1875 SW 47th Terrace Apt. 3211
Gainesville, FL 32607
January 26, 2022

Caitlyn Simonds
University of Florida
201 Criser Hall PO Box 114000
Gainesville, FL 32611

Dear Ms. Simonds:

The main objective of this lab is to gather data on a centrifugal fan using backwards and radial impellers to characterize a fan system and how to engineer this system and impellers. This is accomplished by analyzing different inherent properties of a centrifugal fan system such as head rise, brake horsepower, water horsepower, fan efficiency, and the annual cost of running the fan system. These characteristics were all studied using variable rotational speeds of the shaft and were plotted on graphs to make the comparison and analysis more visual.

Utilizing the conservation of energy equation, the mass flow rate, net rate of heat transfer, net work on the system, the internal energy, the velocity of the flow, the volumetric flow rate, the kinetic energy, and potential energy of the flow can be determined. The internal energy and potential energy are distributed evenly so we can assume their change is negligible. The volumetric flow rate is also the same at the inlet and the outlet because the flow is incompressible, and the inlet and outlet have the same cross-sectional area. The fluid is also assumed to be incompressible. The law of the conservation of energy is used along with Bernoulli's equation to determine the pressure differences and head. This equation will help develop a head vs. flowrate curve to develop a performance curve of the centrifugal fan. This performance curve will work for any fluid that is in the pump. WHP is the water horsepower which is the work added to the flow from the fan. This is the density of the air times gravity, times the volumetric flow rate, and the head available. This equation will help develop a WHP vs. flowrate curve and the data will be used to find the annual cost of running the fan at different RPMS for a specific fan efficiency. BHP is the brake horsepower and is the torque of the shaft times the angular velocity. This will be used to develop a brake horsepower and water horsepower vs. flowrate curve to compare how much work from the shaft is added to the flow and how much it will cost to run the fan for a year. η is the fan efficiency which is the WHP divided by BHP. This is used to develop a graph which is the efficiency curve to see the percent of BHP actually contributed to the WHP.

The empirical methods that will be used to obtain the data needed for this lab will primarily be the VDAS system that is able to capture data from the fan at several points. Pressure sensors are placed to capture the air pressure at four different locations: the nozzle inlet; just before the impeller; after the centrifugal fan; and the ambient pressure of the room. Measurements for air temperature and impeller size will be taken using a thermometer and a standard 12-inch ruler. Data was captured by setting the desired rotational speed. The rotational speeds that were selected for this lab include 1000, 1500, 2000, 2500, and 3000 rpm. After a rotational speed was selected, a slide valve was adjusted from 100% to 0% in increments of 10% at each rotational speed and data was captured at each point. Before any data acquisition occurred, the flow was allowed to stabilize for five seconds to ensure accurate data collection. Eye protection, pants, and close-toed shoes were worn to ensure proper lab safety when in the lab. The data was then analyzed and used to produce the results in our report.

The attempt to gather data on a centrifugal fan featuring different impellers to determine several different properties was done successfully. For the backwards impeller data resulted in head rise was between 8 meters and 88 meters, fan efficiency was between 0% and 81.09%, and the annual cost of running the fan was between $2,816.16 and $85,637.19. When looking at the radial impeller, the head rise was between 9 meters and 102 meters, the fan efficiency was between 0% and 73.50%, and the annual cost of running the fan was between $3,949.58 and $105,440.81.

Sincerely,

Cyril Moran   Nickolas Saavedra

"On my honor, I have neither given nor received unauthorized aid in doing this assignment."

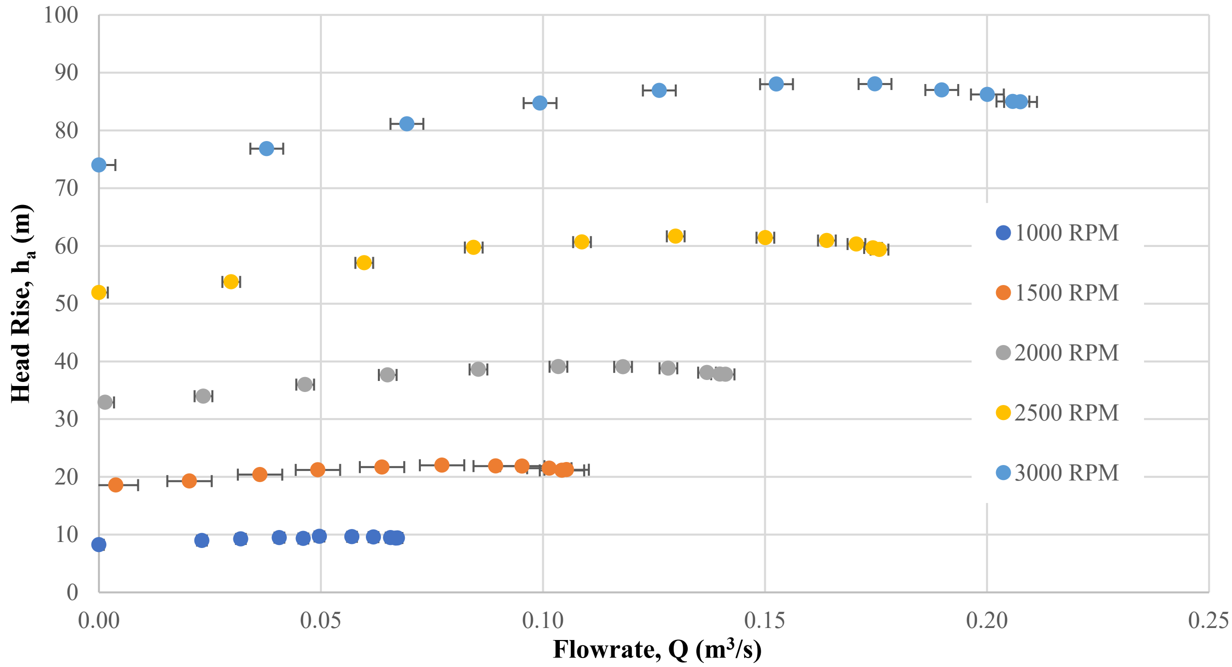

Figure 1: The amount of head added to the flow at a specific rotational speed was plotted vs. flowrate which was measured from the VDAS software. The head increases until the maximum value where it begins to have a negative slope illustrates that pumps generate high flow rates when pumping against low-pressure head but as the pressure-head increases the flow rate decreases. This is due to the fact that as the flowrate increases the pressure at the inlet and outlet decrease respectively which results in a decrease in head.

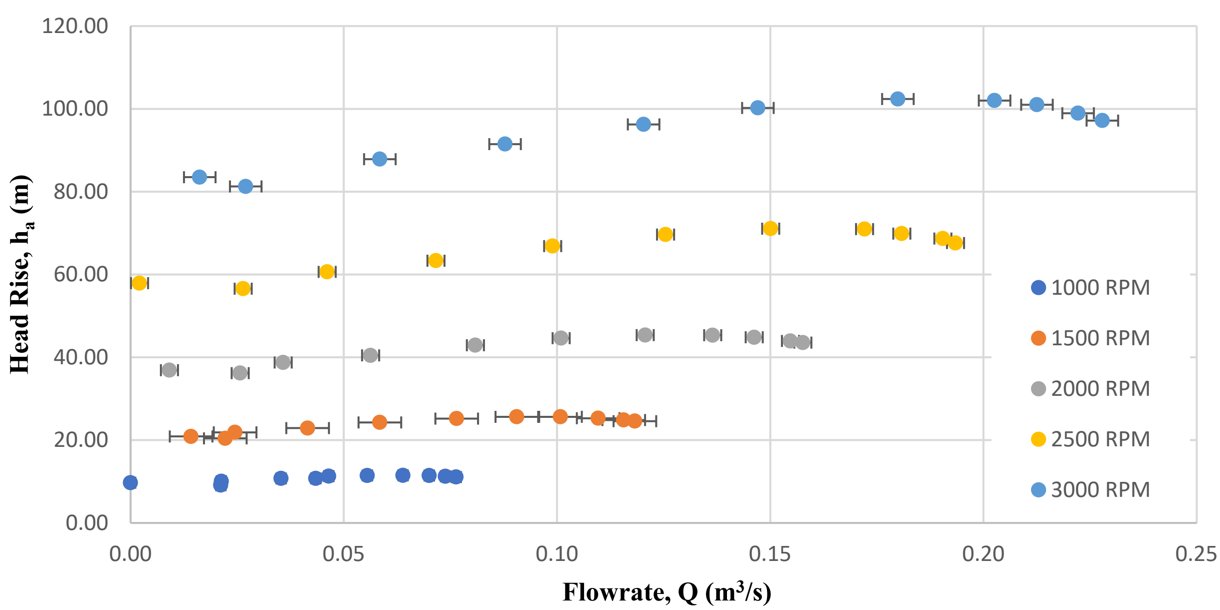

Figure 2: The amount of head added to the flow at a specific rotational speed was plotted vs. flowrate which was measured from the VDAS software. The head increases until the maximum value where it begins to have a negative slope illustrates that pumps generate high flow rates when pumping against low-pressure head but as the pressure-head increases the flow rate decreases. This is due to the fact that as the flowrate increases the pressure at the inlet and outlet decrease respectively which results in a decrease in head. In lower flowrates the head varies which can be due to the impeller shape and an increase in pressure when pumping against specific head values.

