Design and optimization of automotive thermoelectric generators for maximum fuel efficiency improvement

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Abstract

Automotive fuel efficiency can be increased by thermoelectric power generation using exhaust waste heat. A high-temperature thermoelectric generator (TEG) that converts engine exhaust waste heat into electricity is simulated based on a light-duty passenger vehicle with a 4-cylinder gasoline engine. Strategies to optimize TEG configuration and heat exchanger design for maximum fuel efficiency improvement are provided. Through comparison of stainless steel and silicon carbide heat exchangers, it is found that both the optimal TEG design and the maximum fuel efficiency increase are highly dependent on the thermal conductivity of the heat exchanger material. Significantly higher fuel efficiency increase can be obtained using silicon carbide heat exchangers at taller fins and a longer TEG along the exhaust flow direction when compared to stainless steel heat exchangers. Accounting for major parasitic losses, a maximum fuel efficiency increase of 2.5% is achievable using newly developed nanostructured bulk half-Heusler thermoelectric modules.

Nomenclature

Abbreviations

TE thermoelectric

TEG thermoelectric generator

TEM thermoelectric module

Symbols

P power (W)

26	Q	heat flow (W)
27	T	temperature (K)
28	k	thermal conductivity (W/m·K)
29	ZT	thermoelectric figure of merit
30	h	fin height (mm)
31	t	fin thickness (mm)
32	$\dot{V}$	volumetric flow rate (m ³ /s)
33	c_p	specific heat (J/kg·K)
34	Δp	pressure drop (Pa)
35	Pr	Prandtl number

36 *Greek letters*

37	α	Seebeck coefficient (V/K)
38	η	efficiency (%)
39	μ	viscosity (kg/m·s)
40	ρ	density (kg/m ³)
41	σ	electrical conductivity (S/m)

42 *Subscripts*

43	c	cold side
44	h	hot side
45	m	mean
46	t	turbulent
47	wt	weight

48

49 *Keywords:*

50 Automotive fuel efficiency

51 Thermoelectric generator

52 Waste heat recovery

53 Heat exchanger optimization

54

55 **1. Introduction**

56 About 70% of fuel energy is lost as waste heat in automobiles via engine exhaust and coolant
57 [1]. Automotive fuel efficiency can be increased by the conversion of waste heat into electricity.

58 Using no moving parts, thermoelectric generators (TEGs) convert thermal energy in the form of a
59 temperature gradient directly into electricity through the Seebeck effect. The thermoelectric figure
60 of merit (ZT) provides a dimensionless performance metric for thermoelectric materials, and is
61 defined as $ZT = (\alpha^2 \sigma / k) T$, where α , σ , and k are the Seebeck coefficient, electrical conductivity,
62 and thermal conductivity of the material, respectively, and T is the absolute temperature [2].

63 Thermoelectric devices are typically operated at maximum efficiency or maximum power. In waste
64 heat recovery applications where the heat source is virtually free, the cost per watt can be
65 minimized by operating the thermoelectric device at maximum power with efficiency given by

$$\eta_{TE} = \frac{T_h - T_c}{T_h} \frac{ZT_h}{ZT_m + ZT_h + 4}, \quad (1)$$

66 where T_h and T_c are the hot and cold side temperatures of the thermoelectric module, respectively,
67 ZT_h is the ZT at T_h , and ZT_m is the mean ZT between T_h and T_c [3, 4]. The thermoelectric conversion
68 efficiency can be improved by increasing ZT and/or the difference between T_h and T_c .

69 TEGs have been demonstrated on vehicles using commercially available bulk thermoelectric
70 materials [5-8]. However, automotive TEGs have not become an economically viable technology due
71 to low conversion efficiency and lack of inexpensive and reliable thermoelectric modules (TEMs)
72 that can withstand thousands of high-temperature ($>600^\circ\text{C}$) thermal cycles. In recent years,
73 significant improvements in material ZT have been realized in nanostructured materials [9-14].

