2024년도 2학기

기계공학연구 보고서

실험명 : Centrifugal pump experiment

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학 과 : Mechanical Engineering

학 번 :

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1. Experimental Purpose

Centrifugal pump experiment's purpose is calculation the efficiency and various pump's diagram by changing following experimental variables, 'speed', 'types of impeller' and 'flow rate'.through experimental data. About experimental variables' bigger perspective, 'speed control' involves 'speed' and 'flow rate'. 'Impeller replacement' involves 'types of impeller' and 'flow rate'.
Ultimate purpose is understanding this pump's structure and operating characteristics. Therefore, we need to prepare laptop, SW, control panel and centrifugal pump as centrifugal pump experiment's material in advance.

2. Experimental method

i) Speed control
- Lock the outlet v/v completely and connect control panel to centrifugal pump.
- Connect 1x power at the back and 2x wires at the front of panel. After that, turn on all switches and check whether green colored light turns on or not.
- Access SW-view diagram. Rotate only one cycle the outlet v/v to prevent pump's overload.
- Operate pump with changing 'speed' to 60%, 70% and 80% through SW.
- Increase flow rate by opening the outlet v/v repeat it 10x times for each case.

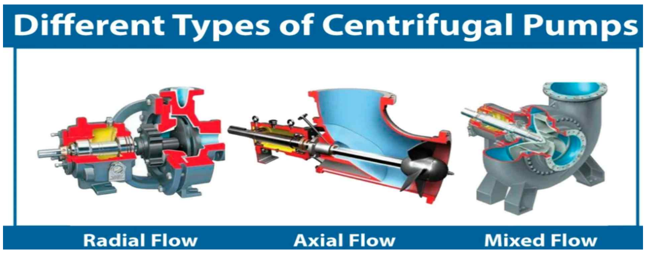
ii) Impeller replacement
- Remove 6x fixed screws and underneath pipe. Next, replace the impeller which has difference about blade's direction and length.
- Assemble with reversed sequence. and operate it with changing impeller.

Caution details on experiment.
- Don't rotate too many times of outlet v/v. Also, Don't set 100% and don't operate for a long time even the pump set as 100%. Pump's overload can be occurred due to them.
- Don't operate such a long time with the locked v/v.
- Be careful of losing 'O-ring' when we assemble the pipe. Next, make sure to tighten the screw well.

3. Background information related to the experiment

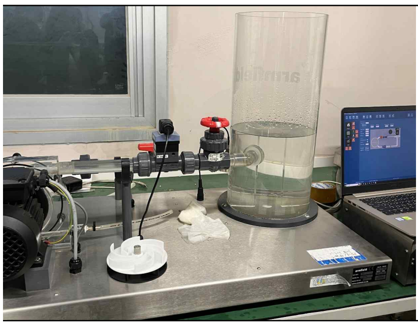
- Turbomachines

Turbomachines the tools that are consisted with rotating blades. Turbomaachines' roles are extraction enery from fluid (turbine) and supplement energy to fluid (pump) through dynamics interaction. Operating mechanism is similar whether the fluid type is liquid or gas.



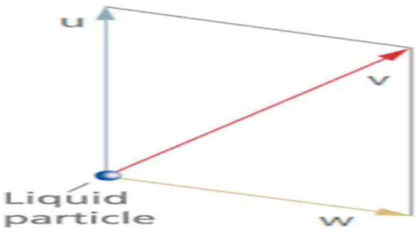
There are three types of turbomachines, axial-flow type pump, radial-flow type pump and mixed-flow type pump. Seperation standard is the predominant direction which generates fluid flowand related to the rotor's axis as the fluid passes turbomachines' blades.

In real CP experiment, we use pump and impellers following pictures.



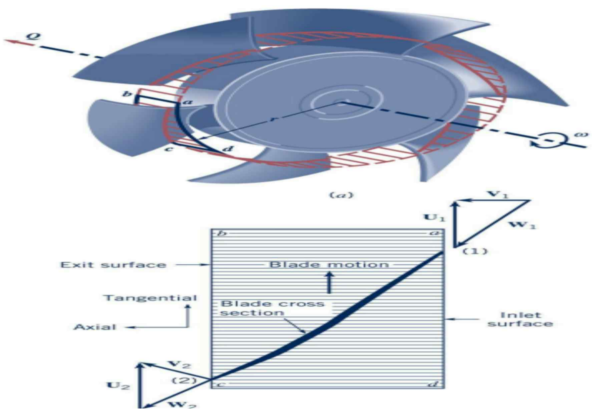
- Dynamic interaction between fluid and turbomachines

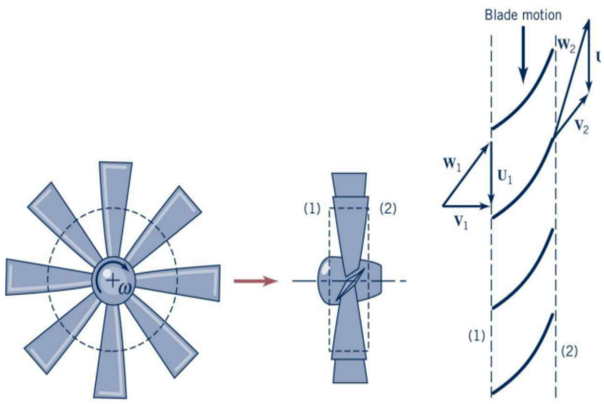
In real, fluid flow in turbomachines is unsteady flow so it is too complicated. However, we can simplify the fluid as following experssion, 'V=W+U'. It can be defined as 'velocity triangles'.



(V: the absolute fluid velocity which the stationary observer observes on the table that accompanies turbomachines blades, W: the relative fluid velocity which the stationary observer observes on the turbomachines blades, U: the blade's speed & U=r*ω, r: blade's radius, ω: blade's angular velocity.)

We are able to have the application of it to pump and turbine with following assumptions, 'Fluid flow runs smoothly through blades.' and 'Fluids enter and exit at the same radius from the axis of rotation'. In other words, $U_1=U_2=r*\omega$ is valid.



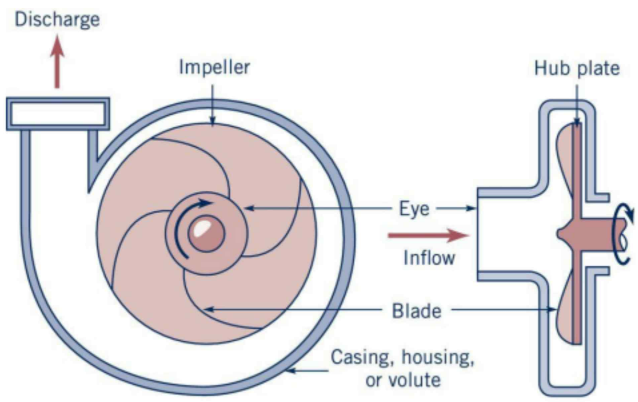


On the left side of the picture, it expresses pump fan's velocity diagram. About inlet surface, we know U_1 and V_1 so we are able to confirm W_1's value. About outlet surface, we know U_2 and W_2 so we are able to confirm V_2's value from velocity triangles. Similarily, on the right side of the picture expresses turbine's velocity diagram expresses U_1 and V_1 in inlet surface so we are able to confirm W_1's value and U_2 and W_2 in outlet surface so we are able to confirm V_2's value. V_θ, which is the velocity of the blade motion is opposite for each case (fan and turbine) and it means law of action and reaction.

- Angular momentum consideration in turbomachines

When we assume the fluid is continuum and the particles passes through the rotor. Then, angular momentum equation, $\sum(\mathbf{r} \times \mathbf{F}) = \int_{cs} (\mathbf{r} \times \mathbf{F})\, \rho\, \mathbf{V} \cdot \hat{n}\, dA$ is valid in control volume. When we simplify it, it can be transformed to $T_{shaft} = -\dot{m}_1(r_1 V_{\theta 1}) + \dot{m}_2(r_2 V_{\theta 2})$ (1: enter, 2: exit) and this form is euler turbomachine equation. Also, $\dot{W}_{shaft} = T_{shaft}\,\omega$ so $\dot{W}_{shaft}$ can be transformed to $-\dot{m}_1(U_1 V_{\theta 1}) + \dot{m}_2(U_2 V_{\theta 2})$. Following definition, w_{shaft} ($\dot{W}_{shaft}$ per mass) $= -U_1 V_{\theta 1} + U_2 V_{\theta 2}$. is valid because $\dot{m}_1 = \dot{m}_2$ is valid in continuity equation.

- Centrifugal pump

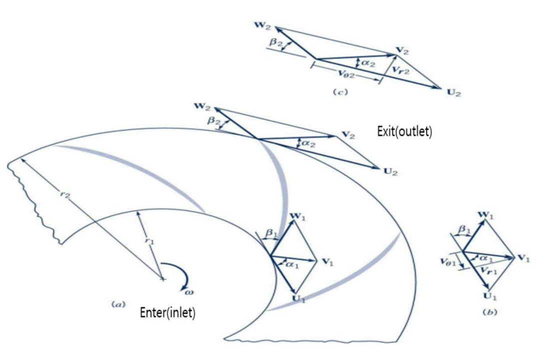


It is the most common turbomachine and its type is radial-flow pump. Operation principle sequence is as follows.

- Fluid flows into casing's eye when the impeller rotates.
- Fluid is driven towards the Hub plate and leaks to outlet.
- Energy transfers to fluid from the impeller due to torque. It expresses pressure increasement in the casing(housing and volute).

