Optimizing the Performance of a Standing Wave Loudspeaker Driven Thermoacoustic Heat Pump

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Abstract— This paper investigates the design and optimization steps of a thermoacoustic refrigerator. Matlab code will be used for optimizing the stack length and its position. DeltaEC version 6.3b11 will be used to do the code used for simulating the refrigerator to identify the optimized operating conditions such as the mean pressure and the oscillating pressure. Behavior of changing the operating conditions effect on the performance is discussed.

Keywords— DeltaEC, Optimization, Performance, Refrigeration, Thermoacoustics.

NOMENCLATURE

A	Resonator area, [m^2]
a	Sound velocity, [m/s]
B	Porosity, …
D	Driver ratio, …
d	Resonator diameter, [mm]
e	Specific internal energy, [kJ/kg]
f	Frequency, [Hz]
h	Specific Enthalpy, [kJ/kg]
K	Thermal conductivity, [W/(m.K)]
L	Length, [m]
P	Pressure, [Pa]
Pr	Prandtl number, …
$\dot{Q}$	Thermal power, [W]
T	Temperature, [K]
U	Volume flow rate, [m^3/s]
v	Velocity, [m/s]
$\dot{W}$	Acoustic power, [W]
X	Stack position, [mm]
y	The Half stack spacing, [mm]

Greek Letters

β	Thermal expansion coefficient, [K^-1]
γ	Specific heats ratio, …
δ_k	Thermal penetration depth, [mm]
δ_v	Viscous penetration depth, [mm]
λ	Wave length, [m]
μ	Dynamic viscosity, [Pa. s]
ρ	Density, [kg/m^3]
θ	Phase angle, [degree]
ω	Angular frequency, [rad/s]
ΔT	Temperature Difference, [K]

Subscripts

0	Spacing
1	Amplitude
k	Thermal
m	Mean
n	Dimensionless
s	Stack
v	Viscous

1 INTRODUCTION

THERMOACOUSTIC refrigeration is a form of the emerging green technologies that overcomes many of the environmental problems related to the ordinary vapor compression cycles based refrigeration. Thermoacoustic refrigeration has advantages of working with environment friendly inert fluids, simple design and possibility of quiet operation [1]. Optimizing of energy systems is important to provide the world need of energy.

Thermoacoustic refrigeration system is composed of simple components as shown in fig. (1). The acoustic driver receives a signal with known frequency and power from the function generator and the amplifier respectively, after that it transmits oscillation inside the resonator with the desired signal to make thermoacoustic effects inside the resonator. Acoustic wave with high pressure side heats up the oscillating near by gas particles and the low pressure side cools down it.

Tijani [2] designed a thermoacoustic refrigerator that reachs a low temperature of -65 °C. A mixture of helium and xenon having 30 % xenon by volume was observed to give relative performance of 17 % and the results were validated using DeltaEC.

The Los Alamos group [3] invented the DeltaEC program (Design Environment for Low-Amplitude ThermoAcoustic Energy Conversion) that is a computer program which calculates the thermoacoustic device performance and helps for equipment design to the desired performance. DeltaEC is used in simulating thermoacoustic devices widely.

Zoontjens et al. [4] made an experimental study of a loudspeaker driven refrigerator. The tools used in the optimization of the thermoacoustic refrigerator are discussed. The performance of the device obtained from measurements and DeltaEC modelling are compared.

Ke et al. [5] simulated numerically thermoacoustic refrigerator driven at large amplitude. The best performance is obtained at the dimensionless plate thickness from 0.28 to 0.33.

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International Journal of Scientific & Engineering Research Volume 7, Issue 9, Septemper-2016
ISSN 2229-5518

Tu et al. [6] experimentally studied the resonance frequency characteristics due to change of the working gas and the stack geometry. The resonance frequency was found to change with the gas type greatly and the stack geometry effect on the resonance was small.

Abakr et al. [7] presented an experimental study to show the effect of wave patterns and frequency on the temperature difference across the stack. They found that the square wave has a better result on the temperature difference than the other wave forms.

