DESIGN OF TURBINE RECUPERATOR FOR POWER SAVING AND REDUCTION OF EXHAUST GASES POLLUTION

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ABSTRACT

A 250W turbo charger comprising of a radial turbine, a centrif ugal compressor with reverse flow combustor coupled to an electric generator to produce power is under development and testing stage at the laboratory. Also a swirling reverse flow recuperator has been designed and fabricated for integration with the turbo charger to recover waste heat coming out of the radial turbine and to improve thermal efficiency of the engine cycle. This paper covers the design, analysis, fabrication and testing of radial turbine and recuperator, with the design objective of higher effectiveness and lower pressure loss.The swirling recuperator with a cylinder surface has been designed and analyzed for stresses due to thermal and structural loads. A test rig set up with compressed air lines and space heaters (combustor) has been designed and fabricated. Simulated engine design conditions such as flow, pressure and temperatures were applied and thermal effectiveness and pressure drop across the model are evaluated, validating the theoretical predictions.A full scale swirling recuperator has also been fabricated for integration with the radial turbine.

Key words: Combustor -Turbocharger –Recuperator-Electric Genarator.

INTRODUCTION

A turbo charger is an emerging class of global power generation technology. They play an important role in evolving power generation both for standalone and for combined cycle application with fuel. Carrillo, R.A.M., (2010). Turbo charger consists of a generator, compressor, combustor, radial turbine and a swirling recuperator, all function together to generate power for small scale utilization. The swirling recuperator is an important component to achieve high thermal efficiency. C.F.McDonald., (2003).

The basic technology used in a turbo charger is derived from diesel engine technologies and automotive designs. Most turbo charger units are currently designed for continuous duty operation and are recuperated to obtain high thermal efficiency. They also have good fuel flexibility. The small radial turbines enable small energy users to generate their own electricity to secure power supply even at peak load periods also at power shortage. They also have advantages over the conventional power systems in compactness, silent running with low emission, multi fuel capabilities; vibration free with low maintenance and moderate to very high fuel utilization efficiency using waste heat recovery. Choi, H.-J., (2013).

The Oak Ridge National laboratories, USA have done a research program to evaluate the thermal efficiency dependence on material selection for various parametric variations of compressor pressure ratio and turbine inlet temperature (fig.1). E.Utriainen, B. Sunde´n., (2012).

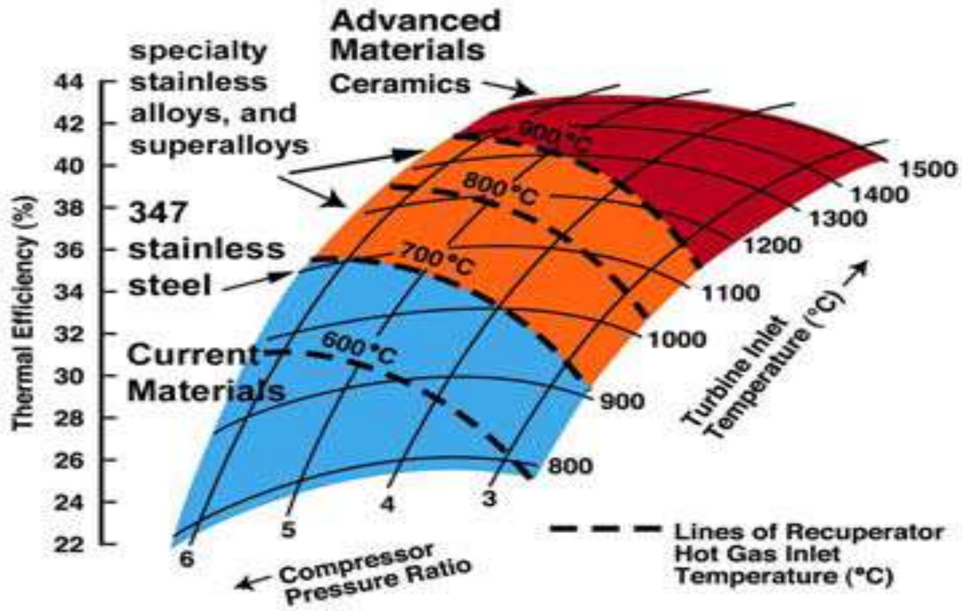

Fig. (1): The thermal efficiency dependence on material selection for various parametric variations of compressor pressure ratio and turbine inlet temperature.

A good recuperator requires minimum weight / volume, high thermal effectiveness, low pressure loss, high reliability and durability. Effectiveness is one of the key thermal performance parameter as indicated in fig (2), which shows the thermal efficiency variation. Fu, L., (2011).

To achieve the highest possible effectiveness, a counter flow recuperator must be used.