About pump's inner structure, the distance between casing and impeller tends to increase. It has designed to convert bigger pressure energy by speed decreasement.

- Centrifugal pump's fluid flow

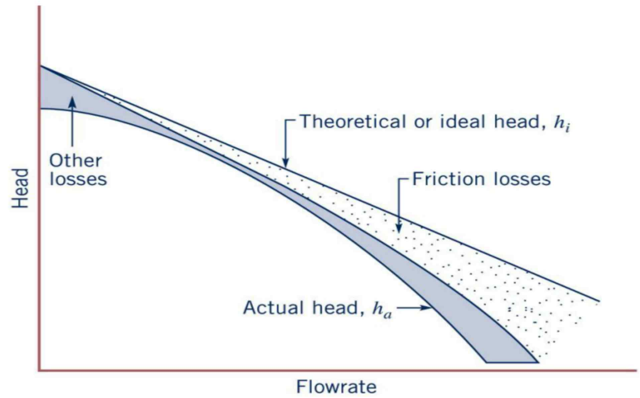


The picture expresses flow structure in centrifugal pump. As I mentioned in dynamic interaction, fluid flow in turbomachines is unsteady flow so it is too complicated in real but we can simplify it by V=W+U. Due to V=W+U, $V_1=W_1+U_1$ ($U_1=r_1*\omega$) is valid when we set inlet is enter and outlet is exit. The efficiency will be maximized when $W_1=U_1$. Similarily, $V_2=W_2+U_2$ ($U_2=r_2*\omega$)is valid. When we compare $V_{\theta 1}$ and $V_{\theta 2}$, we are able to define $V_{\theta 2}$ increases. It means fluid gets work through centrifugal pump so energy increases.

In this moment, $T_{shaft} = \dot{m}(r_2 V_{\theta 2} - r_1 V_{\theta 1}) = \rho Q(r_2 V_{\theta 2} - r_1 V_{\theta 1})$. Therefore, $\dot{W}_{shaft} = \rho Q \omega(r_2 V_{\theta 2} -$

$r_1*V_{\theta 1}$) = $\rho Q(U_2 V_{\theta 2} - U_1 V_{\theta 1})$ from T_{shaft} = $\dot{m}(r_2 V_{\theta 2} - r_1 V_{\theta 1})$. The conclusion is $w_{shaft} = \frac{\dot{W}_{shaft}}{\rho Q} = U_2*V_{\theta 2} - U_1*V_{\theta 1}$. The conclusion is still valid to the rotating shaft. $w_{shaft} = U_2*V_{\theta 2} - U_1*V_{\theta 1} = (\frac{p_{out}}{\rho} + \frac{V^2_{out}}{2} + gz_{out}) - (\frac{p_{in}}{\rho} + \frac{V^2_{in}}{2} + gz_{in})$ + loss in incompressible pump flow. If we divide by g from the equation, $\frac{U_2 * V_{\theta 2} - U_1 * V_{\theta 1}}{g}$ can de defined as $h_i = h_{out} - h_{in} + h_L$ (h_i =ideal head, h_L = head loss). There is the case which h_i can be arranged to $\frac{U_2 * V_{\theta 2}}{g}$ when the fluid enters the impeller. The reason is the fluid doesn't have tangential component of velocity (=$V_{\theta 1}$) or swirl. When we express blade's angle formula to velocity component, $\cot\beta_2$ is $\frac{U_2 - V_{\theta 2}}{V_{r2}}$. Then, h_i can be written as $\frac{U^2{}_2}{g} - \frac{U_2 * V_{r2} * \cot\beta_2}{g}$ and it can be written another form with Q(the flow rate), such as $\frac{U^2{}_2}{g} - \frac{U_2 * V_{r2} * \cot\beta_2}{2\prod r_2 b_2 V_{r2}}Q$ ($Q=2\prod r_2 b_2 V_{r2}$). From this form, we can expect h_i and Q is on linear relation. Additionally, the curve is classified by angle β_2. The curve is forward curve when $\beta_2 > 90°$. The curve is radial curve when $\beta_2 = 90°$. The curve is backward curve when $\beta_2 < 90°$. Generally, engineers design pump with backward curve and design cheaper one with radial curve.

- Theoretical considerations in Centrifugal pump

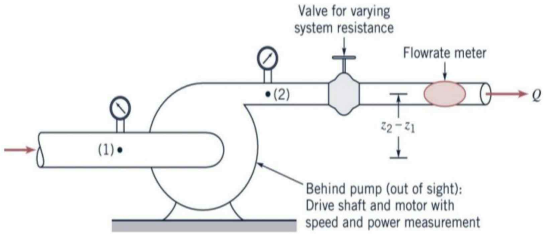


We have already arranged h_i (ideal head) = $\frac{U_2 * V_{\theta 2} - U_1 * V_{\theta 1}}{g} = h_{out} - h_{in} + h_L$. Then, h_i can be defined as $h_a + h_L$ (h_a = The actual rise in head of fluid). The curve, h_a versus flowrate becomes to lie below the ideal head curve. Additionally, we are able to know the h_a versus flowrate curve shows a nonlinear variation with flowrate. This nonlinear variation's expected principles are 'losses'.

- Losses

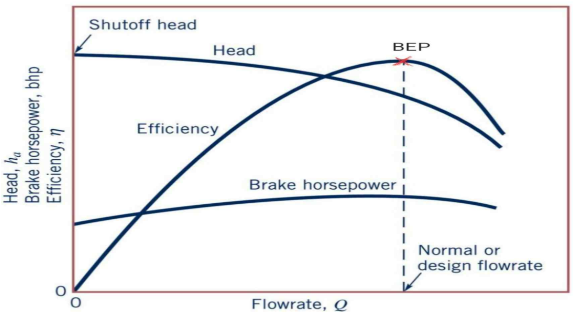
The losses can be classified friction losses and other losses. Other losses can be classified in more detail as 'hydraulic losses', 'volumetric losses' and 'mechanical losses'. Hydraulic losses are flow seperation and caused by skin friction in the blade passages. Also, it occurred due to dimensional difference. Hydraulic losses are 3-dimensional effects but we assumed it as 1-dimension. Therefore, the losses from dimesional difference are inevitable. Volumetric losses are leakage flow and caused by impeller blade-casing clearance flows. Mechanical losses caused by bearings and seals such as 'O-ring'. In conclusion, we need to draw 'pump performance curves' which actual pump performance is determined experimentally through repeated tests on the pump to consider these types of errors.

- Pump performance characteristics

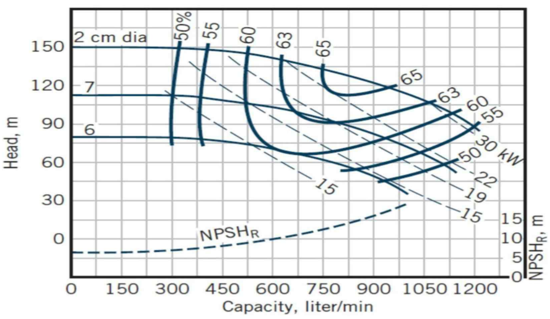


Actual head in this pump system equals $\frac{p_2 - p_1}{\gamma}$ + z_2 -z_1 + $\frac{V^2{}_2 - V^2{}_1}{2g}$. When we calculate flowrate at 'Flowrate meter', $z_2 \approx z_1$, $V_2 \approx V_1$ and h_a can be defined as $\frac{p_2 - p_1}{\gamma}$ ($h_a \approx \frac{p_2 - p_1}{\gamma}$). Power which transfered to fluid P_f is $\gamma Q h_a$ and overall efficiency η equals $\frac{P_f}{W_{shaft}}$ and $\eta_h \eta_m \eta_v$ (η_h = hydraulic efficiency, η_m = mechanical efficiency, η_v = volumetric efficiency).

Pump performance characteristics such as 'brake horsepower', 'efficiency' and 'shutoff head' are important components to draw pump performance curves.



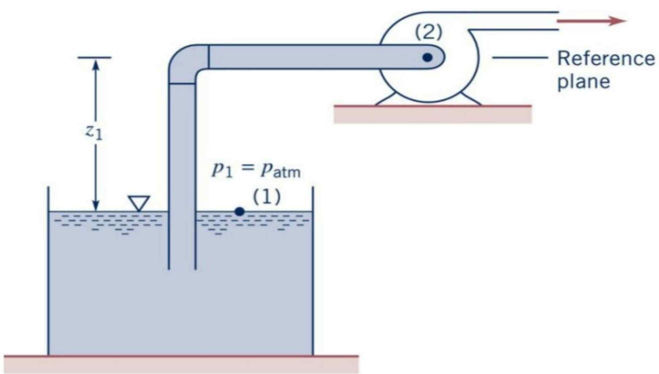
Brake horsepower is the power which can be gained when we control machine's rotation well. Shutoff head is the head which the pump makes when the discharge flowrate is zero. Normal flowrate means the flowrate which has maximum efficiency. BEP is the maximum efficiency point in normal flowrate. Generally, engineers intend to operate pump at BEP point.



Left side's picture represents the pump performance curves of impellers with different diameters running at the same angular velocity. An 'equal efficiency line' appears in a relatively vertical direction, and an 'equal power line' appears in a relatively horizontal direction. NPSH$_R$ is a 'required effective suction head' and is an issue that occurs at the pump suction part.