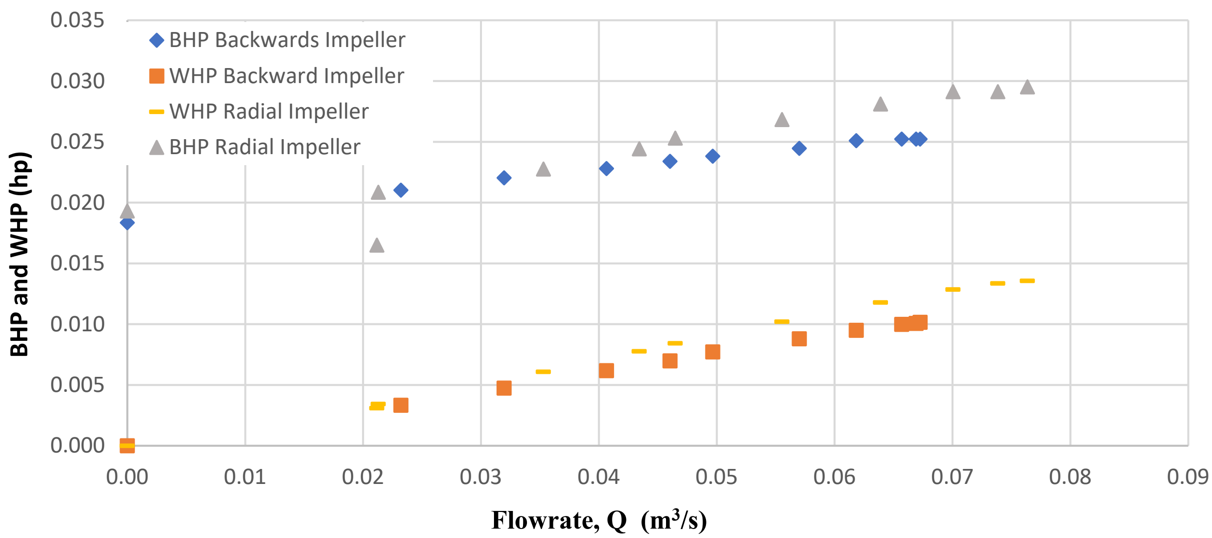

Figure 3: WHP and BHP data was calculated using the density of air, the flowrate, the shaft torque, the rpm, gravity, and the head available. This was plotted vs. flowrate (which was calculated from the VDAS software) and illustrates that the BHP for both impellers are orders of magnitude higher than the WHP for both impellers for the 1000 rpm motor speed. This is due to the inefficiencies within the pump that result in lower WHP.

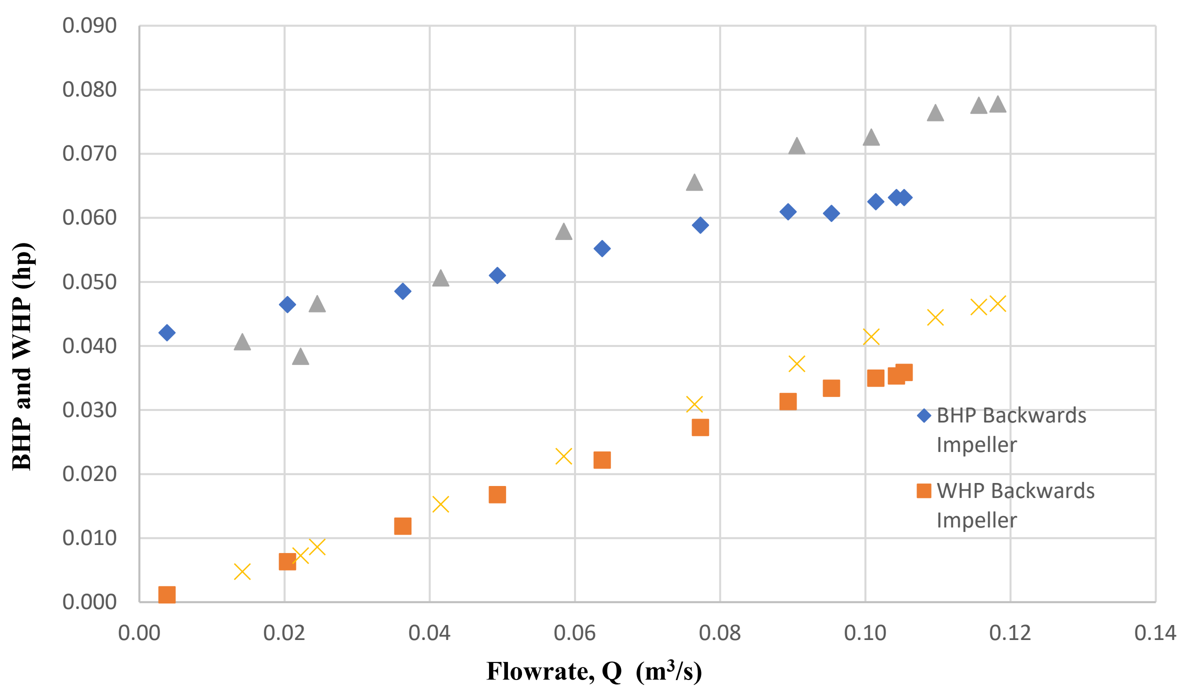

Figure 4: WHP and BHP data was calculated using the density of air, the flowrate, the shaft torque, the rpm, gravity, and the head available. The results were plotted vs. flowrate of the fluid which was measured by the VDAS software. The BHP for both impellers was magnitudes higher than the WHP and the increase in BHP for an increased flowrate illustrates how the flowrate increases with a steady increase of torque which BHP is dependent on. Lower values of WHP can be a result of inefficiencies within the pump.

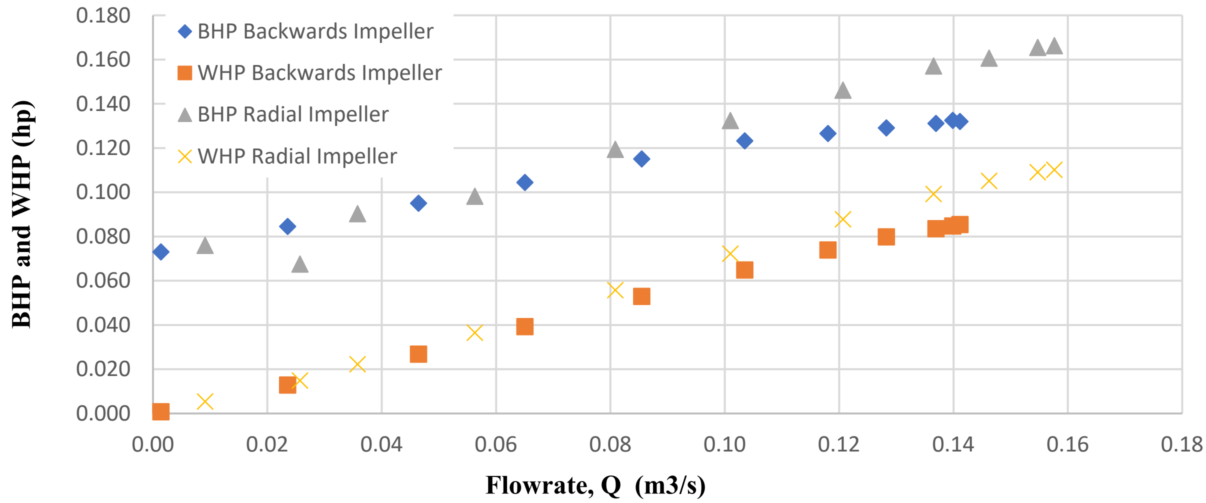

Figure 5: BHP and WHP data was calculated density of air, the flowrate, the shaft torque, the rpm, gravity, and the head available. The data was then plotted vs. flowrate which was measured from the VDAS software. The trends of the graph illustrate that BHP is magnitudes higher than the WHP for both impellers. Also, the linear profile of the BHP is consistent with an increasing flowrate because a higher flowrate produces a higher torque which is a variable of BHP. Lower values of WHP can be a result of inefficiencies within the pump.

