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Research Paper

A comprehensive thermo-economic analysis, optimization and ranking of different microturbine plate-fin recuperators designs employing similar and dissimilar fins on hot and cold sides with NSGA-II algorithm and DEA model

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HIGHLIGHTS

- A detailed thermo-economic optimization is performed for plate-fin recuperators using NSGA-II.
- Triangular, rectangular, offset strip and louver fins are considered for the optimization.
- Recuperators with similar and dissimilar fins on hot and cold sides are comprehensively studied.
- To rank the optimal recuperator designs, Data Envelopment Analysis (DEA) model is employed.
- For the ranking, recuperator effectiveness, cost, pressure drop, volume and mass are considered.

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ABSTRACT

This study aims to perform a comprehensive thermo-economic analysis, optimization and ranking of cross and counter-flow plate-fin recuperators employing rectangular, triangular, offset strip and louver fins. The analysis is mainly conducted for two recuperator structures: (i) fins' configurations on both hot and cold sides are the same; (ii) fins on hot side and cold side are dissimilar in configuration. Considering effective practical optimization constraints and design parameters, Non-dominated Sorting Genetic Algorithm (NSGA-II) is used to maximize the recuperator effectiveness and minimize its total cost, simultaneously. Pareto-optimal fronts are presented to specify the desirable recuperator designs satisfying the constraints. Afterwards, in order to accurately and reliably rank the optimal designs based on significant factors including recuperator effectiveness, total cost, volume, mass and pressure drop, Data Envelopment Analysis (DEA) model is utilized. According to the ranking results achieved from the DEA model, the counter-flow recuperator employing louver fins on the cold side and rectangular fins on the hot side has the best performance among the investigated recuperator structures for which the values of effectiveness, cost, pressure drop, volume and mass are equal to 0.814359, 326688.3 $, 0.716857 kPa, 0.40834 m^3, 419.3277 kg, respectively.

1. Introduction

Continuing growth in world population, energy demand and fossil fuel consumption may result in a wide variety of environmental problems. In this regard, extensive efforts are being made by many scientific and industrial communities to tackle these problems through the deployment of high-efficiency and environmentally friendly power generation systems [1–7]. Gas turbines are one of the desirable combustion engines extensively being utilized in many power plants [8]. These engines follow the Brayton cycle and typically consist of a compressor, a combustor and a turbine [9,10]. Generally, gas turbines can be classified into microturbines (5–200 kW) and miniturbines (200–500 kW) [11]. Microturbines are a promising small-scale technology for electricity generation due to their compact size [12,13], high thermal

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Nomenclature

A	thickness of separation plate, m
A	overall heat transfer area, m^2
A_{fin}	fin heat transfer area, m^2
A_w	wall heat transfer area, m^2
b	fin height, m
Cp	specific heat capacity, J/kg K
c^*	ratio of heat capacity rate
C	heat exchanger cost coefficient, W/K
$Cost$	total cost, $
c_{ope}	operational cost, $
$c_{purchase}$	purchase cost, $
$c_{Capital}$	capital cost, $
$c_{Maintenance}$	maintenance cost, $
D_h	hydraulic diameter, m
f	friction factor
f^*	inflation rate,%
G	mass flux, kg/m^2 s
h	convective heat transfer coefficient, W/m^2 K
i	interest rate, %
j	Colburn number
K_e	exit pressure loss coefficient
K_c	entrance pressure loss coefficient
k_{el}	price of electrical energy, $/MWh
k_{fin}	fluid thermal conductivity, W/m K
k_w	wall thermal conductivity, W/m K
L_n	non-flow stream length, m
L_c	cold flow stream length, m
L_h	hot flow stream length, m
L_p	louver pitch, m
L_f	fin length, m
L_l	louver height, m
LHV	fuel lower heating value, kJ/kg
m	mass flow rate, kg/s
n	recuperator expected life time, year
NTU	number of transfer unit
Nu	nusselt number
P	pressure, kPa
$P_{o,cycle}$	cycle pressure outlet, kPa
p_t	plate pitch, m
Q	rate of heat transfer, W
R_f	fouling factor, m^2 K/W
Re	Reynolds number
s	fin pitch, m
St	stanton number
T	temperature, K
T_{TIT}	turbine inlet temperature, K
T_0	reference temperature, K
t	fin thickness, m
UA	overall heat transfer coefficient, W/K
V_t	volumetric flow rate, m^3/s
w_t	louver length, m
x	offset length, m

Greek abbreviation

ΔT_m	corrected temperature, K
ΔP	pressure drop, kPa
ε	recuperator thermal effectiveness, %
η_{fin}	single-fin efficiency, %
η_s	overall fin efficiency, %
$\eta_{comp.}$	compressor efficiency, %
θ	louver angle, degree
ϑ_m	mean specific volume, m^3/kg
ρ	density, kg/m^3
σ	ratio of minimum free flow area to frontal area
τ	hours of operation per year, hour

Subscripts

a	air
A	first year value
c	cold
f	fuel
g	gas
h	hot
i	inlet
o	outlet
P	present value
w	wall

efficiency [12], system flexibility [14], low noise [2], multi-fuel capability [2], low emissions [14], high reliability [2] and low maintenance cost [14,15]. Microturbine efficiency can be enhanced to 30% or higher by using high-performance recuperators that can recover waste heat from the exhaust stream by preheating the incoming air before it enters the combustion chamber which leads to a considerable reduction in fuel consumption [16–19]. On the basis of heat transfer geometry, recuperators can be categorized as primary surface, plate-fin and tubular recuperators [2]. It is worth noting that plate-fin recuperators are widely utilized in gas–gas applications because of their high thermal efficiency, low weight (95% less than comparable conventional shell-and-tube heat exchangers), low cost and ease of using multiple streams. From a thermo-economic standpoint, incorporating recuperative heat transfer units into microturbines can increase the total capital cost up to 30% [20]. Therefore, comprehensive thechno-economic studies are essential taking into account the recuperator heat transfer area density and pressure drop characteristics, weight, volume, design and cost, simultaneously. Well-designed recuperators for micro gas turbines normally possess high heat transfer effectiveness (>90%), low relative pressure loss (<3) and compact lightweight matrix at low cost [21]. So far, many numerical and experimental analyses have been carried out to develop proper low-cost and highly-efficient recuperators for microturbines.

Dong et al. [22] applied K-ε model for a turbulent flow within wavy-finned heat exchangers to numerically and experimentally investigate the thermo-hydraulic performance of the heat exchangers taking into account the pressure drop and convection heat transfer coefficient. Pandey and Nema [23] experimentally measured the ineffectiveness of a plate heat exchanger with corrugate fins considering exergy loss parameter. Peng and Ling [24] employed both genetic algorithm and back propagation neural networks to minimize the total weight and total cost of plate-fin heat exchangers using rectangular fins. Yousefi et al. [25] employed the imperialist competitive algorithm (ICA) to minimize the total cost and total weight of a cross-flow offset-strip-finned heat exchanger. In another report, Yousefi et al. [26] developed a learning automata based particle swarm optimization (LAPSO) to identify the optimal design of a compact heat exchanger by conducting a three-objective optimization in which total cost, weight and entropy generation units were considered as the objective functions. Traverso and Massardo [12] proposed an approach to evaluate and optimize the performance and capital cost of two types of recuperators employed in microturbines. Cai and Jiang [27] studied the