Namdar et al. [8] used the OpenFOAM software to simulate the thermoacoustic refrigerator. They investigated the refrigerator working at high pressure amplitude. The pressure node was observed to change and the position of the cold heat exchanger was optimized.

Piccolo [9] studied numerically the entropy generation affecting a thermoacoustic refrigerator. The stack and heat exchanger were acting as the strong causes of irreversibility.

The main contribution of this work is to present theoretical study of thermoacoustic refrigerator to optimize the stack parameters and the operating conditions.

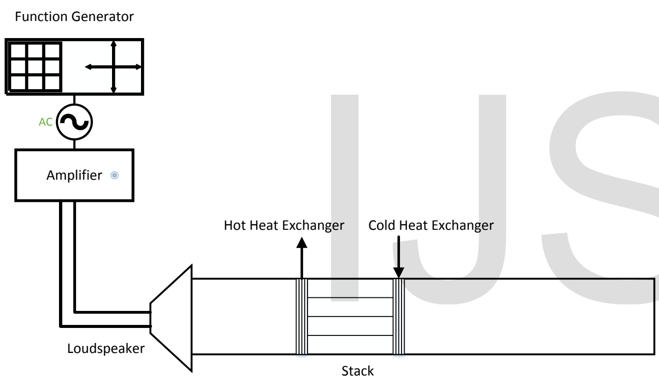

Fig. (1) Schematic view thermoacoustic refrigerator components

2 Governing Equations

The stack length, position and pore size affect the obtained cooling load and the required input acoustic power as shown in (1) and (2) for the dimensionless cooling power ($\dot{Q}_{cn}$) and the dimensionless acoustic power ($\dot{W}_n$) respectively.

$$\dot{Q}_{cn} = \frac{-\delta_{kn} D^2 \sin(2X_{sn})}{8\gamma A(1+Pr)}\left[\frac{\Delta T_{mn}\tan(X_{sn})}{L_{sn}B(1+\gamma)}\frac{1+\sqrt{Pr}+Pr}{1+\sqrt{Pr}} - (1+\sqrt{Pr}+\sqrt{Pr}\,\delta^2_{kn})\right] \quad (1)$$

$$\dot{W}_n = \frac{\delta_{kn} D^2 L_{sn}}{4\gamma}(\gamma-1)B\cos(X_{sn})^2\left[\frac{\Delta T_{mn}\tan(X_{sn})}{L_{sn}AB(1+Pr)} - 1\right] - \frac{\delta_{kn} D^2 \sin(X_{sn})^2\sqrt{Pr}}{4\gamma AB(1+Pr)} \quad (2)$$

The coefficient of performance C.O.P is the ratio between the gained cooling power and the comsumed work:

$$C.O.P = \frac{\dot{Q}_{cn}}{\dot{W}_n} \quad (3)$$

3 Design Steps

A design of thermoacoustic refrigerator required to give a cooling power of 5 W, desired temperature difference, ΔT_m= 15 K, mean operating temperature, T_m= 300 K, mean pressure, P_m= 2 bar, pressure amplitude, P_1= 2 kPa and frequency, f= 500 Hz using helium will be presented using the sequence shown in fig. (2) for designing.

