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Experimental Investigation On Thermo-Acoustic Refrigerator

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ABSTRACT

The importance of thermo-acoustic stream is growing every day, despite the fact that Thermo-Acoustic devices are not as efficient as conventional ones. Thermo acoustic is an environmental friendly technology as it uses gases like air and noble gases as a refrigerant. It possesses potentially high reliability due to simple structure and minimum number of moving parts, and it is reasonable. These characteristics could lead to low manufacturing and maintenance costs. Stack can be called as heart of the Thermo-acoustic Refrigeration Process. Thermo-acoustic effect takes place in a stack. The effect varies with the position of stack inside the resonator tube. In this experimentation we fixed other parameters of thermo-acoustic refrigerator (Operating Frequency, Mean Pressure, Cooling Capacity, etc.) and varied stack position inside the resonator to observe its effect on temperature difference across the stack.

***Keywords*- Thermo-Acoustics, Refrigeration, Stack**

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I.INTRODUCTION

The importance of thermo-acoustic refrigeration is growing every day. Despite the fact that thermo-acoustic refrigerators are not as efficient as conventional ones, there are many other advantages to consider. With continuous technological advancements, thermo-acoustic refrigerators may one day become as efficient as or more efficient than conventional refrigerators. Unfortunately till the date, thermo-acoustic refrigerators are not highly efficient when compared with vapor compression refrigerators. At the present time, the most efficient thermo-acoustic refrigerators have Carnot efficiencies between 20-30% which is half that of the conventional counterpart.

Acoustics has an extensive history, but it was not until the mid-1800s that the connection between heat and sound was made. Physicists determined that the interaction between sound and air is too small to measure, however sound waves in pressurized gas have the power to create intense thermo-acoustic interactions. These interactions have the ability to power potentially energy efficient thermo-acoustic devices, motivating the continued interest in thermo-acoustic improvement. Thermo-acoustic Refrigeration effect depends on the position and thickness of the stack. The thickness can be increased up to some optimum length after which increase in length will decrease the effect; same case is with stack position. Thus it is highly essential to obtain thickness and position of the stack which will provide optimum effect.

Nomenclature

B, Blockage Ratio
Cp, Specific Heat Capacity of Air, kJ kg^{-1} K^{-1}
D, Inner Diameter of Pipe, m
δ_k, Thermal Penetration Depth, mm
δ_k, Viscous Penetration Depth, mm
K, Thermal Conductivity, $Wm^{-2}K^{-1}$
μ, Viscosity of Working Medium
ρ, Density of Working Medium
ω, Angular Frequency of the Sound Wave
ΔT_m, Temperature Difference, K
P_m, Mean Pressure, bar
P_0, Dynamic Pressure, bar
M, Mach No.
a, Sonic Velocity, m/s
R_y, Reynolds No.
D, Drive Ratio
K_s, Thermal Conductivity of Stack Material
f_k, Rott's Function
r_h Hydraulic Diameter, mm
x_s, Centre Stack Position
L_s, Stack Length
Γ, Critical Temperature Gradient

Π, Stack Perimeter
Q_{cn}, Normalized Cooling Capacity, W
W_n, Normalized Power, W
σ, Prandtl No.

II. THERMO-ACOUSTIC THEORY

The first and second laws of thermodynamics place an upper bound on the efficiency of heat engines. If T_H and T_C are the hot and cold thermal reservoirs, respectively, and Q_H and $Q_C$the associated heat flows with W the work flows, in the usual case of cyclic engines operation, $Q_H$and Q_C and W are time averaged powers. The operation is assumed steady-state, so that the time-averaged state of the engine itself does not change. The first law of thermodynamics states that $Q_H - Q_C = W$ The second law states that the entropy generated by the system must be positive or zero. Since the engine is in (time averaged) steady state, the net entropy increase in the reservoirs is $\frac{Q_H}{T_H} - \frac{Q_C}{T_C} \geq 0$, then Performance of the thermo-acoustic refrigerator $COP \leq \frac{T_C}{T_H - T_C}$. Thermo-acoustic systems operate in a similar manner.

Two types of devices are developed viz. Standing Wave and Travelling Wave Devices.

