The UNIVERSITY OF JORDAN

FACULTY OF ENGINEERING AND TECHNOLOGY

SCHOOL OF ENGINEERING

DEPT.OF CHEMICAL ENGINEERING

Chemical Engineering laboratory 1 (0915361)

Section no. (2)

Experiment Number (1)

Losses in piping system

Short report

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1. Abstract

For this experiment, pressure losses for a variety of components, such as valves, pipelines, and fittings, were calculated across a range of flow rates. In systems with flowing fluids, head loss was mostly caused by friction between the pipe wall and the fluid. Two separate circuits, one painted dark blue and the other light blue, with varied pipe system parts in each circuit, were used to measure major and minor pressure losses. Water for both circuits was supplied by the same hydraulic bench. The valves were both downstream of the pipe worked to reduce the chance that turbulence from the valves could change the readings from the pipe work.

Piezometer tubes were used to gauge pressure fluctuations in the pipework's component. A differential pressure gauge was used to determine the pressure difference across the valves.

Key outcome:

-The head loss will increase as the fluid flows more quickly (the higher the flow rate).

- For all fittings, the loss coefficient (K-values) values in the dark blue circuit range from 0.29 to 5.63, which is moderate to slightly high.

- K-values in light circuits typically range from 0.02 to 3.11, which is likewise regarded as moderate.

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1. Dark Blue Circuit

Dark Blue Circuit – Straight pipe :

Table (1-1) : data for dark blue circuit

pressure gauge (bar)	pressure gauge (m H2O)	flow rate (L/s)	Flow rate (m3/s)
0.1	1.02	0.2	0.0002
0.15	1.53	0.182	0.000182
0.2	2.04	0.163	0.000163
0.25	2.55	0.149	0.000149
0.3	3.06	0.128	0.000128
0.35	3.57	0.109	0.000109
0.4	4.08	0.079	0.000079
0.45	4.59	0.042	0.000042
0.5	5.10	0	0
Water Tempreture ©	14		0

Table (1-1): data for dark blue circuit (Continued)

Table (1-1) 1-2. Standard 90 Elbow 90	3-4. straight line (mm)	5-6. 90 miter bend
264	164	326
224	138	278
186	116	238
154	96	190
116	74	142
88	56	104
54	36	62
18	12	22
0	0	0

Standard-Bore Straight Pipe data results :

Table (1-2-A): Parameters of Standard-Bore Straight Pipe (Nominally 914.4 mm of 13.6 mm)

pressure gauge (bar)	pressure gauge (m H2O)	flow rate (L/s)	Flow rate (m3/s)
0.1	1.01971621	0.2	0.0002
0.15	1.529574315	0.182	0.000182
0.2	2.03943242	0.163	0.000163
0.25	2.549290525	0.149	0.000149
0.3	3.05914863	0.128	0.000128
0.35	3.569006735	0.109	0.000109
0.4	4.07886484	0.079	0.000079
0.45	4.588722945	0.042	0.000042

Table (1-2-A): Parameters of Standard-Bore Straight Pipe (Nominally 914.4 mm of 13.6 mm) (Continued)

ΔH (3-4)	Velocity (m/s)	Reynold Number	friction factor (Henry-Darcy)	Friction factor (Blasius)
164	1.38	16065.28	0.025	0.028
138	1.25	14619.40	0.026	0.029
116	1.12	13093.20	0.027	0.030
96	1.03	11968.63	0.027	0.030
74	0.88	10281.78	0.028	0.031
56	0.75	8755.58	0.029	0.033
36	0.54	6345.78	0.035	0.035
12	0.29	3373.71	0.042	0.041

Table (1-2-B): Parameters of Standard-Bore Straight Pipe (Nominally 914.4 mm of 13.6 mm)

K (Experimental)	log (flowrate)	log(ΔH (3-4))
1.70	-3.70	-0.79
1.72	-3.74	-0.86
1.81	-3.79	-0.94
1.79	-3.83	-1.02
1.87	-3.89	-1.13
1.95	-3.96	-1.25
2.39	-4.10	-1.44
2.81	-4.38	-1.92

Table (1-3-A): Parameters of Standard 90 Elbow (Nominally 914.4 mm of 13.6 mm)

pressure gauge (bar)	pressure gauge (m H2O)	flow rate (L/s)	Flow rate (m3/s)
0.1	1.02	0.2	0.0002
0.15	1.53	0.182	0.00018
0.2	2.04	0.163	0.00016
0.25	2.55	0.149	0.00015
0.3	3.06	0.128	0.00013
0.35	3.57	0.109	0.000109
0.4	4.08	0.079	0.000079
0.45	4.59	0.042	0.000042

Table (1-3-A): Parameters of Standard 90 Elbow (Nominally 914.4 mm of 13.6 mm)(Continued)

Flow rate (m3/s)	ΔH (3-4) m H2O	ΔH (1-2) m H2O	hm (m H2O)
0.0002	0.164	0.264	0.1
0.00018	0.14	0.224	0.086
0.00016	0.12	0.186	0.07
0.00015	0.096	0.154	0.058
0.00013	0.074	0.116	0.042
0.000109	0.056	0.088	0.032
0.000079	0.072	0.054	0.018
0.000042	0.046	0.018	0.028

Table (1-3-B): Parameters of Standard 90 Elbow (Nominally 914.4 mm of 13.6 mm)

Velocity (m/s)	Reynold Number	K (Experimental)	Log(flowrate)	Log(hm)
1.38	16065.28	1.03	-3.70	-1.00
1.25	14619.40	1.07	-3.74	-1.07
1.12	13093.20	1.09	-3.79	-1.15
1.03	11968.63	1.08	-3.83	-1.24
0.88	10281.78	1.06	-3.89	-1.38
0.75	8755.58	1.11	-3.96	-1.49
0.54	6345.78	1.19	-4.10	-1.74
0.29	3373.71	6.57	-4.38	-1.55
K (Experimental) average =	***1.78***			
K (Theoritical)	***1.50***			
%Error	***18.67***			

Table (1-4-A): Parameters for 90 Miter Bend

pressure gauge (bar)	pressure gauge (m H2O)	flow rate (L/s)	Flow rate (m3/s)
0.1	1.01971621	0.2	0.0002
0.15	1.529574315	0.182	0.000182
0.2	2.03943242	0.163	0.000163
0.25	2.549290525	0.149	0.000149
0.3	3.05914863	0.128	0.000128
0.35	3.569006735	0.109	0.000109
0.4	4.07886484	0.079	0.000079
0.45	4.588722945	0.042	0.000042

Table (1-4-A): Parameters for 90 Miter Bend (Continued)

ΔH (3-4) m H2O	ΔH (5-6) m H2O	hm (m H2O)	hm (mH2O) after taking the absolute
0.164	0.326	-0.162	0.162
0.138	0.278	-0.14	0.14
0.116	0.238	-0.122	0.122
0.096	0.19	-0.094	0.094
0.074	0.142	-0.068	0.068
0.056	0.104	-0.048	0.048
0.072	0.062	0.01	0.01
0.046	0.022	0.024	0.024

Table (1-4-B): Parameters of 90 Miter Bend

Velocity (m/s)	Reynold Number	K (Experimental)	Log(flowrate)	Log(hm)
1.38	16065.28	1.68	-3.70	-0.79
1.25	14619.40	1.75	-3.74	-0.85
1.12	13093.20	1.90	-3.79	-0.91
1.03	11968.63	1.75	-3.83	-1.03
0.88	10281.78	1.72	-3.89	-1.17
0.75	8755.58	1.67	-3.96	-1.32
0.54	6345.78	0.66	-4.10	-2.00
0.29	3373.71	5.63	-4.38	-1.62
K (Experimental) average =	***2.09***			
K (Theoritical)	***1.10***			
%Error	***90.36***			

Figures

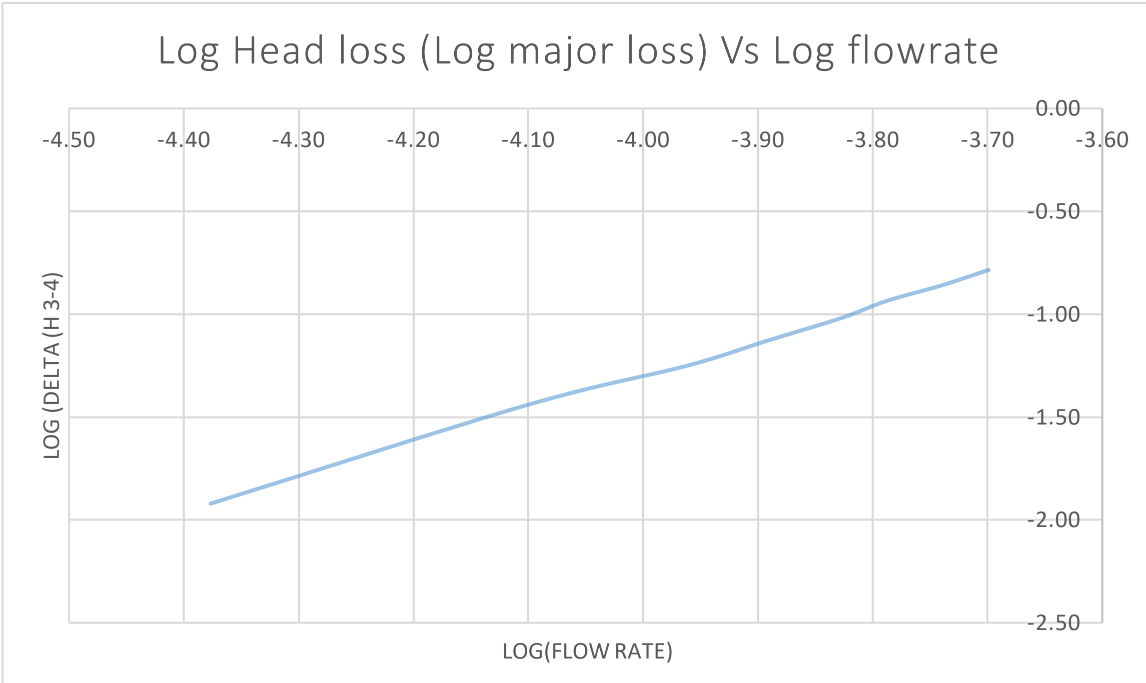

Figure (1): Log Head loss (Log major loss) Vs Log flowrate

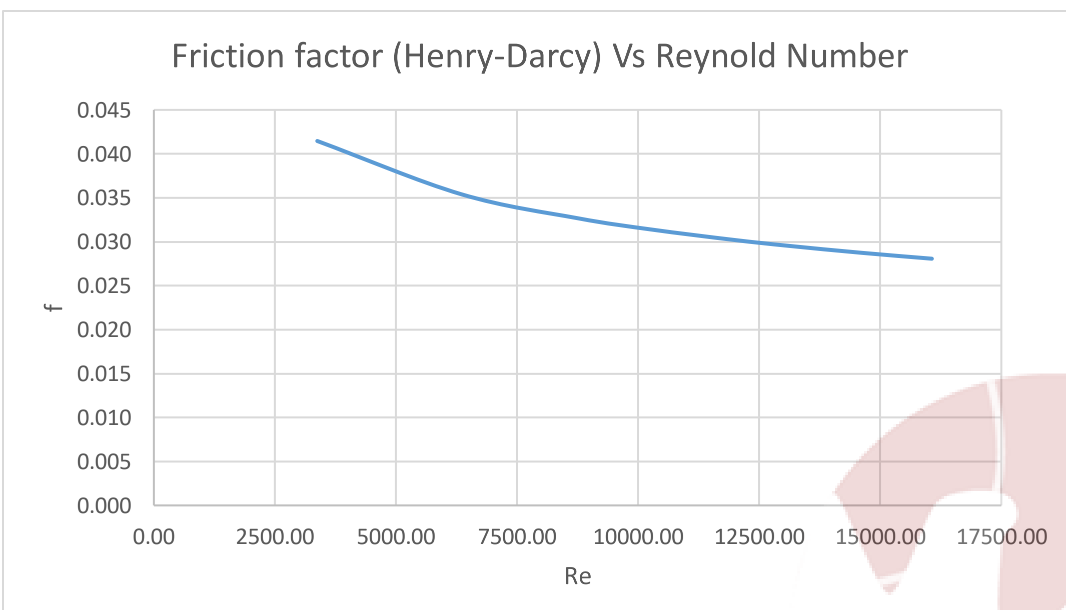

Figure (2): Friction factor (Henry-Darcy) Vs Reynold Number

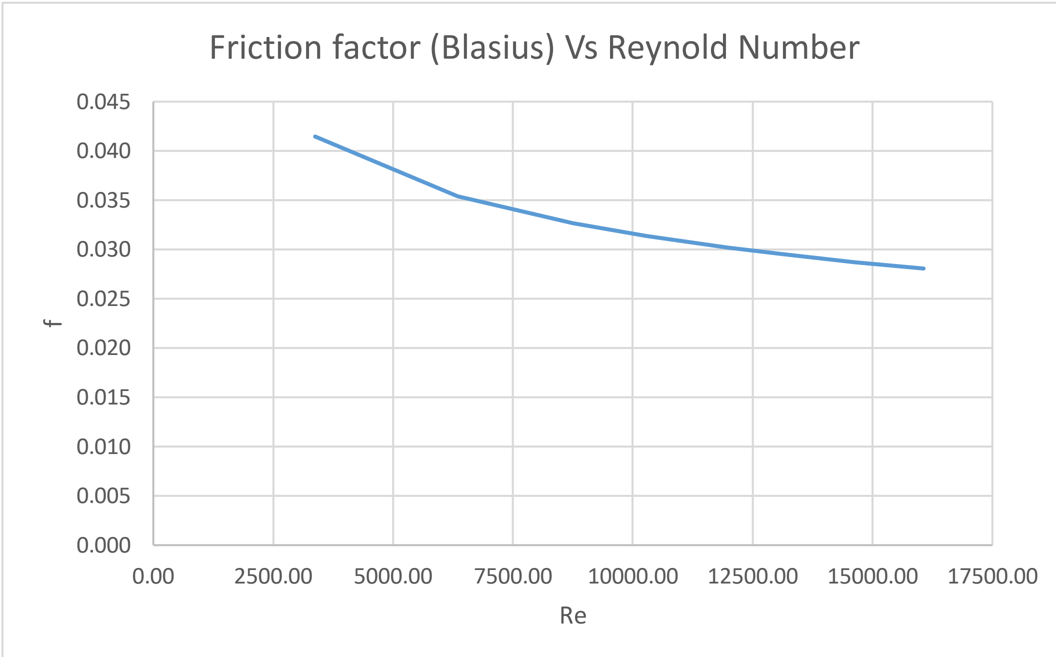

Figure (3): Friction factor (Blasius) Vs Reynold Number

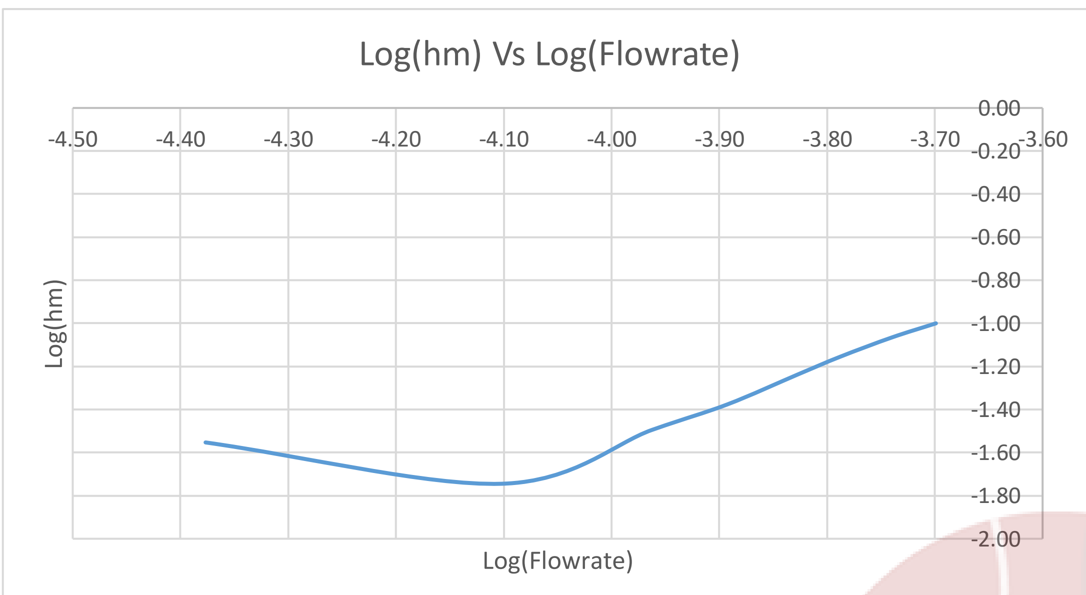

Figure (4): Log(hm) Vs Log(Flowrate)

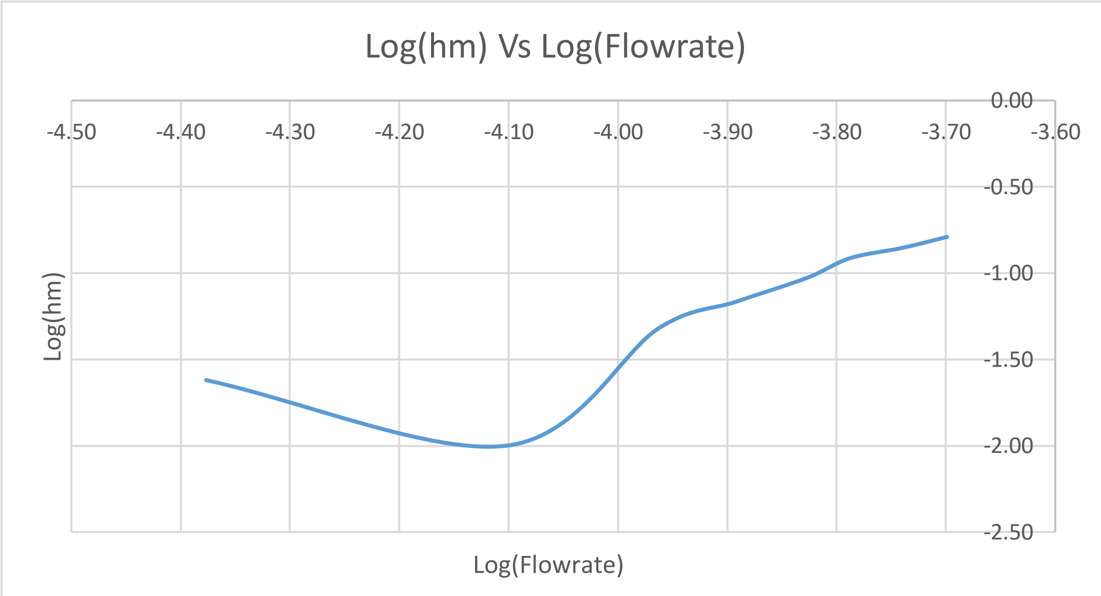

Figure (5): Log(hm) Vs Log(Flowrate)

2. Light Blue Circuit

Light Blue Circuit bends :

Table (2-1): Data for Light Blue Circuit (Globe Valve)

Pressure Gauge (bar)	Pressure Gauge (m water)	Flow rate (L/s)	Flow rate (m3/s)
0.1	1.019	0.22	0.00022
0.15	1.5285	0.197	0.000197
0.2	2.038	0.175	0.000175
0.25	2.5475	0.157	0.000157
0.3	3.057	0.13	0.00013
0.35	3.5665	0.111	0.000111
0.4	4.076	0.075	0.000075
0.45	4.5855	0.036	0.000036
0.5	5.095	0	0

Table (2-1): Data for Light Blue Circuit (Globe Valve)(Continued)

7-8: expansion	8-9: straight	9-10: contraction	11-12: medium rad.	13-14: large rad.	15-16: small rad.
-32	10	182	278	192	286
-31	8	150	238	162	242
-20	6	118	188	128	192
-16	6	94	154	104	156
-10	4	66	112	74	112
-10	4	50	84	58	84
-6	4	24	46	30	42
-2	2	6	16	12	10
0	0	0	0	0	0
Light Blue Circuit Piezometer Tube Heights (mm water)					
Water Temperature (°C) =14					

Table (2-1A): Parameters for small radius (50mm) smooth 90 bend (15-16)

ΔH (15-16) or (hL) in m water	Q (m3/s)	V (m/s)	Re	V^2/2g
0.29	0.00022	1.54	17790.95	0.12
0.24	0.000197	1.37	15930.99	0.10
0.19	0.000175	1.22	14151.89	0.08
0.16	0.000157	1.10	12696.27	0.06
0.11	0.00013	0.91	10512.83	0.04
0.08	0.000111	0.77	8976.34	0.03
0.04	0.000075	0.52	6065.10	0.01
0.01	0.000036	0.25	2911.25	0.00
0.00	0.00	0.00	0.00	0.00
		0.00	0.00	0.00

Table (2-1A): Parameters for small radius (50mm) smooth 90 bend (15-16)(Continued)

f (blasius)	hF	hB	Kb
0.02736	0.22	0.06	2.38
0.02813	0.18	0.06	2.51
0.02897	0.15	0.04	2.53
0.02977	0.12	0.03	2.55
0.03121	0.09	0.02	2.67
0.03246	0.07	0.02	2.75
0.03581	0.03	0.01	3.01
0.04302	0.01	0.00	3.11
#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!
#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!
AVG K= 2.69			

Experimental K-value= 2.687024

Theoretical K-value=2.64052

%Error= 1.76%

Table (2-2): Parameters for medium radius (100mm) smooth 90 bend (11-12)

∆H (11-12) or hL (in m water)	Q (m3/s)	V (m/s)	Re	V^2/2g	f (blasius)	hF	hB	Kb
0.28	0.00	1.52	17673.22	11.26	0.03	0.22	0.06	0.02
0.24	0.00	1.36	15825.56	9.03	0.03	0.18	0.06	0.03
0.19	0.00	1.21	14058.24	7.13	0.03	0.14	0.04	0.03
0.15	0.00	1.08	12612.25	5.74	0.03	0.12	0.03	0.03
0.11	0.00	0.90	10443.26	3.93	0.03	0.09	0.03	0.03
0.08	0.00	0.76	8916.94	2.87	0.03	0.07	0.02	0.03
0.05	0.00	0.52	6024.96	1.31	0.04	0.03	0.01	0.04
0.02	0.00	0.25	2891.98	0.30	0.04	0.01	0.01	0.05
0.00	0.00	0.00	0.00	0.00	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!
AVG K = 0.03								

Average experimental K-value= 0.0312

Theoretical K-value= 2.08

%Error= 98.5%

Table (3-1): Parameters for Large Radius (150mm) smooth 90 bend (13-14)

∆H (13-14) or hL in (m water)	Q (m3/s)	V (m/s)	Re	V^2/2g	f (blasius)	hF	hB	Kb
0.19	0.00	1.52	17673.22	0.12	0.03	0.22	0.02	1.64
0.16	0.00	1.36	15825.56	0.09	0.03	0.18	0.02	1.73
0.13	0.00	1.21	14058.24	0.07	0.03	0.14	0.02	1.73
0.10	0.00	1.08	12612.25	0.06	0.03	0.12	0.02	1.75
0.07	0.00	0.90	10443.26	0.04	0.03	0.09	0.01	1.81
0.06	0.00	0.76	8916.94	0.03	0.03	0.07	0.01	1.95
0.03	0.00	0.52	6024.96	0.01	0.04	0.03	0.00	2.21
0.01	0.00	0.25	2891.98	0.00	0.04	0.01	0.00	3.83
0.00	0.00	0.00	0.00	0.00	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!
AVG K =2.08								

Average experimental K-value= 2.08

Theoretical K-value=1.78

%Error=16.85%

Table (4-1): Sudden Expansion Calculated Parameters (13.6mm to 26.2mm)

ΔH (7-8) (mm water)	ΔH (7-8) (m water)	**Q (m3/s)**	**V1 (m/s)**	**V1^2/2g (m)**	**V2 (m/s)**	**V2^2/2g (m)**	**HL**	**K**
-32	-0.032	0.00022	1.467	0.110	0.408	0.008	0.069	0.631
-31	-0.031	0.000197	1.313	0.088	0.366	0.007	0.050	0.570
-20	-0.02	0.000175	1.167	0.069	0.325	0.005	0.044	0.634
-16	-0.016	0.000157	1.047	0.056	0.291	0.004	0.036	0.636
-10	-0.01	0.00013	0.867	0.038	0.241	0.003	0.025	0.661
-10	-0.01	0.000111	0.740	0.028	0.206	0.002	0.016	0.564
-6	-0.006	0.000075	0.500	0.013	0.139	0.001	0.006	0.452
-2	-0.002	0.000036	0.240	0.003	0.067	0.000	0.001	0.241
0	0	0	0.000	0.000	0.000	0.000	0.000	#DIV/0!
AVG K = 0.5487								

Average experimental K-value= 0.5487

Theoretical K-value=0.533706

%Error=2.81%

Table (5-1): Sudden Contraction Calculated Parameters (26.2 mm to 13.6 mm)

ΔH (9-10) in (mm water)	**ΔH (9-10) in (m water)**	**Q (m3/s)**	**V1**	**V1^2/2g**	**V2**	**V2^2/2g**	**HL**	**K**
182	0.182	0.00022	1.467	0.110	0.408	0.008	0.283	2.583
150	0.15	0.000197	1.313	0.088	0.366	0.007	0.231	2.629
118	0.118	0.000175	1.167	0.069	0.325	0.005	0.182	2.623
94	0.094	0.000157	1.047	0.056	0.291	0.004	0.146	2.606
66	0.066	0.00013	0.867	0.038	0.241	0.003	0.101	2.647
50	0.05	0.000111	0.740	0.028	0.206	0.002	0.076	2.714
24	0.024	0.000075	0.500	0.013	0.139	0.001	0.036	2.806
6	0.006	0.000036	0.240	0.003	0.067	0.000	0.009	2.966
0	0	0	0.000	0.000	0.000	0.000	0.000	#DIV/0!
AVG K = 2.696698								

Average Experimental K-value= 2.7

Theoretical K-value=0.533

%Error=406.5%

Figures

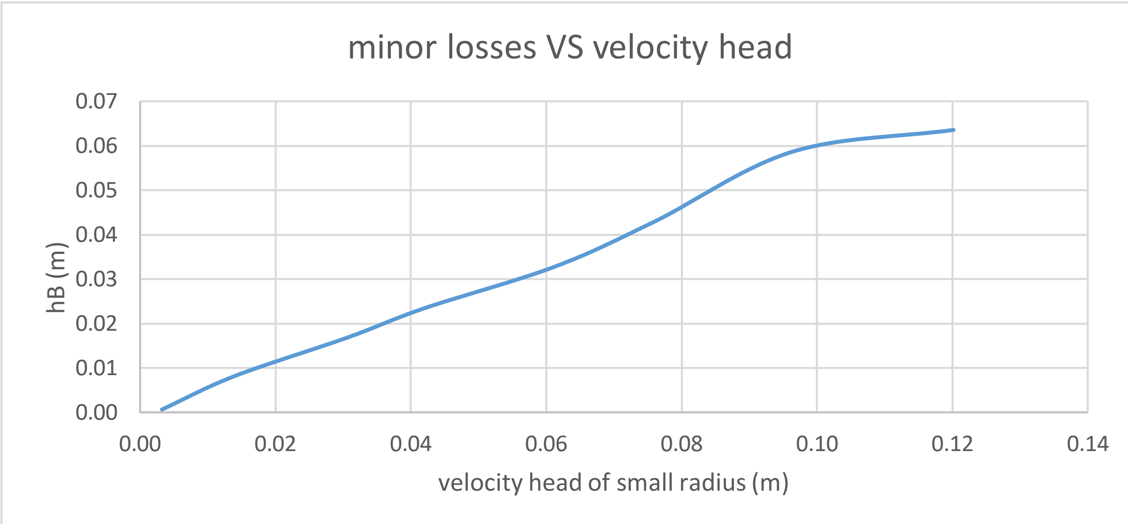

Figure (6): minor losses VS velocity head

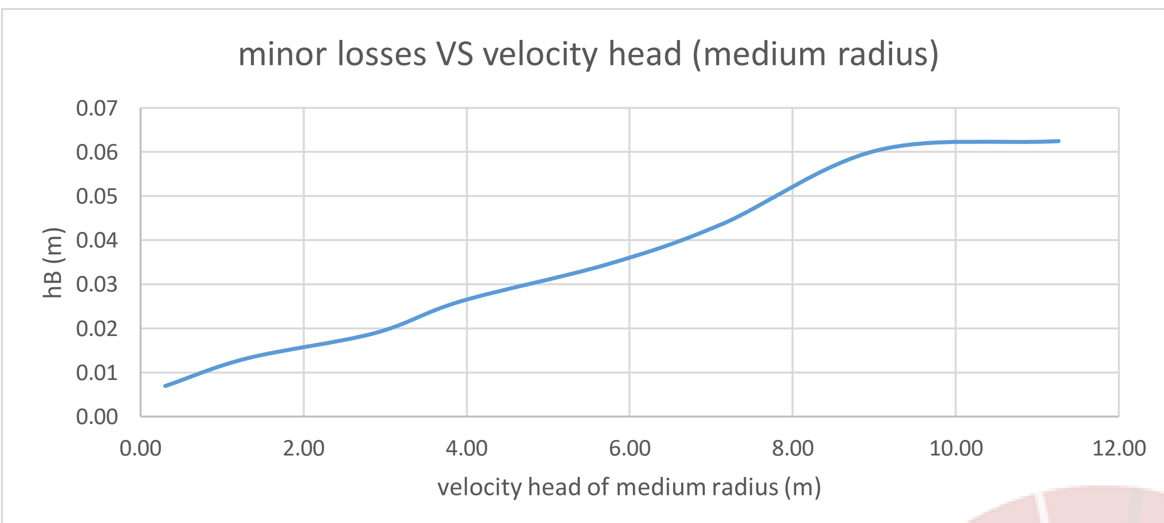

Figure (7): minor losses VS velocity head (medium radius)

