AIN SHAMS UNIVERSITY
FACULTY OF ENGINEERING
MECHANICAL POWER DEPARTMENT

AN INVESTIGATION OF THE PERFORMANCE OF TRANSPOSED-FLUIDS HEAT EXCHANGERS TO BE USED IN STIRLING ENGINE DESIGN

A THESIS

SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS OF THE DEGREE OF THE DOCTOR OF PHILOSOPHY IN MECHANICAL POWER ENGINEERING

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Degree Name: Doctor of Philosophy in Mechanical Power Engineering

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STATEMENT

This dissertation is submitted to Ain Shame University in fulfillment of the requirements for doctor of philosophy degree in Mechanical Engineering.

This work included in the thesis was made by the author during the period from January 2004 to November 2010 at Mechanical Power Engineering Department, Faculty of Engineering, Ain Shams University.

No part of this thesis has been submitted for degree or qualification at any other university or institute.

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TO

MY SMALL FAMILY

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ABSTRACT

According to the objective of this research*, an investigation of the performance of transposed-fluids heat exchangers to be used in Stirling engine design*, eight simple heat exchangers were designed, manufactured, and tested experimentally to be used as a heater and a cooler in the alpha type Stirling engine.

Each specimen was a shell-and-tube, air-to-water, and elbow bend heat exchanger. As a first step a number of five heat exchangers were designed, manufactured, and tested. Each specimen of this group has a different tube bank arrangement and geometrical configuration. The specimen which has minimum dead volume, low pressure drop and high rate of heat transfer was selected as the best one to be improved in the second step. From the discussion of the first step experimental results, it was found that a specimen of (quadrant cross-section) and circular tube arrangement can be selected as the best one.

Three specimens were manufactured similar to the best one. These specimens have the same geometrical configuration and tube bank arrangement, but have different tube spacing. So they have different number of tubes consequently different dead volumes. From the discussion of the second step experimental results, it was found that the specimen of the least number of tubes has the minimum pressure drop, but does not have the minimum dead volume, while it has a reasonably high rate of heat transfer.

Each one of the eight tested specimens was employed individually as a heater and a cooler in a computer program (Excel program) to study the engine performance analytically. Based on the Schmidt analysis the engine work space was divided into three isothermal regions. This program considers the friction losses and consequently the pressure drop due to the working fluid flow through each part of the engine.

The dimensions of the engine parts were optimized analytically according to two schemes to get a maximum output power. In the first scheme, different strokes of the expansion and compression were assumed. In the second scheme, they were assumed to be equal. For all calculations the optimum performance was calculated at a charging pressure, which insures 40 bar maximum pressure inside the machine. The cooling water mass flow rate and its inlet temperature to the cooler were assumed constant. Also the hot gasses mass flow rate and its inlet temperature to the heater were assumed constant. The inner diameter of the expansion and compression spaces was assumed to be equal. The optimum dimensions of the engine were found. The optimum engine power was found for a specimen which has square cross-section, in-line tube bank arrangement.

CONTENTS

NOMENCLATURE

Symbol	Description	Unit
A	Area	m^2
C	heat capacity rate	W/ K
C^*	heat capacity ratio	----
C_p	specific heat at constant pressure	$J/kg\ .\ K$
D	cylinder diameter	m
d	tube diameter	m
f	friction factor	----
G	mass flux of the air based on the minimum flow area	$kg/m^2\ .s$
h	heat transfer coefficient	$W/m^2\ .K$
k	thermal conductivity	$W/m\ .K$
L	effective length of tube	m
L_X	heat exchanger width	m
L_Y	heat exchanger height	m
L_Z	heat exchanger length	m
$\dot{m}$	mass flow rate	kg/s
n	number of water passes	-----
N	engine speed	rpm
N_{tube}	number of tube rows	-----
N_{total}	total number of tubes	-----
Nu	*Nusselt* number	-----
NTU	number of transfer units	-----
P	pressure	Pa
ΔP	pressure drop	Pa

PS	length of the piston stroke	*m*
$\dot{Q}$	heat transfer rate	*W*
R	Crank radii	*m*
r_v	compression ratio	-----
r_p	*pressure ratio*	-----
$\bar{R}_f$	*gas constant*	*J/kg.K*
Re	*Reynolds* number	-----
s	entropy	*J/kg.K*
S	tube spacing (pitch)	*m*
St	Stanton number	-----
S_v	surface area of heat transfer to volume ratio	m^2/m^3
T	temperature	^oK
t	time	*s*
T_i	time interval	*s*
U	overall heat transfer coefficient	$W/m^2.K$
V	volume	m^3
V_d	dead volume	m^3
V_{total}	total volume of the specimen	m^3

Greek symbol

μ	dynamic viscosity	*Pa.s*
ρ	density	kg / m^3
ψ	Porosity	-----
ε	heat exchanger effectiveness	-----
Θ	impact angle of the stream lines	*degree*
θ	crank angle	*degree*
γ	phase angle	*degree*
η	engine efficiency	------

Subscripts

a	air side	*min*	minimum value
ave	average value	*n*	transverse
C	compression	*o*	outer
cl	clearance	*p*	longitudinal
ch	charging	*R*	regenerator
E	expansion	*Sch*	Schmidt
f	working fluid	*s*	tube surface
H	heater	*T*	total
i	inner	*w*	water side
K	cooler	1	inlet
max	maximum value	2	exit