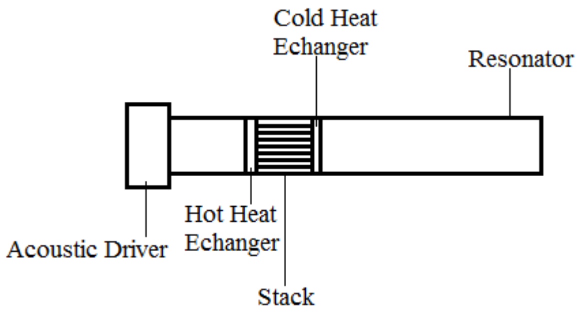

Fig 2.1 Schematic Diagram of Standing Wave Device

Standing-wave devices, such as the representation shown in Figure 2.1, generate useful cooling by pumping heat from the cold heat exchanger to the hot heat exchanger. Pumping heat requires the input of work in the form of acoustic power from the loudspeaker on the left end of the resonator. The cold heat exchanger absorbs heat from an external load, while the hot heat exchanger rejects heat to the surroundings or other heat sink.

The Stack serves to increase the surface area in contact with the oscillating gas and acts as temporary storage for the heat as it is transported from the cold exchanger to the hot exchanger. If the parallel plates in the stack are spaced too far apart, much of the oscillating gas will simply expand and contract adiabatically and will not pump any heat. If the plates are spaced too close together, the thermal contact between the oscillating gas and the stack will be too good, and the gas will expand and contract isothermally and, again, will not pump any heat. The stack does not necessarily have to be made from plates. (For example, Stack made from bundled glass tubing.)

The stack material is also important. The goal is to provide the gas a place to temporarily store the heat as it is shuttled along the length of the stack, as well as to insulate between the hot and cold ends to minimize heat conduction, which would reduce the efficiency of the device. Therefore, the stack is typically made from a material with high heat capacity that does not conduct heat well

Unlike standing-wave engines and heat pumps, where pressure and position are substantially in phase, pressure and velocity are substantially in phase in traveling-wave engines and heat pumps. This will add a more complicated acoustic network than the simple resonators in standing-wave devices. The other major difference is that traveling-wave devices have a regenerator rather than a stack.

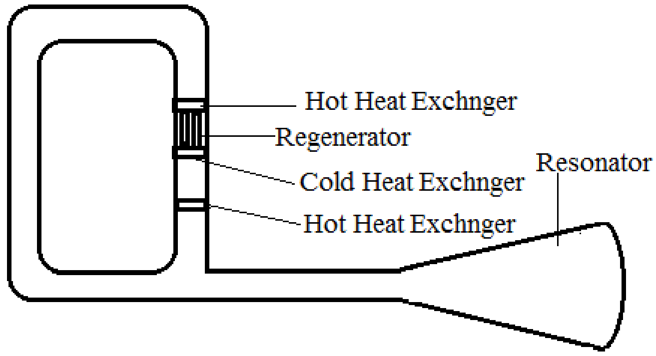

Fig 2.2 Schematic Diagram of Travelling Wave Device

As shown in figure 1.2, the regenerator looks nearly like a stack in most cases. The difference is that the spaces in which the gas oscillates are smaller than in a stack because the goal here is to have very good thermal contact with the gas. A major advantage of traveling-wave devices is that they can theoretically achieve Carnot efficiency equal to compression Refrigeration System, while standing-wave devices are limited to 40-50% of Carnot (Berson et al. 2008). Approaching Carnot efficiency can only be done, however, by having the cooling capacity of the machine approaching zero (Huang et al. 2005).

III. WORKING OF THERMO-ACOUSTIC REFRIGERATOR

The cycle, in heat Refrigerator mode, can be explained by imagining a single parcel of gas: The parcel is compressed adiabatically in the open space outside of the regenerator by the pressure wave, heating up as this happens. As the parcel is displaced and enters the hot end of the regenerator, it rejects heat into the regenerator. Outside of the other end of the regenerator, the parcel is adiabatically expanded, cool down. As the parcel is displaced back through the regenerator, it absorbs heat from the cold end of the regenerator. The heat transport is caused by the parcel of gas depositing the heat of compression to the hot end of the regenerator and absorbing that same amount of heat from the cold end of the regenerator. The thermal expansion and contraction that takes place inside of the regenerator cancel each other out, so there is no net heat transfer in those steps. Unlike stack-based devices, where the displacement of a parcel of gas is a small fraction of the length of the stack, the parcel of gas in a regenerator-based device must be displaced clear through the regenerator and heat exchangers to operate properly.