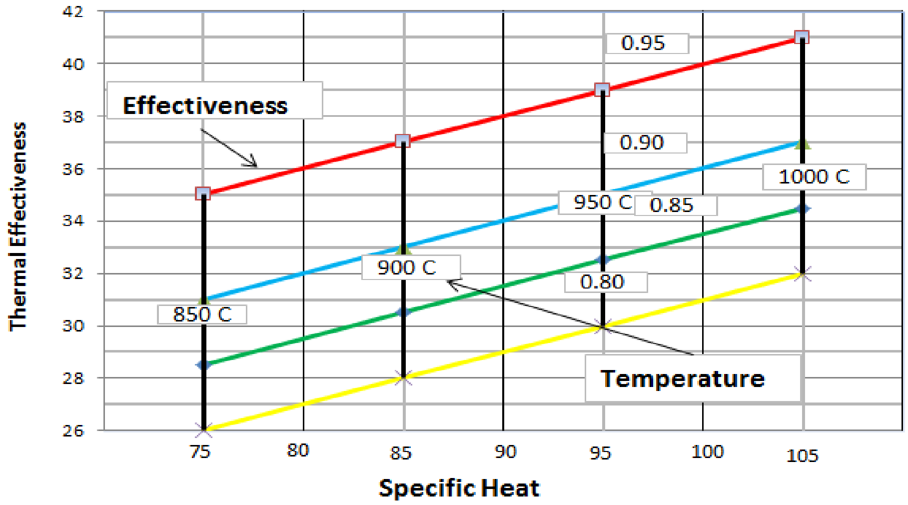

Fig. (2): The completely mapped mesh component

Major factors affecting the effectiveness are overall heat transfer coefficient and large heat transfer area; the same influenced by a number of factors [6, 7] like configuration of gas/air as sages, Reynolds number, Nusselt number and duct geometry. Another key performance parameter is the total relative pressure loss. Gunter, E.G.and Chen, W.J., (2005).

PROBLEM DEFINITION

The problem is increasing the pollution in the air and increasing the fuel consumption to generate electricity.

This paper covers the design, analysis, fabrication and testing of turbocharger, recuperator, and cyclone-scrubbers, with the design objective of higher effectiveness and lower pressure loss. Commercial package (ANSYS) was used for modeling, meshing and analysis geometry of recuperator.

Simulated engine design conditions such as flow, pressure and temperatures were applied and thermal effectiveness and pressure drop across the model are evaluated, validating the theoretical predictions. Also design, fabrication and analysis cyclone-scrubbers.

AIM OF THE RESEARCH

The aim of this research is to design some devices that can help in controlling this pollution protecting the environmental quality and keep it with the required standards and save power by using heat recovery. In the process of the heat recovery, the exhausted heat can be re-used in many applications such as heating cold air, cold water, drying, coating, produce electricity and improving engine performance.

METHODOLGY

The research includes analytical and experimental analysis of the problem:

1. The Analytical part of the present work includes:

Design the exhaust gas heat exchanger (swirling recuperator cylinder) is used to recover heat energy by inter supply air through (swirling recuperator cylinder) to become hot air.

Design turbocharger besides (swirling recuperator cylinder) used heat recovery to rotate the shaft of turbine to generate electric current, rotation motor, blower or compressor.

Combined cyclone and scrubber with the modification of (Cyclone-Scrubber) by fixed all moving part and addition helical rings to reduce pollutants from the waste and harmful to the environment associated with the various combustion processes.

2. The experimental work includes three parts:

A. First experimental part of the present study: Manufacture and testing of a heat exchanger (swirling Recuperator Cylinder) and providing the measurement devices to measure various temperatures and flow rates premade.

B. Second part of the experimental study: Manufacture and testing turbocharger fixed beside heat exchanger (Swirling Recuperator Cylinder) to rotate blower, generator and compressor.

C. Third part of the experimental study: Manufacture and testing the apparatus (Cyclone-Scrubber) is reduced the exhaust gas pollutants and produced a new environment friendly exhaust.

With the data provided, a complete 3D model of the recuperator is created. The basic methodology involved in creating the component is as follows:

- Key points are created for the given dimensions or recuperator.
- Lines are created by joining these key points.
- Area and volumes are created by joining these lines and using Boolean operations.

DESIGN OF RECUPERATOR

1) DESIGN PROCEDURE: The objective of this design problem is to determine the recuperator effectiveness and pressure drop for both the air and gas sides for the specified conditions and the basic heat transfer and flow friction characteristics of the surface. The required Recuperator for the turbo charger is designed for the given specifications and the parameters are derived from a radial gas turbine cycle for optimum performance.

The following parameters are used for the thermal analysis:

1. Thermal conductivity of material k = 20.14 W/m2 K
2. Number of swirling = 9 & 7
3. Fin thickness = 0.0005 m
4. Air and gas side angle of passage are assumed as 60º and 55º respectively see fig (4).
5. Outlet temperature of gas side and air side are assumed so that it is easy to calculate the log mean temperature difference as the temperature is varying along the length.

T_{in} = 877K and T_{out} = 543K

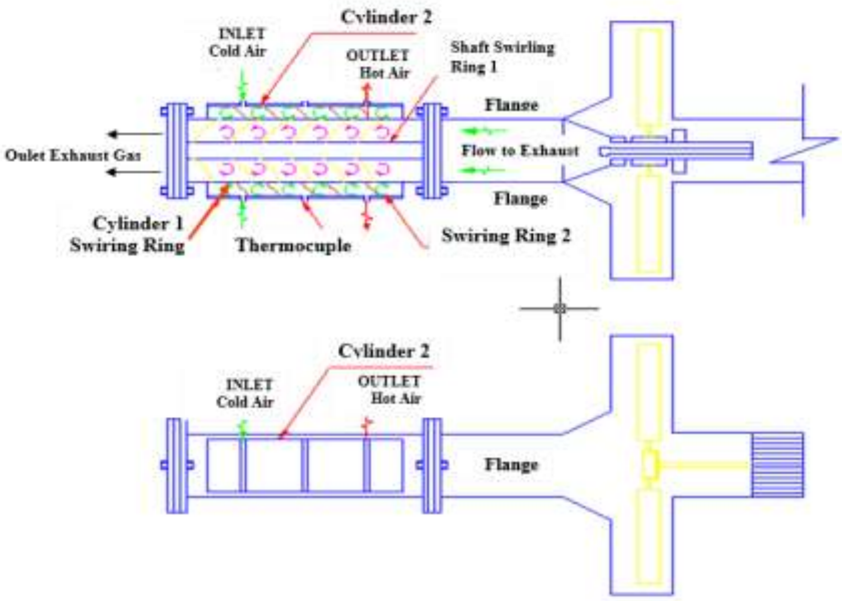

Fig (3) presents a working drawing of the design

2) CORE DIMENSIONS OF THE RECUPERATOR:

Outer diameter of recuperator $Do = 0.23m$

Inner diameter of recuperator $Di = 0.17m$

Length of recuperator $L = 0.80m$

Air inlet ellipse major axis $a = 0.00825m$

Air inlet ellipse minor axis $b = 0.00330m$

Mass flow rate of air	0.1279 kg/s
Air side inlet pressure ($p_{ai)}$	303.9 k_{pa}
Air side inlet temperature	432k
Mass flow rate of gas	0.1289 kg/s
Gas side inlet pressure ($p_{gi)}$	100 k_{pa}
Gas side inlet temperature	955 k

The steps in the design & analysis require the determination of the following factors:

1. Fluid properties.
2. Heat transfer and free flow areas.
3. Reynolds number.
4. Friction factor.
5. Pressure drop.
6. Overall coefficient of heat transfer.

3) **RECUPERATOR (SWIRLING CYLINDER) MODELLING:** For the present work the model is created in the ANSYS itself. In order to create a complete 3D model of the recuperator the data available are the 2D drawing and assembly drawing of the recuperator.

4) **MATERIALS FOR RECUPERATOR:** Steel 37 is used as a material for fabrication of swirling recuperator. Holmes et al., (2004).

5) **BOUNDARY CONDITIONS:** More precisely, boundary conditions in a model must represent everything in the operating environment that is not explicitly modeled. Basically, loads are used to represent inputs to the system of interest. These can be in the form of the forces, moments, pressures, temperatures, or accelerations, whereas constraints on the other hand are typically used as reactions to the applied loads.Non-uniform temperature load, which forms the body load and the recuperator, is subjected to pressure load of 1bar in the hot gas passage and 3bar in air passage.

(6) THERMAL ANALYSIS & STRUCTURAL ANALYSIS: A thermal analysis calculates the temperature distribution and related thermal quantities in a system or component. Typical thermal quantities of interest are the temperature distributions; the amount of heat lost or gained thermal gradients and thermal fluxes. After applying the loads and boundary conditions, the model is solved as a steady state static model. Philip J. Maziasz et al., (2002).The results of the analysis are shown and discussed in the results and discussions section.

RESULTS AND DISCUSSIONS

Design calculations have been carried out for the swirling primary surface counter flow type recuperator for the given specifications and results of the analysis are highlighted below.

Parameter	Air side	Gas side
Reynolds Number	4235	4056
Pressure drop k_{pa}	2.871	3.148
Heat transfer coefficients w/m^2	62.7	82.2
Heat transfer area m^2	0.78	0.78
Effectiveness	70%	

The pressure drop on the air side was 1.8% and pressure drop on the gas side was 6% so total pressure drop in recuperator was 7.8% which was well within the acceptable range of 8%. The overall heat transfer coefficient is one of major parameters, which influence the effectiveness.

The heat transfer coefficients obtained for both air and gas sides were low and so this resulted in an increase in effectiveness. The amount of heat transferred and effectiveness also depends on area of heat transfer. From the design results, the obtained values of area of heat transfer are high i.e. 0.78 m2 so this also contributes to increase of effectiveness. The effectiveness of the recuperator was about 70%, the reason behind this high effectiveness is the high heat transfer coefficients, high heat transfer areas and high compactness of the recuperator. Compactness is the ratio of heat transfer surface area to enclosed volume.

RESULTS OF EXPERIMENTS

The measured performance data along with derived values of effectiveness in percentage and percentage of pressure loss in both air and gas side of model with the percentage variation of design mass flow parameter are listed below. Temperatures were measured to an accuracy of 1 degree centigrade and mass flow rates are set.

Temperatures were measured to an accuracy of 1 degree centigrade and mass flow rates are set to an accuracy of +/- 3 % tolerance see table (1).

Table (1): the variation of effectiveness with gas input mass flow parameter

% design mass flow	% air pressure loss	Model air inlet temp 0C	Model air outlet temp 0C	m/Tm/pmx10-7 (kg/sec/k/N/m2)		% gas pressure loss	Model gas inlet temp 0C	Model gas outlet temp 0C	Effectiveness%
50	1.83	30.4	44.1	2.16	4.97	0.23	590	481	37.55
60	2.42	30.37	43.0	2.54	6.13	0.29	588.5	490.5	34.85
70	3.50	30.35	42.6	3.03	7.25	0.39	588.5	494.5	33.70
80	4.87	30.33	42.15	3.48	8.47	0.50	588.5	497.5	32.50
90	5.72	30.25	42.2	3.8	9.38	0.60	589	499	32.69
100	6.90	30.2	41.4	4.06	10.40	0.74	586	505	30.85
110	8.56	304	423.5	4.72	11.19	0.82	669	588.5	32.74

It is observed +/- 3% variation in effectiveness which is obvious with variation in setting mass flow mentioned above. An average effectiveness obtained from experiments is approximately 70% coinciding with the design value see fig (4).

