# *FLUIDYNE PUMP PARAMETERS ANALYSIS AND OPTIMIZATION*

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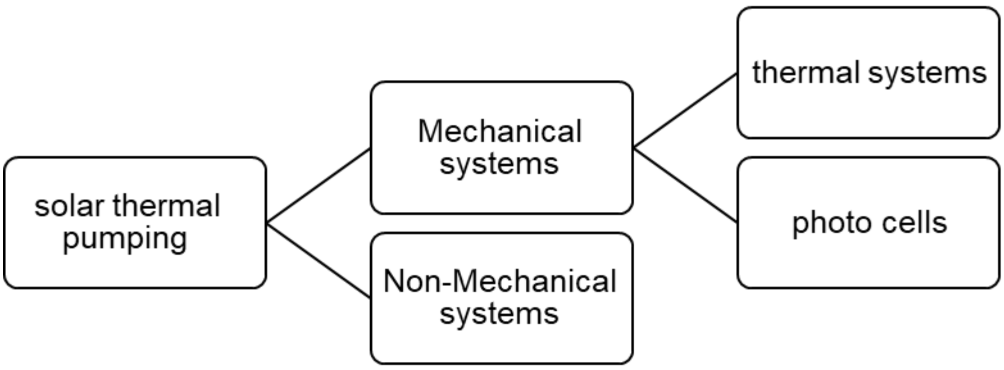
## ABSTRACT

Fluidyne pump or engine is a Stirling engine with liquid pistons. The main governing equations of the fluidyne pump were analyzed and reduced to get the main independent parameters that affect the efficiency of the fluidyne pump. GENETIC algorithm using MATLAB software was applied to get the optimum case (or cases) based on the maximum efficiency. The optimum cases were tested experimentally using different gas and water volumes. The results showed that the main independent parameters are: a proposed temperature factor (depends on the hot and the cold zone temperatures), the working gas volume , the working liquid volume and density, the output tube pressure and diameter. Moreover , the experimental study confirmed that increasing the difference between the hot and the cold zone temperatures increases the efficiency. It was confirmed also that the high performance of the pump requires a working gas volume to be higher than the working liquid volume. A mathematical conditions was proposed by the present study that guarantee the best performance of the fluidyne pump as a function of the independent parameters mentioned.

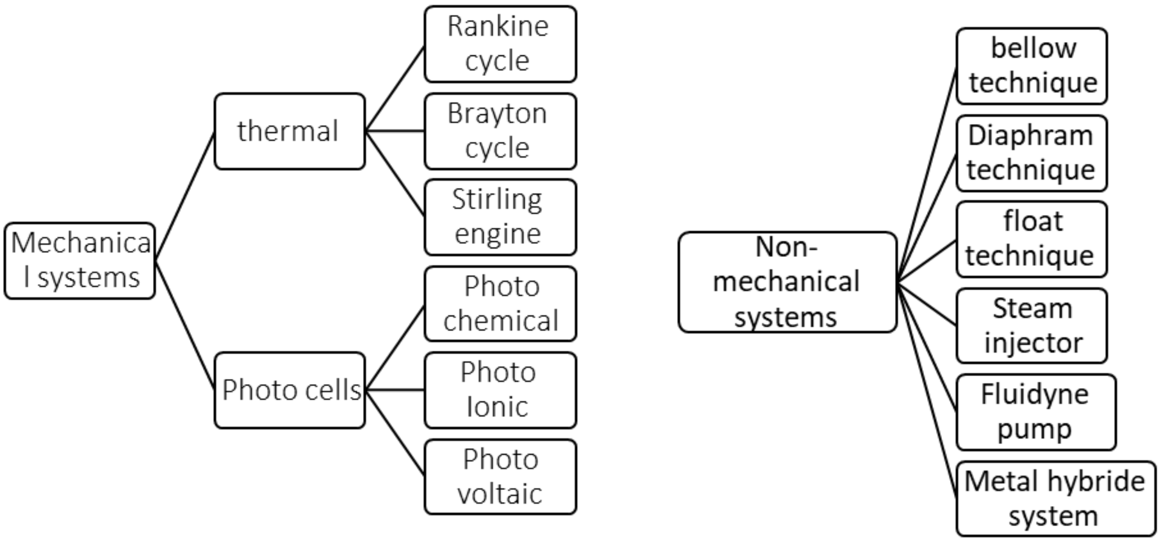
**Keywords:** Fluidyne pump, liquid piston stirling engine, solar pumping.

## 1 INTRODUCTION

Solar energy is considered as the main energy source on the earth. The solar thermal form of this energy is applied in many applications including water heating, air conditioning, power plants, drying, pumping and others. Pumping is one of the most important applications for Agricultural and Industrial activities. Solar pumping is one of the earliest applications as a solar thermal irrigation was built in Egypt in 1913[1] .Generally solar thermal pumping can be classified to mechanical and non-mechanical systems:




**( a )**




(b) (c)

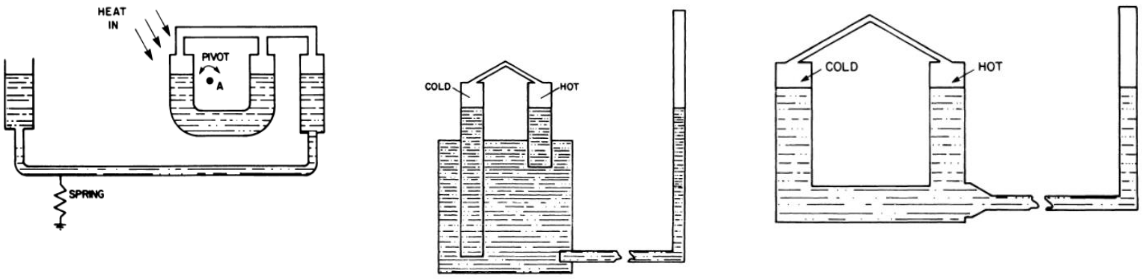
**Figure 1. solar thermal pumping(STPS) classification (a) general classification, (b) mechanical systems, (c) non-mechanical systems**

Each of the mechanical and non-mechanical systems has some advantages and disadvantages which are illustrated in table 1:

**Table1. Advantages and disadvantages of the Mechanical and non-mechanical solar pumping systems**

| Item | MSTP. | NMSTP. |
|---|---|---|
| Output flow rates | High - medium | Medium-low |
| Installation cost | high | Low |
| Maintenance cost | High | low |
| Moving parts | high | Low / non |
| Total efficiency | High - medium | Medium – low |
| Heat exchangers | high | Low |
| loops | high | low |

Fluidyne pump (FP) is a liquid piston engine working according to "Stirling cycle" with two isothermal and two isochoric strokes[2,3,4,5,6,7,8]. It is a very simple device that does not have moving parts, loops or lubrication system. This pump can be used in pumping liquids for different applications. It has a promising application in agriculture and industrial activities. Many research works explained the types of the FP. There are three different design groups of the fluidyne engine as illustrated in (fig.2).




(a) (b) (c)

**Figure 2. Different designs of the fluidyne engine [2] (a) rocking beam (b) pressure feedback (c) jet-stream (The fluidyne heat engine-Mosby, David Cottrell 1978)**

The cycles are composed of two constant volume strokes and tow constant temperature strokes as illustrated in (fig.3). The pressure difference between the hot , the cold sides and the output make a reciprocal movement for the working liquid generating output work..