III. EXPERIMENTAL SETUP

Experimental setup consists of the Loudspeaker Driven Standing Wave thermo-acoustic Refrigerator as shown in fig 3.1. As parameters we are working on are Stack Position and Stack Length our test section is confined to stack holder section in the schematicdiagram(fig 3.1). Experimental setup is consists of three sections Acoustic driver and casing, Stack Holder and Stack, Resonator, Controls and Data Recorder. Design procedure is discussed as bellow. We are using Tijani's approach for designing standing wave thermo-acoustic refrigerator.

IV. DESIGN OF THE DEVICE

A. Design of the Stack

Considering our area of interest, we remain with three main stack design parameters: the center position x_n, the length L_{sn}; and the cross-sectional area A.

The length and position of the stack can be normalized by $\lambda/2\pi$. The thermal and viscous penetration depths can be normalized by the half spacing in the stack y_0. The cold temperature or the temperature difference can be normalized by T_m. Since δ_k, δ_v are related by the Prandtl number σ, this will further simplify the number of parameters. Olson and Swift proposed to normalize the acoustic power W and the cooling power Q_c by the product of the mean pressure P_m; the sound velocity a, and the cross-sectional area of the stack A = P_maA. The amplitude of the dynamic pressure can be normalized by the mean pressure. The ratio p_0/p_m is called the drive ratio D. In practice the stack material can be chosen so that the thermal conductive term in the heat flow expression can be neglected. In this case the parameters of the stack material do not have to be considered in the performance calculations. The porosity of the stack, sometimes called blockage ratio and defined as

$$B = \frac{y_0}{y_0 + l}$$

is also used as a dimensionless parameter for the geometry of the stack. The thermal and viscous penetration depths are given by

and
$$\delta_k = \sqrt{\frac{2K}{\rho_m C_p \omega}}$$

Where K is the thermal conductivity, μ is the viscosity, ρ is the density, C_p is the iso-baric specific heat of the gas, and ω is the angular frequency of the sound wave. The resultant normalized parameters are given an extra index n. The number of parameters can once more be reduced, by making a choice of some operation parameters, and the working gas

$$\delta_v = \sqrt{\frac{2\mu}{\rho_m \omega}}$$

This area is equal to the resonator cross-section at the stack location. By using data for the gas parameters we first optimize the stack geometry parameters by optimizing the performance expressed in terms of the coefficient of performance (COP) which is the ratio of the heat pumped by the stack to the acoustic power used to accomplish the heat transfer. This leads to the determination of x_n and L_{sn} Then the required cooling power will be used to determine the cross-sectional area A. Once these parameters are determined we can design the resonator. The dissipated acoustic power at the cold side of the resonator forms an extra heat load to the cold heat exchanger. This load, and the required cooling power, will form the total heat load that the cold heat exchanger has to transfer to the stack. The first law of thermodynamics states that the total heat load at the hot heat exchanger is the sum of the heat pumped by the stack and the acoustic power used by the stack to realize the heat transfer process. The hot heat exchanger has to remove this heat from the hot side of the stack. The driver has to provide acoustic power for driving the thermo-acoustic heat transport process and compensating for all viscous and thermal dissipation processes in the stack, heat exchangers, and at the resonator wall.