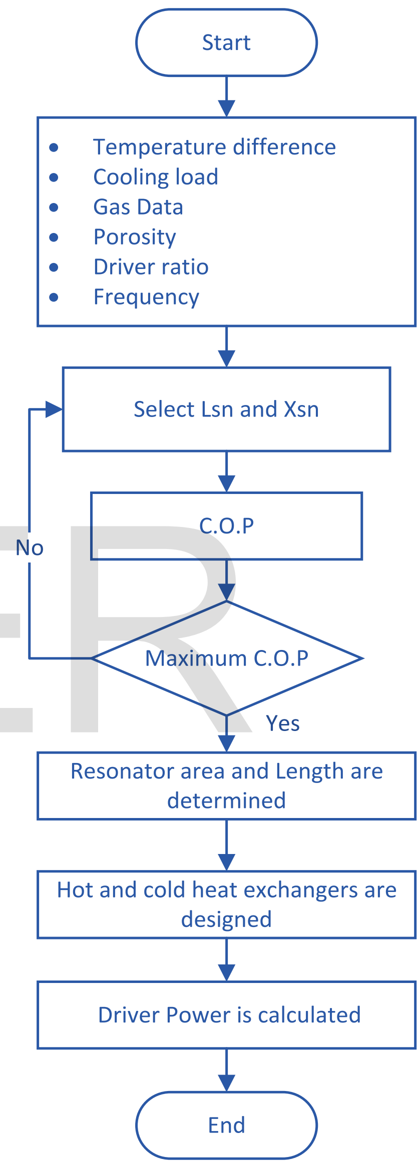

Fig. (2) Design steps for thermoacoustic refrigerator

Equation (1) and (2) are used through a matlab code to optimize the stack parameters. Fig. (3) shows the effect of dimensionless stack length and dimensionless stack position on the stack performance. From this figure; the value of the normalized stack center position is set to X_{sn} = 0.4. This is because a greater value will lead to difficulties in fabricating, so when X_{sn} = 0.4 is chosen, the dimensionless stack length, L_{sn} should be about 0.13 to give stack performance of 3.5702.

International Journal of Scientific & Engineering Research Volume 7, Issue 9, Septemper-2016
ISSN 2229-5518

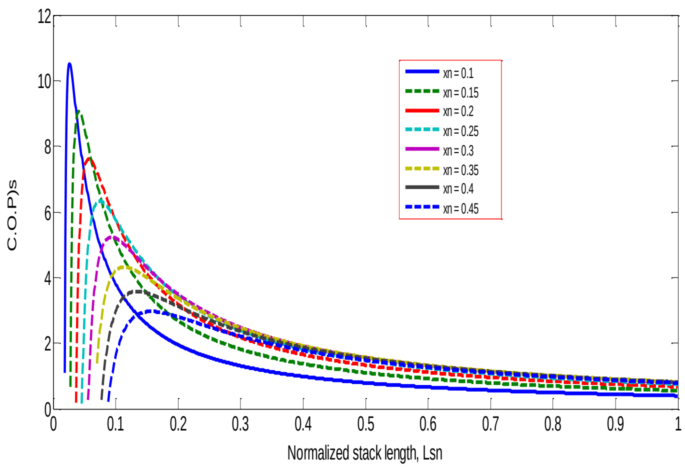

Fig. (3) Stack performance for different dimensionless stack lengths and positions

4 DeltaEC Model

The change of operating conditions effect on the coefficient of performance of the whole thermoacoustic refrigerator at different cooling loads will be presented numerically with the help of the free simulation software DeltaEC version 6.3b11 and using the default DeltaEC convergence tolerance of 10^-8 to do the code used for simulating the design. For DeltaEC program: the pressure equation is concluded from the momentum equation of fluid mechanics and the flow rate equation is concluded from the continuity equation of fluid mechanics. The equations are integrated numerically along the x coordinate to form solutions $P_1(x)$ and $U_1(x)$. DeltaEC has more complicated momentum and continuity equations that include additional effects as the vicous and thermal losses resulting from the dissipation of acoustic power along the sides of ducts. It uses different equations for the different segments to fit local conditions.

The study considers a constant cooling load, and calculate the incoming acoustic power that varies with changing the operating conditions. The boundary conditions in our theoretical model are:

- θ (P) = θ (U) = 0 at the beginning segment to enforce the resonance frequency to be obtained [2] [3].
- Pressure amplitude, P_1 = 2 kPa.

Four targets and four guesses were taken into account. Two targets for the volume flow rate, which will enforce complex flow rate U_1= 0 at the bottom end of the model; the third target is a total energy flow equals zero; this potential target appears in insulated mode and the fourth target is constant hot heat exchanger temperature of 303 K. The three targets related to the flow rate and the total energy will be done using the logical segment HardEnd. The four unknown guesses are: the hot heat exchanger load, the resonance frequency, beginning volume flow rate and the mean temperature. The results of the different thermoacoustic refrigerator design elements will be used through DeltaEC using the boundary conditions, the initial guesses and targets to begin simulation in the DeltaEC program.