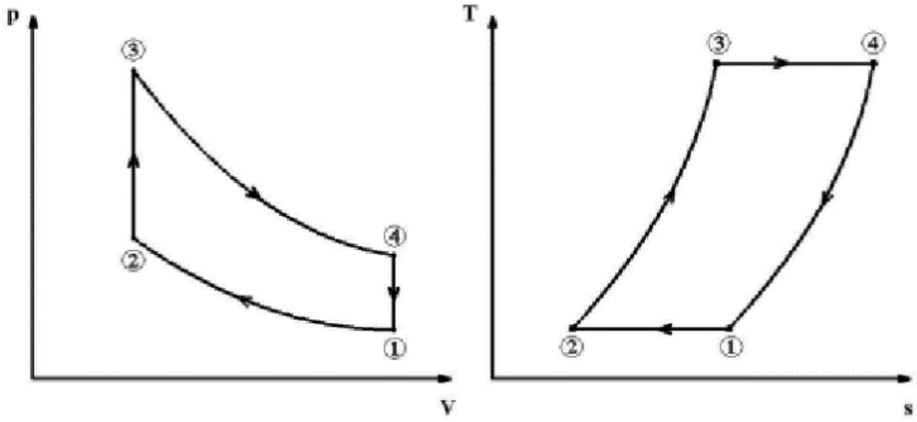



**Figure 3. P-V and T-S diagram of fluidyne engine (The fluidyne heat engine-Mosby, David Cottrell 1978)**

Fluidyne engine had born in the U.K. Atomic energy Lab – 1971 by C. West [2,3,6]. In 1984, the "Stirling engine conference" published a paper in the fluidyne engine classifying it as a type of stirling engine and explained its parts and principles [4]. C. West , Elord and Geissow studied the non-loading system of the FP in the period (1971-1975) [2,3,7]. C. West studied how to get a maximum output power through investigating the frequency and the amplitude of the syst6em. Elord studied the movement of cold, hot and output columns. Geissow investigated the temperature difference needed to start the oscillation and then the pumping working.
The pumping flow rate, amplitude and the onset temperature for the FP design illustrated in fig.4. The study took place with three different gas volumes (0.8458, 0.6846, 0.5233 $in^3$), and a temperature range (140: 210 °F) [2]. The results illustrated that increasing the gas volume and temperature of the hot column increases the output displacement and then the output flow rate.

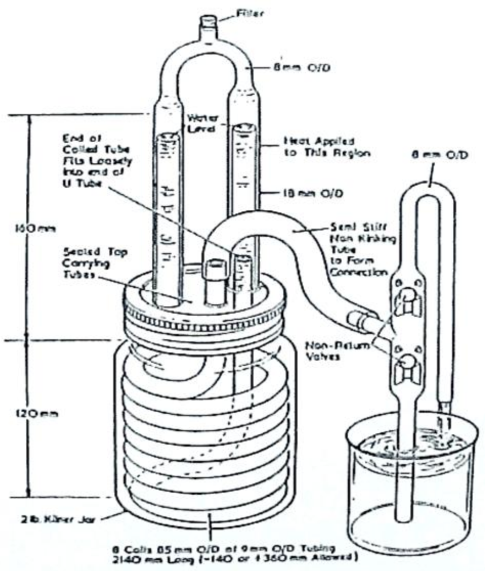
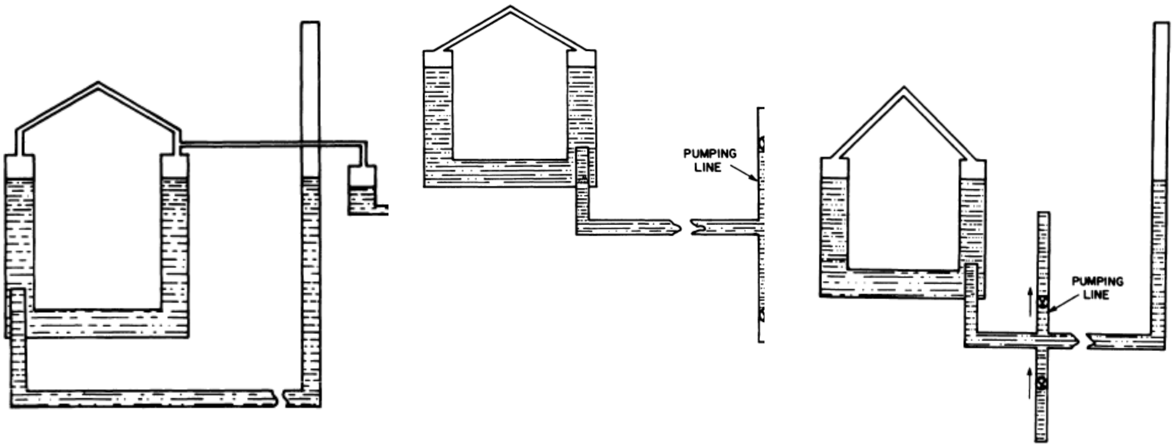



**Figure 4. Mosby's model of the FP (The fluidyne heat engine-Mosby, David Cottrell 1978)**

The Main designs of JSFP are classified according to the output tube position from the other tubes [6] as illustrated in (fig-5)




a- cold tube model b-series model c- Parallel model

**Figure 5. the main deigns for F.P. [6]**

Several systems have been built after that: [6, 7] as shown in table (2)

**Table 2. models of F.P between 1970 and 1981 [6, 7]**

| system | Q - liter/hour | Head (h) – m | η % |
|---|---|---|---|
| West (1970) [3] | 11 | 1.1 | ....... |
| West (1971) [3] | 380 | 1.8 | 0.35 |
| Goldberg et al. (1977) [9] | 36 | 0.7 | 0.12 |
| Goldberg et al. (1977) [9] | 44 | 1 | 0.08 |
| Mosby (1978) [2] | 22 | 0.4 | 0.15 |
| Reader (1979) [10] | ...... | .......... | 0.03 |
| Bell (1979) [11] | 114 | 1.2 | 0.18 |
| Reader et al. (1981) [10] | 8 | 1.1 | 0.52 |
| West et Pandey (1981) [12] | 1740 | 3.3 | 4.7 |
| Pandey (1981) [12] | 9500 | 3.3 | 7 |

Harwell and Metal Box, built the biggest and more complicated model of FP 1981(fig-6) pumped more than 1700 liter/hour, with 5-m head [6]. Their model is illustrated in figure 6...

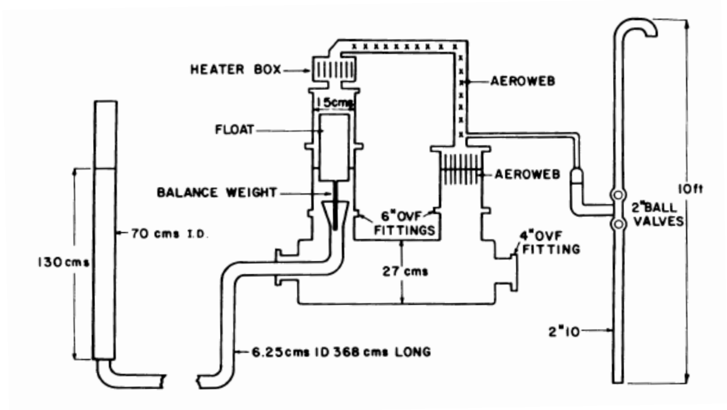



**Figure 6 . Harwell and Metal Box , pump (G.Walker J. R. Senft 1985)**

Yamaguchi 1994 [13] tested two models of a non-loading FP and explained the frequency of the output tube and the displacer tube. They showed that the output frequency depends on the output tube, while the displacer frequency depends on the working liquid length. Another model was presented by

Francois Lanzetta 2004[7] which pumped 6 litter/hour. Jackson W.Mason and James W.Stevens 2014 [4] tested a free-unloaded fluidyne engine using a chair with Fresnel lens to get the positive and negative work for 1.06 Hz oscillation, it was about 12% ratio between negative and positive per cycle, for unloaded system. In the middle of 1990s the PV (Photovoltaic) applications became wide spreading compared to the other solar pumping applications, for example, there were 21000 pumping system working by PV in the world in the end of 1992 [15]. PV pumping systems introduced an acceptable efficiency for different heads and different types of pumps. This led to a little attention to the non-mechanical systems in general and specially the fluidyne engines. Frank Kyei and Obodoako 2006 [24] designed a FP that depends on the output from cold zone as illustrated in (fig-7 ), the pump has 4-Hz frequency and 25 cm amplitude, 5 watts output, and its efficiency reached 3.5%.

