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Alfarawi, Suliman; Al-Dadah, Raya; Mahmoud, Saad

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Potentiality of new miniature-channels Stirling regenerator

S. Alfarawi^{1,a}, R. AL-Dadah¹, S. Mahmoud¹

¹Department of Mechanical Engineering, University of Birmingham, Edgbaston, B15 2TT, UK

^ae-mail: ssa178@bham.ac.uk

Abstract

This paper deals with the development of new Stirling regenerator as one of parallel-geometry regenerators. Circular miniature-channels with different diameters, 1.5, 1.0, 0.6, 0.5, 0.4 (mm) were adopted for the current design. A 3D CFD approach, based on transient conjugate heat transfer was performed on a regenerator sector to obtain the fluid flow and heat transfer characteristics of each configuration. The obtained data was converted into an equivalent porous media to be utilized in a full CFD model of the engine developed by the author in other study. The results revealed that the 0.5mm channel regenerator had good potential close to random fibre regenerator for maximizing engine indicated power. However, engine performance was adversely affected by the excessive heat rejected from the cooler due to the inherit axial conduction loss of channel regenerators compared to conventional regenerator. The results also showed that the proper selection of matrix material with high volumetric heat capacity and low thermal conductivity can alleviate these losses and reduce the heat rejected from the engine. This study suggested further experimental work to investigate the effect of channel regenerator segmentations to minimize the axial conduction losses and improve the regenerator performance.

Keywords: Circular-shaped, Stirling regenerator, miniature-channels.

Nomenclature

A_f	Internal flow area (m ²)	Re	Reynolds number
C_p	Gas heat capacity at constant pressure (J/kg. K)	r_i	Sector inner radius (m)
D	Pipe diameter (m)	r_o	Sector outer radius (m)
D_h	Regenerator hydraulic diameter (m)	S	Fabrication clearance (m)
d_{ch}	Channel diameter (m)	T	Fluid temperature (K)
f	Darcy friction factor	T_m	Mean bulk fluid temperature (K)
h_x	Local heat transfer coefficient (W/m ² . K)	T_w	Wall temperature (K)
K	Permeability (m ²)	u	Fluid velocity (m/s)
k_f	Fluid thermal conductivity (W/m. K)	VSWE	Expansion swept volume (m ³)
k_s	Solid thermal conductivity (W/m. K)	Greek letters	
Nu	Average Nusselt number	β_F	Forchheimer drag coefficient, kg/m ⁴
P_r	Radial pitch (m)	δ	Thermal penetration depth (m)
P_θ	Angular pitch (°)	ρ	Fluid density, kg/m ³
p	Fluid pressure (Pa)	μ	Fluid dynamic viscosity, Pa. s
$''q_x$	Local heat flux (W/m ²)	ε	Porosity

24 1. Introduction

25 The Stirling engine is an externally-heated engine, it is thermally regenerative, simple in construction, virtually
26 quiet, safe in operation, and intrinsically flexible to adopt any heat source such as solar, biomass, geothermal energy
27 or even an industrial waste [1]. However, its low specific power compared to internal combustions engines (ICE)
28 and the highly initial cost of the engine may still hinder further development and optimization of Stirling machines.
29 The successful development of an efficient and cost effective Stirling engine will have a significant impact on the
30 recovery of the available waste heat sources leading to significant reduction in fossil fuel consumption and CO2
31 emissions.

32

33 The regenerator is a key component of the engine, it is an internal heat exchanger that acts as a thermal sponge when
34 absorbing and releasing heat at a portion of the cycle, thus, engine power and efficiency are enhanced. The heat
35 being absorbed and restored to the gas in the regenerator during one cycle is typically four times the heat that passes
36 through the heater during one cycle [2]. For an engine without a regenerator, its heater would need to take in five
37 times as much heat during a cycle as that with a regenerator to generate the same power.

38

39 Stirling engines mostly use wire mesh (woven or random fibre) as a conventional regenerator due to its high
40 convective heat transfer between the solid and the gas resulting from the extended surface area of wires (this is
41 similar to a cross flow over repeated cylinder-shaped wires), and low axial conduction in flow direction. However,
42 the downside of this type of regenerators is the high flow friction resulting from flow separation, eddies associated
43 with stagnation areas that can degrade the engine performance [3]. The regenerator has to have several features for
44 better performance that might be contradicting and this pose a challenge for developers to find the optimum
45 configuration that least fulfils the requirement of maximum convective heat transfer, minimum pressure drop, and
46 minimum axial conduction in flow direction [4, 5].

47

48 Theoretically, a regenerator, with its heat transfer surfaces parallel to the oscillatory flow, has better performance
49 than the conventional mesh regenerator type [6]. This type of regenerator has some interesting features in spite of
50 the high heat transfer such as; (1) small frictional losses due to the minimum flow separation; (2) low dead volume

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51 which means that they can be fabricated as compact as possible. However, the intrinsic axial conduction losses are
52 large due to the continuity of solid material and this crucial disadvantage can degrade its performance.

53

54 The regenerator performance is often evaluated experimentally on a real engine test-stand or on a tailor-made test-
55 stand based on unidirectional flow or oscillatory flow. However, in recent years, analytical, numerical models and
56 the sophisticated CFD packages for Stirling engine analysis have emerged to shift from the tedious and expensive
57 experimental work. In their hierarchal order, they are classified as zeroth, first, second, third and fourth order models
58 [7]. These models are ascending in their complexity and accuracy however, the effects caused by the geometrical
59 variation can only be handled by the adoption of fourth-order analysis or namely computational fluid dynamics
60 (CFD). These models can be used to judge the regenerator performance through a full simulation of the engine or
61 the regenerator alone.

62

63 There have been several studies on the conventional Stirling engine regenerators in literature. Martaj et al [8]
64 presented an energetic, entropic and exegeric analysis of LTD gamma-type Stirling engine based on steady state
65 operation. Using zero-dimensional numerical model, the engine was split into three isothermal control volumes,
66 including the heater, cooler and regenerator. Energy, mass, entropy and exergy balance was carried out on each cell
67 based as a function of crank angle (kinematics-thermodynamic coupling). Their results showed that it was possible
68 to optimize engine components based on best efficiency and minimum production of entropy and that the
69 regenerator's dead volume had strong impact on thermal and exergy efficiencies for the whole engine. Shendage et
70 al [9] investigated the effect of geometrical dimensions of a regenerator on beta-type Stirling engine performance.
71 They used a thermodynamic analysis based on second-order simulation to evaluate engine performance. Different
72 losses were accounted for in the analysis including reheat, shuttle conduction, pumping and heat exchangers
73 effectiveness. The number of wire sheets in the regenerator were varied depending on the pressure drop, dead
74 volume and the thermal penetration depth. They proposed a length of the regenerator of 22 mm that balanced the
75 conflicting increased heat transfer and pressure drop due to the increase in number of sheets which gave a
76 regenerator an optimum effectiveness of 0.965. Mahkamov [10] performed a second-order and 3D CFD analysis on
77 a gamma-type Stirling engine prototype to enhance its power. The CFD results revealed that power reduction was
78 attributed to the high level of hydraulic losses in the regenerator, and the entrapment of the gas in the pipe

