Performance Analysis in Study of Heat Transfer Enhancement in Sinusoidal Pipes

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Keywords
Heat transfer enhancement
Non-dimensional entropy generation
Performance analysis
Sinusoidal pipes

1. Introduction

Methods for performance evaluation of the heat transfer enhancement devices are presented in the literature (Campbell 1999, Kakaç et al. 1987, Rohsenow et al. 1985, Sparrow and Comb 1983, Lee et al. 1985). The aim of most of these methods is to evaluate the parameters among themselves such as pressure loss, flow rate and pumping power which are taken as main characteristics. Next, in the case of one of the parameters mentioned as constant, changes in heat transfer and other parameters reveal the performance of enhancement methods. With this analysis, the appearance of multiple performance criteria causes difficulties in identifying a definite result in enhancement technique for comparison. For an optimal solution, characteristics obtained in result of calculations can be re-evaluated among themselves. Despite all these, Bejan (1997) proposed the evaluation method based on the second law of thermodynamics. In this method, irreversibility and entropy generation are used as performance evaluation criteria in the system to which enhancement technique is applied. Entropy generation in convective heat transfer process, arises from fluid-flow friction and heat transfer occurring in finite temperature difference. Consequently, minimization of the entropy generation yields the criterion for system optimization.

Meanwhile performance analysis by considering the entropy generation can be performed with exergy analysis. At this point, the exergy destruction can be determined by using the exergy analysis. As this destruction is equal to the entropy generation, a different approach to performance analysis has been done. Prasad and Shen (1993), in their study, evaluated the performance of heat transfer enhancement devices by using exergy analysis. The difference of this method from Bejans (1997) is that the effect of axial temperature variation in the passage transferring heat is also taken into consideration. In the study, an exergy analysis was made by taking into consideration the fact that the effect of pressure loss in pipe heat exchangers for liquid (water in this study) to
Symbols/Oznake

$L$ - length of the pipe, m
duljina cijevi

$D$ - inner diameter of the pipe, m
unutrašnji promjer cijevi

$T_w$ - pipe wall temperature, °C
temperatura stijenke cijevi

$T_{fi}$ - inlet temperature of the fluid, °C
ulažna temperatura fluida

$T_{fo}$ - outlet temperature of the fluid, °C
izlazna temperatura fluida

$\Delta P$ - static pressure difference, N/m²
razlika statičkog tlaka

$U_m$ - average velocity of the fluid, m/s
prosječna brzina strujanja fluida

$m$ - mass flow rate of the fluid, kg/s
maseni protok fluida

$f$ - friction factor
faktor trenja

$\rho$ - density of the fluid, kg/m³
gustoća fluida

$\tau$ - non-dimensional inlet temperature difference
bezdimenzijska ulazna temperaturna razlika

$v$ - kinematic viscosity, m²/s
kinematička žilavost

Indices/Indeksi

$m$ - average
prosječni

$w$ - wall
stijenka

$fi$ - fluid inlet
ulažni fluid

$fo$ - fluid outlet
izlazni fluid

enthalpy change is small and this can be neglected. As
a result of this, in the study mentioned the expression of
exergy destruction is valid only in pipe heat exchangers
in which liquid is used as a fluid.

In this present study, a performance analysis of
sinusoidal pipes with various amplitudes (in constant
period) and various periods (in constant amplitude) was
carried out by utilizing the non-dimensional entropy
generation equation ($Ns$) mathematically developed for
constant wall temperature thermal boundary condition in
heating application, as suggested by Nag and Mukherjee
(1987). The heat transfer and fluid-flow friction
characteristics which are necessary for the analysis
were obtained experimentally. The pipe configuration
is characterized by the amplitudes and periods of the
Sine function. The respective expressions are
$\sin 0.25X$, $2\sin 0.25X$ and $4\sin 0.25X$, which are called
sinusoidal pipes in constant period, and $\sin 0.25X$, $\sin
0.5X$, $\sin X$ and $\sin 2X$, which are called sinusoidal pipe
in constant amplitude. The sinusoidal pipes were curved
without decreasing in the cross-section along the axis of
the pipe. Experiments were performed for both laminar
and turbulent flow conditions in the Reynolds number
range from 75 to 23000 and air was used as working
fluid. The test pipes were 812 mm long and 4 mm inner
diameter. Comparative studies of heat transfer and fluid-
flow friction results of the sinusoidal pipes were given by
[1-3] for laminar and turbulent flows respectively.

