CHAPTER 6

CONCLUSIONS AND FUTURE WORK

Chapter (6)

CONCLUSIONS AND FURTHER WORK

6.1. Conclusions

In the present study, the pump was tested in both direct (pump mode) and reverse (turbine mode) experimentally and numerically to investigate their performance for two sets of speeds of rotation. This study is widely applied in developing countries and rural areas, which is very significan solution for energy problems in Egypt.

The main starting points that can be drawn from the experimental results in this research:

- A pump could be operated as a turbine at different speeds of rotation without mechanical problems.
- An inline centrifugal pump which is considered a special type with low power could operate effectively as a turbine.
- An electronic load controller could control the operation of the turbine mode and fix the speed of rotation while the flow rate changes and could control the pump operation as well by adding batteries as a power source.

On the other hand, from the numerical results some conclusions are derived as:

- The flow through the inline pump could be simulated in pump mode and turbine mode.
- The shape of the volute casing has a great effect on the results at the pump mode.
- The numerical results are affected by the losses in the volute casing so the results have a difference with the experimental ones.

The comparison between the computational and experimental results shows good agreement at the turbine mode and shows a slight deviation at the pump mode.

6.2. Recommendations for future work

A parametric study should be conducted for the pump mode and turbine mode to study the effect of different parameters on the performance experimentally. These parameters are:

- The effect of impeller trimming on the performance of the pump and turbine modes.
- The effect of changing the outlet pipe geometry at the turbine mode.
- Adding two pumps in parallel and studying the effect of running two PATs in parallel instead of using a valve which dissipates energy to regulate the flow.
- Testing different inline pumps as turbines with different specific speeds and conclude correlations for each range of specific speed.

- Using a booster pump before the PAT at the turbine mode in order to test the pump at higher flow rates.

On the other hand, concerning the Numerical model a parametric study should be conducted for both modes to study the effect of different parameters on the performance. These parameters are:

- Using a dynamic mesh rather than the multiple frames of reference is more accurate as it eliminates the interpolation errors in the MRF approach.
- Changing the shape of the volute casing to obtain more optimum shape.
- Simulating the pump flow for more sets of rotational speeds.
- The internal losses and other modifications of the pump geometry (i.e. optimum shape) are to be studied which is difficult to measure experimentally.
- Effect of pump blade angle and number of the blades on the performance of turbine mode.

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Appendix (A)

Pressure gauges calibration

The upstream and downstream pressure gauges used in the experimental setup were calibrated in the fluid mechanics laboratory using a pneumatic pressure tester (DRUCK- DPI 610). The calibration curves are as follows:

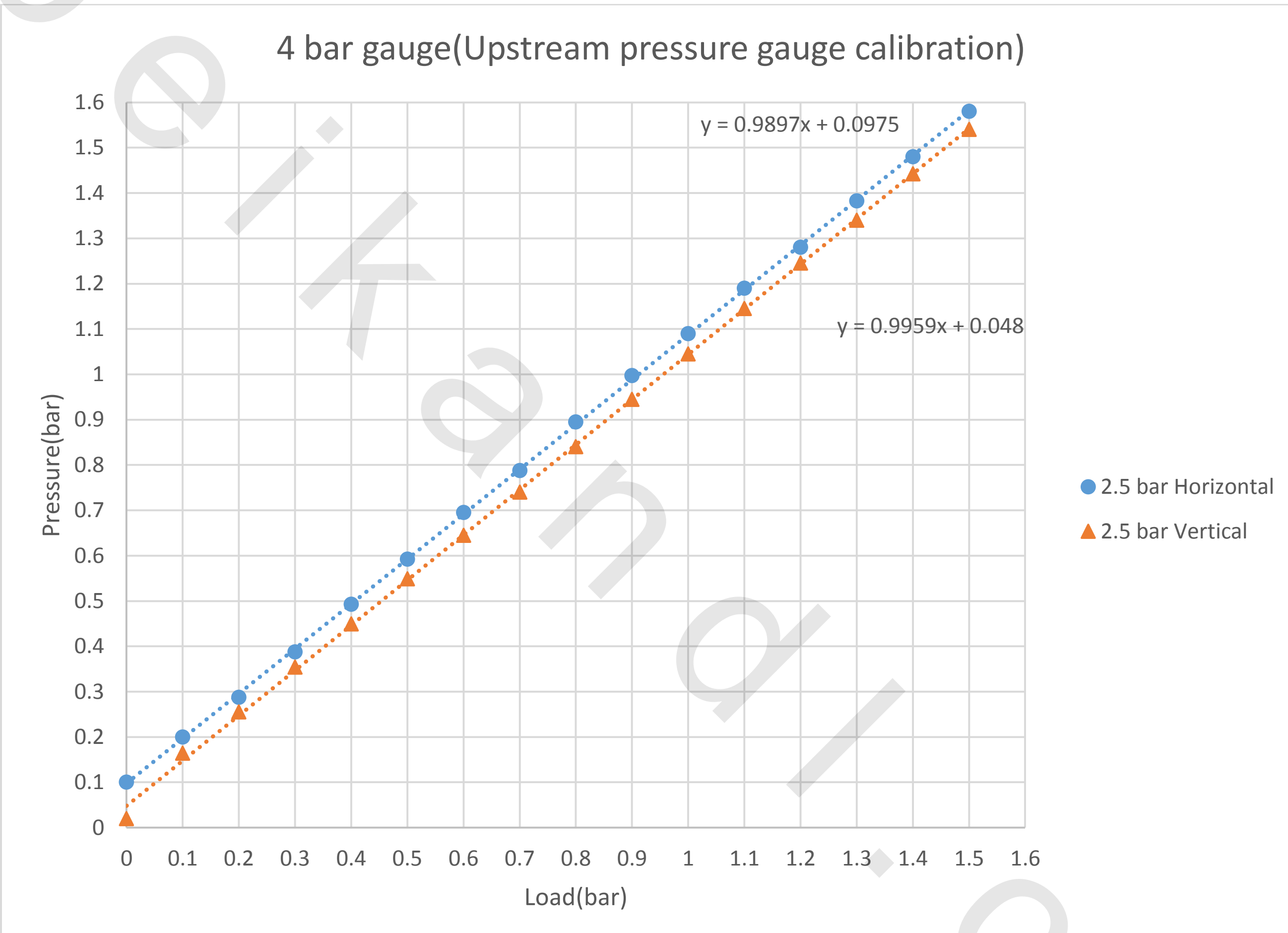
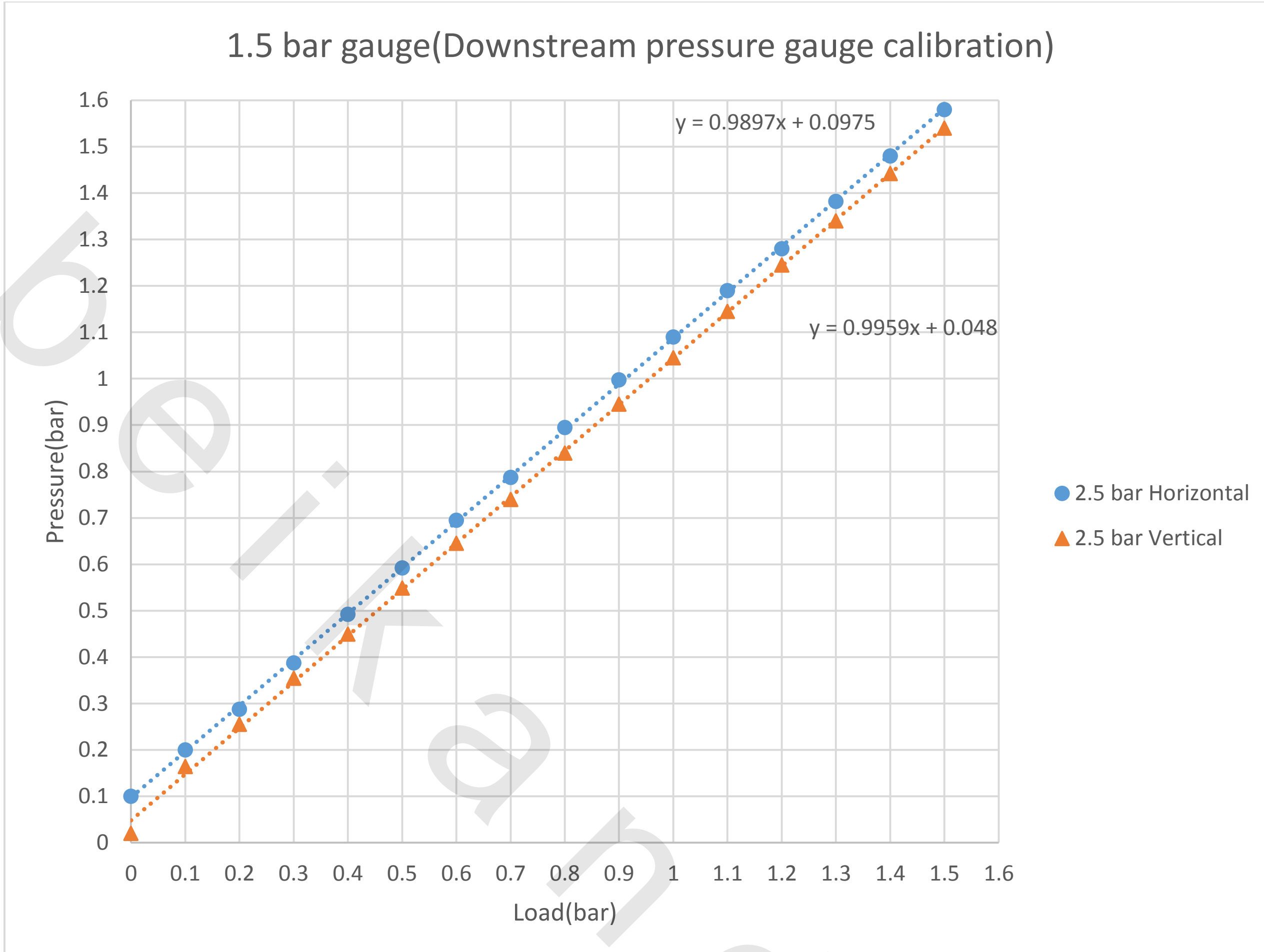