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79 connecting the two parts of the compression space and to its large dead volume. A further improvement in the
80 engine design was only viable by adopting this multi-dimension approach within an acceptable range of accuracy,
81 18% when compared to experimental results. Torre [11] developed a 2D CFD model for 300 cm³ beta-type Stirling
82 engine simulation based on OpenFoam code where turbulence effects and porous media modelling were considered.
83 Parametric study was carried out to investigate the effects of charge pressure, heat input to the heater and
84 regenerator material thermal conductivity on engine power. It was found that increasing both charge pressure and
85 heat input increase engine power. Meanwhile, lower thermal conductivity material of the regenerator enhances the
86 engine performance due the reduced axial conduction in the regenerator. It was reported that when using helium
87 against air as working gas, no significant difference in power output was observed. Costa et al [12] developed a CFD
88 model based on finite volume method (FVM) to derive Nusselt number correlations of two types of regenerator
89 matrices; stacked and wound woven wires. The stacked woven wire correlation was first compared to an existing
90 experimental correlation to validate the numerical method. Then, the correlation of for wound woven wire is
91 proposed for investigated parameters of a diameter range from 0.08 to 0.11 mm and a porosity range from 0.60 to
92 0.68. Chen et al [13] constructed and tested a twin power piston gamma-type Stirling engine. The engine was
93 incorporated with a moving regenerator housed inside its displacer and filled with a woven-screen material. The
94 effects of different regenerator parameters on engine performance, including regenerator material, wire diameter,
95 filling factor and stacking arrangements, were investigated. According to their results, copper material was found
96 superior to stainless steel on engine performance at the tested conditions and optimum filling factor was proposed.
97 Gheith et al [14] conducted an experimental investigation on the optimum regenerator matrix material and porosity
98 for gamma-type Stirling engine. Different materials were tested including stainless steel, copper, aluminium and
99 Monel 400. The results showed that stainless steel matrix with 85% porosity is the best configuration to maximize
100 engine performance. Gheith [15] studied experimentally a new phenomenon associated with gamma-type Stirling
101 engine causing a thermal energy dissipation in the regenerator. They observed that a maximum temperature of 25 °C
102 between regenerator two sides can occur at heater temperature of 200 °C, the asymmetry of the engine and the one
103 inlet of the cooling circuit causes this phenomenon. They carried out multi objective optimization to find the
104 optimum operating values that minimizes this temperature difference in the regenerator and they found that heating
105 temperature of 350 °C, initial pressure of 8 bar and cooling water flow rate 8.1 L/min can minimize this temperature
106 difference to 17 °C.

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107 As micro-fabrication techniques have recently shown significant advances, regular-shaped miniature-channel
108 regenerator type can be a good choice as one of parallel-geometry regenerators. A few attempts were reported in
109 literature for the development of new conceptual parallel-geometry regenerators to have superior performance.
110 Ibrahim et al [16] developed a segmented-involute-foil regenerator to minimize the pressure drop effects of cylinder
111 cross-flow and to avoid the large axial conduction losses such as in the case of parallel plates and tube bundles.
112 Their test results showed that higher figure of merit was achieved closer to the ideal parallel-plate regenerators.

113

114 On the other hand, Li et al [17] proposed a porous-sheet regenerator with hexagonal-shaped flow channels using
115 dynamic mesh CFD simulation for whole engine. Each flow channel has a side length of 0.4mm and a length of
116 73mm. They found that, under the same working conditions, the pressure drop for porous-sheet regenerator is lower
117 compared to wire mesh. Optimizing both regenerators under given operating conditions showed that the porous-
118 sheet regenerator had 38%-51% lower entropy generation rate compared to wire mesh, thus contributing to an
119 increase in power and efficiency of the engine. However, experimental testing of porous-sheet regenerator was not
120 reported in his study.

121

122 Nam et al [18] developed a parallel wire type regenerator. The friction factor was found to be 20-30% lower than the
123 screen mesh regenerator, but the thermal performance of this new type was poor compared to the screen mesh type
124 due to large axial conduction of parallel wires. The axial conduction losses were further alleviated by the
125 segmentation of the continuous wires but the number of segmentation was limited due to the increased number of
126 housings required to hold the wires. Takizawa et al [19] developed a thin porous-sheet regenerator with small
127 rectangular-shaped flow holes. The effect of this type of regenerators and other two wire screens were tested on an
128 experimental engine stand (NS03T), the results showed that the engine power output was improved by 15%
129 compared to 200M wire sheet.

130

131 Isshiki et al [20] investigated experimentally the effect of layered-plate type regenerator on beta-type Stirling
132 engine. They compared the performance of the engine using this type of regenerator to the traditional wire-mesh
133 type. They found that the layered-type had a better performance on the engine compared to the wire-mesh. They
134 justified this to the reduced pressure drop of the layered-type as well as the increases heat transfer due to the

135 extended area of the layered plates. The engine brake power was achieved in the range of 22 to 91.4 W when heater
136 temperature was varied in the range of 180-330 °C. Kato et al [21] proposed a methodology to evaluate the
137 regenerator performance based on experiments. The experimental test facility was similar to that of a 180°-phase
138 angle alpha-type Stirling engine. An electric heater was used to control the hot end temperature while the cold end
139 temperature was controlled by a conditioned air. The LTD engine was tested with polyurethane foam and #18
140 stainless steel as regenerators. The regenerator effectiveness was calculated in terms of the measured temperature
141 fluctuation between the hot and cold ends. The results showed that regenerator effectiveness of the stainless steel
142 mesh layered parallel to gas flow direction was significantly less in comparison to that of the normal mesh layers
143 while a significant pressure loss was observed for the polyurethane foam when fitted in cylinder/displacer gap.

144
145 Based on open literature, there is limited research on micro parallel flow regenerators, therefore in this work, the
146 development of new Stirling engine regenerator based on parallel-geometry configuration using miniature circular
147 channels with diameters of 1.5, 1.0, 0.6, 0.5, 0.4 (mm) were investigated for gamma-type Stirling engine
148 (ST05CNC). A CFD model was developed on a 3D sector of the proposed geometry using transient conjugate heat
149 transfer to determine the fluid flow and heat transfer characteristics of the five configurations for a unidirectional
150 flow. The real geometry of regenerators then is converted into an equivalent porous media so that the engine
151 performance fitted with channels regenerator is evaluated using a full engine CFD model developed by the author in
152 other study [22].

153
154 **1. Design concept**

155 Fig.1 shows the conceptual design of miniature-channel regenerator which is scaled to the original regenerator dead
156 volume of the engine. The proposed geometry is composed of circular-shaped miniature-channel cutting through the
157 solid matrix with constant diameter. Different diameters are selected for the channels 1.5, 1.0, 0.6, 0.5, 0.4 (mm) so
158 that high surface area density (up to 2600 m²/m³) can be achieved, with channels length equals to the regenerator
159 length of 57 mm. The material selected for all samples is stainless steel (SS304), with its properties are listed in
160 **Table 1.**

161

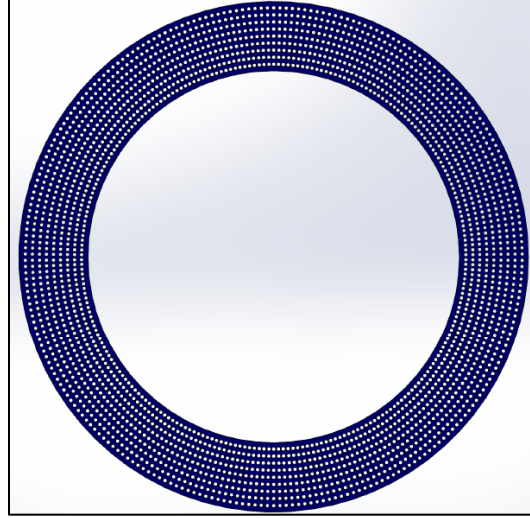


Figure 1: Conceptual design of miniature-channel regenerator

162

163

164 **Table 1**
165 Regenerator material thermal properties (SS304L)