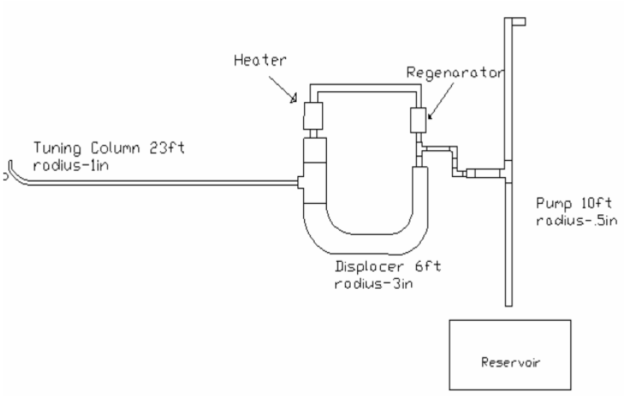



**Figure 7. Frank Kyei FP model (Frank Kyei and Manu Aloysius Obodoako 2006 )**

After the finance crisis 2008 the attention to the non-mechanical systems returned back, especially because of the high cost of P.V. systems [17].

This study will analyze the basic parameters and governing equations of the FP to simplify them and determine the main independent variables. Moreover, optimize those parameters in order to get the maximum efficiency of the FP by artificial intelligence techniques. Then validate the optimum results .experimentally.

## 2 MATHEMATICAL STUDY

### 2.1 Governing Equations

The governing equations of the FP are derived in to two cases, loaded and un-loaded system. The equations are developed between 1971-1979 by C. West &, Elord and Giesow as mentioned in [2, 3]. The main equations for the FP were published by C. West 1979 [3] where the final formula combined the loaded and un-loaded models, treated the frequency and the temperature coefficient based on some assumptions which are as follows :

1.The diameter of displacer tube is the same all over the length , also gas and output tube diameters
2.The pressure in the ambient (gas tube) is the same in both hot and cold zones
3.The cold zone temperature in the gas tube (T1) is constant
4.The Output tube pressure varies with the different sources of water , but the atmospheric pressure case is the base case.

The equations as derived by C. West 1979 [3] are as follows

$$\frac{T_2 - T_1}{T_2 + T_1} = \frac{\rho_w * V_g}{P_o * A_d}\left[\frac{0.5 * g * L_d}{2H + L_d} + \frac{8\pi\mu_w}{(A_o * \rho_w)^2}\left(\frac{2H + L_d}{L_d}\right) b\right] \quad (1)\ldots. \quad [3]$$

$$\omega = \sqrt[2]{\frac{2g}{2H+L_d}}$$ (2)….[2,3]

$L_o=(P_o*A_o)/(\rho_w*V_g*\omega^2)$ (3)......[2,3]

Different parameters of FP are illustrated in (fig.8) and can be classified as :
1-Geometric variables:-
Displacer : Ld ,Hh , Hc , Ao
Output tube zone : Lo , Do, Ho , Ao
Gas tube : Vg
2-Pressure – temperature variables:
Ambient pressure
Output tube pressure (Po)
Hot and cold zone temperature ( T2) & ( T1)
3- Flow variables:-
Volumetric flow rate (Q)
Fluid density : working and output fluids $\rho w$ , $\rho o$
Fluid viscosity: absolute and kinematic $\mu w$, $\gamma w$, $\mu o$, $\gamma o$
Moreover, three additional factors were considered in the present study for simplicity which is as follows:
Working liquid length factor d_i=L_d/(2H+L_d )
Temperature factor Ti=( T_2- T_1)/( T_2+ T_1 )
Displacer factor (D.F.) m = n2 L d n – factor n = Dg / Dd

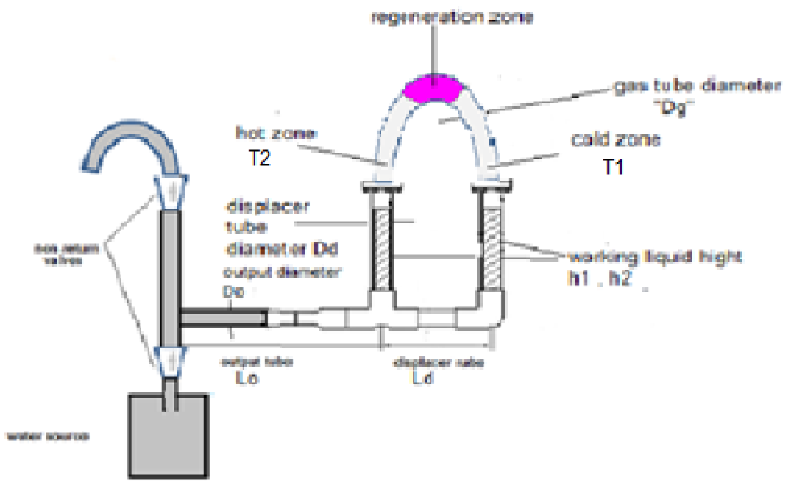



**Figure 8. FP parameters**

## 2.2 Equations simplification:

### 2.2.1 Volumetric flow rate

Pump output Power= $\rho_o\, Q\, g\, h = 0.5b(\omega R)2$ (4) **[3]**

$Q = 2RAo/\text{period time} = \omega RAo/\pi$ (5) **[3]**
$\omega R = Q/Ao$ $(\omega R)^2 = (\pi Q/Ao)^2$ (6) **[3]**
$\therefore b = \rho o\, g\, h / 0.5Q\, (\pi/Ao)2$ (7)

Substituting in (1) $$\frac{T_2-T_1}{T_2+T_1}=\frac{\rho_W*V_g}{P_o*A_d}\left[\frac{0.5*g*L_d}{2H+L_d}+\frac{8\pi\mu_w}{(A_o*\rho_w)^2}\left(\frac{2H+L_d}{L_d}\right)*\left(\frac{\rho_o Qgh}{0.5\left(\frac{\pi Q}{A_o}\right)^2}\right)\right] \quad (8)$$

$$\frac{T_2-T_1}{T_2+T_1}*\left(\frac{P_o*A_d}{V_g}\right)=0.5\rho_w*g\left(\frac{L_d}{2H+L_d}\right)+\frac{8\mu_w}{\rho_w}\left(\frac{2H+L_d}{L_d}\right)*\left(\frac{\rho_o Qgh}{0.5\,\pi Q}\right) \quad (9)$$

$$\frac{T_2-T_1}{T_2+T_1}*\left(\frac{P_o*A_d}{V_g}\right)=0.5\rho_w*g\left(\frac{L_d}{2H+L_d}\right)+\frac{8\,\mu_w}{\rho_w}\left(\frac{2H+L_d}{L_d}\right)*\left(\frac{\rho_o gh}{0.5\,\pi Q}\right) \quad (10)$$

$$\frac{32\,\mu_w*h*g*\frac{\rho_o}{\rho_w}}{\pi Q*d_i}=\frac{2P_o*A_d*T_i}{V_g}-\rho_w*g*d_i \quad (11)$$

$$Q=\frac{\left(\frac{32}{d_i\pi}\right)\mu_w*h*g*\left(\frac{\rho_o}{\rho_w}\right)}{\left(\frac{2P_o*Ad*Ti}{V_g}\right)-(\rho w*g*d_i)} \quad (12)$$

$$Q=\frac{\left(\frac{32}{d_i\pi}\right)\mu_w*h*g*\left(\frac{\rho_o}{\rho_w}\right)}{(\rho_w*g*di)\left[\left(\frac{2Po*Ad*T_i}{\rho w*g*d_i*V_g}\right)-1\right]} \quad (13)$$

$$Q=\frac{\left(\frac{32}{d_i*d_i\pi}\right)\left(\frac{\mu_w}{\rho_w}\right)*h*\left(\frac{\rho_o}{\rho_w}\right)}{\left[\left(\frac{2P_o*A_d*T_i}{\rho_w*g*d_i*V_g}\right)-1\right]} \quad (14)$$

$$Q=\frac{\left(\frac{10.19}{d_i^2}\right)\left(\frac{\mu_w}{\rho_w}\right)\left(\frac{\rho_o}{\rho_w}\right)}{\left[\left[2\left(\frac{P_o}{\rho_w*g}\right)*\left(\frac{A_d}{V_g}\right)*\left(\frac{T_i}{d_i}\right)\right]-1\right]}h \quad (15)$$

$$\frac{\mu_w}{\rho_w}=\gamma_w \quad and, \quad Ad=\frac{\pi}{4}D_o^2$$

*Substituting in equ. :* $$Q=\frac{40.76*\gamma_w*h*\left(\frac{\rho_o}{\rho_w}\right)}{\left[\left[0.26\,T_i\left(\frac{P_o}{\rho_w}\right)*\left(\frac{1}{m}\right)\right]-1\right]} \quad (16)$$