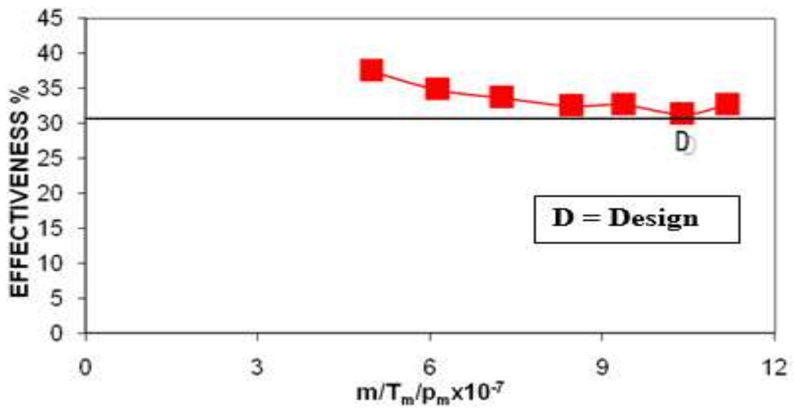

Fig (4) Effectiveness with input mass flow parameter (gas)

Shows variation of percentage pressure loss with gas input mass flow parameter for both air and gas sides. Pressure loss on the air side varies within 1% and the pressure loss in gas side is little higher. This is due to gas entering into the model through the swirling cylinder passage which is not true in actual prototype giving additional loss. Fig (5) & (6)

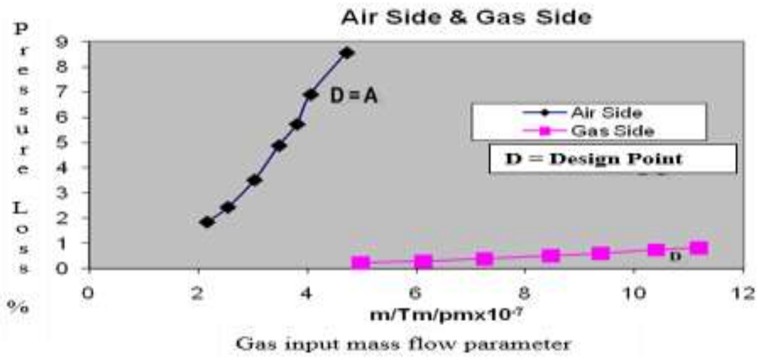

Fig (5) Pressure loss with gas input mass flow parameter

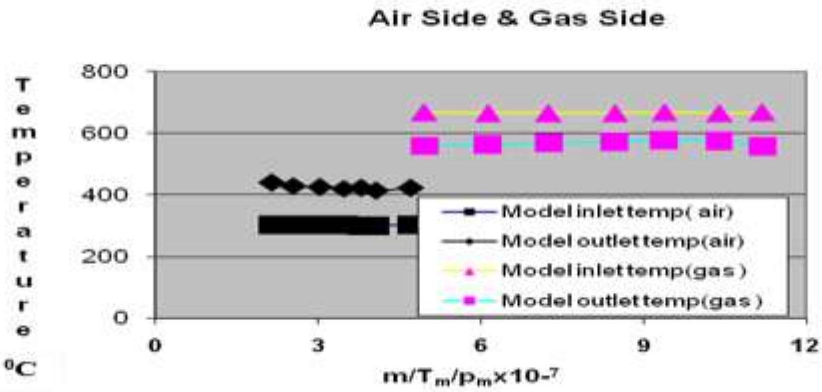

Fig (6) Variation of temperature with input mass flow parameter

Show the variation of temperatures at both inlet and outlet of model for air as well as gas side with mass flow parameter. The results clearly indicate scope for space improvement in effectiveness by increasing the surface area and turbulence levels for future studies.

TURBOCHARGING

1) **Introduction:** Turbochargers are a class of turbo machinery intended to increase the power of internal combustion engines. This is accomplished by increasing the pressure of intake air, allowing more fuel to be combusted.

In the late 19th century, Rudolf Diesel and Gottlieb Daimler experimented with pre-compressing air to increase the power output and fuel efficiency. The first exhaust gas turbocharger was completed in 1925 by the Swiss engineer Alfred Buchi who introduced a prototype to increase the power of a diesel engine by a reported 40%. The idea of turbo charging at that time was not widely accepted. However, in the last few decades, it has become essential in almost all diesel engines with the exception of very small diesel engines. Utriainen, E., (2001).

2) **Turbocharged Engines:** Turbocharged engines made a comeback during the oil shortage in the early 70's due to their inherent increase in fuel efficiency. The advances in rotor dynamic analysis using up-to date computation technology have made the dynamics of a turbocharger's rotor-bearing system a rich area for investigation. Vendors are now looking for more dynamically stable turbochargers to benefit business and increase customer satisfaction. More contributions are needed to have optimum design stability, while assuring continued low cost production. They also require a high level of reliability and efficiency in order to be cost-effective.

There are several ways to reduce the price of turbochargers; the easiest way is to keep the design as simple as possible.A common design assembly in an automotive turbocharger consists of a simple inboard bearing mounting arrangement with a radial outflow compressor and a radial inflow turbine on a single shaft see fig (7).