Property	Value
Density (kg/m ³)	7850
Specific heat capacity (J/kg. K)	475
Thermal conductivity (W/m. K)	16

166

167

168 The circular channels are distributed with radial and angular pitches that satisfy the minimum required thermal
169 penetration depth, which is defined as the distance that heat can diffuse into the solid matrix during single blow
170 time, $t_p = (1/\pi f)$, where f is the frequency of the engine [9]. Since the engine under consideration can operate at
171 speed ranging from 100 to 1000 rpm, then the thermal penetration depth is calculated by Eq. 1 and the results are
172 presented in fig. 2;

173

$$\delta = \sqrt{(k_s/\pi f \rho C_p)} \quad (1)$$

174

175

176

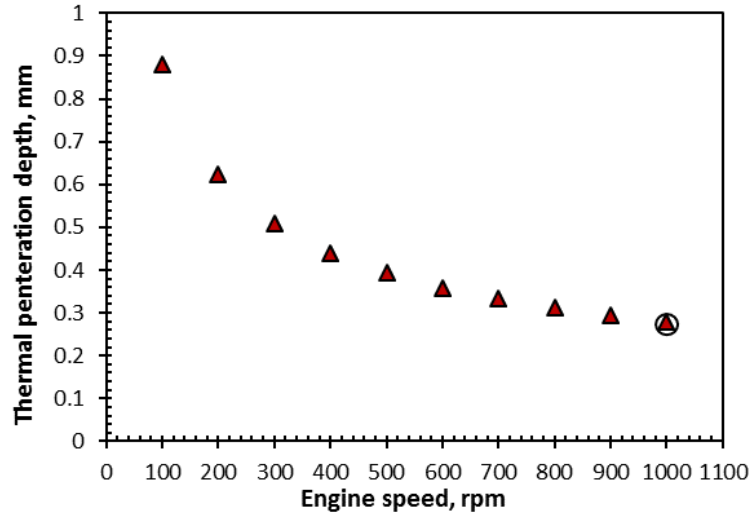


Figure 2: Thermal penetration depth vs. engine speed.

177

178 The radial and angular pitches are not fixed and can be calculated based on the minimum thermal penetration depth
 179 of 0.25 obtained at maximum engine speed as shown in fig.2. The geometrical parameters of the regenerator
 180 configuration as shown in fig.3 can be calculated from Eq.2 and 3, respectively.

181

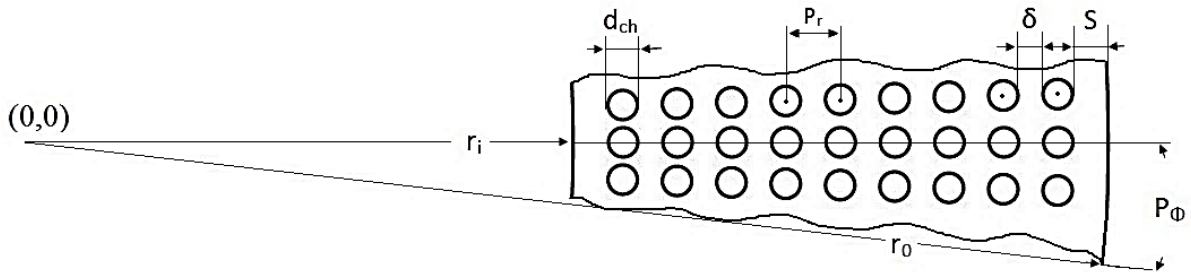


Figure 3: Geometrical parameters of channels regenerator.

182

$$P_r = \delta + d_{ch} \quad (2)$$

183

184 Where the radial pitch is measured from centre to centre of consecutive channels

185

186

$$P_{\phi} = 2 \sin^{-1}[(d_{ch}/2 + \delta/2)/(d_{ch}/2 + r_i + S)] \quad (3)$$

187

188 The angular pitch in (Eq.3) is based on the trigonometric relation exists between the three parameters (d_{ch} , δ and S)

189 at the first row of channels as demonstrated in fig.4.

190

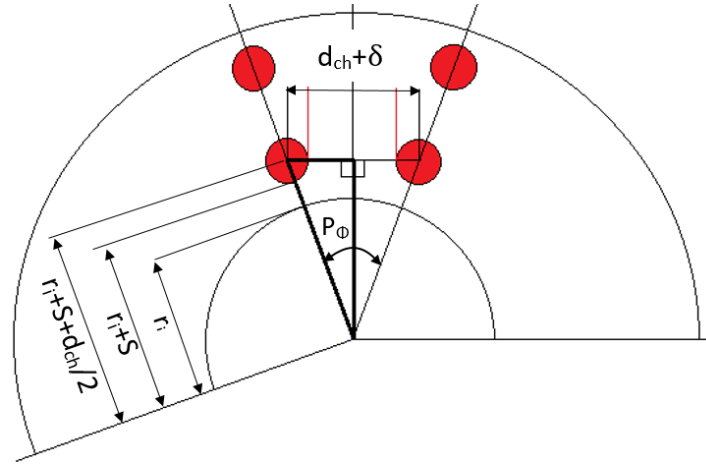


Figure 4: Radial pitch definition in terms of regenerator parameters (not to scale).

191

192

193 The geometrical parameters are calculated for channel diameter ranging from 0.4mm to 1.5 mm and fixed δ value

194 and tabulated in **Table 2**.

195

196 **Table 2**
197 Regenerators configurations and parameters.

Configuration	Angular pitch P_{ϕ} (°)	Vertical pitch P_r (mm)	Tubes/sector	Tubes/annulus	Porosity (%)
1.5mm	2	1.75	9	180	0.433
1.0mm	1.42	1.25	13	253	0.391
0.6mm	0.9	0.85	20	400	0.343
0.5mm	0.85	0.75	22	500	0.286
0.4mm	0.85	0.65	26	500	0.248

198

199

200

201 **2. CFD Model**

202 Since the regenerator is composed of miniature-channel of circular shape, internal fluid flow and heat transfer are
 203 the major physics. COMSOL Multiphysics 5.2 CFD commercial code was used for 3D transient conjugate heat
 204 transfer simulation of the regenerator. Prior to the procedure of computational modelling of the regenerator, the
 205 testing metrics of CFD computations is a prerequisite to evaluate the accuracy of the current results. Since the flow
 206 is developing inside the regenerator channels, a replication of hydrodynamically and thermally developing laminar
 207 tube flow was initially conducted.

208

209 **2.1 Fluid flow and heat transfer in channels**

210 The average friction factor and heat transfer coefficient for a whole tube are enhanced due to the effect of the
 211 entrance length [23]. This enhancement is significant for a shorter tube rather than a longer one as the skin shear
 212 stress and the heat transfer coefficient are large at the entrance of the tube where the boundary layer thickness is
 213 very small. In the developing flow regime, both the velocity and thermal boundary layers are growing in the flow
 214 direction where the velocity and temperature profile are locally varying up to a certain distance downstream of the
 215 tube. The flow is then said to be *fully developed*, and the *parabolic profile* is obtained for laminar flow in a circular
 216 tube. In the entrance region, the boundary layer of the developing flow needs to be captured by the computational
 217 grid for accurate solutions.

