Heat transfer behavior, flow topology and performance improvement in the square duct heat exchanger with various parameters of wavy ring vortex generator

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Abstract
The thermal performance analysis in the heat exchanger duct with wavy ring vortex generator are performed. The numerical simulation (finite volume method with SIMPLE algorithm) is selected to sloved the present problem. The effects of the wavy ring size on heat transfer, friction loss and thermal enhancement factor are investigated for the laminar regime with the Reynolds number around 100 – 2000. The heat transfer characteristic is presented in terms of local Nusselt number on the heat transfer surface, while the flow configuration is depicted in form of streamlines in transverse plane and longitudinal vortex flow. The plots of the average Nusselt number ratio, friction factor ratio and thermal enhancement factor in the heat exchanger duct with wavy ring are also concluded. As the numerical result, it is found that the insertion of the wavy ring in the heat exchanger duct brings the heat transfer rate and thermal performance augmentations in all cases. The optimum gap spacing between the wavy ring and duct walls is an important point for the peak of thermal performance in the heating system.

Keywords: wavy ring, gap spacing, heat transfer rate, heat exchanger, thermal performance.

1. Introduction
In the present day, the demand to improve the thermal system in many industries and various engineering device extremely increases. The augmentation of the thermal efficiency can reduce the operation cost for the process. Therefore, many researchers try to develop the thermal system by both numerical and experimental methods [1 – 15].

For examples, Saravanakumar et al. [16] presented the thermal efficiency of arc shaped rib roughened solar air heater integrated with fins and baffles. They concluded that the lower baffle width and length values give the highest effective efficiency at high mass flow rate. Sahin et al. [17] studied the design of the heat sink with hollow trapezoidal baffles. The corner angle, inclination angle, baffle height, baffle length, baffle width and Reynolds number were investigated. Chai et al. [18] reported the thermo-hydraulic performance of a micro-channel heat sink inserted with triangular rib on one wall. They found that the aligned triangular rib provides higher heat transfer rate around 1.03 – 2.01 times, while the offset triangular ribs performs higher heat transfer rate around 1.01 – 2.01 times when compared with straight micro-channel heat sink. Hu et al. [19] reported the performance development of the baffle type solar collector. Perng et al. [20] investigated the performance augmentation of a plate methanol steam reformer by ribs in the reformer channel. Kumar and Layek [21] reported the heat transfer rate, pressure loss in a solar air heater installed with twisted-rib on the absorber plate for the Reynolds number around 3500 - 21000. The Nusselt number and friction factor correlation of their investigation were also concluded. Du et al. [22] studied the laminar flow (Re = 400 – 1800) and heat transfer characteristic in a tube with sinusoidal rib. The rib height to diameter ratios, rib amplitude to diameter ratios, rib width to diameter ratios, rib pitch to diameter ratios and circumferential rib numbers were investigated. They reported that the heat transfer rate and friction loss are around 4.89 and 5.62 times above the plain tube, respectively. Wang et al. [23] selected the numerical investigation to study the effects of slant rectangular rib in the micro-channel heat sink. They explained that the heat transfer rate increases due to the flow disturbance, the disruption of the boundary layer and the enhancement of the heat transfer area. Chai et al. [24] concluded the thermal performance of the micro-channel heat exchanger inserted with triangular rib on one wall. They claimed that the installation of the triangular rib in the heat exchanger is to reinitialize the thermal boundary layer of the heat transfer surface and also improve the temperature of the fluid mixing. Singh and Singh [25] numerically investigated a solar air heater duct installed with square wave profiled transverse rib. They summarized that the highest heat transfer rate and pressure loss are around 2.14 and 3.55 times above the plain duct, respectively, while the optimum thermal performance is around 1.43.

As the literature reviews above, it is found that the rib or baffle vortex generator gives high thermal performance and heat transfer rate when compared with the other types of the vortex generator. Therefore, the rib vortex generators is selected to develop the heating system in the present work. The rib configuration is developed to support the installation and maintenance in the real system. The modified rib has the configuration similarly as the orifice or vortex ring. The modified ring of the present investigation is called “wavy ring”. Moreover, it is
also found that the wavy ring may give high pressure loss in the heating duct. Therefore, the gap spacing between the edges of the wavy ring and the heating duct is generated. The objectives for the present of the gap spacing are to reduce the pressure loss in the heat exchanger and also increase the vortex strength of the fluid flow. The optimum gap spacing may help to increase the heat transfer rate and thermal efficiency of the heat exchanger.