5 Results

5.1 Mean Pressure Effect

Increasing the mean pressure will decrease the temperature difference as shown in fig. (4).

The driver ratio (D) is the ratio between the amplitude pressure and the mean pressure. The acoustic energy is directly proportional to drive ratio as in (2), so increasing mean pressure will decrease the driver ratio and this leads to decreasing acoustic power and thus increasing C.O.P as shown in fig. (5). The larger temperature difference is desired, but the coefficient of performance should be optimized.

5.2 Amplitude Pressure Effect

Increasing the amplitude pressure will increase the temperature difference as in fig. (6).

As the acoustic power is proportional to the driver ratio, the input acoustic power for fixed mean pressure increases with the increase in the amplitude pressure which means lower C.O.P. This is well illustrated in fig. (7).

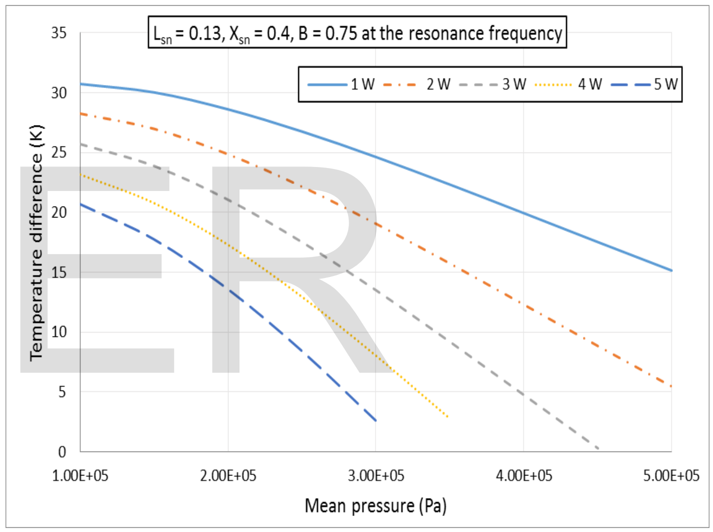

Fig. (4) Mean Pressure effect on the temperature difference

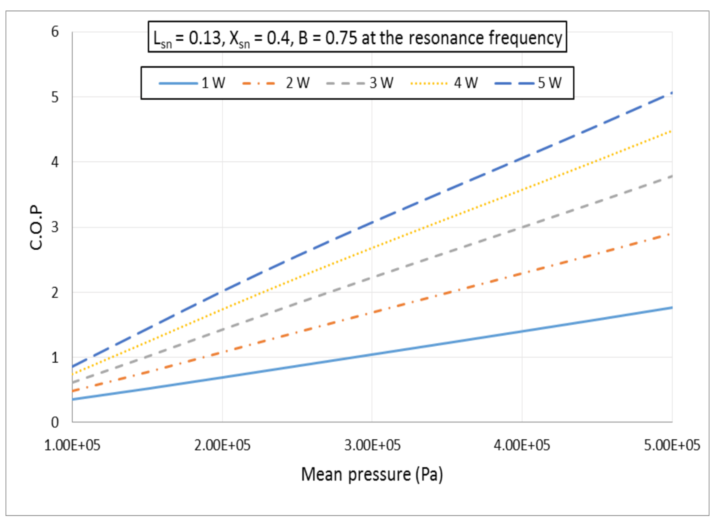

Fig. (5) Mean Pressure effect on the coefficient of performance

International Journal of Scientific & Engineering Research Volume 7, Issue 9, Septemper-2016
ISSN 2229-5518