*Where* $m = n^2L_d$ *and* $n = d_g/d_d$

*Also:* $Q= 2RA_o/period$ *and* $Ad=\frac{\pi}{4}D_o^2=1.57R\,D_o^2$

**Table 3. volumetric flow rate formulas for different working liquids**

| *working liquid* | *Water* | *Mercury* | *Olive oil* |
|---|---|---|---|
| $\rho_w$ *kg/m*$^3$ | 1000 | 13590 | 800 – 920 |
| $\rho_o/\rho_w$ | 1 | 0.075 | 1.12 |
| $\mu_w$ *pas.s* | 0.001 | $1.55*10^{-3}$ | 0.03 |
| $\gamma_w$ *m*$^2$*/s* | 1.0035*10-6 | $0.11*10^{-6}$ | $33.2*10^{-6}$ |
| Q m$^3$/second | $\frac{(40.76*10^{\wedge}-6)*(h)}{\left[\left[0.26*10^{-3}*T_i*\left(\frac{P_o}{m}\right)\right]-1\right]}$ (17) | $\frac{(0.34*10^{-6})*(h)}{\left[\left[0.02*10^{-3}*T_i*\left(\frac{P_o}{m}\right)\right]-1\right]}$ (18) | $\frac{(1.4*10^{-3})*(h)}{\left[\left[0.3*10^{-3}*T_i*\left(\frac{P_o}{m}\right)\right]-1\right]}$ (19) |
| Amplitude –R (m) | $(4/\omega)*\left(\frac{Q}{D_o^2}\right)$ *(20)* | | |

Where di=0.5 and the gas area is half-circle
General formula : from equ. (15)

*Multiplying* $\left(\frac{A_d}{V_g}\right)$ *by* $(L_w/L_w)$

$$Q = \frac{\left(\frac{10.19}{d_i^2}\right)*\gamma_w*h*\left(\frac{V_g}{V_w}\right)*\left(\frac{\rho_o}{\rho_w}\right)}{\left[\left[0.2038\ Ti\left(\frac{P_o}{\rho_w}\right)*\left(\frac{1}{L_d}\right)\right]-\left(\frac{V_g}{V_w}\right)\right]} \qquad (21)$$

## 2.2.2- Output length : -

*From equ. (3)* $\omega^2 = g/L_d$

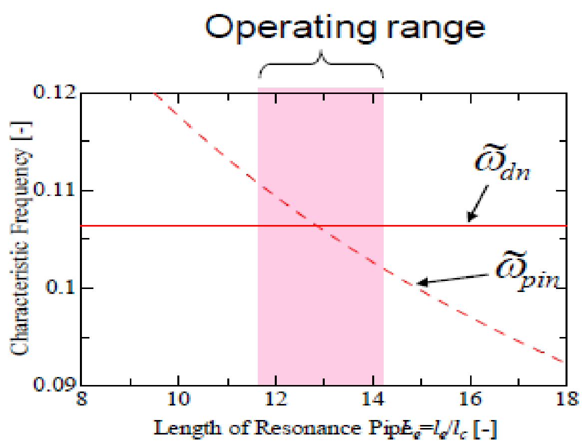
$$\therefore L_o = 0.065\ (P_o/\rho_w)(D_o/D_g)^2 \qquad \text{For 1-atm } P_o \qquad (22)$$

**Table 4. output length approximate formula**

| Working liquid | water | Mercury | Olive oil |
|---|---|---|---|
| Output length | $6.5\ (d_o/d_g)^2$ (23) | $0.48\ (d_o/d_g)^2$ (24) | $7.5\ (d_o/d_g)^2$ (25) |

*Notice : In fact there are two frequencies – not one –*
$\omega -$ *for the displacer and* $\omega -$ *for the output tube* $(fig - 9)$ *They will be equal at the optimum output length* $L_o$ *which assist the condition[13]*

$$Lo = \frac{Po * Ao}{\rho w * Vg * \omega^2}$$




$fig - 9 \quad \omega\ dn - displacer\ \omega\ \&\ \omega\ pin - output\ \omega$ (**STUDY ON WATER-TYPE STIRLING ENGINE AND ITS REVERSE CYCLE-Yoshiyuki Yamaguchi)**

## 2.3 Independent variables

Now it can be re-classified the parameters as dependent and independent variables

**Table 5. independent variables**

| independent variable | | Suggested range | Suggested step |
|---|---|---|---|
| $L_d$ | horizontal length of displacer | 0.05-1.0 m | 0.05 m |
| $D_d$ | Displacer diameter | 0.00625 : .05 m | 0.00625 |
| $D_g$ | Gas tune diameter | | |
| $T_2$ | Hot zone | 45 – 90 °C | 0.5 |

| | | | |
|---|---|---|---|
| | temperature | | |
| h | Total head | * | |
| $P_o$ | Output pressure | ** | |
| $\rho_w$ | Working liquid density | 1000 kg/m$^3$ | 0.5 |
| $T_1$ | cold zone temperature | 25-45°C | |

* Total head=1m
** $P_o$ is atmosphere = 101.0 k pas

## 2.4 Efficiency

Efficiency: $\eta\% = \frac{output\ work}{energy\ added} * 100$ (26)

To get work done : W= power per for period time = ρQgh* period time (27)

∵ Period time = 2π/ω ∴ W = ( ρQgh )* 2π/ω (28)

For water as working liquid and ω = 1 rad / s

Work done = $6.16*10^4$ Q* h (29)

Where Q is for period time of ω , not for 1-one hour

To get energy –heat- added

$Q_{heat} = m\, c_p \Delta T$ (30)

M(mass)= $\rho_g * V_g$

$\rho_g$ = 1.21 kg/m^3 (30 °C ) &$c_p$= 1.006 kJ/kg.K

Mass = $1.5\, D_g^2 L_d$ (31)

$Q_{heat} = m\, c_p\, \Delta T = 1006 * 1.5\, D_g^2 L_d * \Delta T$ (32)

$Q_{heat} = 1509\, D_g^2 L_d * \Delta T$ (33)

So , : $\eta = 1562\, Q\, h / 1509\, D_g^2 L_d * \Delta T$ 1562/1509 ≈ 1

$$\eta = \frac{(\rho Qgh) * 2\pi/\omega}{1509 Dg2 * Ld * \Delta T} \quad (34)$$

$$\eta = \frac{5.34*10^{-4} * (Ld)^{0.5}}{\{[0.26*10^{-3} * Ti * Po * dd^2] - (dg^2 * Ld)\}\, \Delta T} \quad (35)$$

$$For\ general\ formula: \ \eta = \frac{0.002 * h^2}{\omega * \Delta T * V_w \left[\left[0.2038\left(\frac{Ti * P_o}{L_d * \rho_w}\right)\right] - \left(\frac{V_g}{V_w}\right)\right]} \quad (36)$$

Notice : ∵ $b = \rho_o gh / 0.5Q\, (\pi/A_o)^2$ and $Q = 2RA_o$

$\frac{g}{\pi 2} \approx 1$ ,So ,$b = \rho\, h\, A_o / R$ (37)

Work done = force * displacement

∵ $W = (b * \ddot{H}_o) * R$ (38)

$\ddot{H}_o$ is the acceleration of output column displacement

∴ $\ddot{H}_o = (\omega\ R) * g$ (39)

# 3 OPTIMIZATION

4.1 Using "GENTIC" algorithm (MATLAB)

Input data of table (5) and eff. Equation

$$\eta = \frac{5.34*10^{-4} * (Ld)^{0.5}}{\{[0.26*10^{-3} * Ti * Po * dd^2] - (dg^2 * Ld)\}\Delta T}\ h^2 \quad (40)$$

For 1-m head (h2 = 1 ) and Po = 1 atm. (101 k pas) and the water as working liquid and output liquid while air is the working gas.