2. Computational domain of the square duct inserted with wavy ring

The wavy ring is designed as the combination between the wavy rib or double V-rib and the strengthening structure (see Figs. 1 and 2). The objective for the wavy rib part is to generate the vortex flow, swirling flow and
longitudinal vortex flow in the test section, while the strengthening structure is created to support the installation of the wavy ring in the square duct. The test section is square duct with the square channel height, H, around 0.05 m (square channel width = H). The wavy ring height is represented with “b”. The ratio between the wavy ring height to the channel height, b/H or BR, is called blockage ratio. The blockage ratio for the present work is varied in the rang around 0.05 – 0.25. The flow attack angle for the wavy ring is fixed with 30° for all investigated cases. The distance between the wavy ring is fixed around 1H for all cases. To reduce the pressure loss and increase the strength of the flow in the test section, the small space between the edges of the wavy ring and duct walls is produced. The gap spacing is represented with “g”. The ratio between the gap spacing and the channel height, g/H or GR, is called gap spacing ratio. In the present research, the gap spacing ratio is varied in the range around 0 – 0.30. The laminar flow regime with the Reynolds number (at the inlet condition) around 100 – 2000 is focused. The computational domain with the wavy ring at GR = 0 (no gap spacing) is presented as Fig. 3a, while the model of the wavy ring with gap spacing is depicted as Fig. 3b. The studied cases can conclude as Table 1.

![Wavy ring](image1)
![Square duct](image2)

(a)

![Wavy ring](image3)
![Square duct](image4)

(b)

**Fig. 3** Computational domain of the square channel heat exchanger inserted with wavy ring for (a) no gap spacing and (b) with gap spacing.

**Table 1** Studied cases for the wavy ring in the square channel.

<table>
<thead>
<tr>
<th>BR</th>
<th>GR</th>
<th>Re</th>
</tr>
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<tbody>
<tr>
<td>0.05</td>
<td>0 – 0.30</td>
<td>100 - 2000</td>
</tr>
<tr>
<td>0.10</td>
<td>0 – 0.30</td>
<td>100 - 2000</td>
</tr>
<tr>
<td>0.15</td>
<td>0 – 0.30</td>
<td>100 - 2000</td>
</tr>
<tr>
<td>0.20</td>
<td>0 – 0.25</td>
<td>100 - 2000</td>
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<td>0.25</td>
<td>0 – 0.20</td>
<td>100 - 2000</td>
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</table>
2.1 Assumption, boundary and initial conditions
The computational domain of the square channel heat exchanger inserted with wavy ring is developed under following assumption.
- Flow is laminar and incompressible.
- Flow and heat transfer are steady in three dimensions.
- The convective heat transfer is considered, while the radiation heat transfer is ignored.
- Body force and viscous dissipation are disregarded.
- The tested fluid is air (Pr = 0.707) with constant thermal properties at the average bulk mean temperature.

The periodic boundary is applied for inlet and outlet of the computational domain. The concept of the periodic module and related equations are described as Ref. [26]. The uniform temperature around 310K is set for all sides of the square channel walls. The wavy ring is assumed to be an insulator or heat flux around 0 W/m². The air at the inlet condition is around 300K. The Reynolds number is calculated from the inlet condition in all cases.

2.2 Mathematical foundation and numerical method
The mathematical foundation and numerical method for the heat exchanger square channel heat exchanger are referred from Ref. [27]. The numerical problem is solved by the commercial code (finite volume method with SIMPLE algorithm).