218

219 The hydrodynamic entry length, for laminar flow ($Re < 2300$), can be expressed in the form [24]

220

$$\left(\frac{x_{fd,h}}{D}\right) \approx 0.05Re \quad (4)$$

221

222 In the fully developed region, the pressure gradient is constant and the velocity profile is recalled from [24] as

$$u(r) = 2u_m \left[1 - \left(\frac{r}{R}\right)^2\right] \quad (5)$$

223

224 It follows that the ratio of the maximum and averages velocities in the fully developed region

225

$$u_{max} = 2u_m \quad (6)$$

226

227 According to Shah and London [25], The friction factor – Reynolds number product (fRe) for laminar flow is

228

$$fRe = 16 \quad (7)$$

229

230 If tube surface is maintained at either uniform temperature or uniform heat flux, conduction and convection heat
231 transfer occurs and the *boundary layer* begins to develop thermally until the fully developed temperature profile is
232 reached.

233

234 The *thermal entry length* for laminar flow can be expressed as

235

$$\left(\frac{x_{fd,t}}{D}\right) \approx 0.05ReP_r \quad (8)$$

236

237 When a tube flow is considered, heat transfer occurs by conduction when the fluid layers are stationary near the wall
238 and by forced convection when the fluid flowing through the tube. The ratio of the convective heat flux to the
239 conductive heat flux is referred to, Nusselt number (Nu) and it is defined as

240

$$Nu = \frac{h_x D_h}{k_f} \quad (9)$$

241

242

243 In a horizontal tube, under constant wall temperature boundary condition, the thermal characteristics of the flow is
244 defined by the fluid bulk temperature, heat transfer coefficient, heat flux, and the Nusselt number. The fluid bulk
245 temperature, is obtained in terms of the true energy advection over an arbitrary cross section of the pipe.

246

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247 Therefore, T_m is defined by

$$T_{m,x} = \frac{\int \rho u C_p T dA_c}{m C_p} \quad (10)$$

248

249

250 The other important parameter is the heat transfer coefficient which is defined by

251

$$h_x = \frac{q''_x}{(T_w - T_{m,x})} \quad (11)$$

252

253 Where

$$q''_x = k_f \left[\left(\frac{\partial T}{\partial r} \right)_{w,m} \right]_x \quad (12)$$

254

255 Where $\left(\frac{\partial T}{\partial r} \right)_{w,m}$ represents the peripheral average temperature gradient at wall.

256

257 As a special case for a fully developed laminar flow in case of constant wall temperature in a circular tube, *the rate*
258 *of net energy transfer to the control volume by mass flow is equal to the net rate of heat conduction in the radial*
259 *direction*, leads to the common value of Nusselt number,

260

$$Nu = 3.66 \quad (13)$$

261

262 **2.2 Governing equations and solution methodology**

263 For steady-state incompressible viscous flow, the Navier-Stokes equations which describe the velocity, pressure,
264 temperature, and density of a moving fluid based on conservation of mass, momentum and energy, for the
265 computational domain can be written as;

$$\rho \nabla \cdot (\mathbf{u}) = 0 \quad (14)$$

266

$$\rho(\mathbf{u} \cdot \nabla)\mathbf{u} = \nabla \cdot [-p\mathbf{I} + \mu (\nabla\mathbf{u} + (\nabla\mathbf{u})^T)] + \mathbf{F} \quad (15)$$

267

$$\rho C_p \mathbf{u} \cdot \nabla T + \nabla \cdot k \nabla T = Q \quad (16)$$

268

269 These coupled differential equations are solved by finite element-based CFD software (Comsol multiphysics). In
 270 order to carry out the simulation, a prerequisite of mesh sensitivity analysis is initialized to ensure that the numerical
 271 solutions obtained are grid-independent. Using symmetry can provide faster solution in which a quarter of the
 272 current 3D tube is selected as the computational domain as shown in fig.5. The tube has (1 in) diameter and (12 in)
 273 length which is subdivided into six sections in order to calculate the developing Nusselt number downstream of the
 274 inlet.

275

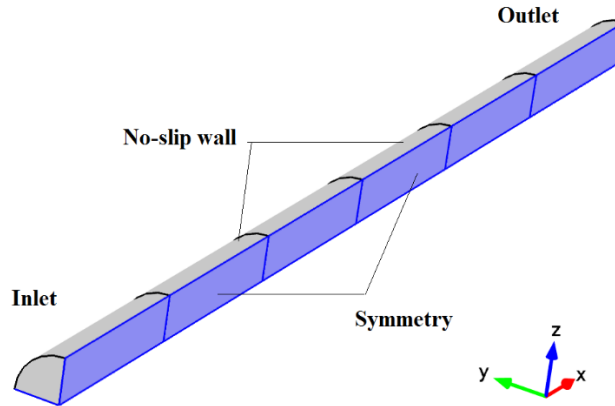


Figure 5: Computational domain of the tube.

276

277 The boundary conditions are applied as follows;

- 278 ▪ Constant wall temperature of 310 K is applied on the outer walls of the pipe.
- 279 ▪ Uniform inlet velocity (based on Re =100) at constant temperature of 300 K is applied at the tube inlet.
- 280 ▪ Outlet pressure of 101,325 kPa with no back flow is applied at the tube outlet.
- 281 ▪ Free-slip symmetry walls with zero normal velocity components and normal gradients of all velocity
- 282 components.

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283 The meshing sequence is varied from coarse to extra-fine size as shown in fig.6. This will define mesh size that
284 gives an accurate solution with less computational time required to perform the CFD simulation. The face elements
285 then are swept over the length of the tube based on each sequence. **Table 3** summarises the details of each meshing
286 sequence and the obtained solution of Nusselt number.

287

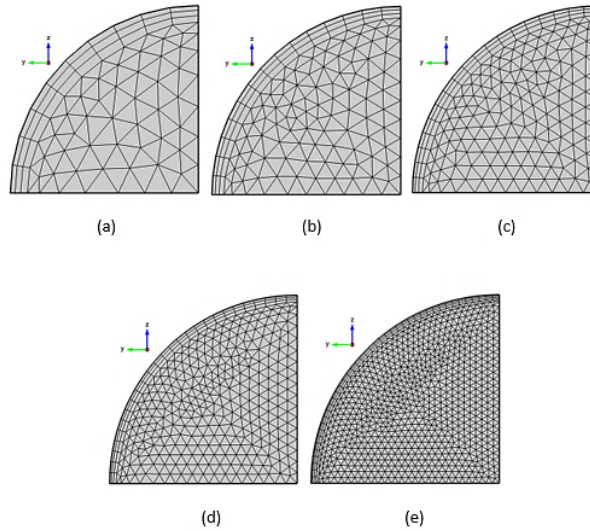


Figure 6: Meshing sequence, a) coarse, b) normal, c) fine, d) finer, e) extra-fine.