The results for Water as working liquid finding max. eff. And comparing between F.P. and Stirling efficiency for the same conditions. It is cleared that the efficiency of F.P. is limited by Stirling efficiency (fig-10) and can't exceed it.

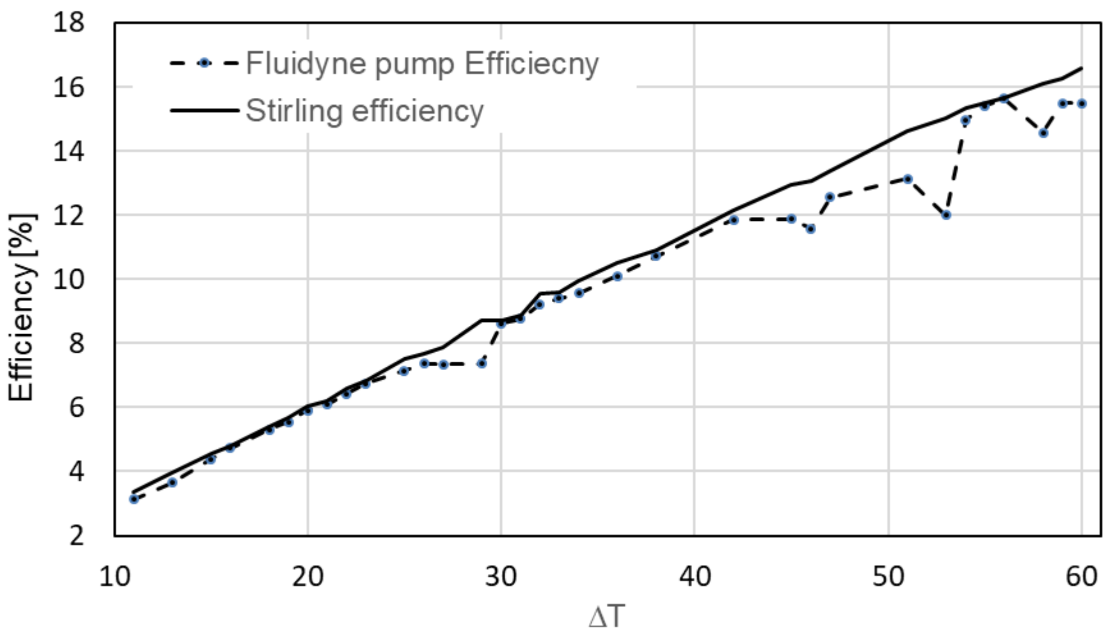



**Figure10. Stirling and fluidyne efficiency**

The efficiency related to gas and liquid volume (and mass) ratio, and temperature factor ($T_i$) showed that the largest area is the efficiencies lower than 5% ( fig-11) while the smallest area expresses the efficiencies greater than 10 %. It showed that the efficiency can maximized by following the mathematical relation showed in equ. (21) and (36), that cleared the mathematical difference reducing the denominator, is the most important relation that increases the output flow rate, work, and efficiency

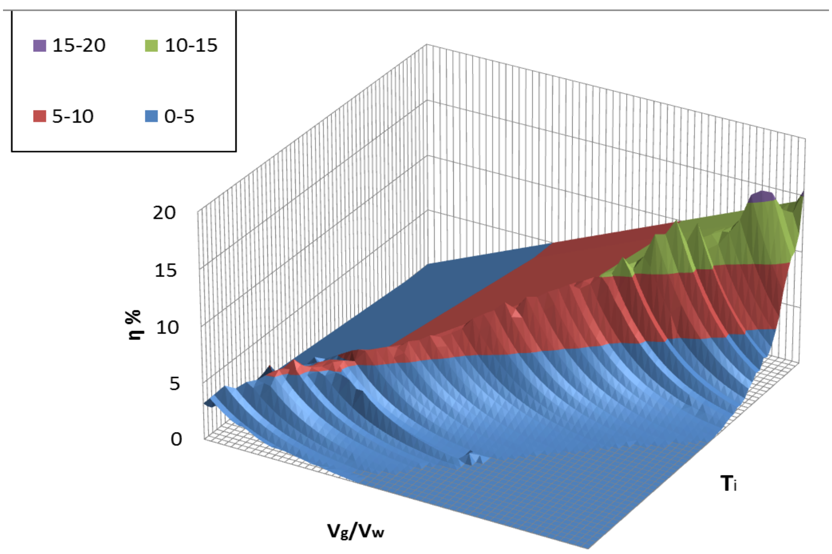



***Figure 11.*** $T_i$ **&** $\frac{V_g}{V_w}$ ***and*** $\eta$ ***relation***

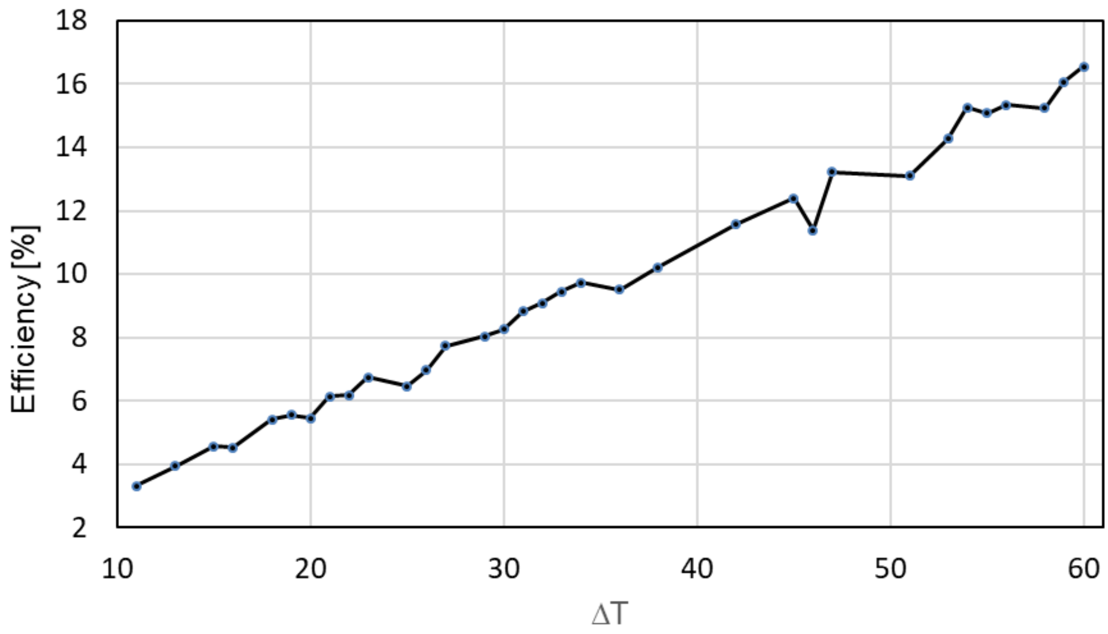



**Figure 12. Effect of ΔT on the efficiency**

Analyzing of temperature difference between hot and cold zones (ΔT) and its effect on efficiency, showed that the clear effect of increasing ΔT (fig- 12) leadsto a marked increase in efficiency, which means how coooling the cold zone is needed.

# 4 EXPERIMENTAL STUDY

## 4.1 Setup

To ensure the effect of different parameters like Ti and ($V_g$ / $V_w$) on the performance of the system in the optimum conditions derived from the mathematical model, experimental validation was necessary. A model of FP is assembled of 0.5 inch (1.125cm) diameter cast iron tubes. Two vertical tubes with 57 cm height represents the hot and cold tubes, while others are horizontal with 20 cm length represent the regeneration, displacer , and output tubes. All the tubes are assembled by cast iron joints : T-joints and $90^0$-elbows - fig.(13,14). To simulate the solar thermal power, a 1000 watt - circular electrical heater surrounding the hot tube and insulated from the ambient by thermal bricks (0.05 w/m. K conductivity) and glass wool (0.04 w/m. K conductivity), and 220 to110 volts AC Variac (1 K.W.) to supply the heater by different AC electrical power. The experiment includes loading and un-loading systems. By using 4-four thermocouples T-type with about ±0.5% limit of error for temperature range 21:100 °C.