The dimensionless variables for the present work are Reynolds number, friction factor, Nusselt number and thermal enhancement factor. The air velocity is reported in term of the Reynolds number, which can be written as Eq. 1.

\[ Re = \frac{\rho u D}{\mu} \]  

(1)

The pressure loss in the heating section is presented in terms of the friction factor value. Eq. 2 shows the calculations of the friction factor across the heating section.

\[ f = \left( \frac{\Delta P / L}{D} \right) \frac{1}{\rho u^2} \]  

(2)

The heat transfer rate is reported with the local Nusselt number (Nu) and average Nusselt number (Nu). The calculations of the local Nusselt number and the average Nusselt number are illustrated as Eqs. 3 and 4, respectively.

\[ Nu_x = \frac{h D}{k} \]  

(3)

\[ Nu = \frac{1}{L} \int Nu_x dx \]  

(4)

The thermal performance in the tested section is presented with the thermal enhancement factor or TEF. The TEF can determine as Eq. 5.

\[ TEF = \left. \frac{h}{h_{pp}} \right| = \frac{Nu}{Nu_{pp}} \right| = \left( \frac{Nu}{Nu_{pp}} \right) \left( \frac{f / f_o}{1.5} \right)^{1/3} \]  

(5)

3. Numerical validation
The validation of the numerical model is necessary for the simulation research. The validation result is an important value to ensure that the creation model has enough reliability to predict flow configuration and heat transfer behavior in the tested section. The numerical model for the square channel heat exchanger inserted with wavy ring is validated in all cases. The validation of the model can separate into two sections; validation of the smooth duct and grid independence.

The values of the Nusselt number and friction factor for the present smooth model are compared with the values from the correlations [28]. The deviations around ±0.7% and ±1.1% are found for the Nusselt number and friction factor.

The hexahedral mesh is generated for the numerical domain. The four numbers of grid cells; 80000, 120000, 180000, 220000, are compared. As the numerical result, the Nusselt number and friction loss are closely found when increasing the grid cell from 120000 to 180000. Therefore, the grid around 120000 cells is selected for all tested cases of the present research. The optimum grid number may help to save time for investigation and also gives high accuracy results.
4. Result and discussion
The numerical results for the heat exchanger duct inserted with the wavy ring can be concluded into two sections; 1. flow and heat transfer configurations and 2. performance analysis. The flow and heat transfer configuration part will report the mechanisms in the tested duct. The heat transfer rate, pressure loss and thermal performance in terms of Nusselt number ratio ($\frac{Nu}{Nu_0}$), friction factor ratio ($\frac{f}{f_0}$) and thermal enhancement factor (TEF), respectively, will conclude in the performance analysis part.

4.1 Flow and heat transfer configurations
Figs. 4, 5 and 6 present the streamline in cross-sectional planes in the square duct inserted with wavy ring at BR = 0.05, 0.10 and 0.20, respectively, with various gap spacing ratios. For BR = 0.05 and GR = 0, the wavy ring can produce the vortex flow through the test section. The configuration of the counter-rotating flow includes the four main vortex flow and small vortices near the corners of the channel for some plane. Considering at the lower pair of the vortex flow, the direction of the counter-rotating flow is common-flow-down.

![Streamline in transverse planes of the square duct inserted with wavy ring at BR = 0.05 and Re = 800 for (a) GR = 0, (b) GR = 0.10 and (c) GR = 0.20.](image)

For BR = 0.05 and GR = 0.10, the vortex flow is also produced by the wavy ring through the test channel. The four main vortex cores are detected in all planes. Seeing 2nd and 3rd plane in the Fig. 4b, the generation of the vortex flow near the channel walls is found. This is because some part of the air passes the gap spacing. The creation of the new vortex flow (around 12 cores) is clearly detected when GR = 0.20 as the Fig. 4c.

Considering at BR = 0.10 and GR = 0, the counter-rotating flow with common-flow-down is found in all cross-sectional planes. The four core flows are detected at the first and last planes. At the 2nd – 4th plane, the eight cores of the vortex flow are formed. When GR > 0, the small vortices near the channel walls around 2 – 3 cores are clearly detected due to some part of the tested fluid flows through the gap.

For BR = 0.20 and GR = 0, the wavy ring creates eight vortex cores (equally size) through the square channel. When GR = 0.10, the size of the vortex core has changed. The vortex core can separate into two groups; main vortex flow and small vortices. For GR = 0.20, the wavy ring performs the four main vortex flows (the counter-rotating flow with common-flow-up).
Fig. 5 Streamline in transverse planes of the square duct inserted with wavy ring at BR = 0.10 and Re = 800 for (a) GR = 0, (b) GR = 0.10 and (c) GR = 0.20.