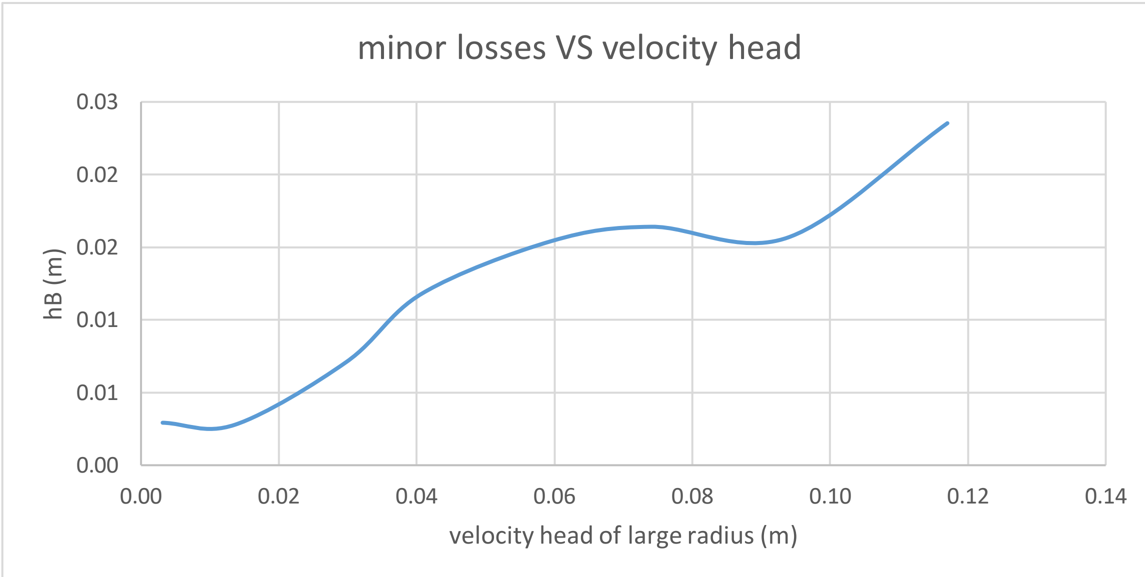

Figure (8): minor losses VS velocity head

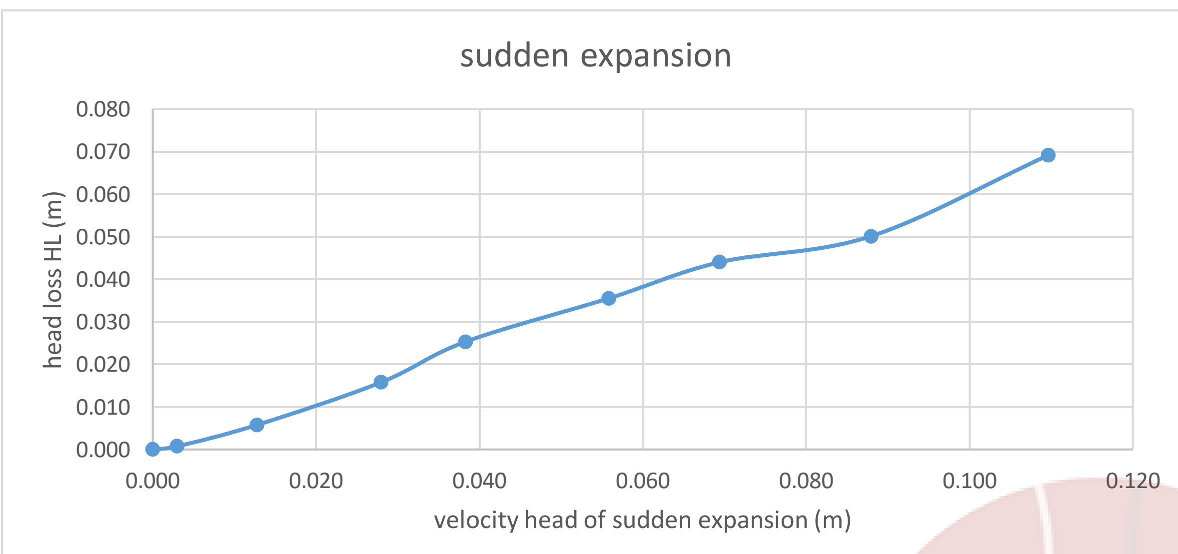

Figure (9): sudden expansion

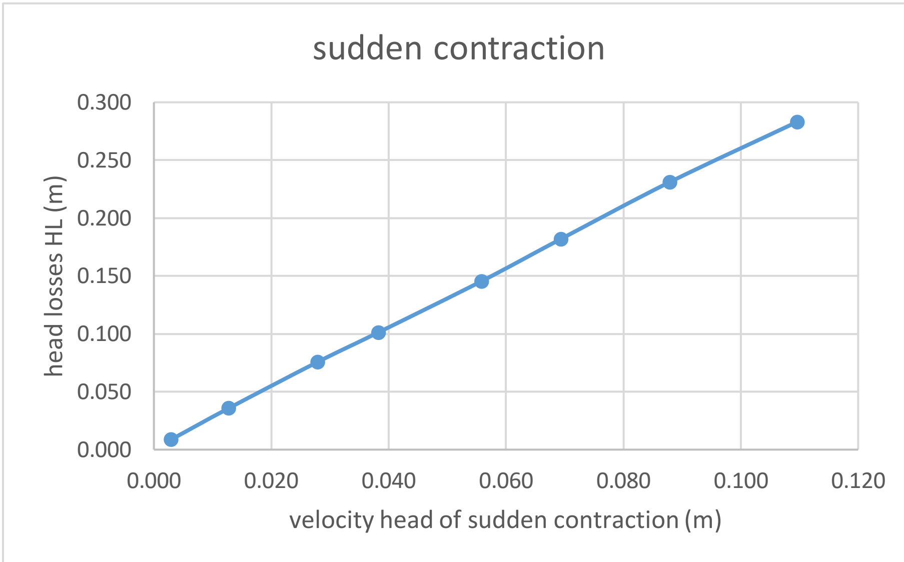

Figure (10): sudden contraction

3. Discussion

Friction between the pipe wall and the flowing fluid is a major contributor to head loss in systems with flowing fluids (Major Loss). The fluid is going through the pipe at a faster speed as a result of the friction.

The head loss will be greater the faster the fluid is flowing (the higher the flow rate). When the valves are almost fully opened and the pressure was at its lowest points (0.1 bar), conjugate manometers displayed the most erroneous readings. The disparities between the heads were therefore at their greatest importance.

Both Henry-Darcy and Blasius equations produced a consistent curve with no deviations when graphing both friction factors derived from Blasius and Henry-equations Darcy's using experimental data for standard-bore straight pipes in figures (3) and (2).

Continuing with the results, figure (4) for the standard 90° elbow illustrates how the head loss is rather substantial as the flow rate increases in accordance with the log scale, and figure (5) for the 90° miter bend.

The flow in the dark blue circuit was controlled by a gate valve, and it includes a straight pipe, an elbow and a 90° miter bend. The values of loss coefficient (K-values) are moderate to slightly high, ranging from 0.29 to 5.63 generally for all fittings. Logarithmic graphs of major head loss versus flow rate exhibited increasing behavior overall.

However, the flow in the light blue circuit, which is controlled by a globe valve, includes bends of small radius (50mm), medium radius (100mm), and large radius (150mm), as well as sudden expansion and contraction pipes. K-values vary from 0.02 to 3.11 generally, which are also considered moderate. Graphs of head losses versus velocity heads exhibit increasing behavior for all fittings, indicating that there’s a direct increasing relation between these two parameters.

It can be observed for sudden expansion that the head losses had negative values, which is understandable because there’s a suction force with varied cross-sectional area of the pipe.

For both circuits, as pressure is increased to 0.5 bar, the head losses approach zero as does the water flow rate. Additionally, readings may deviate due to personal errors and instrumental errors. Sources of personal errors could be from inaccuracy while recording readings from the piezometers.

4. Conclusions and recommendations

major head loss as a result of shear stress between the water and the pipe’s inner wall.

minor head loss is caused by the loss coefficient of the various pipe fittings (valve ,elbows ,sudden expansion/contraction).

For low losses in the piping system, the gate valve is superior than the globe valve.

Lower friction loss in the 90-degree elbow than the 90-meter.

The coefficient (k) rises as the flow falls.

A pipe's radius of curvature has a stronger relationship with the load loss coefficient.

When capturing the data, the eye level must be perpendicular to the reading to prevent parallax error and produce an accurate result.

Reading can vary due to leakage, so the water flow must always be watched .

Sources of Errors:

1. Personal Error:

Uncertainty in data reading, errors in calculations.

2. Experimental error:

Water leakage and reading variation.

5. References

1 ‘Chemical Engineering Laboratory (1) Manual’ 1st edition, Losses in piping system, University of Jordan.

2 Gean Koplis ،Transport Processes Momentum ،Heat and Mass ،Augn and Bacon 1983.

3 Mott, R.L. and Untener, J.A. (2016). *Applied fluid mechanics*. Boston: Pearson

4 Noel de Nevers (2005). *Fluid Mechanics for Chemical Engineers*. McGraw-Hill Companies.

5 www.piping-designer.com. (n.d.). *Loss Coefficient*. [online] Available at: https://www.piping-designer.com/index.php/properties/dimensionless-numbers/2484-loss-coefficient.

6. Appendices

1. Sample of Calculations:

Dark Blue Circuit:

- Taking the first raw in each table :
  Pressure gauge = 0.1 Bar.

Table (1-1) : data for dark blue circuit:

- Pressure gauge (bar) ➔ Pressure gauge (m H2O)
  1.00 bar = 10.19 m H2O
  0.1 bar = ?? H2O

$$0.1\ \text{bar}* \frac{10.19\ \mathrm{m\ H_2O}}{1.00\ \mathrm{bar}} = 1.02\ \mathrm{m\ H_2O}.$$

- Volumetric Flowrate (L/s) ➔ Volumetric Flowrate ($\mathrm{m^3/s}$)
  1.0 L/s = 0.001 $\mathrm{m^3/s}$
  ✓ /s = ?? $\mathrm{m^3/s}$

$$0.2\ \text{L/s}* \frac{0.001\ \mathrm{m^3/s}}{1.00\ \mathrm{L/s}} = 0.0002\ m^3/s$$

- Velocity

$$V = \frac{\text{Volumetric Flowrate}}{\text{Cross Sectional Area}} = \frac{Q}{A} = \frac{Q}{(\frac{\pi}{4}d^2)} = \frac{(0.0002\frac{m^3}{s})}{(\frac{\pi}{4}(13.6\times10^{-3})^2 m^2} = 1.38\frac{m}{s}.$$

- Reynold Number

$$Re = \frac{\rho DV}{\mu} = \frac{1000\frac{Kg}{m^3}*13.6*10^{-3}m*1.38\frac{m}{s}}{1.166*10^{-3}Pa.s} = 16065.28.$$

- Friction Factor (Henry-Darcy)

$$F = \frac{h_f 2dg}{LV^2} = \frac{0.164m\times2\times13.6\times10^{-3}m\times9.81m/s^2}{0.9144\ m\times1.38\frac{m}{s}} = 0.025.$$

- Friction factor (Blasius)

$$F = \frac{0.316}{Re^{0.25}} = \frac{0.316}{16065.28^{0.25}} = 0.028.$$

- Minor head loss Standard 90° Elbow

$$h_m = \Delta h_{1-2} - h_{f_{((3-4)+\Delta k(0 \text{ in bent pipe})}}$$

= 0.264 -0.164 = 0.10 m H2O.

- Loss Coefficient in Standard-Bore Straight Pipe

$$K_{Experimental} = \frac{\Delta h_{3-4}}{\frac{v^2}{2g}} = \frac{0.164 \text{ m H2O}}{\frac{1.38^2}{2*8.81}} = 1.70$$

Light-blue circuit (globe valve)

- Taking the first row of Table (2-1A): (small radius 50mm 90 bend)
  Small radius = 50mm=0.05m
  Given from technical details: r/d=3.7 -> d=0.05/3.7= 0.01351m.

➢ **Volumetric flow rate:**

$Q = 0.22\frac{L}{s} \times \frac{0.001\ m^3}{1\ L}$= 0.00022 m^3

➢ **Velocity**:

$V = \frac{Q}{A} = \frac{Q}{\pi/_4 \times d^2} = \frac{0.00022}{\pi/_4 \times 0.01351^2} = 1.54\frac{m}{s}$.

➢ **Reynold's Number:**

$Re = \frac{\rho.V.d}{\mu} = \frac{\left(1000\frac{kg}{s}\right).\left(1.54\frac{m}{s}\right).(0.01351\ m)}{(1.166 \times 10^{-3} Pa.s)} = 17790.95$

➢ **Velocity head:**

$\frac{V^2}{2g} = \frac{1.54^2}{2(9.81)} = 0.12$ m

➢ **Friction factor from the Blasius equation**:

$f = \frac{0.316}{Re^{0.25}} = \frac{0.316}{17790.95^{0.25}} = 0.0273.$

➢ **Major head loss due to pipe friction from the Darcy equation**:

$h_F = \frac{f.L.V^2}{2dg} = \frac{(0.0273)(0.914)(1.54)^2}{2(0.01351)(9.81)} = 0.22\ m$

➢ **Minor head losses:**

$h_B = h_L - h_F = 0.29 - 0.22 = 0.06m.$

➢ **K-value:**

$K_b = \frac{h_L}{V^2/_{2g}} = \frac{0.29}{0.12} = 2.4$

- Taking the first row of Table (4-1):
  Sudden expansion (The same steps are applied for sudden contraction)

➢ $\Delta H\ (7-8) = -0.032\ m.$

- **Diameters:**

  D1= 13.6mm =0.0136m

  D2=26.2mm=0.0262 m

- **Areas:**

$$A1 = \frac{\pi}{4} D1^2 = \frac{\pi}{4}(0.0136^2) = 0.00015m^2$$

$$A2 = \frac{\pi}{4} D2^2 = \frac{\pi}{4}(0.0262^2) = 0.0005389\ m^2$$

- **Velocity:**

$$V1 = \frac{Q}{A1} = \frac{0.00022}{0.00015} = 1.461\ m.$$

$$V2 = \frac{Q}{A2} = \frac{0.00022}{0.0005389} = 0.408m$$

- **Velocity heads:**

$$\frac{V1^2}{2g} = \frac{1.461^2}{2(9.81)} = 0.11m$$

$$\frac{V2^2}{2g} = \frac{0.408^2}{2(9.81)} = 0.008\ m$$

- **Head loss:**

$$H_L = \Delta H(7-8) + \frac{V1^2 - V2^2}{2g} = \frac{1.461^2 - 0.408^2}{2(9.81)} = 0.069m$$

- **K-value:**

$$K = \frac{H_L}{V1^2/2g} = \frac{0.069}{1.461^2/2(9.81)} = 0.631.$$

2. *Data Sheet:*

Losses in Pipes Experiment Data Sheet

Dark Blue Circuit and Gate Valve: (major losses).

Pressure Gauge (bar)	Pressure Gauge (m water)	Flow rate (L/s) (volumetric) or kg/s (gravimetric)	Volume Flow rate ($m^3.s^{-1}$)	Dark Blue circuit piezometer tube heights (mm water)		
				1-2	3-4	5-6
0.1		0.2	[illegible]	264	164	326
0.15		0.182		224	138	278
0.2		0.163		186	116	238
0.25		0.149		154	96	190
0.3		0.128		116	74	142
0.35		0.109		88	56	104
0.4		0.079		54	36	62
0.45		0.042		18	12	22
0.5		0		0	0	0
Water Temperature: 14°C						

Light Blue Circuit and Globe Valve: (minor losses)

Pressure Gauge (bar)	Pressure Gauge (m water)	Flow rate (L/s) (volumetric) or kg/s (gravimetric)	Volume Flow rate ($m^3.s^{-1}$)	Light Blue circuit piezometer tube heights (mm water)					
				Exp. 7-8	8-9	Contr. 9-10	11-12	13-14	15-16
0.1		0.22	0.00022	582 | 614		604 | 422	278	192	286
0.15		0.197	0.000197	588 | 614		606 | 456	238	162	242
0.2		0.175	0.000175	594 | 614		608 | 490	188	128	192
0.25		0.157	0.000157	598 | 614		608 | 514	154	104	156
0.3		0.130	0.00013	604 | 614		610 | 544	112	74	112
0.35		0.111	0.000111	606 | 616		612 | 562	84	58	84
0.4		0.075	0.000075	612 | 618		614 | 590	46	30	42
0.45		0.036	0.000036	618 | 620		612 | 612	16	12	10
0.5		0	0	0	0	0	0	0	0
Water Temperature: 14°C									

Instructor signature and Date: م. هبة [illegible] 14/3/2023

The University of Jordan

School of Engineering

Department of Chemical Engineering

Chemical Engineering Laboratory I (0915361)

Section Number: (2)

Experiment Number: (2)

Experiment Title: ‘Positive Displacement Pumps Characteristics’.

Report Type: Short Report

Instructor: Prof. Khaled Rawajfeh

Date of Performance: 16/5/2022.

Date of Submission: 23/5/2023.

Abstract:

This experiment focused on the characteristics and performance analysis of positive displacement pumps, specifically piston pumps and vane pumps. The aim of this experiment was to compare and assess the performance characteristics and efficiency of these two varieties of positive displacement pumps under identical condition. The experiment was divided into two parts; the first looked at the impact of delivery pressure at constant speed, while the second dealt with the impact of speed at constant delivery pressure. Positive displacement pump module and oil as the working fluid were used to practice the experiment. The ratio of the force applied to the fluid by the pump to the force applied to drive the pump is known as the pump efficiency.

Based on the results: As the delivery pressure increases, vane pumps have higher volumetric efficiency and shaft power than piston pumps while the speed is constant. Vane pumps outperform piston pumps in terms of volumetric efficiency and shaft power at constant pressure as speed is raised. Consequently, the vane pump is more effective.

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Part one: The Effect of Delivery Pressure at Constant Speed.

A) piston pump: V_s=7.15 cc/rev.

Table 1: Piston Pump raw data at constant speed=500rev/min and varying pressure.

Speed (rpm)	*Torque (N.m)*	*Power (watt)*	*P2 (bar)*	*P2 (bar)*	*Oil temperature (˚C)*	*Flow rate (L/min)*
500	*0.34*	*18*	*-0.03*	*2.2*	*19.5*	*2*
499	*0.39*	*20.5*	*-0.03*	*2.9*	*19.5*	*1.9*
498	*0.47*	*24*	*-0.03*	*4*	*19.5*	*1.9*
498	*0.54*	*28*	*-0.03*	*4.95*	*19.45*	*1.9*
500	*0.59*	*31*	*-0.03*	*5.9*	*19.5*	*1.9*
499	*0.67*	*35*	*-0.03*	*7*	*19.5*	*1.9*
498	*0.73*	*37*	*-0.03*	*7.9*	*19.5*	*1.9*
500	*0.8*	*43*	*-0.03*	*9*	*19.6*	*1.9*

Table 2: Parameters for piston pump at constant speed =500 rev/min

Δ P (Bar)	*ΔP (Pa)*	*Volumetric flow rate Qv (m3/s)*	*Hydraulic power (w)*	*Expected flow rate Qe (m3/s)*	*η (overall) %*	*Volumetric efficiency %*
2.23	*225954.75*	*0.000033*	*7.53*	*0.000060*	*41.84*	*55.94*
2.93	*296882.25*	*0.000032*	*9.40*	*0.000059*	*45.86*	*53.25*
4.03	*408339.75*	*0.000032*	*12.93*	*0.000059*	*53.88*	*53.36*
4.98	*504598.50*	*0.000032*	*15.98*	*0.000059*	*57.07*	*53.36*
5.93	*600857.25*	*0.000032*	*19.03*	*0.000060*	*61.38*	*53.15*
7.03	*712314.75*	*0.000032*	*22.56*	*0.000059*	*64.45*	*53.25*
7.93	*803507.25*	*0.000032*	*25.44*	*0.000059*	*68.77*	*53.36*
9.03	*914964.75*	*0.000032*	*28.97*	*0.000060*	*67.38*	*53.15*

B) vane pump= 6.60cc/rev.

Table 3: Vane Pump raw data at constant speed=500rev/min and varying pressure.

Speed (rpm)	Torque (N.m)	Power (watt)	P1 (bar)	P2 (bar)	Oil temperature (˚C)	Flow rate (L/min)
498	0.79	40.5	-0.05	2.1	20.3	3.8
499	0.91	47	-0.05	3	20.4	3.8
498	1.01	52	-0.05	4	20.5	3.8
500	1.2	63	-0.05	5	20.5	3.7
500	1.28	68	-0.05	6	20.5	3.7
498	1.39	74	-0.05	7	20.5	3.7
500	1.57	82	-0.05	8	20.6	3.7
500	1.66	87	-0.05	9	20.6	3.7

Table 4: Parameters for vane pump at constant speed =500 rev/min.

Δ P (Bar)	*ΔP (Pa)*	*Volumetric flow rate Qv (m3/s)*	*Hydraulic power (Kw)*	*Expected flow rate Qe (m3/s)*	*η (overall) %*	*Volumetric efficiency %*
2.15	*217848.75*	*0.000063*	*13.80*	*0.000055*	*34.07*	*115.61*
3.05	*309041.25*	*0.000063*	*19.57*	*0.000055*	*41.64*	*115.38*
4.05	*410366.25*	*0.000063*	*25.99*	*0.000055*	*49.98*	*115.61*
5.05	*511691.25*	*0.000062*	*31.55*	*0.000055*	*50.09*	*112.12*
6.05	*613016.25*	*0.000062*	*37.80*	*0.000055*	*55.59*	*112.12*
7.05	*714341.25*	*0.000062*	*44.05*	*0.000055*	*59.53*	*112.57*
8.05	*815666.25*	*0.000062*	*50.30*	*0.000055*	*61.34*	*112.12*
9.05	*916991.25*	*0.000062*	*56.55*	*0.000055*	*65.00*	*112.12*

Part Two: The Effect of Speed at Constant Delivery Pressure.

A) piston pump: Vs=7.15 cc/rev.

Table 5: Piston Pump raw data at constant pressure=5 bar and varying speed.

Speed (rpm)	*Torque (N.m)*	*Power (watt)*	*P1 (bar)*	*P2 (bar)*	*Oil temperature (˚C)*	*Flow rate (L/min)*
98	*0.51*	*5*	*-0.015*	*5.4*	*18.9*	*0.2*
200	*0.5*	*10*	*-0.02*	*5.2*	*18.8*	*0.7*
300	*49*	*14*	*-0.02*	*5.1*	*18.9*	*1.1*
401	*0.56*	*23.5*	*-0.03*	*5*	*19*	*1.6*
501	*0.57*	*30*	*-0.03*	*4.9*	*19.3*	*2*
600	*0.6*	*37.5*	*-0.035*	*5*	*19.3*	*2.4*
701	*0.6*	*43.5*	*-0.04*	*5*	*19.2*	*2.7*
800	*0.6*	*50.5*	*-0.04*	*5*	*19.3*	*3.1*
899	*0.6*	*58*	*-0.05*	*5*	*19.4*	*3.5*
1002	*0.625*	*65*	*-0.05*	*5*	*19.5*	*3.9*

Table 6: Parameters for piston pump at constant pressure=5 bar.

Δ P (Bar)	ΔP (Pa)	Volumetric flow rate Qv (m3/s)	Hydraulic power (w)	Expected flow rate Qe (m3/s)	η (overall) %	Volumetric efficiency %
5.42	548674.88	0.000003	1.83	0.000012	36.58	28.54
5.22	528916.50	0.000012	6.17	0.000024	61.71	48.95
5.12	518784.00	0.000018	9.51	0.000036	67.94	51.28
5.03	509664.75	0.000027	13.59	0.000048	57.83	55.80
4.93	499532.25	0.000033	16.65	0.000060	55.50	55.83
5.04	*510171.38*	*0.000040*	*20.41*	*0.000072*	*54.42*	*55.94*
5.04	*510678.00*	*0.000045*	*22.98*	*0.000084*	*52.83*	*53.87*
5.04	*510678.00*	*0.000052*	*26.39*	*0.000095*	*52.25*	*54.20*
5.05	*511691.25*	*0.000058*	*29.85*	*0.000107*	*51.46*	*54.45*
5.05	*511691.25*	*0.000065*	*33.26*	*0.000119*	*51.17*	*54.44*

B) Vane Pump=6.60 cc/rev.

Table 7: Vane Pump raw data at constant pressure=5 bar and varying speed.

Speed (rpm)	*Torque (N.m)*	*Power (watt)*	*P1 (bar)*	*P2 (bar)*	*Oil temperature (˚C)*	*Flow rate (L/min)*
101	*0.98*	*10*	*-0.02*	*5*	*19.6*	*0.7*
199	*1*	*19*	*-0.02*	*4.8*	*19.6*	*1.4*
302	*1.02*	*32*	*-0.03*	*5.1*	*19.7*	*2.3*
400	*1.12*	*47*	*-0.04*	*5.2*	*19.8*	*3*
502	*1.22*	*66*	*-0.05*	*5.2*	*20*	*3.8*
599	*1.26*	*79*	*-0.06*	*5.2*	*20.1*	*4.6*
702	*1.29*	*95.5*	*-0.07*	*5.2*	*20.2*	*5.3*
801	*1.32*	*110*	*-0.07*	*5.2*	*20.2*	*6*
899	*1.37*	*130*	*-0.08*	*5.2*	*20.3*	*6.7*
1002	*1.39*	*146*	*-0.09*	*5.2*	*20.3*	*7.3*

Table 8: Parameters for vane pump at constant pressure=5 bar.

Δ P (Bar)	ΔP (Pa)	Volumetric flow rate Qv (m3/s)	Hydraulic power (W)	Expected flow rate Qe (m3/s)	η (overall) %	Volumetric efficiency %
5.02	508651.50	0.000012	5.93	0.000011	59.34	105.01
4.82	488386.50	0.000023	11.40	0.000022	59.98	106.59
5.13	519797.25	0.000038	19.93	0.000033	62.27	115.39
5.24	530943.00	0.000050	26.55	0.000044	56.48	113.64
5.25	531956.25	0.000063	33.69	0.000055	51.05	114.69
5.26	*532969.50*	*0.000077*	*40.86*	*0.000066*	*51.72*	*116.36*
5.27	*533982.75*	*0.000088*	*47.17*	*0.000077*	*49.39*	*114.39*
5.27	*533982.75*	*0.000100*	*53.40*	*0.000088*	*48.54*	*113.49*
5.28	*534996.00*	*0.000112*	*59.74*	*0.000099*	*45.95*	*112.92*
5.29	*536009.25*	*0.000122*	*65.21*	*0.000110*	*44.67*	*110.39*

Result Analysis: Figure of Data:

Part One: Figure at Constant Speed =500 rev/min.