74 Nanostructured bulk half-Heusler alloys are of particular interest for automotive waste heat
75 recovery applications due to their high peak ZT (1.0 for n-type and 0.9 for p-type) at temperatures
76 around 500°C [15-20], which is close to typical automotive exhaust temperatures. A TEG
77 constructed with these materials produced over 1 kW of electricity from the exhaust waste heat of a
78 diesel engine [21], affirming the potential of the half-Heusler materials for automotive waste heat
79 recovery.

80 The ultimate performance metric for automotive TEGs is fuel efficiency improvement. In order
81 to make TEGs a commercially viable option for automotive applications, comprehensive modeling is
82 required to achieve the maximum fuel efficiency increase.

83 Many theoretical models predicting TEG performances have been developed. Wang et al.
84 simulated an automotive TEG for various temperatures, flow rates, and heat transfer coefficients,
85 among other factors [22]. They found that the heat transfer coefficient on the exhaust side is the
86 limiting factor for TEG performance for water-cooled automotive TEGs. However, methods to
87 increase the heat transfer coefficient were not within the scope of the study. Baker et al. modeled
88 different heat exchanger types, such as porous metal foams and parallel-plate fins with parallel and
89 serpentine duct arrangements [23]. They reported that porous media and serpentine arrangements
90 result in an unacceptably large pressure drop for automotive applications. A constant fin thickness
91 of 2.5-mm was used – a significantly thicker fin than the optimal fin thickness in this work. Kumar
92 et al. developed models for rectangular, hexagonal, and circular TEG configurations with copper
93 heat exchangers and found that a transverse TEG arrangement with only one TEM along the
94 exhaust flow is ideal due to high TEG power and low pressure drop [24, 25]. Karri et al. created a
95 comprehensive water-cooled TEG vehicle model including parasitic losses due to TEG weight and
96 pressure drop in order to test various TEMs with a fixed heat exchanger geometry, and a peak fuel
97 efficiency increase of 1.25% was reported on a gasoline-powered SUV using quantum well
98 thermoelectric materials [7]. At least two TEGs have been demonstrated on diesel engines. A low-

temperature Bi_2Te_3 -based TEG that produced 944 W with a brass heat exchanger was tested by Liu et al [8]. The nanostructured bulk half-Heusler-based TEM implemented in this work was demonstrated on a TEG that produced 1002.6 W from the exhaust of a diesel engine using a nickel heat exchanger [21]. Neither of these studies offer heat exchanger optimization strategies.

Previous models suggest that the exhaust-side heat exchanger is the limiting factor for heat transfer to the TEMs and pressure drop across the TEG [22, 23]. However, existing literature has not provided a strategy to simultaneously optimize the critical heat exchanger and TEG design parameters in order to fully extract the potential of a high-temperature nanostructured TEG for maximum fuel efficiency increase.

Here we report a three-dimensional model for a high-temperature ($>600^\circ\text{C}$) automotive TEG system with emphasis on optimization of heat exchanger parameters and overall TEG configuration in order to achieve the maximum fuel efficiency increase. While this model is applicable to any TEM, the nanostructured bulk half-Heusler TEM is chosen for this study due to its superior performance in the automotive exhaust temperature range. The strategy for the design optimization is to pursue the optimal heat exchanger design and TEG configuration through careful consideration of the trade-off between TEG power output and the parasitic fuel efficiency losses due to the increased vehicle weight and exhaust and coolant pressure drops. The model is validated by comparing simulation results to the experimental results of a TEG prototype tested on a diesel engine [21]. The model shows agreement within 3% for temperature distribution and 6% for pressure drop, respectively.