Use of the dimensionless parameters like the ratio of the temperature gradient along the stack and the critical temperature gradient is done in order to reduce calculation work.

$$\Gamma = \frac{\Delta T_m}{\Delta T_c}$$

can be rewritten as

$$\Gamma = \frac{\Delta T_{mn}}{(\gamma - 1) B L_{ns}} \tan X_n$$

The stack perimeter Π, can be expressed as function of the cross-sectional area as

$$\Pi = \frac{A}{y_0 + l}$$

The expressions of the heat flow and acoustic power can be rewritten in a dimensionless form by using the dimensionless parameters

$$Q_{cn} = -\frac{\delta_{kn} D^2 \sin 2X_n}{8\gamma(1+\sigma)\Lambda} \times \left(\frac{\Delta T_{mn} \tan X_n}{(\gamma-1) B L_{sn}} \frac{1+\sqrt{\sigma}+\sigma}{1+\sqrt{\sigma}} - \left(1+\sqrt{\sigma}-\sqrt{\sigma}\delta_{kn}\right) \right)$$

And

$$W_n = \frac{\delta_{kn} L_{sn} D^2}{4\gamma} (\gamma-1) cos^2 X_n \times \left(\left(\frac{\Delta T_{mn} \tan X_n}{B L_{sn}(\gamma-1)(1+\sqrt{\sigma})\Lambda} - 1 \right) - \left(\frac{\delta_{kn} L_{sn} D^2}{4\gamma} \frac{\sqrt{\sigma} sin^2 X_n}{B\Lambda} \right) \right)$$

where Λ is defined as

$$\Lambda = 1 - \sqrt{\sigma}\delta_{kn} + \frac{1}{2}\sigma\delta_{kn}$$

The thermal conductivity term in Eq. (7) has been neglected. The performance of the stack is expressed in terms of the coefficient of performance

$$COP = \frac{Q_{cn}}{W_n}$$

B. Optimization of the Stack

In the COP calculations, the data shows the performance calculations as a function of the normalized stack length L_{sn}, for different normalized stack positions x_n. The normalized position $x_n = 0$, corresponds to the driver position (pressure antinode). In all cases the COP shows a maximum. For each stack length there is an optimal stack position. As the normalized length of the stack increases, the performance peak shifts to larger stack positions, while it decreases. This behavior is to be understood in the following way: A decrease of the center position of the stack means that the stack is placed close to the driver

This position is a pressure antinode and a velocity node. Equation shows that the viscous losses (second term on the right) are proportional to the square of the acoustic velocity. Thus decreasing the velocity will result in a decrease of the losses and hence a higher COP. It is concluded that the maximum cooling power may be expected at a position roughly halfway between the pressure antinode and pressure node. The COP peak, the cooling powers, and the acoustic power, calculated at the peak position, are plotted as functions of the stack length. The cooling power and the acoustic power increase while the COP decreases as function of the stack length and thus as function of the normalized stack center position. For a normalized stack length above 0.35, the COP is smaller than one. Considering the above remarks and for practical reasons, we have chosen for a normalized stack center position of $x_n = 0.22$ in our setup. To achieve optimum performance this requires a stack length of $L_{sn} = 0.22$. Expressed in terms of the normal stack center position and length, we have $x_s = 8$ cm and $L_s = 8.5$ cm. This is equivalent to place the hot end of the stack at a distance of 3.75 cm from the driver. Under these conditions the dimensionless cooling power is $Q_{cn} = 3.7 \times 10^{-6}$. Since the required cooling power is 4 W, Equation leads to a cross-sectional area A = 12 cm^2 which is equivalent to a radius of r = 1.9 cm for a cylindrical resonator. To pump 4 W of heat, the stack uses 3 W of acoustic power (COP = 1.3)

C. Design of Resonator

The resonator is designed in order that the length, weight, shape and the losses are optimal. The resonator has to be compact, light, and strong enough. The shape and length are determined by the resonance frequency and minimal losses at the wall of the resonator. The cross-sectional area A of the resonator at the stack location is determined in the preceding section. The acoustic resonator can have a $\lambda/2$ or a $\lambda/4$ length, as shown in Figs. 6(a) and (b). The viscous and thermal relaxation dissipation losses take place in the penetration depths, along the surface of the resonator. In the boundary-

$$\frac{dW_2}{dS} = \frac{1}{4}\rho_m |\langle u_1\rangle|^2 \delta_{v\omega} + \frac{1}{4}\frac{|P_1{}^2|}{\rho_m a^2}(\gamma - 1)\delta_{k\omega}$$

layer approximation, the acoustic power lost per unit surface area of the resonator is given by

Where the first term on the right-hand side is the kinetic energy dissipated by viscous shear. The second term is the energy dissipated by thermal relaxation. Since the total dissipated energy is proportional to the wall surface area of the resonator, a $\lambda/4$-resonator will dissipate only half the energy dissipated by a $\lambda/2$-resonator. Hence a $\lambda/4$-resonator is preferable. Hofler shows that the $\lambda/4$-resonator can be further optimized by reducing the diameter of the resonator part on the right of the stack.

