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Thermoacoustic solar cooling for domestic usage sizing software Part (I)

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Abstract

In the project of "Thermoacoustic solar cooling for domestic usage" funded by University of Damascus, which well be built in refrigeration laboratory, it was necessary to start up the project with the ability to use for cooling achieving known temperature and required capacity. As the possible required cooling variable, it was necessary to obtain the sizes of the Thermoacoustic cooler and optimal design pressure as a function of the required cooling capacity in order to insure convenient sound power. In this paper, the modeling equations for design were done and turned into a computer program. The developed program inputs were the possible cooling capacity and some assumptions such as gas and stack type and initial dimensions with the possibility of changes. The outputs were stack, heat exchangers and resonator sizes and acoustics Driver power necessary. The program has been tested on some manufactured models such as Tijani's 5 watts and Normah Mohd Ghazali's 5 watts refrigerator, results was satisfactory.

Keywords: Thermoacoustic, solar refrigeration, stack sizing software, Blockage ratio impact

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1. Introduction

In the really need for friendly environmental green building. Thermoacoustics were discovered. Thermoacoustics have many applications. One is refrigeration. Also it has lower COP, but it is convenient with renewable energy. Has no moving part. Thermoacoustic refrigerators can generally be broken up into four parts. These parts are known as the driver, the resonator, the stack, and the heat exchangers. The four parts are labelled for an example refrigerator in Figure1:

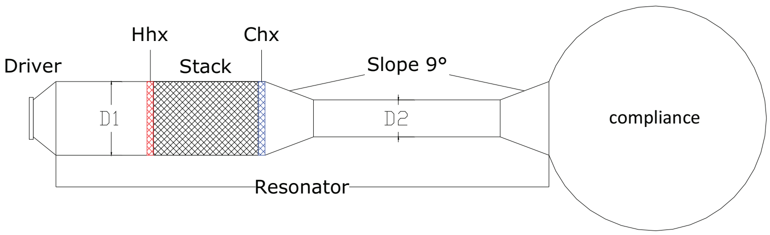


Figure.1: Parts of a thermoacoustic refrigerator

In Figure (1), the parts are shown for a common thermoacoustic refrigerator setup. However, there are many different thermoacoustic refrigerators in existence, some of which look nothing like the thermoacoustic refrigerator shown in Figure .1. This being said, every design in some way carries out the four basic functions shown above [5]. The driver creates either a standing or travelling wave in the refrigerator. The wave created by the driver is generally at or near the resonant frequency of the resonator in which the wave oscillates. The stack is located at some point within the resonator and exists to create more surface area across which the Thermoacoustic effect can take place. Finally, the heat exchangers are used to take heat from a refrigerated region and pump heat to the outside. These components are each described individually in detail in the later sections. Russel [12] describes a cheap and easy to build thermoacoustic refrigerator. This refrigerator is for demonstration purposes and so is not very powerful or efficient. However, it is an excellent starting point for those interested in the field. Actually, Tijani [11] published a paper describing in detail the process used to design a thermoacoustic refrigerator from start to finish. In this paper the relations which been used, will be reviewed from gas thermo physical Properties till acoustic driver.

2. Working Gas thermo physical Properties

Based on the National Institute of Standard and Technology (NIST) Data [9], polynomial equations were developed describing thermo physical properties for Helium, Argon, Air, and Hydrogen been included in the software. Equations were developed by using Graph [10]

3. Design Strategy

Based on Tijani's strategy, with some modification that will be referred to in place, it contains: stack, resonator, heat exchangers, and driver. The strategy shown in figure (2)

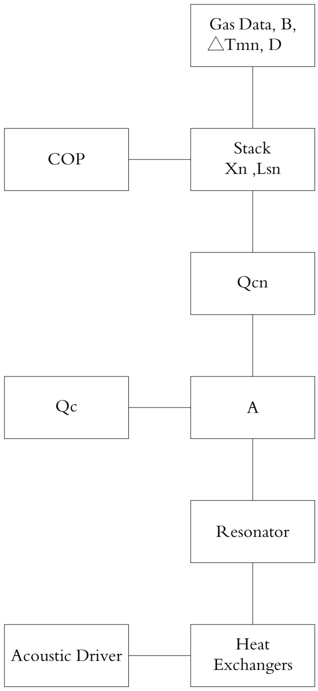


Figure.2: Design strategy of a thermoacoustic refrigerator

3.1. STACK DESIGN

Stack is the most important item in the thermo acoustic device, and the hard one to be designed.