2. Model description

2.1 Nanostructured bulk half-Heusler thermoelectric modules

The nanostructured bulk half-Heusler TEM is designed and constructed for high-temperature automotive waste heat applications [21]. As shown in **Fig. 1** (a), the TEM containing 14 thermoelectric couples has a footprint of $27 \times 27 \text{ mm}^2$ on the hot side, $40 \times 40 \text{ mm}^2$ on the cold side,

124 and is 5 mm thick. Each thermoelectric couple consists of 2 n-type and 2 p-type elements connected
 125 in series using direct bonded copper consisting of two copper layers (0.25 mm thick) direct bonded
 126 to a ceramic substrate (0.6 mm thick), as shown in **Fig. 1** (b). The thermoelectric elements are 2 mm
 127 tall and have $1.8 \times 1.8 \text{ mm}^2$ cross-sectional area.

Fig. 1. (a) Schematic of the half-Heusler TEM containing 14 thermoelectric couples connected in series; (b) Schematic of one thermoelectric couple containing 2 n-type and 2 p-type elements connected in series with direct bonded copper.

128
 129 The thermoelectric couples are sealed in a stainless steel enclosure with argon atmosphere. A
 130 0.25 mm thick graphite foil provides a thermal interface between the hot side of the thermoelectric
 131 couples and the top side of the case (not shown in **Fig. 1**). The cold sides of the couples are soldered
 132 directly to the case bottom.

133 The steady state thermal resistance and power output of the TEM were measured as a function
 134 of hot and cold side temperature using a well-established steady-state method, as described in Ref.
 135 [26]. The TEM is modeled as a $27 \times 27 \times 5 \text{ mm}$ thermal resistance block with effective thermal
 136 conductivity of $2.69 \text{ W/m}\cdot\text{K}$, which was experimentally determined under temperatures
 137 representative of those found in the simulation: $T_h \sim 400^\circ\text{C}$, $T_c \sim 100^\circ\text{C}$, and electric current resulting
 138 in maximum power.

139 As shown in **Fig. 2**, the TEM power was experimentally measured as a function of temperature
 140 difference with the cold side fixed at 95°C, which is near the average value of cold side TEM
 141 temperatures realized in simulation. The TEM power output peaks at 7.9 Watts with the hot side at
 142 595°C and the cold side at 95°C. The power output is slightly higher than the value reported in Ref.
 143 [21] due to a reduction in the ceramic thickness in the TEM and enhanced brazing procedures
 144 resulting in decreased parasitic losses.

Fig. 2. Power output of the nanostructured half-Heusler TEM measured as a function of temperature difference across the TEM. The cold side is fixed at 95°C.

145
 146 The best-fit quadratic through the experimental data is given by

$$P = 2.90(10^{-5})(\Delta T)^2 + 1.46(10^{-3})(\Delta T) - 9.60(10^{-2}) \quad (2)$$

147 Equation (2) has 99.9% quality of fit to the experimental data and is valid for $200^{\circ}\text{C} < \Delta T < 500^{\circ}\text{C}$.
 148 For any given ΔT within this range, there is <2% change in TEM power for any cold side
 149 temperature variations obtained during simulation. Thus, equation (2) is used to determine the
 150 power of each individual TEM during simulation, where the input temperature is the integral
 151 average across the respective TEM surface. By incorporating this experimentally-derived equation
 152 into the TEG system model, uncertainties in the simulation of TEM power output are eliminated.

153 *2.2 TEG model and optimization parameters*

154 The core components of the TEG are the TEMs and the hot and cold side heat exchangers. The
 155 hot side heat exchanger consists of a series of parallel-plate fins that intercept the engine exhaust
 156 flow and transfer heat from the hot exhaust gas to the TEM hot side. The parallel-plate fin heat
 157 exchanger is chosen for its proven performance in exhaust waste heat recovery and relatively low
 158 pressure drop compared to other designs [7, 21, 23-25]. Two hot side heat exchanger materials are
 159 studied due to their corrosion resistance, thermal conductivity, and high-temperature stability:
 160 stainless steel 444 (SS 444) with $k = 26.8 \text{ W/m}\cdot\text{K}$ and silicon carbide (SiC) with $k = 120 \text{ W/m}\cdot\text{K}$.
 161 Despite its proven performance for high-temperature ($>1000^\circ\text{C}$) applications, SiC remains
 162 relatively unexplored as an automotive TEG heat exchanger material. The cold side of the TEM is
 163 cooled by passing the vehicle coolant through a cold plate that runs counter to the exhaust flow,
 164 providing downstream TEMs with the lowest cold side temperatures.