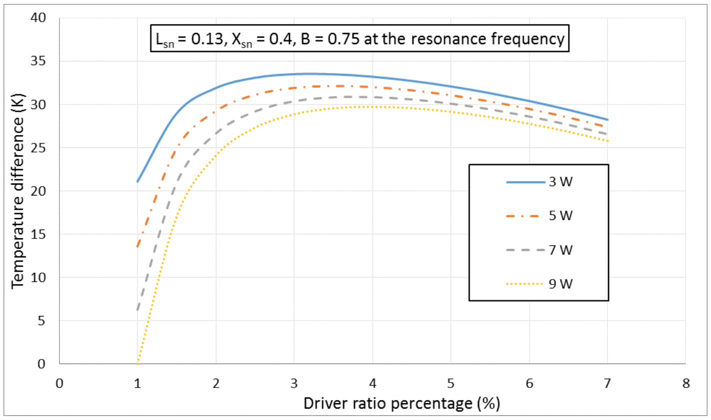

Fig. (6) Driver ratio effect on the temperature difference

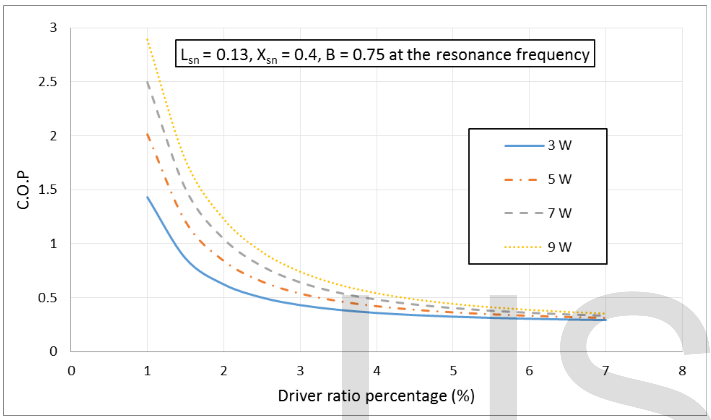

Fig. (7) Driver ratio effect on the coefficient of performance

5.3 Amplitude Pressure distribution

The pressure and the velocity are complex numbers. The real part is more important than the complex part as the the real part value of the pressure is used in the analysis of thermoacoustic devices. DeltaEC calculates the real and imaginary parts. The pressure and velocity components are calculated at the beginning and inside of each segment. The behavior of pressure amplitude will be as shown in fig. (8). There are pressure antinodes at the ends of the resonator and the pressure node nearly is at the middle of the resonator.

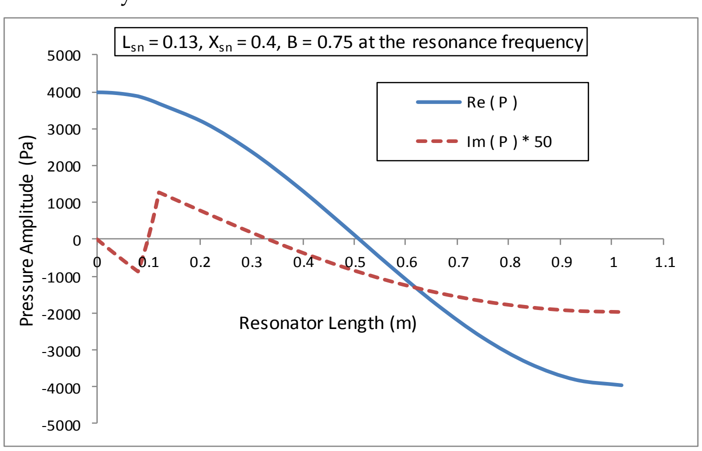

Fig. (8) Amplitude pressure distribution through the resonator

5.4 Flow Rate Propagation

The flow rate is shown in fig. (9). The flow rate at the beginning must have a value because of the dynamic part of the fluctuating loudspeaker and the flow has its maximum value at the loudspeaker facing then it decreases until reaching the stack where there is flow resistance and drops suddenly, after the stack portion is exceeded the flow will have a constant value after that decreasing until reaching the ending of the resonator and then it will have a zero value where there is no flow.