Fig. 6 Streamline in transverse planes of the square duct inserted with wavy ring at BR = 0.20 and Re = 800 for (a) GR = 0, (b) GR = 0.10 and (c) GR = 0.20.
The vortex flow, which is generated by the wavy ring, helps a better fluid temperature mixing between cold fluid at the center of the channel and hot fluid near the channel walls. The vortex flow also disturbs the thermal boundary layer on the heat transfer surface. The change of the thermal boundary layer effects for the increments of the heat transfer rate and thermal performance. The heat transfer rate and thermal performance increase when increasing the strength of the vortex flow.

The present of the gap spacing in the heat exchanger channel is an important factor for the change of the flow structure. The value of the gap spacing can produce the new core of the flow and small vortices near the channel wall. The augmentation on the number of the core flow can help to distribute the fluid temperature in the channel heat exchanger. The change of the flow pattern may decrease or increase the vortex strength. Moreover, the variation of the flow topology in the channel, when changed the gap spacing value, may effect for the change of the heat transfer region on the channel wall due to the change of the thermal boundary layer.

The longitudinal vortex flow in the square duct inserted with wavy ring is depicted as Figs. 7a and b, respectively, for GR = 0 and 0.15. It is clearly seen in the figures that the gap spacing has directly effect for the flow configuration. Some part of the tested fluid flows passes the gap near the channel walls. Therefore, the strength of the flow may increase or decrease depended on the gap spacing value. The heat transfer behavior may also change.

**Fig. 7** Longitudinal vortex flow in the square duct inserted with wavy ring for (a) no gap spacing and (b) with gap spacing.
Fig. 8 Fluid temperature in transverse planes of the square duct inserted with wavy ring at BR = 0.05 and Re = 800 for (a) GR = 0, (b) GR = 0.10 and (c) GR = 0.20.

Fig. 9 Fluid temperature in transverse planes of the square duct inserted with wavy ring at BR = 0.10 and Re = 800 for (a) GR = 0, (b) GR = 0.10 and (c) GR = 0.20.
Fig. 10 Fluid temperature in transverse planes of the square duct inserted with wavy ring at BR = 0.20 and Re = 800 for (a) GR = 0, (b) GR = 0.10 and (c) GR = 0.20.

Figs. 8, 9 and 10 plot the distribution of the fluid temperature in cross-sectional plane in the heat exchanger square channel at BR = 0.05, 0.10 and 0.20, respectively, at various gap spacing values. The plot of the temperature in transverse plane is an indicator to observe the change of the thermal boundary layer on the heat transfer surface. In general, the low temperature fluid with blue contour is found at the center of the channel, while the high temperature fluid with red contour is detected near the channel walls. The change of the thermal boundary layer or heat transfer behavior is found when inserting the wavy ring in the heating section. The better fluid mixing is clearly found. The red layer near the channel wall is found to be thinner, while the blue contour distributes from the center of the channel. The vortex strength increases when augmenting the BR value, therefore, the heat transfer rate enhances when increasing the height of the wavy ring. Some gap value can increase vortex strength, therefore, also increases the heat transfer rate, but, some case of the gap value may produce the reverse trend.

Figs. 11, 12, and 13 report the variation of the heat transfer rate in term of local Nusselt number on the channel walls of the heat exchanger square channel placed with wavy ring at BR = 0.05, 0.10 and 0.20, respectively, with various gap values. The wavy ring with zero gap value gives the peak of heat transfer rate at the upper and lower walls of the test section in all BRs. When GR increasing, the heat transfer rate at the left and right sidewalls increases, while the heat transfer rate at the upper and lower walls performs in the opposite result.