165 The total number of TEMs is fixed at 72 based on the space constraint to install the TEG in the
 166 vehicle exhaust system. Hence, the TEG configuration may be described by the aspect ratio N_L/N_A ,
 167 where N_L is the number of TEMs along the exhaust flow and N_A is the number of TEMs in the cross
 168 section, as illustrated in **Fig. 3**.

169

Fig. 3. Left: Schematic of single exhaust flow passage showing heat exchangers (red), TEMs (green), and cold plates (blue); Right: Various TEG configurations with six different TEG aspect ratios N_L/N_A .

170

171 Six TEG aspect ratios are simulated: $N_L/N_A = 1/72, 2/36, 3/24, 4/18, 6/12$, and $9/8$. Since the
 172 overall number of TEMs is constant, $N_L \times N_A = 72$ for all cases. The heat exchanger width w and
 173 length l are both fixed at 40 mm to accommodate the size of the TEM. To minimize computational
 174 expenses, a single exhaust flow passage is modeled assuming the mass flow rate is evenly distributed
 175 among all passages. The exhaust flow passage has two orthogonal planes of symmetry parallel to the flow
 176 direction, so only $1/4$ of the domain was simulated. The mesh was refined until grid independence was
 177 achieved. The number of elements in the final mesh ranged from ~ 1 million to ~ 20 million depending on
 178 the TEG aspect ratio, fin height, fin thickness, and fin packing fraction. An example mesh for a TEG with
 179 a single TEM along the exhaust flow direction (aspect ratio $N_L/N_A = 1/72$) and 1.0 mm thick fins with a
 180 packing fraction of 0.3 is shown in **Fig. 4**.

Fig. 4. Mesh images of a TEG with a single TEM along the exhaust flow direction (aspect ratio $N_L/N_A = 1/72$) showing heat exchanger (red), TEM (green), water and cold plate (blue) and symmetry boundary conditions (yellow). (a) Cut plane parallel to flow direction; (b) Cut plane normal to flow direction with inset showing increased mesh density near the walls.

As shown in **Fig. 4** (a), the non-conformal mesh is selectively refined from the entry region to the HEX with gradient refinement to minimize the number of hanging nodes. The full 200 mm long entry region is not shown. The mesh density is increased near the walls and in areas of high velocity gradient. Meshes for models with larger TEG aspect ratio are similar but are longer along the exhaust flow direction, and thus have a greater number of elements. **Fig. 4** (b) shows the mesh as viewed along the exhaust flow direction with gradient nodal spacing between fins.

The TEG aspect ratio N_L/N_A , fin height h , fin thickness t , fin packing fraction, and heat exchanger material are the design parameters studied in this work. Table 1 shows the range of optimization parameters.

Table 1 Optimization parameters.

Parameter	Value
TEG aspect ratio N_L/N_A	1/72, 2/36, 3/24, 4/18, 6/12, 9/8
Flow passage height h (mm)	10, 20, 30, 40, 50, 60
Fin packing fraction	0.1, 0.2, 0.3, 0.4, 0.5
Fin thickness t (mm)	0.2, 0.4, 0.6, 0.8, 1.0, 1.2
Heat exchanger material	SS 444, SiC

2.3 Fluid properties and boundary conditions

Of all driving conditions, highway driving has perhaps the highest potential for fuel efficiency improvement via TEGs due to relatively steady exhaust conditions along with high exhaust temperature and flow rate. Table 2 summarizes key input parameters, including the mass flow rate and temperature of the exhaust and coolant for the 2.0-L 4-cylinder vehicle under a highway driving condition at 107 ± 23 km/hr.

