DEVELOPMENT OF A RECUPERATOR FOR A 10 KW MICRO GAS TURBINE

by

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ABSTRACT

A 10kW Micro Gas Turbine (MGT) comprising of a radial turbine, a centrifugal compressor with reverse flow annular combustor coupled to an electric generator to produce power is under development and testing stage at the Propulsion Division, National Aerospace Laboratories, Bangalore. Also an annular reverse flow finned recuperator has been designed and fabricated for integration with the MGT to recover waste heat coming out of the radial turbine and to improve thermal the efficiency of the engine cycle. This paper covers the design, analysis, fabrication and testing of the recuperator, with the design objective of higher effectiveness and lower pressure loss.

The annular recuperator with a plain surface has been designed and analyzed for stresses due to thermal and structural loads. A finite element commercial package (ANSYS) was used for modeling, meshing and analysis of recuperator geometry. The analysis indicate that stress levels are within yield the strength of material

A test rig set up with compressed air lines and space heaters has been designed, fabricated and erected in the Propulsion Division for testing a sector test section model with alternate three air and four gas passages for experimental evaluation. 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 SS annular plain finned recuperator has also been fabricated for integration with the MGT. Various fabrication technologies such as spark EDM, wire EDM, Laser welding along with normal conventional machining processes were used in the fabrication and construction of the recuperator.

1. INTRODUCTION

Micro and mini gas turbines are an emerging class of global power generation technology. They play an important role in evolving power generation both for stand-alone and for combined cycle application with fuel cells (1). A micro gas turbine consists of a generator, compressor, combustor, turbine and a recuperator, all function together to generate power for small scale utilization. The recuperator is an important component to achieve high thermal efficiency; it is more cost effective but much more labor intensive to fabricate.

The basic technology used in an MGT is derived from aircraft auxiliary power systems, diesel engine technologies and automotive designs. Most MGT units are currently designed for continuous duty operation and are recuperated to obtain high thermal efficiency. They also have a good fuel flexibility. The small turbines enable small energy users to generate their own electricity to secure power supply even at peak load periods also at power shortage. In many geographical places, the MGT could be employed to produce power at a competitive cost. 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. MGT with thermal efficiency of about 30% are well suited to meet energy needs of small users such as schools, apartment buildings, restaurants, offices and small business centers.

Micro and mini gas turbines are designed and developed in the power range of 5kW-500kW in many developed countries for distributed power generation. For very small simple cycle gas turbines demonstrated to date, their efficiencies have been modest. While a recuperator may be a user's option for larger industrial gas turbine, it is mandatory to achieve acceptable efficiency for small units and the majority of MGTs being developed use a recuperated cycle, as the inclusion of a recuperator is vital to obtain thermal efficiency of over 30%. (1) Also recuperators significantly muffle the noise (2, 3) which is important in many potential applications.

Extensive literature survey made on recuperators reveal the characteristics of various types, design and analysis, giving baseline for design of a recuperator suitable to our MGT. As recuperators represent 25-30% of the overall cost of MGT, selection of material plays major role considering the tensile strength, resistance to corrosion/oxidation and creep deformation. The Oak Ridge National laboratories, USA (4) 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). 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. To achieve the highest possible effectiveness, a counter flow recuperator must be used. (5) 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 passages, Reynolds number, Nusselt number and duct geometry. Another key performance parameter is the total relative pressure loss.

2. DESIGN OF RECUPERATOR

2.1 CONSTRUCTION DETAILS

A primary surface annular counter-flow type recuperator is considered for design The recuperator geometry is formed by a single spool of thin foil which is stamped and folded to make passages for air and gas flow and this folded geometry is press fitted between two hollow cylinders and welded to the ends of the air passages at both ends and closed and welded by flanges. Gas enters the recuperator at one end and flows axially along the length of the recuperator whereas air enters the recuperator from opposite end radially through elliptical holes formed at both ends of the recuperator radially, takes a longitudinal path along the length of recuperator and comes out of it again through an elliptical passage by following a radial path. Fig.3 shows a view of the annular recuperator assembly.

2.2 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 micro gas turbine is designed for the given specifications and the parameters are derived from a micro gas turbine cycle for optimum performance.