288

289

290 **Table 3**

291 Mesh sequence and CFD results

Mesh sequence	No of elements	Mesh quality	Growth rate	Nu
Coarse	19908	0.6558	1.344	6.2728
Normal	57354	0.7324	1.317	6.1937
Fine	103840	0.7747	1.269	6.1387
Finer	283920	0.8139	1.229	6.0173
Extra-fine	1,116,220	0.8614	1.189	5.9435

292

293

294 From the data obtained in **Table 3**, finer meshing sequence can be selected for the investigation as the change of
295 Nusselt number is less than 1.3% compared to the extra-fine sequence. The velocity and temperature contours are
296 plotted in fig. 7 and 8. It is clearly shown that both hydrodynamic and thermal boundary layers are developing
297 downstream of the tube inlet. A first check to carry out was to calculate the ratio of the maximum velocity

298 magnitude to the average velocity magnitude at the end of the pipe and the product of Reynolds number with friction
 299 factor. It is recalled that from the analytical solutions available in literature, Eq.6 that this ratio is double since the
 300 pressure gradient is constant in the fully developed region. Since the entry length required for this tube is calculated
 301 from Eq.7 equals 12.7 cm which means that the flow should be hydrodynamically fully developed at the end of the
 302 pipe and the product of Reynolds number with friction factor equals to 16. The comparison made in **Table 4** has
 303 shown good agreement with the available analytical solution.

304

305

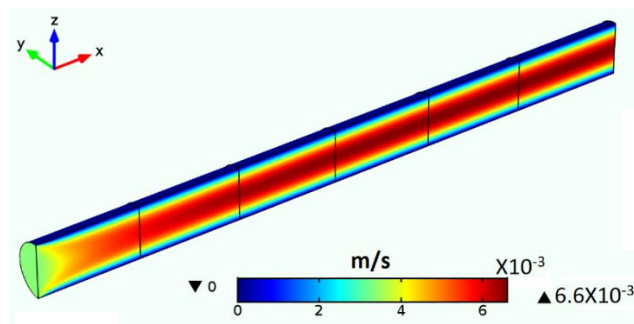


Figure 7: Velocity contours along the tube.

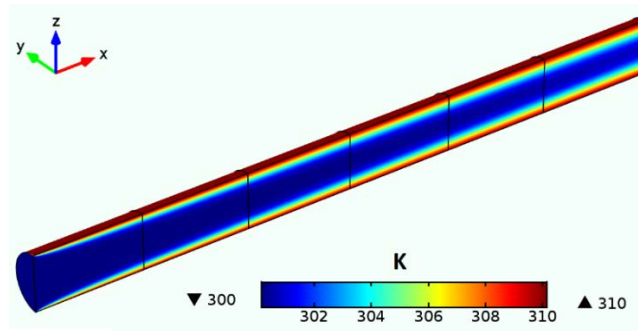


Figure 8: Temperature contours along the tube.

306

307 **Table 4**
 308 Comparison between CFD results and analytical data for fully developed tube flow.

Parameter	CFD	Analytical solution [24]
u_{\max}/u_{mean}	1.97	2
fRe	17.4	16

309

310 In order to validate the CFD results of the obtained Nu number in the case of developing flow, [26] was
 311 selected for the comparison which is recommended for a combined entry length under constant wall boundary
 312 condition. The results of Nusselt number was compared to Hausen's correlation in fig.9. As shown, both
 313 results are reasonably matched with a deviation in the range of 5-11%.

314

$$\overline{Nu_D} = 3.66 + \left[\frac{0.0668 G_{ZD}}{1 + 0.04 G_{ZD}^{\frac{2}{3}}} \right] \quad (17)$$

315

316 Where G_{ZD} is a dimensionless parameter defined as,

317

318

$$G_{ZD} = \left(\frac{D}{x} \right) \cdot Re \cdot Pr \quad (18)$$

319

320

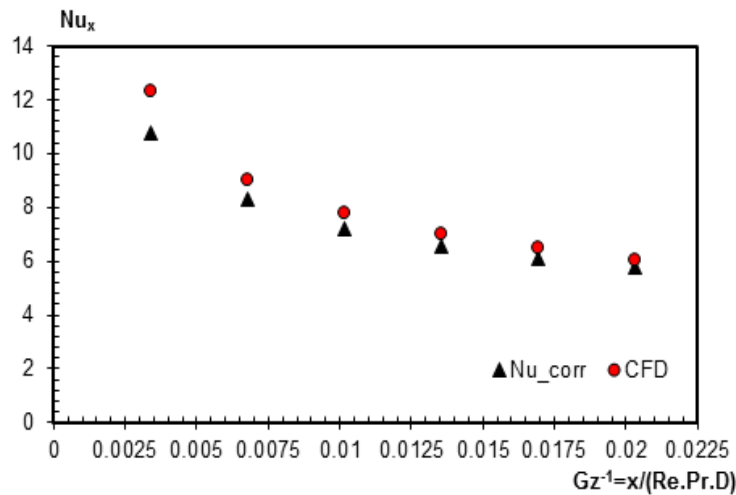


Figure 9: Nusselt number for laminar developing pipe flow.

321

322

3. Regenerator modelling

323 From a microscopic point of view, it is not feasible to model the regenerator as a whole as depicted in fig.1.

324 Therefore, more simplification is realized when a certain sector is repeated in the angular-direction with a certain

325 pitch (P_θ), it resembles the flow through regenerator annulus. The depth of the sector is taken as whole regenerator

326 length. Inlet velocity at constant temperature of 650 °C is specified at channels inlet, while pressure outlet at 10 bar

327 with outflow boundary is prescribed at the outlet. These boundary conditions are similar to the actual operating
328 conditions of the engine during single hot blow period. The maximum inlet velocity to the sector channels, is varied
329 according to the engine frequency and displacer swept volume, and is calculated by;

330

$$v_{max} = \pi \cdot f \cdot VSWE / A_f \quad (19)$$

331

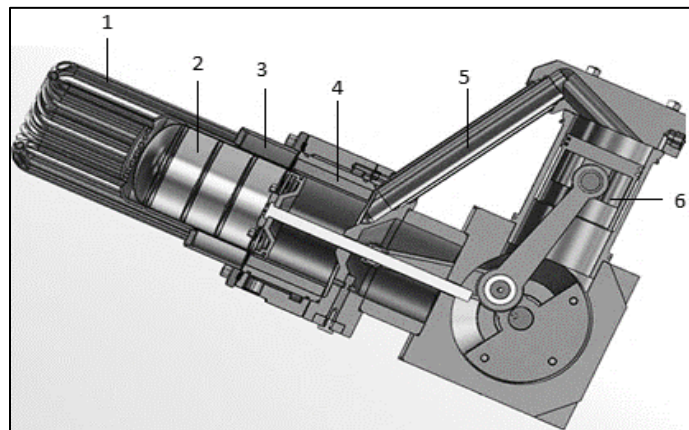
332 The outer walls and the inner wall of the sector (except the channels) are adiabatic and the reference pressure is the
333 charge pressure of the engine. Transient conjugate heat transfer simulations based on compressible laminar flow
334 were carried out for all configurations following the same procedure of meshing sequence described in section 3.

335

336 4. Engine description

337 The engine under study is a gamma-type that was first designed by Dieter Viebach in 1992 in Germany to promote
338 microgeneration with biomass fuels and since then was opened for research development [27]. The engine, shown in
339 fig. 10, consists of power and displacer pistons with 90° phase angle, and three heat exchangers (heater, cooler and
340 regenerator) and a connecting pipe. The expansion and compression spaces are connected via a 30-mm concentric-
341 cooled pipe. Meanwhile, the engine is heated up to 650 °C by an external electric heating unit and cooled by a
342 circuit of cooling water normally at 15°C. The geometrical and operational parameters of the engine are listed in
343 **Table 5.**

344



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346 Figure 10: Engine components: 1-Heater, 2-Displacer piston, 3-Regenerator, 4-Cooler, 5-Connecting pipe, 6-Power
347 piston.