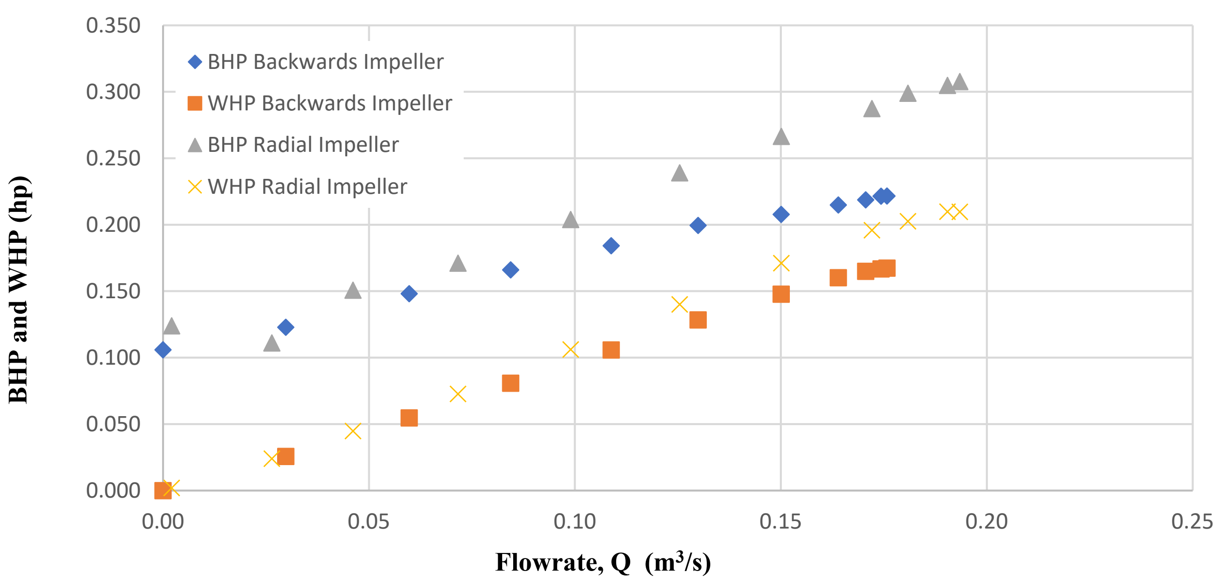

Figure 6: BHP and WHP data was calculated density of air, the flowrate, the shaft torque, the rpm, gravity, and the head available. The data was then plotted vs. flowrate which was measured from the VDAS software. The WHP is orders of magnitude smaller than the BHP and its slope is much smaller than BHP since BHP is dependent on rpm and torque. Lower values of WHP can be a result of inefficiencies within the pump.

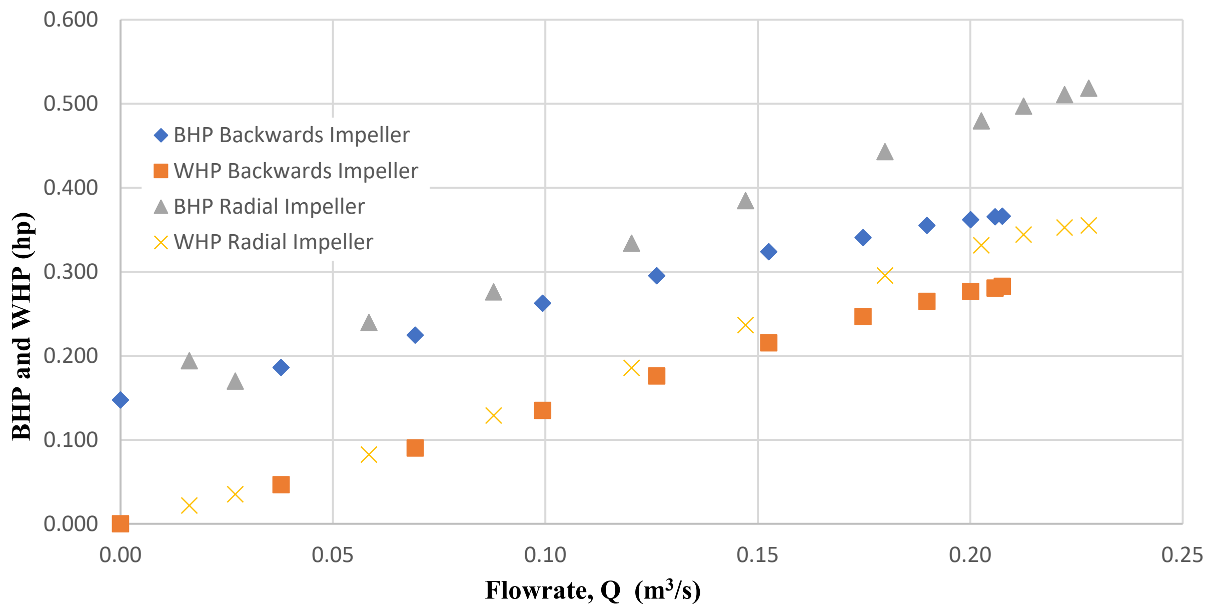

Figure 7: BHP and WHP data was calculated density of air, the flowrate, the shaft torque, the rpm, gravity, and the head available. The data was plotted vs. flowrate which was given from the VDAS software. The trends illustrates that the BHP for both impellers is much greater than the WHP. The WHP for the radial impeller is also greater than that of the backwards impeller demonstrating a higher efficiency since the BHP of the backwards impeller is greater than that of the radial impeller. Lower values of WHP can be a result of inefficiencies within the pump.

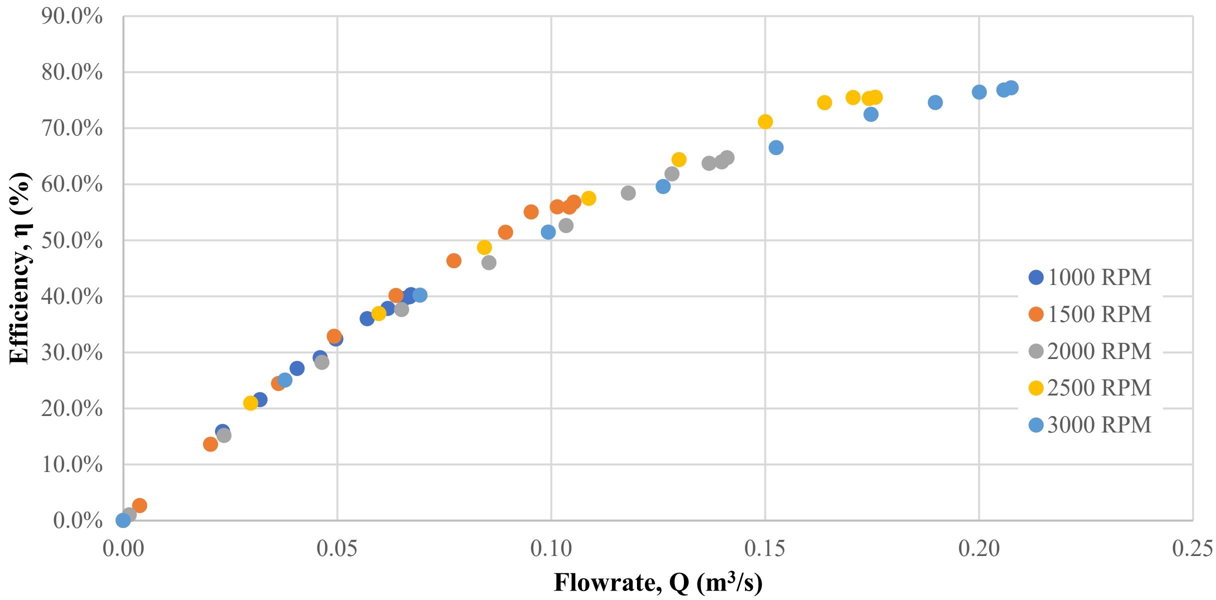

Figure 8: The efficiency of the backwards impeller is plotted vs. the flowrate (which was found using the VDAS software). The efficiency was calculated by dividing the WHP by the BHP. The graph demonstrates exponential type behavior with a high slope in the beginning and a decrease in slope as flowrate increases. This illustrates that the efficiency of power being transferred to the fluid will increase to a maximum value but will plateau meaning that there is the most efficient flowrate value.

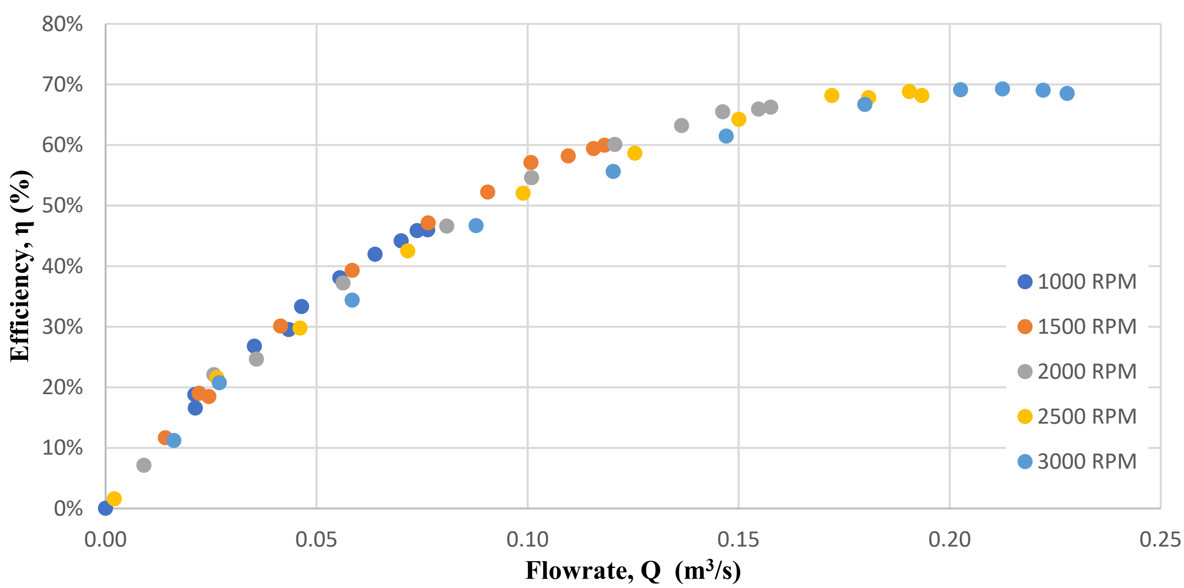

Figure 9: The efficiency of the radial impeller is plotted vs. the flowrate (which was found using the VDAS software). The efficiency was calculated by dividing the WHP by the BHP. The graph demonstrates exponential type behavior with a high slope in the beginning and a decrease in slope as flowrate increases. This illustrates that the efficiency of power being transferred to the fluid will increase to a maximum value but will plateau meaning that there is the most efficient flowrate value.