The following parameters are used for the thermal analysis

1. Thermal conductivity of material k = 20.14 $W/m^2 K$
2. Number of fins = 100
3. Fin thickness = 0.0005 m
4. Air and gas side angle of passage are assumed as 4º and 3.2º respectively
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. $T_{a0} =$ 877K and $T_{g0} =$ 543K

Core dimensions of the recuperator:

Outer diameter of recuperator D_o = 0.23m
Inner diameter of recuperator D_i = 0.17m
Length of recuperator =0.134m
Air inlet ellipse major axis a = 0.00825m
Air inlet ellipse major axis b = 0.00330m

Mass flow rate of air	0.1279kg/s
Air side inlet pressure P_{ai}	303.9kPa
Air side inlet temperature	432K
Mass flow rate of gas	0.1289kg/s
Gas side inlet pressure P_{gi}	100kPa
Gas side inlet temperature	955K

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. Prandtl Number and Stanton Number
7. Unit film conductance
8. Overall coefficient of heat transfer.
9. NTU and exchanger effectiveness

The Annexure A shows a sample design calculation and results

3. FINITE ELEMENT 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 of the recuperator fins and assembly drawing of the recuperator

METHODOLGY

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

Fig. 4 shows the completely mapped mesh component.

3.1 MATERIALS FOR RECUPERATOR

Stainless steel 347 is used as a material for fabrication of recuperator. The variation of modulus of elasticity and thermal conductivity with temperature are plotted in Fig. 5 & 6 with the respective recuperator hot gas inlet temperature and the corresponding modulus of elasticity and thermal conductivity are highlighted respectively. .

3.2 BOUNDARY CONDITIONS

In a finite element analysis, boundary conditions refer to the loads and constraints that each component or system of components experiences in its working environment. 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, where as 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. Fig. 7 & 8 show the thermal and structural loads applied to as body and surface load on the model.

3.3 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. The results of the analysis are shown and discussed in the results and discussions section.

4. TEST RIG SET UP, INSTUMENTATION AND SIMULATION

With a view to validate the predictions, an experimental test rig set up was constructed and erected in the CLOCTER(Closed Circuit Centrifugal Compressor Test Rig) Laboratory, Propulsion Division. The test rig consisted of two, 2" compressed air lines for simulating the compressor air outlet and the turbine gas outlet of the MGT. 2 kW/1.5 kW space heaters (6 Nos) of total heat capacity 9 KW were used in one of the compressed air line for simulation of the turbine gas outlet temperature. They were connected to a power input through 3 phase auto-transformers for variation of heat input with mass flow variation. A sector test section model with plain fin surface with three air and four gas passages was designed and fabricated. Fabrication methodology was as follows:- 1. Fabrication of outer/ inner sector shells with grooves by wire EDM, 2. Forming elliptical holes on the inner sector shell by spark EDM, 3. Insert the fins in between the grooves of the outer and inner shells and weld the fins to the outer/ inner shells by a Laser welding technique. A photographic view of fabricated and instrumented sector model test section is shown in Fig. 9. Valves were provided at the end of the model test section outlets to enable maintain the required flow and pressure in the inlet of the air and gas lines. D and D/2 orifice meters were designed and used in both the lines for measurement of the flow. Cr/Al thermocouples were provided at various locations and were connected to a DORIC multi channel digital temperature indicator for recording the temperatures. Static pressure tapings were made at desired locations and instrumented to Druck multi channel digital pressure indicators for measurement of pressures.

As it took considerable time to reach the steady state desired value of temperature with space heaters, the experiments were carried out keeping the major parameter temperature ratio i.e., (turbine outlet to compressor outlet) as constant with compressed air line temperature at ambient level. The other major parameters such as pressures and mass flows were kept at the designed conditions. For simulated and set mass flow variation of 50% to 110% of designed values, experiments were carried out and pressure and temperature outputs recorded to evaluate mass flows, pressure loss and effectiveness characteristics. The results are discussed in the next section.

5. RESULTS AND DISCUSSIONS

Design calculations have been carried out for the annular 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 kPa	2.871	3.148
Heat transfer coefficients W/m2	62.7	82.2
Heat transfer area m2	0.39	0.39
Effectiveness	35%	

The pressure drop on the air side was 0.9% and pressure drop on the gas side was 3.01% so total pressure drop in recuperator was 3.9% which was well within the acceptable range of 4%. 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 a decrease 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 very low i.e. $0.39m^2$ so this also contributes to decrease of effectiveness. The effectiveness of the recuperator was only 36%, the reason behind this low effectiveness is the low heat transfer coefficients, low heat transfer areas and low compactness of the recuperator. Compactness is the ratio of heat transfer surface area to enclosed volume. So compactness of the recuperator has to be increased to get the higher effectiveness. A typical value of compactness required is $700m^2/m^3$. The value used at present is $200m^2/m^3$.