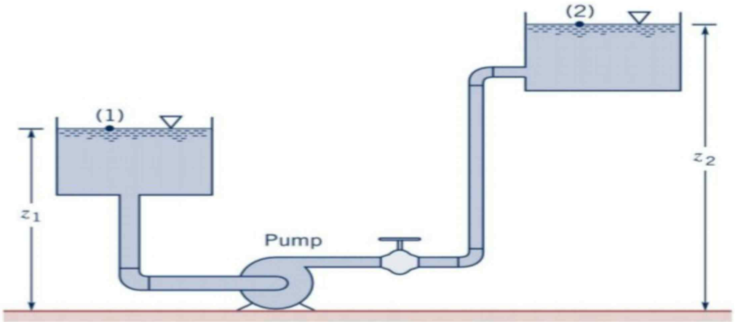
- NPSH (Net Positive Suction Head)

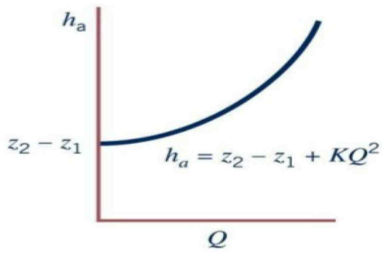
There is the low pressure on pump's suction side and 'cavitation', the boiling phenomenon by pressure's decreasement occurs due to it. To characterize the potential for cavitation, we can use following formula, NPSH = (Total head - Vapor pressure head) = $\frac{p_s}{\gamma} + \frac{V_s{}^2}{2g} - \frac{p_v}{\gamma}$.



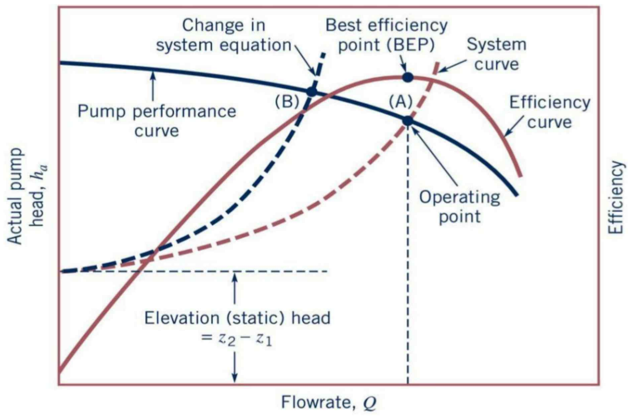
There are two types of NPSH, 'The required NPSH(NPSH$_R$)' and 'The available NPSH(NPSH$_A$)'.The type selection depends on standard points. NPSH$_R$'s the head that cavitation will not occur and NPSH$_A$'s the real suction head that occurs in specific flow system. NPSH$_A$'s calculation formula is $\frac{p_{atm}}{\gamma}$ = z_1 - $\sum h_L$ $-$ $\frac{p_v}{\gamma}$. The condition that cavitation does not occur is $NPSH_A \geq NPSH_R$.

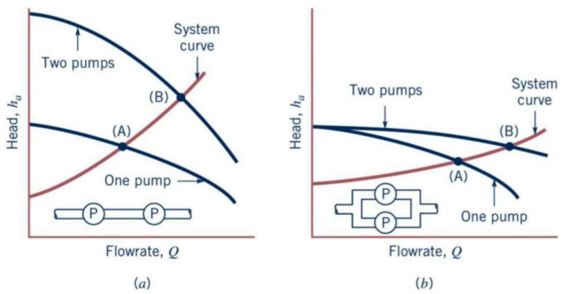
- System Characterisitics & Pump Selection





The actual headd equals $(z_2-z_1)+\sum h_L$ and it can be arranged as $(z_2-z_1)+KQ^2$. This formula is head formula which system needs about flowrate passes system.





'The pump performance curve' and 'The system curve' are drawn simultaneously to select pump and reasonable system. The point of contact between two curves is 'Operating point'. Additionally, we can confirm flowrate Q doesn't increase after A point. Its reasons is the actual head and pump's RPM are pump and system's specification. In other words, there is a limit to increasing the flow rate beyond the pump and system specifications.

About types of pump's circuit connection, there is head flowrate difference in series type and parallel type. In series type, operating point (A) changes to (B) point. Therefore, we can get pump performance curve by addition head from decreased flowrate. In parallel type, we can get pump performance curve by addition flowrate from same head.

- Dimensionless Parameters and Similarity laws

If the three dependent variables of the pump are organized through dimensional analysis of the variables that affect 'actual head elevation h_a', 'refractory power', and 'efficiency', $f(D, l_i, \varepsilon, Q, \omega, \mu, \rho) = \phi(\frac{l_i}{D}, \frac{\varepsilon}{D}, \frac{Q}{\omega D^3}, \frac{\rho \omega D^2}{\mu})$. 6 dependent variables reduced to 4 dependent variables and it means it became more comfortable to anaylze the variables' dimension. 'Head rise coefficient' $C_H = \frac{gh_a}{\omega^2 D^2} = \phi_1(\frac{l_i}{D}, \frac{\varepsilon}{D}, \frac{Q}{\omega D^3}, \frac{\rho \omega D^2}{\mu})$, ;Power coefficient' $= \frac{W_{shaft}}{\rho \omega^3 D^5} = \phi_2(\frac{l_i}{D}, \frac{\varepsilon}{D}, \frac{Q}{\omega D^3}, \frac{\rho \omega D^2}{\mu})$ and 'The efficiency' $\eta = \frac{\rho g Q h_a}{W_{shaft}} = \phi_3(\frac{l_i}{D}, \frac{\varepsilon}{D}, \frac{Q}{\omega D^3}, \frac{\rho \omega D^2}{\mu})$. About $\frac{\rho \omega D^2}{\mu}$, high Re value, $\frac{\varepsilon}{D}$ and $\frac{l_i}{D}$ can be neglected. $\frac{\varepsilon}{D}$'s neglect reason is irregular pump chamber's shape and $\frac{l_i}{D}$'s neglect reason is geometrical similarity. Therefore, these formulas can be arranged following other forms, $\frac{gh_a}{\omega^2 D^2} = \phi(\frac{Q}{\omega D^3})$, $\frac{W_{shaft}}{\rho\omega^3 D^5} = \phi(\frac{Q}{\omega D^3})$ and $\frac{\rho g Q h_a}{W_{shaft}} = \phi(\frac{Q}{\omega D^3})$. $\frac{Q}{\omega D^3}$ is flow coefficient.

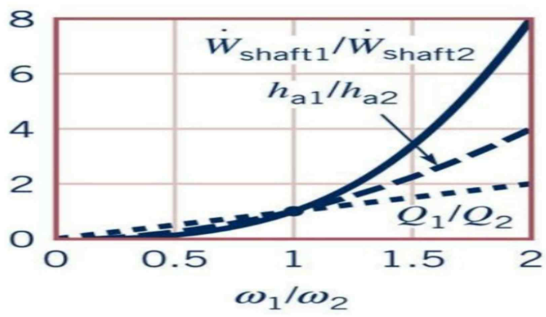
- Special Pump Scaling laws

When the two pumps which are geometric similarity have the same flow coefficient, $(\frac{Q}{\omega D^3})_1 = (\frac{Q}{\omega D^3})_2$ is valid. It follows that $(\frac{gh_a}{\omega^2 D^2})_1 = (\frac{gh_a}{\omega^2 D^2})_2$, $(\frac{W_{shaft}}{\rho\omega^3 D^5})_1 = (\frac{W_{shaft}}{\rho\omega^3 D^5})_2$ and $(\eta)_1 = (\eta)_2$ are valid due to pump scaling laws. It is possible to predict the properties corresponding to other pumps which are geometric similar through these equations.

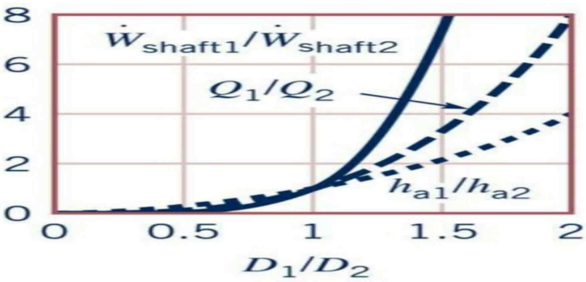
$$\frac{Q_1}{Q_2} = \frac{\omega_1}{\omega_2}$$

$$\frac{h_{a1}}{h_{a2}} = \frac{\omega_1^2}{\omega_2^2}$$

$$\frac{\dot{W}_{shaft\,1}}{\dot{W}_{shaft\,2}} = \frac{\omega_1^3}{\omega_2^3}$$



Left side of the picture's equations are valid when $D_1 = D_2$ and $\omega_1 \neq \omega_2$. On the right side of the picture, it is about the two pumps operating at different angular velocities in the flow coefficient (similarity law). However, the operating points of the two pumps operating at different angular velocities are not converted. It is difficult to say that the operating point is uniformly converted because it depends on the system characteristics as well as the pump characteristics. Exceptionally, if there is no system deformation, and the system shape is a condition where there is no pressure head or pressure head, direct conversion of the operating point is possible.