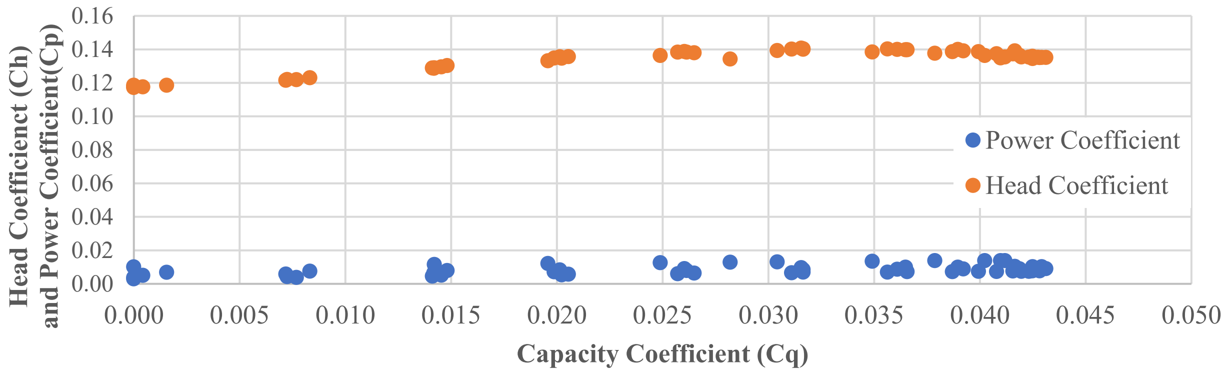

Figure 10: The head coefficient and power coefficient are plotted vs. capacity coefficient. These are non-dimensional parameters which allows us to use this data for much bigger models and scales. The C_P has a linear relationship with C_Q which illustrates that the power added to flow is increased as the volume of water will flow over a pressure drop. The C_h has a parabolic profile which illustrates that the head added to the flow will reach a maximum value.

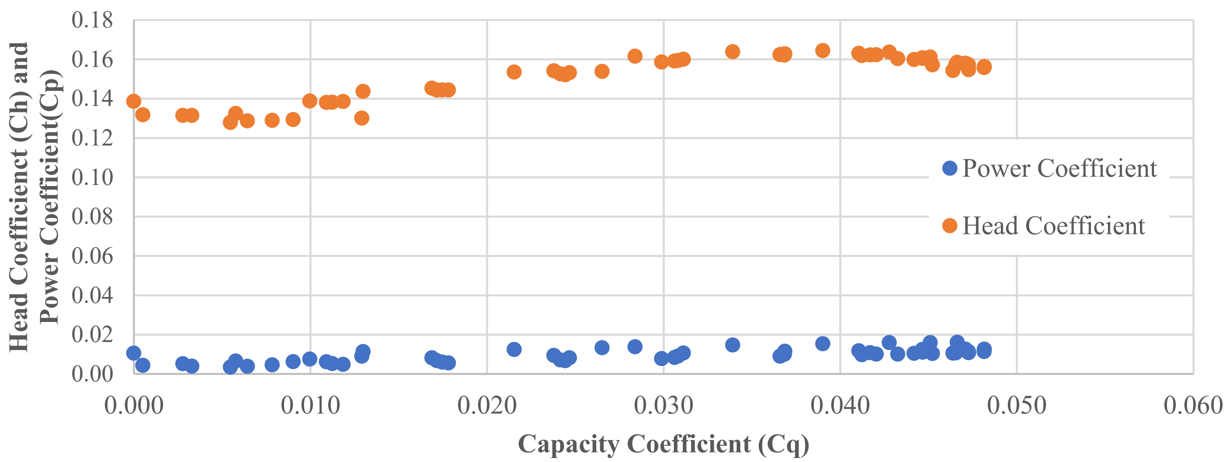

Figure 11: The head coefficient and power coefficient are plotted vs. capacity coefficient. These are non-dimensional parameters which allows us to use this data for much bigger models and scales. The C_p has a linear relationship with C_Q which illustrates that the power added to flow is increased as the volume of water will flow over a pressure drop. The C_h has a parabolic profile which illustrates that the head added to the flow will reach a maximum value.

Table 1: The amount of money it will cost to run the centrifugal fan each year for 2800 hours at 80% efficiency. The annual cost will increase exponentially because BHP increases exponentially comparatively to the WHP which results in the annual cost growing rapidly for an increase in BHP.

Shaft Rotational Speed, ω (rev/min)	**BHP Required, BHP_R (hp)**	**Annual Cost ($)**
1000	0.01258	3.94
1500	0.04416	13.83
2000	0.10678	33.45
2500	0.20843	65.28
3000	0.35072	109.84

Table 2: Experimental data and calculated values for backward impeller and radial impeller are showcased in tables 2 through table 11 below. The fan speed was measured from the VDAS software. The torque was measured from the VDAS software. The total power was measured from the VDAS software. All the pressure values were measured from the VDAS software. The volumetric flow rate was measured from the VDAS software. The mechanical power loss was measured from the current rpm value. The head rise was calculated from the difference in P_3 and P_2 divided by the specific gravity of air. The fan efficiency was calculated from the brake horsepower and water horsepower. The following table is the data collected and calculated from the backwards impeller at 1000 rpm.

Slider Valve (%)	**Speed ω (rev/min)**	**Torque T (N·m)**	**Total Power P_T (W)**	**ΔP_1 (Pa)**	**P_2 (Pa)**	**P_3 (Pa)**	**Flowrate Q (m^3/s)**	**Head h_a (m)**	**Efficiency η**	**Power Loss P_L (W)**
100	998.00	0.18	19.00	-150.00	-97.00	15.18	0.07	9.38	0.40	4
90	998.00	0.18	19.00	-151.54	-98.36	14.27	0.07	9.42	0.40	4
80	998.00	0.18	19.00	-144.00	-92.91	20.45	0.07	9.48	0.40	4
70	998.00	0.18	19.00	-128.00	-82.91	31.55	0.06	9.57	0.38	4
60	998.00	0.17	18.45	-108.73	-69.91	45.18	0.06	9.62	0.36	4
50	998.00	0.17	18.00	-82.55	-56.64	59.18	0.05	9.69	0.32	4
40	998.00	0.16	17.27	-55.27	-38.82	74.55	0.04	9.48	0.27	4
30	998.00	0.16	16.73	-34.18	-24.45	86.27	0.03	9.26	0.22	4
20	998.00	0.15	16.00	-18.00	-12.82	94.45	0.02	8.97	0.16	4
10	998.00	0.17	15.64	-6.00	-54.34	57.24	0.05	9.33	0.29	4
0	998.00	0.13	14.09	0.00	4.36	102.91	0.00	8.24	0.00	4

Table 3: The data collected and calculated for the backwards impeller at 1500 rpm.

Slider Valve (%)	Speed ω (rev/min)	Torque T (N·m)	Total Power P_T (W)	ΔP_1 (Pa)	P_2 (Pa)	P_3 (Pa)	Flowrate Q (m^3/s)	Head h_a (m)	Efficiency η	Power Loss P_L (W)
100	1500.00	0.30	47.00	-363.82	-214.64	38.00	0.10	21.13	0.56	5
90	1500.00	0.30	47.00	-371.27	-218.00	36.00	0.11	21.24	0.57	5
80	1489.18	0.30	46.55	-344.18	-205.91	51.18	0.10	21.50	0.56	5
70	1495.73	0.29	45.83	-304.00	-184.64	76.73	0.10	21.86	0.55	5
60	1497.00	0.29	46.00	-263.27	-160.73	100.82	0.09	21.87	0.51	5
50	1497.00	0.28	43.91	-194.18	-124.55	138.73	0.08	22.02	0.46	5
40	1497.00	0.26	41.18	-130.73	-88.55	170.91	0.06	21.70	0.40	5
30	1497.00	0.24	38.27	-77.27	-54.27	199.36	0.05	21.21	0.33	5
20	1497.00	0.23	37.00	-41.27	-30.45	213.36	0.04	20.39	0.24	5
10	1498.00	0.22	34.00	-12.00	-8.36	222.09	0.02	19.27	0.14	5
0	1498.00	0.20	30.82	0.55	10.18	232.27	0.00	18.57	0.03	5

Table 4: The data collected and calculated for the backwards impeller at 2000 rpm.