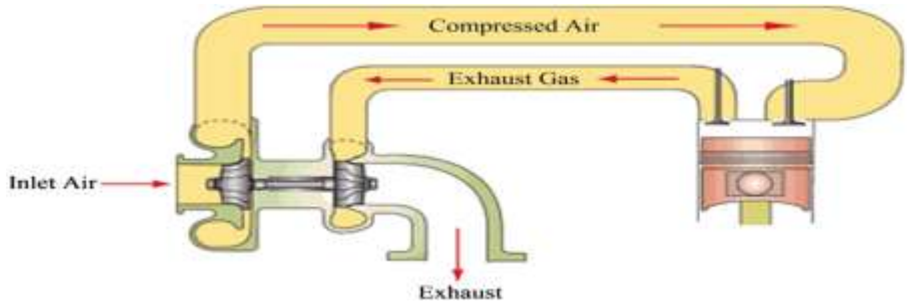

Fig (7): typical turbocharger space with compressor wheel and turbine

<u>Thermodynamic processes consist of:</u> Housing-Compressor-Recuperator-Combustion chamber –Turbine - Exhaust-gas heat-exchanger

<u>X- Rapid Design and Flow Simulations for Turbocharger Components:</u>

<u>(1) Simulated Cases:</u> Three Impellers, One Volute

1. Impeller without Tip Clearance
2. Impeller with 0.2 mm Tip clearance
3. Impeller with 0.4 mm Tip clearance

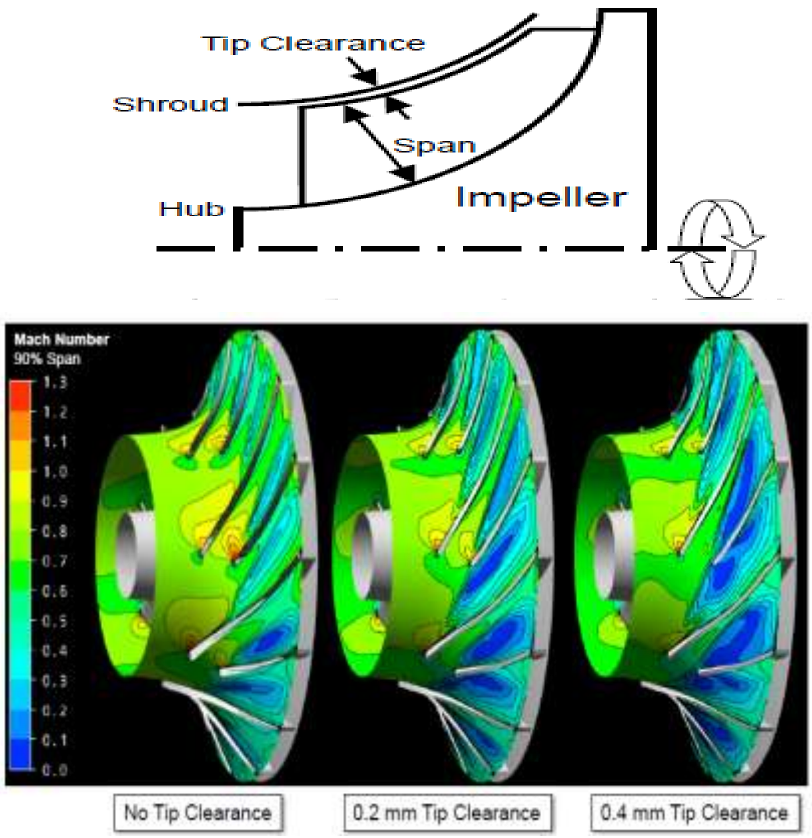

Fig. (8): Differences tip clearance in Mach number Distribution

(2) Performance:

1- Efficiency& mass flow rate

- No tip clearance
- mm tip clearance
- 0.4 mm tip clearance

Table (2): Relation between isentropic efficiency & mass flow rate at different tip clearance.

No tip clearance					
Isentropic Efficiency %	0.78	0.80	0.63	0.47	0.25
Mass flow rate kg/s	0.25	0.32	0.41	0.43	0.44
0.2 m_m tip clearance					
Isentropic Efficiency %	0.78	76	0.61	0.45	0.24
Mass flow rate kg/s	0.27	0.31	0.38	0.41	0.42
0.4mm tip clearance					
Isentropic Efficiency %	0.77	0.76	0.60	0.43	0.42

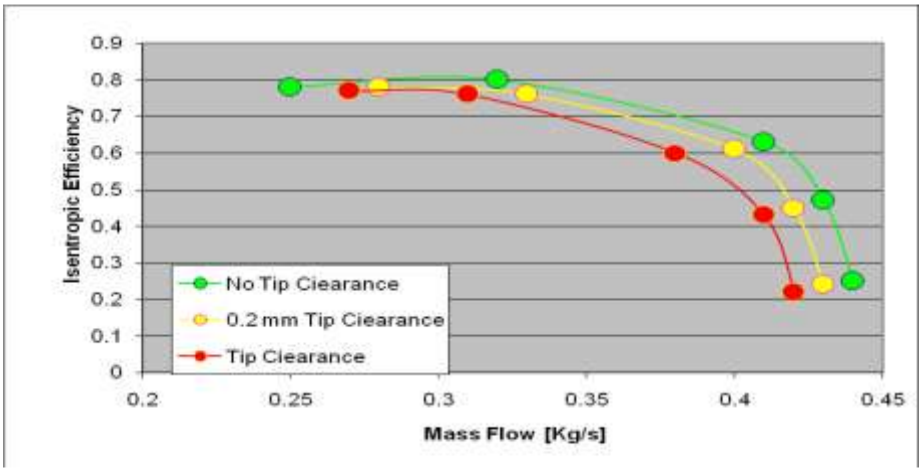

Fig. (9): Relation between isentropic efficiency & mass flow rateat different tipclearance