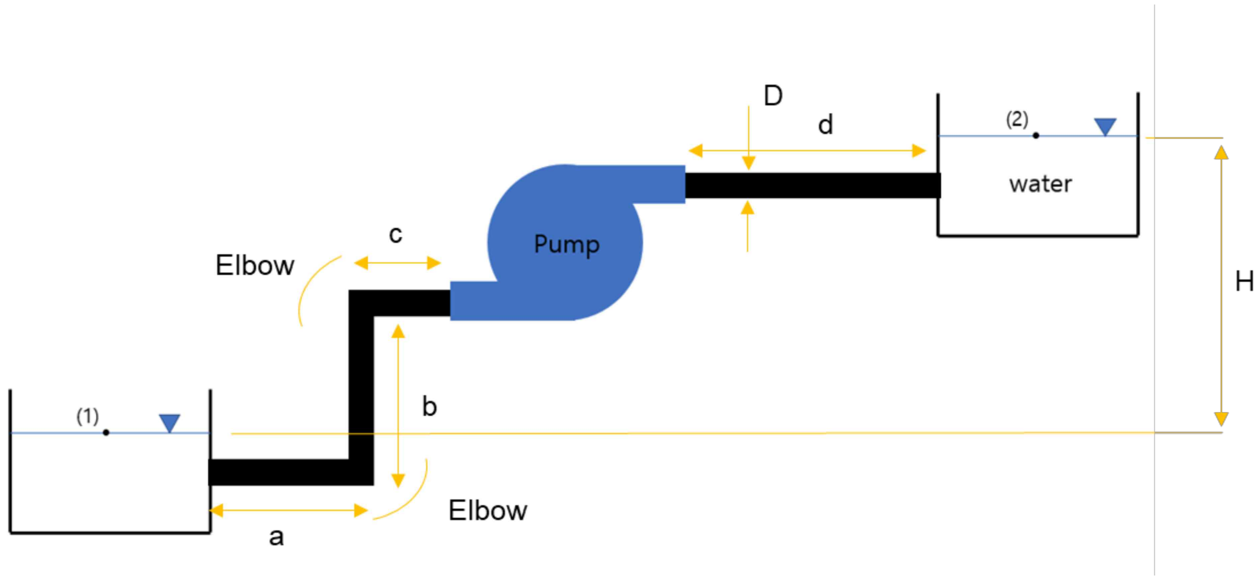


When we find out how the change in impeller diameter D affects the pump characteristics at a pump that is geometrically similar at the same angular velocity, the above relationship means scaling so that all other important geometric variables become similar when the impeller diameter is converted.

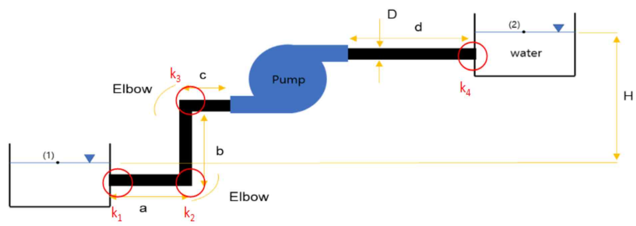
However, it is difficult to completely follow the above law due to the difficulty of manufacturing the pump. Until now, the parts for the 'viscosity' and 'surface roughness effect' have been ignored in the equations, but the smaller the pump, the greater the influence of the above variables. The experimental equation approximating the effect of reducing the size of the pump on the efficiency is as follows. $\frac{1-\eta_2}{1-\eta_1} \approx (\frac{D_1}{D_2})^{1/5}$.

4. Experiment results

Using the pump we tested, we will try to apply it to the system we have set up. The material of the pipe is galvanized iron, and the total length and diameter are 2m and 0.02m, respectively. Also, use incompressible water at room temperature and in a steady state. Water flows in through a slightly rounded corner inlet and discharges through a slightly rounded outlet, and there are two elbows with a standard flange joint of 90 degrees. The difference in surface height of the two open tanks is 2 m. (Solve the given problem by using the data provided through the LMS.)



1) List the areas where minor loss occurs in the above experimental system and the loss coefficients of each area.



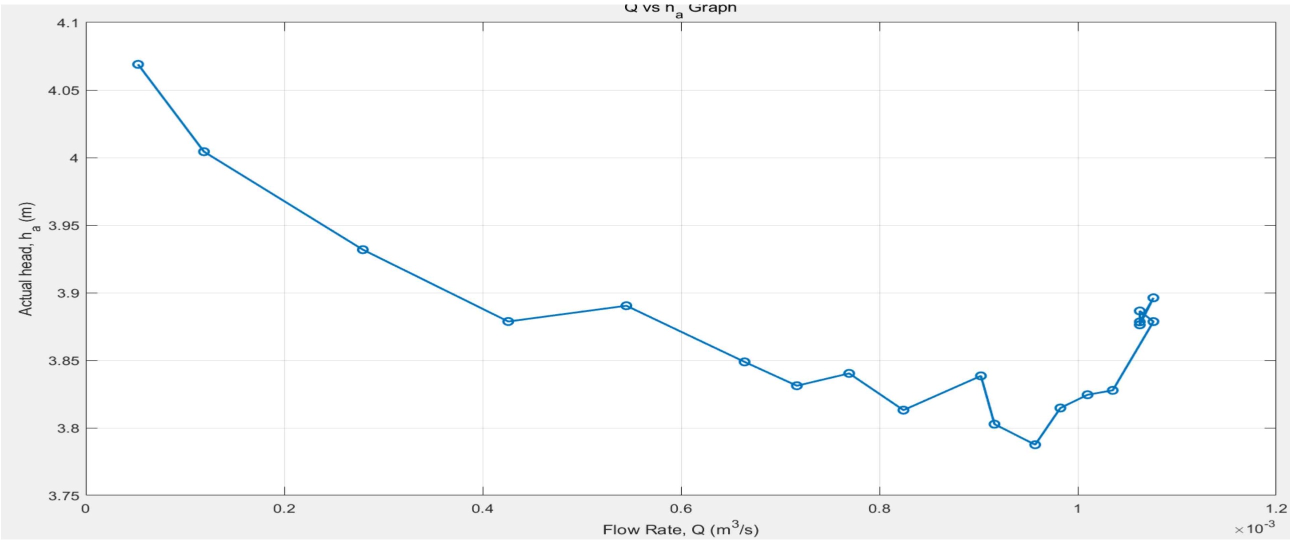
Component	K_L
a. Elbows	
Regular 90°, flanged	0.3
Regular 90°, threaded	1.5
Long radius 90°, flanged	0.2
Long radius 90°, threaded	0.7
Long radius 45°, flanged	0.2
Regular 45°, threaded	0.4

There are 4 points(k_1~k_4) where the minor loss occurs in the system. Loss coefficients values on each points are k_1=0.2, k_2=0.3, k_3=0.3 and k_4=1.0. Therefore, the amount of $k_L(\sum k_L)$ is 1.8.

2) Draw a performance curve (Actual head curve + Shaft curve + Efficiency curve) and explain the method.

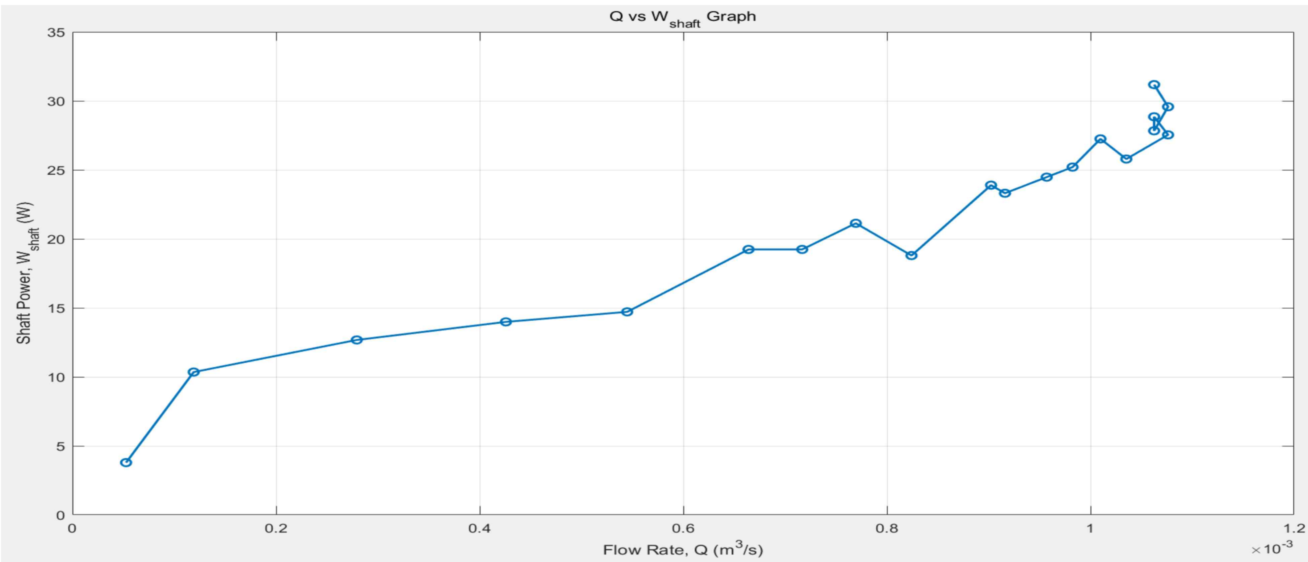
- Draw and explain the actual head curve.