Slider Valve (%)	Speed ω (rev/min)	Torque T (N·m)	Total Power P_T (W)	ΔP_1 (Pa)	P_2 (Pa)	P_3 (Pa)	Flowrate Q (m^3/s)	Head h_a (m)	Efficiency η	Power Loss P_L (W)
100	2000.36	0.47	100.00	-666.55	-382.18	69.27	0.14	37.75	0.65	9
90	2000.00	0.47	100.36	-655.27	-387.18	64.64	0.14	37.78	0.64	9
80	1999.00	0.47	99.45	-628.36	-359.45	95.55	0.14	38.05	0.64	9
70	1999.00	0.46	98.00	-550.73	-319.45	144.64	0.13	38.81	0.62	9
60	1999.18	0.45	95.27	-466.55	-275.82	191.18	0.12	39.05	0.58	9
50	1999.00	0.44	91.82	-358.36	-216.18	251.09	0.10	39.07	0.53	9
40	1999.00	0.41	86.00	-244.55	-153.27	308.36	0.09	38.60	0.46	9
30	2000.00	0.37	78.36	-141.45	-91.00	359.27	0.07	37.65	0.38	9
20	2001.00	0.34	71.18	-72.18	-47.18	383.09	0.05	35.98	0.28	9
10	2001.00	0.30	63.18	-18.55	-11.82	394.45	0.02	33.97	0.15	9
0	2001.73	0.26	55.00	0.36	20.73	414.09	0.00	32.89	0.01	9

Table 5: The data collected and calculated for the backwards impeller at 2500 rpm.

Slider Valve (%)	Speed ω (rev/min)	Torque T (N·m)	Total Power P_T (W)	ΔP_1 (Pa)	P_2 (Pa)	P_3 (Pa)	Flowrate Q (m^3/s)	Head h_a (m)	Efficiency η	Power Loss P_L (W)
100	2508.00	0.63	165.82	-1018.36	-602.73	110.55	0.17	59.65	0.75	12
90	2509.00	0.63	165.82	-1035.27	-607.45	102.36	0.18	59.36	0.76	12
80	2509.00	0.62	164.18	-974.91	-574.36	147.00	0.17	60.32	0.75	12
70	2509.00	0.61	161.00	-900.36	-507.73	220.91	0.16	60.93	0.75	12
60	2509.00	0.59	157.00	-754.36	-427.45	307.00	0.15	61.42	0.71	12
50	2509.00	0.57	148.27	-565.27	-327.55	409.82	0.13	61.66	0.64	12
40	2510.00	0.52	137.82	-396.55	-232.55	492.82	0.11	60.66	0.57	12
30	2510.64	0.47	125.36	-238.55	-148.27	565.82	0.08	59.71	0.49	12
20	2511.45	0.42	110.91	-119.82	-75.73	606.64	0.06	57.06	0.37	12
10	2512.36	0.35	93.45	-29.64	-20.55	622.36	0.03	53.76	0.21	12
0	2514.00	0.30	79.00	0.00	32.27	653.18	0.00	51.92	0.00	12

Table 6: The data collected and calculated for the backwards impeller at 3000 rpm.

Slider Valve (%)	Speed ω (rev/min)	Torque T (N·m)	Total Power P_T (W)	ΔP_1 (Pa)	P_2 (Pa)	P_3 (Pa)	Flowrate Q (m^3/s)	Head h_a (m)	Efficiency η	Power Loss P_L (W)
100	2998.09	0.87	273.45	-1419.27	-860.00	156.55	0.21	85.01	0.77	14
90	2998.00	0.87	274.00	-1444.00	-871.91	143.73	0.21	84.93	0.77	14
80	2998.00	0.86	271.00	-1341.09	-809.36	221.64	0.20	86.22	0.76	14
70	2998.00	0.84	266.09	-1206.55	-729.91	310.36	0.19	86.99	0.75	14
60	2998.00	0.81	256.73	-1022.91	-601.18	451.55	0.17	88.03	0.72	14
50	2999.00	0.77	241.73	-780.00	-453.00	599.36	0.15	88.00	0.66	14
40	3000.00	0.70	222.27	-533.45	-316.91	722.45	0.13	86.92	0.60	14
30	3001.55	0.62	196.82	-330.73	-197.27	815.91	0.10	84.73	0.51	14
20	3003.00	0.53	167.82	-161.27	-101.64	868.55	0.07	81.13	0.40	14
10	3004.36	0.44	138.36	-48.00	-31.36	887.45	0.04	76.83	0.25	14
0	3006.82	0.35	111.82	0.00	46.45	931.45	0.00	74.01	0.00	14

Table 7: Radial Impeller Data for 1000 RPM

Slider Valve (%)	Speed ω (rev/min)	Torque T (N·m)	Total Power P_T (W)	ΔP_1 (Pa)	P_2 (Pa)	P_3 (Pa)	Flowrate Q (m^3/s)	Head h_a (m)	Efficiency η	Power Loss P_L (W)
100%	1001	0.21	22	-198	-115	18	0.08	4	11	55.60
90%	1001	0.21	22	-186	-107	27	0.07	4	11	55.60
80%	1001	0.21	22	-168	-97	40	0.07	4	11	55.60
70%	1001	0.19	20	-140	-81	56	0.06	4	11	56.30
60%	1001	0.19	20	-104	-65	73	0.06	4	12	50.00
50%	1001	0.18	19	-72	-47	88	0.05	4	11	40.00
40%	1001	0.16	17	-42	-27	101	0.04	4	11	38.50
30%	1002	0.15	16	-16	-12	108	0.02	4	10	25.00
20%	1002	0.14	15	0	-2	114	0.00	4	10	0.00
10%	1002	0.13	14	12	6	119	0.02	4	9	20.00
0%	1002	0.12	13	16	14	122	0.02	4	9	22.20

Table 8: Radial Impeller Data for 1500 RPM

Slider Valve (%)	Speed ω (rev/min)	Torque T (N·m)	Total Power P_T (W)	ΔP_1 (Pa)	P_2 (Pa)	P_3 (Pa)	Flowrate Q (m^3/s)	Head h_a (m)	Efficiency η	Power Loss P_L (W)
100%	1501	0.37	59	-470	-255	40	0.12	5	25	64.80
90%	1501	0.37	59	-450	-242	54	0.12	5	25	63.00
80%	1501	0.36	57	-402	-219	84	0.11	5	25	63.50
70%	1501	0.34	54	-338	-189	119	0.10	5	26	63.30
60%	1501	0.34	53	-270	-153	153	0.09	5	26	56.30
50%	1502	0.31	49	-192	-109	193	0.08	5	25	52.30
40%	1502	0.27	43	-110	-68	223	0.06	5	24	44.70
30%	1503	0.24	39	-54	-36	236	0.04	5	23	32.40
20%	1503	0.22	34	-18	-14	248	0.02	5	22	20.70
10%	1503	0.19	30	6	7	256	0.01	5	21	12.00
0%	1504	0.18	28	18	21	266	0.02	5	20	26.10

Table 9: Radial Impeller Data for 2000 RPM

Slider Valve (%)	Speed ω (rev/min)	Torque T (N·m)	Total Power P_T (W)	ΔP_1 (Pa)	P_2 (Pa)	P_3 (Pa)	Flowrate Q (m^3/s)	Head h_a (m)	Efficiency η	Power Loss P_L (W)
100%	2001	0.59	125	-846	-448	76	0.16	9	44	71.60
90%	2001	0.58	123	-820	-433	93	0.16	9	44	71.90
80%	2001	0.58	123	-746	-389	145	0.15	9	45	69.30
70%	2001	0.56	117	-628	-337	207	0.14	9	45	68.50
60%	2002	0.52	109	-496	-266	276	0.12	9	45	66.00
50%	2002	0.47	100	-346	-191	342	0.10	9	45	59.30
40%	2003	0.41	86	-204	-120	392	0.08	9	43	51.90
30%	2004	0.35	75	-106	-66	419	0.06	9	41	40.90
20%	2005	0.33	69	-44	-30	433	0.04	9	39	28.30
10%	2005	0.28	59	2	7	450	0.01	9	37	6.00
0%	2005	0.24	52	22	34	467	0.03	9	36	25.60