5.1 RESULTS OF ANALYSIS

In order to have an assessment of the behavior of the recuperator subjected to varying aerodynamic and temperature loads, a cyclic sector model of the recuperator was considered. All the necessary boundary conditions and loadings are applied to the model and solved. Fig. 10 and Fig. 11 show the results of thermal and stress analysis contours. From the results obtained the following conclusions are drawn and they are as follows.

The thermal stresses are concentrated near hot end section of the recuperator. The temperature of the material at the hot end section is 680^0C and yield strength of the material at this temperature is 240 MPa. The stresses in the hot end section are 96MPa which are within yield limit and allowable limit. The deformation of material due thermal expansion is 1.52mm in the radial direction and 1.2mm in axial direction at the hot gas inlet passages and so an appropriate gap has to be provided at the flanges which can allow these expansions.

5.2 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 to an accuracy of +/- 3 % tolerance.

%design mass flow	% air Pr Loss	Model air inlet Temp K	Model air Outlet Temp K	$m\sqrt{T_{mi}}/P_{mi}$ X 10^{-7} (kg/Sec√K)/(N/m²)		% gas Pr Loss	Model gas inlet Temp K	Model gas Outlet Temp K	Effective-ness %
				Air	Gas				
50	1.83	304	441	2.16	4.97	0.23	668.8	559	37.55
60	2.42	303.7	430.3	2.54	6.13	0.29	667	564	34.85
70	3.50	303.5	426	3.03	7.25	0.39	667	569	33.70
80	4.87	303.3	421.5	3.48	8.47	0.50	667	573	32.50
90	5.72	302.5	422	3.81	9.38	0.60	668	577	32.69
100	6.90	302	414	4.06	10.40	0.74	665	575.5	30.85
110	8.56	304	423.5	4.72	11.19	0.82	669	588.5	32.74

Fig.12 shows the variation of effectiveness with gas input mass flow parameter.

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 35% coinciding with the design value. Fig.13 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 elliptical tube passage which is not true in actual prototype giving additional loss. Fig.14 shows 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 with redesign and tests with straight, cross and double cross corrugated fin configurations leading to increase in surface area and turbulence levels for future studies. The work has already been taken up and evaluation process will be carried out shortly.

6.0 CONCLUSIONS

The design of a recuperator for a 10kW MGT 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%. A 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. From the static analysis the maximum stresses were found to occur at hot gas inlet region and the stresses obtained were well below the yield limit and allowable limit. The maximum radial and axial deformation of recuperator was at the hot end flange region and this deformation was within acceptable limit. Experiments carried out with a plain fin surface sector model validates very well with our analysis for effectiveness and pressure loss measurements show variation in the range 2-8% on gas side. Further redesign and experiments are planned with straight corrugated/ cross corrugated finned sector model to improve effectiveness.

7.0 REFERENCES

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ANNEXURE-A

THERMAL DESIGN CALCULATIONS OF ANNULAR RECUPERATOR.							
INPUT DATA:-				**KNOWN QUANTITIES:-**			
Outer diameter of recuperator	D_o	0.23	m	Air side inlet pressure	P_{a1}	303900	N/m2
Inner diameter of recuperator	D_i	0.17	m	Inlet temperature of air	T_{ai}	432.71	K
Number of fins	N_f	100		Outlet temperature of air	T_{ao}	877	K
Air inlet ellipse major axis	a	0.00825	m	Mass flow rate of air	m_a	0.1279	kg/s
Air inlet ellipse minor axis	b	0.0033	m	Mass flow rate of gas	m_g	0.1289	kg/s
Length of recuperator for air side	L_a	0.13	m	Gas side inlet pressure	P_{g1}	104400	N/m2
Length of recuperator for gas side.	L_g	0.13	m	Gas side inlet temperature	T_{gi}	955.32	K
Fin thickness between air and gas.	t	0.0005	m	Gas side outlet temperature	T_{go}	543	K
Air side angle of pessage	θ	4					
Thermal conductivity of material	K_w	20.76882	J/s-mK				
Circumference of outer dia.	C_o	0.722566	m				
Circumference of inner dia.	C_i	0.53407	m				
AIR SIDE CALCULATIONS:				**GAS side CALCULATIONS:-**			
Number of air passages	N_p	50		Number of gas passages	N_p	50	
Angle from the origin for air side	θ_a	4	deg.	Angle from the origin for gas side	f_{ig}	3.2	Deg
Reynold's number of air in passage	R_{ea}	4235.279		Reynolds no of gas in passage.	R_{eg}	4066.4387	
pressure drop due to air flow	P_{dp}	240.821	N/m2				
Total pressure drop due to air flow	P_{da}	*2871.674*	N/m2	Pressure drop due to gas flow	P_{dg}	*3148.36*	N/m2
Prandtl Number	N_{pr}	0.69847		Prandtl Number	N_{pr}	0.7023346	
Stanton Number	N_{st}	0.004479		Stanton Number	N_{st}	0.0044937	
Total heat transfer area of air side	A_a	0.39	m2				
Effectiveness of recuperator	e	34.12948					