The thermal loss increases monotonically as function of the ratio D1=D2, but the viscous losses decrease rapidly up to about D1=D2 =0.5 and then increase slowly. As a result the total loss (sum) has a minimum at about D1=D2 =0.54. Hofler and Garrett used a metallic spherical bulb to terminate the resonator. The sphere had sufficient volume to simulate an open end. But at the open end, which is a velocity antinode, the velocity is maximum so that an abrupt transition from the small diameter tube to the bulb can generate turbulence and so losses occurs

D. Cold heat exchanger

The whole resonator part on the right of the stack in Fig. 1.1 cools down so a cold heat exchanger is necessary for a good thermal contact between the cold side of the stack and the small tube resonator. An electrical heater is placed at the cold heat exchanger to measure cooling power. The length of the heat exchanger is determined by the distance over which heat is transferred by gas. The optimum length corresponds to the peak-to-peak displacement of the gas at the cold heat exchanger location. The displacement amplitude is given by

$$x_1 = \frac{u^{(1)}}{\omega} = \frac{P_0{}^{(1)}}{\omega \rho_m a}\sin(kx)$$

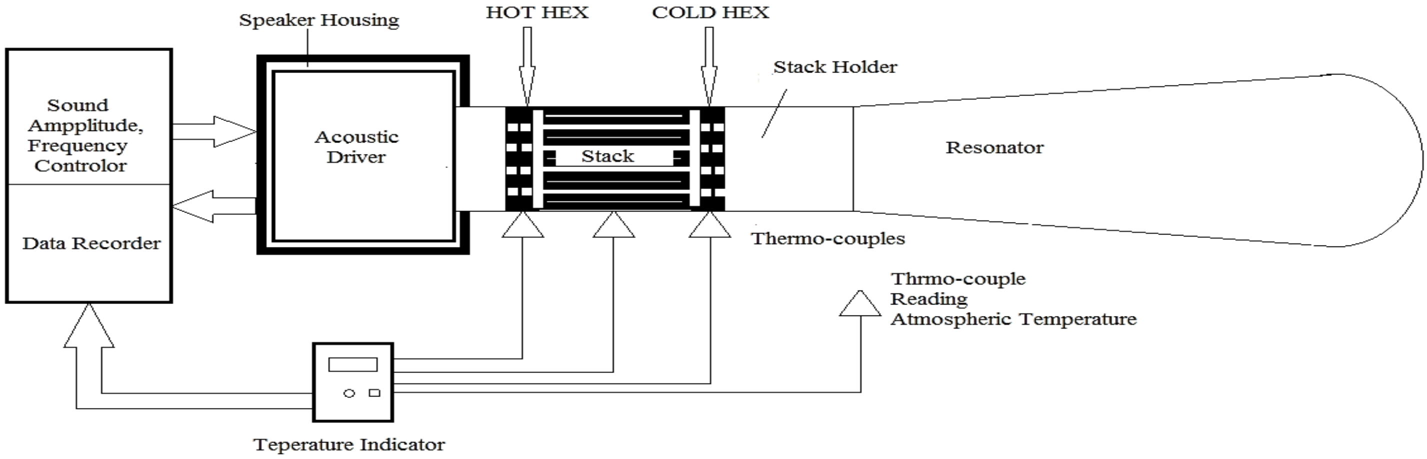

Fig 3.1 Schematic of Experimental Setup

E. Hot heat exchanger

The hot heat exchanger is necessary to remove the heat pumped by the stack and to reject it to the circulating cooling water. As discussed in the precedent section, the optimal length of the heat exchanger is equal to the peak-to-peak displacement amplitude of the gas at the heat exchanger location. But since the hot heat exchanger has to reject nearly twice the heat supplied by the cold heat exchanger, the length of the hot heat exchanger should be twice that of the cold heat exchanger (6 mm). Substituting the position of the hot heat exchanger x_n = 0.10, the length L_{hn} = 0.016 and Γ = 0 , we obtain an estimation for the acoustic power dissipated in the hot heat exchanger which is W_{hhx} = 0.33 W.