Table 10: Radial Impeller Data for 2500 RPM

Slider Valve (%)	Speed ω (rev/min)	Torque T (N·m)	Total Power P_T (W)	ΔP_1 (Pa)	P_2 (Pa)	P_3 (Pa)	Flowrate Q (m^3/s)	Head h_a (m)	Efficiency η	Power Loss P_L (W)
100%	2502	0.88	231	-1344	-723	84	0.20	12	67	73.50
90%	2501	0.87	228	-1220	-686	135	0.19	12	69	72.20
80%	2502	0.85	224	-1164	-623	213	0.19	12	70	73.10
70%	2503	0.81	214	-972	-545	303	0.17	12	71	71.30
60%	2503	0.76	200	-722	-419	432	0.15	12	71	66.00
50%	2504	0.68	178	-546	-296	537	0.13	12	70	63.90
40%	2504	0.58	154	-336	-183	614	0.10	12	67	55.60
30%	2506	0.48	127	-170	-102	654	0.07	12	63	47.00
20%	2508	0.43	113	-72	-48	679	0.05	12	61	33.70
10%	2509	0.36	96	0	6	698	0.00	12	58	0.00
0%	2509	0.31	82	24	50	719	0.03	12	56	25.70

Table 11: Radial Impeller Data for 3000 RPM

Slider Valve (%)	Speed ω (rev/min)	Torque T (N·m)	Total Power P_T (W)	ΔP_1 (Pa)	P_2 (Pa)	P_3 (Pa)	Flowrate Q (m^3/s)	Head h_a (m)	Efficiency η	Power Loss P_L (W)
100%	3002	1.23	387	-1804	-1044	119	0.23	14	97	71.80
90%	3002	1.21	381	-1590	-986	196	0.22	14	99	69.80
80%	3003	1.18	373	-1498	-893	310	0.21	14	101	70.50
70%	3005	1.14	358	-1402	-769	450	0.20	14	102	72.10
60%	3005	1.05	332	-1104	-594	625	0.18	14	102	69.20
50%	3008	0.91	289	-744	-411	790	0.15	14	100	64.70
40%	3011	0.79	249	-488	-269	882	0.12	14	96	58.70
30%	3012	0.65	205	-264	-152	946	0.09	14	92	50.80
20%	3014	0.57	180	-114	-71	977	0.06	14	88	36.70
10%	3016	0.46	148	-10	10	1008	0.02	14	83	12.70
0%	3017	0.40	127	24	70	1044	0.03	14	81	23.00

The sample calculations will be of the backwards impeller at 3000 rpm, and a slider position of 100%.

In the following equation ρ is the density of air, P_{atm} is the atmospheric pressure, R is the specific gas constant for dry air, and T is the ambient temperature.

$$\rho = \frac{P_{atm}}{R \cdot T}$$

$$\rho = \frac{768.1\ mmHg}{\left(287\ \frac{J}{kg \cdot K}\right) \cdot (295.15\ K)}$$

$$\rho = \frac{(768.1\ mmHg) \cdot \frac{(133.32\ Pa)}{(1\ mmHg)}}{\left(287\ \frac{J}{kg \cdot K}\right) \cdot (295.15\ K)}$$

$$\rho = \frac{\left(\frac{102403.1\ mmHg \cdot Pa}{1\ mmHg}\right)}{\left(84708.05 \frac{K \cdot J}{kg \cdot K}\right)}$$

$$\rho = \frac{\frac{(102403.1\ \cancel{mmHg} \cdot \frac{kg \cdot \cancel{m}}{\cancel{s^2} \cdot m^2})}{\cancel{mmHg}}}{\left(84708.05 \frac{\cancel{K} \cdot \frac{\cancel{kg} \cdot m^{\cancel{2}}}{\cancel{s^2}}}{\cancel{kg} \cdot \cancel{K}}\right)}$$

$$\rho = 1.208\ \frac{kg}{m^3}$$

In the following equation Q is the flowrate, $\dot{m}$ is the mass flowrate, and ρ is the density of the fluid (air).

$$Q = \frac{\dot{m}}{\rho} \quad \textit{Bergman, p.551, Eqn.12.2}$$

$$Q = \frac{0.2460\ \frac{kg}{s}}{1.208\ \frac{kg}{m^3}}$$

$$Q = 0.2036\ \frac{\frac{\cancel{kg}}{s}}{\frac{\cancel{kg}}{m^3}}$$

$$Q = 0.2036\ \frac{m^3}{s}$$

In the following equation A_C is the cross-sectional area of the backwards impeller, D is the diameter of the impeller, and π is the ratio of the impeller's circumference and its diameter.

$$A_C = \frac{1}{4} \cdot \pi \cdot D^2$$

$$A_C = \frac{1}{4} \cdot \pi \cdot (0.25\ m)^2$$

$$A_C = 0.049\ m^2$$

In the following equation h_a is the head available, P_3 is the pressure at the outlet, P_2 is the pressure at the inlet, ρ is the density of air, and g is the gravitational constant.

$$h_a = \frac{(P_3 - P_2)}{\rho \cdot g} \quad \textit{Bergman, p.558, Eqn. 12.20}$$

$$h_a = \frac{(931.45\ Pa - 46.45\ Pa)}{\left(1.208\ \frac{kg}{m^3}\right) \cdot \left(9.81\ \frac{m}{s^2}\right)}$$

$$h_a = \frac{885.00\ Pa}{\left(11.85\ \frac{m \cdot kg}{m^3 \cdot s^2}\right)}$$

$$h_a = 74.68\ \frac{\frac{\cancel{kg}}{\cancel{s^2} \cdot \cancel{m}}}{\frac{\cancel{m} \cdot \cancel{kg}}{m^{\cancel{3}} \cdot \cancel{s^2}}}$$

$$h_a = 74.68\ m$$

In the following equation WHP is the water horsepower, g is the gravitational constant, P_3 is the pressure at the outlet, and P_2 is the pressure at the inlet.

$$WHP = Q(P_3 - P_2) \quad \textit{Bergman, P.558, Eqn.12.21}$$

$$WHP = 0.21\ \frac{m^3}{s} \cdot (156.55\ Pa + 860.00\ Pa)$$

$$WHP = 0.21\ \frac{m^3}{s} \cdot (1016.55\ Pa)$$

$$WHP = 213.48\ \frac{N \cdot m^{\cancel{3}}}{s \cdot \cancel{m^2}}$$

$$WHP = 213.48\ W$$

$$WHP = (213.48\ \cancel{W}) \cdot \left(\frac{1\ hp}{745.7\ \cancel{W}}\right)$$

$$WHP = 0.281\ hp$$

In the following equation *BHP* is the brake horsepower, T_{shaft} is the torque of the fan shaft, and ω is the angular velocity of the shaft.

$$BHP = T_{shaft} \cdot \omega \quad \textit{Bergman, p. 555, Eqn.12.10}$$

$$BHP = (0.87\ N \cdot m) \cdot \left(2998.09\ \frac{rev}{min}\right)$$

$$BHP = (0.87\ N \cdot m) \cdot \left[\left(2998.09\ \frac{\cancel{rev}}{\cancel{min}}\right) \cdot \left(\frac{2\pi\ rad}{1\ \cancel{rev}}\right) \cdot \left(\frac{1\ \cancel{min}}{60\ s}\right)\right]$$

$$BHP = (0.87\ N \cdot m) \cdot \left(313.96\ \frac{1}{s}\right)$$

$$BHP = 273.14\ W$$

$$BHP = (273.14\ \cancel{W}) \cdot \left(\frac{1\ hp}{745.7\ \cancel{W}}\right)$$

$$BHP = 0.366\ hp$$

In the following equation η is the fan efficiency, *BHP* is the brake horsepower, and *WHP* is the water horsepower.

$$\eta = \frac{WHP}{BHP} \quad \textit{Bergman, p.559, Eqn. 12.23}$$

$$\eta = \frac{(0.281\ \cancel{hp})}{(0.366\ \cancel{hp})}$$

$$\eta = 0.768$$

In the following equation C_Q is the flow coefficient, *Q* is the flowrate, *D* is the diameter of the backwards impeller, and ω is the angular velocity of the shaft.

$$C_Q = \frac{Q}{\omega \cdot D^3} \quad \textit{Bergman, p.567, Eqn. 12.32}$$

$$C_Q = \frac{\left(0.21\ \frac{m^3}{s}\right)}{\left(2998.09\ \frac{rev}{min}\right) \cdot (0.25\ m)^3}$$

$$C_Q = \frac{\left(0.21\ \frac{m^3}{s}\right)}{\left[\left(2998.09\ \frac{\cancel{rev}}{\cancel{min}}\right) \cdot \left(\frac{2\pi\ rad}{1\ \cancel{rev}}\right) \cdot \left(\frac{1\ \cancel{min}}{60\ s}\right)\right] \cdot (0.25\ m)^3}$$

$$C_Q = \frac{\left(0.21\ \frac{m^3}{s}\right)}{\left(313.96\ \frac{1}{s}\right) \cdot (0.25\ m)^3}$$

$$C_Q = \frac{\left(0.21\ \frac{\cancel{m^3}}{\cancel{s}}\right)}{\left(4.91\ \frac{\cancel{m^3}}{\cancel{s}}\right)}$$

$$C_Q = 0.042$$

In the following equation C_P is the power coefficient, *BHP* is the brake horsepower (in Watts), ω is the angular velocity of the shaft, ρ is the density of air, and D is the diameter of the backwards impeller.