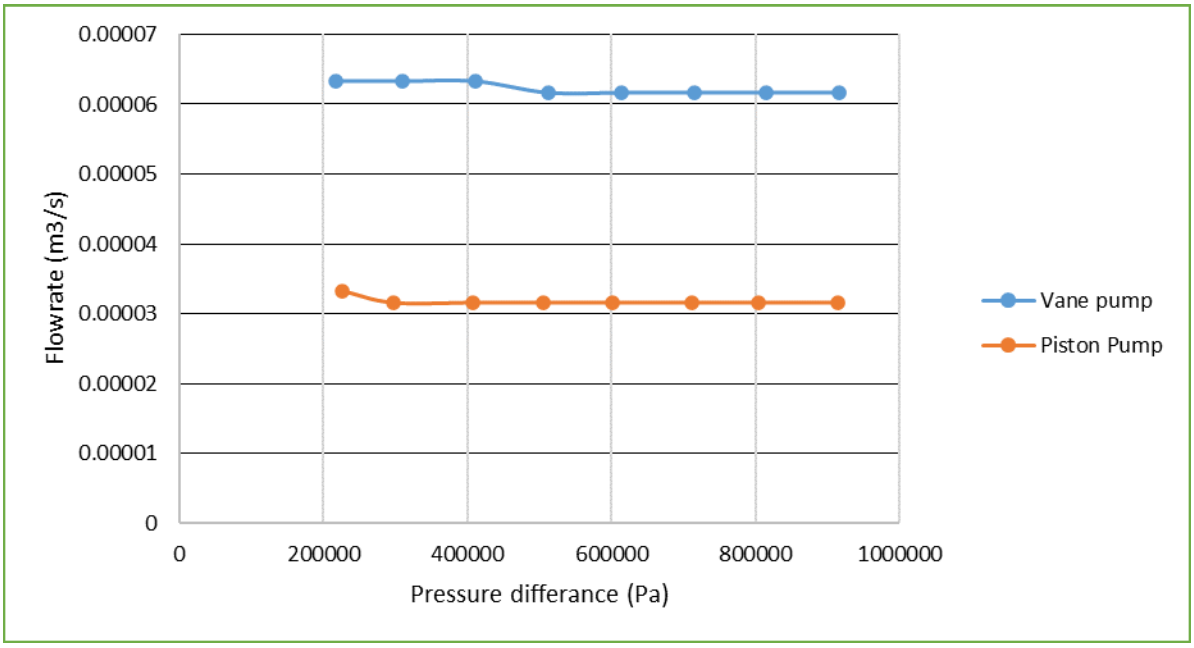

Figure (1): Flow rate vs. Pressure difference for piston and vane pumps.

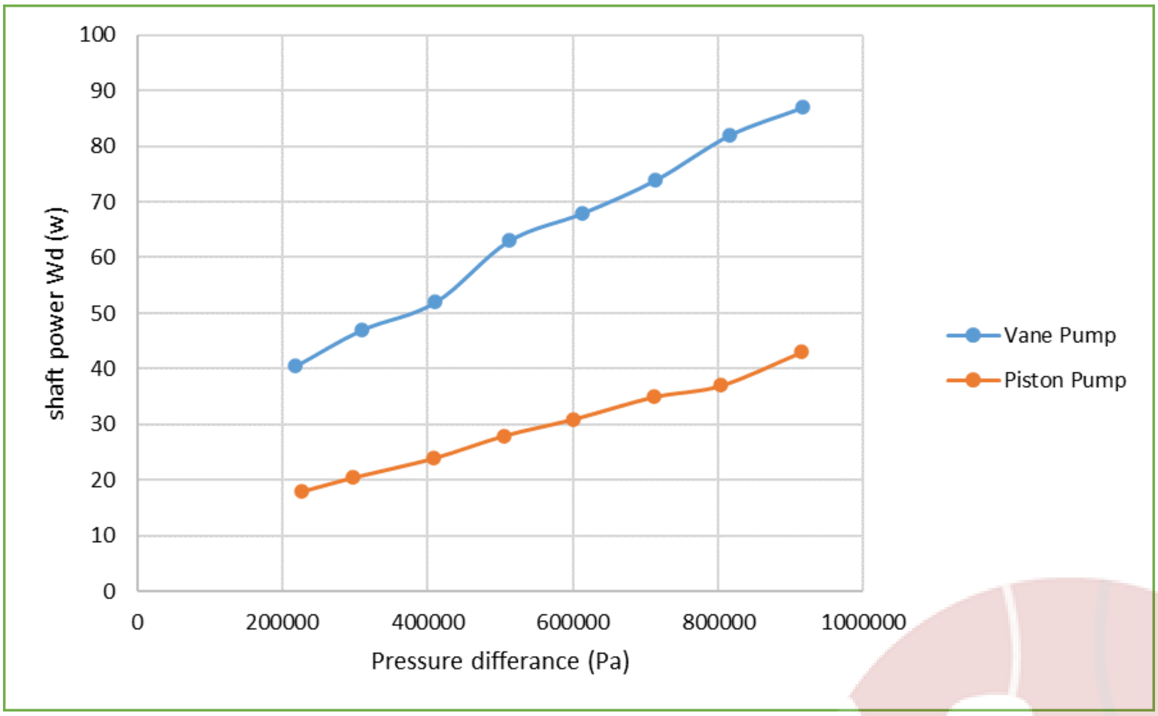

Figure (2): Shaft power vs. Pressure difference for piston and vane pumps.

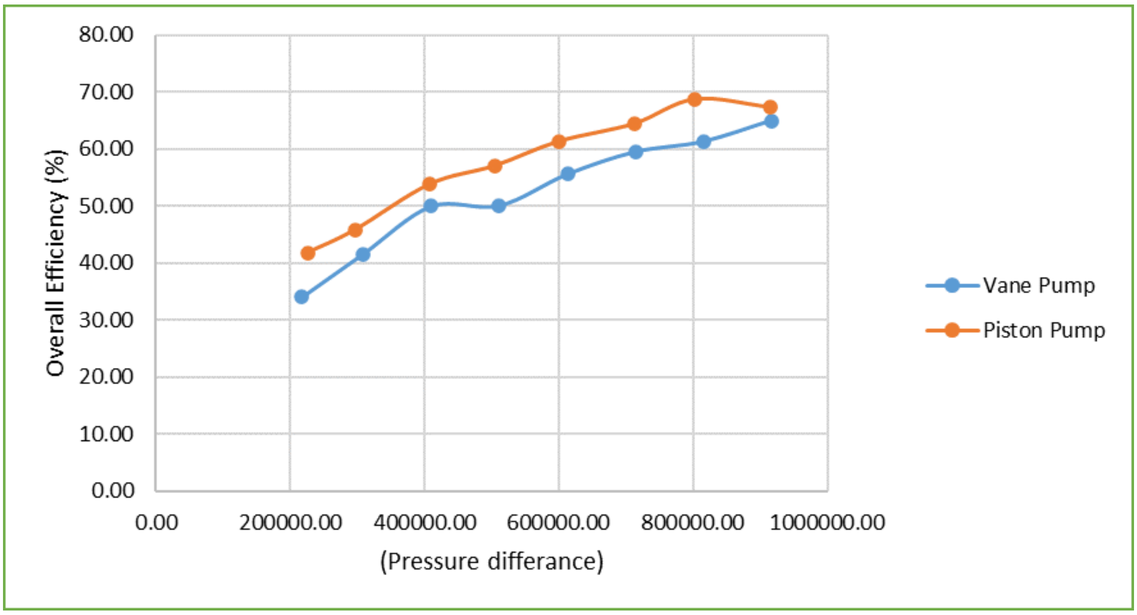

Figure (3): Overall efficiency vs. Pressure difference for piston and vane pumps.

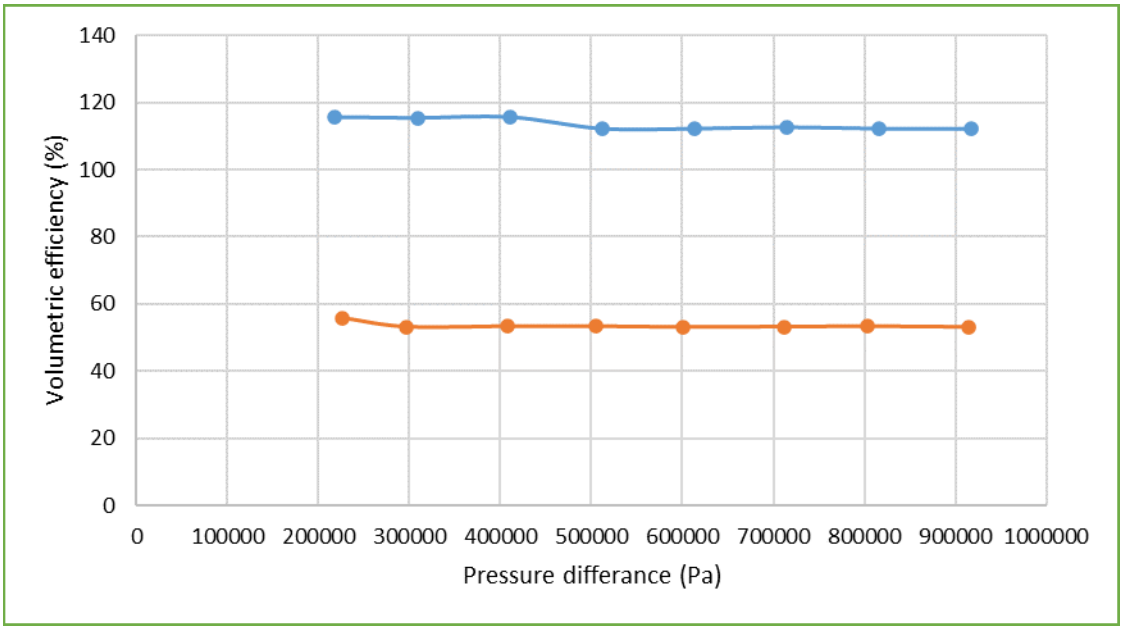

Figure (4): Volumetric efficiency vs. Pressure difference for piston and vane pumps.

Part Two: Figure at Constant Delivery Pressure =5 bar.

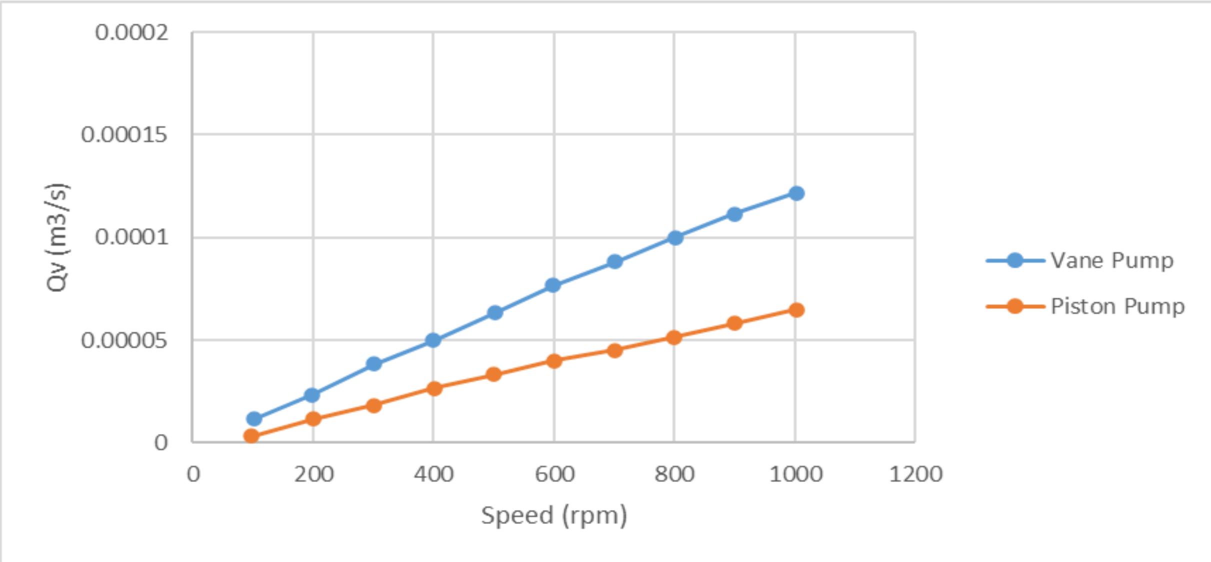

Figure (5): Flow rate vs. Pump Speed for piston and vane pumps.

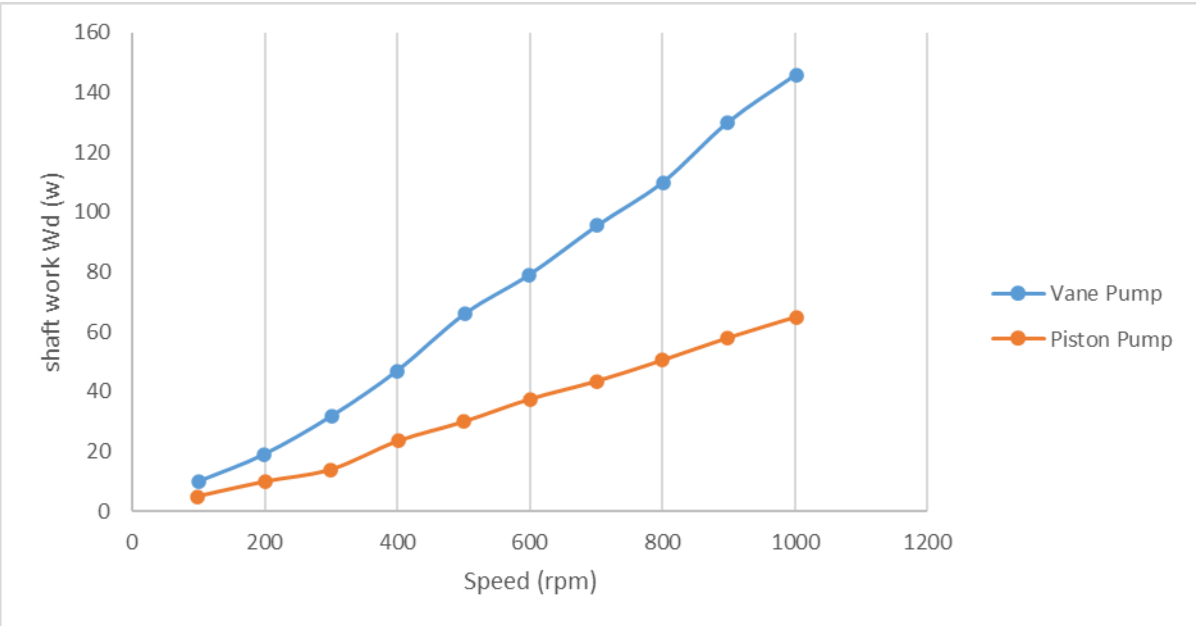

Figure (6): Shaft power vs. Pump speed for piston and vane pumps.

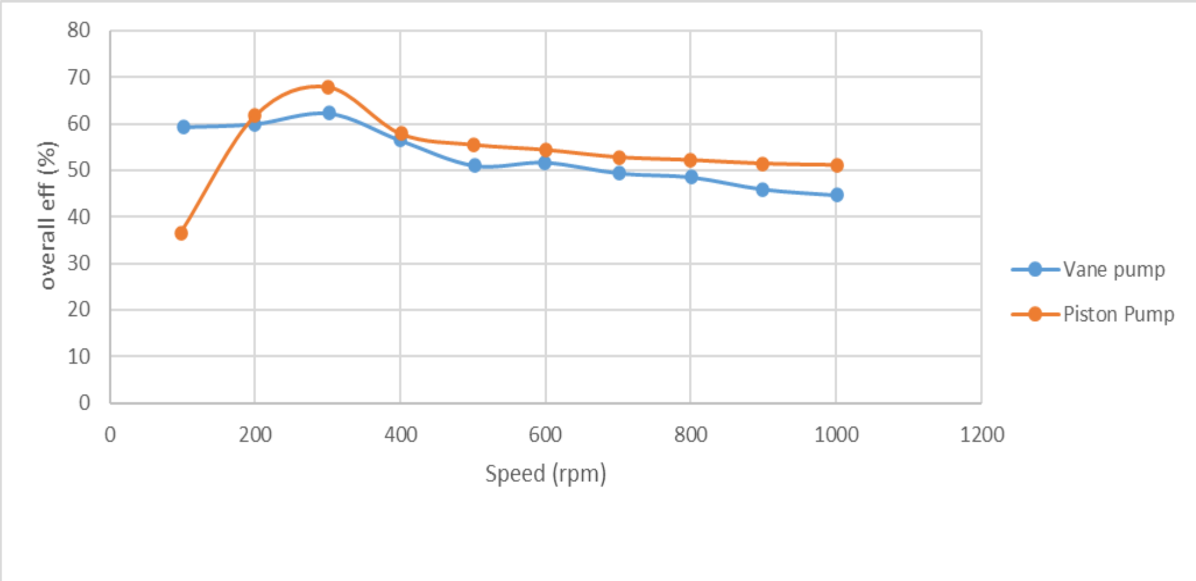

Figure (7): Overall efficiency vs. Pump Speed for piston and vane pumps.

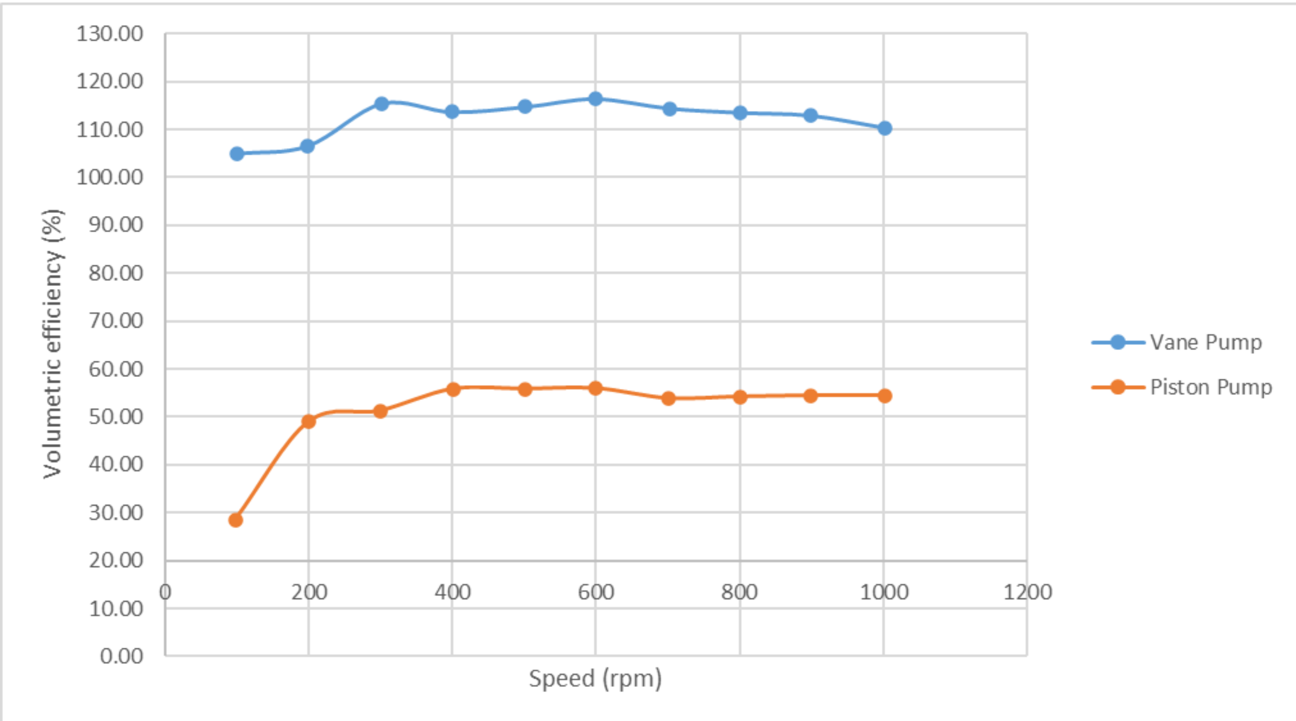

Figure (8): Volumetric efficiency vs. Pump Speed for piston and vane pumps.

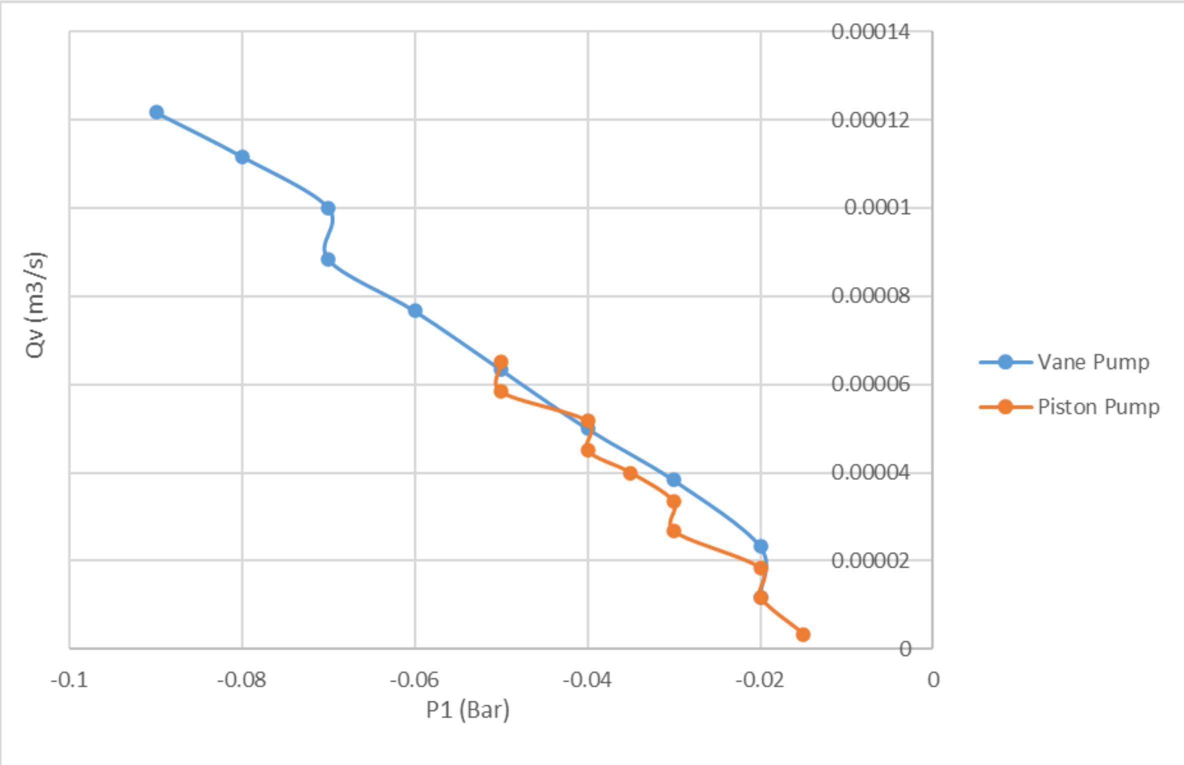

Table (9): Flow rate vs. Inlet pressure for piston and vane.

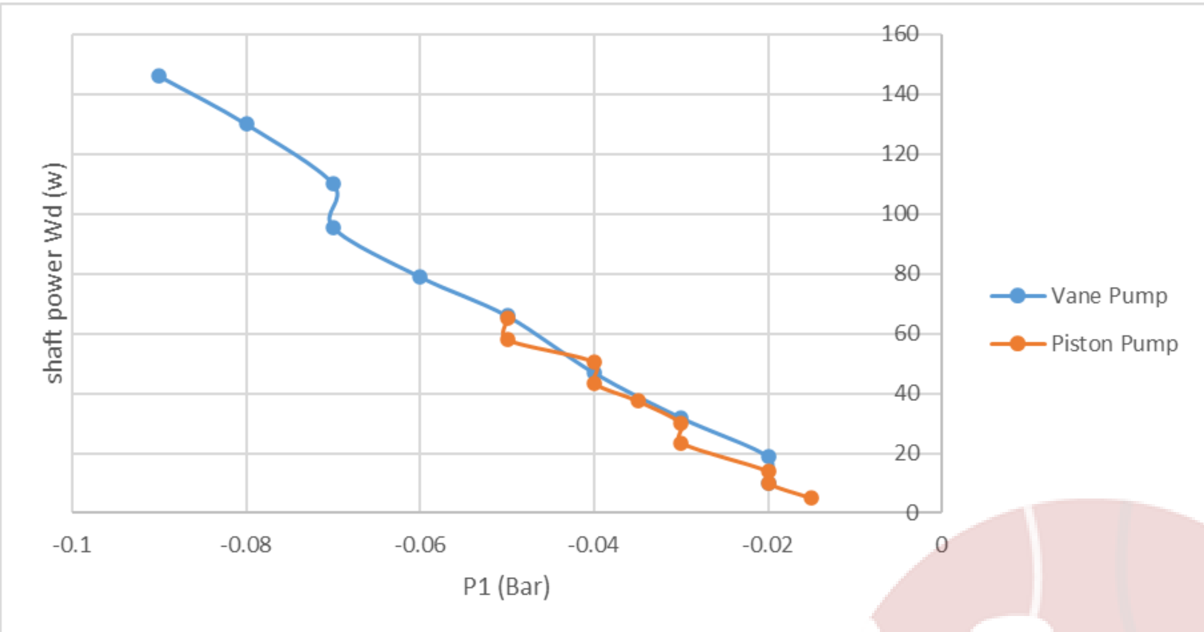

Table (10): shaft power vs. Inlet pressure for piston and vane.

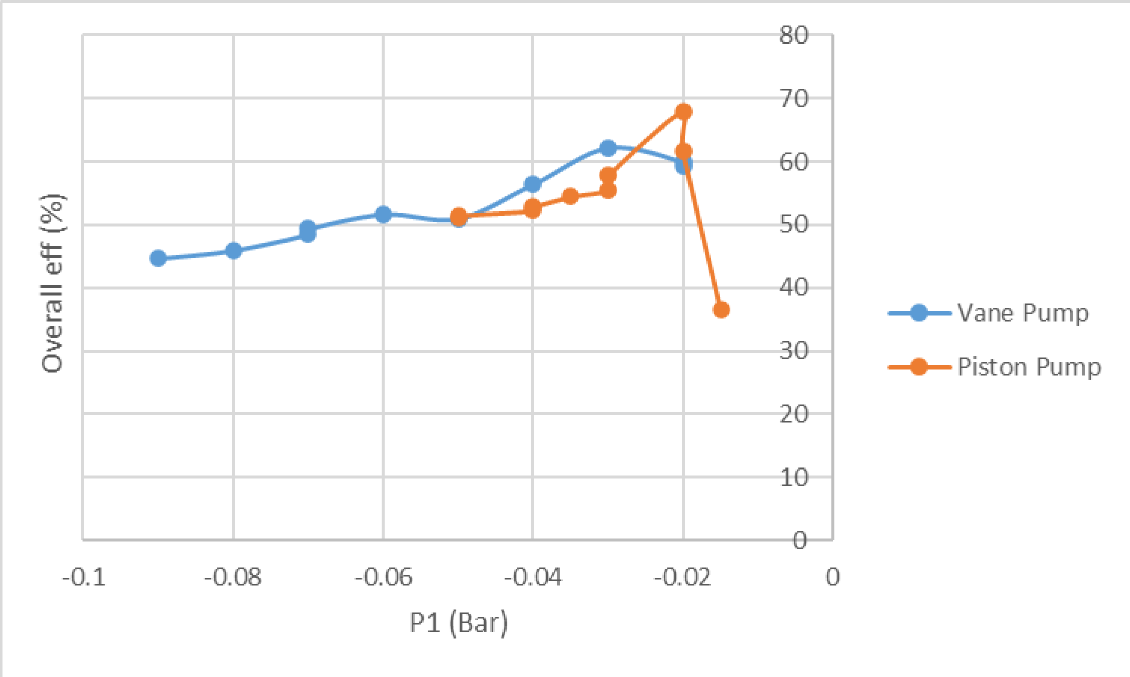

Table (11): Overall efficiency vs. Inlet pressure for piston and vane.

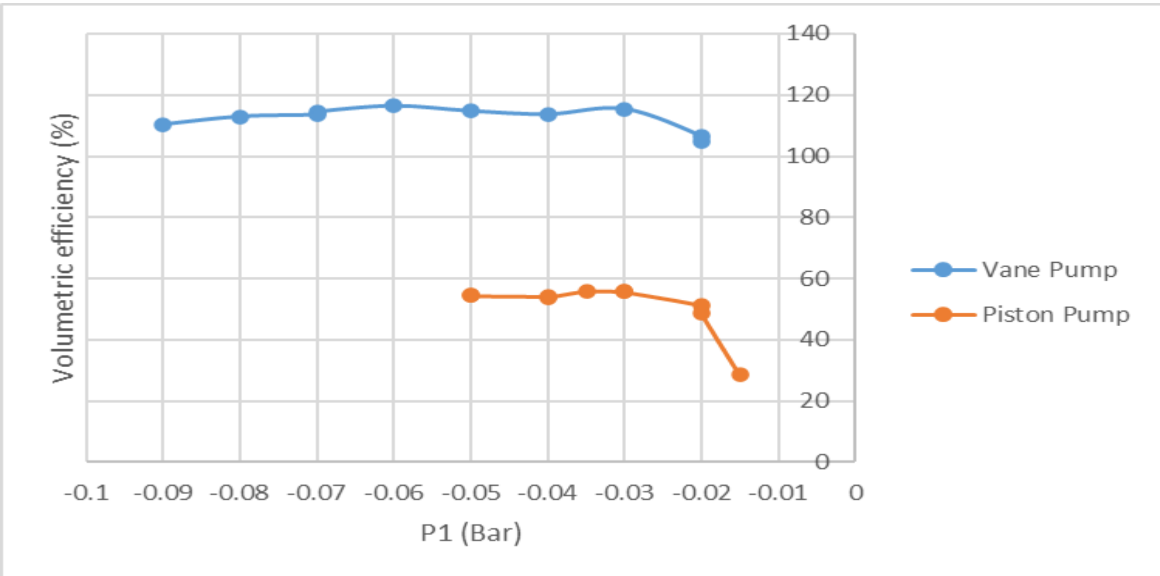

Table (11): Volumetric efficiency vs. Inlet pressure for piston and vane.