Table 2 Average values for parameters collected during highway driving.

Parameter	Value
Exhaust Temperature (°C)	708
Exhaust Flow Rate (g/s)	21.9
Coolant Temperature (°C)	82.0
Coolant Flow Rate (L/min)	14.9
Engine Speed (RPM)	2500
Engine Power (kW)	18.2
Vehicle Speed (km/hr)	107

200

201 The exhaust temperature is taken at the outlet of the primary catalytic converter. The mean
202 exhaust temperature is 708 °C for highway driving with associated mass flow rate 21.9 g/s. The
203 temperature of the coolant is between 81.2 °C and 83.2 °C for the entire drive cycle with an average
204 of 82.0 °C. The exhaust is approximated as air with temperature-dependent viscosity and thermal
205 conductivity [27]. The coolant is modeled as water at 82°C. Convection and radiation heat transfer
206 on the TEM free surfaces are neglected since in practice the TEMs are covered with thermal
207 insulation material [21].

208 *2.4 Governing equations*

209 The Mach number for the maximum exhaust flow condition is less than 0.1, so the flow may be
210 treated as incompressible with reasonable accuracy [28]. The conservations of mass and
211 momentum for a steady, incompressible flow are

$$\nabla \cdot \vec{v} = 0 \quad (3)$$

$$\nabla \cdot (\rho \vec{v} \vec{v}) + \nabla p - \nabla \cdot \bar{\bar{\tau}} - \rho \vec{g} = 0, \quad (4)$$

212 where $\vec{v}$ is the velocity vector field, $\bar{\bar{\tau}}$ is the stress tensor, and $\vec{g}$ is the acceleration due to gravity
213 [29]. The Reynolds number for flow in the exhaust pipe at the mean highway driving condition is

214 ~13,000. Therefore, an algebraic turbulence model (zero equation) is employed with turbulent
215 viscosity defined as

$$\mu = \rho l^2 S, \quad (5)$$

216 where l is the size of the turbulent eddies and S is the modulus of the mean rate of strain tensor
217 [29]. The fluid turbulent thermal conductivity is then computed as

$$k_t = \frac{\mu c_p}{Pr}, \quad (6)$$

218 where c_p is the specific heat and Pr is the Prandtl number [29]. Finally, the energy balance is given
219 by

$$\nabla \cdot (\rho h \vec{v}) - \nabla \cdot ((k + k_t) \nabla T) = 0 \quad (7)$$

220 where h is the enthalpy and k is the molecular thermal conductivity [29]. These equations are
221 solved numerically with ANSYS.

222 *2.5 Parasitic losses*

223 The parasitic losses associated with the TEG include the pumping power of the engine necessary
224 to overcome the increased exhaust pressure drop across the TEG, the pumping power of the coolant
225 through the cold plates, and the rolling resistance due to the weight of the TEG.

226 *2.5.1 Exhaust and coolant pumping power*

227 The power required to overcome the exhaust pressure drop across the TEG is related to the
228 engine speed ω as

$$P_h = \dot{V}_h \Delta p_h = \frac{\pi \omega N_p s b^2}{8} \Delta p_h, \quad (8)$$

229 where $\dot{V}_h$ is the volumetric flow rate of the exhaust, Δp_h is the exhaust pressure drop across the TEG,
230 N_p is the number of pistons, s is the piston stroke, and b is the piston bore. For the mean highway
231 driving condition, $\dot{V}_h = 2880$ L/min.

232 The power required to pump the coolant through the cold plate is $P_c = \dot{V}_c \Delta p_c$, where $\dot{V}_c$ is the
 233 volumetric flow rate of the coolant. The flow rate of the coolant for highway driving conditions is
 234 14.9 L/min.