Figure (A-1) Pressure gauge calibration (4 bar)



Figure(A-2) pressure gauge calibration (1.5 bar)

Orifice calibration:

The orifice meter was calibrated in the Fluid mechanics laboratory. A tank of volume 26.719 liter was used in the calibration process.

The Foxboro Integral Orifice Flow Meter Assembly (IFOA) and the honed-orifice meter run are widely used in line sizes of ½ to 1 ½. They have corner taps, since flange or D and D/2 taps would be located in regions where pipe friction would influence the differential [26].

$$C = C_{\infty} + b/R_D^n$$

Where C_{∞} is a factor which could be estimated from the following equation:

$$C_{\infty} = a + b * \beta^{2.1} + c * \beta^8 + d * \frac{\beta^4}{1-\beta^4} + e * \beta^3$$

Coefficient b could be estimated from the following equation:

$$b = f + g * \beta + h * \beta^2 + i * \beta^3 + j * \beta^4$$

<u>Integral flow orifice assembly IFOA</u>

Square-edged D=1in

$$C_\infty = 0.6050 - 0.1837 * \beta^2 + 0.6615 * \beta^4 - 1.094 * \beta^8$$

Coefficient b equation

$$b = 1.646 * \beta^{0.5} + 2.394 * \beta^{\frac{5}{2}} - 4.899 * \beta^{\frac{9}{2}}$$

Exponent n=0.5

Recommended accuracy and restrictions for equations

<u>Honded meter runs</u>

Nominal pipe dia. D (in)	Beta Ratio β	Pipe Re R_D	Coefficient accuracy %
½ - 1½	0.1-0.8	>1000	±0.75

<u>Integral flow orifice assembly IFOA (Foxboro)</u>

Nominal pipe dia. D (in)	Beta Ratio β	Pipe Re R_D	Coefficient accuracy %
	STD sizes	$1500/\beta \le R_D/\beta$	±0.75

Plate thickness = 0.125 in for D<6 in (D*<150mm) & 0.2<β<0.75

C from the theoretical calculations is 0.67576 while it is 0.754 from the experimental results. This difference is due to the special design of the corner taps.

The relation between the velocity and C_D is plotted using calibration results. In addition, the relation between the Re and C_D based on bore diameter is plotted. The trend of both curves is similar to the real ones.

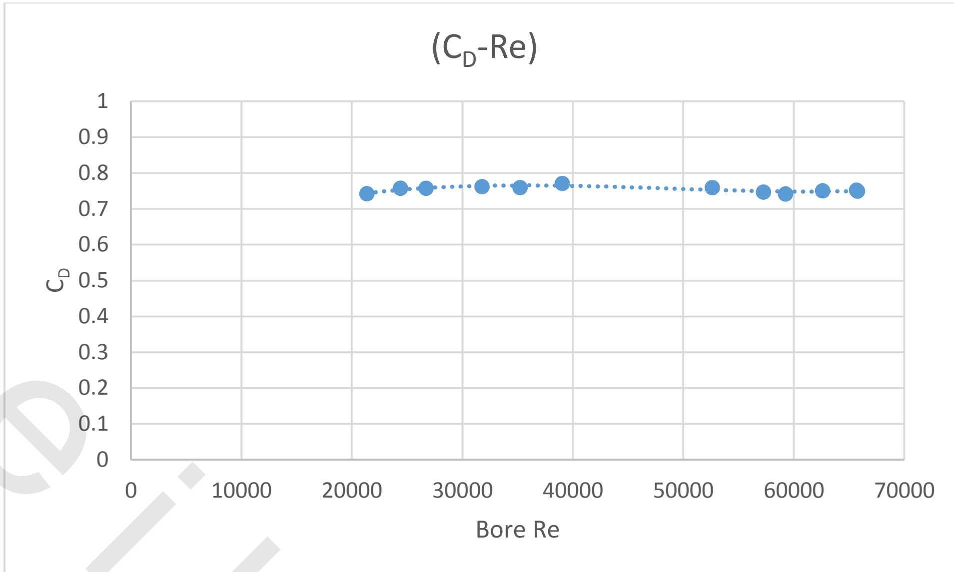


Figure (A-3) Calculated curve of orifice meter (Variation of orifice coefficient of discharge C_D with Reynolds number Re (based on orifice diameter)