Discussion:

Two types of pumps are demonstrated; piston pump and vane pump, both of which are under the category of positive displacement pumps. Positive displacement pumps move a fluid by repeatedly enclosing a fixed volume and so, the fluid is moved mechanically (powered by a motor) through the pump and leaves with hydraulic power, as a result from the increase in pressure and flow. The pumping action is cyclic and is powered by pistons or vanes.

The piston pump is a reciprocating positive displacement pump and it works by the back-and-forth movement of a piston. The volumetric flow rates and overall efficiency are calculated once at a constant speed of 500 rpm, and another at a constant delivery pressure of 5 bar. It is found that at constant speed and increasing delivery pressure, the overall efficiency of the pump and shaft power are increased. For a constant delivery pressure and an increasing speed, the volumetric efficiency is increased due to increasing flow rate.

The vane pump is a rotary positive displacement pump which uses a set of movable vanes that maintain a closed seal against the casing wall causing the fluid to be discharged. At a constant speed of 500 rpm, delivery pressure is increasing causing the overall efficiency to increase as well. At a constant delivery pressure of 5 bar, the increasing speed causes an increase in the flow rate, and consequently, the volumetric efficiency increases too.

Figure (2) indicates an increasing pressure difference with increasing shaft power for both types of pumps, as well as an increase in the overall efficiency for both pumps as seen in figure (3).

(Ref: https://www.michael-smith-engineers.co.uk/resources/useful-info/positive-displacement-pumps).

Conclusion:

A few essential factors, including flow rate, pressure (inlet and delivery), shaft power, and efficiency (overall and volumetric), are the most important performance criteria to take into account when choosing positive displacement pumps.

The effect of constant speed and delivery pressure is affected by many factors. When the speed is constant and delivery pressure varies, the volumetric flowrate and its efficacy have a slight change in both vane and piston pumps. But shaft power and overall efficacy were increasing for both piston and vane pumps.

When the pumps are operating at constant delivery pressure and the speed varies, the volumetric flow rate for both pumps increase with increasing speed, but the volumetric efficiency has a slight change. For overall efficiency, it increases with increased speed. This is because the shaft power is increasing for both pumps.

Sources of Errors:

1.Experimental Errors:

Vibrating of the apparatus, fluctuating in readings.

2.Personal Error:

Errors in reading data, calculation errors.

References used:

1.F.A. Holland, “Fluid Flow for Chemical Engineers ", Arnold, 1980.

2. J.M. Coulson and FF Richardson," Chemical Engineering" Vol.1, Third Edition, 1980, Pergamon Press.

Sample of Calculations:

Starting with the piston pump at constant speed

Second row of each table:

- Pressure difference in bar: $\Delta P = P_2 - P_1 = 2.9 - (-0.03) = 2.93\ bar$.
- Pressure difference in Pascal: $\Delta P = 2.93\ bar(101325) = 296882 Pa$.

Volumetric flow rate: $Q_v = \frac{1.9L}{min} \times \frac{0.001\ m^3}{L} \times \frac{1\ min}{60\ s} = 3.16667 \times 10^{-5}\ m^3/s$

- Hydraulic power: $W_P = \Delta\mathrm{P}(Q_V) = 296882(3.16667 \times 10^{-5}) = 9.40\ W$.
- Expected flow rate: $Q_E = \frac{7.15 \times 499 \times 10^{-3}}{60 \times 1000} = 5.94 \times 10^{-5}\ m^3/s$
- Overall efficiency: $\eta_{overall} = \frac{W_P}{W_D} \times 100\% = \frac{9.40}{20.5} \times 100\% = 45.86\%$
- Volumetric efficiency: $\eta_v = \frac{Q_V}{Q_e} \times 100\% = \frac{3.16667 \times 10^{-5}}{5.94 \times 20^{-5}} \times 100\% = 53.25\%$

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speed = 500rpm , P2 from 2→9 (1 bar step)

Piston pump / Constant speed / (dis/rev) = 7.15 cc/rev

Speed	Torque	Power	Inlet Pressure (P1)	Delivery Pressure (P2) bar	oil temp	Flow Rate
500	0.34	18	-0.03	2.2	19.5	2
499	0.39	20.5	-0.03	2.9	19.5	2
498	0.47	24	-0.03	4	19.5	1.9
498	0.54	28	-0.03	4.95	19.45	1.9
500	0.59	31	-0.03	5.9	19.5	1.9
499	0.67	35	-0.03	7	19.5	1.9
498	0.73	37	-0.03	7.9	19.5	1.9
500	0.80	43	-0.03	9.0	19.6	1.9

Speed = 100 → 1000 rpm ±2 (100 step) , P2 = 5 bar ± 0.2

Piston pump Constant pressure (dis/rev) = 7.15 cc/rev

Speed rev/min	Torque N*m	Power W	Inlet Pressure (P1) bar	Delivery Pressure (P2) bar	oil temp °C	Flow Rate L/min
98	0.51	5	-0.015	5.3 - 5.8	18.9	0
200	0.5	10	-0.02	5.2	18.8	0.7
300	0.49	14	-0.02	5.7	18.9	1.2
401	0.56	23.5	-0.03	5	19	1.6
501	0.57	30	-0.03	4.9	19.3	2
600	0.6	37.5	-0.035	5	19.3	2.4
701	0.6	43.5	-0.04	5	19.2	2.7
800	0.6	50.5	-0.04	5	19.3	3.1
897	0.6	58	-0.05	5	19.4	3.5
1002	0.625	65	-0.05	5	19.5	3.9

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Vane pump / Constant speed / (dis/rev) =

Speed	Torque	Power	Inlet Pressure (P1)	Delivery Pressure (P2)	oil temp	Flow Rate
500				2 → 9		
~~1000~~ 498	~~+~~0.79	40.5	-0.05	2.1	20.3	3.8
499	0.91	47	-0.05	3	20.4	3.8
498	1.01	52	-0.05	4	20.5	3.8
500	1.20	63	-0.05	5	20.5	3.7
500	1.28	68	-0.05	6	20.5	3.7
498	1.39	74	-0.05	7	20.5	3.7
500	1.57	82	-0.05	8	20.6	3.7
500	1.66	87	-0.05	9	20.6	3.7

Vane pump Constant pressure (dis/rev) =

100→1000 rev/min Speed	Torque	Power	Inlet Pressure (P1)	Delivery Pressure (P2) 5 bar	oil temp	Flow Rate
101	0.98	10	-0.02	5.50	19.6	0.7
199	1	19	-0.02	4.8	19.6	1.4
302	1.02	32	-0.03	5.1	19.7	2.3
400	1.12	47	-0.04	5.2	19.8	3
502	1.22	66	-0.05	5.2	20	3.8
599	1.26	79	-0.06	5.2	20.1	4.6
702	1.29	95.5	-0.07	5.2	20.2	5.3
801	1.32	110	-0.07	5	20.2	6
899	1.37	130	-0.08	5.2	20.3	6.7
1002	1.39	146	-0.09	5.2	20.3	7.3

16/5/2023

The UNIVERSITY OF JORDAN

FACULTY OF ENGINEERING

& TECHNOLOGY

SCHOOL OF ENGINEERING

DEPT.OF CHEMICAL ENGINEERING

Chemical Engineering laboratory 1 (0915361)

Section no. (2)

Experiment Number (3)

Comparative Fluid Measurements

Full report

Instructor: Prof. Dr. Khaled Rawajfeh

Date of Submission: 18/4/2023

Abstract

The fluid flow is a unit of measurement for the volume of fluid that a vessel contains. In this experiment, the flow was measured using three hydraulic instruments: a rotameter, an orifice meter, and a venturi meter. A digital hydraulic bench, which also includes a water tank and a pump that supplies water to the hydraulic devices, was used to link these devices. Flow was calculated using the difference in pressure between two sites in each hydraulic device. Determining the discharge coefficient (Cd) of an orifice meter and a venturi meter at various Reynolds numbers (Re) and comparing the pressure drops between the two meters were the goals of this experiment. To build a calibration curve for the rotameter as well. According to the outcome of the experiment, the venturi meter is more precise than other hydraulic devices and the flow rates are closer to their actual values.

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Introduction

There are different forms of fluid flow measurement because each type has distinct needs for things like accuracy, cost, and how to use the flow data to get the results that are needed. When deciding which sort of meter is best to use to measure a specific flow, the nature of the fluid to be measured must be taken into consideration. The flow's characteristics are also very important.

The nature of the fluid to be measured must be taken into account while choosing the optimum type of meter to measure a certain flow. Characteristics of the flow are also crucial. The best flow measurement is needed in custody transfer metering in order to treat the two parties to the transactions fairly. This experiment takes into account the variables that must be taken into account while selecting a meter to measure fluids in various scenarios.

The goals of this experiment are to develop a rotameter calibration curve, compare pressure drops via an orifice and venturi meter, and determine the discharge coefficient of each device at various Reynolds numbers.

Theory

There are numerous additional devices in this device, and to compare them in terms of flow, these additional devices mostly rely on using Bernoulli's equation to calculate the flow rate from the pressure differential in each device. For Venturi, Orifice and rotameter:

$$\frac{P_1}{\rho.g} + \frac{{u_1}^2}{2.g} + z_1 = \frac{P_2}{\rho.g} + \frac{{u_2}^2}{2.g} + z_2 + \Delta h_{12} \dots\dots\dots (1)$$

Where u is the water's velocity and (Δh12) is the head loss as a result of friction and localized effects (area change or fitting).

$$\Delta H = (h_1 - h_2) + \frac{{u_1}^2 - {u_2}^2}{2.g} - \Delta h_f \dots\dots\dots (2)$$

$$\Delta h_{12} = \Delta H + \Delta h_f \dots\dots\dots (3)$$

Where the head loss due to friction (Δhf) and the head loss due to fitting (ΔH) are both present. If the length is brief, (Δhf) can be neglected. Typically, the head loss is stated in terms of the loss coefficient (K), which is given as follows:

$$k = \frac{\Delta H}{\left({u^2}/{2g}\right)} \dots\dots\dots (4)$$

Where (u) is the velocity in the smaller pipe.

A) <u>Venturi meter</u>

Since the distance between the ends of a contracting duct, h12, is small, using equation (1) between pressure tappings (A) and (B) results in:

$$\frac{P_A}{\rho.g} + \frac{{u_A}^2}{2.g} = \frac{P_B}{\rho.g} + \frac{{u_B}^2}{2.g} \dots\dots\dots (5)$$

and since, by mass balance:

$$\dot{m}_A = \rho.u_A.A_A = \dot{m}_B = \rho.u_B.A_B \dots\dots\dots (6)$$

Substitute (6) in (1) will get:

$$u_B = \left[\frac{2g}{(1-(A_B/A_A)^2)} \times \left(\frac{P_A}{\rho.g} - \frac{P_B}{\rho.g}\right)\right]^{\frac{1}{2}} \dots\dots\dots (7)$$

Now,

$$Q_{Th} = A_B.u_B$$

$$Q_{Th} = A_B.\left[\frac{2g}{(1-(A_B/A_A)^2)} \times \left(\frac{P_A}{\rho.g} - \frac{P_B}{\rho.g}\right)\right]^{\frac{1}{2}} \dots\dots\dots (8)$$

This is theoretical valve.

$$Q_{act} = C_v.A_B.\left[\frac{2g}{\left(1-\left(\frac{A_B}{A_A}\right)^2\right)} \times (h_A - h_B)\right]^{\frac{1}{2}} \dots\dots\dots (9)$$

where (Qact) is the actual flow rate and (Cv) may found from experiment.

B) <u>Orifice meter</u>

Applying equation (1) between (E) and (F) results in head losses that are by no means insignificant. Equation should be rewritten using the proper symbols.

$$\frac{{u_F}^2}{2.g} - \frac{{u_E}^2}{2.g} = \frac{P_E}{\rho.g} - \frac{P_F}{\rho.g} \dots\dots\dots (10)$$

The following equation will be created by reducing equation (10) in the exact same manner as for the venturi meter:

$$Q_{act} = C_d.A_F.\left[\frac{2g}{\left(1-\left(\frac{A_F}{A_E}\right)^2\right)} \times (h_E - h_F)\right]^{\frac{1}{2}} \dots\dots\dots (11)$$

Where Cd is the coefficient of discharge.

Apparatus

The device for Flow Measurement Methods, Through a Venturi meter, which has a lengthy, gradually diverging portion, a throat, and a gradually convergent section, water from the Hydraulic Bench enters the apparatus. The flow continues down a settling length and through an orifice meter from a plate with a hole of decreasing diameter after changing cross-section through a quickly diverging section. Following a bend, the water travels up through a flow meter made of rotameters. The H10 features eleven manometers, nine of which are connected to pipework tappings and two of which are left open for further measurements.

Illustration (1) : The device for fluid flow measurements methods

A) *<u>Digital Hydraulic Bench</u>*

A hydraulic bench is a device for measuring volumetric flow that has a water reservoir at the bottom via which water is forced up to an open channel via a control valve. Finally, this water is released into the volumetric tank, where the change in water level, as indicated by the calibrated scale, determines the flow rate. It may also be used to supply water to other apparatuses for their working.

Illustration (2-A) : Digital Hydraulic Bench

Digital Hydraulic Bench construction:

(1) ***On/Off Switch for Motor***	*In order to start the motor of centrifugal pump, the motor switch is turned on. This results in initiation of the pumping process.*
(2) ***Discharge Control Valve***	***This valve controls the flow rate into the volumetric tank. A clockwise rotation tends to decrease the discharge and a counter clockwise rotation increases the flow rate.***
(3) ***Sight Tube and Scale***	***A sight tube present close to the discharge control valve gives an instantaneous value of flow rate in to volumetric tank and the value can be read from the scale present beside the tube.***
(4) ***Volumetric Tank***	***This tank is present downstream of the open channel. It receives water from the open channel. However, the flow rate into this tank is governed by the discharge control valve. The change in level of water in the tank is determined by noting the difference in level of water given by the sight tube.***
(5) ***Open Channel***	***This channel receives pumped water from the sump tank and the flow rate into this channel is governed by flow control valve present beneath it.***
(6) ***Centrifugal Pump***	***This pump converts the mechanical energy of motor into hydraulic energy by the action of centrifugal force.***

*(7) **Sump Tank***	***A sump tank is actually a water reservoir present at the bottom of the hydraulic bench. The water is pumped from the sump tank into the open channel. In addition, when the water level in the volumetric tank reaches a certain value, it empties into sumo tank and the process is repeated.***
*(8) **Flow Control Valve***	***This valve controls the flow from the sump tank into the open channel.***
*(9) **Inlet Stilling Baffle***	***It acts as a stopper for water to retain in the open channel. When water travels up into the open channel, the inlet stilling basin helps dispersing the water stream, thereby, keeping water in the channel. See illustration (2-C)***
*(10) **Tank Stilling Baffle***	***When water enters the volumetric tank from the open channel, a tank stilling basin ensures smooth gliding of water. If this is not provided, the water that strikes the tank bottom may produce some local turbulence. This may result in continuous fluctuation of water level that can be seen from the sight tube. Therefore, the discharge measurement may give misleading results.***
*(11) **Dump Valve Handle***	***A dump valve handle is used to drain water from the volumetric tank back into the sump tank. This can be achieved by raising the handle. Additionally, the overflow in the volumetric tank puts a limit on its maximum capacity, after which water dumping indispensable.***

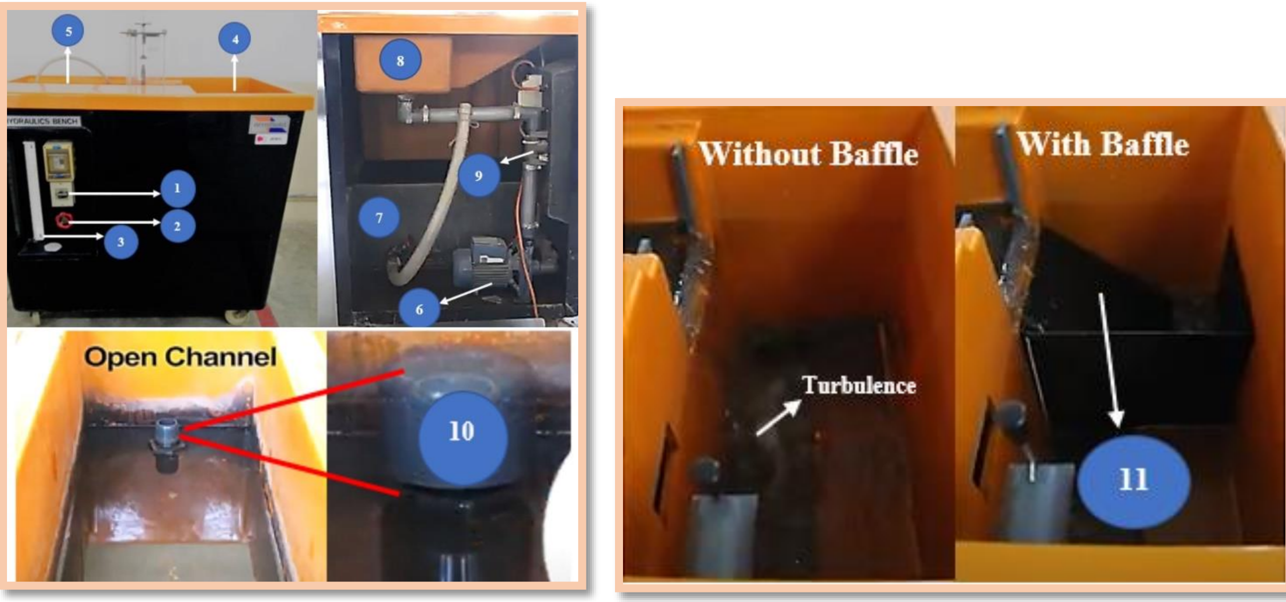

Illustration (2-B) : Digital Hydraulic Bench construction

Illustration (2-C) : Inlet Stilling Baffle

The Uses of Hydraulic Bench:

There are several hydraulic procedures that hydraulic benches can be used for, including the following:

1) It guarantees the continuous and controlled supply of water for a variety of scientific experiments.
2) It aids in flow rate or discharge measurement.
3) It functions as a mobile, self-contained unit since it circulates wate

B) _Venturi Meter_

The flowrate can be calculated using venturi meters, which use a converging segment of pipe to produce a rise in flow velocity and a corresponding pressure decrease. They have long been in widespread usage, particularly in the water supply sector.

The pipeline is connected to a venturi meter, the constriction at the throat increases the fluid's velocity as it passes past the venturi meter.

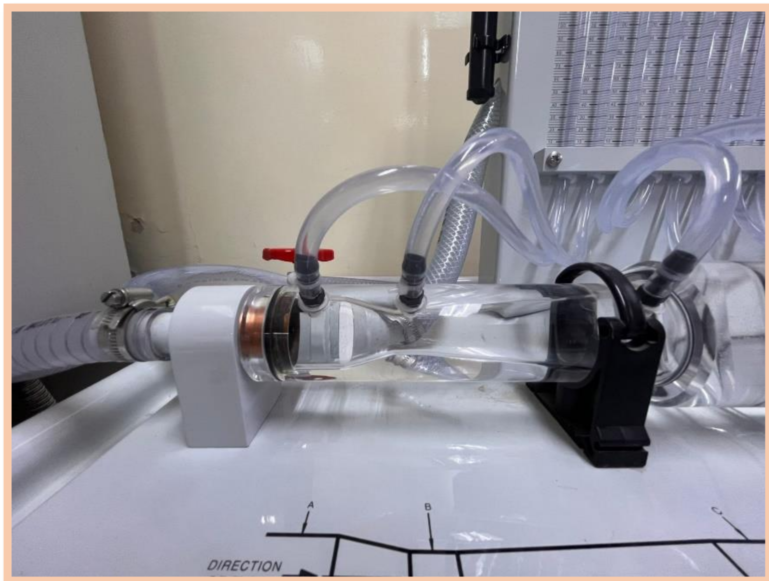

Illustration (3-A): Venturi meter

<u>Venturi meter Construction</u>

It has two tapering portions and is placed into a pipeline that is constricted in the middle where the meter is located. Typically, the downstream cone is shorter than the upstream cone. The tapers are gradual and smooth. Points 1 and 2 are connected to a manometer (or a piezometer), which measures the pressure differential. As shown in illustration (3-3-B):

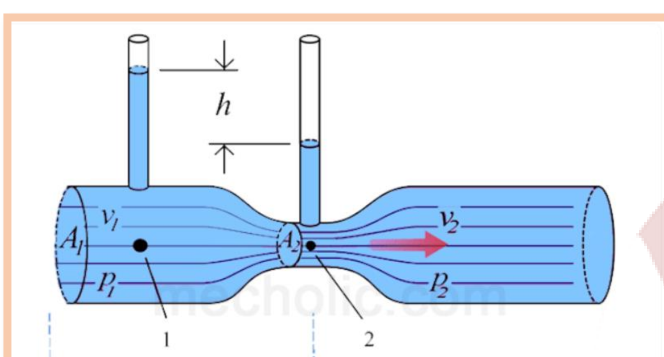

Illustration (3-3-C): Venturi meter construction

The Uses of Venturi meter

1) It is frequently used for both gases and liquids, such as water.
2) It is applied anywhere high-pressure recovery is required.
3) It can be used to measure the flow rates of water, gaseous liquids, suspended liquids, and dirty liquids.
4) Enables high flow rate measurement in pipelines with a few meters of diameter.

C) Orifice meter

Clean liquid, gas, and stream mass flow are frequently measured with differential pressure flow meters of the orifice plate variety. It can measure fluid flows in bigger pipes (above 6" in diameter) and is particularly cost-effective if the pressure loss it needs is free. It is available for all pipe sizes. Numerous standards bodies have also authorized the orifice plate for the custody transfer of liquids and gases.

Illustration (4-A) : Orifice meter

Orifice meter Construction:

A linear portion that extends from the device and serves as an end connection for the fluid moving inside is known as the inlet section, the pressure of the gas or fluid expelled is determined in the outlet section, which is a linear segment comparable to the input section and between the outlet and inlet segments, there is an orifice plate that is utilized to create a pressure drop that allows the flow rate to be measured. Also, a flow conditioner is a device that is positioned in the meter tube's inlet portion and used to improve linear flow there.

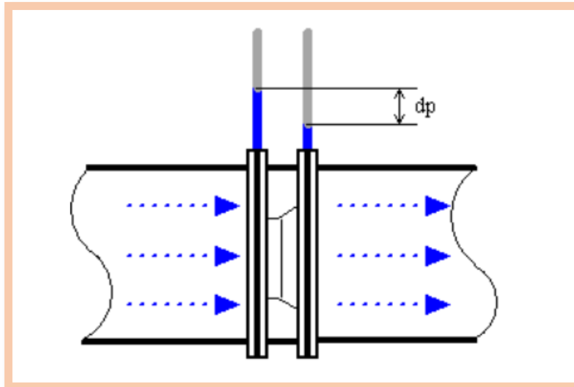

Illustration (4-B) : Orifice meter construction

The uses of Orifice meter

It is used to gauge the flow rate of substances in their pure states, such as gaseous or liquid. Additionally, it can be used to gauge the flow rate of mixtures of fluids, such as wet steam or natural gas and water, which can exist in both gaseous and liquid phases.

D) *Rotameter*

A dependable, straightforward, and affordable flow measurement tool is the Rotameter. In order to measure the flow rate of a liquid or a gas, this instrument is utilized. This meter has a tapered tube that passes through a floating internal component. Rotameters may also go by the names gravity flow meters, mechanical flow meters, or variable area flow meters. The term "gravity" refers to the requirement that the rotameter be positioned vertically in gravity flow meters.

Illustration (5-A) : Rotameter

<u>Rotameter Construction:</u>

Transparent tubes, scales, floats, and transmitter are a few of the components that can be used to build rotameters.

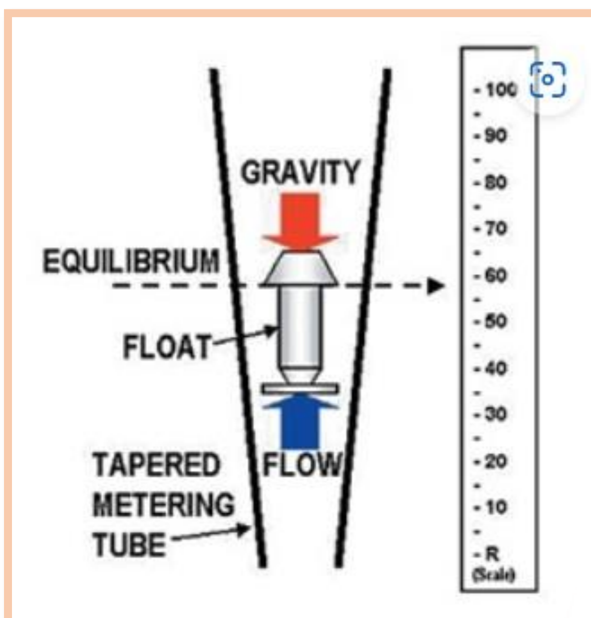

Illustration (5-B) : Rotameter construction

The transparent tube in the construction has a conical shape and floats inside it together with a scale. This clear tube is quite useful for visually inspecting the measurements. A float is a tiny, precisely sized object that fits into the tube of a rotameter. The float, which can be made of plastic, glass, or metal, is used to show how quickly liquid is flowing through the tube. This meter's scale shows the flow readings by using float. As opposed to physically observing the scale, transmitters are highly helpful in accurately documenting the flow data. It consists of a float and a tapered tube, with the float placed inside the tube. Nets are organized through a pipeline at the device's two ends utilizing flanged connections. In pipelines, rotameters are always attached vertically, and a scale is available on the tube for checking the flow rate values immediately.

<u>Rotameter working:</u>

A portion of the liquid that flows from the tapering tube's base strikes the float's bottom directly, while the remainder flows away from it. Due to gravity and drag, respectively, the float in the rotameter is subject to two forces acting in the opposite directions.

The float is propelled upward from gravity by the liquid flow. After a while, the flowing zone reaches a place where the force applied to the floating body is exactly equal to the weight of the float. Therefore, the float will reach equilibrium once the area where it is moving causes enough drag to equal its own weight.

E) *<u>Piezometer</u>*

Is a device that measures the height to which a liquid column rises against gravity in a system.

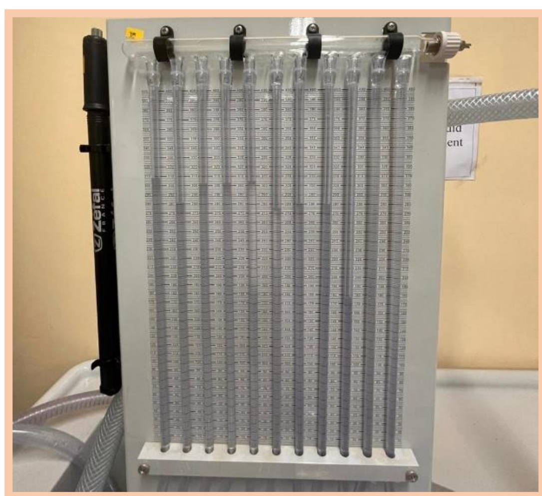

Illustration (6-A) : Piezometer

Procedure

Press the on/off switch on the hydraulic bench to turn on the pump. Then, after ensuring a constant flow, open the apparatus valve until the Rotameter reads a measurement of approximately 10 mm. Next, gauge the flow using the hydraulic bench. During this period, keep track of the manometer readings. Repeat this procedure until the manometer's maximum pressure values can be recorded, which should take place after a number of evenly spaced Rotameter readings.