3.1.1. The stack material must have low thermal conductivity, k_s and heat capacity cs larger than the heat capacity of the working gas, cp. So Mylar is chosen as its has a low heat conductivity, Ks = 0.16 W/m.K

3.1.2. There are many types of stack; parallel, circular, pin array, triangular etc. due to simplicity of fabrication reasons; we choose the parallel plates type of stack. For the parallel plates stack we must have that rh = yo .

3.1.3. The thermal penetration depth, δ_k and viscous penetration depth (δv) are given by,:

$$\delta_k = \sqrt{\frac{2K}{\rho_m c_p \omega}} \quad (1)$$

Where
K = Gas thermal Conductivity
ρm = Gas density
cp = Gas heat capacity

$\omega = 2\pi f$ angular frequency
Where:
f = Operating frequency

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

Where:
μ = Gas Viscosity
As can see from figure below, for parallel plates stack lm(-fk) has a maximum for rh/δk = yo/δk = 1.1

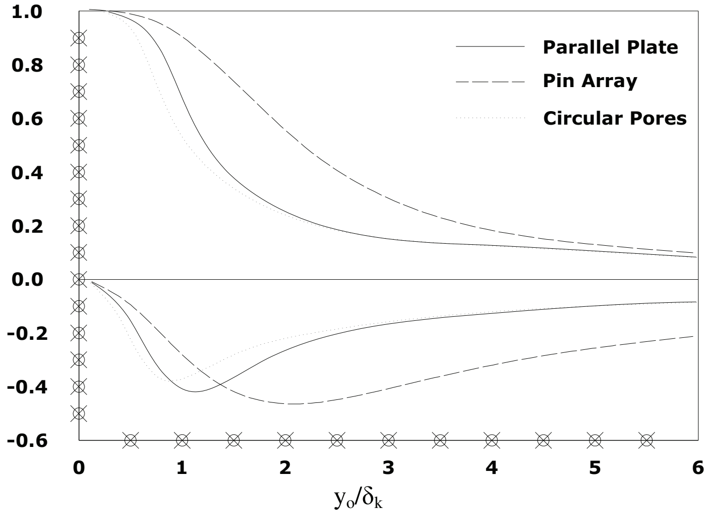


Figure 3: Imaginary and real parts of the Rott function fk as function of the ratio of the hydraulic radius and the thermal penetration depth.

Wheatley [4] has stated that in order not to alter the acoustics field, a spacing of 2δk to 4δk must be used. The blockage ratio or also known as porosity of the stack, B is given by

$$B = \frac{y_0}{y_0 + l} \tag{3}$$

Where:
l = Half stack plate thickness

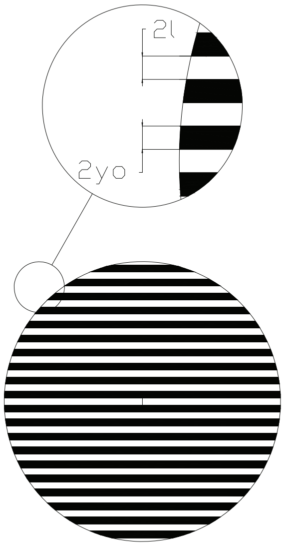


Figure 4: Illustration of cross sectional view of parallel plate stack, the plates are 2l thick and are 2yo spaced

3.1.4. From the figure below we can see the relation between the value of COP and normalized stack center and normalized stack length, COP$\rightarrow$xn, Lsn.