$$C_P = \frac{BHP}{\rho \cdot \omega^3 \cdot D^5} \quad \textit{Bergman, p.566, Eqn. 12.30}$$

$$C_P = \frac{(273.14\ W)}{\left(1.208\ \frac{kg}{m^3}\right) \cdot \left(2998.09\ \frac{rev}{min}\right)^3 \cdot (0.25\ m)^5}$$

$$C_P = \frac{(273.14\ W)}{\left(1.208\ \frac{kg}{m^3}\right) \cdot \left[\left(2998.09\ \frac{\cancel{rev}}{\cancel{min}}\right) \cdot \left(\frac{2\pi\ rad}{1\ \cancel{rev}}\right) \cdot \left(\frac{1\ \cancel{min}}{60\ s}\right)\right]^3 \cdot (0.25\ m)^5}$$

$$C_P = \frac{(273.14\ W)}{\left(1.208\ \frac{kg}{m^3}\right) \cdot \left(313.96\ \frac{1}{s}\right)^3 \cdot (0.25\ m)^5}$$

$$C_P = \frac{(273.14\ W)}{\left(1.208\ \frac{kg}{m^3}\right) \cdot \left(313.96\ \frac{1}{s}\right)^3 \cdot (0.25\ m)^5}$$

$$C_P = \frac{(273.14\ W)}{\left(36508.16\ \frac{kg \cdot m^5}{m^3 \cdot s^3}\right)}$$

$$C_P = 0.0074\ \frac{\frac{\cancel{kg} \cdot \cancel{m^2}}{\cancel{s^3}}}{\frac{\cancel{kg} \cdot \cancel{m^5}}{\cancel{m^3} \cdot \cancel{s^3}}}$$

$$C_P = 0.0074$$

In the following equation C_h is the head coefficient, g is the gravitational constant, h_a is the head, ω is the angular velocity of the shaft, and D is the diameter of the backwards impeller.

$$C_h = \frac{g \cdot h_a}{\omega^2 \cdot D^2} \quad \textit{Bergman, p.566, Eqn. 12.29}$$

$$C_h = \frac{\left(9.81\ \frac{m}{s^2}\right) \cdot (85.01\ m)}{\left(2998.09\ \frac{rev}{min}\right)^2 \cdot (0.25\ m)^2}$$

$$C_h = \frac{\left(9.81\ \frac{m}{s^2}\right) \cdot (85.01\ m)}{\left[\left(2998.09\ \frac{\cancel{rev}}{\cancel{min}}\right) \cdot \left(\frac{2\pi\ rad}{1\ \cancel{rev}}\right) \cdot \left(\frac{1\ \cancel{min}}{60\ s}\right)\right]^2 \cdot (0.25\ m)^2}$$

$$C_h = \frac{\left(9.81\ \frac{m}{s^2}\right) \cdot (85.01\ m)}{\left(313.96\ \frac{1}{s}\right)^2 \cdot (0.25\ m)^2}$$

$$C_h = 0.135\ \frac{\frac{\cancel{m^2}}{\cancel{s^2}}}{\frac{\cancel{m^2}}{\cancel{s^2}}}$$

$$C_h = 0.135$$

In the following equation *cost* is the annual cost to keep the fan running, and *BHP* is the brake horsepower.

$$cost = BHP \cdot \left(\frac{\$\ 0.15}{kW \cdot h}\right) \cdot \left(\frac{2800\ h}{yr}\right) \cdot \left(\frac{0.7457\ kW}{1\ hp}\right)$$

$$cost = (0.2615\ \cancel{hp}) \cdot \left(\frac{\$\ 0.15}{\cancel{kW} \cdot \cancel{h}}\right) \cdot \left(\frac{2800\ \cancel{h}}{yr}\right) \cdot \left(\frac{0.7457\ \cancel{kW}}{1\ \cancel{hp}}\right)$$

$$cost = \frac{\$\ 109.85}{yr}$$

The uncertainty calculations will be of the backwards impeller at 3000 rpm, and a slider position of 100%.

In the following uncertainty calculation ρ is the density of air, P_{atm} is the atmospheric pressure, R is the specific gas constant for dry air, T is the ambient air, $U(\rho)$ is the uncertainty in the density, $U(P_{atm})$ is the uncertainty in the atmospheric pressure, and $U(T)$is the uncertainty in the ambient air. First the equation of density will be shown, and then the Root Square Sum equation for density along with all the necessary calculations.

$$\rho = \frac{P_{atm}}{R \cdot T}$$

$$U(\rho) = \sqrt{\left(\frac{\partial\rho}{\partial P}U(P_{atm})\right)^2 + \left(\frac{\partial\rho}{\partial T}U(T)\right)^2}$$

$$U(\rho) = \sqrt{\left(\frac{1}{RT}U(P_{atm})\right)^2 + \left(\frac{-2P}{RT^2}U(T)\right)^2}$$

$$U(\rho) = \sqrt{\left(\frac{1}{\left(287\ \frac{J}{kg \cdot K}\right) \cdot (295.15\ K)} \cdot (0.001\ Pa)\right)^2 + \left(\frac{-2 \cdot (768.1\ mmHg)}{\left(287\ \frac{J}{kg \cdot K}\right) \cdot (295.15\ K)^2} \cdot (0.05)\right)^2}$$

$$U(\rho) = \sqrt{\left(\frac{1}{\left(287\ \frac{\cancel{kg} \cdot m^2}{\cancel{kg} \cdot \cancel{K} \cdot \cancel{s^2}}\right) \cdot (295.15\ \cancel{K})} \cdot \left(0.001\ \frac{\frac{kg \cdot m}{\cancel{s^2}}}{\cancel{m^2}}\right)\right)^2 + \left(\frac{-2 \cdot (768.1\ \cancel{mmHg}) \cdot \left(\frac{133.32\ \frac{\frac{kg \cdot m}{\cancel{s^2}}}{m^2}}{1\ \cancel{mmHg}}\right)}{\left(287\ \frac{\cancel{kg \cdot m^2}}{\cancel{kg \cdot K \cdot s^2}}\right) \cdot (295.15\ \cancel{K})^2} \cdot (0.05)\right)^2}$$

$$U(\rho) = 0.00819\ \frac{kg}{m^3}$$

In the following uncertainty calculation h_a is the head available, P_3 is the pressure at the outlet, P_2 is the pressure at the inlet, ρ is the density of air, g is the gravitational constant, $U(h_a)$ is the uncertainty of head, $U(P_3)$ is the uncertainty in P_3, $U(P_2)$ is the uncertainty in P_2, and $U(\rho)$ is the uncertainty in the density of air.

$$h_a = \frac{(P_3 - P_2)}{\rho \cdot g} \quad \textit{Bergman, p.558, Eqn. 12.20}$$

$$U(h_a) = \sqrt{\left(\frac{\partial h_a}{\partial P_3} U(P_3)\right)^2 + \left(\frac{\partial h_a}{\partial P_2} U(P_2)\right)^2 + \left(\frac{\partial h_a}{\partial \rho} U(\rho)\right)^2}$$

$$U(h_a) = \sqrt{\left(\frac{(860.00)}{(1.208)\cdot(9.81)} \cdot (2.15)\right)^2 + \left(\frac{(156.55)}{(1.208)\cdot(9.81)} \cdot (1.3)\right)^2 + \left(\frac{-(156.55 + 860.00)}{(1.208)^2 \cdot (9.81)} \cdot (0.00819)\right)^2}$$

$$U(h_a) = 156.97\ m$$

In the following uncertainty calculation Q is the flowrate, $\dot{m}$ is the mass flowrate, ρ is the density of air, $U(Q)$ is the uncertainty in the flowrate, $U(\dot{m})$ is the uncertainty in the mass flow rate, and $U(\rho)$ is the uncertainty in the density of air.

$$Q = \frac{\dot{m}}{\rho} \quad \textit{Bergman, p.551, Eqn.12.2}$$

$$U(Q) = \sqrt{\left(\frac{\partial Q}{\dot{m}} U(\dot{m})\right)^2 + \left(\frac{\partial Q}{\partial \rho} U(\rho)\right)^2}$$

$$U(Q) = \sqrt{\left(\left(\frac{1}{1.208}\right) \cdot (0.0037)\right)^2 + \left(\frac{-(0.2542)}{(1.208)^2} \cdot (0.00819)\right)^2}$$

$$U(Q) = 0.00337\ \frac{m^3}{s}$$

In the following uncertainty calculation *WHP* is the water horsepower, g is the gravitational constant, P_3 is the pressure at the outlet, P_2 is the pressure at the inlet, $U(WHP)$ is the uncertainty of *WHP*, $U(P_3)$ is the uncertainty of P_3, and $U(P_2)$ is the uncertainty of P_2.