In addition, in order to estimate unavoidable errors
in experimental results, an uncertainty analysis was
performed as well, which is appropriate for the literature
[1, 6, 12].

2. Theoretical Study

2.1. Test setup

This study is a theoretical study using data which have
been obtained experimentally. The experiments were
performed in sinusoidal pipes with various amplitudes
and various periods for hydrodynamically fully developed,
thermally developing and fully developed laminar and
turbulent flows.

The schematic drawing of experimental setup is shown
in Figure 1. The pipe configuration is characterized by the
amplitude and period of the Sine function. The respective
expressions are $\sin 0.25X$, $2\sin 0.25X$ and $4\sin 0.25X$
for constant periods and Sin 0.25X, Sin 0.5X and Sin X
for constant amplitudes. The test pipes were 812 mm in
length and 4 mm in inner diameter. All pipes were made
of copper, resulting in small conductive resistance in
the pipe wall. For constant periods of sinusoidal pipes,
the entire pipe encompassed one corrugated cycle. For
constant amplitude experiments, the pipes encompassed
1, 2, 3.5 and 5 cycles respectively. All the schematic
drawings of the test pipes including straight pipe are
shown in Figure 2.

Experiments were performed under constant wall
temperature boundary condition. A constant temperature
bath was constructed in dimensions of 1050x300x350
mm. The bath was made of Plexiglas, which has very
much lower thermal conductivity than the metallic wall,
to minimize the heat loss of surroundings. Constant wall
temperature was accomplished by sending saturated
steam at atmospheric pressure to the test section, where
this saturated steam was condensed on the surface of
sinusoidal test pipes.

By this method, 100±1 °C constant wall temperature
boundary condition was sustained. This boundary
condition was continuously checked by measuring surface
temperature at 12 different locations on the test pipes.
These temperature measurements were performed by using
copper-constant thermocouples attached to pipes with 65
mm intervals. These thermocouples were placed in holes
drilled from the rear face of the pipe. All thermocouples
were made from 0.127 mm diameter teflon-coated copper
and constantan wire that had been specifically calibrated
for the present experiments. In addition, the equipment
used in experiments includes a compressor, a manometer
for pressure difference measurements between inlet
and outlet, two thermocouples for inlet and outlet air
temperature measurements, a calibrated rotameter for
mass flow rate measurement, a steam generator unit, test
section etc.

The sinusoidal pipe was connected to the compressor
from which air was supplied to the system. An inlet pipe
was used in the system between compressor and test pipe
to provide for fully developed turbulent flow. First, the
straight pipe measurements were performed for different
Reynolds numbers and the correctness of these results
was verified by comparing them with literature. Then,
two different groups of tests were performed. The first
group was called constant period experiments in which
the pipe was bent as a Sine function that has constant
amplitude but different periods. The second group was
called constant period experiments in which the period
was fixed and amplitude was varied.