348 **Table 5**
349 Engine's geometrical and operational parameters.

Parameter	Value/description
Nominal rotational speed (rpm)	500
Stroke (mm)	75
Power piston bore (mm)	85
Displacer piston bore (mm)	96
Charge pressure (bar)	10
Working gas	N ₂
Heater type	Tubular
Cooler type	Finned
Regenerator type	Annular/Random fibre
Wire diameter (Micron)	31
Porosity	0.9
Hot source temperature (°C)	650
Inlet water temperature (°C)	15
Water flow rate (L/min)	3.5
Water cooling power (kW)	2.3
Compression ratio	1.3

350
351

352 A full CFD model of the engine developed by the author in a previous study [22] will be used to investigate the
353 effect of channels regenerator on engine performance in section 5. The model was based on a realistic Local
354 Thermal Non-Equilibrium (LTNE) approach for porous domains in the engine (cooler and regenerator),
355 compressible laminar Non-Isothermal flow for nonporous domains and moving boundaries of the pistons. The
356 model results showed an acceptable degree of accuracy of 9% and 5%, respectively when comparing with
357 experimental results in predicting the indicated and cooling powers at different heating temperatures.

358 **Results and discussion**

359 A sample of the temperature contours of 1.5 mm channels regenerator at two extreme engine speeds were presented
360 in figs. (11-12). Most of the heat is transferring in the radial direction depending on the gas inlet velocity through the
361 channels. As the inlet velocity to the regenerator sector increases from 100rpm (fig.11) to 1000rpm (fig.12), more
362 energy is transferred to the solid matrix due to the forced convection giving a rise to the average solid temperature of
363 the matrix. As can be seen that the minimum temperature of the matrix at 100rpm is 500 °C compared to 865 C° at
364 1000 rpm.

365

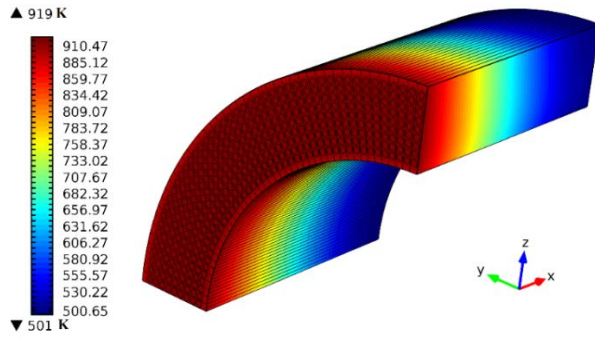


Figure 11: Temperature contours for 1.5mm channels regenerator at speed of 100 rpm and time of 30s.

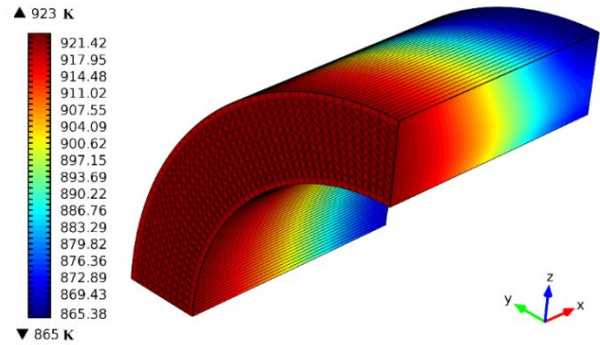


Figure 12: Temperature contours for 1.5mm channels regenerator at speed of 1000 rpm and time of 30s.

366

367

368 The fluid flow and heat transfer characteristics were obtained from CFD simulations for each configuration and are
 369 plotted in fig.13-15. As can be seen that data of pressure loss (fig.13) shows almost linear trends with gas inlet
 370 velocity for all regenerator configurations. The inertial loss part is not significant in the channels due to the absence
 371 of flow separation and vortices. The highest pressure loss is observed for 0.4mm channels regenerator.

372

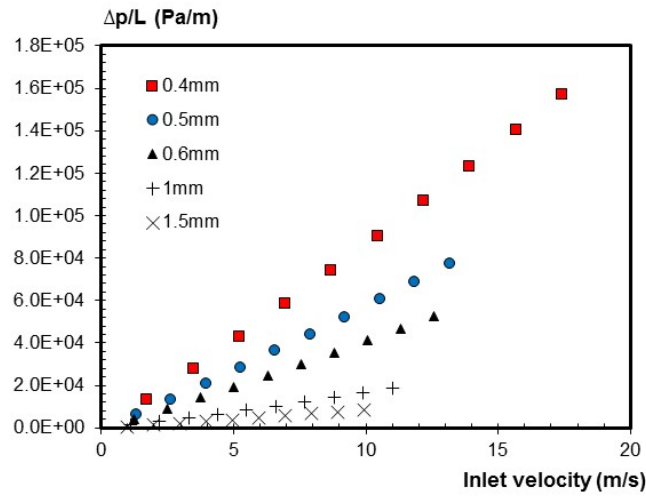


Figure 13: Pressure loss per unit length vs. gas inlet velocity to the regenerator for different channels regenerators.

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374 The simulation results for friction factor and Nusselt number were plotted in fig.14-15. The average friction factor
375 shows a decreasing trend with increasing Reynolds number for all configuration, in which the friction factor
376 correlations are close to Darcy friction factor for laminar flow.
377

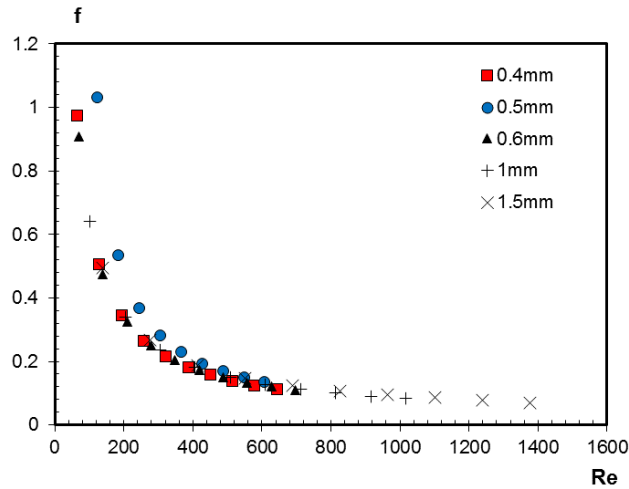


Figure 14: Darcy friction factor vs. Reynolds number for different channels regenerators.

378
379 Fig.15 shows that the average Nusselt number increases with the fluid flow velocity. This result is natural and it
380 agrees with the principles of forced convection. For larger diameter channels, higher values of Nusselt number is
381 depicted in fig.15 due to the higher values of hydraulic diameters. However, the small channels regenerators indicate
382 faster thermal response at a small range of Reynolds number (100 to 700) compared to larger diameter channels
383 (100 to 1400). In terms of heat transfer coefficient, larger diameter channels experience low heat transfer
384 coefficients due to the limited heat transfer surface area.