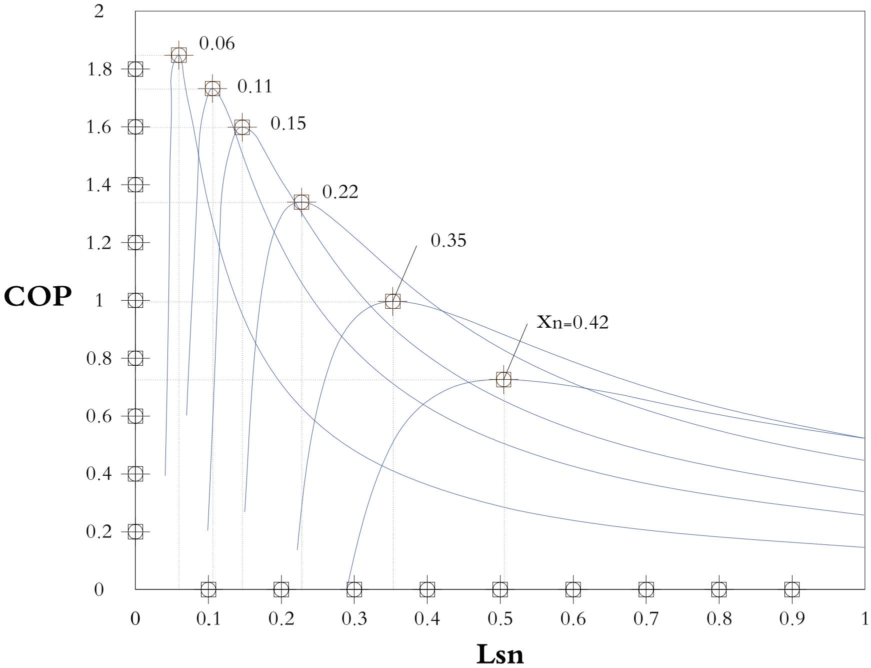
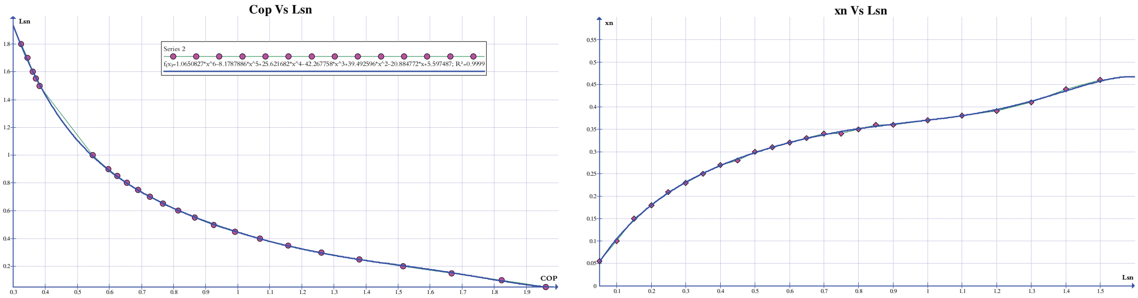


Figure 5 : Performance calculations for the stack, as a function of the normalized stack length Lsn and normalized stack center xn.

To achieve this values the figure may tend to be two equations as follow (using Graph[10]):



a b

Figure 6 : Performance calculations for the stack, as a function of the normalized stack length Lsn (a) and normalized stack length Lsn normalized stack center xn. (b)

$$Lsn=f(COP)=1.0650827*COP^6-8.1787886*COP^5+25.621682*COP^4-42.267758*COP^3+39.492596*COP^2-20.884772*COP+5.597487 \tag{4}$$

$$x_n=-0.57544186*Lsn^6+2.8230118*Lsn^5-5.4519528*Lsn^4+5.5207892*Lsn^3-3.3916463*Lsn^2+1.4571952*Lsn-0.011895787 \tag{5}$$

the relations between Normalized stack operation and stack operation are:

$$Lsn=k\,Ls \tag{6}$$

Where:
Lsn = normalized stack length, Ls = Stack length, k = Wave number

$$xn=kxs \tag{7}$$

Where
xn = normalized stack center, xs = stack center, k = Wave number
Wave number is given by the equation:

$$k=\frac{\omega}{a} \tag{8}$$

Where: ω = angular frequency and a = sound velocity

3.1.5. The dimensionless cooling power and dimensionless acoustics power are:

The dimensionless cooling power is given by:

3.1.5.1.

$$\mathrm{Qcn}=-\frac{\delta_{kn}D^2\sin(2.x_n)}{8.\gamma.(1+\mathrm{Pr}).\Lambda}\left(\Gamma.\frac{1+\sqrt{\mathrm{Pr}}+\mathrm{Pr}}{1+\sqrt{\mathrm{Pr}}}-\left(1+\sqrt{\mathrm{Pr}}-\sqrt{\mathrm{Pr}}.\delta_{kn}\right)\right) \tag{9}$$

Where:
Qcn = Dimensionless / Normalized cooling power
$D=\dfrac{Po}{Pm}$: = Drive ratio
Where: Po = Dynamic Pressure, and Pm = Average Pressure

δkn = Normalized thermal penetration depth
xn = Normalized center position
γ = Ratio of isobaric to isochoric specific heats
Pr = Prandtl Number

$$\Lambda = 1 + \sqrt{\Pr}.\delta_{kn} + \frac{1}{2}\Pr(\delta_{kn})^2 \tag{10}$$

$$\delta_{kn} = \delta_k / y_0 \tag{11}$$

Where yo = half plate spacing

$$\Gamma = \frac{\nabla Tm}{\nabla Tc} = \frac{\Delta Tmn.\tan(x_n)}{(\gamma - 1)B.Lsn} \tag{12}$$

Dimensionless parameters, the ratio of the temperature gradient along the stack and the critical temperature gradient
Where:
ΔTmn = Normalized Temperature difference
B = Blockage ratio or porosity
Lsn = Normalized stack length

$$Qcn = Qc / Pm\, a\, A \tag{13}$$

Where:
Qc= cooling power
Pm= Average Pressure
a = sound velocity
A = Stack cross sectional area
That yield:

$$A = Qc / (Qcn.Pm.a) \tag{14}$$

The dimensionless acoustics power is given by:

3.1.5.2.

$$Wn = -\frac{\delta_{kn} LsnD^2}{4.\gamma}(\gamma - 1)\mathrm{B}\cos^2(.x_n)\left(\Gamma.\frac{1}{(1+\sqrt{\Pr})\Lambda} - 1\right) - \frac{\delta_{kn} LsnD^2}{4.\gamma\,\mathrm{B}\,\Lambda}\sqrt{\Pr}\sin^2(.x_n) \tag{15}$$

$$Wn = W / Pm\, a\, A \tag{16}$$

Wn = Dimensionless / Normalized acoustic power
W= Acoustic Power

4. Resonator:

a λ/4-resonator will dissipate only half the energy dissipated by λ/2-resonator. Hence a λ/4-resonator is preferable. Hofler [7] presents that the λ/4-resonator can be further optimized by reducing the diameter of the resonator part on the right of the stack

As a result of total loss has a minimum at about:

$$D2/D1 = 0.54 \tag{17}$$

Hofler [7] and Garrett [8] used a metallic spherical bulb to terminate the resonator. The sphere had sufficient volume to simulate an open end. The calculation optimization of the half-angle of the cone for minimal losses has been determined to be 9^o.A gradual tapering is also used between the large diameter tube and small diameter tube [11].

The lengths of the two parts of resonator are given by:

$$l \cong x_s + Ls/2 + 2.x_1 \tag{18}$$

Where:

l is the length of the big diameter tube.

$$\cot(k\,l) = \left(\frac{D1}{D2}\right)^2 \tan\left(k\left(L_t - l\right)\right) \tag{19}$$

$$\Rightarrow \left(L_t - l\right) = \frac{\mathrm{atan}\left(\left(\frac{D2}{D1}\right)^2 \cot(k\,l)\right)}{k} \tag{20}$$

Where: Lt - l is the length of the small diameter tube, k = Wave number, l = Length of the large diameter tube, D1 = Diameter of large tube (stack side), D2 = Diameter of small tube, Lt = Total length of resonator

5. Heat exchangers

Heat exchangers are necessary to transfer the heat of the thermoacoustic cooling process. The design of the heat exchangers is a critical task in thermoacoustics.