235 2.5.2 Rolling resistance

236 The power required to transport the TEG weight is estimated as

$$P_{wt} = \frac{\mu V}{\eta} W_{TEG}, \quad (9)$$

237 where the rolling resistance coefficient μ and drivetrain transmission efficiency η are taken to be
 238 0.012 and 0.9, respectively, and V is the vehicle speed [7]. The weight of the TEG W_{TEG} includes the
 239 weight of the cold plates, the TEMs, and the heat exchangers.

240 2.6 Fuel efficiency calculation

241 The change in fuel efficiency is a function of the engine power P_{eng} , the power produced by the
 242 TEG P_{TEG} , the alternator efficiency η_{alt} , and the parasitic losses:

$$\Delta FE = \left(\frac{P_{eng}}{P_{eng} - \frac{P_{TEG}}{\eta_{alt}} + P_h + P_c + P_{wt}} - 1 \right) \times 100 \quad (10)$$

243 The alternator efficiency is 55% for the highway driving condition, a typical value for belt-driven
 244 alternators [30].

245 3. Results and Discussion

246 3.1 Effect of TEG configuration and heat exchanger fin height

247 In this section, the TEG aspect ratio N_L/N_A and fin height h are optimized for a heat exchanger
 248 with the optimal fin thickness and packing fraction for each heat exchanger material. The optimal
 249 packing fractions for SS 444 and SiC are 0.36 and 0.30, respectively, while both materials have the
 250 optimal fin thickness of 0.6 mm (optimization of fin thickness and packing fraction is discussed in
 251 section 3.2). **Fig. 5** shows total TEG power, exhaust pumping power, and net fuel efficiency increase
 252 as a function of N_L and h for SS 444 and SiC heat exchangers.

Fig. 5. (a) TEG power and exhaust pumping power and (b) fuel efficiency increase vs TEG aspect ratio for multiple fin heights with a SS 444 heat exchanger with a packing fraction of 0.36 and 0.6 mm thick fins; (c) TEG power and exhaust pumping power and (d) fuel efficiency increase vs TEG aspect ratio for multiple fin heights with a SiC heat exchanger with a packing fraction of 0.30 and 0.6 mm thick fins.

As shown in **Fig. 5** (a) and (c), an increase in TEG power is observed with shorter fin heights due to increased exhaust flow speed resulting in higher heat transfer coefficients and TEM hot side temperatures. The trade-off is increased pressure drop resulting in greater pumping power. As the fin height reduces to below 20 mm, the pressure drop increases drastically while TEG power increase saturates as the total fin surface area decreases. **Fig. 5** (c) shows that TEG power with 10

mm tall SiC fins is less than that with 20 mm tall fins. TEM hot side temperatures begin to drop with fins shorter than 20 mm due to insufficient heat transfer area. The same trend is not seen in **Fig. 5 (a)** because the optimal SS HEX has a greater packing fraction of 0.36 compared to 0.30 for the SiC HEX.

The fin height of a SiC heat exchanger has less influence on TEM power compared to SS 444 because the significantly higher thermal conductivity of SiC results in similar TEM hot side temperatures for fin heights up to 50 mm. As the fin height increases, the increased heat transfer surface area balances the reduced flow velocity and heat transfer coefficient. Both TEG power and exhaust pumping power increase with increasing TEG aspect ratio N_L/N_A as the heat transfer rate and pressure increase with increasing exhaust flow velocity. Since the heat exchanger fin thickness and packing fraction are fixed, the power to transport the TEG weight P_{wt} is linearly proportional to the fin height h with $P_{wt} = 14.6, 20.9, 27.1$, and 33.4 W for 10, 20, 30, and 40 mm tall SS 444 heat exchangers, respectively. For SiC heat exchangers with 20, 30, 40, 50, and 60 mm tall fins, $P_{wt} = 12.6, 14.6, 16.7, 18.8$, and 20.9 W, respectively.