Standard Working Gas Parameters are shown in the table bellow.

Table I
Working Gas Parameters

Working Gas Parameters		Air	
Thermal Conductivity	K	0.0257	W/m.K
Sound Velocity	a	343	m/s
Ratio of Specific Heats	γ	1.48	
Heat Capacity	Cp	1005	J/kg.K
Gas Density	ρm	1.189	Kg/m³
Dynamic Viscosity	μ	0.000018	kg/s.m
Prandtl Number	Pr	0.71	

Choise of operating parameter depends on many factors like stack plate thickness stack position and working gas etc.

IV. SELECTION OF PARAMETERS

For this investigation we choose to design a refrigerator for a temperature difference of ΔT_m = 75 K and a cooling power of 4 W. In the following, we will discuss the selection of some operation parameters, the gas and stack material

A. Average Pressure

Since the power density in a thermo-acoustic device is proportional to the average pressure p_m, it is favorable to choose p_m as large as possible. This is determined by the mechanical strength of the resonator. On the other hand, δ_k is inversely proportional to square root of P_m, so a high pressure results in a small δ_k and small stack plate spacing. This makes the construction difficult. Taking into account these effects and also making the preliminary choice for helium as the working gas, the maximal pressure is 12 bar. We choose to use 10 bar. To minimize the heat conduction from the hot side of the stack to the cold side, we used a holder made of a material with low thermal conductivity (e.g. POM-Ertacetal).

B. Frequency

As the power density in the thermo-acoustic devices is a linear function of the acoustic resonance frequency an obvious choice is thus a high resonance frequency. On the other hand δ_k is inversely proportional to the square root of the frequency which again implies a stack with very small plate spacing. Making a compromise between these two effects and the fact that the driver resonance has to be matched to the resonator resonance for high efficiency of the driver, we choose to use a frequency of 400 Hz.

C. Dynamic Pressure

The dynamic pressure amplitude P_0 is limited by two factors namely, the maximum force of the driver and non-linearities. The acoustic Mach number, defined as

$$M = \frac{p_0}{\rho_m a^2}$$

has to be limited to M ≈ 0.1 for gases in order to avoid nonlinear effects.

From many experimental studies on the structure of turbulent oscillatory flows, it has unanimously been observed that transition to turbulence in the boundary layer took place at a Reynolds number (R_y) based on Stokes boundary-layer thickness, of about 500 – 550, independent of the particular flow geometry (pipe, channel, oscillating plate). Since we intend to design a refrigerator with moderate cooling power we will use driving ratios D < 3%, so that M < 0.1 and R_y< 500

E. Working Gas

Helium is widely used as working gas. The reason for this choice is that helium has the highest sound velocity and thermal conductivity of all inert gases. Furthermore, helium is cheap in comparison with the other noble gases. A high thermal conductivity is wise since δ_k is proportional to the square root of the thermal conductivity coefficient K.. But air can also be used as refrigerant effectively at it is freely available

F. Stack Material

The heat conduction through the stack material and gas in the stack region has a negative effect on the performance of the refrigerator. The stack material must have a low thermal conductivity K_s and heat capacity C_s larger than the heat capacity of the working gas, in order that the temperature of the stack plates is steady. The material Mylar is chosen, as it has low heat conductivity (0.16 W/m K) and is produced in many thicknesses between 10 and 500 μm.

G. Stack Geometry

There are many geometries which the stack can have: parallel plates, circular pores, pin arrays, triangular pores, etc. The geometry of the stack is expressed in Rott's function f_k . This function is given for some channel geometries in the literature [11]. It is shown that the cooling power is proportional to Im (- fk) . Fig shows the real and imaginary parts of f_k for some geometry as functions of the ratio the hydraulic radius r_h and the thermal penetration depth.