## 4.2 Precautions

1. The system includes a 0.5" plug in the end of cold tube, to adjust the level of water in the vertical tubes by noticing the level in the output tube only. In the beginning of the experiment , the plug must be opened to adjust the pressure on the water surface, that a due to make the water level is the same in the vertical tubes. The plug then is closed tightly to avoid any leakage.
2. To make sure that the thermocouples measure the air temperature, not the tube's wall temperatures, the thermo-couple wires entrance for about 0.6 cm in the tubes and then insulated.
3. The points of thermocouples entrance (especially in the hot and cold zone) are above the water levels by a suitable height – 30 cm from the bottom. The vertical distance is far from

the maximum expected amplitude for the maximum level of water ( max. height of water is 20 cm , max. amplitude is 7 cm) that ensure that the wires doesn't touch the water surface.

## 4.3 Experiment procedure

By opening the plug, the pressure will be the same for all the tubes. Then water is entered from the end of the output tube, or the plug hole. As a result, the level of water will be the same for the three vertical tubes. The chosen points of the $V_g$ / $V_w$ are determined by the output level scale (table 6 ) . After raising the water to the required level , the plug is closed tightly. The volume of the gas and working liquid (water) are calculated by using the geometric formulas (table-6) , and the ratio ($V_g$ / $V_w$) can be then calculated.

Then adjusting the "Variac" at the chosen voltage , while the "clamping amperometer" will measure the current , that allow knowing the heater input power from the relation : power = current * voltage * power factor where power factor equals to 0.8:0.9

The 1 kw variac will supply the heater by the required power for heating for all the selected power input points . Stop watch used for knowing the on-set time , and the period time by measuring the total cycles for 1-minute for every case and then the frequency , ω , and flow rate. The experiment included three cases for gas to liquid volume ratio ($V_g/V_w$) of 3,4 and 5 (table. 6) with five cases for the input heating power (60,7 0,80 ,90 ,and 100 watts) .

Reading $T_1$ &$T'_1$&$T'_2$ and $T_2$ using T-type thermo-couples with a thermocouple reader (AX4 – 1A , multi input and multi output) for T-type sensor , 0°C to +400°C, and ±0.03°C error fig.14. (fig-16, 17) and then calculating the mean temperature for hot and cold tube ($T'_2$ and $T'_1$ are the temperatures of regeneration zone edges) , then calculated ( $T_i = \frac{T2-T1}{T2+T1}$) .

Volumetric flow rate can be calculated, and then the output work and efficiency .The output water will start after "on-set" point which is calculated by a stop watch , and then the pumped water quantity is calculated by using a balance .

## 4.4 Measured parameters

1- temperatures of : hot zone , regeneration zone edges and cold zone temperatures – fig 14
2- The on-set time - minutes
3- The time period (seconds) , frequency and ω.(rad/second)
4- The volumetric flow rate (Q) – $m^3$/s.
5-The other parameters can be calculated from the measuring data , like the FP efficiency and the output work.

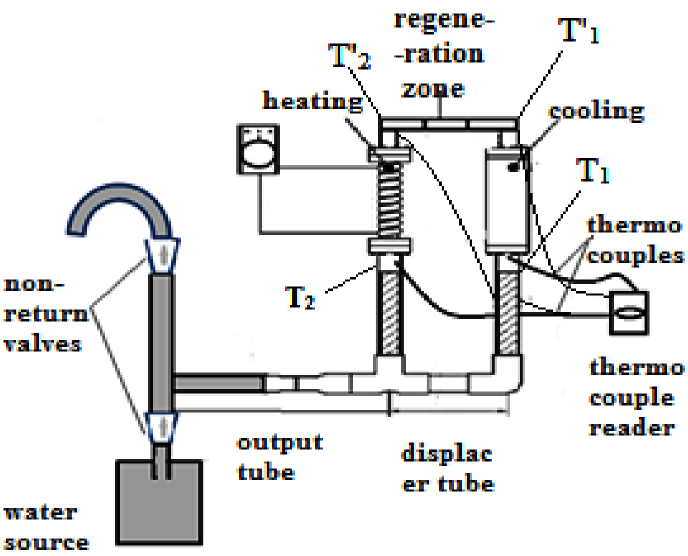



**Figure13. test-rig of sketch of FP parameters**


**Figure14. Test rig of un-loading FP**

**1-1 kw Variac** **2- cold zone tube - 0.5" diameter**
**3-hot zone tube – 0.5" diameter** **4- clamping amperometer**
**5-0.5" plug** **6- .5" regeneration tube**
**7- glass wool surrounded the thermal bricks and the circulating heater**
**8- Thermocouples reader** **9- 0.5" output diameter**
**10- Thermocouples T-type**


**Figure 15. loading system test rig**

**1- Non-return valves 2- On/Off valve 3-Circulated heater surrounded by thermal bricks and glass wool 4-Thermo-couple reader 5-Thermo-couples**

**Table 6. Model's dimensions**

| case | $D_g$ cm | $D_d$ cm | $D_o$ cm | $L_d$ cm | $L_g$ cm | $L_w$ cm | $L_o$ cm | $V_g$ / $V_w$ | $H_o$ cm |
|---|---|---|---|---|---|---|---|---|---|
| 1 | 1.25 | | | 20 | 125 | 25 | 50 | 5 | 2.5 |
| 2 | 1.25 | | | 20 | 120 | 30 | 50 | 4 | 5 |
| 3 | 1.25 | | | 20 | 112 | 38 | 50 | 3 | 9 |

## 4.5 Experimental results

The reading of 15-fiveteen cases parameters, and calculating other parameters (work and efficiency) has been listed and comparing with the theoretical results .

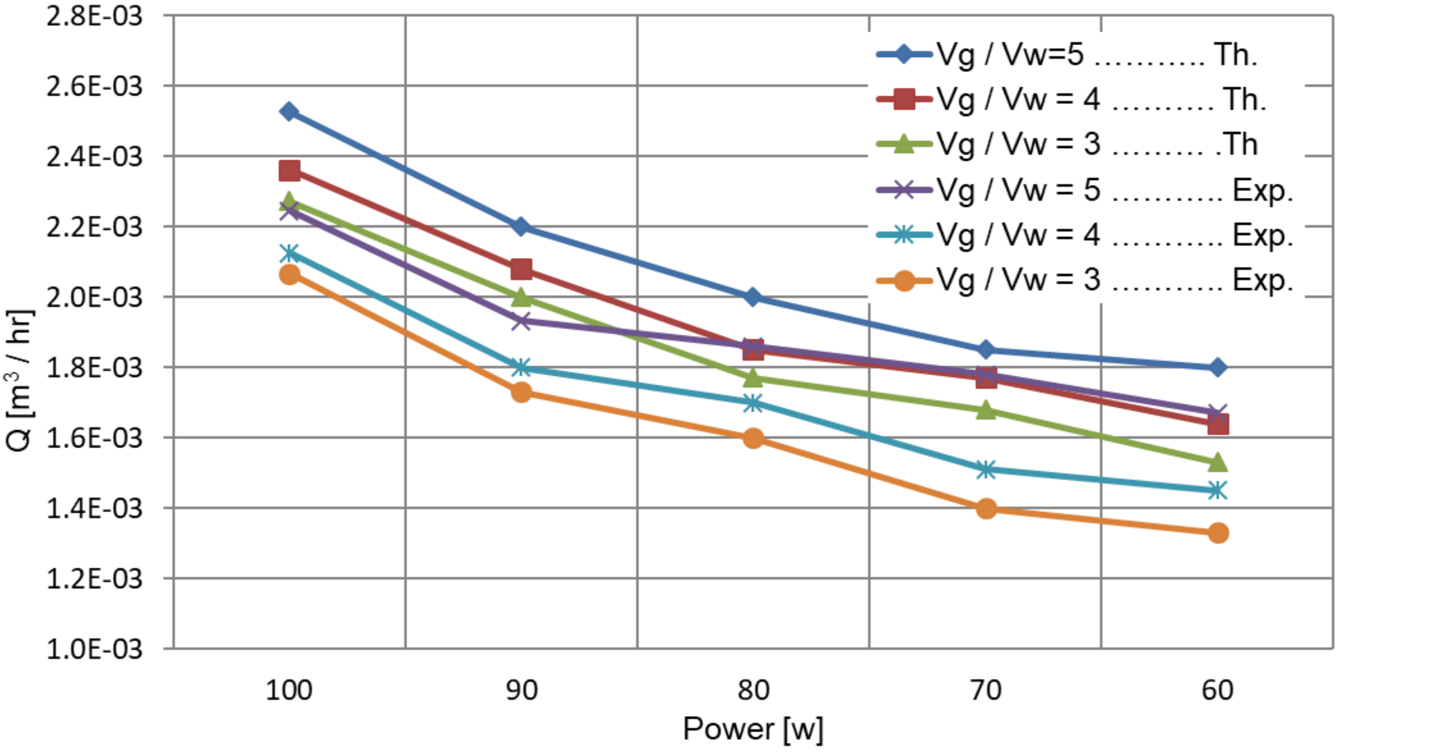
Figures (16 a, b)showed that the output increases with 'working gas to liquid' ratio increasing, and input power; while the efficiency seems higher with the maximum ratio of gas to liquid, and the minimum input power. This result – for efficiency- didn't include the on-set time; in other words, the efficiency is measured after start lifting the water, so , the power consumed from start heating to start lifting the water isn't within calculating the efficiency.