Appendix (B)

Repeatability test

The repeatability of the results were tested in different speeds of rotation of the DC motor at the Fluid mechanics laboratory. The repeatability results are shown in Figures(B-1 to B-7)

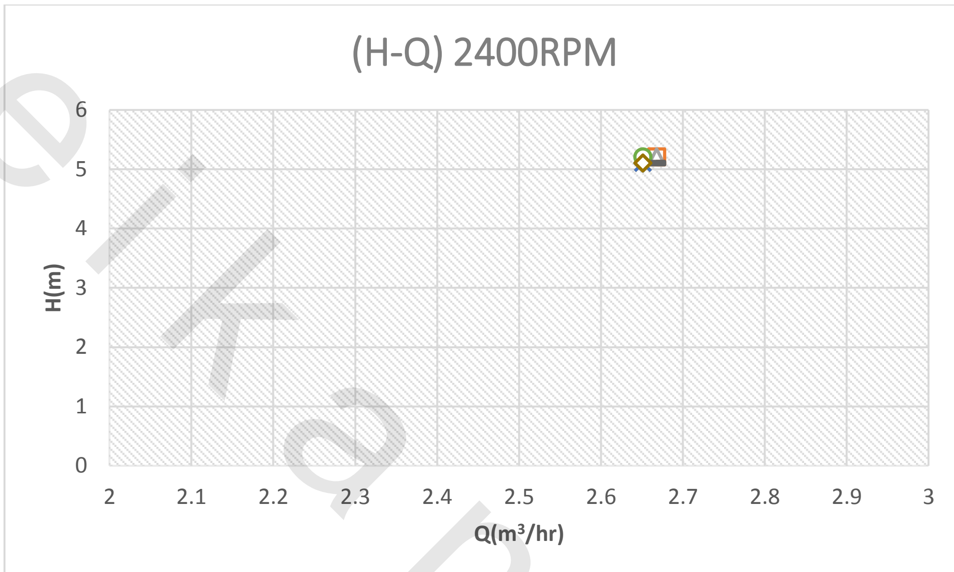


Figure (B-1) Repeatability test at 2400RPM

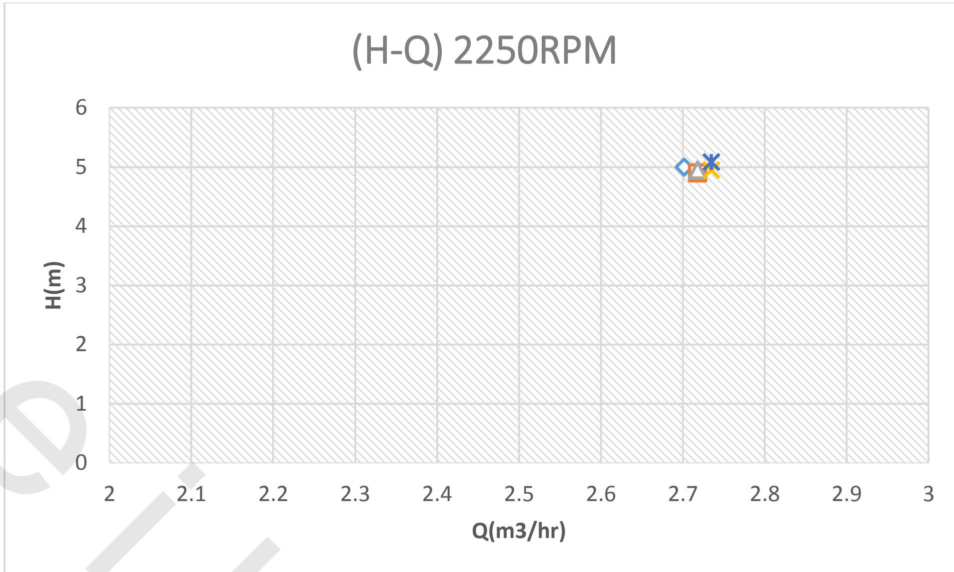


Figure (B-2) Repeatability test at 2250RPM

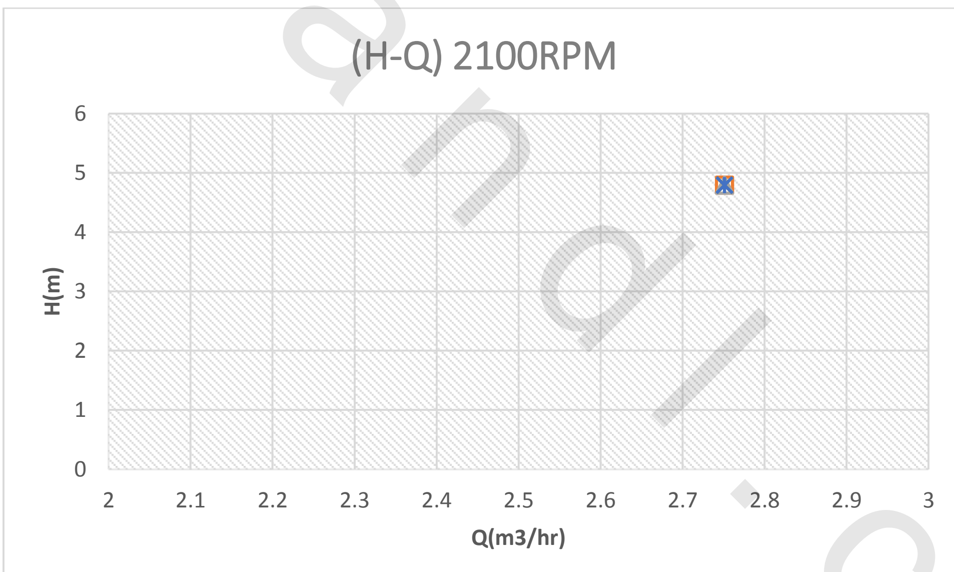


Figure (B-3) Repeatability test at 2100RPM