Results

Table 1: Experiment Parameters

Pressure (mm Hg)	676.00
Temperature (°C)	16.00
Gravity Acceleration (m2/s)	9.81
Water's density (kg/m3)	998.9
Water's viscosity (cP)	1.11

Table 2: Raw Data

Rotameter (cm)	Flow rate (L/min)	Flow rate (m^3/s)	h_A	h_B	h_C	h_D	h_E	h_F	h_G	h_H	h_I
10	3.8	6.31E-05	306.00	300	302	304	304	300	300	300	196
20	4.9	8.13E-05	304.00	276	300	300	302	294	296	296	194
30	6.2	1.03E-04	304.00	292	302	302	304	288	290	290	190
40	7.6	1.26E-04	306.00	286	300	300	302	282	286	284	184
50	8.8	1.46E-04	306.00	280	300	300	302	276	280	280	180
60	10.1	1.68E-04	308.00	274	300	302	304	268	276	274	172
70	11.6	1.93E-04	310.00	268	302	304	306	260	268	266	164
80	13.00	2.16E-04	314.00	260	304	306	308	250	260	258	158
90	14.2	2.36E-04	318.00	254	306	306	310	242	252	252	150
100	15.7	2.61E-04	320.00	246	308	308	314	230	244	240	138
120	18.7	3.10E-04	332.00	226	314	316	322	204	222	220	114
140	21.7	3.60E-04	344.00	202	322	324	332	174	198	194	86
160	24.9	4.13E-04	362.00	176	334	336	346	136	168	164	54
180	28.2	4.68E-04	382.00	144	348	352	364	94	134	130	14

**Note that all head losses are measured in mmH2O.*

Table 3: Venturi Meter Calculated Parameters

Pressure Difference (mmH2O)	Pressure Difference (m H2O)	Pressure Difference (Pa)	Qactual (m3/s)	Cv
6.00	0.006	58.836	6.31E-05	8.55E-03
8.00	0.008	78.448	8.13E-05	9.54E-03
12.00	0.012	117.672	1.03E-04	9.86E-03
20.00	0.02	196.12	1.26E-04	9.36E-03
26.00	0.026	254.956	1.46E-04	9.51E-03
34.00	0.034	333.404	1.68E-04	9.54E-03
42.00	0.042	411.852	1.93E-04	9.86E-03
54.00	0.054	529.524	2.16E-04	9.74E-03
64.00	0.064	627.584	2.36E-04	9.78E-03
74.00	0.074	725.644	2.61E-04	1.01E-02
106.00	0.106	1039.436	3.10E-04	1.00E-02
142.00	0.142	1392.452	3.60E-04	1.00E-02
186.00	0.186	1823.916	4.13E-04	1.01E-02
238.00	0.238	2333.828	4.68E-04	1.01E-02

Table 3: Venturi Meter Calculated Parameters (Continued)

u_B (m/s)	u_A (m/s)	$u_B^2/2g$	**ΔH (m H2O)**	K	Re
0.37	0.14	0.007	0.012	1.713	5350.60
0.43	0.16	0.009	0.016	1.713	6178.80
0.53	0.20	0.014	0.024	1.712	7568.40
0.68	0.27	0.023	0.040	1.713	9768.90
0.77	0.29	0.030	0.052	1.712	11139.80
0.88	0.33	0.040	0.068	1.712	12738.60
0.98	0.37	0.049	0.084	1.712	14158.50
1.11	0.42	0.063	0.11	1.712	16054.50
1.21	0.46	0.075	0.13	1.712	17477.30
1.30	0.49	0.086	0.15	1.712	18793.30
1.56	0.59	0.12	0.21	1.712	22493.10
1.80	0.68	0.17	0.28	1.712	26034.20
2.07	0.78	0.22	0.37	1.712	29794.60
2.34	0.89	0.28	0.48	1.712	33703.60

Table 4: Diffuser Calculation

h_C	h_D	Pressure Difference (mmH2O)
302.00	304.00	-2.00
300.00	300.00	0.00
302.00	302.00	0.00
300.00	300.00	0.00
300.00	300.00	0.00
300.00	302.00	-2.00
302.00	304.00	-2.00
304.00	306.00	-2.00
306.00	306.00	0.00
308.00	308.00	0.00
314.00	316.00	-2.00
322.00	324.00	-2.00
334.00	336.00	-2.00
348.00	352.00	-4.00

Table 5: Orifice Meter Calculated Parameters

Pressure Difference (mmH2O)	Pressure Difference (m H2O)	Pressure Difference (Pa)	Qactual (m3/s)	Cd
4.00	0.004	39.22	6.31E-05	7.16E-03
8.00	0.008	78.45	8.13E-05	6.53E-03
16.00	0.016	156.90	1.03E-04	5.84E-03
20.00	0.02	196.12	1.26E-04	6.41E-03
26.00	0.026	254.96	1.46E-04	6.51E-03
36.00	0.036	353.02	1.68E-04	6.34E-03
46.00	0.046	451.08	1.93E-04	6.45E-03
58.00	0.058	568.75	2.16E-04	6.43E-03
68.00	0.068	666.81	2.36E-04	6.49E-03
84.00	0.084	823.70	2.61E-04	6.46E-03
118.00	0.12	1157.11	3.10E-04	6.49E-03
158.00	0.16	1549.35	3.60E-04	6.51E-03
210.00	0.21	2059.26	4.13E-04	6.48E-03
270.00	0.27	2647.62	4.68E-04	6.47E-03

Table 5: Orifice Meter Calculated Parameters (continued)

uF (m/s)	uE (m/s)	uf^2/2g	ΔH (m H2O)	K	Re
0.28	0.042	0.004	0.008	1.955	5111.70
0.40	0.059	0.008	0.016	1.955	7229.30
0.57	0.084	0.016	0.032	1.955	10223.50
0.63	0.094	0.020	0.04	1.955	11430.20
0.72	0.11	0.027	0.052	1.955	13031.90
0.85	0.13	0.037	0.072	1.955	15333.40
0.96	0.14	0.047	0.092	1.955	17333.70
1.08	0.16	0.059	0.12	1.955	19463.90
1.17	0.17	0.070	0.14	1.955	21074.70
1.30	0.19	0.086	0.17	1.955	23430.30
1.54	0.23	0.12	0.24	1.955	27761.10
1.78	0.26	0.16	0.32	1.955	32124.20
2.05	0.31	0.22	0.42	1.955	37034.0
2.33	0.35	0.28	0.54	1.955	41994.20

Table 6: Bend Calculation

hG	hH	Pressure Difference(mm H2O)
300.00	300.00	0.00
296.00	296.00	0.00
290.00	290.00	0.00
286.00	284.00	2.00
280.00	280.00	0.00
276.00	274.00	2.00
268.00	266.00	2.00
260.00	258.00	2.00
252.00	252.00	0.00
244.00	240.00	4.00
222.00	220.00	2.00
198.00	194.00	4.00
168.00	164.00	4.00
134.00	130.00	4.00

Table 7: Rotameter Calculation

hH	hl	Pressure Difference (mmH2O)
300.00	196.00	104.00
296.00	194.00	102.00
290.00	190.00	100.00
284.00	184.00	100.00
280.00	180.00	100.00
274.00	172.00	102.00
266.00	164.00	102.00
258.00	158.00	100.00
252.00	150.00	102.00
240.00	138.00	102.00
220.00	114.00	106.00
194.00	86.00	108.00
164.00	54.00	110.00
130.00	14.00	116.00

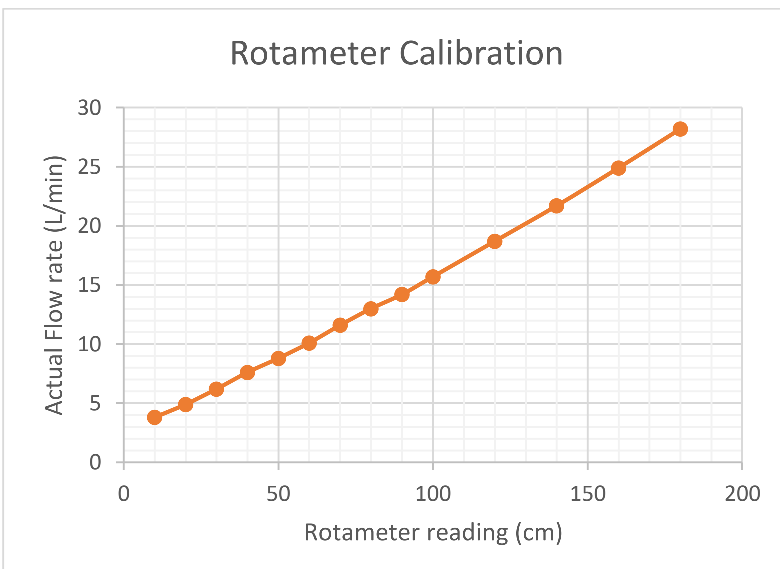

Figure (1): Rotameter Calibration

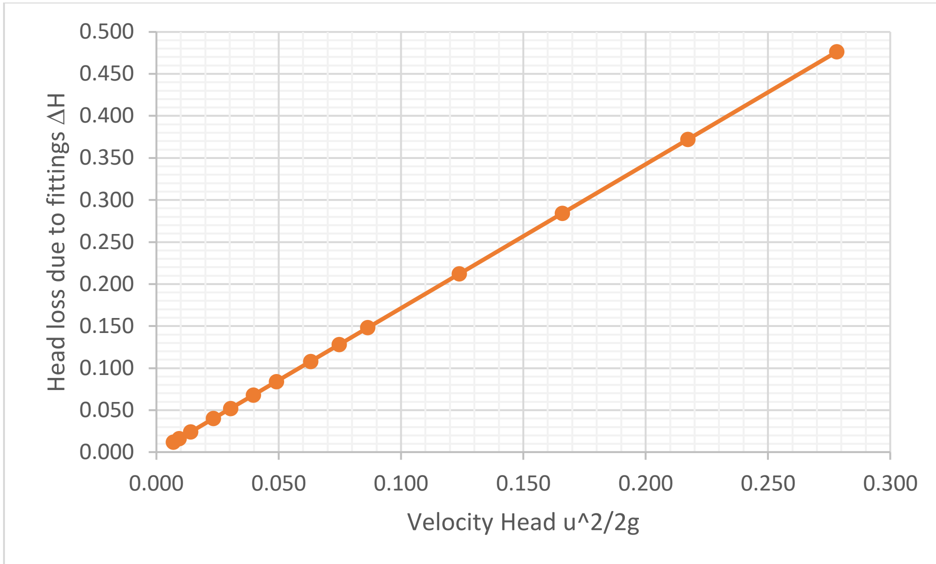

Figure (2): Venturi's *ΔH (head loss) vs* $(u^2/2g)$

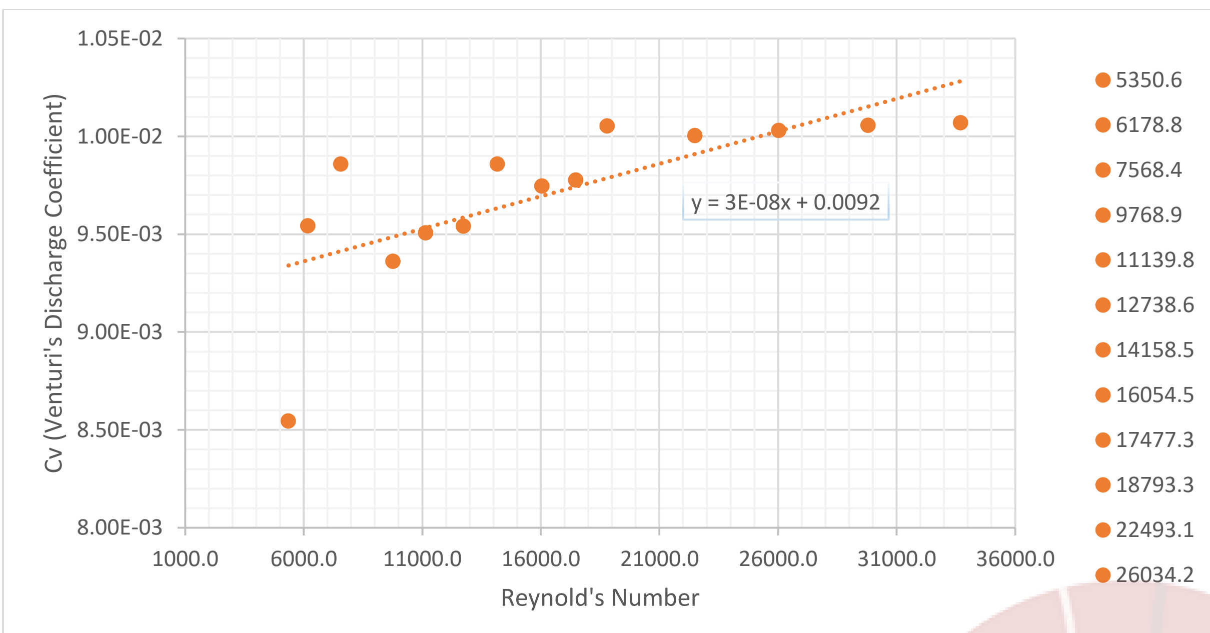

Figure (3): Discharge Coefficient VS. Reynold's Number (Venturi Meter)

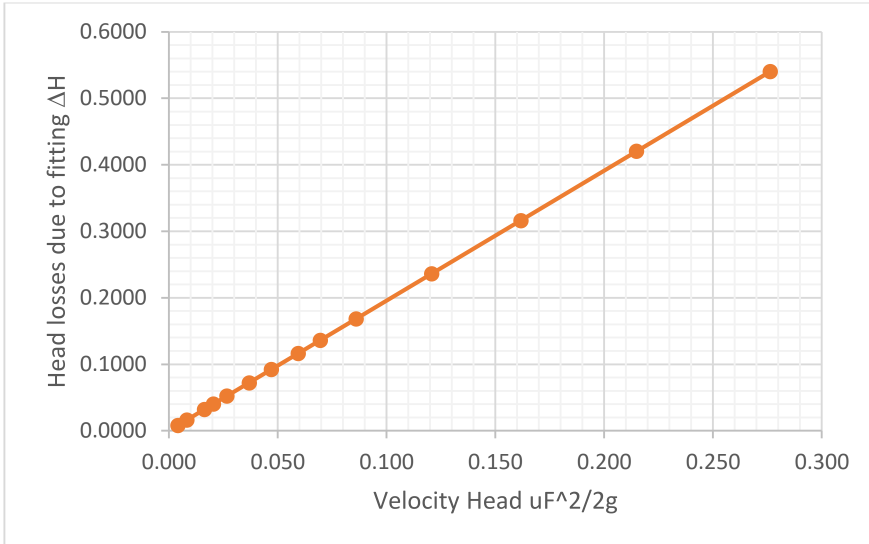

Figure (4): Orifice's *ΔH (head loss) Vs. (u^2/2g)*

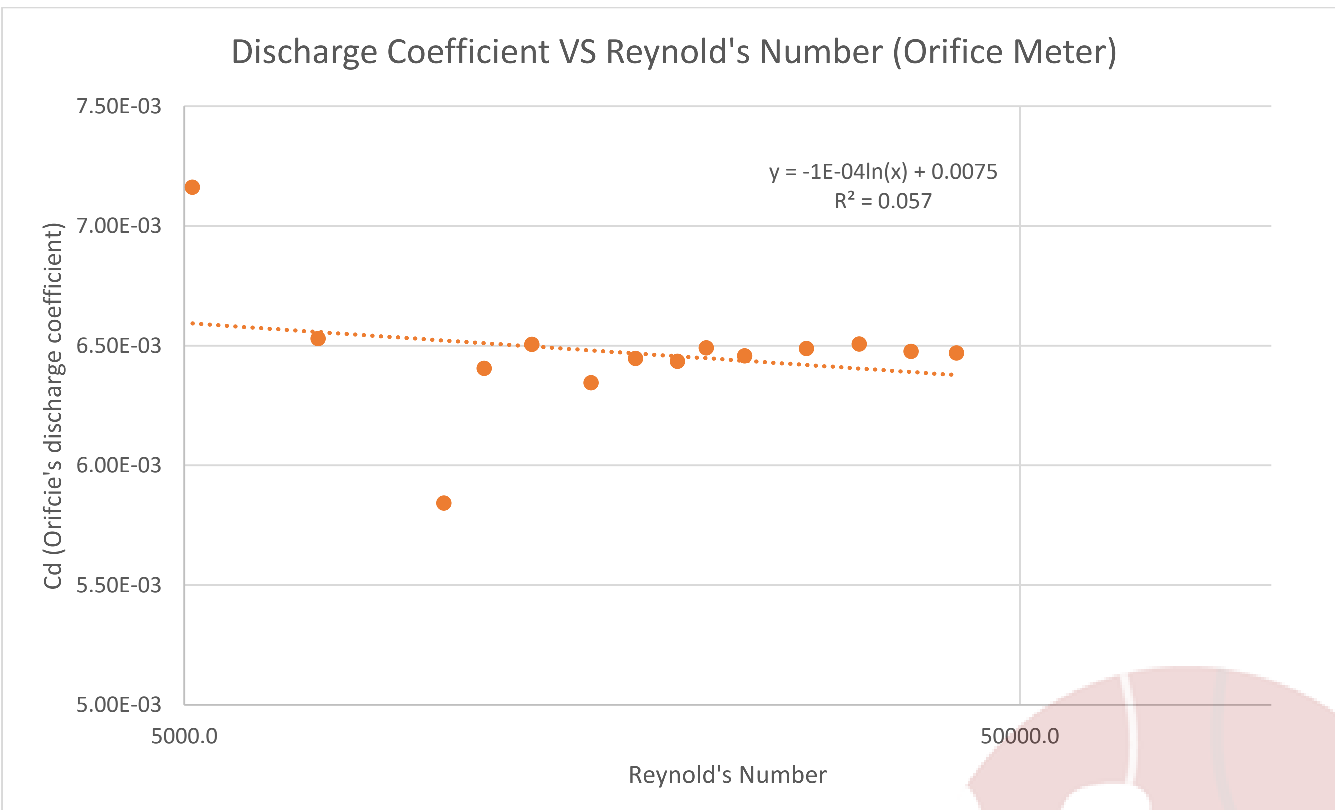

Figure (5): Discharge Coefficient Vs. Reynold's Number (Orifice Meter)

Discussion

According to the experimental findings, as the water flow rate (L/min) is increased, the Rotameter height (cm) also increases, causing a gradual increase in the values of the discharge coefficients, inlet and exit velocities, loss coefficients, head losses, and, finally, the Reynolds number. The experimental variable is the water flow rate through the Rotameter.

The Venturi meter exhibits lower values for pressure differentials, head losses, and Reynolds numbers than an orifice meter. For both flow meters, Figure (2) depicts an increasing linear relationship between the head loss resulting from fittings and the exit velocity head.

The head loss numbers in the orifice meter are higher than those in the Venturi meter because of the abrupt change in the orifice's cross sectional area, which causes a more turbulent flow and is demonstrated by the higher values of Reynolds number. The loss coefficient of the venturi meter is (1.712), while that of the orifice meter is (1.955). This demonstrates that the venturi meter has less losses than the orifice meter.

Despite the venturi meter appearing to be more linear than the orifice meter, both exhibit some nonlinearity when the drag coefficient is plotted against Reynolds number. This tendency arises from the experiment's neglect for friction losses, systematic instrumental errors caused by variations in the pipe system, and individual errors caused by erroneous piezometer readings. These facts show the venturi meter to be a more useful instrument than the orifice meter.

Conclusion and Recommendations

Head loss brought on by changing cross sectional resistance and water. The data indicated that the orifice meter was the location of the largest pressure drop, the location of the flow velocity increase at the orifice meter opening due to the abrupt narrowing of the cross section, and the location of the modest energy loss. However, as the fluid continues to flow, the energy losses increase and the stream's velocity decreases. Because there was less pressure loss, it was thought that the Venturi meter was more accurate. The ideal flow (frictionless flow) and the actual flow are therefore nearly equal.

The Rotameter has a lower pressure drop, but it has the drawback of relying on the fluid to move the body up, which is influenced by the fluid's many qualities, particularly density and temperature and the Venturi meter was more exact. Error sources include: Personal Error and uncertainty in data reading from the device.

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Notations

Symbol	Definition	[Unit]
P	Pressure	[mmH2O]
ρ	Density	[Kg/m^3]
g	Gravity acceleration	[m/s^2]
u	Fluid Velocity	[m/s]
z	Fluid height	[m]
ΔH	Head loss	[mmH2O]
K	Loss coefficient	
A_A	Cross-sectional Area at point A	$[m^2]$
Qth	Theoretical volumetric flowrate of the fluid	$[m^3/s]$
Qact	Actual volumetric flowrate of fluid	$[m^3/s]$
Cv	Valve flow Coefficient	
Cd	Coefficient of discharge	
Re	Reynold Number	

Appendices

1. ***Sample of Calculations:***

Given the following experiment parameters:

Pressure	676 mmHg
Temperature	16°C
Gravity Acceleration	9.81 m^2/s
Water's density	998.9 kg/m^2
Water's viscosity	1.1076 cP

The following values are found in the manual Figure (23):

Venturi meter (A: inlet, B: outlet)	$\boldsymbol{d_A = 26mm, A_A = 530.9mm^2}$ $\boldsymbol{d_B = 16mm, A_B = 201.1\ mm^2}$
Orifice meter (E: inlet, F: outlet)	$d_E = 51.90\ mm, A_E = 2116\ mm^2$ $d_F = 20mm, A_F = 314mm^2$

For the venturi meter, taking the first row:

❖ Pressure difference:

$$h_A - h_B = 306 - 300 = 6\ mmH_2O = 0.006\ mH_2O = 58.836\ Pa.$$

❖ Using Bernoulli's Equation for the further calculations:

$$\frac{P_1}{\rho.g} + \frac{u1^2}{2.g} + z_1 = \frac{P_2}{\rho.g} + \frac{u2^2}{2.g} + z_2 + \Delta h_{12}.$$

❖ Drag Coefficient (Cv):

$$C_V = \frac{Q_{actual}}{A_B\sqrt{\frac{2.g}{(1-(\frac{A_B}{A_A})^2)}\times(h_A-h_B)}} = \frac{6.31\times10^{-5}}{(0.0002011)\sqrt{\frac{2(9.81)}{(1-(\frac{201.1}{530.9})^2)}\times(58.836)}} = 8.5\times10^{-3}.$$

Noting that $Q_{actual} = 3.8\ \frac{L}{min}\times\frac{1\ m^3}{1000L}\times\frac{1\ min}{60\ s} = 6.3\times10^{-5}m^3/s.$

❖ Outlet velocity:

$$u_B = \sqrt{\frac{2g}{(1-\left(\frac{A_B}{A_A}\right)^2)} \times (\frac{(h_A - h_B)}{\rho . g})} = \sqrt{\frac{2(9.81)}{(1-\left(\frac{201.1}{530.9}\right)^2)} \times \frac{58.836}{(998.9)(9.81)}} = 0.37\frac{m}{s}.$$

❖ Inlet velocity: (using continuity equation):

$$u_A = u_B \frac{A_B}{A_A} = (0.37)\left(\frac{201.1}{530.9}\right) = 0.140\frac{m}{s}.$$

❖ Outlet velocity head:

$$\frac{uB^2}{2g} = \frac{0.37^2}{2(9.81)} = 0.007\ m.$$

❖ Head loss due to fitting:

$$\Delta H = (h_A - h_B) + \frac{uB^2 - uA^2}{2.g} = 0.006 + \frac{0.37^2 - 0.14^2}{2(9.81)} = 0.012\ m.$$

❖ Loss coefficient:

$$K = \frac{\Delta H}{\frac{u^2}{2g}} = \frac{0.012}{0.007} = 1.713$$

❖ Reynold's number:

$$Re = \frac{\rho u_B d_B}{\mu} = \frac{(998.9)(0.37)(0.016)}{1.1076 \times 10^{-3}} = 5350.6.$$

2. *Data Sheet*

Comparative Fluid Flow Measurement Data Sheet

Atmospheric pressure: 676 mmHg

Atmospheric temperature: 16°C

Rotameter (cm)	Flow Rate ($l.m^{-1}$)	h_A (mmH_2O) in A	h_A out B	h_B (mmH_2O) in C	h_B out D	h_E (mmH_2O) E	h_E F	h_F (mmH_2O) G	h_F H	I
10	3.8	306	300	302	304	304	300	300	300	196
20	4.9	304	296	300	300	302	294	296	296	194
30	6.2	304	292	302	302	304	288	290	290	190
40	7.6	306	286	300	300	302	282	286	284	184
50	8.8	306	280	300	300	302	276	280	280	180
60	10.1	308	274	300	302	304	268	276	274	172
70	11.6	310	268	302	304	306	260	268	266	164
80	13	314	260	304	306	308	250	260	258	158
90	14.2	318	254	306	306	310	242	252	252	150
100	15.7	320	246	308	308	314	230	244	240	138
120	18.7	332	226	314	316	322	204	222	220	114
140	21.7	348	202	322	324	332	174	198	194	86
160	24.9	362	176	334	336	346	136	168	164	54
180	28.2	382	144	348	352	364	94	134	130	14

Instructor signature: [illegible] 4/4/2023

Date:

The University of Jordan

School of Engineering

Department of Chemical Engineering

Chemical Engineering Laboratory I (0915361)

Section Number: (2)

Experiment Number: (5)

Experiment Title: ‘shell and tube heat exchanger’.

Report Type: Short Report

Instructor: Prof. Khaled Rawajfeh

Date of Performance: 28th March 2023.

Date of Submission: 4th April 2023.

Abstract:

Heat can move from one fluid to another without mixing or coming into touch when it passes through a heat exchanger. Due to the co-current and counter-current flow patterns involved in this experiment, an equipment like Shell and Tube Heat Exchanger is used. In this experiment, it was found that increasing the flow rate of cold fluid, increase the overall heat transfer coefficient, the counter current flow was more efficient than the co current flow because in a counter-current heat exchanger, there is always a significant and uniform temperature difference not like the co-current flow that showed some sort of disturbance and irregularity during increasing the flow rate.