The results clarified a difference between the theoretical and experimental results( fig.16 ,17) gotten from equations (21) and (36), this difference seems logically for this reasons :-

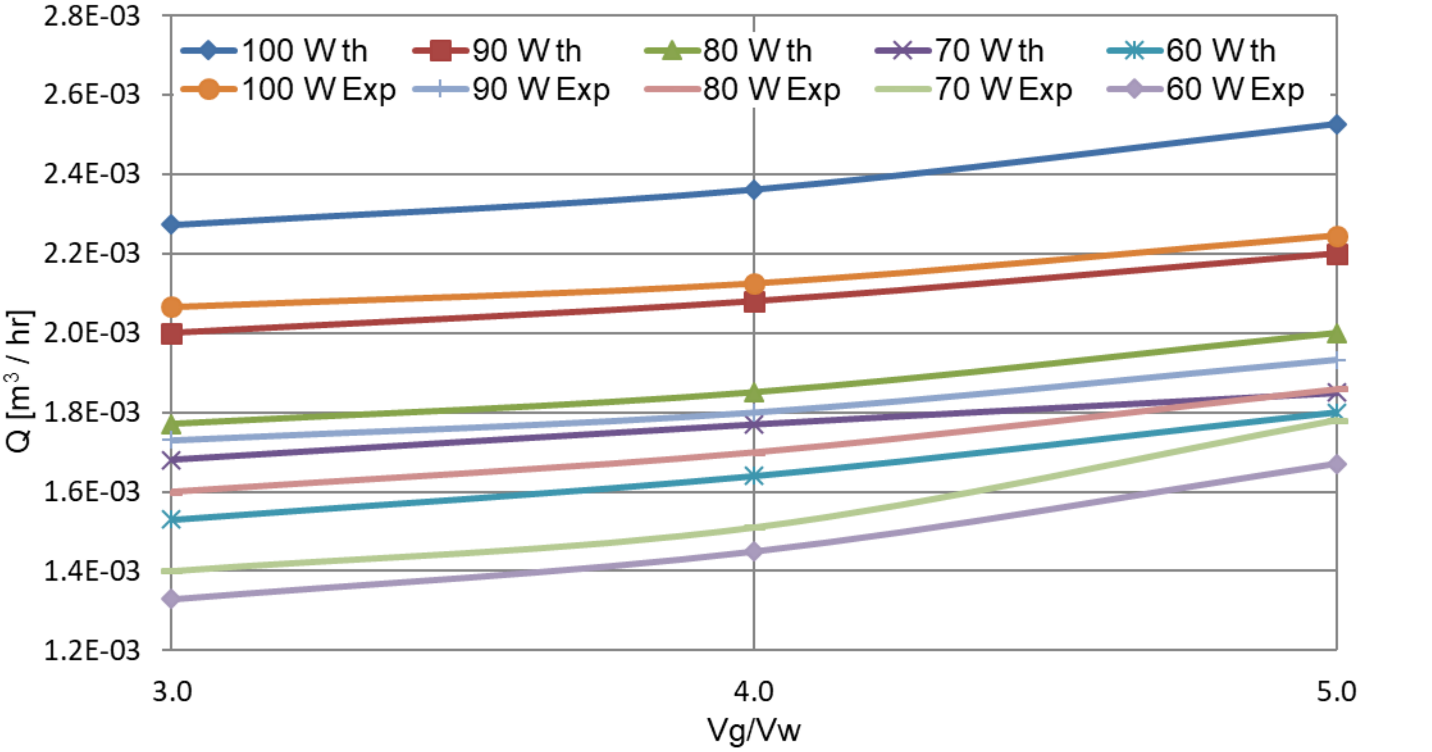
1.The theoretical results were remarked for the 1 atm- output pressure, $d_i$=0.5 and $L_g = ( \prod /2 ) *L_d$ , where the experiment study includes a different cases of $d_i$, and $L_g$.

2.Experiment cleared that the pressure in the output is higher than 1 atm. Because of the type of non-return valve ,so , the spring of the valves , mitigated .Also, this reason (output pressure > 1 atm.) made "$T_i$" higher than the theoretical , and then the efficiency lower than the theoretical.

3.The heater in the experiments was put surrounding the cast iron tube which consumed amount of input power that leads top power losses.

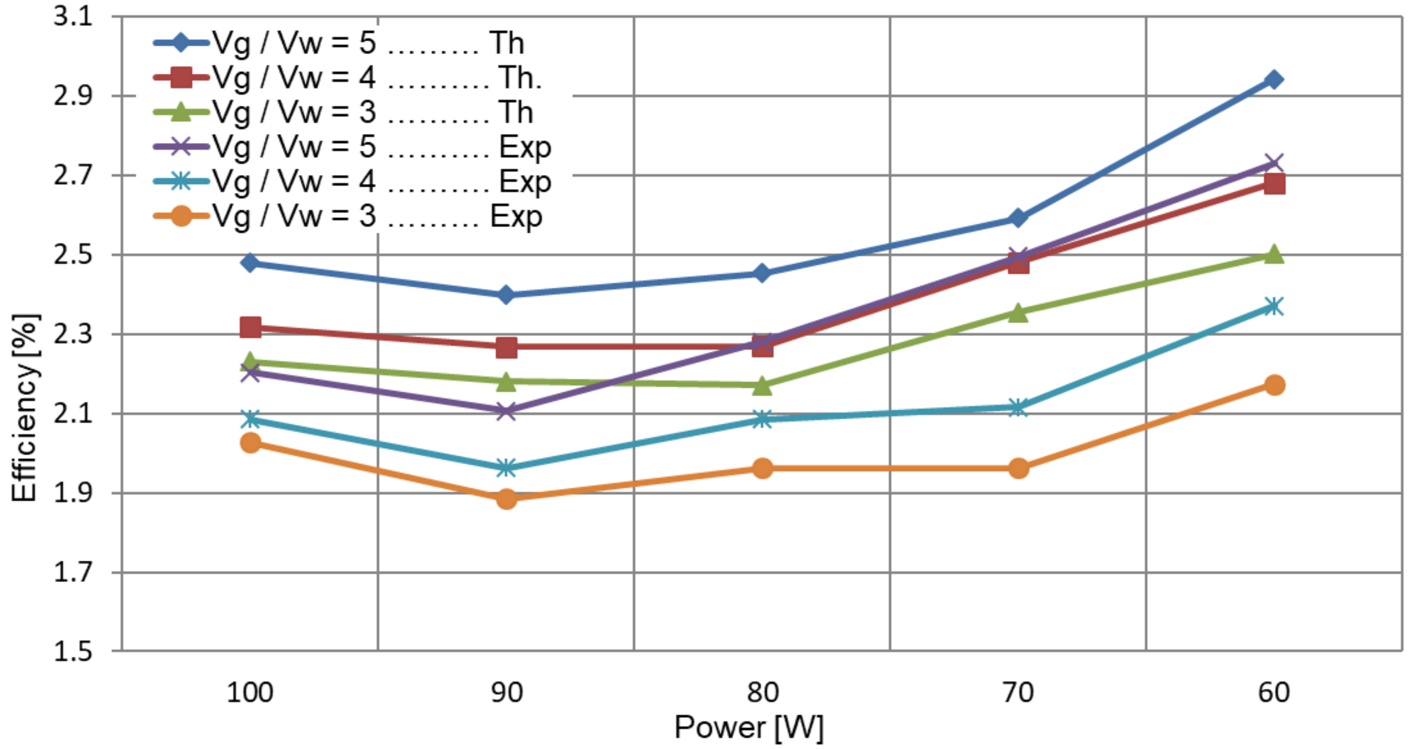



(a)