2- Total pressure ratio & mass flow rate:

Table (3): Relation between total pressure ratio & mass flow rate at different tip clearance

No tip clearance						
total Pressure ratio	2.2	2.1	2	1.75	1.38	1.20
Mass flow rate (kg/s)	0.26	0.32	0.36	0,4	0.43	0.44
0.2mmtip clearance						
Total pressure ratio	2.13	1.97	1.60	-	1.39	1.20
Mass flow rate (kg/s)	0.28	0.33	0.38	-	0.41	0.43
0.4mm tip clearance						
Total pressure ratio	2.07	1.96	1.60	1.38	1.19	-
Mass flow rate (kg/s)	0.27	0.31	0.37	0.40	0.42	-

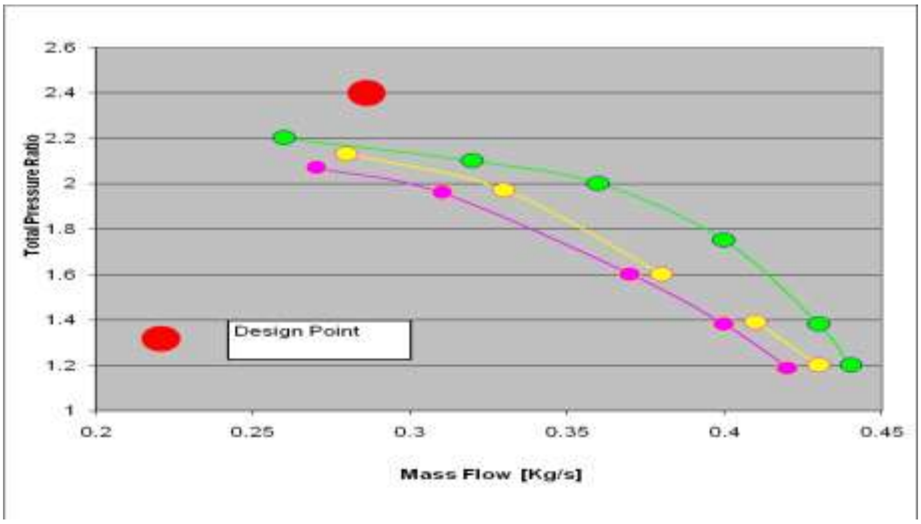

Fig. (9): relation between total pressure ratio & mass flow rate

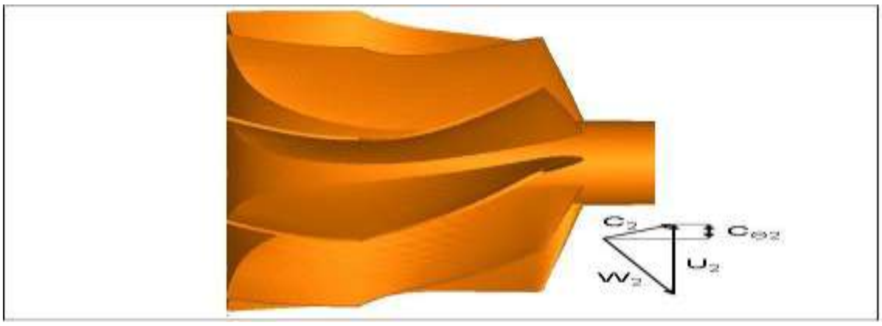

Fig. (10): Rotor inlet and outlet velocity triangles.

Another way to increase the efficiency of the turbine is to use back swept blades, with an inlet tip diameter of 90 mm.

They studied three different back sweep blade angles; 0, 15, and 30. At design condition, the efficiency was almost equally, while at off-design conditions, the efficiency was improved by 2% for the 30 back sweep angle blade due to a strongly reduced tip vortex at the leading edge.

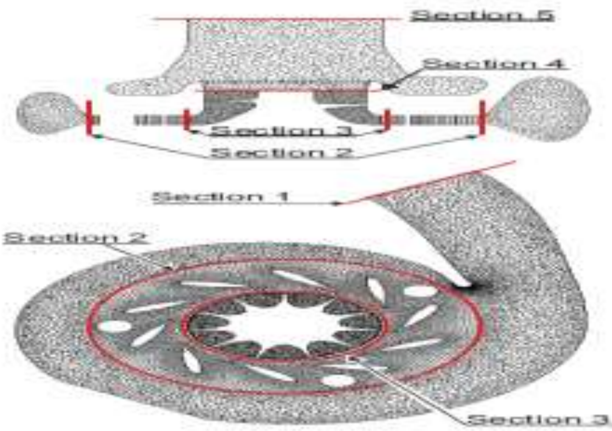

Fig. (11): Schematic representation of the turbine geometry and 3D computational mesh.

Table (4): Turbocharger Specifications

Engine capacity (L)	Up to 7
Output range (hp)	100 to 310
Air flow(max)	0.46 kg/s
Length (mm)	250
Width (mm)	240
Height (mm)	220
Mass(kg)	17

Table (5): shows Thermal source main data.

Parameter	Burner
Mass flow rate (kg/s)	0.15
Exhausts temperature (K)	1123
Pressure (kPa)	290
Average composition (per cent by volume)	CO = 0.041; CO2 = 2.74; O2 = 17.14, CxHy ≤ 0.03

In order to find out the feasibility of running, a DC dynamo connected by the turbo charger. The engine was allowed to run at different speeds the output of the generator was also noted.