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Results:

Result for co-current:

Table (1): Raw data for co-current flow in shell and tube heat exchanger

hot water flow rete (l/min)	T_1 Hot In (℃)	T_2 Hot Out (℃)	T_3 cold in (℃)	T_4 Cold out (℃)	Hot W flowrate (m^3/s)	Cold W flowrate (l/min)	Cold W flowrate (m^3/s)
2	50.5	46.3	14.2	18.4	0.00003	1.62	0.000027
3	49.9	46.5	14.3	19.4	0.00005	1.61	0.000027
4	50	47.1	14.3	20.1	0.00007	1.59	0.000027
5	50	47.5	14.7	20.1	0.00008	1.57	0.000026

Table (2: A): parameters for co-current flow in shell and tube heat exchanger

Cold Stream Average Temp (k)	Density Cold (kg/m3)	C_P Cold (J/kg. k)	Cold Mass flowrate (kg/s)	Hot stream Average Temp (K)	Density Hot (kg/m3)	C_P Hot (J/kg. k)	Hot mass flowrate (kg/s)
289.45	998.89	4184.55	0.027	321.55	988.75	4180.62	0.03
290.00	998.80	4184.00	0.027	321.35	988.84	4180.54	0.05
290.35	998.74	4183.79	0.026	321.70	988.68	4180.68	0.07
290.55	998.71	4183.67	0.026	321.90	988.59	4180.67	0.08

Table (2: B): parameters for co-current flow in shell and tube heat exchanger

Q_e (W)	Q_a (W)	Q_l (w)	ΔT_1	ΔT_2	TLMTD	U (W/m2. k)
578.70	474.00	104.70	36.30	27.90	31.92	996.27
702.76	571.89	130.87	35.60	27.10	31.16	1239.31
799.12	642.24	156.88	35.70	27.00	31.15	1409.65
861.04	590.39	270.65	35.30	27.40	31.18	1517.14

Table (2: C): parameters for co-current flow in shell and tube heat exchanger

ΔT cold	ΔT hot	ΔT max	η cold %	η hot %	η exchanger %	η overall %
4.20	4.20	36.30	11.57	11.57	11.57	81.91
5.10	3.40	35.60	14.33	9.55	11.94	81.38
5.80	2.90	35.70	16.25	8.12	12.18	80.37
5.40	2.50	35.30	15.30	7.08	11.19	68.57

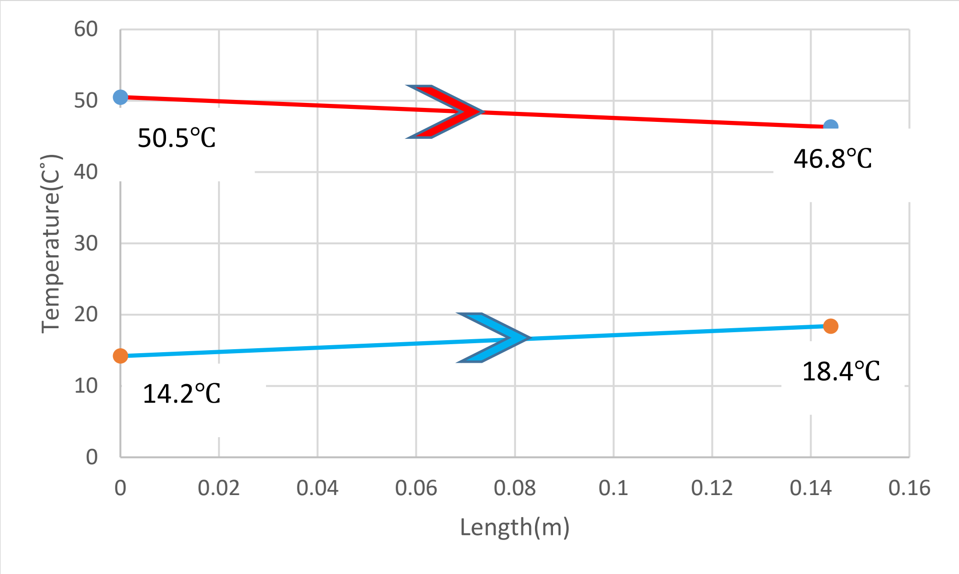

Figure (1): Temperature profile in co-current shell and tube

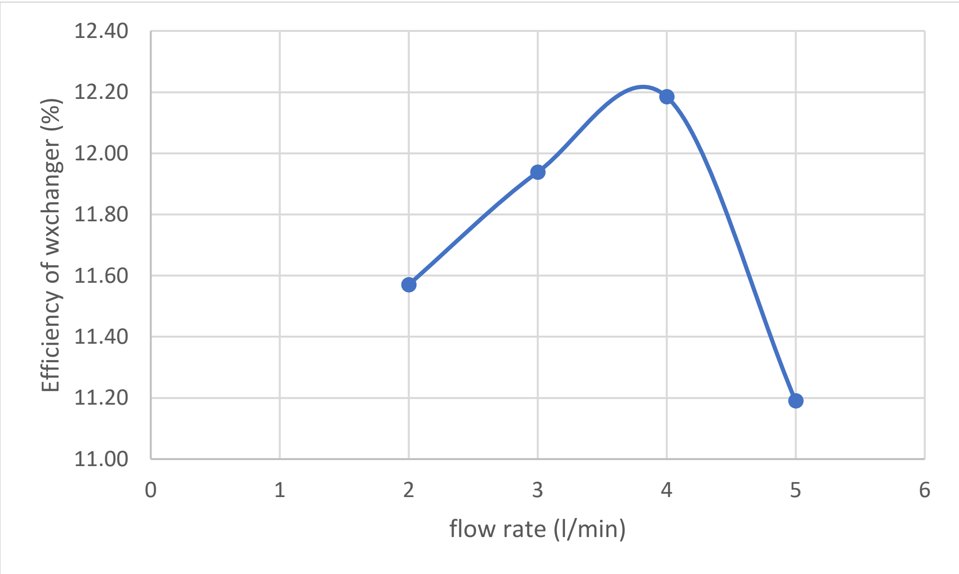

Figure (2): Efficiency VS hot flow in co-current shell and tube

Result for counter- current:

Table (3): Raw data for counter current flow in shell and tube heat exchanger

hot water flow rete (l/min)	T_1 Hot In (℃)	T_2 Hot Out (℃)	T_3 cold in (℃)	T_4 Cold out (℃)	Hot W flowrate (m^3/s)	Cold W flowrate (l/min)	Cold W flowrate (m^3/s)
2	49.7	45.8	15.4	20.3	0.00003	1.13	0.000019
3	50	46.8	15.5	21.4	0.00005	1.17	0.000020
4	49.9	47.1	15.6	22.2	0.00007	1.20	0.000020
5	49.9	47.5	15.7	23	0.00008	1.13	0.000019

Table (4: A): parameters for counter-current flow in shell and tube heat exchanger

Cold Stream Average Temp (k)	Density Cold (kg/m3)	C_P Cold (J/kg. k)	Cold Mass flowrate (kg/s)	Hot stream Average Temp (K)	Density Hot (kg/m3)	C_P Hot (J/kg. k)	Hot mass flowrate (kg/s)
291.00	998.62	4183.24	0.0188	320.90	989.04	4180.36	0.03
291.60	998.51	4182.82	0.0195	321.55	988.75	4180.62	0.05
292.05	998.43	4182.52	0.0200	321.65	988.71	4180.66	0.07
292.50	998.34	4182.25	0.0188	321.85	988.62	4180.74	0.08

Table (4: B): parameters for counter-current flow in shell and tube heat exchanger

Q_e (W)	Q_a (W)	Q_l (w)	ΔT_1	ΔT_2	TLMTD	U (W/m2. k)
537.49	385.51	151.98	34.30	25.50	29.68	994.93
661.37	480.52	180.86	34.50	25.40	29.72	1222.80
771.58	551.23	220.35	34.30	24.90	29.35	1444.47
826.63	574.04	252.60	34.20	24.50	29.08	1561.83

Table (4: C): parameters for counter-current flow in shell and tube heat exchanger

ΔT cold	ΔT hot	ΔT max	η cold %	η hot %	η exchanger %	η overall %
4.90	3.90	34.30	14.29	11.37	12.83	71.72
5.90	3.20	34.50	17.10	9.28	13.19	72.65
6.60	2.80	34.30	19.24	8.16	13.70	71.44
7.30	2.40	34.20	21.35	7.02	14.18	69.44

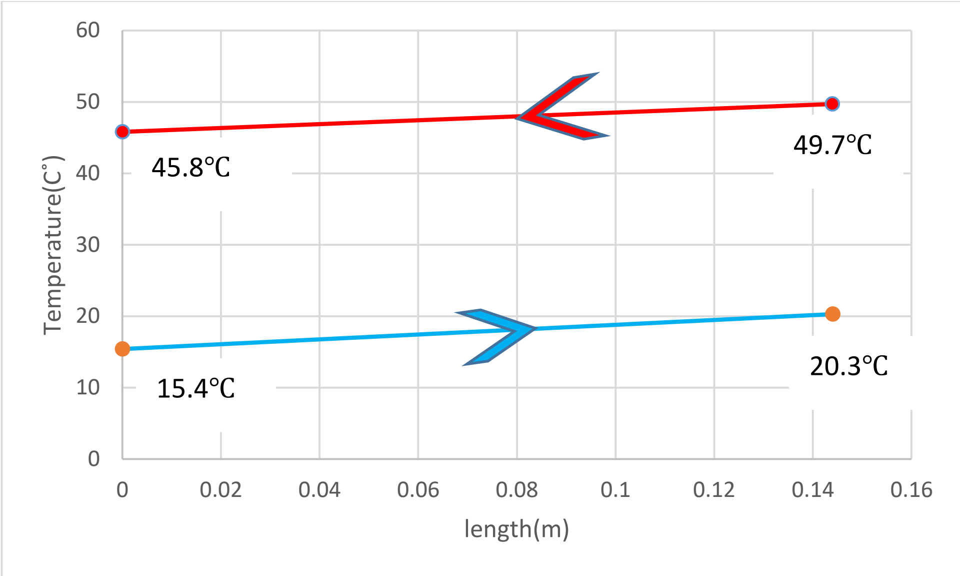

Figure (3): Temperature profile in counter-current shell and tube

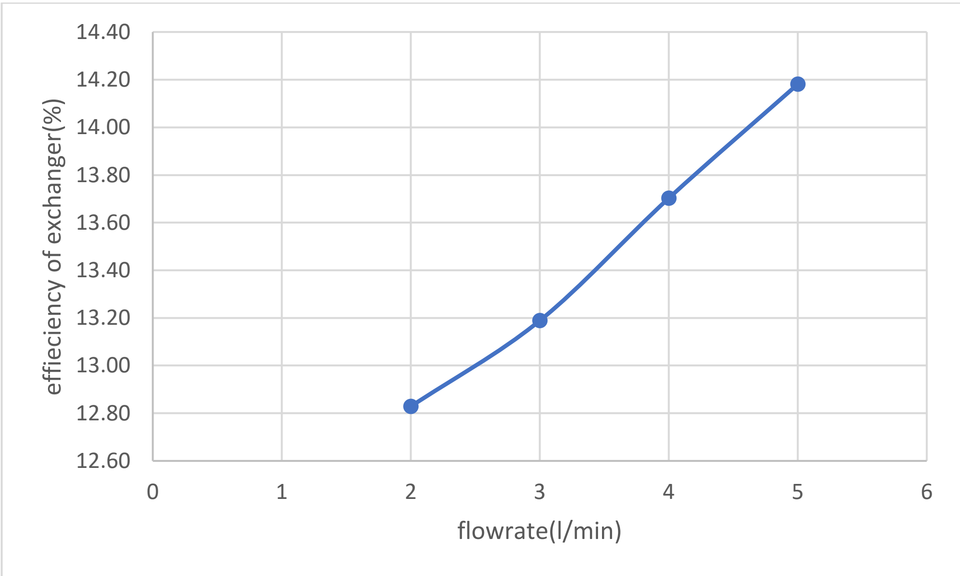

Figure (4): Efficiency VS hot flow in counter-current shell and tube

Discussion:

The trial was conducted to learn how parallel flow and counter flow heat exchangers function and to look into how fluid temperature affects the efficiency of these heat exchangers. Based on the results, it was found that the co-current flow, the efficiency of the device was increasing at flow rate = 4 L/s and then it started decreasing. But the counter current flow showed otherwise, it was always increasing when the flow rate was increase. This will lead to that the counter current flow is more efficient than the co-current flow as shown in figure (2) and (4).

Because the required surface will decrease as LMTD rises, counter-current heat exchangers are preferable to co-current ones in terms of cost and efficiency. The force of heat transfer was another factor in the temperature differential. In a counter-current heat exchanger, there is always a significant and uniform temperature difference; however, in co-current heat exchangers, this temperature difference is very small at the exchanger's fluid-exiting end.

As shown in table (2:B), raising the flow rate of cold fluid results in an increase in the overall heat transfer coefficient because doing so raises the Reynolds number, which raises the overall heat transfer coefficient. Some source of errors may be affected on the results such as systematic and personal error.

Conclusion:

The following conclusions from this experiment using co-current and counter-current water flow in shell and tube heat exchangers:

1) The heat absorbed or emitted by the water rises with increasing the mass flow rate of water in the two kinds and the amount of heat loss from the heated water is less than the amount of heat gain from the cold water because of the heat loss to the surrounding.

2) Heat transfer in turbulent flow is more efficient than in laminar flow because the overall heat transfer coefficient (U) rises with flow rate. Because of the higher heat flow that will be produced, counter current heat exchangers are more effective than co-current ones.

References used:

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Appendix:

Sample of calculations for co current shell and tube heat exchanger:

Table (1): first row

- Hot Water Flowrate $[m^3/\text{s}] = \frac{\text{Hot Water Flowrate [l/min]}}{1000\times60} = \frac{2}{1000\times60} = 0.00003[\frac{m^3}{s}]$

Table (2: A): first row

- cold stream average temperature $= \left(\frac{T_3+T_4}{2}\right) + 273.15 = \left(\frac{14.2+18.4}{2}\right) + 273.15 = 289.45K$
- cold mass flowrate $(m_c) = \rho_{cold} \times V_{cold} = 998.89 \times 0.000027 = 0.027[\frac{\text{Kg}}{\text{s}}]$
- hot mass flowrate$(m_h) = \rho_{hot} \times V_{hot} = 988.75 \times 0.00003 = 0.03[\frac{\text{Kg}}{\text{s}}]$

Table (2: B): first row

- heat power emitted from hot fluid $(Q_e) = m_h \times cC_{phot} \times (T_1 - T_2) = 0.03 \times 4180.62 \times (50.5 - 46.3) = 578.70\text{W}$
- hear power absorbed by cold fluid $(Q_a) = m_c \times C_{pcold} \times (T_4 - T_3) = 0.027 \times 4184.55 \times (18.4 - 14.2) = 474.00\text{W}$
- heat power lost or gained $(Q_l) = Q_e - Q_a = 578.70 - 474.00 = 104.70\text{W}$
- $\Delta T_1 = T_1 - T_3 = 50.5 - 14.2 = 36.3℃$
- $\Delta T_2 = T_2 - T_4 = 46.3 - 18.4 = 27.9℃$
- The mean temperature difference between the two fluids $(L_{MTD}) = \frac{\Delta T_1 - \Delta T_2}{ln\frac{\Delta T_1}{\Delta T_2}} = \frac{36.3-27.9}{ln\frac{36.3}{27.9}} =$

  31.9
- Arithmetic mean diameter $(d_m) = \frac{d_{inner\ tube\ inside\ diameter} + d_{inner\ tube\ outside\ diameter}}{2} = \frac{0.00635+0.00515}{2} = 0.00575m$
- Heat transfer area required in the exchanger (A)= $\pi \times d_m \times 1.008 = \pi \times 0.00575 \times 1.008$= 0.0182$m^2$
- The overall heat transfer coefficient (U)= $\frac{Q_e}{A\times L_{MTD}} = \frac{578.70}{0.0182\times31.9} = 996.23$ W/m2.K

Table (2: C): first row

- $\Delta T_{cold} = T_4 - T_3 = 18.4 - 14.2 = 4.2℃$
- $\Delta T_{hot} = T_1 - T_2 = 50.5 - 46.3 = 4.2℃$
- $\Delta T_{max} = T_1 - T_3 = 50.5 - 14.2 = 36.3℃$

- Thermal efficiency for the cold side $(\eta_{cold}) = \frac{\Delta T_{cold}}{\Delta T_{max}} = \frac{4.2}{36.3} = 0.1157 \times 100 = 11.57\%$

- Thermal efficiency for the hot side $(\eta_{hot}) = \frac{\Delta T_{hot}}{\Delta T_{max}} = \frac{4.2}{36.3} = 0.1157 \times 100 = 11.57\%$
- Efficiency of the exchanger $(\eta_{exchanger}) = \frac{\eta_{cold+}\eta_{hot}}{2} = \frac{11.57+5=11.57}{2} = 11.57\%$
- Overall Efficiency$(\,\eta_{overall}) = \frac{Q_a}{Q_e} = \frac{474.00}{578.70} = 81.91\ \%$
-

Sample of calculations for counter current shell and tube heat exchanger:

Table (2: A): first row

- cold stream average temperature $= \left(\frac{T_3+T_4}{2}\right) + 273.15 = \left(\frac{15.4+20.3}{2}\right) + 273.15 = 291.15K$
- hot stream average temperature $= \left(\frac{T_1+T_2}{2}\right) + 273.15 = \left(\frac{49.7+45.8}{2}\right) + 273.15 = 320.90K$

Table (2: B): first row

- heat power emitted from hot fluid $(Q_e) = m_h \times C_{phot} \times (T_1 - T_2) = 0.03 \times 4180.36 \times (49.7 - 45.8) = 537.49\text{W}$
- hear power absorbed by cold fluid $(Q_a) = m_c \times C_{pcold} \times (T_4 - T_3) = 0.02 \times 4183.24 \times (20.3 - 15.4) = 385.51\text{W}$
- heat power lost or gained $(Q_l) = Q_e - Q_a = 537.49 - 385.51 = 151.98\text{W}$
- $\Delta T_1 = T_1 - T_3 = 49.7 - 15.4 = 34.30℃$
- $\Delta T_2 = T_2 - T_4 = 45.8 - 20.3 = 25.50℃$
- The mean temperature difference between the two fluids $(L_{MTD}) = \frac{\Delta T_1 - \Delta T_2}{ln\frac{\Delta T_1}{\Delta T_2}} = \frac{34.30-25.50}{ln\frac{34.30}{25.50}} =$

  29.68
- Arithmetic mean diameter $(d_m) = \frac{d_{inner\ tube\ inside\ diameter} + d_{inner\ tube\ outside\ diameter}}{2} = \frac{0.00635+0.00515}{2} = 0.00575m$
- Heat transfer area required in the exchanger (A)= $\pi \times d_m \times 1.008 = \pi \times 0.00575 \times 1.008$= $0.0182m^2$
- The overall heat transfer coefficient (U)= $\frac{Q_e}{A \times L_{MTD}} = \frac{537.49}{0.0182 \times 29.68} = 859.0832467$ W/m2.K

Table (2: C): first row

- $\Delta T_{cold} = T_4 - T_3 = 20.3 - 15.4 = 4.90℃$
- $\Delta T_{hot} = T_1 - T_2 = 49.7 - 45.8 = 3.90℃$

- $\Delta T_{max} = T_1 - T_3 = 49.7 - 15.4 = 34.30℃$

- *Thermal efficiency* for the cold side $(\eta_{\text{cold}}) = \frac{\Delta T_{cold}}{\Delta T_{max}} = \frac{4.90}{34.30} = 0.1429 \times 100 = 14.29\%$
- Thermal efficiency for the hot side $(\eta_{hot}) = \frac{\Delta T_{hot}}{\Delta T_{max}} = \frac{3.90}{34.30} = 0.1137 \times 100 = 11.37\%$
- Efficiency of the exchanger $(\eta_{exchanger}) = \frac{\eta_{cold+}\eta_{hot}}{2} = (\frac{14.29+11.37}{2}) \times 100 = 12.83\%$

Overall Efficiency($\eta_{overall}) = \frac{Q_a}{Q_e} = (\frac{385.51}{537.49}) \times 100 = 71.72\%$

The University of Jordan

School of Engineering

Department of Chemical Engineering

Chemical Engineering Laboratory I (0915361)

Section Number: (2)

Experiment Number: (6)

Experiment Title: ‘Concentric tube heat exchanger.

Report Type: Short Report

Instructor: Prof. Khaled Rawajfeh

Date of Performance: 25th April 2023.

Date of Submission: 2nd May 2023.

Abstract:

The main purpose of the concentrated tube heat exchanger was to demonstrate how industrial heat exchangers worked. A heat exchanger is a device made to efficiently transfer heat from one medium to another. The aim of the experiment was to demonstrate the operation of a concentric tube heat exchanger under both co-current and counter-current flow conditions. The ultimate objective was to demonstrate the performance characteristics of a concentric tube heat exchanger operating under counter-current flow conditions, both warm and chilly. According to these objectives, the results showed that countercurrent flow led to more efficient heat exchange, thermal equilibrium was reached, and some degree of heat exchange occurred when an indefinite length of cocurrent concentric heat exchanger was installed.

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Table of Tables:

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Results:

Result for co-current:

Table (1): Specification of the concentric heat exchanger

Heat transfer area(m^2)	0.067
Heat transfer length(m)	1.50
Cold water flow rate(cm^3/min)	1300.00

Table (2): Raw data for concurrent flow

Hot water flow rate (L/min)	Hot water flow rate (m^3/min)	$T_{Hot\ in}$ ($C°$)	$T_{Hot\ out}$ ($C°$)	$T_{Cold\ in}$ ($C°$)	$T_{Cold\ out}$ ($C°$)	Cold water flow rate (cm^3/min)
1000.00	0.0010	58.00	46.00	19.00	28.00	1500.00
2000.00	0.0020	58.00	50.00	19.00	34.00	1500.00
3000.00	0.0030	58.00	51.00	19.00	36.00	1500.00

Table (3): parameters for co-current flow in concentric tube heat exchanger

Hot water mass flow (Kg/s)	ρ for hot water (kg/m^3)	Cp for hot water (J/kg.K)	Cold water (m^3/min)	ρ for cold water (kg/m^3)	Cp for cold water (J/kg.K)	Cold water mass flow (Kg/s)
0.016	987.06	4175.00	0.0015	997.38	4180.00	0.025
0.033	986.19	4176.00	0.0015	996.61	4177.00	0.025
0.049	985.71	41763.00	0.0015	996.34	4176.00	0.025

Table (4): Calculated data for Co-cuurent flow

Qe (W)	Qa (W)	η overall	ΔT1 (K)	ΔT2 (K)	LMTD (K)
824.20	938.031	113.81	39.00	18.00	27.16
1098.22	1561.10	142.15	39.00	16.00	25.81
14408.11	1768.30	12.27	39.00	15.00	25.12

Table (4): Calculated data for Co-cuurent flow (Continued)

Heat transfer area (m^2)	U (over all heat transfer coefficient)	ΔT hot (K)	ΔT cold(K)	ΔT max(K)	η hot%	η c old%	η exchanger %
0.067	452.92	12.00	9.00	39.00	30.77	23.10	26.92
0.067	634.97	8.00	15.00	39.00	20.513	38.46	29.49
0.067	8561.63	7.00	17.00	39.00	17.95	43.59	30.77

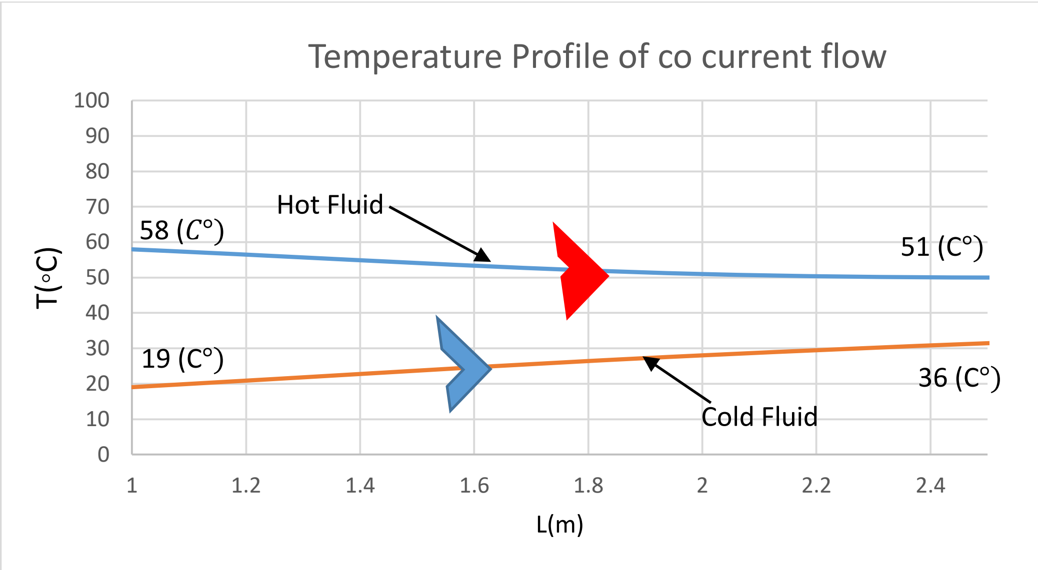

Figure (1): Temperature profile in co-current flow in concentric tube heat exchanger

Result for counter- current:

Table(5): Raw Data for Counter-current flow

Hot water flow rate (L/min)	Hot water flow rate (m^3/min)	$T_{Hot\ in}$ $(C°)$	$T_{Hot\ out}$ $(C°)$	$T_{Cold\ in}$ $(C°)$	$T_{Cold\ out}$ $(C°)$	Cold water flow rate (cm^3/min)
1000.00	0.001	58.00	43.00	18.00	27.50	1500.00
2000.00	0.002	58.00	47.00	18.00	31.00	1500.00
3000.00	0.003	58.00	49.00	18.00	34.00	1500.00

Table (6): Parameters for Counter-current flow

Hot water mass flow (Kg/s)	ρ for hot water (kg/m^3)	Cp for hot water (J/kg.K)	Cold water flow rate (m^3/min)	ρ for cold water(kg/m^3)	Cp for cold water (J/kg.K)	Cold water mass flow (Kg/s)
0.016	987.78	4174.00	0.0015	989.96	4174.50	0.025
0.033	986.82	4175.00	0.0015	988.45	4175.40	0.025
0.049	986.39	4176.00	0.0015	987.06	41760	0.025

Table (7): Calculated data for Counter flow

Qe (W)	Qa (W)	η overall	ΔT1 (K)	ΔT2 (K)	LMTD (K)
1030.76	981.49	95.22	30.5	25.00	27.66
1510.66	1341.33	88.79	27.00	29.00	27.99
1853.62	1648.76	88.95	24.00	31.00	27.35

Table(7) : Calculated data for Counter flow (continued)

Heat transfer area (m^2)	U (over all heat transfer coefficient)	ΔT hot (K)	ΔT cold (K)	ΔT max (K)	η hot%	η cold%	η exchanger%
0.067	556.22	15.00	9.50	40.00	37.50	23.75	30.63
0.067	805.60	11.00	13.00	40.00	27.50	32.50	30.00
0.067	1011.52	9.00	16.00	40.00	22.5	40.00	31.25

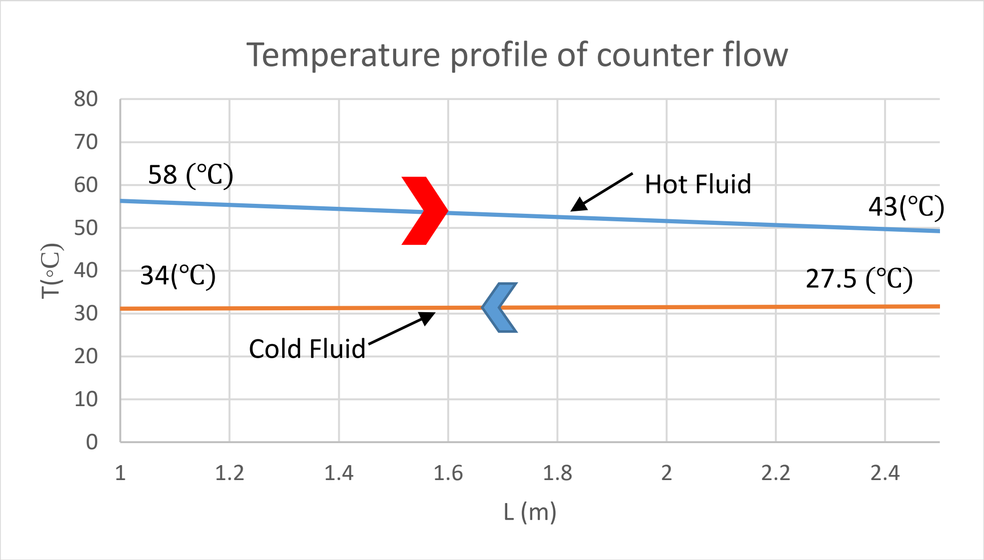

Figure (3): Temperature profile in counter-current flow concentric tube heat exchanger

Discussion:

Concentric tube heat exchangers are being used in the experiment to evaluate the operation of the heat exchanger at varied flow rates. The experiment is divided into two categories: parallel flow and counter flow. Water is employed as the experiment's medium. The hot water inlet temperature is maintained at 58°C, and the cold water flow rate is maintained at 1500 cm3/min. After that, the temperature of both hot and cold water is noted.

Counter-current Heat exchangers are preferable to Co-current because they would require less surface area with a higher LMTD, which would result in lower costs, another factor that caused heat transfer was the temperature difference.

As can be seen, increasing the flow rate of a hot fluid causes an increase in the overall heat transfer coefficient since doing so raises the Reynolds number, which in turn raises the overall heat transfer coefficient.

Increasing hot water flow rate leads to increasing heat power emitted and Increasing cold water flow rate leads to increasing heat power absorbed.

Some of error may effect in the results such as systematic error and personal error during reading from the device

Conclusion:

It is important to understand the reasons why counter-current processes are more efficient than co-current processes. Temperature profiles for both counter-current and co-current concentric heat exchangers were analyzed in figures (insert figure numbers) which demonstrated that driving forces in co-current processes tend to decrease as the length of the concentric heat exchanger increases. If an infinitely long co-current concentric heat exchanger is used, thermal equilibrium will eventually be reached and some degree of heat exchange will occur. However, in contrast to co-current processes, figure (insert figure number) shows that the driving force (temperature difference) for counter-current concentric heat exchangers remains at its maximum throughout the length of the exchanger, resulting in a higher heat exchange rate and a more efficient process.

References used:

1. Incropera, F. P. (2013). *Fundamentals of heat and mass transfer*. Wiley.

Appendix:

A) Sample of calculations

Co-current flow: (taking the first row of each table)

- Log-mean temperature difference:

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln(\frac{\Delta T_1}{\Delta T_2})} = \frac{39-18}{\ln(\frac{39}{18})} = 27.160 \text{ K.}$$

- Overall heat transfer coefficient:

$$U = \frac{Q_e}{A\,(LMTD)} = \frac{824.195}{(0.067)(27.160)} = 452.924\ \frac{W}{m^2\ K}.$$

- Thermal efficiency for the cold side:

$$\eta_{cold} = \frac{\Delta T_{cold}}{\Delta T_{max}} = \frac{T_{cold,out} - T_{cold,in}}{T_{hot,in} - T_{cold,in}} = \frac{28-19}{58-19}(100\%) = 23.07\%$$

- Thermal efficiency for the hot side:

$$\eta_{hot} = \frac{\Delta T_{hot}}{\Delta T_{max}} = \frac{T_{hot,in} - T_{hot,out}}{T_{hot,in} - T_{cold.in}} = \frac{58-46}{58-19}(100\%) = 30.769\%$$

- Efficiency of the exchanger:

$$\eta_{exchanger} = \frac{\eta_{hot} + \eta_{cold}}{2} = \frac{30.769+23.07}{2}(100\%) = 26.923\%$$

- Overall efficiency:

$$\eta_{overall} = \frac{Q_a}{Q_e} = \frac{938.031}{824.195}(100\%) = 113.81\%$$

Counter-current flow: (taking the first row of each table)

- Log-mean temperature difference:

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln(\frac{\Delta T_1}{\Delta T_2})} = \frac{30.5 - 25}{\ln(\frac{30.5}{25})} = 27.659\ K$$

- Overall heat transfer coefficient:

$$U = \frac{Q_e}{A\ (LMTD)} = \frac{1030.764}{(0.067)(27.659)} = 556.22\ \frac{W}{m^2\ K}$$

- Thermal efficiency for the cold side:

$$\eta_{cold} = \frac{\Delta T_{cold}}{\Delta T_{max}} = \frac{T_{cold,out} - T_{cold,in}}{T_{hot,in} - T_{cold.in}} = \frac{27.5 - 18}{58 - 18}(100\%) = 23.75\%$$

- Thermal efficiency for the hot side:

$$\eta_{hot} = \frac{\Delta T_{hot}}{\Delta T_{max}} = \frac{T_{hot,in} - T_{hot,out}}{T_{hot,in} - T_{cold,in}} = \frac{58 - 43}{58 - 18}(100\%) = 37.5\%$$

- Efficiency of the exchanger:

$$\eta_{exchanger} = \frac{\eta_{hot} + \eta_{cold}}{2} = \frac{37.5 + 23.75}{2}(100\%) = 30.625\%$$

- Overall efficiency:

$$\eta_{overall} = \frac{Q_a}{Q_e} = \frac{981.485}{1030.764}(100\%) = 95.219\%$$

B) Data sheet

Concentric Tube Heat Exchanger Data Sheet

Heat transfer area: 0.067 m^2

Heat transfer length: 1.5m

Cold water flow rate: 1500 cm^3/min

Co-current flow:

Hot water flow rate (cm^3/min)	$T_{hot,in}$ (°C)	$T_{hot,out}$ (°C)	$T_{cold,in}$ (°C)	$T_{cold,out}$ (°C)
1000	58	46	19	28
2000	58	50	19	34
3000	58	51	19	36

Counter-current flow:

Hot water flow rate (cm^3/min)	$T_{hot,in}$ (°C)	$T_{hot,out}$ (°C)	$T_{cold,in}$ (°C)	$T_{cold,out}$ (°C)
1000	58	43	18	27.5
2000	58	47	18	31
3000	58	49	18	34

Instructor signature:	
Date: 18/4/2023	

The University of Jordan

School of Engineering

Department of Chemical Engineering

Chemical Engineering Laboratory I (0915361)

Section Number: (2)

Experiment Number: (7)

Experiment Title: ‘Heat Conduction’.