According to Tijani [1]:

5.1. .Cold heat exchanger

The whole resonator part on the right of the stack in Fig. 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. 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

$$x_1 = \frac{u^{(1)}}{\omega} = \frac{p_o^{(1)}}{\omega\,\rho_m\,a}\sin\left(k\,x\right) \tag{21}$$

The optimum length of the cold heat exchanger is thus about $2x_1$. To avoid as much as possible entrance problems of the gas when leaving the stack and entering the cold heat exchanger or vice versa (continuity of the volume velocity), the porosity of the cold heat exchanger must match the porosity of the stack. This implies that a same blockage ratio has to be used in the design of the cold heat exchanger.

Acoustic power is also dissipated in the cold heat exchanger. Eq. (16) can be used to estimate the dissipated power. Substituting the position of the cold heat exchanger *xn* the length *Lsn*, and $\Gamma = 0$ (uniform mean temperature), yields that the cold heat exchanger will dissipate *Wchx*

5.2. Hot heat exchanger

The hot heat exchanger is necessary to remove the heat pumped by the stack and to reject it. 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. Substituting the position of the hot heat exchanger *xn*, the length *Lhn* and $\Gamma = 0$, we obtain the acoustic power dissipated in the hot heat exchanger *Whhx*

6. Acoustic driver

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

$$W_t = W_s + W_{res} + W_{chx} + W_{hhx} \tag{22}$$

Taking into account the power dissipated in the different parts, the performance of whole refrigerator becomes

COP =Qc/Wt (23)

This value must be lower than the stack alone

7. Software and Results

The software was done with aid of VB6 Language. The interface is shown in figure (7).

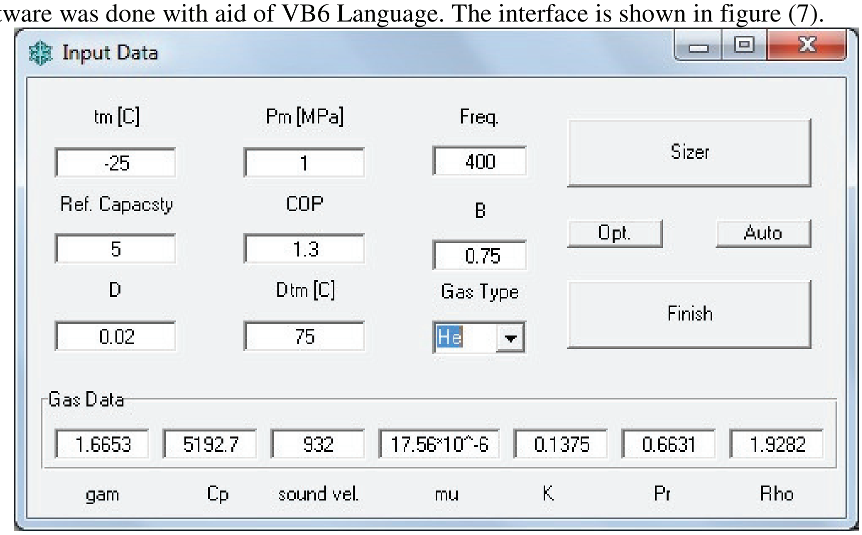


Figure 7-a: Input form showing desired Parameters used to calculate refrigerator sizes