Table (6): Relation between pressure& mass flow rate and power

Pressure(KPa)	Mass flow rate(kg/s)	Power (W)	Exhausts temperature(K)
25= 0.25 bar	0.0183	442	1123
50 = 0.5 bar	0.0367	884	
100 = 1 bar	0.0692	6394	
120 =1.2 bar	0,0789	8221	
140 = 1.4 bar	0.0877	9925	
160 =1.6 bar	0.0957	11504	
180 = 1.8 bar	0.102	13466	
210 =2.1 bar	0.112	15590	
230 = 2.3 bar	0.122	16981	
250 =2.5 bar	0.133	18512	
270 =2.7 bar	0.144	20043	
290 = 2.9 bar	0.154	21435	

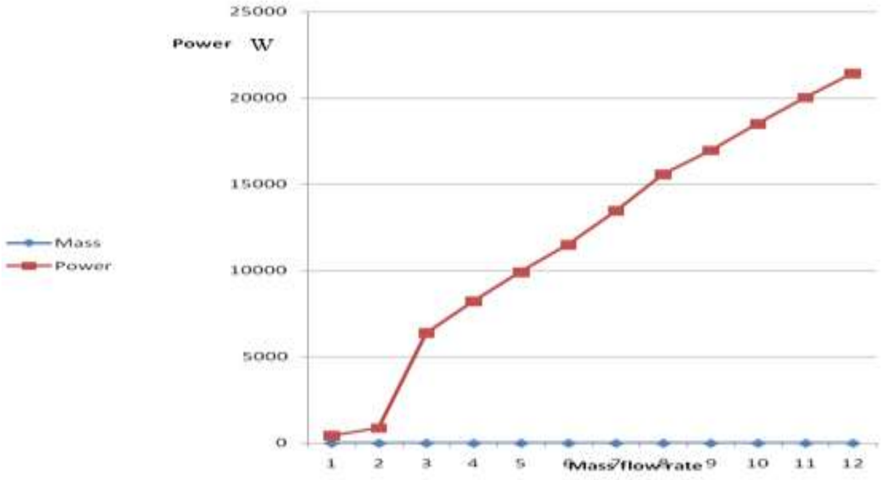

Fig (12) Relation between power& mass flow rate

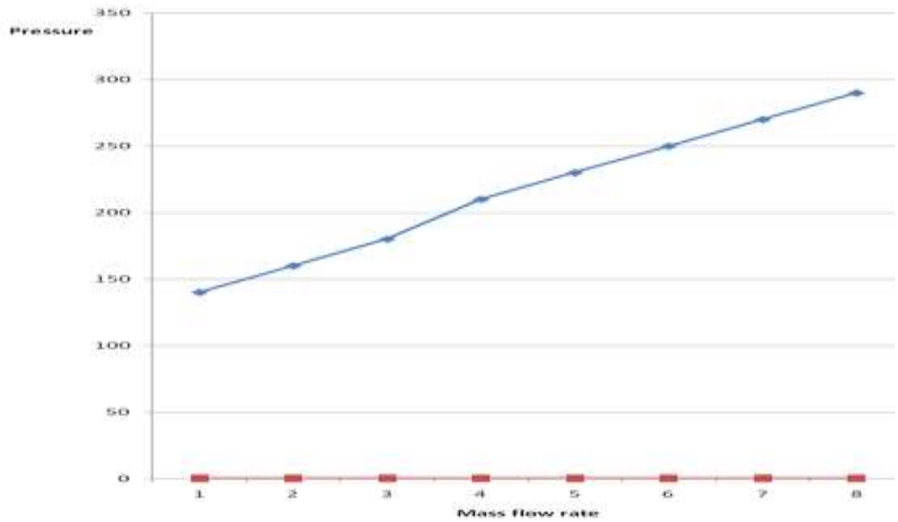

Fig. (13): Relation between pressure (Kpa) & mass flow rate (kg/s)

Power produced by the electrical machine

Power (P) = Voltage x Current

(P) = 9 V x 0.48 A = 4.3Watts

Table (7): Relation between speed& voltage&power & ampere

Engine Speed in KRPM	**2.4**	**6**	**9.6**	**14.4**	**24**	**38.4**	**56.4**	**70.8**	**85.2**	**99.6**	**115**	**130**	**140**
Output voltage of the Alternator	9	22.5	36	54	90	144	212	266	320	374	432	488	526
Power (w)	4.3	10.8	17	10.8	43	69	101	128	154	180	207	234	252
Ampere	0.48												

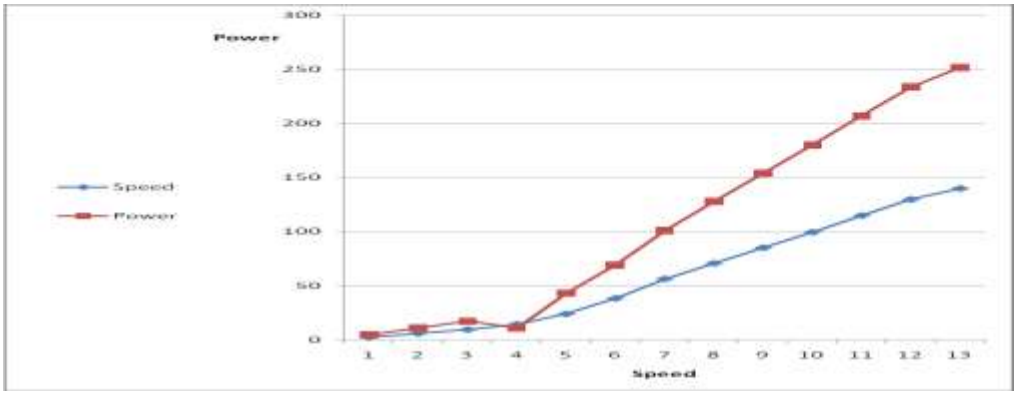

Fig. (14): Relation between rpm &power (w)

CONCLUSIONS

- The design of a recuperator for a 250W small turbine has been carried out for given operating conditions.
- Design calculations were also carried out for plain fin geometry for thermal and structural analysis.
- To begin with an annular primary surface recuperator was considered for the design.
- The results obtained from the design are, total pressure drop of recuperator was within allowable limit but effectiveness of the system was only 35%.