4.2 Performance analysis

Figs. 14a, b, c, d and e report the variations of the Nu/Nu₀ with the Reynolds number in the square channel fitted with the wavy ring at various gap values for BR = 0.05, 0.10, 0.15, 0.20 and 0.25, respectively. The heat transfer rate tends to increase when augmenting the Reynolds number in all tests. The placement of the wavy ring in the heating section can enhance the heat transfer rate higher than the smooth channel (Nu/Nu₀ > 1). For BR = 0.05, the Nu/Nu₀ is around 1.00 – 6.24, 1.00 – 4.51, 1.00 – 3.49, 1.00 – 3.39, 1.00 – 3.75, 1.04 – 3.74 and 1.16 – 3.38, respectively, for GR = 0, 0.05, 0.10, 0.15, 0.20, 0.25 and 0.30. Considering at BR = 0.10, the heat transfer rate is higher than the smooth square duct around 1.00 – 8.56, 1.00 – 5.18, 1.00 – 4.43, 1.00 – 5.53, 1.11 – 5.50, 1.29 – 5.16 and 1.44 – 5.29, respectively, for the gap spacing ratios of 0, 5, 10, 15, 20, 25 and 30%. The BR = 0.15 provides heat transfer rate around 1.00 – 6.50, 1.01 – 5.93, 1.01 – 6.63, 1.20 – 6.91, 1.45 – 6.51, 1.67 – 5.97 and 1.73 – 5.50 times over the smooth duct with no wavy ring for GR = 0, 0.05, 0.10, 0.15, 0.20, 0.25 and 0.30, respectively. Seeing at BR = 0.20, the Nusselt number is around 1.23 – 5.42, 1.21 – 4.68, 1.34 – 8.51, 1.64 – 7.96, 1.93 – 7.16 and 2.01 – 5.51 times above the smooth section for gap spacing around 0, 5, 10, 15, 20 and 25%. At BR = 0.25, the Nusselt number ratio is found to be around 1.57 – 15.06, 1.64 – 12.57, 1.88 – 10.30, 1.99 – 8.72 and 2.35 – 7.15, respectively, for GR = 0, 0.05, 0.10, 0.15 and 0.20.
Fig. 11 Local Nusselt number distribution on the duct walls of the square duct inserted with wavy ring at BR = 0.05 and Re = 800 for (a) GR = 0, (b) GR = 0.05, (c) GR = 0.10, (d) GR = 0.15, (e) GR = 0.20, (f) GR = 0.25 and (g) GR = 0.30.

Fig. 12 Local Nusselt number distribution on the duct walls of the square duct inserted with wavy ring at BR = 0.10 and Re = 800 for (a) GR = 0, (b) GR = 0.05, (c) GR = 0.10, (d) GR = 0.15, (e) GR = 0.20, (f) GR = 0.25 and (g) GR = 0.30.
Fig. 13 Local Nusselt number distribution on the duct walls of the square duct inserted with wavy ring at BR = 0.20 and Re = 800 for (a) GR = 0, (b) GR = 0.05, (c) GR = 0.10, (d) GR = 0.15, (e) GR = 0.20 and (f) GR = 0.25.

Fig. 14 Nu/Nu₀ vs Re for (a) BR = 0.05, (b) BR = 0.10, (c) BR = 0.15, (d) BR = 0.20 and (d) BR = 0.25.
Figs. 15 $f/f_0$ vs Re for (a) BR = 0.05, (b) BR = 0.10, (c) BR = 0.15, (d) BR = 0.20 and (d) BR = 0.25.