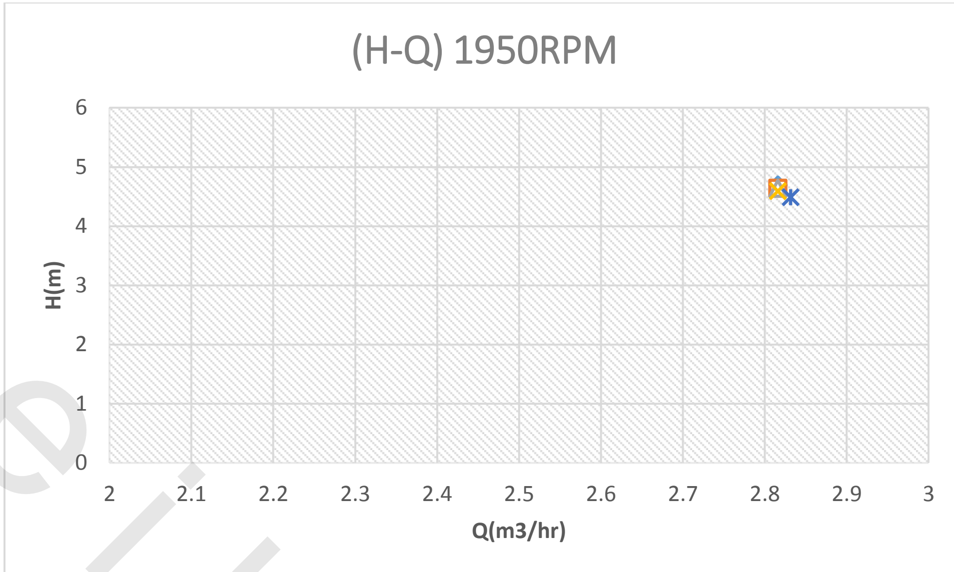


Figure (B-4) Repeatability test at 1950RPM

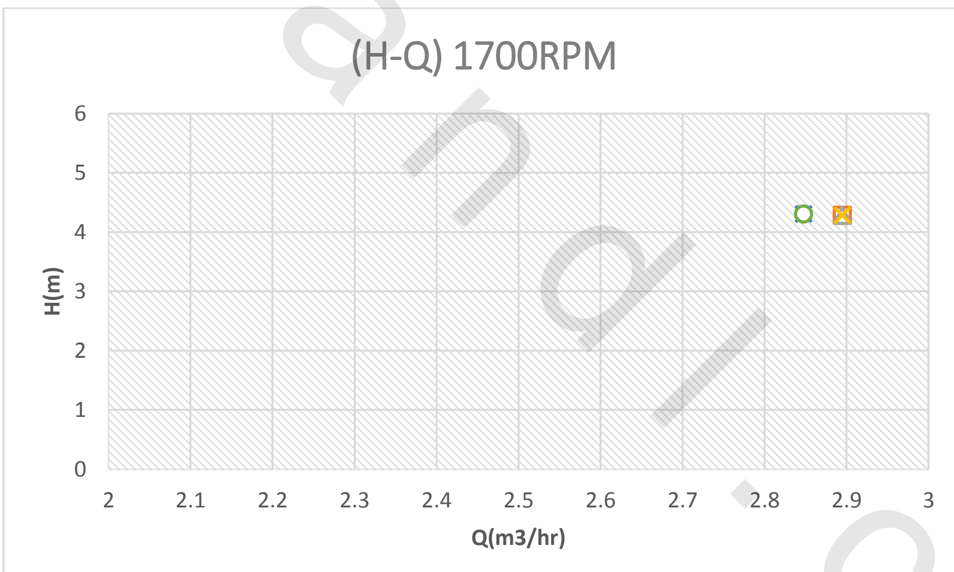


Figure (B-5) Repeatibility test at 1700RPM

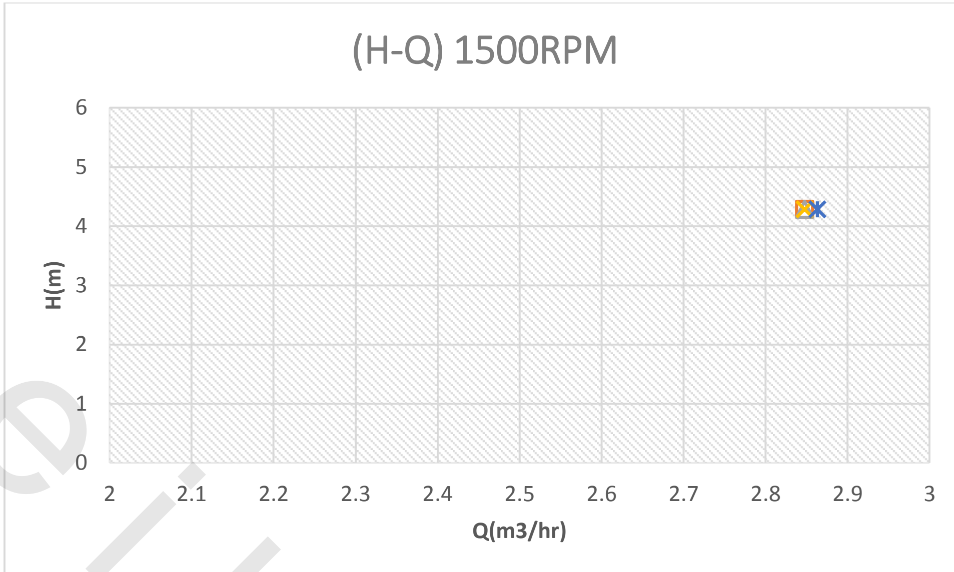


Figure (B-6) Repeatability test at 1500RPM

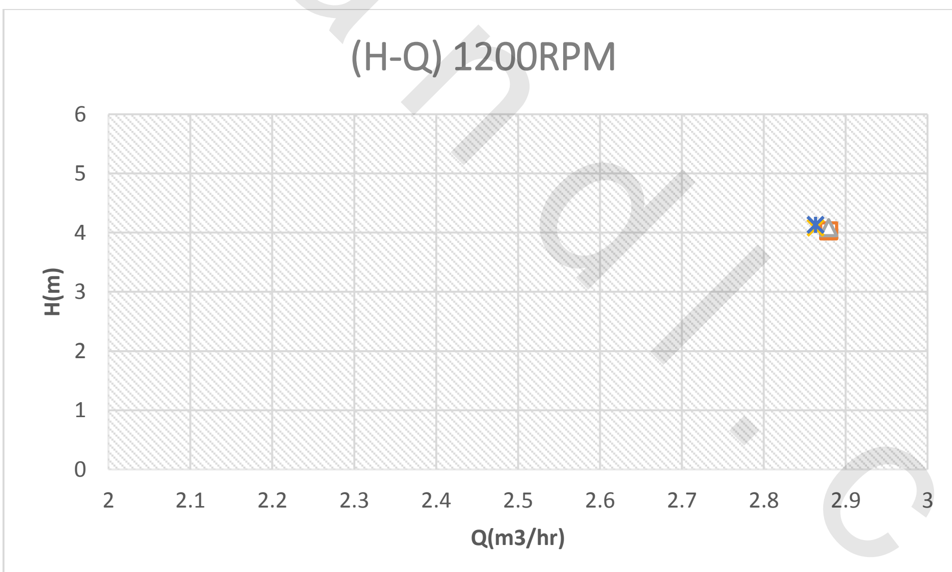


Figure (B-7) Repeatability test at 1200RPM

Uncertainty analysis:

Uncertainty analysis of volume flow rate

The volume flow rate passing through the orifice meter is a function of the U-tube manometer differential pressure reading, y, which is related to the pressure head, h, as follows:

$$Q = c_d a_o \sqrt{2gh}$$

Where $h = y\left(\frac{S.G_u}{S.G_f} - 1\right)$

Where c_d is the discharge coefficient which is calibrated experimentally to be 0.754, a_o is the bore diameter of the orifice, $S.G_u$ and $S.G_f$ are the specific gravity of the differential fluid and the fluid flowing through the pipe respectively.