$$WHP = Q(P_3 - P_2) \quad \textit{Bergman, P.558, Eqn.12.21}$$

$$U(WHP) = \sqrt{\left(\frac{\partial WHP}{\partial P_3} U(P_3)\right)^2 + \left(\frac{\partial \mathrm{WHP}}{\partial P_2} U(P_2)\right)^2 + \left(\frac{\partial \mathrm{WHP}}{\partial Q} U(Q)\right)^2}$$

$$U(WHP) = \sqrt{(0.21 \cdot (1.30))^2 + (-0.21 \cdot (2.15))^2 + \left((156.55 + 860.00) \cdot (0.00337)\right)^2}$$

$$U(WHP) = 0.00464\ hp$$

In the following uncertainty calculation *BHP* is the brake horsepower, T_{shaft} is the torque of the fan shaft, ω is the angular velocity of the shaft, *U(BHP)* is the uncertainty of *BHP*, $U(T_{shaft})$ is the uncertainty in the torque, and $U(\omega)$ is the uncertainty in the angular velocity of the shaft.

$$BHP = T_{shaft} \cdot \omega \quad \textit{Bergman, p. 555, Eqn.12.10}$$

$$U(BHP) = \sqrt{\left(\frac{\partial BHP}{\partial T_{shaft}} U(T_{shaft})\right)^2 + \left(\frac{\partial \mathrm{BHP}}{\partial \omega} U(\omega)\right)^2}$$

$$U(BHP) = \sqrt{(313.96 \cdot (0.005))^2 + (0.87 \cdot (8.23))^2}$$

$$U(BHP) = 0.02227\ hp$$

In the following uncertainty calculation η is the fan efficiency, *BHP* is the brake horsepower, *WHP* is the water horsepower, $U(\eta)$ is the uncertainty in the fan performance, *U(BHP)* is the uncertainty of *BHP*, and *U(WHP)* is the uncertainty of *WHP*.

$$\eta = \frac{WHP}{BHP} \quad \textit{Bergman, p.559, Eqn. 12.23}$$

$$U(\eta) = \sqrt{\left(\frac{\partial \eta}{\partial WHP} U(WHP)\right)^2 + \left(\frac{\partial \eta}{\partial BHP} U(BHP)\right)^2}$$

$$U(\eta) = \sqrt{\left(\frac{1}{0.365} \cdot (0.00464)\right)^2 + \left(\frac{-0.281}{(0.365)^2} \cdot (0.02227)\right)^2}$$

$$U(\eta) = 0.0486$$

In the following equation C_Q is the flow coefficient, *Q* is the flowrate, *D* is the diameter of the backwards impeller, ω is the angular velocity of the shaft, $U(C_Q)$ is the uncertainty of C_Q, *U(Q)* is the uncertainty of *Q*, *U(D)* is the uncertainty of *D*, and $U(\omega)$ is the uncertainty in ω.

$$C_Q = \frac{Q}{\omega \cdot D^3} \quad \textit{Bergman, p.567, Eqn. 12.32}$$

$$U(C_Q) = \sqrt{\left(\frac{\partial C_Q}{\partial Q} U(Q)\right)^2 + \left(\frac{\partial C_Q}{\partial \omega} U(\omega)\right)^2 + \left(\frac{\partial C_Q}{\partial D} U(D)\right)^2}$$

$$U(C_Q) = \sqrt{\left(\frac{1}{(313.96 \cdot (0.25)^3} \cdot (0.00337)\right)^2 + \left(\frac{-0.21}{313.96^2 \cdot (0.25)^3} \cdot (8.23)\right)^2 + \left(\frac{-3 \cdot (0.21)}{313.96 \cdot (0.25)^4} (0.006)\right)^2}$$

$$U(C_Q) = 0.003351$$

In the following uncertainty calculation C_P is the power coefficient, *BHP* is the brake horsepower (in Watts), ω is the angular velocity of the shaft, ρ is the density of air, *D* is the diameter of the backwards impeller, U(C_P) is the uncertainty in C_P, U(BHP) is the uncertainty of BHP in watts, U(D) is the uncertainty of D, U(ω) is the uncertainty of the angular speed, and U(ρ) is the uncertainty in the density of air.

$$C_P = \frac{BHP}{\rho \cdot \omega^3 \cdot D^5} \quad \textit{Bergman, p.566, Eqn. 12.30}$$

$$U(C_P) = \sqrt{\left(\frac{\partial C_P}{\partial BHP} U(BHP)\right)^2 + \left(\frac{\partial C_P}{\partial \omega} U(\omega)\right)^2 + \left(\frac{\partial C_P}{\partial D} U(D)\right)^2 + \left(\frac{\partial C_P}{\partial \rho} U(\rho)\right)^2}$$

$$U(C_P) = \sqrt{\left(\frac{1}{1.208 \cdot (313.96)^3 \cdot (0.25)^5} \cdot (0.02227)\right)^2 + \left(\frac{-3 \cdot (0.365)}{(313.96)^4 \cdot 1.208 \cdot (0.25)^5} \cdot (8.23)\right)^2 + \left(\frac{-5 \cdot (0.365)}{1.208 \cdot (313.96)^3 \cdot (0.25)^6} \cdot (0.006)\right)^2 + \left(\frac{-0.365}{(1.208)^2 \cdot (313.96)^3 \cdot (0.25)^5} \cdot (0.00819)\right)^2}$$

$$U(C_P) = 0.00167$$

In the following uncertainty calculation C_h is the head coefficient, *g* is the gravitational constant, h_a is the head, ω is the angular velocity of the shaft, *D* is the diameter of the backwards impeller, $U(C_h)$ is the uncertainty of C_h, *U(D)* is the uncertainty of the diameter, $U(h_a)$ is the uncertainty of the head, and $U(\omega)$ is the uncertainty of the angular speed.

$$C_h = \frac{g \cdot h_a}{\omega^2 \cdot D^2} \quad \textit{Bergman, p.566, Eqn. 12.29}$$

$$U(C_h) = \sqrt{\left(\frac{\partial C_h}{\partial h_a} U(h_a)\right)^2 + \left(\frac{\partial C_h}{\partial \omega} U(\omega)\right)^2 + \left(\frac{\partial C_h}{\partial D} U(D)\right)^2}$$

$$U(C_h) = \sqrt{\left(\frac{9.81}{(313.96)^2 \cdot (0.25)^2} \cdot 156.27\right)^2 + \left(\frac{-2 \cdot 9.81 \cdot (85.01)}{(313.96)^3 \cdot (0.25)^2} \cdot (8.23)\right)^2 + \left(\frac{-2 \cdot 9.81 \cdot (85.01)}{(313.96)^2 \cdot (0.25)^3} \cdot (0.006)\right)^2}$$

$$U(C_h) = 0.251$$

Lab Report Participation Log

Name	Date	Hours Worked	Description of Tasks Performed
Cyril Moran	1\13\2022	2	Organized all the data and started the head vs. flowrate graph for the backwards impeller.
Cyril Moran	1\15\2022	1	Finished the head vs. flowrate graph of the backwards impeller.
Cyril Moran	1\16\2022	2	Completed the letter formatting and the first two paragraphs of the backwards impeller.
Cyril Moran	1\18\2022	5	Completed the BHP and WHP vs flowrate graphs for all the backwards impeller speeds.
Cyril Moran	1\21\2022	6	Completed the efficiency graph for the backwards impeller, the non-dimensional parameters graphs for the backwards impeller, the annual cost for the backwards impeller, and the sample calculations.
Cyril Moran	1\25\2022	2	Completed the uncertainty calculations and checked formatting with Nickolas.
Nickolas Saavedra	1\26\2022	3	Completed letter formatting, performed calculations for radial impeller, and created head vs. flowrate graphs for radial impeller.
Nickolas Saavedra	1\30\2022	4	Completed efficiency graphs for radial impeller, non-dimensional parameter graphs for radial impeller, and calculated annual cost for radial impeller
Nickolas Saavedra	1\31\2022	8	Corrected errors in radial impeller graphs, imported radial impeller graphs and information into word and completed paragraphs. Revised formatting.
Nickolas Saavedra	2\1\2022	1	Read over report and ensured submission readiness.
Cyril Moran	2\1\2022	2	Checked report and verified that report is ready to submit.
Cyril Moran	2\4\2022	5	Looked over all of the comments from Jaxton, fixed the first two paragraphs of the letter, and reorganized all the data for the backwards impeller to make the graphs easier.

Cyril Moran	2\7\2022	12	Fixed the head graph for the backwards impeller by adding the uncertainty bars, fixed the WHP and BHP vs. flowrate graph for the backwards impeller by averaging the 11 data points for every slide valve position, fixed the efficiency curve of the backwards impeller by accurately calculating the WHP and BHP values, fixed the non-dimensional parameter curves by calculating them correctly and plotting the averages, fixed the annual cost by using the BHP instead of the WHP of the system, fixed all of the sample calculation formatting and equation errors, completed the uncertainty calculation corrections, and then I completed the participation log.
Nickolas Saavedra	2\8\2022	7	Looked over Jaxtons comments and fixed graphs, tables, and paragraphs referring to radial impeller. Looked over backwards impeller graph, tables, and paragraphs to ensure Jaxtons comments were corrected. Adjusted formatting. Double checked report requirements. Completed participation log. Submitted.
	Total Hours:	60	