385

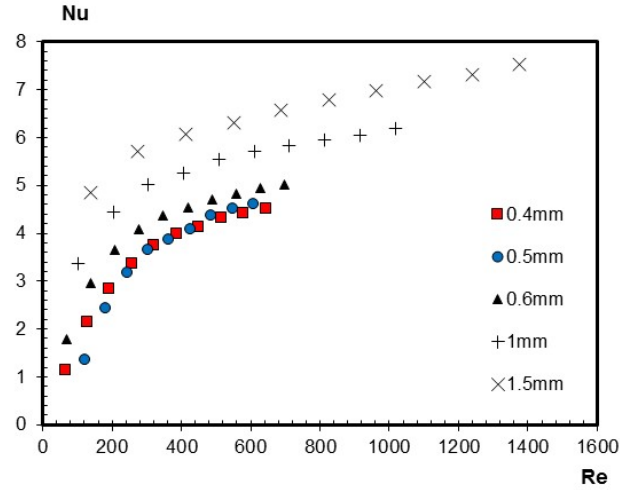


Figure 15: Nusselt number vs. Reynolds number for different channels regenerators.

386

387 The pressure drop-velocity data of each regenerator was fitted to Forchheimer-Darcy equation (Eq.20) using the
 388 least square method in order to obtain the equivalent porous media parameters (porosity, Forchheimer drag
 389 coefficient and permeability).

$$\frac{\nabla p}{L} = \frac{\mu}{K} u + \beta_F u^2 \quad (20)$$

390

391 All simulation results such as Nusselt number correlations and porous media parameters were tabulated in **Table 6**.

392 The specific solid surface area is used in porous media modelling to quantify the non-equilibrium source term when
 393 solving both energy equations of the gas and solid phases. It is defined as the ratio of the matrix surface area
 394 exposed to the gas to the volume of the matrix and can be calculated in terms of porosity (ϵ) and hydraulic
 395 diameter (d_h) from [22]

$$a_{fs} = \frac{4\epsilon}{D_h} \quad (21)$$

396

397 This data of each configuration will be used as input parameters to the engine CFD model to evaluate the
 398 performance of the engine with each configuration. This will include evaluating the indicated and cooling powers of
 399 the engine for the comparison.

400

401 **Table 6**
 402 Equivalent porous media characteristics of the three proposed regenerators.

Regenerator type	Permeability, K (m ²)	Forchheimer drag coefficient, β_F (kg/m ⁴)	Nusselt number	Specific surface area (1/m)
0.4mm	4.95E-09	61.375	$0.124Re^{0.5747}$	2475
0.5mm	7.73E-09	59	$0.195Re^{0.5126}$	2617
0.6mm	1.1E-08	49.2	$0.34Re^{0.4249}$	2284
1.0mm	3E-08	32.925	$1.143Re^{0.2488}$	1565
1.5mm	6..26E-08	23	$2.018Re^{0.1812}$	1156

403

404 The variation of matrix temperature versus the length of regenerator at the end of the 5th cycle for 0.5 mm channels
 405 regenerator is depicted in fig.16. It is worth noting how the temperature variation remarkably deviates from the well-
 406 known linear trend with the appearance of large curvature during the cold and hot blow times. This curvature
 407 caused a considerable exchange of energy between the gas and the matrix. The axial conduction loss is a key
 408 mechanism that occurs in channel regenerators and tends to lower the NTU of the matrix due to the large
 409 temperature gradient between the gas and the solid. This can be more illustrated in fig.17 when the solid and gas
 410 temperatures are plotted over the cycle.

411

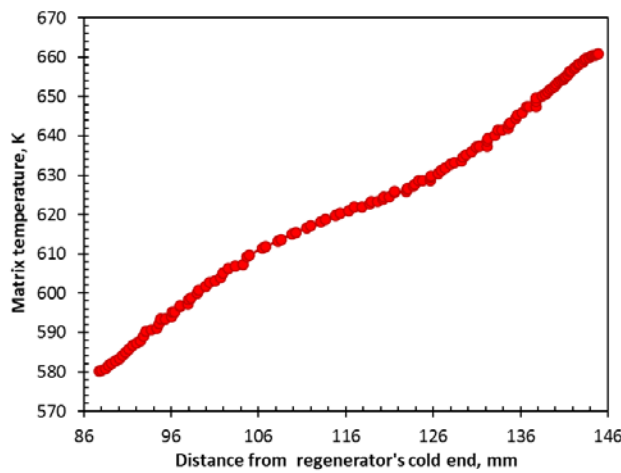


Figure 16: Axial temperature variation of regenerator matrix for the 0.5 mm channels regenerator.

412

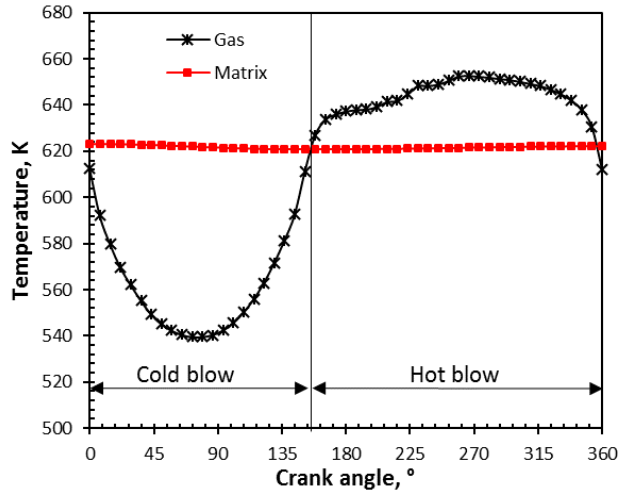


Figure 17: Gas and matrix temperatures variation in the regenerator over one cycle.

413

414 An amplitude of several degrees was observed between the gas and matrix temperatures during the cycle. As can be
 415 seen that the process of heat regeneration occurs due to heat release and storage in the matrix. During cold blow
 416 period, the gas enters the regenerator with a temperature lower than the matrix temperature. Meanwhile, in the hot
 417 blow period, the gas temperature becomes higher than the matrix one.

418

419 Fig.18 shows the engine indicated power for channels regenerators and random fibre at a heating temperature of 650
 420 °C. As depicted, the indicated power of 0.5mm regenerator is 4% lower than that of the random fibre regenerator.
 421 As the channels diameter increases, the indicated power deteriorates reaching a reduction of 31% for 1.5mm
 422 regenerator compared to that of random fibre. Although pressure drop of channels regenerator is significantly lower
 423 than that of conventional regenerators, the performance of the engine is governed by the heat transfer characteristics
 424 of the regenerator.

425

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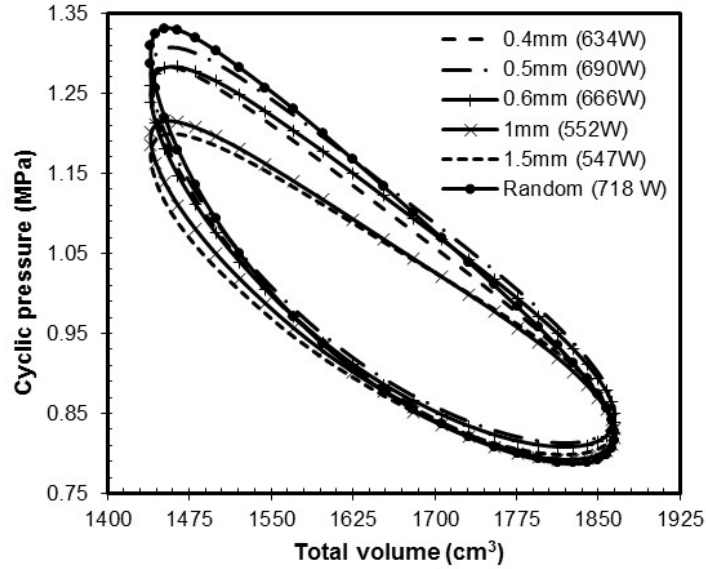


Figure 18 Indicated PV diagrams for channels regenerator and random fibre.