2.2. Performance Analysis

As stated above, in this study, performance analysis
of sinusoidal pipes with various amplitude (in constant
period) and various periods (in constant amplitude) was
carried out by utilizing the non-dimensional entropy
generation equation (N_t) mathematically developed for
constant wall temperature thermal boundary condition
in heating application, as suggested by [9]. The equation
developed for this purpose in this study is as follows:

\[ N_t = \frac{1}{\rho \cdot T_w \cdot c_{pm}} \left( -\frac{\text{d}P}{\text{d}x} \right) \ln \left( \frac{T_w}{T_{in}} \right) \]

where \( \tau = \frac{T_w - T_{in}}{T_w} \)

where \( T_w \) is the wall or surface temperature of the pipe
and \( T_{in} \) is the inlet temperature of the working fluid. In
Equation 1, the value of the \( \gamma \) was calculated as follows

\[ \gamma = \frac{h_m \cdot \pi \cdot D}{\dot{m} \cdot c_{pm}} \]

where \( h_m \) is the average heat transfer coefficient, \( D \) is the
inner diameter of the pipe, \( \dot{m} \) is the mass flow rate of
working fluid, which is air in this study and \( c_{pm} \) is the
average specific heat of working fluid. At the same time,
in Equation 1, the pressure gradient was calculated as follows

\[ \left( -\frac{\text{d}P}{\text{d}x} \right) = \frac{f \cdot \rho \cdot U_m^2}{2 \cdot D} \]

where \( f \) is the friction factor, \( U_m \) is the average velocity
of the working fluid and \( \rho \) is the density of working fluid. Besides, in Equation, \( L \) is the length of the pipe.

In Equation 3 and 4, the variables of \( h_m \), \( f \) and
\( U_m \) were calculated with the help of the following
equations:

\[ h_m = \frac{\dot{m} \cdot c_{pm}}{\pi \cdot D \cdot L} \ln \left( \frac{T_w - T_{in}}{T_{in} - T_{ot}} \right) \]

\[ f = \frac{( \frac{\Delta P}{L} )}{\rho \cdot U_m^2 / 2} \]
Figure 1. Schematic drawing of the experimental setup

Slika 1. Shematski prikazi eksperimentalnog uređaja

Figure 2. Schematic drawing of the sinusoidal pipes: a) Straight pipe, b) Sinusoidal pipes with various periods (in constant amplitude), c) Sinusoidal pipes with various amplitudes (in constant period)

Slika 2. Shematski crtež sinusoidalnih cijevi: a) Ravna cijev, b) Sinusoidalne cijevi s varijabilnim periodima (s konstantnim amplitudama), c) Sinusoidalne cijevi s varijabilnim amplitudama (s konstantnim periodom)
In Equation 5, \( T_0 \) is the outlet temperature of the working fluid. In Equation 6, \( \Delta P \) is the static pressure difference between the inlet and outlet of the working fluid passing through the pipe.

### 2.3. Uncertainty Analysis

Using conventional techniques, the uncertainties for all of the experimental values measured were estimated. The thermocouples used in the heat transfer section are calibrated individually, and the maximum error is estimated to be ±0.5 °C. The rotameters have been calibrated and the error is less than ±1%. Other errors were also included in the analysis to estimate overall uncertainty appeared during the mass flow rate measurements. Corrections were also made to allow for changes in temperature and pressure during the tests. The manometer has been calibrated to give a maximum error of less than ±2%. An important contribution to the pressure drop uncertainty is the error due to air leakage in the test section. Likely, other errors in pressure drop measurements were included in the analysis to estimate overall uncertainty. Besides, the geometric quantities of the pipes, length and inner diameter, were determined within ±5 mm and ±0.1 mm respectively.

Finally, a root-sum-square combination of the effects of each of individual sources of error yields estimates of the uncertainties as shown in Table 1. Measurements were shown to be reproducible well within these limits.

### 3. Results and discussion

In this study, a performance analysis was carried out by calculating the values of non-dimensional entropy generation utilizing Equation 1 mathematically developed for a constant wall temperature thermal boundary condition in heating application. The values of non-dimensional entropy generation obtained from the experimental data for sinusoidal pipes in constant period and in constant amplitude are shown in Figures 3 and 4 for laminar and turbulent flows respectively. At this
point, the values of non-dimensional entropy generation for each pipe, including straight pipe, were determined by considering the second law of thermodynamics. Comparing the straight pipe, sinusoidal pipe in which the non-dimensional entropy generation is minimum will be the best (optimum) pipe enhancing (augmenting) heat transfer. In other words, comparing the straight pipe, sinusoidal pipe in which the non-dimensional entropy generation is minimum, will be the pipe having the highest performance.