- Thermal and structural analysis is carried for an annular recuperator; the temperature distribution obtained from thermal analysis was applied as body load in the structural analysis.
- In average effectiveness obtained from experiments is approximately 70% coinciding with the design value.
- The variation of percentage pressure loss with gas input mass flow parameter for both air and gas sides.
- Pressure loss on the air side varies within 1% and the pressure loss in gas side is little higher. This is due to gas entering into the model through the swirling cylinder passage which is not true in actual prototype giving additional loss.
- The variation of temperatures at both inlet and outlet of model for air as well as gas side with mass flow parameter. The results clearly indicate scope for improvement in effectiveness by increasing the surface area and turbulence levels for future studies.

From turbine we notice that:

- From no tip clearance we notice that the efficient increase about 80%, but from both 0.2mm tip clearance & 0.4 mm tip clearance is lower than 80%.
- To increase the efficiency of the turbine is to use back swept blades, with an inlet tip diameter of 90 mm.
- The design point between total pressure ratio and mass flow rate= 2.4/0.28 and 1.3/0.22.
- One the speed increase the voltage is increase and the power is increase.

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CFD Computational Fluid Dynamics
HSG High Speed Generator
FEA Finite Element Analysis
ISA International Standard Atmosphere
RIT Royal institute of technology

Symbols are listed in alphabetical order. Values are listed for symbols denoting constants:

Symbol (According to SI system)	Meaning	Units
þ	Static pressure	Pa
C	Chord	mm
H	Total specific enthalpy change	j/kg
U	Internal energy	j
V	Absolute velocity	m 2/s
U	Tangential velocity	m/s
W	Relative velocity	m/s
H	Specific enthalpy; specific loss	j/kg
N	Number of blades	radial
R	Radius	mm
S	Blade pitch, gap between Blade and housing	mm
B	Blade height	mm
L	Channel length	mm
W	Width	mm
m.	Mass flow rate	Kg/s
µ	Absolute Viscosity	m 2/s
Ω	Angular velocity	s-1
F	Friction factor, equation of state	-
Re	Reynolds number	-
R	Radial of velocity	r/s
D	Diameter	mm
I	Impeller	mm
EE AA	Egyptian Environment Affairs Agency	-
IGSR	Information Gas System Research	_
EIMP	Environmental Information and Monitoring Project	–

	Fundamental parameters	Non dimensional parameters	Quasi-non dimensional parameters	Referred parameters
Mass flow	$\dot{m}$	$\frac{m\sqrt{RT_0/\gamma}}{p_0 D^2}$	$\frac{m\sqrt{T_0}}{p_0}$	$\frac{m\sqrt{T_0/T_{ref}}}{p_0/p_{ref}}$
Pressure ratio	PR	PR	PR	PR
Rot. speed	N	$\frac{N \cdot D}{\sqrt{\gamma \cdot R \cdot T_0}}$	$\frac{N}{\sqrt{T_0}}$	$\frac{N}{\sqrt{T_0/T_{ref}}}$
Efficiency	η	η	η	η

تصميم مبادل حراري تربيني لتوفير الطاقة وتقليل تلوث غازات العادم

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١) قسم هندسة القوى الميكانيكي، كلية الهندسة، جامعة عين شمس ٢) قسم هندسة القوى الميكانيكي، كلية الهندسة، أكاديمية الدراسات المتخصصة

المستخلص

يتكون الشاحن التربينى من توربينة ½ قطرية (اشعاعية) – ضاغط طارد مركزى – تدافق هواء داخل اسطوانة احتراق – مولد كهربائى لانتاج طاقة قدرها ٢٥٠ وات تحت التطور ومراحل الاختبار فى المعمل. تم تصميم وتصنيع مسترجع حرارى اسطوانى حلزونى ليكون تدفق العادم تدفق عكسى ليتكامل مع شاحن التربينى لأستعادة حرارة العادم الناتجة من التوربينات القطرية (الشعاعية) لتحسين الكفاءة الحرارية لدورة المحرك.

يتناول البحث تصميم وتحليل وتصنيع وإختبار توربينة قطرية (شعاعية) ومسترجع حرارى اسطوانى حلزونى والهدف من التصميم هو زيادة فعالية وإنخفاض فقدان الضغط. وقد أستخدمت حزمة تجارية محدودة العنصر(انسيس) للنمذجة والتأكد ولتحليل الهندسى للمسترجع الحرارى الأسطوانى الحلزونى كما تم عمل محاكاة على التدفق والضغوط ودرجات الحرارة وإنخفاض الضغط عبر النموذج والتحقق من صحة التنبؤات النظرية.

الكلمات المفتاحية: شاحن تربينى– مسترجع حرارى اسطوانى حلزونى– مولد كهربائى– محرك أو ولاعة احتراق