$h_a = \frac{p_2 - p_1}{\gamma} + z_2 - z_1 + \frac{V^2{}_2 - V^2{}_1}{2g}$ ($z_2 - z_1$=2m, $\gamma=\rho g=\frac{9.81 * 997}{1000}$=9.78[kN/m^3]). I assumed that the temperature difference on each measurement is almost zero and calculated γ.



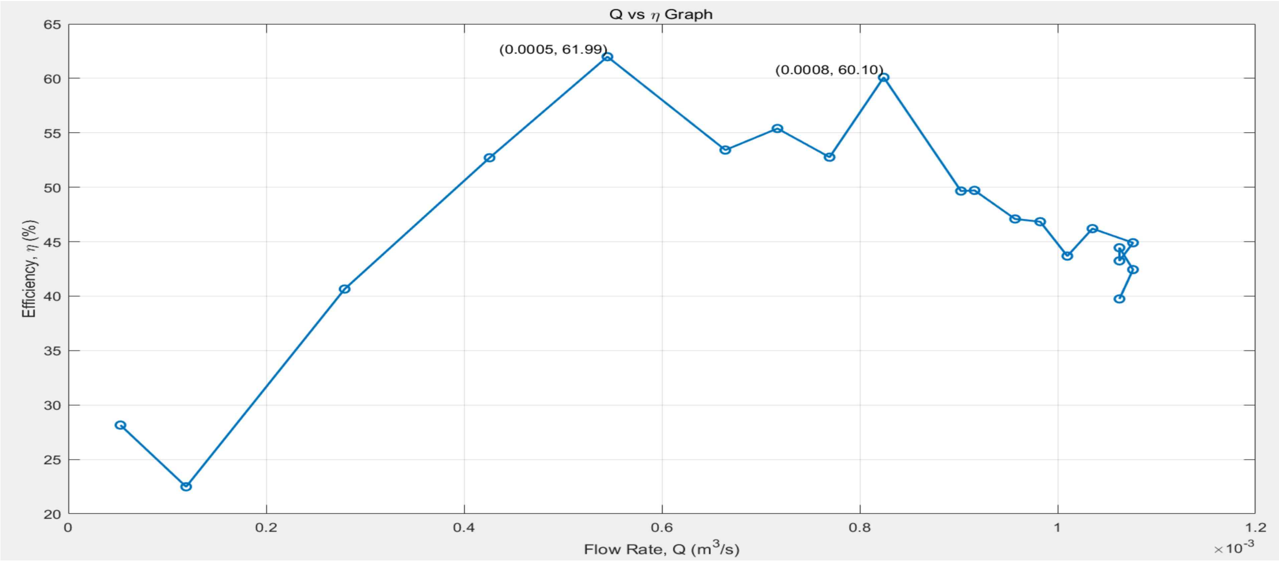
According to ideal actual head graph, the gradient should be decreased as gradually curve. However, this case is decreasing graph but it's not gradual decreasement curve. We can expect the reason of this difference is Re's rapid increasement by fluid's rapid increasement in the latter part. Additionally the graph looks like oscillation overally.

- Draw and explain the shaft curve.



$\dot{W}_{shaft} = T_{shaft}\omega * (\omega = \frac{RPM * 2\prod}{60})$. We can confirm Q and $\dot{W}_{shaft}$ are on almost linear relation and proportional increasement through graph picture.

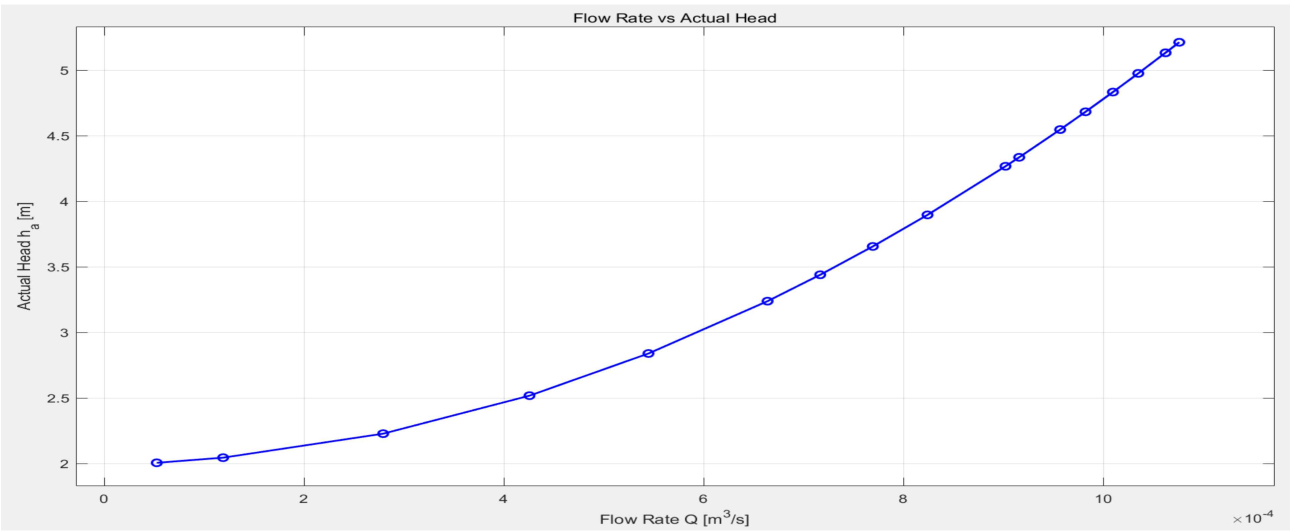
Draw and explain the efficiency curve.



$\eta = \dfrac{P_f}{W_{shaft}}$ and P_f equals $\gamma Q h_a$. Therefore, efficiency can be calculated as $\dfrac{\gamma Q h_a}{W_{shaft}} \times 100$ [%]. There is the picture Q vs efficiency graph below this sentence. In this picture, we can confirm the graph represents Q and η are on almost linear relation in the beginning.

Then, there is the maximum efficiency at the 5th point and the maximum efficiency value is 61.99%. Furthermore, another maximum efficiency point(9th point) can be observed and its value is 60.10%. According to theory, there should be only one maximum efficiency point. However, we are able to confirm there are two maximum efficiency points(BEP) in this graph. Additionally, the graph looks like oscillation overally.

3) Draw a system curve for the above system and describe the process of obtaining it.

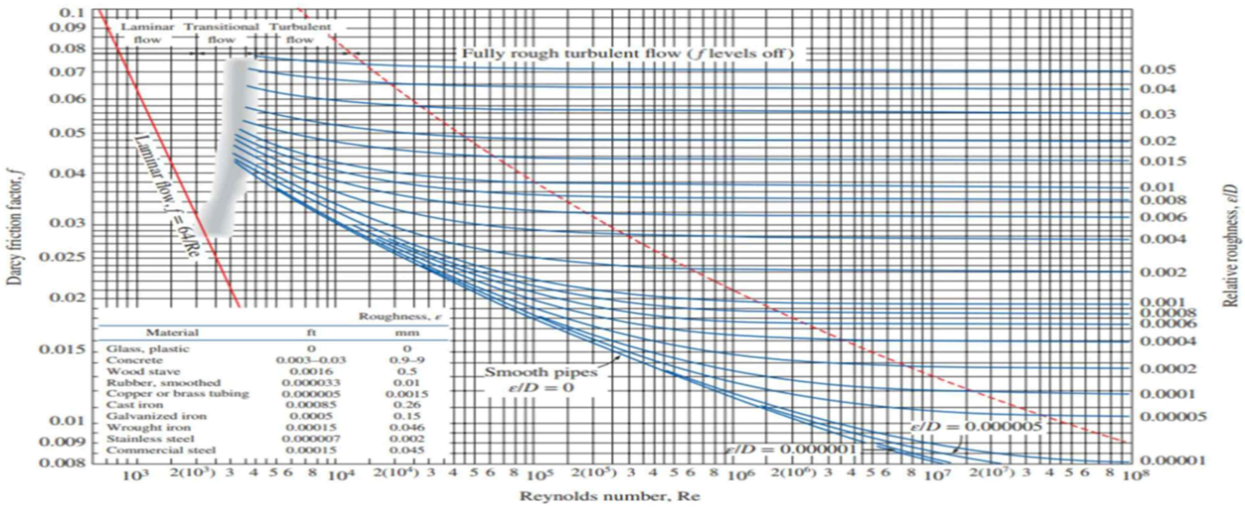


According to theory, system curve's $h_a=(z_2-z_1)+kQ^2$ ($k = \dfrac{f * \frac{l}{d} + \sum K_L}{2gA^2}$, $\sum K_L = 1.8$, $z_2 - z_1 = 2$[m]).

f is $\frac{64}{\mathrm{Re}}$ in laminar and transient flow. In turbulent flow, f is $\frac{0.25}{\log_{10}[(\frac{\varepsilon}{3.7})+(\frac{5.74}{\mathrm{Re}^{0.9}})]^2}$ from Haaland equation (ε=0.15[mm]). We can calculate Re by $\frac{\rho VD}{\mu}$. ρ can be calculated through $998.2+\frac{995.7-998.2}{(30-20)}(T_{measured}-20)$ [kg/m³] and μ can be calculated through $1.005.E-03+\frac{8.010.E-04-1.005.E-03}{(30-20)}(T_{measured}-20)$ [Pa/s] by linear interpolation. D is 0.02[m] and $V=\frac{4Q}{\prod(0.02/2)^2}$ [m/s]. When we observe the Q vs h_a graph, it look's like exponential function.