Report Type: Short Report

Instructor: Prof. Khaled Rawajfeh

Date of Performance: 21st March, 2023.

Date of Submission: 28th March, 2023.

Abstract:

In this experiment, the total heat transfer coefficient and the thermal conductivity of a brass were determined using Fourier's law. The temperature gradients at various heat sources were investigated using eight thermocouples. It was demonstrated that the thermal conductivity slightly varies with temperature and the temperature gradient widens with increasing heat input. When heat is transferred linearly along a uniform planar wall by conduction, different voltages were applied, which had an impact on the temperature distribution and thermal conductivity. Water was used as a coolant and a heat absorber.

The result:

The thermal conductivity (k) decreases with increasing heat input. The average experimental value of k =227.42 W/m.k

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Results:

Table 1 : Voltage = 60.00 V

Date	Time (s)	T1	T2	T3	T4	T5	T6	T7	T8	Input Volts	Input Amps	Water flow
3/21/2023	1205.00	22.89	22.40	21.52	20.03	18.96	17.64	17.68	16.80	63.00	0.067	43.70

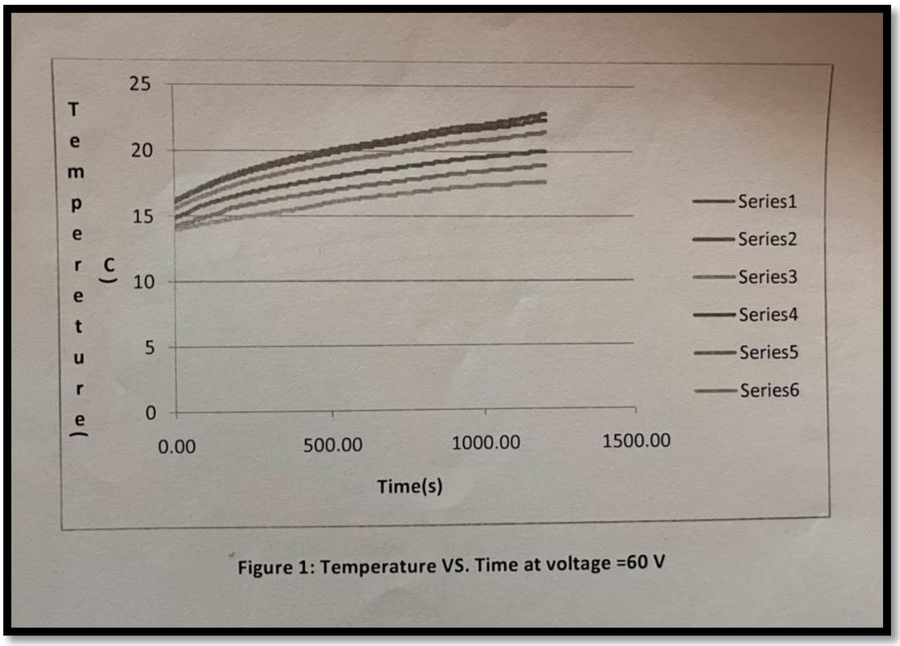

Figure 1: Temperature VS. Time at voltage =60 V

Table 2: Voltage = 80.00 V

Date	Time (s)	T1	T2	T3	T4	T5	T6	T7	T8	Input Volts	Input Amps	Water flow
3/21/2023	**1220.00**	**29.03**	**28.16**	**26.67**	**24.07**	**22.25**	**19.81**	**19.22**	**17.97**	**75.9**	**0.081**	**43.88**

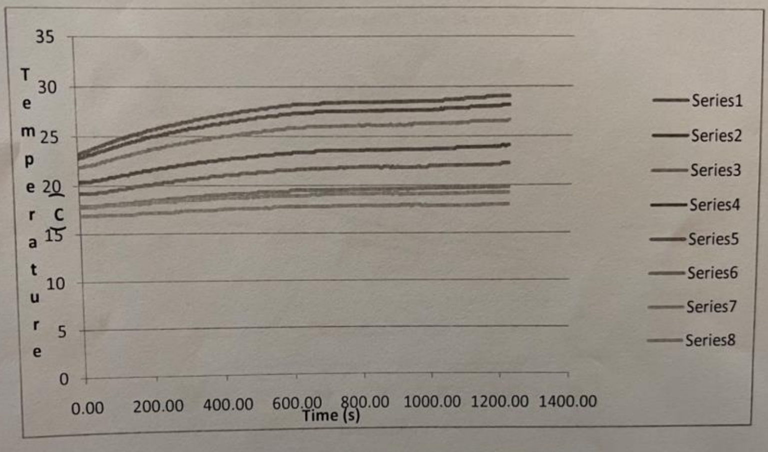

Figure 2 : Temperature VS. Time at voltage =80V

Table 3: Voltage = 100.00 V

Date	Time (s)	T1	T2	T3	T4	T5	T6	T7	T8	Input Volts	Input Amps	Water flow
3/21/2023	1200.00	40.73	36.68	36.12	31.15	28.09	23.66	21.82	19.58	101.34	0.11	40.86

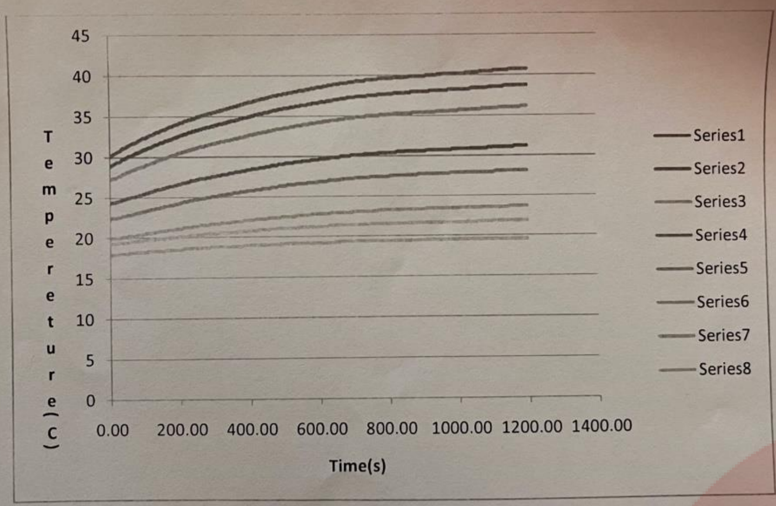

Figure 3: Temperature VS. Time at voltage =100V

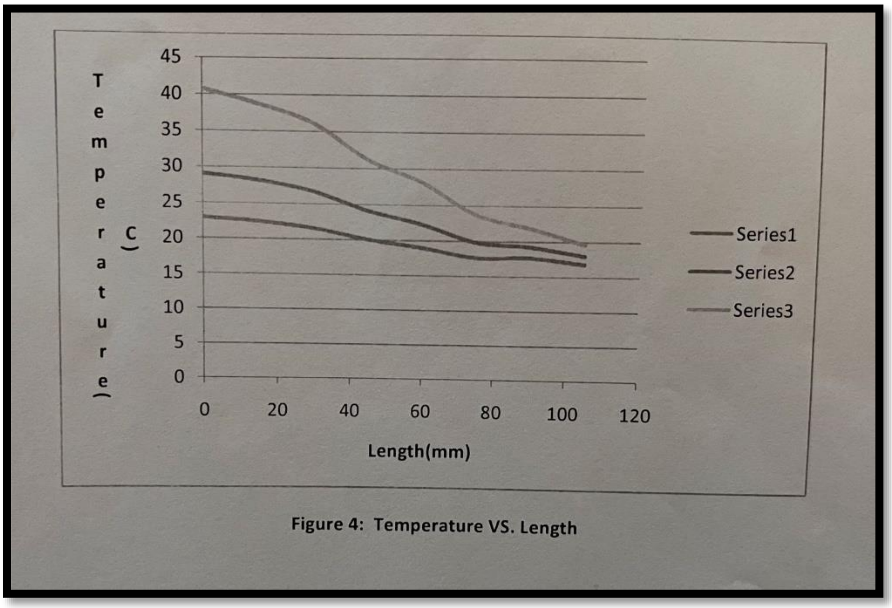

Figure 4: Temperature VS. Length

Table 4: Calculated data for heat conduction at three different voltages

Voltage (volt)	Current (A)	Area (m^2)	T_4 (◦C)	T_5 (◦C)	ΔT (◦C)	Q (W)	K (W/m.k)
63.00	0.067	0.00049	20.03	18.96	-1.07	4.22	241.52
79.90	0.081	0.00049	24.07	22.25	-1.82	6.47	217.71
101.34	0.11	0.00049	31.15	28.09	-3.06	11.15	223.037

Brass theoretical heat conductivity of 109 (W/m.k)
Average Experimental K-value= 227.42 W/m.K
%Error=108.64%

Discussion:

By conducting this experiment, it can be seen that by varying the voltage, the heat input is changed accordingly and so is the thermal conductivity (K). Using Fourier's fundamental law for heat conduction, the values of (K) are calculated for a brass specimen (which is a mixture of zinc and copper). The temperatures T_4 and T_5 are used in calculation because these are the temperatures that measure the difference between the heating section and the cooling section of the unit.

For uniform conduction through a uniform bar, at constant area, there is a direct increasing relationship between the heat input and increased voltage (60V, 80V, 100V). As a result, the thermal conductivity (K) decreases with increasing heat input. The average experimental value of K= 227.42 W/m.K .

Experimental values of (K) may exhibit a respectable amount of deviation from the theoretical value (K_{brass}=109 W/m.K), due to instrumental errors, such as defects in the insulation layer of the unit which results in heat lost from the system.

Conclusion:

In the experiment on linear heat conduction. It is evident that increasing voltage raises heat input/duty (Q), and that heat input is inversely correlated with temperature. The temperature gradient is wider as the temperature rises. Referring to table (4), the experimental values deviated from the theoretical value due to both experimental errors, such as heat losses to the environment through the system, and personal errors, such as inaccurate data reading and calculation errors, etc. Brass has a theoretical thermal conductivity of 109 (W/m.k). It is recommended to turn off the heater by lowering the voltage to zero and to wait until it has cooled down completely before turning off the water.

References used:

1. Celisa (2022). Fundamentals of Thermal Resistance. Retrieved December 25th , 2022, from https://celsiainc.com/heat-sink-blog/fundamentals-of-thermal-resistance.
2. Christie J. Geankolis, “Transport Processes and Unit Operations”, 3rd Edition, Perntic Hall.
3. Incropera's Principles of Heat and Mass Transfer, 8th Edition, Global Edition.

Sample of Calculations:

For trial 1 :

- $\Delta T = T_5 - T_4$

ΔT =18.96-20.03 = -1.07K

- $Q = I \times V$

Q =0.067×63 = 4.221W

- $\text{Area} = \frac{\pi}{4} D^2$

$\text{Area} = \frac{\pi}{4} \times (0.025)^2$

A = 0.00049 m^2

- $K = \frac{-QL}{A\Delta T}$

$= \frac{-4.221 \times 0.03}{0.00049 \times (-1.07)}$

K =241.52 W/m.K

The UNIVERSITY OF JORDAN

FACULTY OF ENGINEERING AND TECHNOLOGY

SCHOOL OF ENGINEERING

DEPT.OF CHEMICAL ENGINEERING

Chemical Engineering laboratory 1 (0915361)

Section no. (2)

Experiment Number (9)

Free and forced convection

Short report

Instructor's Name: Prof. Dr. Khaled Rawajfeh &

Eng. Aya Ghanem

Date of Submission: 28th– May-2023

1. Abstract

The three modes of heat transfer are convection, conduction, and radiation. Convection is the process of transferring heat by moving a fluid (either a liquid or a gas). Free convection and forced convection are the two basic ways it can happen. In this experiment, both free and forced convection will be used to assess the heat transfer coefficient. Free convection is the term used to describe fluid that is naturally moving due to buoyancy forces brought on by temperature variations. Forced convection is when fluid is forced to travel by external devices like fans and pumps.

main results:

The heat transfer coefficient for free convection is measured as 20.99W/m^2.K . In forced convection, it is measured as 16.4 W/m^2.K .

A direct relation exists between power and temperature difference for the pin surface.

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2. Results

Table (1) :Free Convection

	Power (W)	T_2 © surface temp	T_1 © inlet temp	Difference
	25	66.3	22.4	43.9
	0	20.8	20.6	0.2
Average	**12.5**			**22.05**

Table (2) :Forced Convection

Air Velocity (m/s)	T_2 © surafce temp	T_1 © inlet temp	Difference (T_2-T_1)
1	54.6	22	32.6
1.5	45.6	22	23.6
2	41.7	22.1	19.6
2.5	38.7	22	16.7
3	37.5	21.9	15.6

Table (3) :HTC (P=25 W, V=3 m/s)

Duct Position (mm)	T_1 © inlet temp	T_2 ©surface temp	T_3 © probe temp	T_2-T_1	T3-T_1
0.5	22.1	37.4	23.5	15.3	1.4
1	22.1	37.4	23.6	15.3	1.5
1.5	22.0	37.3	23.4	15.3	1.4
2	22.0	37.3	23.4	15.3	1.4
2.5	22.0	37.3	23.1	15.3	1.1
3	22.1	37.3	23.0	15.2	0.9
3.5	22.1	37.3	22.8	15.2	0.7
4	22.0	37.3	22.6	15.3	0.6
4.5	22.1	37.3	22.5	15.2	0.4
5	22.2	37.4	22.4	15.2	0.2
5.5	22.1	37.5	22.1	15.4	0
6	22.1	37.5	22.1	15.4	0
6.5	22.1	37.6	22.2	15.5	0.1
7	22.1	37.6	22.2	15.5	0.1
7.5	22.3	37.6	22.2	15.3	-0.1
8	22.3	37.6	22.2	15.3	-0.1
T average	22.1	37.4	22.7	15.3	0.6

Table (4) : Heat transfer coefficient and LMTD for pinned surface

	LMTD ◦C	Heat transfer coefficient W/m2.K
Free Convection	22.05	20.99
Forced Convection	56.4	16.4

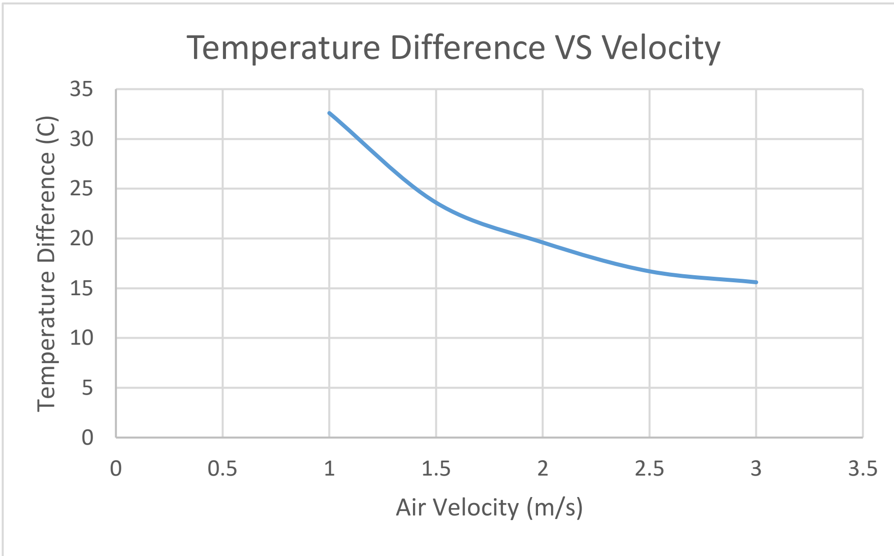

Figure (1): Temperature Difference VS Velocity

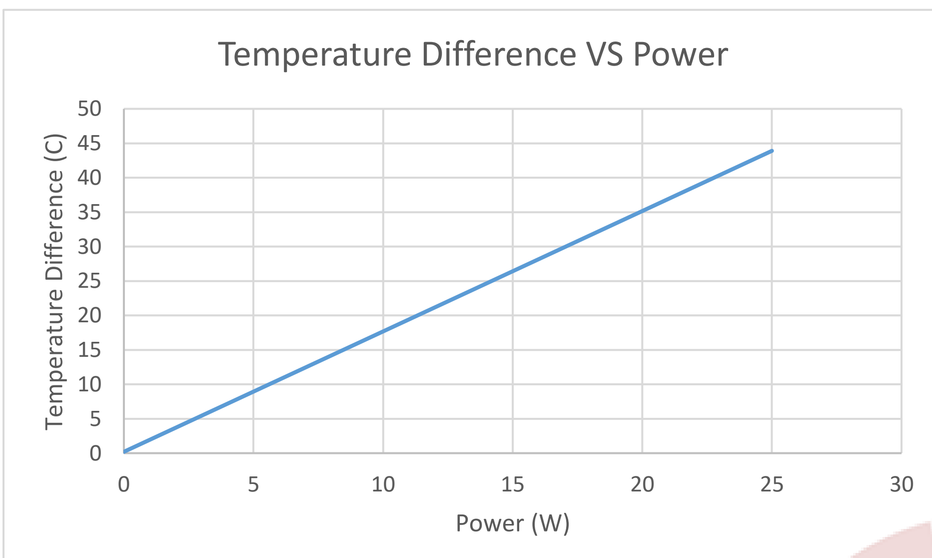

Figure (2):Temperature Difference VS Power

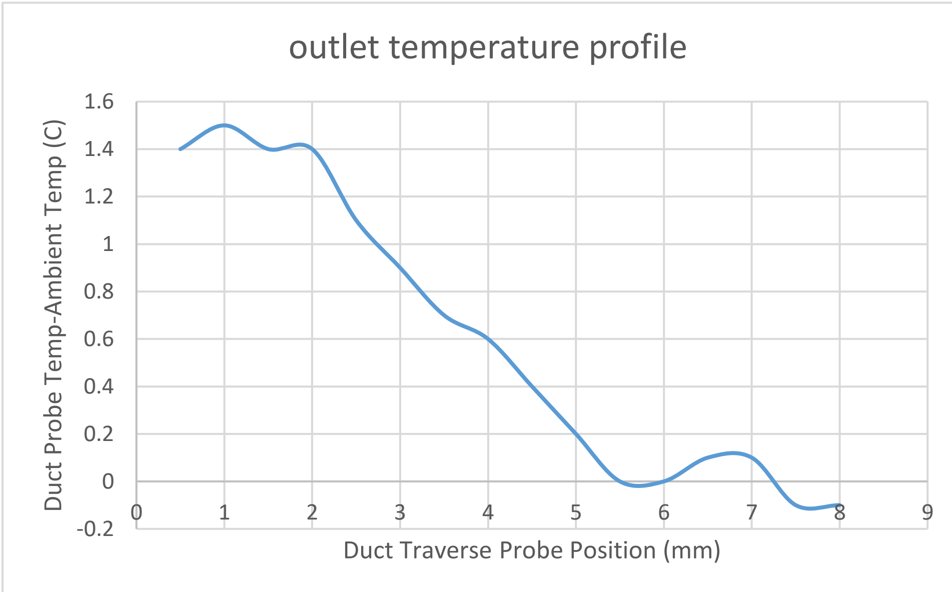

Figure (3) :outlet temperature profile

3. Discussion

Convection is a fundamental process in heat transfer that involves the movement of fluid to transfer heat.

Since the surface area of a pinned surface is greater than that of a flat plate or a finned plate, it facilitates a more effective transfer of heat.

In this experiment, it is determined experimentally that the free convection heat transfer coefficient is 20.99 W/m^2·K. Forced convection is found to have a heat transfer coefficient of 16.44 W/m^2·K.

The convection heat transfer coefficient is usually higher in forced convection because the heat transfer coefficient depends on the fluid velocity, and forced convection involves higher fluid velocities. If difference is found it will be due to either the experimental error or error in data collection.

In forced convection, hot air is moved by another air that has velocity and comes from the fan. As seen in Figure (1), increased air velocity reduces the temperature difference between the inlet and surface, resulting in the air temperature passing through the pins not significantly increasing.

Figure (2) show the direct relation between power and temperature difference (for pin surface).

The pins temperature decrease moving far from the base which means the heat transfer also decreases.

4. Conclusions

In this experiment, the forced heat transfer convection is greater than the free heat transfer convection. This is due to the fact that the heat transfer coefficient for forced convection is higher than the free convection but, in this experiment, it was found the exact opposite this because of non-availability of data in the free convection part and systematic error from the device. In figure (1), as the velocity increases, the driving force (temperature difference) decreases. Also, it is concluded that as the power input increases, the driving force will also increase because the power input indicates the heat input, so according to Newton's law of cooling, if the heat input increases, this will lead to a higher driving force.

5. References

1. Basic Engineering Thermodynamics, by Rayner Joel, published by Longman ISBN 0-582-25629-1
2. Engineering Thermodynamics, by G.F.C Rogers and Y.R Mayhew, published by Longman, ISBN 0-582-02704-7
3. Heat transfer, by J.P Holman, published by McGraw Hill, ISBN 978-0-07- 352936-3

6. Appendices

1. *Sample of Calculations:*

<u>Taking the first raw in each table</u>

<u>Table 1 : free convection</u>

- Temperature difference (ΔT) = $T_2(°C) - T_1(°C)$

$$= 66.3\ °C - 22.4\ °C = 43.9\ °C$$

<u>Table 2 : forced convection</u>

- Temperature difference (ΔT) = $T_2(°C) - T_1(°C)$

$$= 54.6\ °C - 22\ °C = 32.6\ °C$$

<u>Table 3 : HTC (P=25 W, V=3 m/s)</u>

- Temperature difference (ΔT) = $T_2(°C) - T_1(°C)$

$$= 37.4\ °C - 22.1\ °C = 15.3\ °C$$

<u>Same procedure for ΔT between T_3 and T_1</u>

- Log mean temperature difference for forced convection= $\frac{T_{out\ avg} - T_{in\ avg}}{\log(\frac{(T_{s\ avg} - T_{out\ avg})}{T_{s\ avg} - T_{in\ avg}})}$

$$\frac{(0.6-22.1)}{\log((\frac{15.3}{(37.4-0.6)}))} = 56.4\ ℃$$

- Heat transfer coefficient for forced convection (h_{fc}) = $\frac{Q}{A*LMTD}$

$$= \frac{25\ W}{0.027\ m^2 * 56.4℃} = 16.42\ \text{W}/m^2.℃$$

- Log mean temperature difference for free convection= $\frac{T_{out\ avg} - T_{in\ avg}}{\log(\frac{(T_{s\ avg} - T_{out\ avg})}{T_{s\ avg} - T_{in\ avg}})}$ = 22.05
- Heat transfer coefficient for free convection (h_{fc}) = $\frac{Q}{A*LMTD}$
- $\frac{12.5\ W}{0.027\ m^2 * 22.05\ ℃}$ = 20.99 W/m^2.K

2. Data Sheet

P=25 W
air flow = 0.1 w/s

23/5/2023

Table 1 : result table for flat plate

Heat Transfer Surface: flat plate			
	T_2	T_1	
Power (W)	Surface T_S (°C)	Duct Inlet (ambient) T_{IN} (°C)	Difference $T_S - T_{IN}$ (°C)
0			
5			
10			
15			
20			
25	66.3	22.4	43.9
30			
35			
40			
45			
50			

4min

23/5/2023

Table 2: Results for exp. 2

Heat transfer Surface: Power:			
	T_2	T_1	
F Air velocity ($m.s^{-1}$)	Surface Ts (°C)	Duct Inlet (ambient) Tin (°C)	Difference $T_S - T_{IN}$ (°C)
1	54.6	22	32.6
1.5	45.6	22.0	23.6
2	41.~~7~~	22.1	19.6
2.5	38.7	22	16.7.
3	37.5	21.9	15.6

23/5/2023

Duct Traverse Probe Position (mm)	T_1 Ambient Temperature (Probe) Tin (°C)	T_2 Heat Transfer Surface Temperature Ts (°C)	T_3 Duct Traverse Probe Tp (°C)	$T_2 - T_1$ Ts-Tin (°C)	$T_3 - T_1$ Tp-Tin (°C)
0.5	22.1	37.4	23.5		
1.0	22.1	37.4	23.6		
1.5	22	37.3	23.4		
2	22	37.3	23.4		
2.5	22	37.3	23.1		
3	22.1	37.3	23		
3.5	22.1	37.3	22.8		
4	22.0	37.3	22.6		
4.5	22.1	37.3	22.5		
5	22.2	37.4	22.4		
5.5	22.1	37.5	22.1		
6	22.1	37.5	22.1		
6.5	22.1	37.6	22.2		
7	22.1	37.6	22.2		
7.5	22.3	37.6	22.2		
8	22.3	37.6	22.2		

Result analysis:

The University of Jordan

School of Engineering

Department of Chemical Engineering

Chemical Engineering Laboratory I (0915361)

Section Number: (2)

Experiment Number: (10) + (11)

Experiment Title: ‘Reynolds Number’ and ‘Pressure Gauge Calibrator’.

Report Type: Short Report

Instructor: Prof. Khaled Rawajfeh

Date of Performance: 11th April, 2023.

Date of Submission: 18th April, 2023.

Abstract:

A) Experiment 10 : Reynold Number:

The Reynolds number experiment is used to explore the characteristics of the liquid flow in the pipe and to calculate the Reynolds Number for each flow condition. Laminar, transitional, and turbulent flow types are those that are characterized. Fluids with a high viscosity and low speed experience laminar flow. When low viscosity fluids move at a high speed, there is a turbulent flow. transitional when the flow is transitioning from turbulent to laminar. It was found that while the velocity is low in laminar flow, the dye forms a narrow thread line, gradually swirls as the velocity increases, and completely swirls before dispersing at higher velocity, which is typical of turbulent flow. which shows The Reynolds number likewise rose in proportion to the flow.

B) Experiment 11: Pressure Gauge Calibrator :

Using a dead weight tester, this experiment was done to calibrate the pressure on a bourdon tube pressure gage. The piston is loaded with weights, and the suggested Bourdon tube pressure was determined. We added weights up to 6.2 kg (including the mass of the piston) to weights on a plate on a dead weight tester so that the weights applied a known force to a piston with a known area. then indicated the increasing gauge pressure reading and recorded the decreasing pressure readings after reversing the process. According to the experimental results, increasing the weight on the loading platform leads to an increase in gauge pressure readings.