As a result, the uncertainty of the volume flow rate depends on the uncertainty of the differential pressure reading of the manometer. Therefore, the uncertainty of the differential pressure reading or the deviation could be calculated using calculus as the deviation in Q which is a single-variable function Q(y), the deviation in Q can be related to the deviation in y using calculus as follows:

$$\delta_Q = (\frac{dQ}{dy})\delta_y$$

Thus, taking the square and the average:

$${\delta_Q}^2 = (\frac{dQ}{dy})^2 {\delta_y}^2$$

And using the definition of σ, the standard error, we get:

$$\sigma_Q = \left|\frac{dQ}{dy}\right| \sigma_y$$

By applying these equations on the flow rate:

$$\frac{dQ}{dy} = const. * \frac{1}{\sqrt{y}}$$

Where $const. = \frac{c_d a_o \sqrt{2g}}{2}$ =0.001471044

$$\sigma_Q = const.* \frac{1}{\sqrt{y}} * \sigma_y$$

In order to calculate σ_y for N measurements rules of measuring the uncertainty in repeated measurements are applied as follows:

1. Sum all the measurements and divide by 5 to get the average or mean:

$$\mathrm{y_{avg}} = \frac{0.092 + 0.092 + 0.092 + 0.091 + 0.091}{5} = 0.0916 \text{ m}$$

2. Now, subtract this average from each of the 5 measurements to obtain 5 "deviations":
   $y_1 - \mathrm{y_{avg}}$ =0.0004
   $y_2 - \mathrm{y_{avg}}$ =0.0004
   $y_3 - \mathrm{y_{avg}}$ =0.0004
   $y_4 - \mathrm{y_{avg}}$ =-0.0006
   $y_5 - \mathrm{y_{avg}}$ =-0.0006

3. Square each of these 5 deviations and add them all up and divide the result by (N-1), and then take the square root to obtain the standard deviation,s,

$s = \sqrt{\frac{0.0004^2+0.0004^2+0.0004^2+0.0006^2+0.0006^2}{4}}$ =0.000547723

$\sigma_y = \frac{s}{\sqrt{N}}$ =0.000244949

$\therefore \sigma_Q = \frac{0.001471044}{\sqrt{0.0916}} * 0.000244949$ =1.19057E-06 m^3/s=0.004286 m^3/hr

$\therefore Q = Q_{avg} \pm \sigma_Q$

Where

$Q_{avg} = \frac{2.87909+2.879091+2.879091+2.863401+2.86340108}{5}$ =2.879091 m^3/hr

$\boxed{\therefore Q = 2.879091 \pm 0.004286\, m^3/Hr}$

So the uncertainty of the flowrate measurement is $\pm 0.4286\%$

<u>Uncertainty analysis of pressure</u>

The uncertainty of the pressure is easily calculated as it is not a function of any variable. The same steps of getting σ_y is applied.

$\therefore \Delta p = \Delta p_{avg} \pm \sigma_p$

$\therefore \Delta p = 0.399 \pm 0.001870829$ bar

So the uncertainty of the pressure measurement is $\pm 0.187\%$

Uncertainty analysis of power

The power is a function of two variables: the volt and the current.

In the case where P depends on two or more variables, the derivation above can be repeated with minor modification. For two variables, P(V, I), we have:

$$\delta P = \left(\frac{\partial P}{\partial V}\right) \delta V + \left(\frac{\partial P}{\partial I}\right) \delta I$$

Thus taking the square and the average, we get the **law of propagation of uncertainty**:

$$(\delta P)^2 = \left(\frac{\partial P}{\partial V}\right)^2 (\delta V)^2 + \left(\frac{\partial P}{\partial I}\right)^2 (\delta I)^2 + 2\left(\frac{\partial P}{\partial V}\right)\left(\frac{\partial P}{\partial I}\right) \delta V \delta I$$

If the measurements of x and y are uncorrelated, then , $\delta V \delta I$ =0 and using the definition of σ_P, we get:

$$\sigma_P = \sqrt{\left(\frac{\partial P}{\partial V}\right)^2 {\sigma_V}^2 + \left(\frac{\partial P}{\partial I}\right)^2 {\sigma_I}^2}$$

$$\frac{\partial P}{\partial V} = I, \qquad \frac{\partial P}{\partial I} = V$$

$$\therefore \sigma_P = \sqrt{I^2 {\sigma_V}^2 + V^2 {\sigma_I}^2}$$

$\therefore \sigma_P = \sqrt{0.432^2 * 0.04^2 + 11.34^2 * 0.002^2}$=0.0285

$\therefore Power = 4.899 \pm 0.0285$ watt

So the uncertainty of the power measurement is $\pm 2.85\%$

APPENDIX (C)

