted with screw-tape inserts | 1000- 10000 | 1.65 |
| P. Murugesan et. al [30] | Circular Tube Fitted with Twisted Tape inserts | 2000-12000 | 1.27-1.33 |
| S.K. Saha [31] | Rectangular Ducts With Axial Corrugation Roughness and Twisted Tapes | 100-10000 | 0.81-1.32 |
| S.K. Saha [32] | Circular tube having helical corrugated with screw-tape inserts | 100-10000 | 1.22-1.33 |
| V. Zimparov[8] | Three start spirally corrugated tubes with a twisted tape | 3000-70000 | 1.15-1.25 |
| L. Yang[33] | Corrugated Tubes With Twisted-Tape Insert | 3000-10000 | 1.15-1.53 |
| Present work | Corrugated tube with MFCI | 1600-4000 | 1.55-2.5 |

**8. Conclusions**

In present paper, the flow of fluid as well as the heat transfer performance in a smooth and corrugated tube having a number of copper foam cylindrical inserts in a double tube heat exchanger is evaluated. And, the effects of Nusselt number, pressure drops, factor of thermal performance, and effectiveness with various Reynolds no. ranges are regarded beneath variable working circumstances. Also, the subsequent conclusions can be drawn from this study:

* The combinations of cylindrical inserts made of metal foam inside the corrugated tube can enhance the heat transfer by 74-90% compared with smooth tube. And, the study was conducted beneath isothermal fluid circumstances for Reynolds number range (1600-4000).
* The pressure drop (friction factor) for the suggested MFCI model in comparison with the model of smooth tube yields higher values of around (96- 97%).
* For the thermal performance factor in the situation of utilizing the suggested MFCI model, the max. and min. thermal performance factor is around (1.9 and 1.37), (2.1 and 1.4), and (1.6 and 2.5) with cylindrical inserts N = 2, 3, and 4, correspondingly. Also, it was observed that the PEFs with implementing MFTT suggested in the present investigation are generally above the unity.
* The obtained average effectiveness via the MFCI employment in the utilized Reynolds number range used is determined to be around 82%, 92%, and 93% higher than that for the smooth tube.
* The maximum loss of exergy when using four cylindrical inserts inside the heat exchanger is about 0.45.

**Nomenclature**

A Area, m2

Cp Specific heat, J/kg K

di inner diameter of tube, mm

D inner diameter of Perspex tube, mm

E Exergy loss, W

e Corrgeted depth, mm

exDimensionless exergy loss

Friction Factor

h Coefficient of Heat Transfer, W/m2 K

k Thermal Conductivity, W/m2 K

Length of pitch, mm

L Length of cylindrical, mm

m Rate of Mass flow, kg/sec

N Numbers of copper foam cylindrical inserts

Nu Nusselt number

Pr Prandtl number

P Pressure, Pa

Rate of Heat transfer, W

ReReynolds number

thermal resistance, K/W

S Specific entropy, J/K kg

T Temperature, oC

U Overall heat transfer coefficient, W/m2 K

V Velocity, m/sec

**Greece symbols**

Drop of Pressure, Pa

Logarithmic mean temperature difference, oC

Density, kg/m3

Dynamic viscosity, kg/m sec

Effectiveness

Thermal performance factor

**Subscripts**

c cold

f fluid

h hot

i inner

o outer

s smooth

w wall

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