The hydraulic radius is defined as the ratio of the cross-sectional area and the perimeter of the channel. Pin arrays stacks are the best, but they are too difficult to manufacture. Hence, we choose to use a stack made of parallel-plates. We note that for parallel-plate stack r_h = y_0.The selection of a frequency of 400 Hz, an average pressure of 10 bar, and helium as working gas, determines the thermal and viscous penetration depths. Using Eqs. (2.2) and (2.3) we have for our system δ_k = 0.1 mm and δ_v = 0.08 mm. As can be seen from Fig. 2 for a parallelplates stack Im(-fk) has a maximum for $r_h/\delta_k = y_0/\delta_k = 1.1$. Since the spacing in the stack is $2y_0$, this means that the optimal spacing is 0.22 mm.

Using an analogous analysis Arnott et al. obtained an optimal spacing of about 0.26 mm. In order not to alter the acoustic field, it has stated to use a spacing of 2 δ_k to 4 δ_k. We choose to use a spacing of about 0.3 mm. The experimental study of the effect of the pore dimensions in the stack on the performance of the refrigerator is reported elsewhere. The remaining stack geometrical parameters are the center stack position x_s, the length of the stack L_s; and the cross-sectional area A. These parameters are determined from the performance optimization of the stack

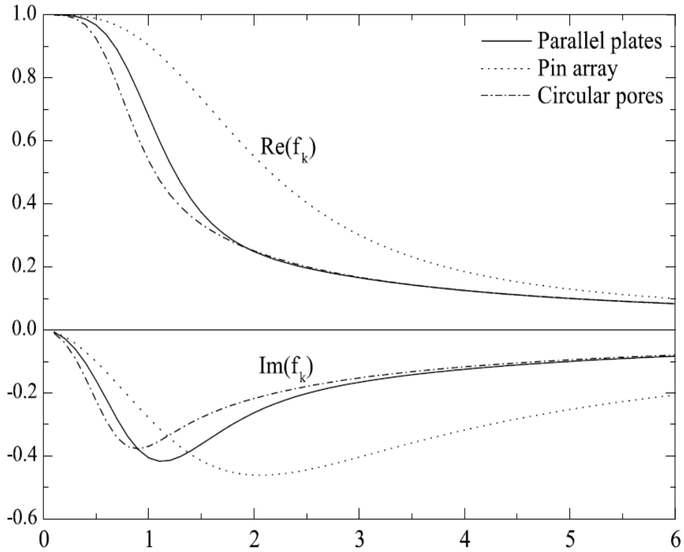

Fig 4.1 Real and Imaginary Part of Rott's Function (Tijani 2002)

Table II
Operating Parameters

Operating Parameter		Overall System	
Operating Frequency	f	400	Hz
Average Pressure	Pm	6	bar
Dynamic Pressure	P0	0.12	bar
Drive Ratio	D	0.02	
Mean Temperature	Tm	313	K
Temperature Difference	ΔTm	15	K
Normalized Temp. Difference	ΔTmn	0.047923	
Cooling power	Q_c	5	W

H. Acoustic driver

The driver has to provide acoustic power to compensate power used by the stack to transfer heat and dissipated in the different parts, thus

$$W_t = W_s + W_{res} + W_{chx} + W_{hhx}$$

thus Electromagnetic speaker of 12 W capacity is used for the same

Table III
Stack Parameters

Stack Parameters		Mylar	
Thermal Conductivity	Ks	0.16	W/m.K
Heat Capacity	Cs	1100	J/Kg.K
Blockage Ratio	B	0.75	
Density	ρ_s	1400	m^3/kg
Stack Plate Spacing	$2y_0$	0.3	mm
Stack Plate Thickness	$2l$	0.1	mm
Stack Radius	r	2.15	cm
Stack length	Xs	81.4785	mm