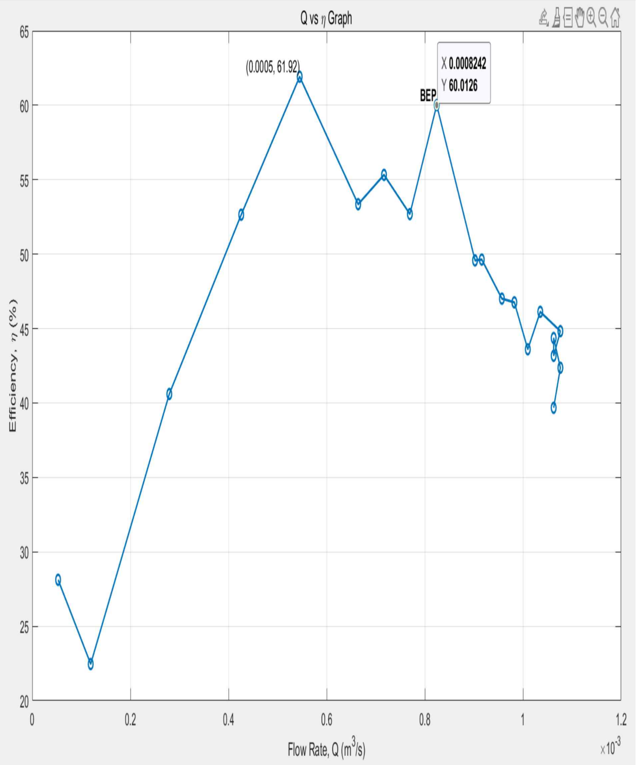
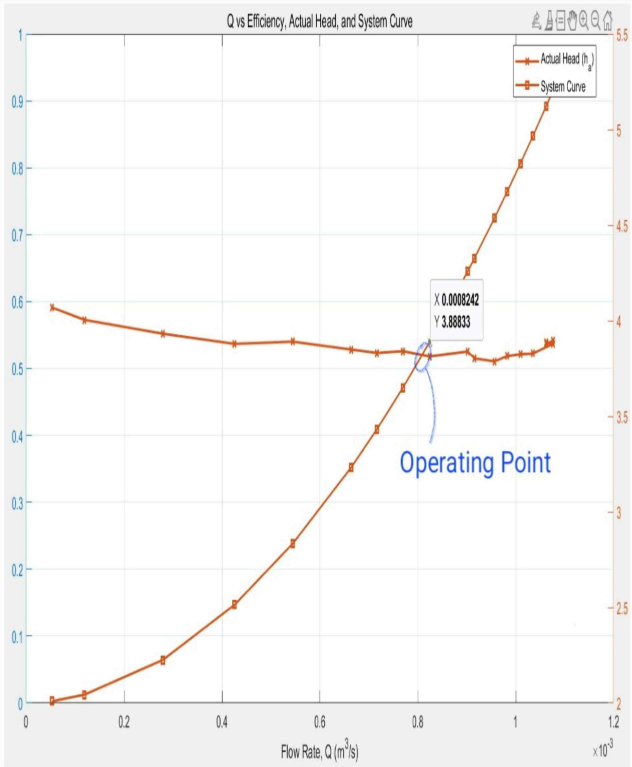
There are Re, ε/D and f calculation values for each data by following table.

Data number	Re	ε/D	f	Data number	Re	ε/D	f
1	3712.34	0.0075	0.01724	11	65035.41	0.0075	0.03589
2	8456.02	0.0075	0.04229	12	67717.11	0.0075	0.03584
3	19852.53	0.0075	0.03838	13	69514.41	0.0075	0.03581
4	30129.52	0.0075	0.03722	14	70816.10	0.0075	0.03579
5	38513.69	0.0075	0.03670	15	72678.37	0.0075	0.03576
6	47044.48	0.0075	0.03635	16	76582.38	0.0075	0.03570
7	50550.05	0.0075	0.03624	17	75266.93	0.0075	0.03572
8	54327.45	0.0075	0.03613	18	74929.45	0.0075	0.03572
9	58059.02	0.0075	0.03604	19	75980.77	0.0075	0.03571
10	63990.50	0.0075	0.03591	20	75097.80	0.0075	0.03572



From Moody's chart, we can know that cases of data number 2 ~ 20 almost located on the Moody's chart graph.

4) Find the BEP (highest pump efficiency point) and OP (operating point), and if they do not match, explain why. Also, find a solution to see what methods can be applied to make the two points coincide.

BEP	OP
	

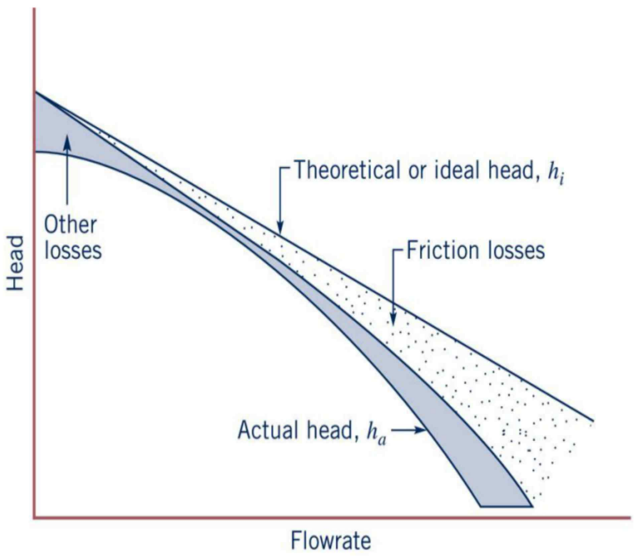
We can check the BEP and OP's Q points are almost same but it is not match 100%. It looks like OP is on the right side of BEP (=OP's Q is less than BEP's Q). We can define it definitely X=0.0008242 (= BEP) in OP picture is not OP's index point. To match BEP and OP completely, make the actual head's curve rapidly or make the system curve's gently. However, there are not independent component. Therefore, we need to choose head control by serial connection or flowrate control by parallel connection.

5) When comparing the ideal performance curve and the performance curve of the given data, the ideal performance curve appears linear, while the performance curve of the given data does not appear that way. Explain why.

As I mentioned on the upper page, the graphs had oscillation overally. We are able to expect why the curve cases are different with ideal cases and its causes can be analyzed on 'hydraulic losses', 'volumetric losses' and 'mechanical losses', the main causes of friction head.

About 'hydraulic losses', it is the main cause of friction head. Turbulent flow's extreme change depends on how much the inlet valve is opened and Re will increase extremely. In dimensional difference's perspective, hydraulic losses are 3-dimensional effects but we assumed it as 1-dimension and set epuation in 1-dimension. Therefore, the losses from dimesional difference (=error) are inevitable and it works as friction head. Additionally, the vortex occurred in blades and outlet valve and we can expect it interupted the flow. Due to it, we can find the twisted phenomenon in the second half of performance curve.

About ‘volumetric losses’, I expect it has occurred partial pressure difference in pump from clearance between impeller’s blades and casing. Other expected reason can be ‘leaked fluid’.
About ‘mechanical losses’, we can expect reasons as huge friction occurrence in bearing and seal’s bad attachment condition. About the more specific reason, it can be occurred because geometric tolerance was not satisfied the standard in design stage, assembly condition and component’s clearance.
Therefore, we can understand oscillation occurred because these friction head’s causes are not constant and it is not easy to predict these cause. In BEP’s perspective, it can be understood that it corresponds to the efficiency (BEP) that can be farthest due to these causes. The most efficient use of the equipment is possible when the pump is operated at the flow rate of the corresponding BEP.
In other perspective, its cause can be considered as head loss.



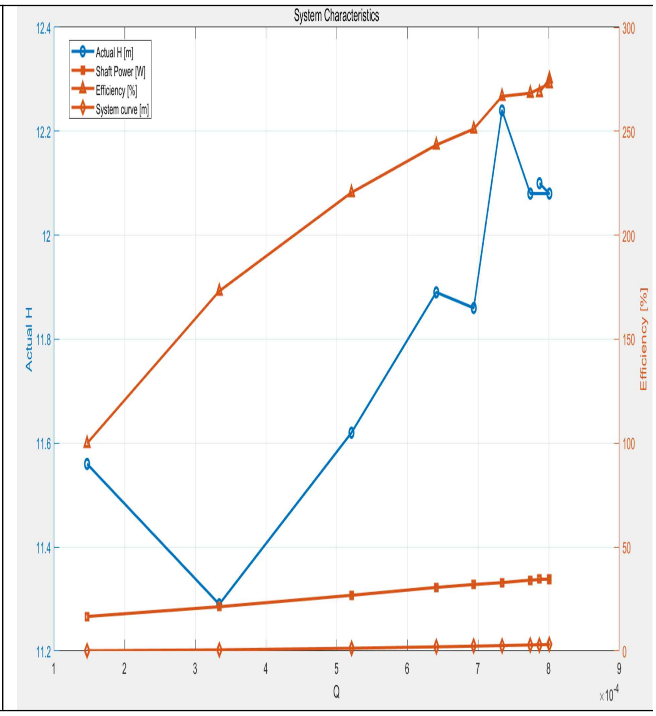
As I explained on theory, We have already arranged h_i (ideal head) $= \dfrac{U_2 * V_{\theta 2} - U_1 * V_{\theta 1}}{g} = h_{out} - h_{in} + h_L$. Then, h_i can be defined as $h_a + h_L$ (h_a = The actual rise in head of fluid). The curve, h_a versus flowrate becomes to lie below the ideal head curve. Additionally, we are able to know the h_a versus flowrate curve shows a nonlinear variation with flowrate. This nonlinear variation’s expected principles are ‘losses’. In other words, it means that h_i is on a linear relation with Q(flowrate) in ideal case but the real case is a little bit different due to friction and head losses.