As shown in **Fig. 5 (b)**, the optimal balance between TEG power, pumping power, and TEG weight for a SS 444-based TEG occurs with aspect ratio $N_L/N_A = 3/24$ and 20 mm tall fins, resulting in a peak fuel efficiency increase of 2.0%. The maximum fuel efficiency of 2.5% occurs with a SiC-based TEG with aspect ratio $N_L/N_A = 6/12$ and 50 mm tall fins, as indicated in **Fig. 5 (d)**. However, similar increases in fuel efficiency can be obtained using 30 or 40 mm tall fins, which may be preferable if overall TEG volume is to be minimized. The increase in fuel efficiency with the higher thermal conductivity material is attributed to higher hot side TEM temperatures attainable with taller fins that result in significantly lower pressure drop. Additionally, the higher temperatures boost the TE material conversion efficiency, leading to greater TEG power.

Material choice determines not only the maximum TEG power but also the optimal TEG configuration and heat exchanger geometry, and in turn, the back pressure experienced by the

284 engine. Indeed, the optimized TEG configuration for a SS 444 heat exchanger includes an aspect
285 ratio of $N_L/N_A = 3/24$ with 20 mm tall fins while the optimized configuration for a SiC heat
286 exchanger includes an aspect ratio of $N_L/N_A = 6/12$ with 50 mm tall fins.

287 *3.2 Effect of fin thickness and packing fraction*

288 In this section, the effect of fin thickness and packing fraction is optimized in order to obtain the
289 maximum fuel efficiency increase based on the optimized TEG configurations. **Fig. 6** shows the total
290 TEG power, exhaust pumping power, and net fuel efficiency increase as a function of fin thickness
291 and packing fraction for the optimized SS 444 and SiC TEG configurations. Some packing fractions
292 are modified from the values in Table 1 to ensure an integer number of fins.

Fig. 6. (a) TEG power and exhaust pumping power and (b) fuel efficiency increase vs packing fraction for multiple fin thicknesses with TEG aspect ratio $N_L/N_A = 3/24$ and 20 mm tall SS 444 heat exchangers. (c) TEG power and exhaust pumping power and (d) fuel efficiency increase vs packing fraction for multiple fin thicknesses with TEG aspect ratio $N_L/N_A = 6/12$ and 50 mm tall SiC heat exchangers.

For a given packing fraction, TEGs with thicker fins have fewer heat transfer surfaces resulting in lower TEM hot side temperatures and TEG power, as shown in **Fig. 6** (a) and (c). To offset this, higher packing fractions may be used to increase heat transfer and TEG power at the cost of increased TEG weight and exhaust pumping power. Increases in TEG power diminish for packing fractions above 0.4. For a given fin thickness, there is an exponential increase in pumping power

299 with increasing packing fraction due to the greater exhaust flow velocity and number of heat
300 transfer surfaces.

301 **Fig. 6** (b) shows the peak fuel efficiency increase of 2.0% occurs at 0.6 mm thick fins with a
302 packing fraction of 0.36 using SS 444. A comparable increase in fuel efficiency can be obtained with
303 either 0.4 or 0.8 mm thick fins, but fuel efficiency improvement drops with thicker fins due to
304 greater TEG weight. **Fig. 6** (d) shows the maximum fuel efficiency increase of 2.5% is obtained using
305 0.6 mm thick SiC fins with a packing fraction of 0.3. As with SS 444, there are multiple combinations
306 of fin thickness and packing fraction that yield similar increases in fuel efficiency for values within
307 the optimized range. When considering TEG weight and cost, the lowest packing fraction, and thus
308 the thinnest fin, would be best. On the other hand, TEGs with thinner fins have a larger pressure
309 drop that may adversely affect engine performance. Hence, the ideal fin thickness and packing
310 fraction must be determined through analysis of parameters such as maximum allowable pressure
311 drop, heat exchanger material cost, and TEG performance at other exhaust conditions.

312 *3.3 Temperature distribution*

313 The temperature distributions across the HEX and TEMs for the optimal TEG configuration and
314 HEX geometry for the two HEX materials are shown in the following figures.