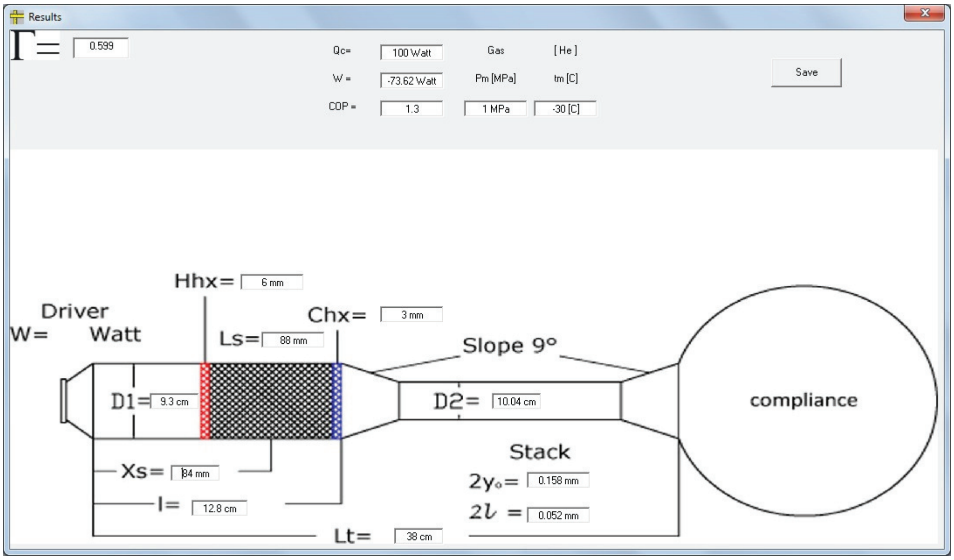


Figure 7-b: Output form showing calculated refrigerator sizes

It has two forms, one for data input fig (7-a), the other is for output data (results) fig (7-b). As we see, in fig. (7-a) the software has a number of commands:

The command named "Sizer" witch calculate and size the refrigerator with the specific Parameters written in the Input form.

The software is used to design a refrigerator with cooling capacity of 30 watt for usage in domestic purposes. The input data was as in table (1), Results are shown in table (2).

Table (1) input data used for sizing refrigerator

f [Hz]	**COPs**	**ΔTm[C]**	**B**	**D**	**Qc [Watt]**	**Tm [C]**	**Pm [Mpa]**
400	1.3	75	0.75	0.02	100	10	1
gas	**γ**	μ	a	cp	K	Pr	ρ
[He]	1.6654	$19.2*10^{-6}$	995	5192.4	0.1505	0.6621	1.6911
[Ar]	1.6992	$21.8*10^{-6}$	314	534	0.0172	0.6772	17.0864
[Air]	1.4206	$17.92*10^{-6}$	338	1022.9	0.0251	0.7313	12.3427
[H2]	1.4091	$8.63*10^{-6}$	1291	14272.9	0.1783	0.6905	0.8509

As we can see in table (2) also Argon refrigerator has the less driver input power, the refrigerator using Helium is the best choice can be fabricated. H2 and Air aren't suitable.

The command named "Auto" calculates and size the refrigerators for all the range desired. The result will be saved in a text file has the name of desired fixed Parameters.

The command named "Opt" witch calculate all the range for:

Blockage ratio of 0.1 to 0.9, Mean pressure of 0.1 to 3 [MPa], Mean temperature -30 to 30 [C]

With other inputs such as Gas type, Dtm, D, refrigeration capacity, frequency, and COPs. The result will be saved in a text file has the name of desired fixed Parameters.

Mean temperature studied were focused on tm =10 [C] because it is suitable as Air Conditioning supply temperature. And tm =-30 [C] for severe cold weather.

Table (2) refrigerator sizes for the input data in table (1)

tm = 10 [C]					*Pm= 1 [Mpa]*				*Qc=100 [Watt]*		
gas	**W [Watt]**	**Lt [cm]**	**2Yo [mm]**	**2l [mm]**	**Ls [mm]**	**Xs [mm]**	**D1 [cm]**	**D2 [cm]**	**Chx [mm]**	**Hhx [mm]**	**Driver [Watt]**
[He]	-65.64	40.7	0.35	0.12	94	90	16	8.64	3	6	-78
[Ar]	-66.88	12.8	0.12	0.04	30	28	28.2	15.23	1	2	-77.7
[Air]	-129.55	13.8	0.12	0.04	32	30	50	27	1	3	-164.7
[H2]	-143.52	52.6	0.32	0.11	122	116	27.8	15.01	5	10	-186.5

Results for tm =15 [C] for Air, and tm =-30 [C] for hydrogen are reported in figure (8) and figure (9) consequently.