5. Discussion

- Analysis of the speed control in centrifugal pump experiment [Case-1] results

i) [Case1-60] : Setting60 (900RPM)

N	Q	Actual H	Shaft Pow	Efficiency['	system curve[m]
1	0.000147	11.56	16.6	99.9	0.11
2	0.000334	11.29	21.3	173.1	0.55
3	0.000521	11.62	26.8	220.5	1.32
4	0.000641	11.89	30.6	243.4	2.00
5	0.000694	11.86	32.0	251.1	2.34
6	0.000734	12.24	32.9	266.8	2.62
7	0.000774	12.08	34.1	268.3	2.91
8	0.000801	12.08	34.7	272.9	3.11
9	0.000801	12.08	34.4	275.2	3.11
10	0.000787	12.10	34.7	268.6	3.01

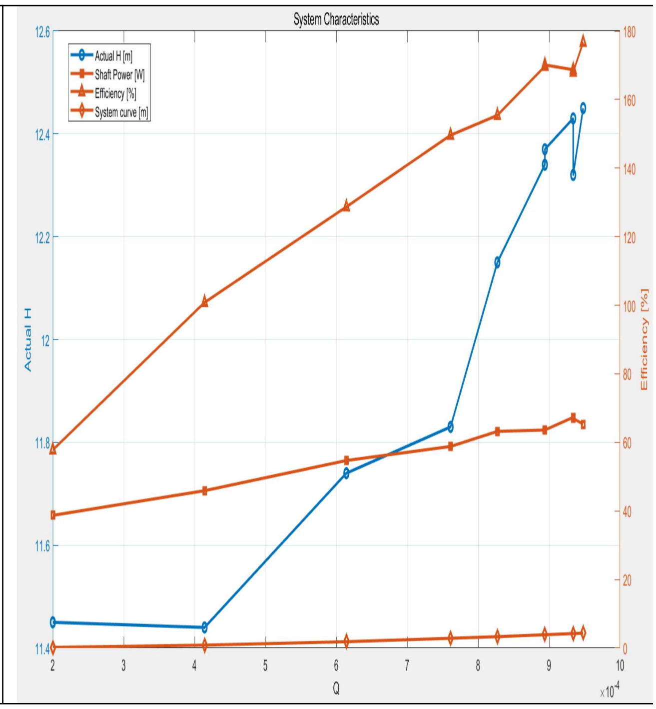


As we opened the valve, the flowrate, shaft power and system curve's values increased.

Flow rate at OP [m^3/s]	0.000801
Head at OP [m]	12.08
Shaft power at OP [W]	34.4
Efficiency at BEP [%]	275.2

ii) [Case1-70] : Setting70 (1050RPM)

N	Q	Actual H [	Shaft pow	Pump effi	System curve[m]
1	0.000200	11.45	38.7	57.8	0.20
2	0.000414	11.44	45.9	100.8	0.84
3	0.000614	11.74	54.7	128.7	1.84
4	0.000761	11.83	58.8	149.6	2.81
5	0.000827	12.15	63.2	155.4	3.32
6	0.000894	12.34	63.6	169.7	3.87
7	0.000894	12.37	63.6	170.1	3.87
8	0.000934	12.43	67.3	168.7	4.22
9	0.000934	12.32	67.0	168.0	4.22
10	0.000948	12.45	65.3	176.7	4.34

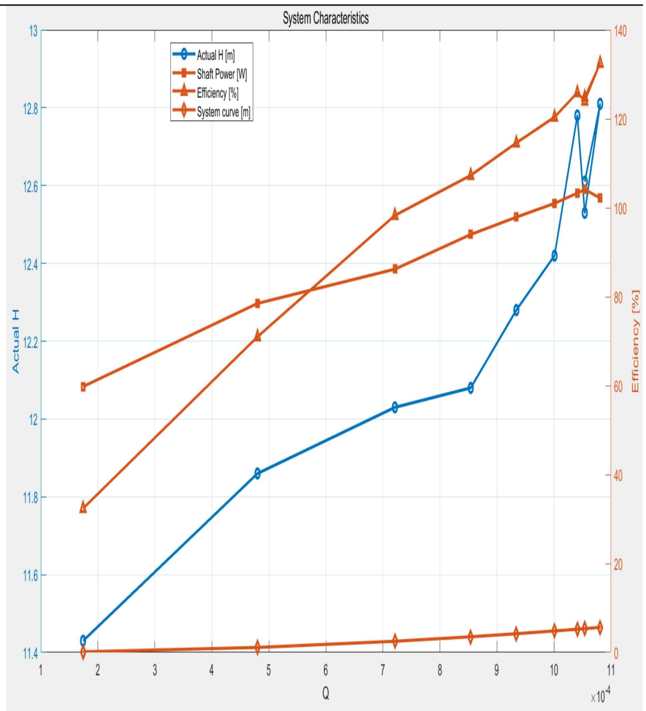


As we opened the valve, the flowrate, shaft power and system curve's values increased.

Flow rate at OP [m^3/s]	0.000894
Head at OP [m]	12.37
Shaft power at OP [W]	63.6
Efficiency at BEP [%]	170.1

iii) [Case1-80] : Setting80 (1200RPM)

N	Q	Actual H [	Shaft pow	Pump effi	System curve[m]
1	0.000174	11.43	59.8	32.4	0.15
2	0.000480	11.86	78.5	71.0	1.13
3	0.000721	12.03	86.2	98.3	2.52
4	0.000854	12.08	94.0	107.3	3.53
5	0.000934	12.28	97.9	114.6	4.22
6	0.001001	12.42	101.0	120.4	4.84
7	0.001041	12.78	103.3	125.9	5.23
8	0.001054	12.53	104.1	124.0	5.37
9	0.001081	12.81	102.2	132.5	5.64
10	0.001054	12.61	104.1	124.9	5.37



As we opened the valve, the flowrate, shaft power and system curve's values increased.

Flow rate at OP [m^3/s]	0.001081
Head at OP [m]	12.81
Shaft power at OP [W]	102.2
Efficiency at BEP [%]	132.5

- Consideration os Speed Control results

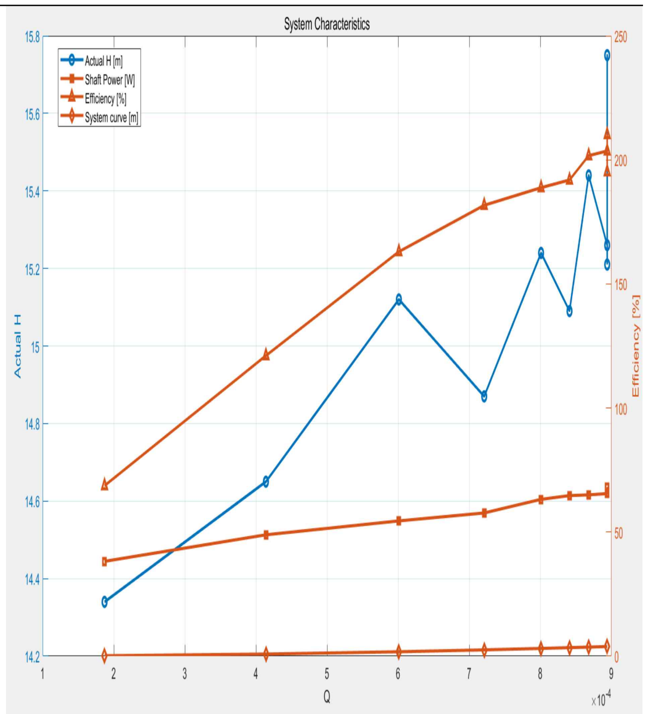
Case	Case1-60	Case1-70	Case1-80
RPM	900	1050	1200
Flow rate at OP [m^3/s]	0.000801	0.000894	0.001081
Head at OP [m]	12.08	12.37	12.81
Shaft power at OP [W]	34.4	63.6	102.2
Efficiency at BEP [%]	275.2	170.1	132.5

I have compared the case-1 experimental data values. Before specific discussion, I would like to explain first that 'operating point', the point of contact between two curves has not discovered in real experiment. Therefore, I had to assume and set OP where it has same Q value with BEP's. When RPM has increased, OP's flowrate has increased, either. Also, when RPM has increased, OP's head and shaft power have increased. However, when RPM has increased, efficiency at BEP has decreased. To reduce error(difference between OP and BEP), we need to fix pump's system structure by series or parallel connection.