V. EXPERIMENTAL SETUP AND EXPERIMENTATION

A. Stack

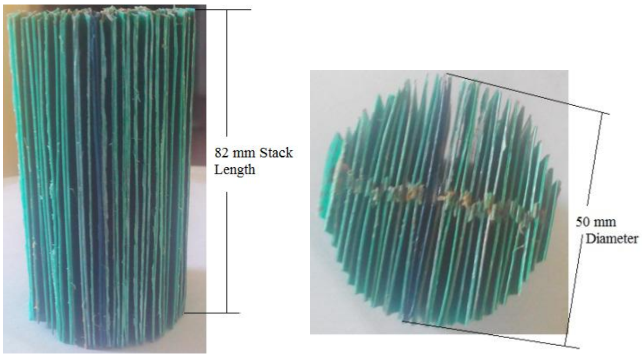

Fig 5.1 Fabricated Stack

Parallel Pate Stack arrangement is used as it is simpler to design and construct. Total Stack Length is taken as 82 cm according to the design calculation with 50 mm diameter. We kept stack spacing to 0.5 mm .

B. Resonator

There are two possible arrangements of resonator as $\lambda/2$ and $\lambda/4$. Among that $\lambda/4$ arrangement is having higher losses compared to $\lambda/2$ arrangement but $\lambda/4$ can be considered over $\lambda/2$ arrangement as is simpler to construct.

Figure 5.2 shows $\lambda/4$ type resonator with total length of 40 cm which is closed at one end and other end is connected to stack holder. For Experimentation different section with different lengths are used.(viz. 5 cm, 10 cm, 15 cm, 20 cm, 25 cm , 30 cm , 35 cm , 40 cm)

Fig 5.2 Resonator And Stack Arrangement

C. Experimental Setup

Fig5.3 Actual Experimental Setup

As we can see in the diagram 400 Hz Sine Wave Audio Frequency is Provided to the 12 W Loud-Speaker.Which Generates Sound Waves which Passes through the Stack and resonator section. Thermo-acoustic Effect is carried out in the stack which produced temperature difference between two ends of the stack which is measured with the help of thermocouples.

VI. RESULT

For different stack position experimentation was carried out. Setup was kept running for 15 min and then reading were note down from the temperature indicators provided. Stack Position is varied in the range of cm. Temperature Difference is tabulated with respect to the stack position as follows

Stack Position	Temperature Difference
0	4.7
5	4.6
10	5.2
15	4.3
20	1.2
25	-1.4
30	-4.4
35	-3.2

Mean Pressure. From which Generalized conclusion can be drawn.

The same is plotted using MS Excel Tool

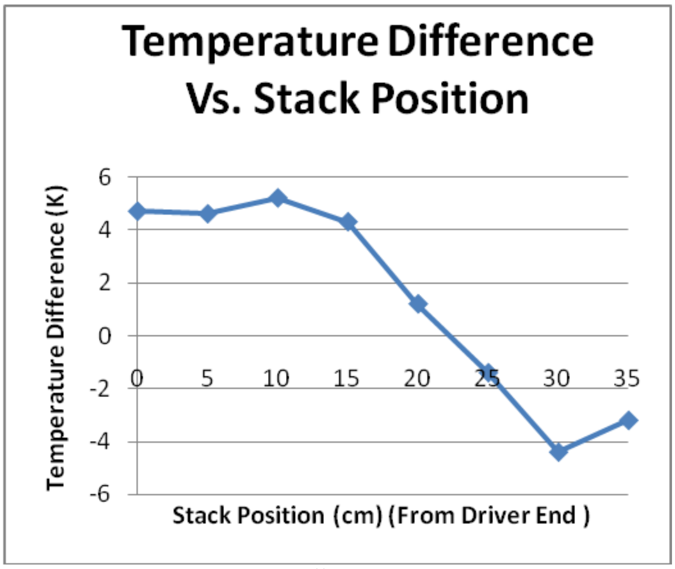

Fig 6.1 Temperature Difference Vs. Stack Position

VI. CONCLUSION

Stack is the key component in the thermo-acoustic refrigerator system . Stack position is one of the parameter which is essentially should be considered during design of the system. As we can see with air as a working medium maximum temperature difference is occured at the stack position of 10 cm (from the Driver End). The distribution shows that temperature difference initially increases with the increase in the distance then at 10 cm stack position it is maximum then it starts decreasing. If stack position is kept changing one position occurs where negative temperature difference is occurred (Temperature increases)

Further this experimentation can be repeated using different working gas medium, operating frequency

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