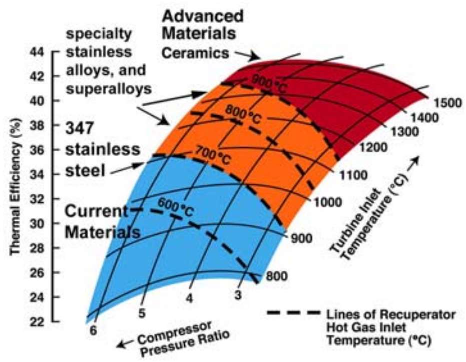

Fig.1 Recuperated Micro Turbine Efficiency

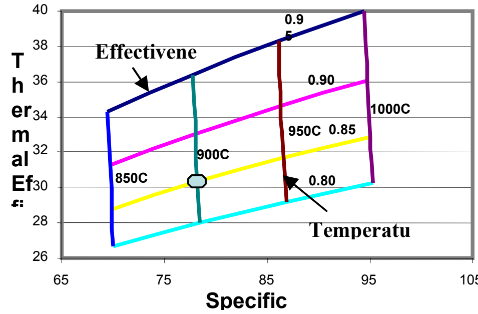

Fig.2 Variation of gas turbine efficiency with effectiveness

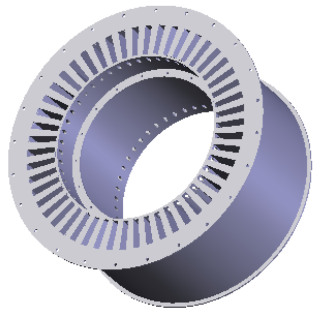

Fig.3 Annular recuperator assembly

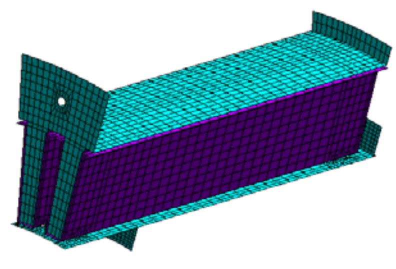

Fig.4 Meshed model nomenclature

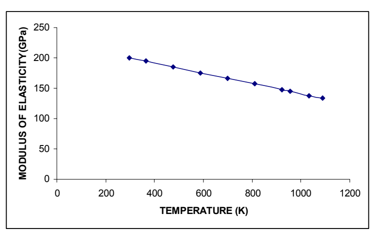

Fig.5 Modulus of elasticity with temperature(4)

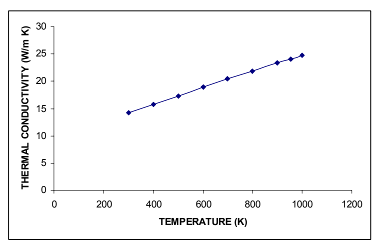

Fig.6 Thermal conductivity with temperature(4)