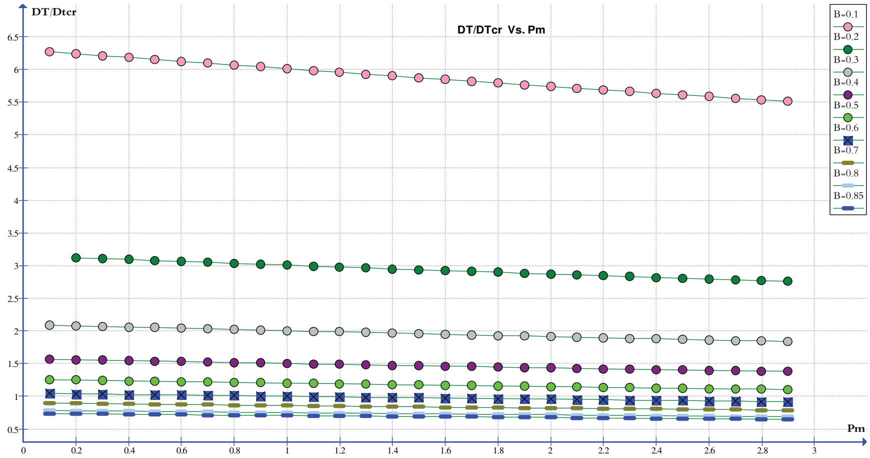


Figure 8: Output for calculated "Γ" as a function of "Pm" for all range of B when: Gas= Air, and tm=15 [C].

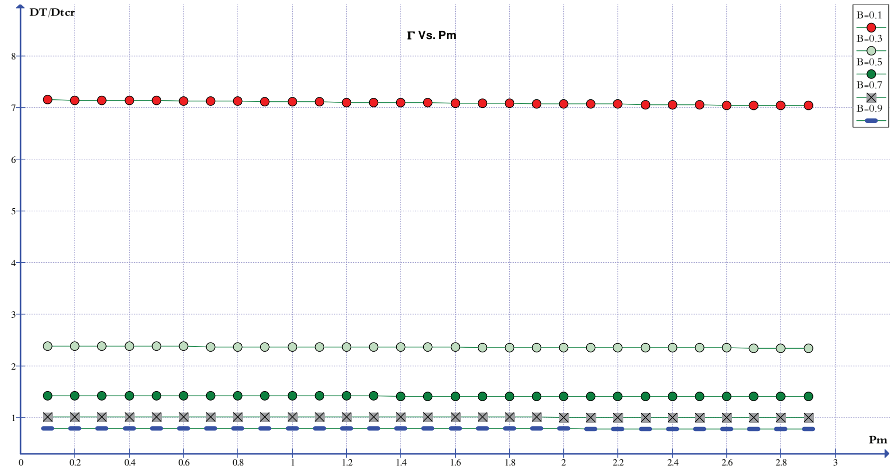


Figure 9: Output for calculated "Γ" as a function of "Pm" for all range of B when: Gas= H2, and tm=-30 [C].

8. Conclusion

Software was achieved by VB6 code using equations proposed by Tijani and others. Software was tested by some tested on some manufactured models such as Tijani's 4 watts and Normah Mohd Ghazali's 5 watts refrigerator, results was satisfactory. The software was used to design a refrigerator with cooling capacity of 100 watt for usage in domestic purposes. The refrigerator using Helium is the best choice can be fabricated.

Hydrogen is better for prime mover than heat pump, because of higher Γ, And it is a suitable solution for cold weather prime movers. Pm has no high impact on the performance of refrigerator using "H2" with B>0.8. Nor prime movers with B<0.7

Air is good for each prime mover and heat pump. Pm has no impact on the performance of refrigerator using "Air" with B>0.8. But it has for prime movers with B<0.7. The "B" lower, the impact higher.

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