Table of Contents:

Table of Tables:

Table of Figures:

Results

A) <u>Experiment 10 : Reynold Number</u>

Table(10-1) : Raw data

Trial	Volume (mL)	Temperature C°	Volume (L)	Time (Sec)
1	80.00	18.00	0.08	4.95
2	198.00	18.00	0.198	3.57
3	182.00	18.00	0.182	2.45

Table(10-2) : Useful Parameters for Reynold Number Calculations

Test Pipe Suction diameter	10.00 mm
Test Pipe Suction length	675.00 mm
Cross sectional Area	75.00 mm^2
Density of water	998.90 Kg/m^3
Viscosity of water	1.1081 mPa.s
Temperature	16.00 C°

Table(10-3) : Reynold Number Calculations

Trial	*Volume (m^3)*	*Volumetric flow rate (m^3/s)*	*Velocity (m/s)*	*Reynold Number*	*Observation Regime*
1	0.00008	1.61E-05	0.22	1983.19	Laminar
2	0.000198	5.54E-05	0.74	6666.21	Turbulent
3	0.00018	7.43E-05	0.99	8928.67	Turbulent

B) Experiment 11: Pressure Gauge Calibrator

Table (11-1) : Raw data

Mass (Kg)	Force W (N)	Applied Pressure (Kpa)
1.00	9.81	31.14
2.00	19.62	62.29
3.00	29.43	93.43
4.00	39.24	124.57
5.00	49.05	155.71
5.50	53.96	171.29
6.00	58.86	186.86
6.20	60.82	193.09

Table (11-1) : Raw data (continued)

Gauge Reading (Kpa)		
Increasing	**Decreasing**	**Average**
30.00	45.00	37.50
54.00	51.00	52.50
83.00	75.00	79.00
110.00	102.00	106.00
132.00	133.00	132.50
150.00	147.00	148.50
163.00	162.00	162.50
167.00	167.00	167.00

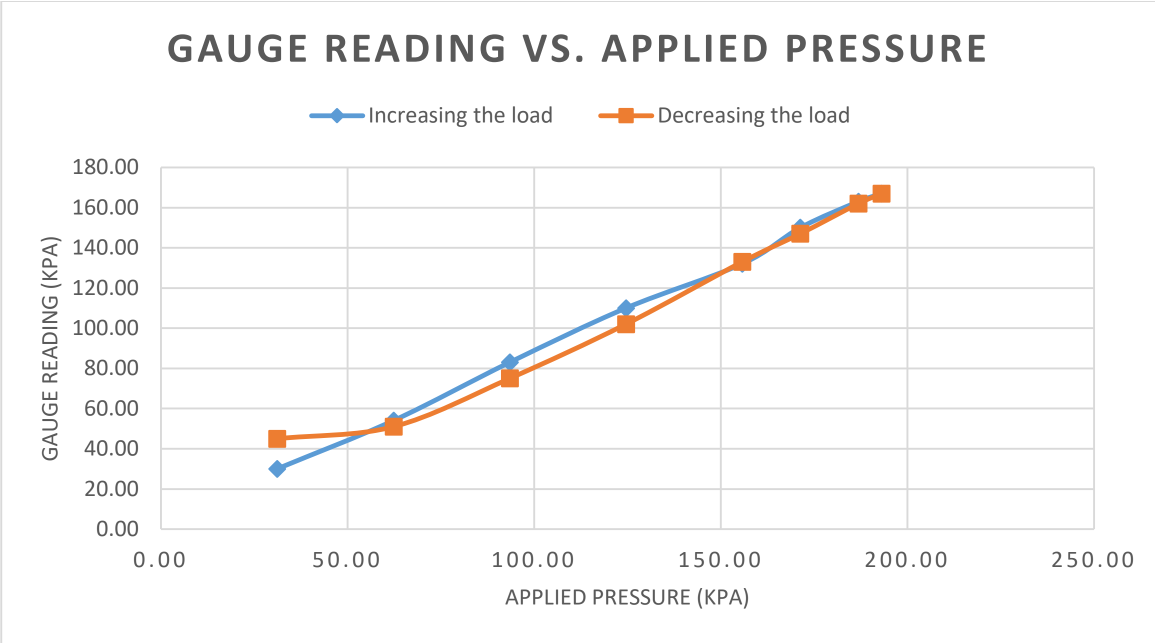

Figure (11-1): Gauge Reading Vs. Applied Pressure

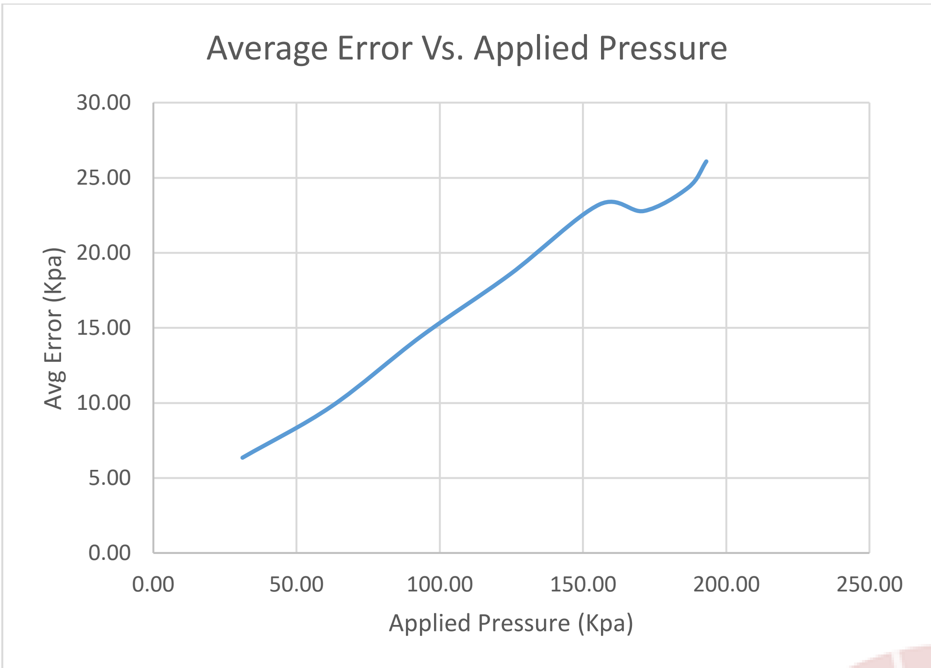

Figure (11-2): Average Error Vs. Applied Pressure

Discussion:

A) Experiment 10 : Reynold's Number

Reynold's number is a dimensionless parameter that describes the ratio of inertial forces to the viscous forces. It is used often to categorize fluids based on their flow behaviour either as laminar ($Re \leq 2300$), transient, or turbulent flow ($Re > 2300$). This is especially important to know when predicting flow behaviour in a costly process.

According to the experimental results obtained, the flow pattern in each flow state differs. As seen in table (…), in the laminar stage (where $Re \cong 2000$) the apparatus shows a clear, well-defined line of ink, and its mixing with the water is minimal due to molecular diffusion. In transient flow, the ink stream wanders about and shows intermittent bursts of mixing followed by a more laminar behaviour. For turbulent flow (where $Re \cong 9000$), the ink rapidly mixes with the water causing turbidity and this is due to the substantial lateral movement.

B) Experiment 11 : Pressure Gauge calibrator

The Bourdon pressure gauge is used to measure the amount of change in a coiled or a semi-circular metal tube by a pressurized fluid inside, when the tube is pressured by external forces, it regains its circular form after flattening. In this experiment, gauge readings (in kPa) are taken when the piston is loaded by different masses starting from 1 kg till 6.2 kg. Then reversibly, gauge readings are recorded by reducing the weights on the piston from 6.2 kg till 1 kg.

It is found that increasing the weights on the piston increases the gauge pressure, which indicates a direct relationship. However, the calculated values of applied pressure are greater than the Bourdon gauge readings, which shows an average relative error of 18.165%. This deviation from ideality is due to external factors, personal errors (inaccuracy in reading the pressure), and systematic error from the instrument.

Conclusion:

A) Experiment 10 : Reynold Number

• The value of Re is influenced by the fluid's physical characteristics, such as density and viscosity, as well as by the flow rate and cross-section of the pipe

• As the flow increased, the Reynolds number also increased.

• There were 3 types of flow: a laminar flow when the Reynolds number was < 2300, a transitional flow when the Reynolds number was..., and a turbulent flow when the Reynolds number was

B) Experiment 11 : Pressure Gauge calibrator

In this experiment, there were two different kinds of error. Hysteresis in the gauge was brought on by increasing and decreasing gauge readings. The reading error across the pressure range, which was the second error, gets worse as the applied pressure gets stronger.

References used:

1. Chemical Engineering Laboratory (1) Manual' (2022), 1st Edition, University of Jordan.
2. Engineering Toolbox (2003). Water - Density, Specific Weight and Thermal Expansion Coefficient. [online] Engineeringtoolbox.com. Available at: https://www.engineeringtoolbox.com/water-density-specific-weight-d_595.html.
3. Hyperphysics.phy-astr.gsu.edu. (n.d.). Viscosity of Liquids and Gases. [online] Available at: http://hyperphysics.phy-astr.gsu.edu/hbase/Tables/viscosity.html.

Appendices

A) *Experiment 10 : Reynold Number*

Taking the first Trial in each table

Table 1 : raw data

- Volume (ml) ➔ Volume (L)

  1 ml = 0.001 L

  So , Volume (L) = $80 \text{ ml} \frac{0.001 \text{ L}}{1 \text{ ml}}$ = 0.08 L

Table 3 : Reynold Number Calculation

- Volume (m^3) ➔ Volume (L)

  1 m^3 = 1000 L

  So , Volume (m^3) = $0.08\, L \frac{1\, m^3}{1000\, L}$ = 0.00008 m^3

- Volumetric Flow rate $(m^3/s) = \frac{Volume\ (m^3)}{Time(s)}$

  $= \frac{0.00008\, m^3}{4.95\, s} = 0.000016\, m^3/s$

- Velocity (m/s) $= \frac{Volumetric\ Flow\ rate}{Cross\ sectional\ Area}$

  $= \frac{0.000016\, m^3/s}{75 \times 0.0000001\, m^2}$ = 0.22 m/s

- Reynolds Number= $\frac{\rho D V}{\mu}$

  $= \frac{998.90 \frac{Kg}{m^3} * 10\ mm * 0.21 \frac{m}{S}}{1.1081\ mPa.s}$ = 1983.19

  Based on the value of Reynold Number the regime is Laminar (Re < 2300)

B) Experiment 11 : Pressure gauge Calibrator

Taking the first row in table 1:

- Force W (N) = m×g

  = 1 Kg ×9.81 $\mathrm{m/s^2}$ = 9.81 N
- Applied pressure = F/A

  Given Area = 315 mm^2

  = $\frac{9.81\times 0.001\ KN}{315\times 0.000001\ m^2}$ = 31.14 Kpa

  Or Applied Pressure = $m\ \times k$

  $$= 1\ Kg * 31.14 = 31.14\ \ Kpa$$

- Average Error = | Applied Pressure – Average Gauge reading |

  = | 31.14 – 37.5|

  = 6.36Kpa.

- Error (% full scale) = $\frac{Average\ Error}{Full\ Scale\ in\ gauge\ Pressure} \times 100\%$

  = $\frac{6.36\ Kpa}{200.00} \times 100\% = 3.18\%$ Kpa

Data sheet

A) Experiment 10 : Reynold Number

trial	Volume(ml)	Temperature(˚c)	Volume(L)	Time(sec)
1	80	18	0.08	4.95
2	198	18	0.198	3.57
3	182	18	0.182	2.45

B) Experiment 11 : Gauge Pressure Calibrator

Pressure gauge calibrator Data Sheet

Piston Area (A): 315 mm^2

Mass (m) (kg)	Force (W) (N)	Applied Pressure (p) $kN.m^{-2}$	Gauge Reading ($kN.m^{-2}$) Increasing	Decreasing	Average	Average Error ($kN.m^{-2}$)	Error (% of full scale)
1kg			30	45			
2			35.54	51			
3			83	75			
4			110	102			
5			137	133			
5.5			150	147			
6			163	162			
6.2			167	167			

Instructor sign: م. هبة طلافحة

Date: 11/4/2023

The University of Jordan

School of Engineering

Department of Chemical Engineering

Chemical Engineering Laboratory I (0915361)

Section Number: (2)

Experiment Number: (12)

Experiment Title: ‘Efflux time for a tank with exit pipe’.

Report Type: Short Report

Instructor: Prof. Khaled Rawajfeh

Date of Performance: 9th May, 2023.

Date of Submission: 16th May, 2023.

Abstract:

In this experiment, the efflux time of an exit pipe with different diameters and lengths was calculated experimentally and theoretically for a glycerine-filled cylindrical tank. Raw data was recorded with elapsed time for two categories: 3 pipes with same diameter, and 3 pipes with same length. It was seen that for the first category of pipes with the same diameter; as length of the pipe increased the efflux time increased as well, and that is due to the increased friction. For the second category of pipes with the same length; with increasing diameter the efflux time decreased.

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Results

Table 1 : Raw data and calculated parameters

H1(m)	0.21
H2(m)	0.16
Room Temperature (˚C)	26.70
Viscosity (Pa.s)	0.03
Density (Kg/m3)	1167.00
Internal Diameter of the tank (m)	0.16
Internal Height of the tank (m)	0.26
Volume of the tank (m^3)	0.001

A) *Pipes have the same diameter and different length*

Table 2 : Raw and calculated data for Pipes with the same diameter and different length

Constant diameter = 0.00535 m						
Area = 0.0000225 m^2						
L (m)	**t (s) (average)**	**Q (m3/s)**	**u (m/s)**	**Re**	**F (Laminar)**	**t_{eff} (s)**
0.09	117.00	0.0000085	0.38	79.13	0.20	42.84
0.16	146.00	0.0000068	0.30	63.41	0.25	62.56
0.32	180.00	0.0000056	0.25	51.43	0.31	84.29

Table 3 : ratio of the Experimental Efflux time to the Calculated Efflux time at constant diameter

t_{exp}/t_{eff}		
Pipe 1 (L = 0.0874 m)	**Pipe 2(L = 0.1634 m)**	**Pipe 3 (L = 0.3184 m)**
2.73	2.34	2.14

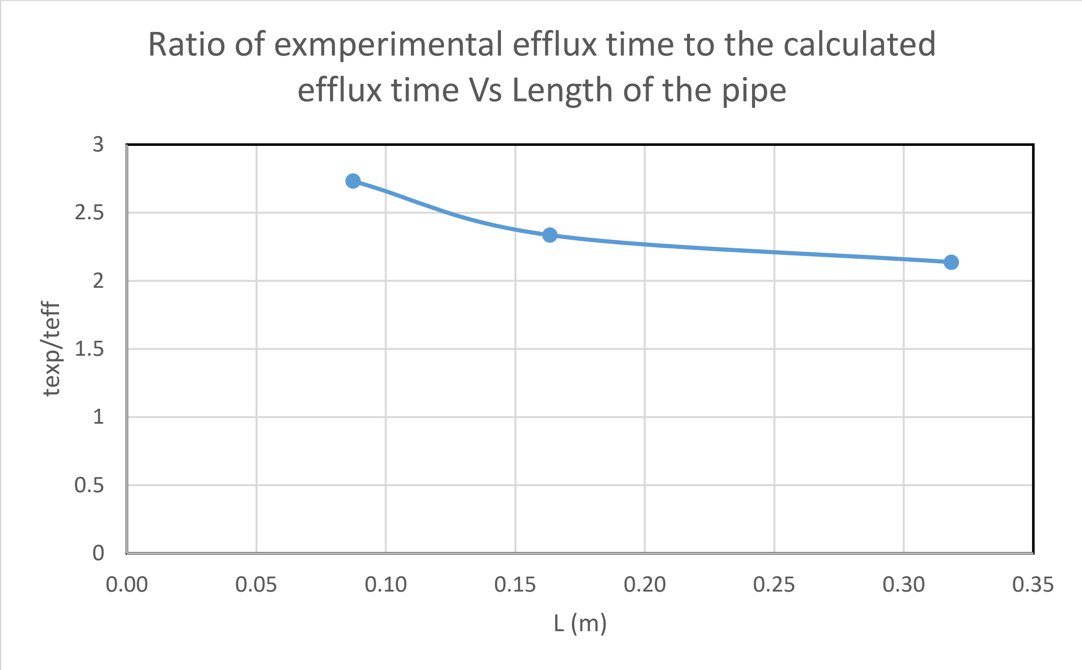


Figure 1: Ratio of experimental efflux time to the calculated time (effective) Vs. Length of the pipe

B) *Pipes have the same Length and different diameter*

Table 4: Raw and calculated data for the same pipe length and different diameter

L = 0.6324 m							
D (m)	t (s) (average)	Area (m^2)	Q (m^3/s)	u (m/s)	Re	f (laminar)	t_{eff} (s)
0.00210	2100.00	0.0000035	0.00000048	0.14	11.37	1.41	4341.07
0.00535	206.00	0.000022	0.00000491	0.22	45.48	0.35	103.05
0.00840	25.00	0.000055	0.0000405	0.73	238.71	0.067	16.96

Table 5 : ratio of the experimental Efflux time to the Calculated Efflux time at constant length

t_{exp}/t_{eff}		
Pipe 1 (D = 0.0021 m)	**Pipe 2 (D = 0.00535 m)**	**Pipe 3 (D = 0.0084 m)**
0.483	1.99	1.47

Table 6: ratio of the diameter of the tank to the diameter of the Pipe

$\mathbf{DT/d}$		
Pipe 1 (D = 0.0021 m)	**Pipe 2 (D = 0.00535 m)**	**Pipe 3 (D = 0.0084 m)**
76.45	30.00	19.11

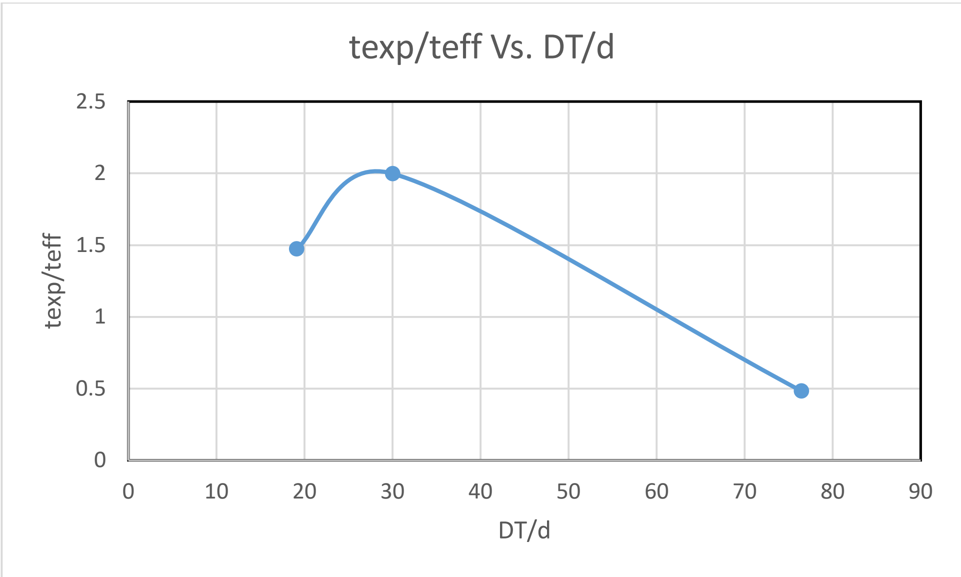


Figure 2 : t_{exp}/t_{eff} Vs. DT/d

Discussion:

For pipes of the same diameter (5.35 mm) with varying lengths, the efflux time increases with increasing pipe length, and this is due to the increasing friction. Since the cross-sectional area is constant (A=$2.25 \times 10^{-5} m^2$), it can be seen that the volumetric flow rate decreases with increasing pipe length, moreover, decreasing glycerine exit velocity, and consequently, laminar flow is dominant in this part.

For pipes of the same length (623.4 mm) with varying diameters, increasing the diameter decreases the efflux time needed to empty 1 L of the tank (constant head) , because there is less friction with larger diameters therefore increasing the volumetric flow rate and thus the exit velocity of the glycerine. Since the diameters are varied, then the cross-sectional area is no longer constant, unlike the first part. Laminar flow is also dominant based on the results. As can be seen in figure (2), increasing the ratio of tank diameter to tube diameter increases the ratio of experimental efflux time to calculated efflux time until a maximum of 2, then gradually decreases.

Some experimental errors occurred which is believed to have affected the deviation between calculated efflux time and the experimental efflux time.

Conclusion:

For a constant pipe diameter, pipe length and efflux time have a direct relationship. And consequently, pipe length and outlet volumetric flow rate have an inverse relationship, and so exit velocity decreases with increasing pipe length.

For a constant pipe length, pipe diameter and efflux time have an inverse relationship. Increasing the diameter leads to an increase in the outlet volumetric flow rate and so the exit velocity.

In both cases, the flow exhibited laminar behaviour, because Reynolds number <2100. Moreover, a slight deviation is noticed between the experimental efflux time and calculated efflux time, which is due to impurities in the glycerine-water mixture, which affects the values of calculated density and viscosity.

Experimental errors are due to impurities in the glycerine-water mixture, and friction between this mixture and the walls of the tank and pipes. Personal errors could be due to precise but inaccurate reading of the tank calibrator.

References used:

1. Department of Chemical Engineering, (2022). Chemical Engineering Laboratory (1) Manual (Version No.1), Efflux Time for a Tank with Exit Pipe. University of Jordan.
2. Geankoplis, C.J., A Allen Hersel and Lepek, D.H. (2018). Transport processes and separation process principles. 5th ed. Boston: Prentice Hall.

Appendices

1. Sample of calculations

<u>Taking the first row of each table in the results:</u>

- Mass of Empty bottle (pycnometer) = 34.3 g
- Mass of water + bottle = 83.6 g
- Density of water At 26.7 $^{\circ}C$ = 996.66 Kg/m^3
- Volume of the bottle = $\frac{83.6-34.3}{0.9966} = \ 49.5\ ml$
- Mass of bottle + mixture = 92.1 g
- Density = $\frac{(Mass\ of\ Bottle+mixture)-(Mass\ of\ Empty\ bottle)}{Volume\ of\ \ empty\ bottle} = \frac{57.8\ g}{49.5\ ml} = 1.167\ g/ml$ = 1167 Kg/m^3
- Volume of the tank= $\frac{\pi}{4}(internal\ diameter\ of\ tank^2)(H_1 - H_2)$
$$= \frac{\pi}{4} \times (0.16054^2)(0.21 - 0.16) = 0.001 m^3.$$

A) <u>Pipes have the same diameter and different length</u>

- $Area = \frac{\pi}{4}(0.00535^2) = 2.248 \times 10^{-5} m^2.$
- $Volumetric\ flow\ rate: Q = \frac{V}{t} = \frac{0.001}{117} = 8.54 \times 10^{-6}\ \frac{m^3}{s}.$
- $velocity: u = \frac{Q}{A} = \frac{8.54\times10^{-6}}{2.248\times10^{-5}} = 0.38\frac{m}{s}.$
- Reynold's Number: $Re = \frac{\rho u D}{\mu} = \frac{1156(0.00042735)(0.00535)}{(0.03)} = 79.13$
- Friction factor: $f = \frac{16}{Re} = \frac{16}{78.33} = 0.20$
- Efflux time:
$$t_{eff} = \frac{32\mu L D_t{}^2}{\rho g d^4}\ln\left[\frac{L+H_1}{L+H_2}\right] = \frac{32(0.03)(0.0874)(0.16054^2)}{(1156)(9.81)(0.00535^4)}\ln\left[\frac{0.0874+0.21}{0.0874+0.16}\right] = 42.84s$$
- $\frac{t_{exp}}{t_{calc}} = \frac{117}{42.84} = 2.73$ (for the first pipe)

B) <u>Pipes have the same Length and different diameter</u>

- Area: $A = \frac{\pi}{4}D^2 = \frac{\pi}{4}(0.0021^2) = 3.46 \times 10^{-6} m^2.$
- Volumetric flow rate: $Q = \frac{V}{t} = \frac{0.001}{2100} = 4.81 \times 10^{-7} m^3/s$
- Velocity: $u = \frac{Q}{A} = \frac{4.81\times10^{-7}}{3.46\times10^{-6}} = 0.14\ m/s$

- Reynold's Number: $Re = \frac{\rho u D}{\mu} = \frac{1167(0.14)(0.0021)}{0.03} = 11.37$
- Friction factor: $f = \frac{16}{Re} = \frac{16}{11.37} = 1.41$

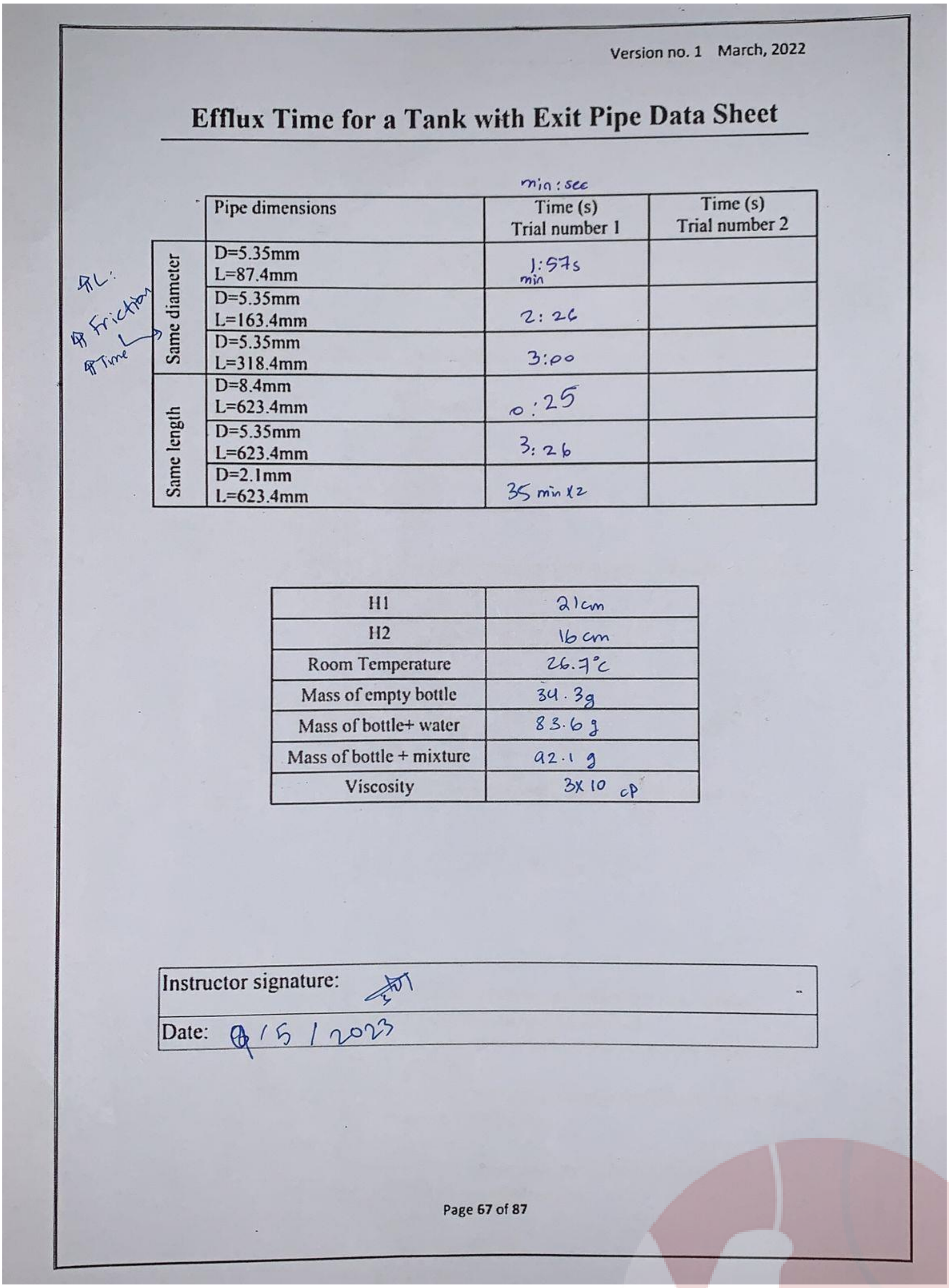
- Efflux time:

$$t_{eff} = \frac{32\mu L D_t{}^2}{\rho g d^4} \ln\left[\frac{L+H_1}{L+H_2}\right] = \frac{32(0.03)(0.6234)(0.16054^2)}{1156(9.81)(0.0021^4)} \ln\left(\frac{0.6234+0.21}{0.6234+0.16}\right) = 4341.07 \text{ s}$$

- $\frac{t_{exp}}{t_{calc}} = \frac{2100}{4341.7} = 0.4837$ (for the first pipe)

- $\frac{d_{tank}}{d_{pipe}} = \frac{0.16054}{0.0021} = 76.44$ (for the first pipe)

2. *Data sheet*



Version no. 1 March, 2022

Efflux Time for a Tank with Exit Pipe Data Sheet

↑L: ↑ Friction → ↑Time

	Pipe dimensions	Time (s) Trial number 1 (min : sec)	Time (s) Trial number 2
Same diameter	D=5.35mm L=87.4mm	1:57s min	
Same diameter	D=5.35mm L=163.4mm	2:26	
Same diameter	D=5.35mm L=318.4mm	3:00	
Same length	D=8.4mm L=623.4mm	0:25	
Same length	D=5.35mm L=623.4mm	3:26	
Same length	D=2.1mm L=623.4mm	35 min x2	

H1	21cm
H2	16 cm
Room Temperature	26.7°c
Mass of empty bottle	34.3g
Mass of bottle+ water	83.6 g
Mass of bottle + mixture	92.1 g
Viscosity	3x 10 cP

Instructor signature:
Date: 9/5/2023

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