Fig. 7. Temperature contours of the optimized TEG with SS 444 heat exchangers. (a) HEX inlet viewed along flow direction; (b) Isometric view (quarter model); (c) Cut plane through center fin; (d) TEM hot side surfaces with average surface temperatures listed below and average cold side surface temperatures in parentheses.

Fig. 8. Temperature contours of the optimized TEG with SiC heat exchangers. (a) HEX inlet viewed along flow direction; (b) Isometric view (quarter model); (c) Cut plane through center fin; (d) TEM hot side surfaces with average surface temperatures listed below and average cold side surface temperatures in parentheses.

As shown in **Fig. 7** and **Fig. 8**, the temperature distribution along the exhaust flow direction is highly non-uniform with upstream TEMs receiving significantly higher hot side temperatures than downstream TEMs due to decreasing exhaust temperatures along the flow direction. Meanwhile, the temperature differences between the HEX and TEM are much higher for SS HEX than SiC HEX due to the significant difference in thermal conductivity of these materials. To compensate the effect of the temperature decrease along flow direction, a gradient HEX and TEM design can be implemented. In order to increase the number of TEMs operating near peak power, higher thermal conductivity HEX materials and greater fin packing fractions can be employed in the downstream together with TE materials of peak ZT at lower temperatures.

3.4 Results summary

Table 3 summarizes the simulations results, including temperatures, pressure, and TEG power for the optimized geometry of each heat exchanger material.

Table 3 Results for optimal TEG configuration and heat exchanger geometry for each material.

	Stainless Steel 444	Silicon Carbide
TEG aspect ratio N_L/N_A	3/24	6/12
Heat exchanger height h (mm)	20	50
Fin thickness t (mm)	0.6	0.6
Packing fraction	0.36	0.3
TEG volume (L)	2.3	4.0
Max TEM hot side temperature (°C)	495	573
Mean TEM hot side temperature (°C)	391	417
Mean TEM cold side temperature (°C)	92.2	90.9
Exhaust outlet temperature (°C)	350	316
Coolant outlet temperature (°C)	86.2	86.6

TEG Power P_{TEG} (W)	224	267
Pressure drop Δp (Pa)	572	436
Pumping power P_{pump} (W)	27.9	21.3
TEG efficiency $\eta_{TEG} = P_{TEG}/Q_h$ (%)	2.60	2.84
Fuel efficiency increase ΔFE (%)	2.0	2.5

4. Conclusions

A three-dimensional TEG model integrated with experimental testing results from a high-temperature nanostructured bulk half-Heusler TEM is developed for automotive waste heat recovery applications. Accounting for major parasitic losses, the model is used to simultaneously optimize heat exchanger design (material, fin thickness, packing fraction) and overall TEG configuration (length/flow area, fin height) for maximum increase in fuel efficiency for highway driving. The model is validated by comparing simulation results to the experimental results of a TEG prototype tested on a diesel engine. Major conclusions of this study include

- Heat exchanger material has a strong influence on maximum fuel efficiency increase. The fuel efficiency improvement from the optimized TEG with SiC HEX material is 25% greater than the TEG with SS 444 HEX material.
- For a fixed number of TEMs, both the optimal TEG aspect ratio and fin height increase with the thermal conductivity of heat exchanger fin material.
- Pressure drop and heat transfer effectiveness are both important considerations in automotive TEG heat exchanger design, as the additional exhaust pumping power of the engine can dominate any gains provided by the TEG while, on the other hand, insufficient heat transfer can limit TEG power output.

348 The optimized TEG consisting of 72 nanostructured bulk half-Heusler TEMs yields 2.5% and
349 2.0% increases in fuel efficiency on the highway with a SiC heat exchanger and a SS 444 heat
350 exchanger, respectively. Additional gains in fuel efficiency can be realized with increases in material
351 ZT , implementation of cascaded/segmented modules with higher thermoelectric conversion
352 efficiency, and/or by employing a multi-stage TEG with low-temperature TEMs farther down the
353 exhaust stream.

354

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358

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