Figs. 15a, b, c, d and e illustrate the relations of the $f/f_0$ with the Reynolds number for the BR = 0.05, 0.10, 0.15, 0.20 and 0.25, respectively, of the wavy ring in the square channel heat exchanger. Generally, the friction loss enhances when augmenting the air flow rate in all studied tests. The installation of the wavy ring in the test section provides higher pressure loss over than the smooth duct with no wavy ring ($f/f_0 > 1$). At BR = 0.05, the $f/f_0$ is around 1.02 – 6.68, 1.20 – 6.66, 1.46 – 6.78, 1.78 – 7.22, 2.12 – 7.31, 2.39 – 6.95 and 2.50 – 6.78 for the gap values of 0, 5, 10, 15, 20, 25 and 30%, respectively. Considering at BR = 0.10, the pressure loss is upper than the base case around 1.25 – 16.62, 1.71 – 15.50, 2.23 – 16.12, 2.88 – 16.66, 3.48 – 15.82, 3.71 – 14.45 and 3.54 – 14.01 times, respectively, for GR = 0, 0.05, 0.10, 0.15, 0.20, 0.25 and 0.30. The BR = 0.15 wavy ring gives the friction loss around 1.93 – 30.57, 2.93 – 29.33, 3.79 – 31.14, 4.83 – 30.57, 5.33 – 26.32, 5.08 – 23.37 and 4.48 – 25.68 times over than the smooth duct for GR = 0, 0.05, 0.10, 0.15, 0.20, 0.25 and 0.30, respectively. The pressure loss of the BR = 0.20 wavy ring is found to be about 3.74 – 33.44, 5.55 – 33.37, 6.87 – 58.66, 7.75 – 46.35, 7.46 – 38.52 and 6.56 – 33.90 times higher than the smooth square duct, respectively, for GR = 0, 0.05, 0.10, 0.15, 0.20 and 0.25, respectively. Seeing at BR = 0.25, the friction factor is around 8.04 – 189.54, 11.12 – 116.85, 11.79 – 94.16, 11.32 – 67.02, 9.95 – 44.69 times upper the base case at GR = 0, 0.05, 0.10, 0.15 and 0.20, respectively.
Figs. 16a, b, c, d and e depict the relations of the TEF with the Reynolds number for the square section inserted with the BR = 0.05, 0.10, 0.15, 0.20 and 0.25 wavy rings, respectively. Almost cases, the TEF slightly increases when raising the Reynolds number. The TEF is higher than the unity. The optimum TEF is found at GR = 0 for BR = 0.05, 0.10 and 0.25, while found at GR = 0.15 for BR = 0.15 and 0.20. The maximum values of the TEF is around 3.50, 3.35, 2.24, 2.22 and 2.62 for BR = 0.05, 0.10, 0.15, 0.20 and 0.25, respectively.

6. Conclusion
The numerical analysis on heat transfer behavior, flow topology and thermal performance enhancement in the square duct heat exchanger fitted with wavy ring at various blockage ratio and gap spacing ratio are performed. The numerical simulation is considered in the rang of laminar flow regime with the Reynolds number around 100 – 2000. The outcomes of the present investigations can conclude as follows:
The wavy ring can produce in vortex flow, swirling flow and longitudinal flow through the test section in all tested cases. The vortex flow disturbs the thermal boundary layer on the heat transfer surface that the cause for heat transfer rate and thermal performance augmentations. The vortex flow also assistance a better fluid-temperature mixing between the core of the channel and near the channel walls. However, the present of the wavy ring in the heating section brings a higher pressure loss over than the smooth duct.

Effect of the Reynolds number, the increment of the Reynolds number leads to higher heat transfer rate, friction loss and thermal performance due to the enhancement of the vortex strength.

Influence of the wavy ring height, the heat transfer rate and pressure loss increase when augmenting the wavy ring height due to the increment of the vortex strength. The \( \text{BR} = 0.25 \) performs the peakest on both heat transfer rate and pressure loss, while the \( \text{BR} = 0.05 \) gives the reverse trend.

Effect of gap spacing, the gap spacing in the test section has effect for the change on both heat transfer behavior and flow pattern. The present of gap can reduce or increase the heat transfer rate and pressure loss in the heating system. The optimum of gap spacing for each blockage ratio and Reynolds number may bring higher heat transfer rate and thermal performance and also reduces the pressure loss in the system.

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NOMENCLATURE

- **BR**: flow blockage ratio (=b/H)
- **b**: wavy ring height, m
- **D_h**: hydraulic diameter of channel, m
- **f**: friction factor
- **GR**: gap spacing ratio, (=g/H)
- **g**: gap spacing, m
- **H**: square duct height, m
- **h**: convective heat transfer coefficient, W m\(^{-2}\) K\(^{-1}\)
- **k**: thermal conductivity, W m\(^{-1}\) K\(^{-1}\)
- **Nu**: Nusselt number (=hD_h/k)
- **p**: static pressure, Pa
- **Re**: Reynolds number
- **T**: temperature, K
- **\( \bar{u} \)**: mean velocity in channel, m s\(^{-1}\)

Greek letter

- \( \alpha \): flow attack angle, degree
- \( \text{TEF} \): thermal enhancement factor (=\(\frac{\text{Nu}}{\text{Nu}_0}/(f/f_0)^{1/3}\))
- \( \rho \): density, kg m\(^{-3}\)

Subscript

- **0**: smooth square channel
- **pp**: pumping power

References