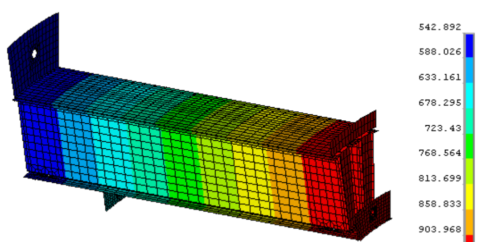

Fig.7 Temperature as body load

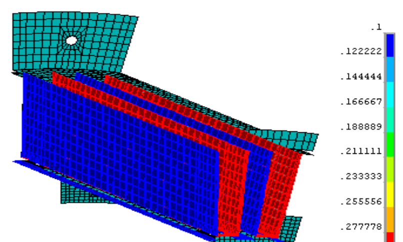

Fig.8 Pressure as surface load

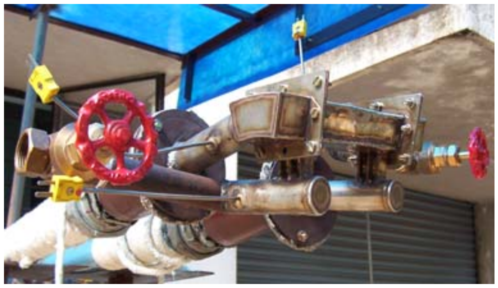

Fig 9 Photographic view of sector model

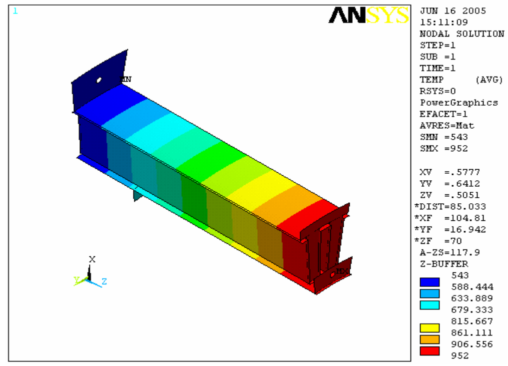

Fig10 Temperature distribution plot

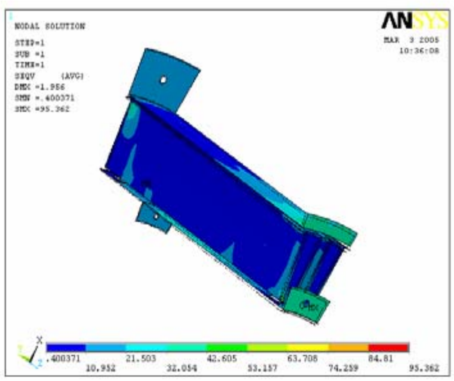

Fig.11 Von Mises stress distribution

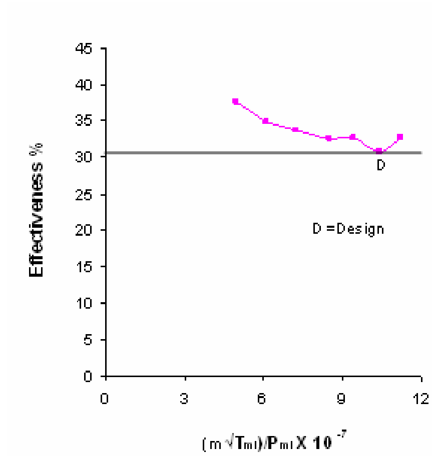

Fig.12 Effectiveness with input mass flow parameter (gas)

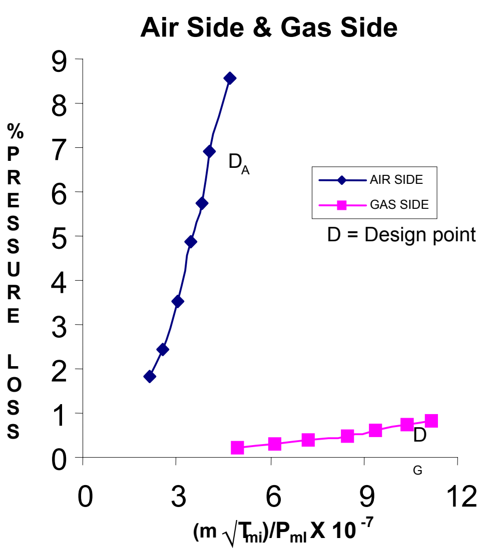

Fig.13 Pressure loss with input mass flow parameter

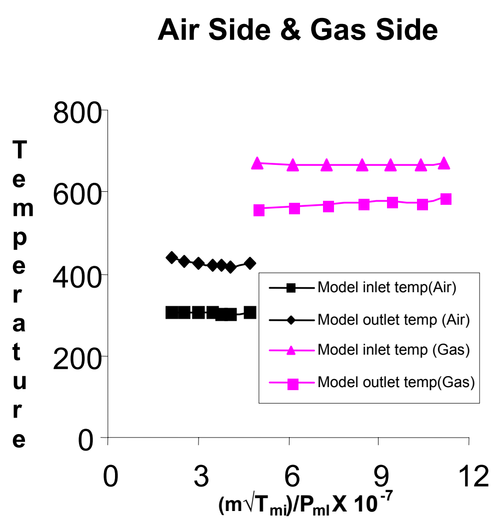

Fig.14 variation of temperature with input mass flow parameter