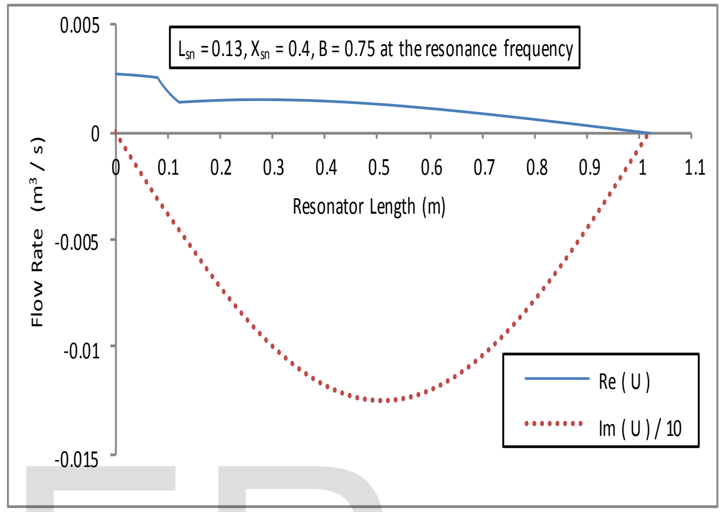

Fig. (9) Flow rate through the resonator

5.5 Acoustic Power Flow

The acoustic energy flow is shown at fig. (10). The acoustic power comes from the driver at which it will have its maximum value after that due to thermal and viscous dissipations the acoustic power will be consumed in the ambient duct before the stack; after that the acoustic power flow reaches to the hot heat exchanger and there will be also losses in that small part due to friction, the stack region is the place where the acoustic power is consumed to pump heat so the main drop in the acoustic power will be in the stack region after that in the cold heat exchanger and cold duct there will be thermal and viscous losses and the acoustic power drops again which will be soon converting as additional cooling load at the cold heat exchanger.

5.6 Total Power Flow

Total power is the sum of the acoustic power flow, plus the dissipated heat and the power flow at heat exchangers. The total power only changes with x in heat exchanger segments as shown in fig. (11). and it is constant for the regions at the regions where there are no heat exchangers.

5.7 Temperature distribution through the resonator

As shown in fig. (12); the resonator can be classified into three regions. The ambient duct is the first region (the duct before the hot heat exchanger); the cold duct (the duct after the cold heat exchanger) is the second region and the third region is the stack where the temperature gradient takes place.

International Journal of Scientific & Engineering Research Volume 7, Issue 9, Septemper-2016
ISSN 2229-5518

5.8 Air-Helium Comparison

Air and helium are compared and a constant cooling load of 1 W is applied to the cold heat exchanger to investigate the working fluid effect on the temperature difference across the stack and the coefficient of performance of the thermoacoustic refrigerator. Mean pressure of 2 bar, amplitude pressure of 2500 Pa and the four guesses and targets in sec. 4 are used to begin DeltaEC simulation. The temperature difference is observed to be higher for helium when used as a working fluid rather than air as shown in fig. (13). The contrast effect is observed in fig. (14) for the coefficient of performance where the air case has higher performance. The Prandtl number is lower for helium than air. This demonstrates the high temperature differences for helium as the Prandtl number is proportional directly with the viscous penetration depth and reversely to the thermal penetration depth. The gas particle velocity in case of helium is higher than air particles velocity and this draws more input acoustic power for helium which will have lower performance. Helium is preferred in low temperatures but this will lead to high power consumption. A stack spacing to thermal penetration value of 4 is optimum for both the temperature difference and the temperature difference for air and helium.