- Analysis of the speed control in centrifugal pump experiment [Case-2] results
We conducted same experiment with different variable. The variable is type of impeller and week 2's experiment's impeller had shorter blade and different direction compared with week 1's impeller.

iv) [Case2-60] : Setting60 (900RPM)

N	Q	Actual H [	Shaft pow	Pump effi	System curve[m]
1	0.000187	14.34	38.2	68.7	0.17
2	0.000414	14.65	48.9	121.1	0.84
3	0.000601	15.12	54.5	163.0	1.76
4	0.000721	14.87	57.7	181.7	2.52
5	0.000801	15.24	63.2	188.8	3.11
6	0.000841	15.09	64.7	191.9	3.42
7	0.000868	15.44	65.0	201.7	3.64
8	0.000894	15.26	65.5	203.6	3.87
9	0.000894	15.75	65.5	210.2	3.87
10	0.000894	15.21	68.2	195.2	3.87

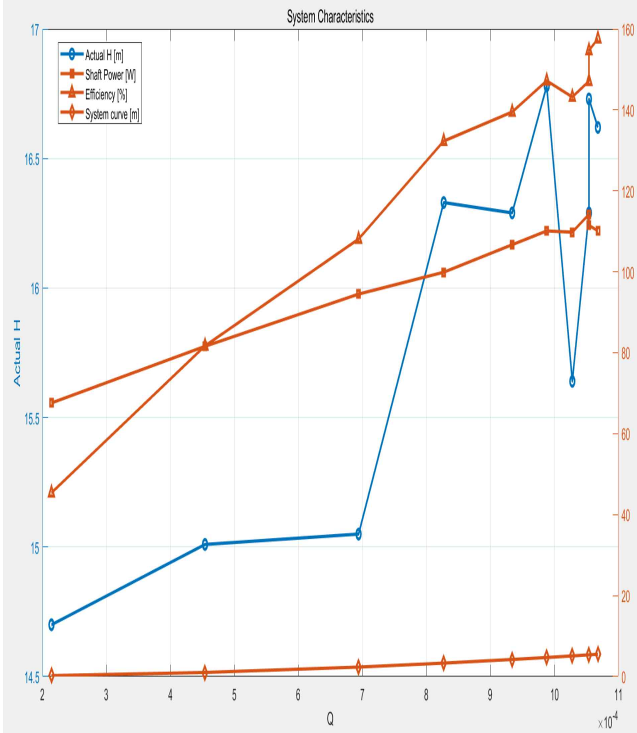


As we opened the valve, the flowrate, shaft power and system curve's values increased.

Flow rate at OP [m^3/s]	0.000894
Head at OP [m]	15.75
Shaft power at OP [W]	65.5
Efficiency at BEP [%]	210.2

v) [Case2-70] : Setting70 (1050RPM)

N	Q	Actual H	Shaft pow	Pump effi	System curve[m]
1	0.000214	14.70	67.6	45.4	0.23
2	0.000454	15.01	81.6	81.7	1.01
3	0.000694	15.05	94.5	108.1	2.34
4	0.000827	16.33	99.9	132.3	3.32
5	0.000934	16.29	106.7	139.5	4.22
6	0.000988	16.78	110.1	147.2	4.71
7	0.001028	15.64	109.8	143.2	5.10
8	0.001054	16.29	114.2	147.1	5.37
9	0.001054	16.73	111.5	154.7	5.37
10	0.001068	16.62	110.1	157.6	5.50

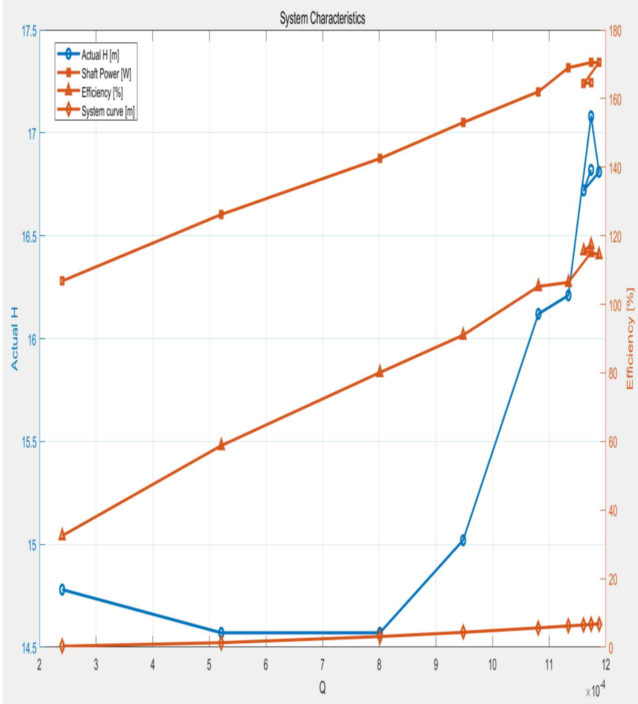


As we opened the valve, the flowrate, shaft power and system curve's values increased.

Flow rate at OP [m^3/s]	0.000988
Head at OP [m]	16.78
Shaft power at OP [W]	110.1
Efficiency at BEP [%]	147.2

vi) [Case2-80] : Setting80 (1200RPM)

N	Q	Actual H	Shaft pow	Pump effi	System curve[m]
1	0.000240	14.78	106.8	32.5	0.29
2	0.000521	14.57	126.2	58.8	1.32
3	0.000801	14.57	142.5	80.1	3.11
4	0.000948	15.02	153.0	91.0	4.34
5	0.001081	16.12	162.0	105.2	5.64
6	0.001134	16.21	169.0	106.4	6.20
7	0.001174	17.08	170.5	115.1	6.65
8	0.001188	16.81	170.5	114.5	6.80
9	0.001161	16.72	164.3	115.6	6.50
10	0.001174	16.82	164.7	117.3	6.65



As we opened the valve, the flowrate, shaft power and system curve's values increased.

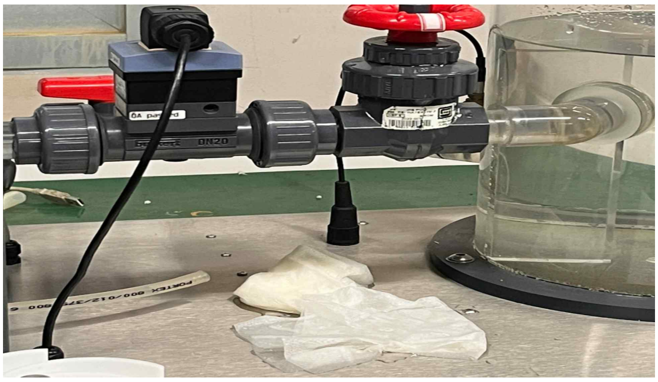
Flow rate at OP [m^3/s]	0.001188
Head at OP [m]	16.81
Shaft power at OP [W]	170.5
Efficiency at BEP [%]	115.1

- Consideration os Speed Control results

Case	Case2-60	Case2-70	Case2-80
RPM	900	1050	1200
Flow rate at OP [m^3/s]	0.000894	0.000988	0.001188
Head at OP [m]	15.75	16.78	16.81
Shaft power at OP [W]	65.5	110.1	170.5
Efficiency at BEP [%]	210.2	147.2	115.1

I have compared the case-2 experimental data values. Before specific discussion, I would like to explain first that 'operating point', the point of contact between two curves has not discovered in real experiment. Therefore, I had to assume set OP where it has same Q value with BEP's this time, too. Like case-1, when RPM has increased, OP's flowrate has increased, either. Also, when RPM has increased, OP's head and shaft power have increased. However, when RPM has increased, efficiency at BEP has decreased.

As theory, shaft power, system curve have increased by flow rate's increasement. Flow rate's increasement represents the more valve valve is opened and RPM's increasement. In fact, the actual head should be decreased due to friction loss between fluid and impeller when impeller rotates. However, all of cases don't show decreasement graph. These cases are definitely different with general case, theory. The expected reasons of this phenomenon could be less cavitation valve's position and condition. About less cavitation, in the initial stage of opening the valve, local resistance (vortex or cavitation) is reduced within the pipe or valve, allowing the pump to deliver higher pressure efficiently. Therefore, the actual head can be increased. About valve's position, if the valve is located immediately near the pump, the local flow characteristics induced by the pump may change when the valve is opened. In other words, turbulence generated by operating the valve may have been reduced, allowing the pump to deliver higher energy and it shows higher actual head.



About valve's condition, we can find the wet tissue under the pump's valve. The wet tissue means there was the water leak during the water's flow. We can't ignore this situation because it is unexpected variable. The water leak must not be occurred in ideal centrifugal pump. Therefore, there is the possibility of water leak affected this error and definitely different graph.

Case	Case1-60	Case1-70	Case1-80
RPM	900	1050	1200
Flow rate at OP [m^3/s]	0.000801	0.000894	0.001081
Head at OP [m]	12.08	12.37	12.81
Shaft power at OP [W]	34.4	63.6	102.2
Efficiency at BEP [%]	275.2	170.1	132.5

Case	Case2-60	Case2-70	Case2-80
RPM	900	1050	1200
Flow rate at OP [m^3/s]	0.000894	0.000988	0.001188
Head at OP [m]	15.75	16.78	16.81
Shaft power at OP [W]	65.5	110.1	170.5
Efficiency at BEP [%]	210.2	147.2	115.1

About BEP's efficiency, overal efficiency decreased when we replaced impeller previous one to shorter and opposite direction. What can be seen from this is that flow rate, head, shaft power and BEP's efficiency can be changed depending on the length and direction of the impeller. Namely, when we replaced impeller previous one to shorter and opposite direction, flow rate, head and shaft power increase with BEP's efficiency decreasement.

In conclusion, we need to look into various expected error causes in detail. However, we couldn't conduct it this time due to limited experiment time. There is the possibility of reduced error or getting satisfied values through multiple times of experiment. Also, to reduce error(difference between OP and BEP), we need to fix pump's system structure by series or parallel connection.