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Fig.19 shows a comparison of the heat rejected from engine’s cooler for channels regenerator and random fibre where the maximum heat rejected from the engine is observed for 1.5mm channel regenerator which is 17% higher than that of the random fibre. One important factor of regenerator performance is the specific surface area exposed to the working gas. Since heat transfer resistance between the gas and the solid is much higher than the resistance inside the regenerator. Therefore, transferring the heat from the gas to the regenerator can be enhanced by increasing the specific surface area.

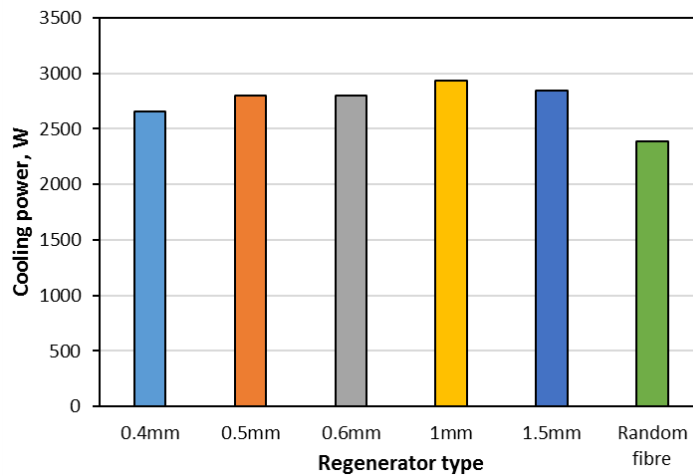


Figure 19: Cooling power for channels regenerator and random fibre.

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437 A closer comparison of the solid specific areas between the random fibre and channel regenerators showed that the
438 surface area of all channel regenerators is limited which is lower than that of random fibre. However, the volumetric
439 heat capacity of the channel regenerator is significantly higher than that of random fibre due to a lower porosity. The
440 ineffectiveness in channels regenerators was their excessive conduction loss due to high longitudinal thermal
441 conductivity. Therefore, the regenerator's NTU is reduced due to the reduction of the temperature difference
442 between the working gas and matrix.

443
444 However, the ineffectiveness of channels regenerators can be alleviated by optimizing the matrix material and the
445 segmentation of the whole regenerator. The thermal performance of the regenerator is governed by the choice of its
446 material. A material with good heat capacity allows to absorb/release maximum energy to/from the working gas and
447 a material with lower thermal conductivity reduces conduction losses. Different materials such as Stainless steel,
448 Monel 400, ceramic (ZrO₂), copper and aluminium with different thermal properties, were investigated using the
449 engine full CFD model. The thermal properties of those material are listed in **Table 7**.

450

451

452

453

454

455 **Table 7**

456 Material thermal properties of regenerator matrix [14].

Properties	Stainless steel (304L)	Copper	Aluminium	Monel 400	Ceramic (ZrO₂)
Density (kg m ⁻³)	7850	8920	2700	8800	6050
Thermal capacity (J kg ⁻¹ K ⁻¹)	475	385	902	430	460
Thermal conductivity (J kg ⁻¹ K ⁻¹)	26*	390	237	22	3

457 *recommended for Stirling engine environment based on oscillatory flow condition [28]

458

459 Monel 400 and Stainless steel have the highest volumetric heat capacity (material density times the specific heat
460 capacity) among those materials and lower thermal conductivity. Therefore, it is expected that those two materials
461 will show better performance compared to copper and aluminium with higher thermal conductivities. In order to

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462 investigate the effect of those materials on engine performance, the maximum heater temperature of the engine was
463 maintained at 500 °C to avoid approaching the melting point of aluminium. As depicted in fig.20, higher indicated
464 power was obtained using Monel 400 and Stainless steel, while copper has the lowest indicated power. In terms of
465 the cooling power rejected from the engine, ceramic (ZrO₂) and Monel 400 had the lowest cooling power while
466 copper had the highest. This proves that the best regenerator material should have a reasonable volumetric heat
467 capacity and a lower thermal conductivity.

468

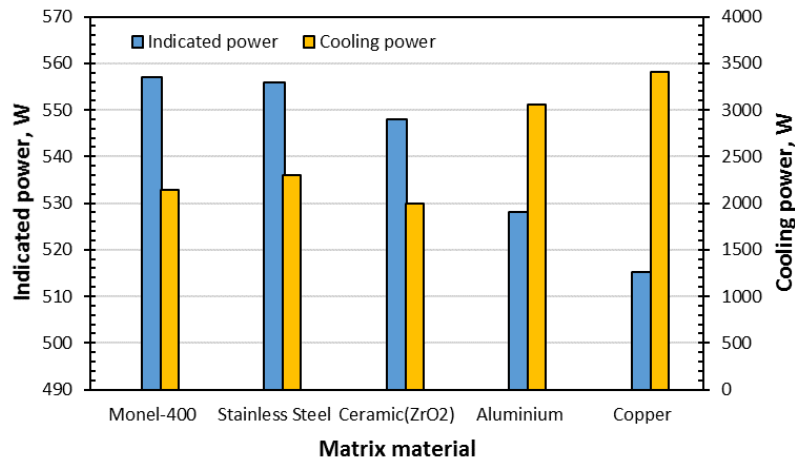


Figure 20: Indicated and cooling powers for different matrix materials.

469

470

471 Segmentation of the regenerator adds thermal contact resistance to each consecutive sheets so that the axial
472 conduction loss is minimized due the interruption of the solid continuity. Similar behaviour was observed by authors
473 [18, 29] where they conducted an experimental study to find the optimum number of segmentations for parallel wire
474 regenerator. The theoretical investigation of segmentation of a parallel geometry regenerator was not reported in
475 literature due to the lack of defining thermal contact characteristics that resembles the real cutting properties of the
476 sheets such as surface roughness (height and slope), hardness of the material, contact pressure and the gap
477 conductance. Therefore, the next phase of this design will be based on regenerator segmentation and experimental
478 investigation of heat transfer and fluid flow characteristics of the different configurations of channel regenerators.

479

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482

483 **5. Conclusion**

484 Potentiality of new Stirling regenerator as one of parallel-geometry regenerators was investigated using CFD
485 simulations. Circular miniature-channels with different diameters in the range of (0.4 to 1.5mm) were adopted in this
486 study. The metrics of CFD results was initially set up to replicate fluid flow and heat transfer characteristic for a
487 developing laminar pipe flow. A transient conjugate heat transfer simulations were performed on a 3D regenerator
488 sector to obtain heat transfer and fluid flow characteristics for all configuration of channels regenerators. The
489 obtained data was converted into equivalent porous media characteristics and then was utilized for full engine CFD
490 simulations. The results showed that small channels regenerators (0.5mm) can have good potential to generate
491 power as random fibre. It was found that the higher cooling power of the engine using channels regenerators is
492 posed by the axial conduction loss. The matrix material of channel regenerator can have strong impact on
493 regenerator performance. The results showed that using materials such as ceramic (ZrO₂) and Monel 400 can
494 alleviate the axial conduction loss due to the low thermal conductivity and enhance the engine performance. A
495 further 3D CFD study on an oscillatory pipe flow will follow this initial design with an experimental verification of
496 regenerator segmentations to further optimize the channels regenerators performance.

497

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506

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