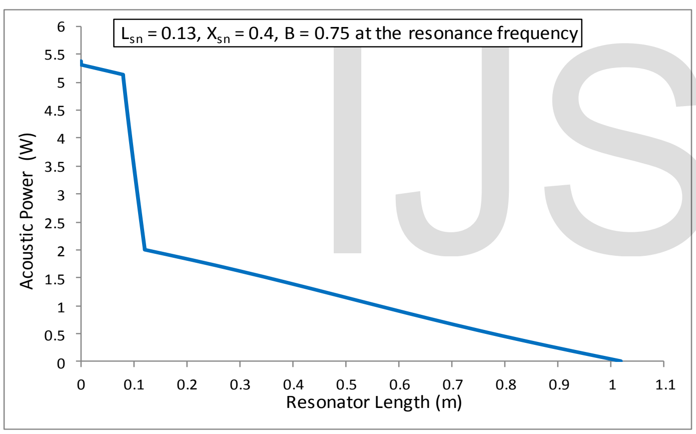

Fig. (10) Acoustic power through the resonator

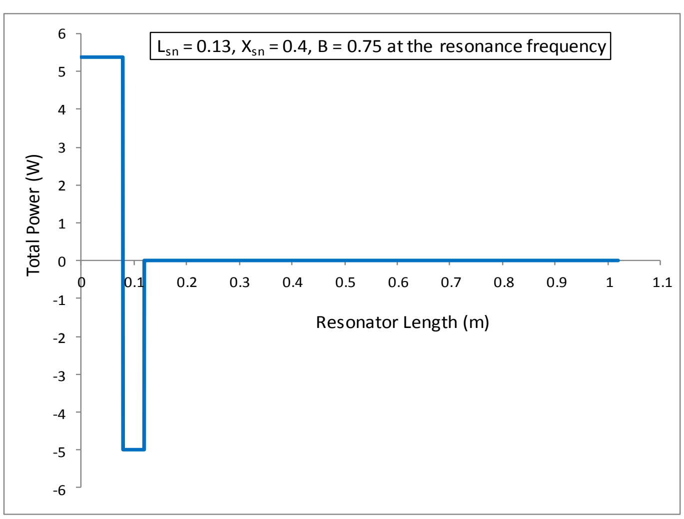

Fig. (11) Total power through the resonator

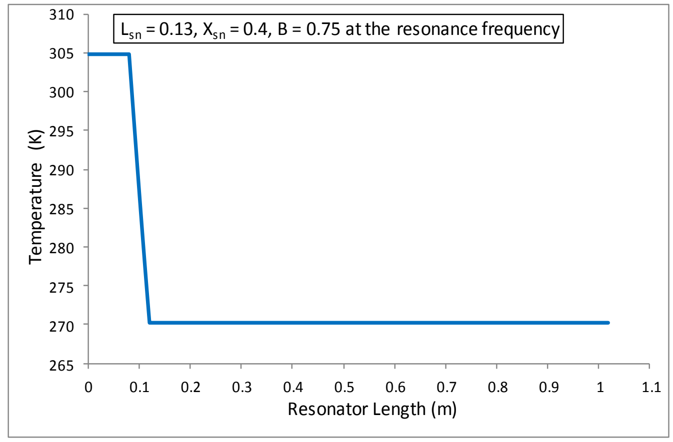

Fig. (12) Temperature distribution through the resonator

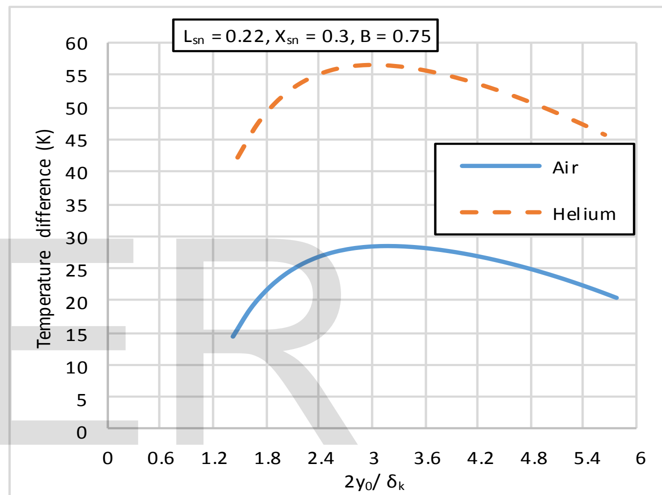

Fig. (13) Temperature difference for air and helium versus stack spacing to thermal penetration depth