Arduino Code of the electrical circuit

```
int sensorPin = A1;    // select the input pin for the potentiometer
int ledPin = 13;      // select the pin for the LED
int sensorValue = 0;  // variable to store the value coming from the sensor
int pwm=9;
void setup() {
  // declare the ledPin as an OUTPUT:
  pinMode(ledPin, OUTPUT);


}


void loop() {
setPwmFrequency(pwm, 128);
  // read the value from the sensor:
  sensorValue = analogRead(sensorPin);
sensorValue=sensorValue/4.0;
  analogWrite(sensorValue, pwm);
    TCCR2B = TCCR2B & 0b11111000 | mode;
}
}
void setPwmFrequency(int pin, int divisor) {
  byte mode;
  if(pin == 5 || pin == 6 || pin == 9 || pin == 10) {
    switch(divisor) {
```

```
      case 1: mode = 0x01; break;
      case 8: mode = 0x02; break;
      case 64: mode = 0x03; break;
      case 256: mode = 0x04; break;
      case 1024: mode = 0x05; break;
      default: return;
    }
    if(pin == 5 || pin == 6) {
      TCCR0B = TCCR0B & 0b11111000 | mode;
    } else {
      TCCR1B = TCCR1B & 0b11111000 | mode;
    }
  } else if(pin == 3 || pin == 11) {
    switch(divisor) {
      case 1: mode = 0x01; break;
      case 8: mode = 0x02; break;
      case 32: mode = 0x03; break;
      case 64: mode = 0x04; break;
      case 128: mode = 0x05; break;
      case 256: mode = 0x06; break;
      case 1024: mode = 0x7; break;
      default: return;
    }
```

ملخص الرسالة

نظرا لزيادة التطبيقات في استخدام المضخات كتوربينات، فإنه من الضروري أن نحصل على معلومات عملية حقيقية تتعلق بهذا الموضوع. الهدف من هذه الرسالة هو تمثيل مضخة الطرد المركزيذات المحور الواحد كتوربينة عمليا و حسابيا و دراسة العلاقة بين منحنيات الأداء لهما.

الجهاز العملي مكون من مضخة طرد مركزي متصلة مباشرة بمحرك كهربي ذو تيار مباشر.أداء المضخة في الوضع المباشر يتم اختبارها عن طريق التحكم في فتحة صمام الخروج حتى نحصل على منحنى الأداء. المحرك كهربي ذو التيار المباشر يتم التحكم فيه عن طريق دائرة كهربية للتحكم في سرعة دورانه. علاوة على ذلك، فإن المضخة يتم احتبارها في الوضع المعاكس كتوربينة عن طريق التحكم في صمام الخروج السابق ذكره للحصول على منحنيات الأداء لنفس سرعات الدوران للوضع المباشر.من هنا، العلاقة بين منحنيات الأداء يتم تعريفها بوضوح.

السريان خلال المضخة يتم استنتاجه عن طريق ديناميكا الموائع الحسابية باستخدام برنامج "Fluent" و النتائج تم اختبارها و التحقق من صحتها باستخدام النتائج المعملية. النموذج العددي تم استخدامه للتنبؤ بمنحنيات الأداء للمضخة و التوربينة عند سرعات دوران مختلفة عن السرعات المعملية.

و في هذه الرسالة تم التنبؤ بالنتائج النظرية و التحقق م صحتها باستخدام النتائج المعملية و التي أثبتت أن النتائج تتوافق في سلوك التغيير و يوجد فرق بين القيم العملية و النظرية تحتاج إلى دراسات مستقبلية لتقليل الفروق.

و قد توصلت الدراسة إلى:

- إمكانية استخدام المضخات ذات الطرد المركزي في عملية توليد الطاقة الكهرومائية الصغيرة المقدار بطريقة سريعة و رخيصة.
- إمكانية استخدام ميكانيكا الموموائع العددية في التنبؤ بأداء المضخة الطاردة المركزية ذات الخط الواحد عندما تعمل كمضخة أو توربين حيث يمكن اختيار المضخة المناسبة في الموقع المناسب إذا علم كمية التصرف و الرافع الخاص بالطاقة المائية المتاحة.

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دراسة استخدام مضخات الطرد المركزي ذات الخط الواحد كتوربينات

مقدمة إلى قسم الهندسة الميكانيكية

كلية الهندسة – جامعة الإسكندرية

ضمن متطلبات الحصول على درجة

الماجستير في الهندسة الميكانيكية

آية صادق زكريا كساب

2014

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دراسة استخدام مضخات الطرد المركزي ذات الخط الواحد كتوربينات

مقدمة من المهندسة

آية صادق كساب

للحصول على درجة

ماجستير العلوم الهندسية

في

الهندسة الميكانيكية

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