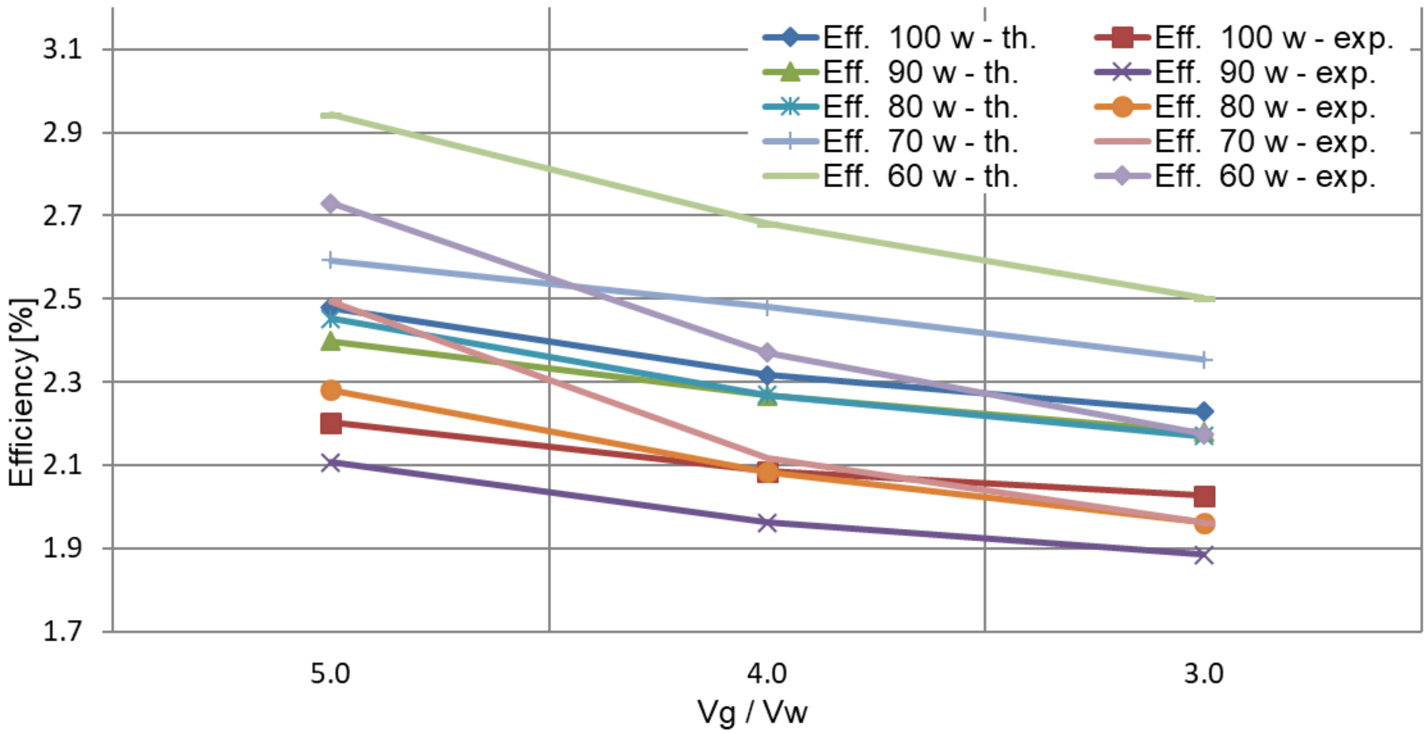



(b)

**Figure 16. (a) Theoritical and experimental flow rate WRT power , (b)Theoritical and experimental flow rate WRT Vg/Vw**




**(a)**




(b)

**Figure 17. (a) Theoritical and experimental efficiency wrt power , (b) Theoritical and experimental efficiency wrt Vg/Vw**

4.The "$V_g$ /$V_w$" is not completely accurate because the working air zone includes the regeneration zone, where the regeneration material occupies a space . This space is calculated by making a ball or cylinder of the regeneration material and then deducted from the total space of the working gas. This method is not accurate because the regeneration material is not coherent.

## 5 CONCLUSION

The study included the analysis of the basic equations to obtain the main non-dependent variables, and then try to reach the highest efficiency of the pump, using optimization programs, and then check the results experimentally. It is cleared that F.P. can used efficiently by reducing the mathematical difference that related with output pressure, input power , and geometrical dimensions of working gas and liquid zones. The results can summarized in :

1. The main independent values are : $V_g$ ,$V_w$ ,$T_i$ ,$D_o$ ,$\rho_w$ ,and $P_o$
2. The higher $\Delta T$ , the higher efficiency , so, cooling of cold zone is preferred .
3. Higher efficiency needs $V_g$ larger than $V_w$ ($V_g$ / $V_w$ > 1)
4. FP efficiency is limited by stirling engine ( $\eta_{fluidyne} \leq \eta_{stirling}$ )
5. Maximum FP output requires the difference : {(0.2038 $P_o T_i / \rho_w L_d$ )-($V_g$ / $V_w$ ) } as less as possible

## ABBREVIATIONS

$A_d$ :cross section area of displacer tube[m$^2$]
$A_o$ :cross section area of output tube[m$^2$]
$A_g$ :cross section area of gas tube[m$^2$]
$V_g$ : working gas volume[ m$^3$]
$V_w$ : working liquid volume [ m$^3$]
$D_d$ : diameter of displacer tube [m]
$D_g$ diameter of gas tube[m]
$D_o$: diameter of output tube[m]
$H_h$ : heights of output column [m]
$H_c$ : heights of cold column[m]
$H_o$: heights of output column[m]
H : H= $H_o$= $H_c$= $H_h$ , for balancing case
g: gravity acceleration[m/s$^2$]
$L_d$ : horizontal length of displacer [m]
$L_o$: total length of output tube [m]
$L_w$: total length of working liquid tube [m]
$L_g$ : total length of working gas tube [m]
$P_o$ : output tube pressure [pas]
$P_a$ : atmospheric pressure [101 k.pas]
$\gamma_w$ : Kinematic viscosity of working liquid[m$^2$/s]
$\rho_o$ : density of output liquid[kg/m$^3$]
$\rho_w$ : density of working liquid[kg/m$^3$]
ω: radian frequency [rad/sec]
R: Amplitude [m]
h : total head[m]
Q : volumetric flow rate [m$^3$/s]
$T_1$ : temperatures of cold zone [k$^0$,C$^0$]
$T_2$: temperatures of hot zone [k$^o$, C$^o$]
b : velocity dependent load coefficient[N.m/s]
$\mu_w$ : absolute viscosity for working liquid[pas.s]

Subscription :
n : non-dimensional ratio – n = $D_g/D_d$ [-]
$d_i$ :working liquid length factor $d_i=\frac{Ld}{2H+Ld}$ [-]
$T_i$ : temperature factor $T_i=\frac{T2-T1}{T2+T1}$ [-]
m – displacer factor m = $n^2 * L_d$ [m]

**ABBREVIATIONS :**
STPS: solar thermal pumping systems
MSTPS: mechanical solar thermal pumping systems
NMSTPS : non-mechanical solar thermal pumping systems
FP. : fluidyne pump
JSFP : jet-stream fluidyne pump
DF : displacer factor
PV : photovoltaic

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