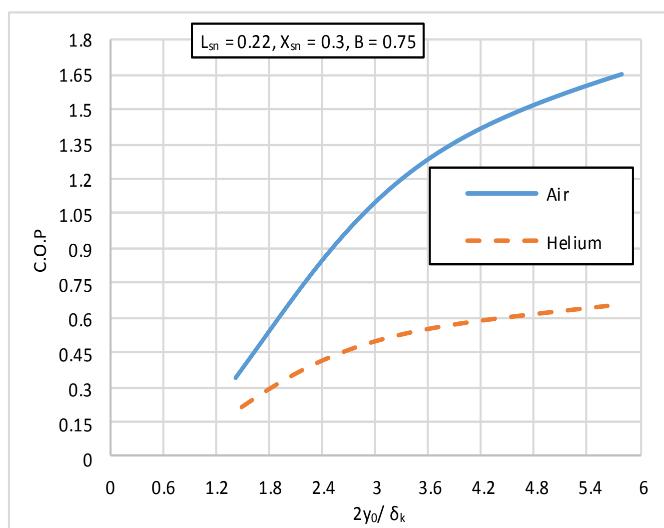

Fig. (14) Coefficient of performance for air and helium versus stack spacing to thermal penetration depth

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ISSN 2229-5518

5.9 Results Validation

Tejani [2] designed a thermoacoustic refrigerator using a buffer volume at the end of the resonator to reduce the total length and thus reducing losses. The working medium was helium at frequency of 409 Hz and 10 bar mean pressure. Tejani made experiments using the previous conditions. In additions; hot heat exchanger temperature was set to be constant at 289 K and the experiments were executed at three driver ratios of 0.7 %, 1.4 % and 2.1 %. A code using DeltaEC simulation program has been done to simulate Tejani work. The simulation Temperature differences results from DeltaEC at driver ratio of 1.4 % is compared with Tijani experimental work. A good agreement as shown in fig. (15) is found.

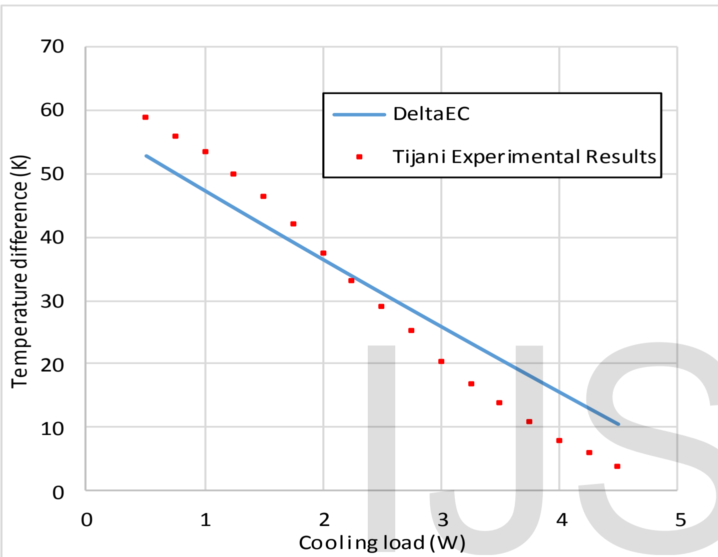

Fig. (15) Effect of cooling load change on the temperature difference at D = 1.4%

6 Conclusion

A thermoacoustic refrigerator is designed and then it is simulated through DeltaEC to get optimized operating condition. The following conclusions can be drawn:

- The temperature difference should be put into consideration when optimizing the performance.
- Increasing the mean pressure will increase the performance for the thermoacoustic refrigerator.
- Increasing the amplitude pressure will decrease the performance for the thermoacoustic refrigerator.
- The stack is the place where the temperature gradient takes place and the power is consumed through it in adition to the thermal and viscous losses at the resonator walls.
- Air gives higher performance coefficients than helium with lower temperature differences across the stack.

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