The values of non-dimensional entropy generation obtained for the sinusoidal pipes with constant amplitude and constant period in laminar flow are given in Figure 3. For constant period, $2\sin 0.25 \times X$ pipe shows the minimum values of non-dimensional entropy generation by approximately $Re = 700$ while the minimum non-dimensional entropy generation occurs in straight pipe after this point. At approximately $Re = 2000$, all sinusoidal pipes with constant amplitude show similar values of non-dimensional entropy generation. Although minimum non-dimensional entropy generation is expected to be in a straight pipe, $2\sin 0.25 \times X$ pipe is the one by approximately $Re = 700$, in which minimum non-dimensional entropy generation occurs. In constant amplitude, $\sin 2 \times X$ pipe shows the minimum non-dimensional entropy generation by approximately $Re = 1300$ while the maximum value of the non-dimensional entropy generation is reached in this pipe above approximately $Re = 1300$. After $Re = 1300$, $\sin 0.5 \times X$ pipe shows the minimum non-dimensional entropy generation.

The value of non-dimensional entropy generation determined in sinusoidal pipes with constant period and constant amplitude for turbulent flow are given in Figure 4. As shown in Figure 4a, very closed values of non-dimensional entropy generation appear for the constant period. As shown in Figure 4b, in sinusoidal pipes with constant amplitude, the maximum non-dimensional entropy generation takes place in $\sin 2 \times X$ pipe. However, the minimum non-dimensional entropy generation occurs in $0.25 \times X$ pipe after the straight pipe. Besides, another pipe showing very closed value of non-dimensional entropy generation to $0.25 \times X$ pipe is $0.5 \times X$ pipe.

According to results of uncertainty analysis performed to determine the accuracy of experimental results, in which unavoidable measurement errors are estimated, it is concluded that the maximum uncertainty values can result from errors in measurements of mass flow rate, air outlet temperature and pressure drop. The attention in measurements of mass flow rate, air outlet temperature and pressure drop will decrease considerably. In this study, the results of uncertainty analysis agree with the literature [12].

4. Conclusions

In this study, performance analysis of the sinusoidal pipes with various amplitude (in constant period) and various periods (in constant amplitude) was carried out by utilizing the non-dimensional entropy generation equation ($N_s$) mathematically developed for constant wall temperature thermal boundary condition in heating application. In the performance analysis carried out with characteristics obtained from the experiments, $2\sin 0.25 \times X$ pipe shows the best performance in constant period while

![Figure 4. Variation of non-dimensional entropy generation with Reynolds number for turbulent flow: a) Sinusoidal pipes with various amplitudes (in constant period); b) Sinusoidal pipes with various periods (in constant amplitude)](image-url)

Slika 4. Utjecaj Reynoldsova broja na bedimenzijsku generaciju pri turbulentnom strujanju: a) Sinusoidalne cijevi s promjenljivim amplitudama (uz konstantni period); b) Sinusoidalne cijevi s promjenljivim periodima (uz konstantnu amplitudu)
Sin 0.5X and Sin 2X pipes show the best performance in constant amplitude for laminar flow. All sinusoidal pipes show almost the same performance as a straight pipe while Sin 0.25X and Sin 0.5X pipes show the best performance in constant amplitude for turbulent flow.

Sinusoidal pipes, except for Sin 2X pipe, can be used in the heat transfer enhancement application without extra pumping power. For this purpose, the sinusoidal pipes can be used in solar collectors, refrigeration equipment and heat exchangers, in which heat transfer is needed to augment. Although some difficulties in serial manufacturing of sinusoidal pipes are present, it is possible to overcome these restrictions thanks to new developing technologies.

REFERENCES


